Volume 1



Industrial Ventilation Design Guidebook

Fundamentals

Edited by

Howard Goodfellow
Risto Kosonen

INDUSTRIAL VENTILATION DESIGN GUIDEBOOK

SECOND EDITION

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Volume 1: Fundamentals

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Preface

The revised Industrial Ventilation Design Guidebook—IVDGB (2020) builds on the work done by the original contributors to IVDGB (2001). The IVDGB (2020) represents the advances in science and engineering over the past 20 years. The IVDGB is being published in two volumes. Volume 1 is titled Fundamentals and covers Chapters 1−9. Volume 2 in the series is titled Engineering Design and Applications which covers Chapters 10−16. This scientific textbook represents for the first time advances in global R&D and the best practices in engineering design based on contributions from over 40 experts in the global industrial ventilation field. This revised textbook represents the first truly global scientific textbook in this field with major contributions by experts in the industrial ventilation field from Asia, Europe, and North America.

The publication of this reference book is very timely as the awareness of the role of ventilation from engineering design, operations, and maintenance has increased to new levels in 2020. The driving force with a high level of urgency has been global health concerns (hospitalization and deaths) of infection control surrounding COVID-19 pandemic and the critical role that ventilation systems play in the control of this pandemic. It is known that infection risk depends on aerosol concentration and occupancy time. Ventilation systems become a key technology to provide a safe and healthy environment for occupants in the control of airborne viruses such as COVID-19, designers of ventilation systems for all residential, commercial, and industrial ventilation spaces will face new challenges on the proper design of ventilation systems for infection control measures. The challenge will be to provide healthy indoor environments for all occupants during all seasons (especially winter) while being energy efficient. IVDGB (2020) will be an authoritative reference textbook in the ventilation field for policymakers and designers of ventilation systems for COVID-19 pandemic and for future viruses.

For the contents of Volume 1, the reader is referred to Chapter 1, Introduction, of this book which covers in details the history/state-of-the-art of industrial ventilation, background, and contents of IVDGB (2001) and background and contents of the revised IVDGB (2020). The last section in Chapter 1, Introduction, explores future directions and opportunities in the science and engineering of industrial ventilation systems. I will expand on this area and add some details.

The science and technology of ventilation (residential, commercial, and industrial) is at a crossroads and key decisions need to be made at the global level to capitalize on the unbounded opportunities. Three key areas to be pursued are

- better communication (scientific and engineering community of ventilation and contaminant control),
- develop a global collaborative community, and
- embrace disruptive technologies [sensors, modeling, automation, artificial intelligence (AI), etc.].

In the first area of better communication, it is important to recognize that there are many common areas of scientific research and engineering in the ventilation and contaminant control field for residential, commercial, and industrial ventilation spaces. Unfortunately silos exist between these different sectors and there is very little sharing of technologies and R&D activities. The goal is to develop a holistic approach for the science and engineering of ventilation for any occupied spaces.

The second area is to develop a global collaborative network in the ventilation technology field. This global network would include scientific research (academic and research institutes), professional associations (ASHRAE, REHVA, SHASE, etc.), international technical conferences (such as international industrial ventilation conferences started in 1985 at the University of Toronto and held every 3 years at different regions of the world), low carbon economy, disruptive technologies (Industrial 4.0, AI, sensors, etc.), and scientific publications such as revised IVDGB (2020). Success depends on a holistic, multidisciplinary, and sustainable funding model. An excellent example of this goal to breakdown the silos is the leadership shown by ASHRAE President, Professor Olesen in his August 2017 article in *ASHRAE Journal* entitled, Extending our

Community. The specific goals outlined in the report were the urgent need for the ventilation community to "extend the global community" and "to extend the technological horizons."

In October 2019, ASHRAE announced the incorporation of the Indoor Environmental Quality Global Alliance (IEQ-GA) during a ceremony at the 40th AIVC conference in Ghent, Belgium. The mission of the IEQ-GA is to promote and advocate for acceptable indoor environmental quality (thermal environment, indoor air quality, lighting, and acoustics) for building occupants globally while ensuring the knowledge from IEQ research is implemented in practice.

"We are inspired by the forward-thinking approach IEQ-GA has in the building industry," said 2019−20 ASHRAE President Darryl K. Boyce, P. Eng. "It is critically important that we advocate for the well-being of the people who occupy our buildings. Through the collaborative efforts and resources of IEQ-GA member organizations, we will continue to provide safe buildings for generations to come." The COVID-19 pandemic has accelerated this forward-thinking approach and the challenge to develop innovative and cost-effective solutions in a timely fashion across the different sectors. A fundamental understanding of the science of ventilation and contaminant control is a key component to develop cost-effective engineering solutions and to deliver a high performance of safety for all occupants from a health perspective.

"An important attribute of IEQ-GA is that it is a non-industrial alliance among societies representing members that promote IEQ," said Bjarne Olesen, 2017−18 ASHRAE Presidential Member. "The intent of IEQ-GA is to work with all partners of the value chain for indoor environmental quality including building research, comfort and health research building design, installation, commissioning, operation, and occupant behavior. Furthermore IEQ-GA will work with industry organizations to help their members to provide products and services that promote IEQ."

The founding members represent professionals from various disciplines linked to indoor environment and are committed to work together and promote education, research, and knowledge exchange on a global scale to develop standard guidelines for advocacy of the general public in indoor environments around the world.

The third area to be pursued is to embrace disruptive technologies and to be bold and take risks where there are significant opportunities for rewards. Disruptive technologies are happening at an accelerated rate and will have a major impact on the future directions of the science and technology of ventilation of the three sectors (residential, commercial, and industrial). Disruptive technologies, such as robotics, AI, models for low carbon economy, innovative sensors, etc., are impacting many sectors and researchers must embrace the cross transfer of these technologies. For example, innovative sensors are being developed that are wireless, non-invasive, cheap, remote, and in situ. Many of these sensors developed for autonomous vehicles have wide applications in the measurement and control fields for many different sectors for advanced design of cost-effective ventilation systems. We must not miss these opportunities to embrace disruptive technologies in the ventilation fields.

In summary, IVDGB (2020) is an updated and exhaustive scientific textbook prepared by global experts in the industrial ventilation field. IVDGB (2020) has the potential to accelerate the implementation into practice the latest research and development activities at a global level.

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I would like to express my thanks to my longtime friend Esko Tahti who was our leader and my coeditor as we worked together on the preparation of the first Industrial Ventilation Design Guidebook more than 25 years ago. I will always be grateful for the confidence that Finland had in me to work with their scientific team and the EU to deliver a world class reference textbook. I thought of our earlier journey often as I worked with the new team to prepare the revised textbook some 20 years later. The legacy of our earlier work has been the basis for our updated revised textbook based on new engineering and scientific R&D work in the global industrial ventilation field.

My alma mater (University of Toronto) has continued to provide support and encouragement to myself and my companies as we pursued projects in the areas of leading edge R&D in the industrial ventilation and contaminant control field. I have been privileged to be an Adjunct Professor in the Department of Chemical Engineering and Applied Chemistry for more than 40 years and value many of my colleagues who have made significant contributions to my career.

I dedicate this book to Karen Goodfellow, my wife, and partner for more than 54 years, who has provided unfailing support and love during my professional career. I also acknowledge the support and encouragement from my family (Geoff, Anne, Jen, Peter, Jessie, Will, Juliana, Ryan, and Caitlyn). All of you as a family have been my true source of inspiration and solace during this journey.

CHAPTER

1

Introduction

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1.1 Goals/benefits

1.1.1 Goals

As stated in the preface of the original *Industrial Ventilation Design Guidebook—IVDGB* (2001),[1] the primary goal of IVDGB (2001) was to develop a systematic approach to the engineering design of industrial ventilation systems. This goal was achieved by assembling a global team of scientific researchers and engineers to prepare a comprehensive definitive international handbook. Our plan was to update the IVDGB on a regular basis in order to meet our objective of including the most current scientific knowledge on a global basis. The revised *Industrial Ventilation Design Guidebook—IVDGB* (2020) represents the advances in science and engineering over the last 20 years based on global research and development and best practices in engineering design on a global basis.

In the 1980s a young scientist said "I have never seen such a complex scientific area such as industrial ventilation where so little scientific research and brain power has been applied." This is one of the major reasons research and development activities in the industrial ventilation field were started in the 1980s in Finland. The young scientist was right. The challenges faced by designers and practitioners in the industrial ventilation field, compared to comfort ventilation, in the residential and/or commercial buildings, are much more complex. In industrial ventilation it is essential to have an in-depth knowledge of modern computational fluid dynamics (CFD), three-dimensional heat flow, complex fluid flows, steady state and transient conditions, operator issues, contaminants inside and outside the facility, environmental conditions, etc.

In all ventilation the condition of the indoor environment, called indoor air quality (IAQ), and the exposures for the occupants are important. IAQ is a subset of indoor environmental quality (IEQ), which refers to overall conditions of the building, including not only air but also lighting, acoustics, thermal comfort, electromagnetic radiation, etc. In industrial facilities the contaminant emission rates may be 10−100 times higher than in nonindustrial facilities, but for many contaminants the design IAQ levels may be the same. The first priority is to consider the process, but other important issues, such as occupant exposures, energy, environment, and corporate image, must also be considered.

Energy is a key issue and is closely linked with the environment. Global environmental issues must be addressed, where the energy chain from resources to end users is of vital importance.

It is recognized that some countries are leaders in the area of scientific research and experimental development in the industrial ventilation field. Scandinavian countries were in the forefront of implementing leading-edge technology for good environmental practice and energy-efficient plants in the 1980s and 1990s. Russia, Japan, and North America (United States, Canada) have also made significant scientific contributions in the ventilation field. The challenge is the implementation of the best industrial ventilation technology and practice to all workplaces on a global basis.

The objectives of new innovations, procedures, systems, and equipment to fulfill the end user's needs must be included as a part of the ongoing research and development program. Significant advances in technology presented in IVDGB (2001) were target levels, systematic design methodology, IAQ strategies, control of flow in facilities, and air cleaning devices. The objective is to make the leading-edge science work in practice and to continue the development process. IVDGB (2020) covers recent developments in these areas and includes for the first time leading-edge

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research and development and engineering design in the industrial ventilation field from China.

When a comparison is made between industrial ventilation and comfort ventilation, it is clear that the task is very challenging for ventilation engineers and scientists. To fulfill all the needs of the end user is often impossible. If the IAQ is fulfilled, the amount of air may be so large to create excessive velocities in the workplace. We must also have the courage to say what is possible and what is not. We also have to create tools to validate system performance based on rigorous scientific and engineering principles.

1.1.2 Benefits

The benefits of state-of-the-art industrial ventilation technology are as follows:

- improved health of workers and reduced absenteeism as a result of better IAQ;
- improved worker satisfaction, higher productivity, and reduced production failures as a result of improved IAQ;
- reduction in maintenance costs for the building envelopes, machinery, and products;
- reduction in energy consumption as a result of improved usage patterns and reduced airflow rates;
- increased awareness, and therefore improved selection, of new energy-efficient systems in ventilation design, which results in reduced energy consumption;
- cleaner plant surroundings, and thus an improved image of the company, resulting from improved systems and equipment;
- reduction in environmental pollution due to lower energy usage and lower emissions to the surroundings and a decrease in greenhouse gas emissions to the atmosphere; and
- improved life cycle economy resulting from the use of high-level and rugged industrial ventilation technology systems and equipment.

Field studies have revealed the potential for significant energy saving based on the proper engineering design and application of modern industrial ventilation technology. For example, one study revealed great variation in energy consumption (a ratio of 5:1) in welding halls of similar size and production levels. This study showed the best IAQ; hence, worker exposure was achieved in the hall with the lowest energy consumption. With commercially available high-level design concepts, it is possible to decrease the contaminant load by about 90% and the heat load by about 60% compared with medium-level applications.

The abovementioned result is just one example to show the need to increase the level of knowledge from "rules of thumb" to a more "rigorous scientific procedure" based on validated data and design methods. IVDGB (2020) will fill an important gap in the development of a rigorous scientific approach.

In general, industrial ventilation systems can be classified into the following four types:

- industrial air-conditioning
- general ventilation systems
- local ventilation systems
- process ventilation systems

A brief description of each type of ventilation system is as follows. A more detailed description is presented in Chapter 2, Terminology, and subsequent chapters of the IVDGB (2020).

Industrial air-conditioning systems control air quality and thermal environment for both human occupancy and processes. General ventilation systems only control partially some indoor air parameters. Target levels are usually lower than for air-conditioning. Local ventilation systems are used for local controlled zones. These systems are based on local capture of contaminants. In process ventilation the target is to maintain defined conditions to ensure process performance (e.g., paper machine hoods). Any of the abovementioned ventilation systems may have air cleaning equipment to remove contaminants resulting in removal of contaminants before discharge to the atmosphere.

1.2 History and state of the art

Prior to IVDGB (2001) the available systematic information regarding industrial air technology is scarce. There are some handbooks, such as those of Hemeon (United States),[2] Baturin (1972),[3] Heinsohn (United States),[4] Goodfellow (Canada),[5] and ACGIH (United States),[6] but they do not cover the whole field of industrial ventilation technology. There is no internationally accepted handbook available, and the designer has no validated solutions at his disposal. According to the present state of the art, both capturing and ventilating systems are designed in general based on know-how rules (e.g., air exchange rate) and rarely achieve the targeted heat and contaminant load removal without over dimensioning and excessive costs. This expertise is not generated by systematic investigations but by experience with various plants under construction and in operation. This is obviously due to a total lack of approved design criteria and a lack of International or European standardization, which make effective ventilation design impossible.

1.3 *Industrial Ventilation Design Guidebook—IVDGB* (2001)

In 2001 a scientific textbook edited by Howard D. Goodfellow and Esko Tahti was published by Academic Press. The *Industrial Ventilation Design Guidebook* addressed the design of ventilation systems for the control of contaminants in industrial workplaces such as processing and manufacturing plants and for other specialized and complex engineering design of facilities such as health-care operating theaters, mine ventilation, and data centers. It covered the basic theories and science behind the technical solutions for industrial ventilation technology and included publication of new fundamental research and design equations contributed by more than 40 engineers and scientists from over 18 countries.

Readers were presented with scientific research and data for improving the IAQ in the workplace and reducing emissions to the outside environment. The guidebook represented, for the first time, a single source for all current scientific information available on the subject of industrial ventilation on a global basis and the more general area of ventilation for contaminant control. Specific features of the guidebook include the following:

- It presents technology for energy optimization and environmental benefits.
- It is a collaborated effort from more than 40 ventilation experts throughout 18 countries.
- It is based on more than 50 million dollars of research and development focused on industrial ventilation (mostly from Finland, France, Germany, Russia, United States, and Canada).
- It includes significant scientific contributions for the first time from leading ventilation experts in Russia.
- It presents new innovations, including a rigorous design methodology and target levels.
- It contains extensive sections on design with modeling techniques.
- Its content is well organized and easily adaptable to computer applications.

This comprehensive digest of scientific know-how gained its origin from the *International Industrial Ventilation Conferences* that were conceived by Profs. Jim Smith and Howard D. Goodfellow from the University of Toronto (September 1985).[7,8] Fig. 1.1 conveys the ventilation conferences held every 3 years with the most recent conference being held in Finland in 2018. These specialized conferences have resulted in the development of a critical global mass of engineers and scientists working in the industrial ventilation field. Since the conference inception, there have been 12 international symposiums with more than 3000 attendees and over 1250 technical papers. The *13th International Industrial Ventilation Conference* (Vent 2021) is now being planned for August 15–18, 2021 in Toronto, Canada, and is being organized by ASHRAE.

1.4 *Industrial Ventilation Design Guidebook—IVDGB* (2020)

The proposed revised guide book covers the area of ventilation for contaminant control based on global research by world-class researchers. This reference book is unique because it brings together global researchers and engineers to allow designers and

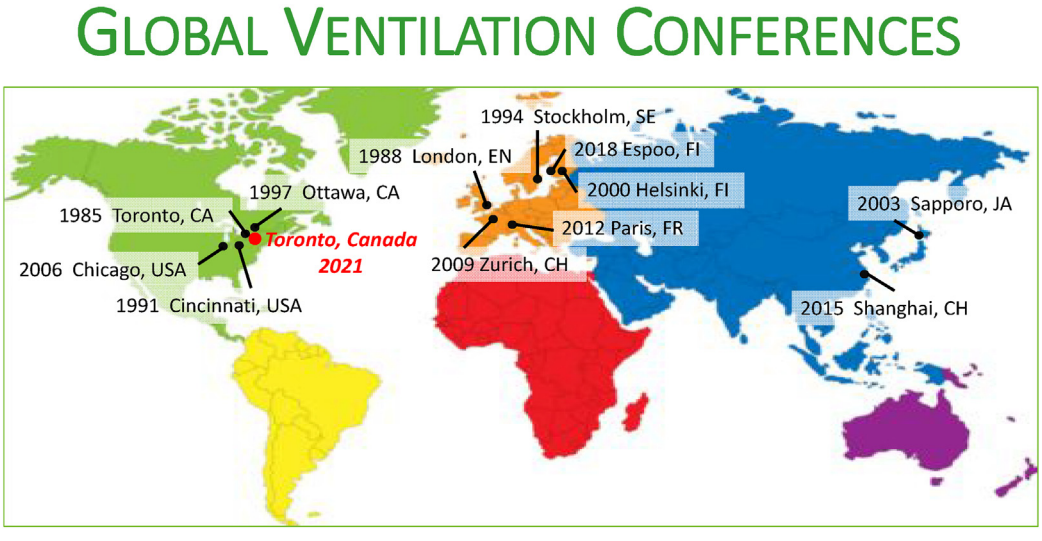


FIGURE 1.1 International Industrial Ventilation Conference locations.

engineers to solve complex ventilation problems using state-of-the-art design equations. Most of the equations and other scientific terms can be used in all ventilation and air-conditioning fields, not only for industrial ventilation but also residential and commercial applications. The IVDGB (2020) represents a significant advancement in the goal of harmonization of engineering design across all sectors of the scientific field of ventilation and contaminant control.

The recent awareness of climate change and a push by all industrial countries to embrace a low-carbon economy has put significant pressure on industry to reduce their environmental footprint. European countries have taken a leadership role with the introduction of Industry 4.0～automation plus sensors. For this to be implemented, engineers and scientists will be looking for a single reference source to find design equations and methodology to develop control algorithms for automation. Another key scientific component is the measurement of process parameters in real time using state-of-the-art sensors in the air and contaminant fields. These are specific areas that will be presented in depth for the first time in a detailed format based on global research in the sensor technology fields. Data will be presented both for leading-edge sensor technology and well-proven technology on a global basis.

The revised *Industrial Ventilation IVDGB* (2020) will be unique in the marketplace as it will present a single source for a holistic approach on a global basis to industrial ventilation for contaminant control. Details will be presented for the four key steps:

Step 1: Design methodology
Step 2: Design equations
Step 3: Design toolkits
Step 4: Specific industrial examples of best practice for more than 10 major sectors

The reason for proposing this newly revised edition is because of the wealth of increase in new research technology in the broad field of ventilation for contaminant control on a global scale since the original *Industrial Ventilation Design Guidebook* was published in 2001. The preparation of the original book took 10 years and major contributors from Europe and Russia where the level of science was the highest. Specific areas of advancement presented in IVDGB (2020) include design methodology for ventilation systems for contaminant control, use of high-speed computers in modeling capabilities of airflow and contaminant levels in both the workplace environment and the external environment, commercialization of the latest sensor technology such as lasers, and the breakthrough of practical application of deep learning in the artificial intelligence (AI) field.

Our approach is to achieve harmonization of ventilation technologies on a global basis. Our extensive list of global experts will present for the first time multisector crosscutting technology based on a holistic integrated approach of scientific research and engineering in the industrial ventilation field.

Fig. 1.2 is a schematic that illustrates the major chapters being planned for the revised *Industrial Ventilation IVDGB* (2020). The revised book will have Prof. Howard D. Goodfellow (University of Toronto) as Editor-in-Chief. The coeditors will be Prof. Risto Kosonen (Aalto University, Finland) for *Volume I—Fundamentals* and Prof. Yi Wang (Xi'an University, China) for *Volume 2—Engineering Design and Applications*. A brief description of chapters in Volumes 1 and 2 is as follows.

Volume 1: Fundamentals

1. Introduction
This chapter introduces the goals and benefits, history, state-of-the-art contents of IVDGB (2001) and IVDGB (2020), and future directions and opportunities in the industrial ventilation field.

2. Terminology
This chapter outlines the major terminology and components of industrial ventilation systems.

3. Design Methodology
Design methodology is the systematic description of the technical design process of industrial ventilation technology as an elementary part of the whole life cycle of the industrial plant. Practical designs for different types of industrial systems, including descriptions and new designs for ducted systems are presented.

4. Physical Fundamentals
This chapter introduces the important topics of fluid flow, properties of gases, heat and mass transfer, and physical/chemical characteristics of contaminants. The aim is to assist all engaged in industrial ventilation technology in understanding the physical background of the issues involved.

5. Physiological and Toxicological Considerations
This chapter introduces fundamentals of human physiology and health requirements from a toxicological perspective relevant to the control of indoor environment within industrial buildings.

6. Target Levels
The chapter presents further development in a new concept called target levels. It outlines the role of target levels in the systematic design methodology, the scientific and technical grounds for assessing target levels for key parameters of

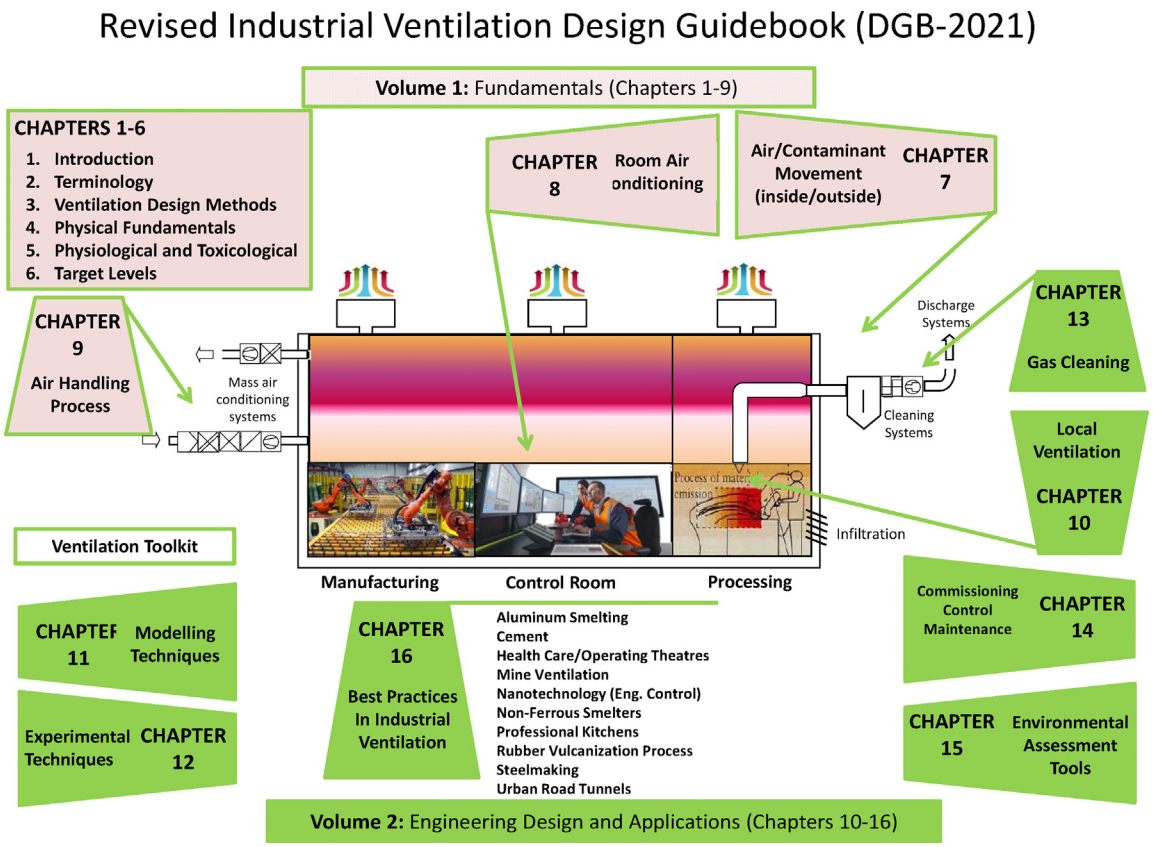


FIGURE 1.2 Outline of IVDGB (2020). *IVDGB,* Industrial Ventilation Design Guidebook.

industrial air technology, and the hierarchy of different target levels, as well as some examples of quantitative targets.

7. Principles of Air and Contaminant Movement inside and around Buildings

   This chapter presents the basic processes of air and contaminant movement, such as jets, plumes, and boundary flows, inside ventilated spaces. Major factors are summarized and classified.

8. Room Air-Conditioning

   This chapter describes the room air-conditioning process, including the interaction of different flow elements: room air distribution, heating and cooling methods, process sources, and disturbances. Air handling equipment, including room air heaters, is discussed in the form of "black boxes" as far as possible.

9. Air Handling Processes

   This chapter describes the fundamentals of air handling processes and equipment and gives answers to questions relating to the theoretical background of air handling unit and ductwork dimensioning and building energy systems optimization.

Volume 2: Engineering Design and Applications

10. Local Ventilation

    This chapter describes aerodynamic principles, models, and equations for local ventilation systems, which are used to transport contaminants or heat from the occupancy zone. The chapter covers the three main categories of local ventilation systems: exhaust hoods, supply air systems, and combinations of exhaust hoods and supply systems. In addition to the introduction of traditional local ventilation equipment, some innovative local ventilation systems developed recently are described.

11. Design with Modeling Techniques

    This chapter describes calculation models for building energy demand and airflow in and around industrial buildings based on recent advances in modeling. Special attention is paid to simulation of airborne contaminant control.

    Four methods for industrial air technology design are presented: CFD, thermal building dynamics simulation, multizone airflow models, and integrated airflow and thermal modeling. In addition to the basic physics of the problem, the purpose of the methods, recommended applications, limitations, cost and effort, and examples are provided.

12. Experimental Techniques
    This chapter covers a description of conventional measurement techniques used in ventilation as well as other related topics such as flow visualization, laser-based measurement techniques, and scale model experiments. Advancements in sensor technology and applications to industrial ventilation are discussed.
13. Gas Cleaning Technology
    This chapter describes the fundamentals of gas cleaning technology for removal of particulates and gaseous compounds from off-gas systems. Recent developments in gas cleaning technology to meet new statutory requirements are discussed.
14. Commissioning, Control and Maintenance of Ventilation Systems
    This new chapter covers the initial steps for commissioning through to startup, control and maintenance for both industrial ventilation and air-conditioning for buildings and local exhaust ventilation.
15. Environmental Assessment Tools
    Environmental assessment is a scientific method to evaluate and predict the environmental quality in a specific area in accordance with a certain standard. Specific examples are presented as they relate to the efficiency of industrial ventilation.
16. Best Practices in Industrial Ventilation
    This is a new chapter on best practices for specific industrial sectors and will be based on more than 10 major industrial sectors. The template for industrial sectors for this chapter on "Best Practices" for Industrial Ventilation for Contaminant Control—Industry Specific Sectors for selected industrial sectors includes:
    a. Overview—role of ventilation
    b. Design methodology
    c. Design equations
    d. Design toolkits
    e. Case studies—best practice
    f. Future challenges/opportunities
    g. Selected bibliography

This proposed template can be used to expand to other industrial sectors in the future.

New features of IVDGB (2020) will be as follows:

- Major new innovative technologies from researchers and engineers from China (book will become truly global) as well as Japan, Europe, and North America.
- Further validation of design methodology and target levels based on plant experience.
- Integration of automation and sensors (Industry 4.0).
- Closer collaboration with engineering schools and end users and the design/consulting communities.
- Focus on gaps in ventilation using new powerful search engines to ensure all recent developments and innovations are included on a global basis.
- A new and expanded section on sensors technology and methodology of selecting the best sensor for each unique application.
- Section on modeling and its practical applications will be expanded based on recent advances in research.

Table 1.1 illustrates the features and benefits of the revised *Industrial Ventilation IVDGB* (2020).

The scientific and professional engineering audience faces many issues. The first is that the literature (research and engineering) is highly fragmented in the scientific world (no specific home for ventilation and often in different disciplines in different countries). The proposed book will provide a single source for relevant research and engineering in the industrial ventilation for contaminant control field.

A second issue is that many of the text books, reference books, and engineering books in this field have not been updated for a couple of decades or so and do not reflect state-of-the-art for ventilation technology today and do not include the significant innovations in design criteria, modeling, sensors, AI (deep learning), and machine learning, which are available to meet the new challenges of sustainability and a low-carbon economy. The proposed book will update IVDGB (2001) and will focus on these recent developments.

The target audience will be at two levels and for a multisector industrial approach for processing plants and manufacturing and specialized areas such as hospital operating theaters, data centers, mining, and professional kitchens. The proposed two levels for this revised IVDGB (2020) will be to bring researchers, engineers (both design and plant), and scientists to develop a fundamental scientific understanding of ventilation and to provide trained engineers to implement this state-of-the-art ventilation technology on a global basis. It is envisaged that the revised IVDGB (2020) can be used as a core text book in an academic setting for mechanical engineers and process engineers. It is envisaged that it can be used as a background for specific industry based 1- to 5-day workshops and for plant and process engineers looking for a design methodology, sensors, and control algorithms for specific industrial operations to meet the challenging low-carbon economy. The textbook will also be a valuable reference book for consulting engineers working in the design of air pollution and

TABLE 1.1 *Industrial Ventilation IVDGB* (2020)—features/benefits.

Features	Benefits
Systematic holistic approach to design	Accelerate implementation of best practice for end users
With new section on best practices for more than 10 selected industries	
Global team of researchers and engineers as contributors, including China for the first time	Single source of all recent research and best practice for industrial end users. Benefit for training of future researchers, designers, and engineers to use IOT[a] to achieve energy efficiencies, Cleantech, climate change, etc.
Innovative state-of-the-art development of sensors, modeling, and design equations	Provide key technical inputs required for challenges of low-carbon economy and Industrial 4.0

[a]IOT, *Internet of Things*.

sustainability for their industrial clients (processing and manufacturing) to meet stringent standards for a green economy in the area of Cleantech.

1.5 Future directions and opportunities

1.5.1 Background

The manufacturing and processing industries are facing many challenges today in a global competitive world. The fourth industrial revolution (Industrial 4.0) is happening at an accelerated rate and many companies are embracing disruptive technologies. This technology is an extension of the automation field, which has seen the use of largely automatic equipment, including robots in a system of manufacturing or other production processes. Successful companies must adapt quickly and new technical skills are required to implement this revolution. Many companies do not have technical personnel to implement these technological changes. Skills include AI leaders, researchers, technologists, data scientists, and engineers. All of these innovative changes have a major impact on the proper design of industrial ventilation systems for the specific processes. Other disruptive technologies besides Industrial 4.0 and AI include robotics, innovative sensors, models for low-carbon economy, green economy, carbon taxes, and Cleantech. These disruptive technologies will have a profound impact on the design and planning of industrial ventilation and contaminant control systems.

A brief description of some selected leading-edge research and development and new policies and standards being developed in the different regions of the world [China, Europe, North America (United States, Canada), Japan] will be presented in the following sections. These descriptions will include policies, standards, guidelines, and design procedures as they impact on the design of industrial ventilation and contaminant control.

1.5.2 China

In recent years in the field of atmospheric pollution control, the Chinese government has issued a series of emission standards for different industrial sectors (magnesium, titanium, lead, steel, cement, etc.), which strictly limits the allowable emissions from these industrial sectors to the atmosphere. In the field of industrial building environment and energy conservation, the Chinese government has issued two standards: "Design Standards for Heating, Ventilation and Air Conditioning of Industrial Buildings" and "Unified Standard for Energy Efficiency Design of Industrial Buildings." Prof. Wang and coauthors have published two recent papers in the *Indoor and Built Environment Journal*, which cover the practical application of these two standards. A paper published in 2017[9] covers Industrial Building Environment: Old problems and challenges. The paper published in 2019[10] covers Energy Efficiency of Industrial Buildings.

There are two standards on green factories in China—"GB/T 36132-2018: General principles for assessment of green factory" and "GB/T 50787-2013: Evaluation standard for green industrial building." The first one focuses on green production and the second one focuses on green buildings. A short introduction and context of the two standards are presented next.

GB/T 50787-2013: Evaluation standard for green industrial building

Ministry of Housing and Urban-Rural Construction of the People's Republic of China, 2013

Introduction

This standard is formulated for the purpose of implementing the national policy of green development and building a resource-conserving and environment-friendly society, implementing the state laws and regulations on industrial policies, equipment policies, cleaner production, environmental protection, resource conservation, circular economy, safety and health of industrial construction, promoting the sustainable development of industrial buildings, and standardizing the evaluation of green industrial buildings.

This standard is applicable to new construction, expansion, reconstruction, relocation, restoration of the construction of industrial buildings, and applicable to existing industrial buildings in various industries factories, or applicable to the main production plants and all kinds of auxiliary production buildings of industrial complex.

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1. General Provisions
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  3.1. General Requirements
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Innovation

Appendix A Weightings and Credits

Appendix B Scope, Calculation, and Statistical Method for Energy Consumption Quota of Industrial Building

Appendix C Scope, Calculation, and Statistical Method for Water Utilization Quota of Industrial Building

Explanation of Wording in This Standard

List of Quoted Standards

Addition: Explanation of Provisions

GB/T 36132-2018: General principles for assessment of green factory

State Administration for Market Regulation, 2018

Introduction

Green manufacturing is not only an important means to solve national resources and environmental problems, an important task to achieve industrial transformation and upgrading, an effective way for the industry to achieve green development, but also an inevitable choice for enterprises to take the initiative to assume social responsibility. Factory is the main body of green manufacturing. The evaluation of green factories is helpful to set a benchmark in the industry, and to guide and standardize the implementation of green manufacturing.

Based on the existing relevant evaluation indexes and requirements, and the principles of comprehensiveness and systematization, this standard establishes the evaluation model of green factory that meets the needs of industrial development, aiming to give the comprehensive evaluation indexes and requirements of green factory.

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Foreword
Introduction

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11. Evaluation
  Appendix A (normative appendix) Calculation method of green factory performance index
  Appendix B (informative appendix) Example of green factory evaluation index
  Appendix C (informative appendix) Technical framework of green factory evaluation standard formulated according to this standard
  Reference

1.5.3 Europe

Europe has taken a leadership role in the development of energy standards for buildings and standards for components of HVAC systems. The bibliography section in the Appendix of this book contains a partial list of European Standards and ISO Standards for

ventilation systems. An editorial by Joap Hogeling (Editor-in-Chief, *RHEVA Journal*)[11] in the October *RHEVA Journal* describes what can the HVAC and building sector can do to meet the global warming challenge. As outlined in the amended European Union (EU) directive 2012/27/EU on Energy Efficiency, the approach is to push for more efficient technologies in heating and cooling in buildings and industry. The targets are a 32.5% reduction in energy by 2030 and a commitment to decarbonize the energy sector and to transition to a net-zero greenhouse gas emission by 2050. The key components are making use of efficient HVAC in buildings and industry based on need reduction, more efficient systems, and the use of renewable energy. For the EU countries, half of the energy used is for heating and cooling and 80% is consumed in buildings. The HVAC community must be fully engaged in order to meet the Paris Agreement targets of 2°C or preferably 1.5°C. It is also recognized that this transition must be cost-effective and include all environmental costs and must be implemented on a global scale.

Joap Hogeling stated clearly in his article that we have the technology today to reach the zero carbon emission buildings. What is lacking is a standard way to calculate the costs of carbon emissions and other environmental impacts in an honest and transparent way. A new set of EPB (Energy Performance Building) standards (www.epb.center) has been developed by CEN, the European Standards Association, to assess the energy performance of our buildings and systems in a transparent and consistent way. The *RHEVA Journal* (October 2019, p. 6–38)[11] outlines the status of implementation and challenges faced by some EU members (Italy, United Kingdom, Switzerland, Netherlands, Croatia, Romania, France) in implementing EPBD (Energy Performance Buildings Directive) and in connection to the use of EPB standards. A review of these articles from different countries outlines the challenges to adopt a rigorous, consistent, and transparent process in the evaluation of Energy Performance in the EU on EPBD of Buildings. This work in the EU on EPBD to develop laws and regulations and administrative procedures has been underway since 2003. The challenge will be to implement similar policies on a global basis and to have a transparent and valid model.

1.5.4 North America (example given is based on Ontario, Canada)

Industrial Ventilation within Ontario's environmental regulatory framework

Ontario's local air quality regulation O.Reg.419/05: Air Pollution—Local Air Quality (O.Reg.419/05)[12,13] works within the province's air management framework by regulating air contaminants released into communities by various sources, including local industrial and commercial facilities. The regulation aims to limit exposure to substances released into air that can affect human health and the environment, while requiring industry to operate responsibly under a set of rules that are publicly transparent. The regulation includes three compliance approaches for industry to demonstrate environmental performance and make improvements when required. Facilities can:

- meet the provincial air standard;
- request and meet a site-specific standard; or
- register and meet the requirements under a sector-based technical standard (if available).

Provincial air standards are used to assess a facility's individual contribution of a contaminant to air. Those facilities that demonstrate to the ministry in an Emission Summary and Dispersion Modelling Report (ESDM Report) using air dispersion models, which they can meet the air standards, generally do not need to take any further action.

The ministry publishes Guideline A-10[14]: Procedure for Preparing an Emission Summary and Dispersion Modelling Report. The guideline describes requirements for emissions and the modeling of the emissions to predict contaminant concentrations at points of impingement. The guideline requires that all sources of contaminants need to be included in the ESDM Report, including fugitive sources such as incomplete capture from ventilation of process operations (such as hooding) and particulate emissions from roof vents on a process building. The guideline also discusses a number of methods to estimate emissions such as emission factors, mass balance calculations, source testing, and engineering calculations. The engineering estimate can be based on operating conditions, data from literature, thermodynamic and physical properties as long as the approach is demonstrated through documentation and references to be based on sound scientific and engineering principles.

Under the technical standard compliance approach, industrial ventilation is a key part of two technical standards; the Foundries—Industry Standard[15] and the Metal Finishers—Industry Standard[16]. The ministry's analyses for both these sectors determined that fugitive emissions from processes were dominant sources that contributed to the maximum point of impingement concentrations (e.g., maximum off-site concentrations). Requirements related to the monitoring, assessment, and change management of industrial ventilation are included in these technical standards.

1.5.5 Japan

Profs. Kobayashi and Ito[17] of Tokyo Polytechnic University, Japan, presented a paper in 2003 entitled "Current Status of Research Activity for Industrial Ventilation and future Problems in Japan." The regulations for the working environment is covered by the Industrial Safety and Health Act in 1972. It is a typical regulation of specifications, which are continuously applied for the current regulations systems. Research and development is carried out on equipment such as ventilators and air pollution control equipment at industrial sites but technical publications on their research and development are very limited because of competitive concerns.

Fig. 1.3 shows regulations and standards for the indoor and outdoor environment.

For the indoor environment the concentration measurement for the specified contaminants is the whole working space. This system differs from the individual exposure management, which is the approach used in Europe and North America.

The Industrial Safety and Health Law: This law was enacted for the following three purposes:

1. maintenance of health,
2. prevention of exposure to hazardous contaminants, and
3. healthy workplace environment.

Specifications required for ventilation equipment in the regulation include:

1. the notification of an equipment plan (ventilation plan),
2. performance of installation of a total ventilation and local ventilation devices,
3. exhaust capability,
4. performance of a duct,
5. performance of a fan,
6. performance of air purification equipment,
7. performance of a total ventilation system, and
8. periodical and independent inspection of local ventilation system.

Japan Society for Occupational Health has given advice on the values of acceptable concentration of chemical materials and particulates for worker in factory. The acceptable concentration signifies the concentration, based on the criteria that the substance does not give harmful effect on the majority of workers in the space if the concentration of the substance is less than this value, even though they are exposed everyday by this concentration. This means the same as Threshold Limit Values of ACGIH.

In ventilation design of a factory, all engineers follow the regulations of performance of ventilation equipment and design for an industrial workplace so that worker exposure is less than the acceptable concentration recommended by the Japan Society for Occupational Health. Design ventilation systems include equipment such as canopy, hood, push–pull type ventilation, or a dust collector or air purification devises.

1.5.5.1 *The Basic Environmental Law*

It is the law by which the basic idea about environmental preservation is provided. The Basic Environmental Law defines the desirable standard



FIGURE 1.3 Governmental regulations and standards concerning industrial ventilation in Japan.

about the conditions on the environment in connection with air pollution.

1.5.5.2 Air Pollution Control Law

Air Pollution Control Law was defined and carried out for the purpose that regulates discharge of the detrimental substance generated in connection with the activity in a factory. This law defines the discharge standard of exhaust from a factory.

1.5.5.3 The Building Standard Law

Building Standard Law has defined the standard about the window or opening for ventilation of a room. This law has the technical standard of natural ventilation equipment and mechanical ventilation equipment.

1.5.5.4 Heating, Air-Conditioning and Sanitary Standard 102 Ventilation Standard

The ventilation standard Heating, Air-Conditioning and Sanitary Standard (HASS) 102 of The Society of Heating, Air-Conditioning and Sanitary Engineers Japan (SHASE Japan) is a technical standard about the ventilation in Japan. It has been renamed SHASE-S 102.

This standard was revised in 1997 and applies to the ordinary indoor environment mechanically ventilated such as habitable room, office space, attached spaces to those rooms, and the spaces for various facilities. Working space such as a factory is not specifically designed by this standard, but the concept to keep good condition of IAQ by ventilation and the technical process of SHASE-S 102 must have applicability to working spaces.

This paper by Kobayashi and Ito focuses on the framework of the standard, design criteria for acceptable concentration of indoor air pollutants, calculation method for ventilation requirement, and technical principles for construction of ventilation equipment. Main points for SHASE-S102 are as follows:

1. The amount of ventilation requirement is obtained by the emission rate and the design criteria for acceptable concentration of indoor air pollutants. In other words, the amount of ventilation requirement is calculated in consideration of the situation of space usage and the condition of air pollutant generation.
2. The kinds of indoor air pollutant prescribed for in this standard are CO_2, CO, suspended particulate, NO_2, SO_2, and HCHO.
3. Design criteria for acceptable concentration of CO_2 are provided by a general indoor quality index (1000 ppm) as well as one of pollutants influencing occupant's health (3500 ppm). It describes how to properly use these two indexes for each pollutant source.
4. When pollutants are not perfectly mixed with the room air, ventilation effectiveness is taken into account for calculation of the amount of ventilation requirements.
5. Also prescribed are technical principles for construction of ventilation equipment and test methods of ventilation performance after the construction.

1.5.5.5 Activity of Society of Heating, Air-Conditioning, and Sanitary Engineers of Japan

In Japan, Committee of Ventilation Design Method and Committee of Industrial Ventilation have existed from 1990 and 2000, respectively.

The Committee of Ventilation Design Method targets ventilation design in a general environment and the investigation of the latest overseas trend and the research activity is followed in order to have an up to date standard based on best practice from a global perspective. The main results of this committee are as follows:

1. Constitution of SHASE-S 102 Ventilation Standard
   SHASE-S 102 2011 Ventilation Standard was designed and revised by Committee of Ventilation Design Method.
2. Constitution of HASS 115 Measurement Method for Ventilation Efficiency in Occupied Zone has a present name of SHASE-S 115 2017
   SHASE-S 102 Ventilation Standard applies when pollutants are not perfectly mixed with the room air. The ventilation design using the concept of ventilation efficiency, for example, normalized concentration in an occupied zone "Cn", is recommended. In order to evaluate and measure the Cn value, HASS 115 was designed in 2002 and the latest version is SHASE-S 115 2017

The Committee of Industrial Ventilation targets ventilation design of working environment and the investigations of the latest overseas trends concerning industrial ventilation. The committee has made the survey of actual environmental condition in some industrial facilities and also investigated ventilation system of commercial kitchen, push−pull type ventilation equipment. The committee also makes a strong effort to build a network between researchers or engineers for exchanging technical information.

Prof. Toshio Yamanaka of Osaka University is the current chairperson of the technical committee of ventilation facilities in SHASE. At present, there are four small committees covering

1. Ventilation Effectiveness
2. CFD applications for Design of Environment and Building facilities
3. Modelling of Ventilation Components for CFD

4. Indoor Air Quality

New relevant Standard were added to the family of ventilation standard of SHASE

present name: SHASE-S 116
latest version: SHASE-S 116−2011
title of standard: Ventilation Rate Measurement of a Single Room Using Tracer Gas Technique
present name: SHASE-S 117
latest version: SHASE-S 117−2017
title of standard: Field Measurement Methods of Air Flow Rate for Ventilation and Air Conditioning Systems

1.5.5.6 Activity of other academic societies

In addition to SHASE, there are several academic societies that are intended to achieve a good environment for workers in Japan. These are Society of Industrial Health of Japan, Japan Industrial Safety and Health Association, Japan Association for Working Environment Measurement, and so on. The activities of these societies are mainly to inspect the ventilation equipment installed and to conduct surveillance for keeping a good environment by measurements of air quality and temperature in spaces, management of industrial safety and health, improvement measures against chemical substances, and promotion of making a safe and healthy workplace.

1.6 Opportunities

There is an urgent need for a simple holistic model to provide technical guidance for implementation of Industrial 4.0. Goodfellow[8] has proposed framework for a Smart Cleantech Model (SCM). Cleantech is a general term used to describe products, processes, or services that reduce waste and require as few nonrenewable resources as possible. The goal is to develop a simple generic software platform for a wide range of industries to improve global competitiveness. This needs to be a multidisciplinary approach (ventilation, sensors, AI) in a lab environment. This approach would require the integration of best practices for modeling, sensors, big data, and optimization (AI, deep learning). Fig. 1.4 identifies the framework for what SCM would look like for air (similar structures for the Cleantech model could be developed for water and solid waste).

The science and technology of industrial ventilation is at a crossroads and key decisions need to be made to capitalize on the unbounded opportunities.[8] Three key areas to be pursued are:

- Build better communication (scientific and engineering community of ventilation and contaminant control).
- Develop a global collaborative community.
- Embrace disruptive technologies (sensors, modeling, automation, AI, etc.).

In the area of better communications, it is important to recognize that there are many common areas of scientific research and engineering in the ventilation and contaminant control field. The goal is to develop a holistic approach for the science of ventilation. Fig. 1.5 illustrates many of the common areas of science and technology for the residential, commercial, and industrial sectors.

The second area is to develop a global collaborative network in the ventilation technology field. This global network would include scientific research (academic, research institutes), professional associations [ASHRAE, the Federation of European Heating, Ventilation and Air Conditioning (REHVA), SHASE, Industrial Ventilation Committee of ACGIH], international technical conferences (such as ventilation conferences), low-carbon economy, disruptive technologies (Industrial 4.0, AI, sensors, etc.), and scientific publications [revised *Industrial Ventilation IVDGB* (2020)]. Success depends on a holistic, multidisciplinary, and a sustainable funding model. An excellent example of this goal to breakdown the silos is the leadership shown by ASHRAE President Prof. Olesen in his August 2017 article in *ASHRAE Journal* entitled, Extending our Community.[18] The specific goals outlined in the report were the urgent need for the ventilation community to "extend the global community" and "to extend the technological horizons." A description of advances in this field by ASHRAE and founding members is listed next.

IEQ-GL (http://ieq-ga.net)

In October 2019 ASHRAE announced the incorporation of the IEQ Global Alliance (IEQ-GA) during a ceremony at the 40th AIVC Conference in Ghent Belgium.

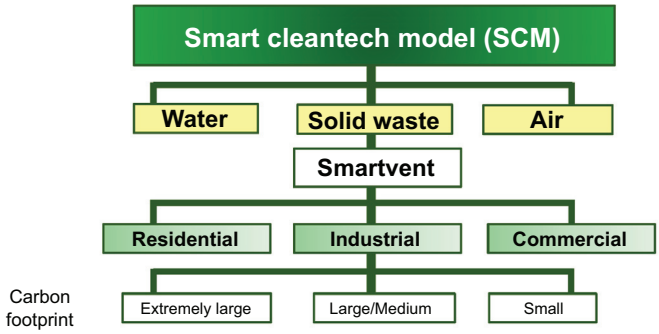


FIGURE 1.4 Smart Cleantech Model (SCM).

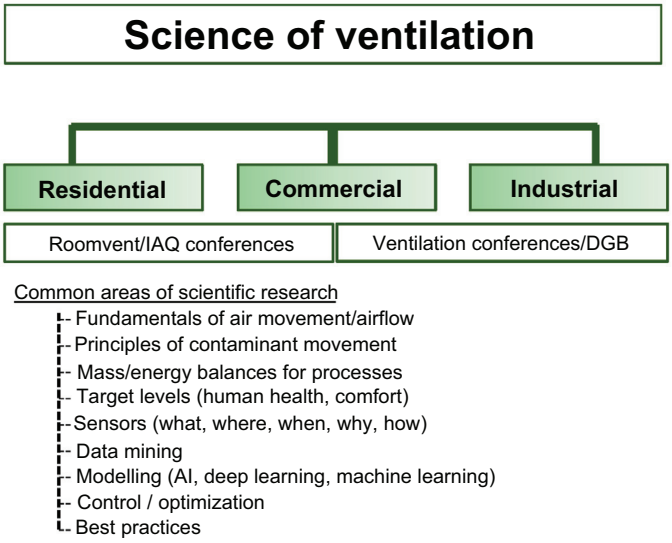


FIGURE 1.5 Science of ventilation.

The mission of the IEQ-GA is to promote and advocate for acceptable IEQ (thermal environment, IAQ, lighting, and acoustics) for building occupants globally while ensuring the knowledge from IEQ research is implemented in practice.

Founding members of the corporation are the Italian Association of Air Conditioning, Ventilation and Refrigeration (ACARR), and the American Industrial Hygiene Association (AIHA), the Air Infiltration and Ventilation Center (AIVC), the Indian Society of Heating, Refrigerating and Air Conditioning Engineers (ISHRAE), REHVA, and ASHRAE.

A recent press release by ASHRAE provides more details; "We are inspired by the forward-thinking approach IEQ-GA has in the building industry," said 2019–20 ASHRAE President Darryl K. Boyce, P.Eng. "It is critically important that we advocate for the well-being of the people who occupy our buildings. Through the collaborative efforts and resources of IEQ-GA member organizations, we will continue to provide safe buildings for generations to come."

The creation of the IEQ-GA was the result of a presidential initiative of Bill Bahnfieth, 2013–14 ASHRAE Presidential Member and current IEQ-GA Vice President, based on the report of a presidential ad hoc committee chaired by Bjarne Olesen, 2017–18 ASHRAE Presidential Member and current ASHRAE IEQ-GA Alternate Director. The committee was tasked with exploring ways in which industry groups could work together to address all aspects of IEQ and health.

A memorandum of understanding was established between the Air & Waste Management Association, the Indoor Air Quality Association, AIHA, AIVC, REHVA, and ASHRAE to form IEQ-GA. AiCARR and ISHRAE later became members of the alliance.

"An important attribute of IEQ-GA is that it is a nonindustrial alliance among societies representing members that promote IEQ" said Olesen. "The intent of IEQ-GA is to work with all partners of the value chain for indoor environmental quality including building research, comfort and health research building design, installation, commissioning, operation and occupant behavior. Furthermore IEQ-GA will work with industry organizations to help their members to provide products and services that promote IEQ."

The founding members represent professionals from various disciplines linked to indoor environment and are committed to work together and promote education, research, and knowledge exchange at a global scale, to develop standards, codes, guidelines, and advocacy of the general public in indoor environments around the world.

The IEQ-GA is currently seeking new members from all the involved sectors and disciplines to join forces. Sectors include architects, consulting and design, engineers, environmental professionals, industrial/occupational hygienists, health sector specialists, researchers, and industrial ventilation and contaminant control specialists.

About ASHRAE (ASHRAE.org)

Founded in 1894, ASHRAE is a global professional society committed to serve humanity by advancing the arts and sciences of heating, ventilation, air-conditioning, refrigeration, and their allied fields.

As an industry leader in research, standards writing, publishing, certification, and continuing education, ASHRAE and its members are dedicated to promoting a healthy and sustainable built environment for all, through strategic partnerships with organizations in the HVAC&R community and across related industries.

About AiCARR (www.aicarr.org)

Since 1960 AiCARR (Associazione Italiana Condizionemento dell'Aria, Riscaldamento e Refrigerazione), the cultural network for energy efficiency, has created and promoted culture in the field of technological systems for production, distribution, and use of thermal energy in both residential and industrial buildings.

About AIHA (aiha.org)

The AIHA is the premier association of occupational and environmental health and safety professionals. AIHA's 10,000 members play a crucial role in the front line of worker health and safety every day. Members represent a cross section of industry, private business, labor, government, and academia.

About AIVC (AIVC.org)

The AIVC is one of the projects/annexes running under the Energy in Buildings and Community Systems implementing agreement within the context of the

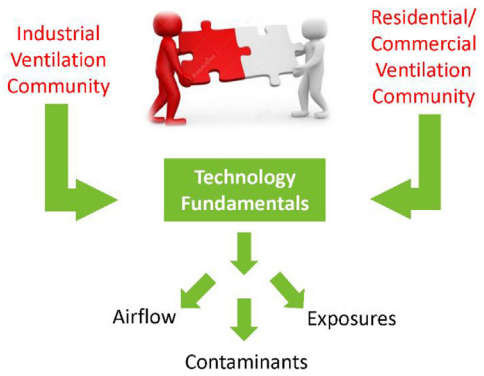


FIGURE 1.6 Communications model.

International Energy Agency. With the support of 17 member countries as well as key experts and two associations (REHVA and IBPSA), the AIVC offers the building sector, policy makers, and research organizations technical support aimed at better understanding the ventilation challenges and optimizing energy-efficient ventilation.

About ISHRAE (ishrae.in)

The ISHRAE has over 12,000 HVAC&R professionals as members and additionally there are 10,500 student-members. ISHRAE operates from over 40 chapters.

About REHVA (rehva.eu)

The Federation of European Heating, Ventilation and Air Conditioning associations was founded in 1953. It is an umbrella organization that represents over 120,000 HVAC designers, building services engineers, technicians, and experts across 27 European Countries.

Fig. 1.6 illustrates a model for better communications with the science and engineering community. The IEQ-GA corporation is a positive step to improve communication and to facilitate implementation of best practices for end users.

Disruptive technologies are happening at an accelerated rate and will have a major impact on the future directions of the science and technology of industrial ventilation. Disruptive technologies such as robotics, AI, models for low-carbon economy, and innovative sensors are impacting many sectors and researchers must embrace the cross-transfer of these technologies. For example, innovative sensors are being developed that are wireless, noninvasive, cheap, remote, and in situ. Many of these sensors have wide applications for many different sectors (i.e., autonomous vehicles) for advanced design of ventilation systems.

Advanced design of industrial ventilation systems based on best practices on a global basis will play a major role in meeting our common goal to meet climate change objectives on a global basis. The "big picture" goal for IVDGB (2020) is to provide a road map for the future of industrial ventilation on a global basis.

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CHAPTER

2

Terminology

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2.1 Main definitions

Fig. 1.1 outlines the main terminology and components of industrial ventilation systems. These technologies and systems are described in technical details Volume 1 in Chapter 7, Principles of Air and Contaminant Movement Inside and Around Buildings; Chapter 8, Room Air Conditioning; and Chapter 9, Air-Handling Processes, of Volume 1. In Volume 2, engineering design for industrial ventilation systems is covered Engineering Design and Applications (Chapter 10, Local Ventilation and Chapter 13, Gas Cleaning). In the following sections the main definitions for zones, industrial air-conditioning systems, local ventilation systems, and gas cleaning are discussed.

2.2 Zones

Typically, industrial premises have, in one space, zones with different activities that require different target levels for the indoor environment and its control. These target levels may be determined for the whole area or locally, if only a part of the space needs to be controlled. In addition to the main controlled zone, there may be one or more local controlled zones with targets different from those in the main controlled zone. For example, machines equipped with electrical components require a very clean and accurately controlled indoor environment, while the unoccupied zone near the ceiling needs only roughly controlled protection against structural damages.

In industrial premises the target levels of indoor air quality, as well as other targets (e.g., emissions), shall be specified zone by zone.

A *controlled zone* is a zone in which the thermal and air purity (quality) conditions are controlled to their specified levels. The two categories of controlled zones are as follows:

- The *main controlled zone* is normally a large area, which is often the same as the occupied zone.
- A *local controlled zone* is an area where the air is controlled locally; the control requirements may be for worker protection and comfort, for process control, or for production protection.

An *uncontrolled zone* is a zone in which the thermal and air purity (quality) conditions are not specified or controlled.

Note: There may also be uncontrolled zones near to the processes inside the main controlled zone.

Capture zones are zones in which source emissions will be captured by a source-capturing system, (local exhaust ventilation) and where the capture efficiency is determined and shall be maintained over the working period. From the pollutant concentration point of view, the capture zone is uncontrolled (e.g., workers shall not enter a capture zone without additional protection).

2.3 Industrial air-conditioning systems

Room air-conditioning (see Fig. 2.1) systems are used to control the main controlled zone. Systems can be divided into subsystems, for example:

- air-handling systems
- air distribution systems (ductwork)
- room air distribution systems
- ventilation systems
- room heating and cooling systems
- main exhaust systems
- discharge systems: stacks, environmental dispersion

Note: Air distribution systems are not ventilation or air-conditioning systems. For example, mixing air distribution and displacement air distribution are methods to bring the supply air to the treated space.

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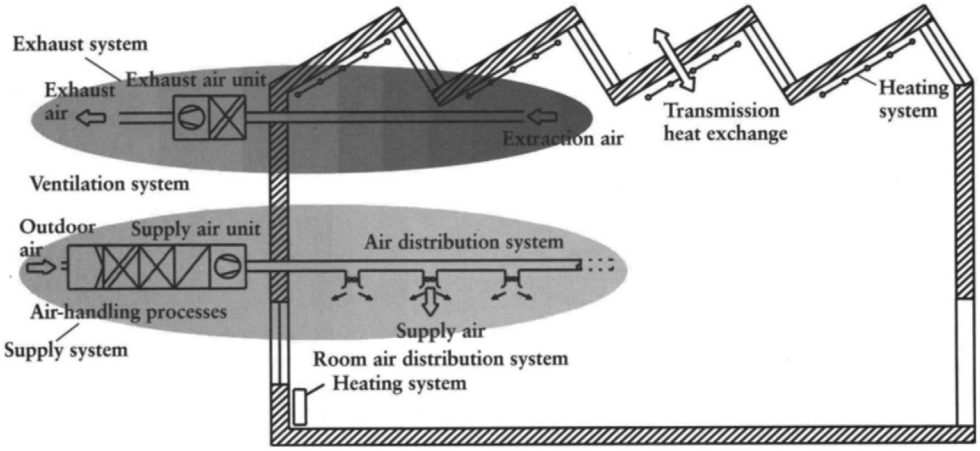
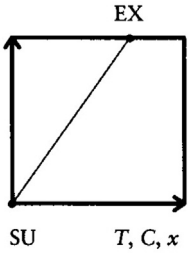
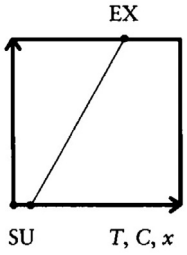
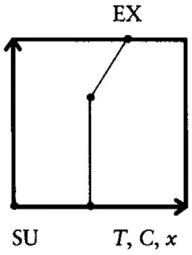
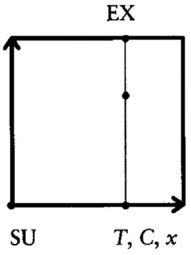
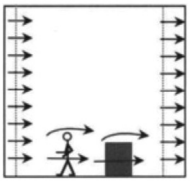
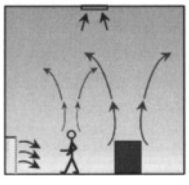
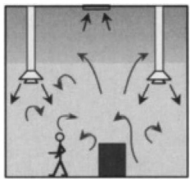
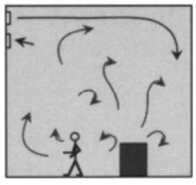


FIGURE 2.1 Main air-conditioning systems.

TABLE 2.1 Ideal room air-conditioning strategies.

	Strategy			
	Piston	**Stratification**	**Zoning**	**Mixing**
Description	To create unidirectional airflow field over the room area by supply air	To support flow filed created by density differences by replacing the airflow out from the room area with supply air	To control air conditions within the selected zone in the room by the supply air and allow stratification of heat and contaminants in the other room areas	To provide uniform conditions throughout the ventilated space
Heat, humidity, and contaminant distribution				
Main characteristics	Room airflow patterns controlled by low-momentum unidirectional supply airflow, strong enough to overcome disturbances	Room airflow patterns controlled mainly by buoyancy; supply air distribution with low momentum	Room airflow patterns controlled partly by supply and partly by buoyancy	Room airflow patterns controlled typically by high-momentum supply airflow
Ideal contaminant and heat removal efficiency				
Typical application (example of a general room air distribution method)				

x-Axis: *C*, mg/m^3, g/kg; *y*-axis: room dim. (e.g., height). *EX*, Exhaust; *SU*, supply.

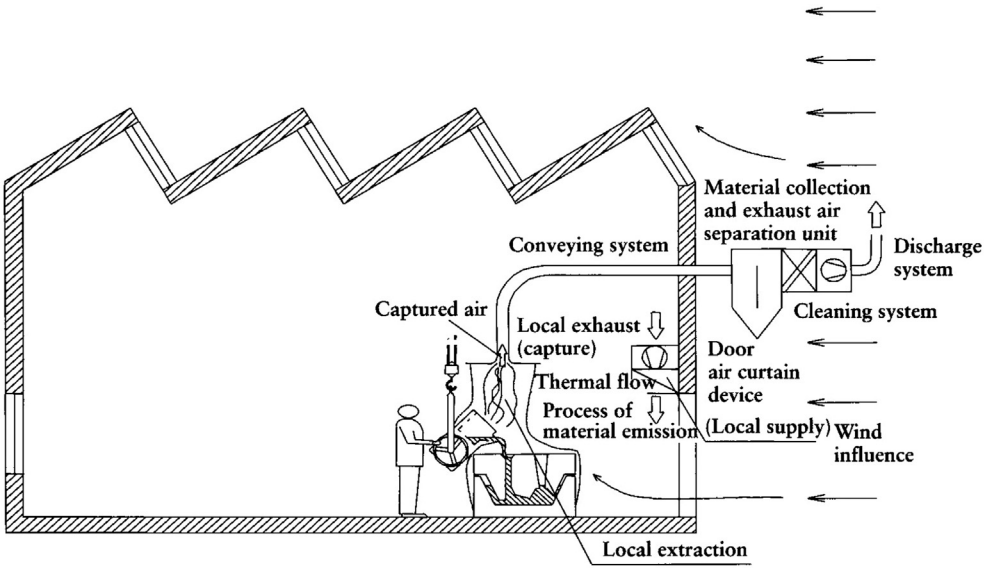


FIGURE 2.2 Local ventilation systems.

Discharge systems are used to discharge exhaust air to the outdoors in such a way that harmful spreading of pollutants to the environment and back indoors is avoided.

A detailed description of the methods for room air conditioning is presented in Chapter 8, Room Air Conditioning. Table 2.1 summarizes the strategies.

2.4 Local exhaust ventilation systems

Local ventilation systems (see Fig. 2.2) are used for local controlled zones. These systems are based on engineering design methods for local protection. Primarily, local protection should be made using process methods such as encapsulation or process modification (see "Design Methodology," Chapter 3: Industrial Ventilation Design Method, and also Chapter 10 in Volume 2). Another use for local ventilation systems is source capturing.

2.5 Gas-cleaning systems

These include equipment for supply air and equipment for exhaust air and gases. Cleaning of supply air is normally called air filtering, when the contaminant concentration upstream from the air filter is less than, for example, 1–2 mg/m^3. Also, chemical filtration can be applied to supply air.

There are many types of cleaning systems and equipment, for example:

- dynamic separators (cyclones)
- fabric filters
- wet separators (scrubbers)
- electrostatic precipitators

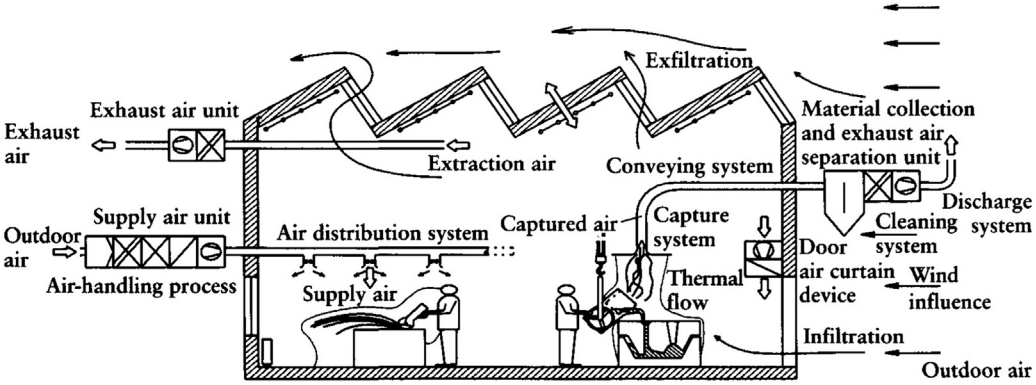


FIGURE 2.3 Definitions of air.

- desulfurization equipment (SO_x control)
- denitrification equipment (NO_x control)

These types of gas-cleaning equipment are described in technical detail in Chapter 13 (Volume 2).

2.6 Definitions of types of air

The definitions of the types of air are presented in Fig. 2.3. Fig. 1.1 also shows the types of air for manufacturing, control rooms, and processing plants:

- supply air
- exhaust air
- extract (main, local) air
- recirculated air
- outdoor air
- transferred air (two types—direct or via an air-handling system)
- infiltration
- exfiltration
- indoor air

Transferred air can be intentionally or unintentionally transferred. Exhaust air is air leaving the building. Extract air is air leaving the room (it may be partly returned into the room).

CHAPTER

3

Industrial ventilation design method

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3.1 General

Environmental issues are being addressed more and more heavily in today's society. Thus it is natural that in industrial processes and in their design, environmental effects are also considered over the whole life cycle. The life cycle of the production process can be divided into four parts: design, construction, operation, and end of the process. Each part consists of different tasks. Design methodology is a part of the whole process during whole life-span period. The life cycle of the production process is illustrated in Fig. 3.1. Also, in Table 3.1, short descriptions and lists of tools for different tasks are given. Moreover, the ventilation methods such as traditional mixing ventilation and displacement ventilation are clarified in detail. Additionally, a novel ventilation method—attachment ventilation was proposed by Angui Li. Attachment ventilation combines the advantages of both mixing ventilation and displacement ventilation, and avoids a series of shortcomings such as low temperature efficiency for mixing ventilation and occupying work spaces for displacement ventilation. Attachment ventilation focuses on the environment control of occupied or conditioned zone. The theory and design method of attachment ventilation is presented. Furthermore, novel low-resistance components of ventilation duct system have received more and more attention for their industrial applications. This chapter also introduces the low resistance components and design methods including tee, elbow and coupling bends, etc.

Common items should be taken into account as follows:

- energy consumption
- ecological issues
- costs (construction, life cycle)

The design methodology is a description of a technical design process that covers the whole lifetime of the production process. Most decisions concerning industrial ventilation are made at the design stage and are reflected in construction, operation, maintenance, service, etc.

The first and most important aim of design methodology is to produce, by systematic analysis, a description of the design procedure that is commonly accepted and used in every process in different markets. The idea is to make a description of the technical process of design, in other words, to answer two questions:

- What is to be made clear and done during the design procedure?
- In which order are the tasks to be done?

The design methodology *does not* take a position on who does this or that task. That is part of administrative or commercial flow that varies in different parts of the world and even in different projects in one country.

3.2 Design methodology description

3.2.1 Explanations of the design process

Basic elements in the methodology can be presented in several ways. Table 3.1 gives an idea of the whole contents. In addition, decision trees are needed, because the design process requires many back couplings that cannot be illustrated in table form. The decision tree technique is a tool for dividing a process, here design methodology, into subtasks, which have their accurate inputs and outputs. The order of the tasks is chosen so that the data needed to do a task are given or calculated before that task to minimize the number of back couplings. Thus the tree guides the right execution order of

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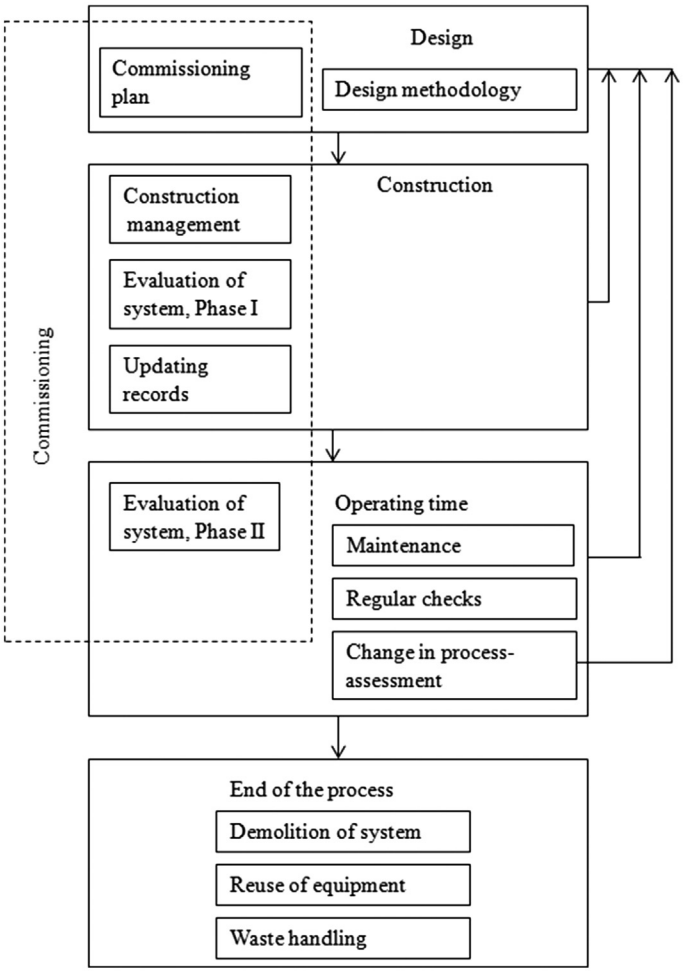


FIGURE 3.1 Life cycle of the production process.

the subtasks. It also serves as an internal quality guidance tool for design process, because the quality of the preceding subtasks' results will be assessed in the next task, where they are used as input data.

A decision tree for design methodology is illustrated in Fig. 3.2.

Explanations of Fig. 3.2

Step 1: Given data

- Identify and collect data that depend only on the site location and that do not change during the design process, such as outdoor conditions.
- The division of the data is shown in Fig. 3.3.
- The *tools* for this task are:
  - databases and
  - weather models.

Step 2: Process description

- Identify the industrial process and subprocesses.
- Identify possible emission sources, occupational areas, effects of environmental parameters, needs for enclosure, and ventilation equipment.
- Divide process into parts such that their inputs from and outputs to the environment can be defined.
- When the process or subprocess is not well defined during the initial period of design, obtain the data from similar processes based on recent successful practices. Obtain and use more precise data as soon as possible.

Step 3: Building layout and construction

- Collect properties of building layout, structures, and openings and their properties as basic values for load calculations.
- Complete zoning of building based on division of the process and building layout.
- Make space reservations and add structures needed for ventilation equipment.

Step 4: Target level assessment

- Define target levels for indoor (zones) and outdoor (exhaust) conditions.
- Specify design conditions in which the target levels are to be met.
- Define target levels for the ventilation system, such as reliability, energy consumption, investment, and life cycle costs.

For the decision tree of the target level assessment, see Fig. 3.4.

Explanations of Fig. 3.4

1: Musts

- Clarify requirements due to laws, regulations, and standards related to legislation, processes, and equipment.

2: Needs

- Clarify standards not related to legislation, such as those related to human comfort, guidelines, codes of practice, and custom needs.

3: Target levels

- Decide target levels based on musts and needs.

4: Design conditions

- Suggest and agree with customer on the outdoor conditions in which the target levels have to be met, for example, absolute maximum temperature versus 95% temperature.

5: Reliability

- Study the reliability requirements of the process with the customer.
- Define and get the customer's acceptance of the needs for ventilation system reliability, for example, what is the allowed break-off time.

The *tools* for this task include:

- laws,
- regulations,
- standards, and
- guidelines, codes of practice.

TABLE 3.1 Design methodology and associated tools.

Design criteria		Tools
Given data	Data dependent only on the site location and do not change during design process.	Database weather model
Process description	Purpose: Identification of possible emission sources, occupational areas, effects of environmental parameters to production, needs for enclosure, and ventilation. Division of process into such parts that their inputs and outputs can be defined.	Expert system databases
Building layout and construction	Collection of properties of building layout, constructions, windows as basic values for load calculations.	Databases
Target level assessment	Prediction of target levels for indoor and outdoor conditions based on requirements of laws and orders, human health, production processes and equipment, and type of premises and construction. Needed as a standard to which system solutions are compared.	Classification regulations
Source description	Characteristics of sources and calculations methods for load calculation.	Calculation models
Calculation of local loads	Calculation of loads from different subprocesses.	Building model
System performance		
Local protection	Examination of subprocesses in order to provide proper working conditions by it or to reduce emissions to environment. In case use of local protection system effect on exposure of the process, load calculations shall be revised.	Calculation models for prefabricated products
Calculation of total loads	Calculation of total loads from different subprocesses and environment.	
System selection	Based on technical calculations, conditions achievable by different systems are compared to target levels to identify acceptable systems, which are compared to each other, and the most suitable system is selected on the basis of different parameters: Power and energy consumption and investment and life cycle costs.	System description and characterization Heat, mass, and energy balances
Equipment selection	Based on technical specification, acceptable equipment is identified. Final selection is made on the same basis as in selection of system.	Equipment selection programs and diagrams
Detailed design	Includes the following subtasks: detailed design of ventilation systems, design of adjustment, and control system, commissioning plan.	Duct design programs and diagrams CAD solutions (drawing tools)
Construction management		Mounting design Materials handling Commissioning plan
Evaluation of system, Phase II	Inspections and start-up and functional performance tests.	Performance tests Checks Measurements
Updating records	System descriptions user instructions.	CAD programs (drawing tools)
User training	Training of operating and maintenance people.	Lectures Practical training
		Participating in the evaluation
Operating time (use)		
Evaluation of system, Phase	Functional performance tests in different situations.	Performance tests
Maintenance	Measures to keep ventilation system operating at the specified level economically.	Maintenance plan Monitoring Health surveillance
Regular checks	Measures to secure that system and equipment performance are unchanged. In addition, evaluation of the system toward new requirements.	Energy audits Environment audits Assessment (COSSH)

(*Continued*)

TABLE 3.1 (Continued)

Design criteria		Tools
Process changes	Adoption of the process changes by evaluating influences to ventilation system and to conditions. When needed, renewing of ventilation system to meet targets.	Assessment (COSSH) System simulation
End of process		
Demolition of system	Design and completion of demolition, taking into account possible risks (e.g., asbestos).	Assessment of the risk to health Special working methods
Reuse of equipment	Evaluation of the value and usefulness of the equipment and components.	Condition analysis of the equipment
Waste handling	Separation of different types of waste. Handling of problem waste. Recycling materials.	Records of materials used Marking of components

Administrative Flow—Quality Assurance: Prestudies, Design, Construction, and Maintenance.

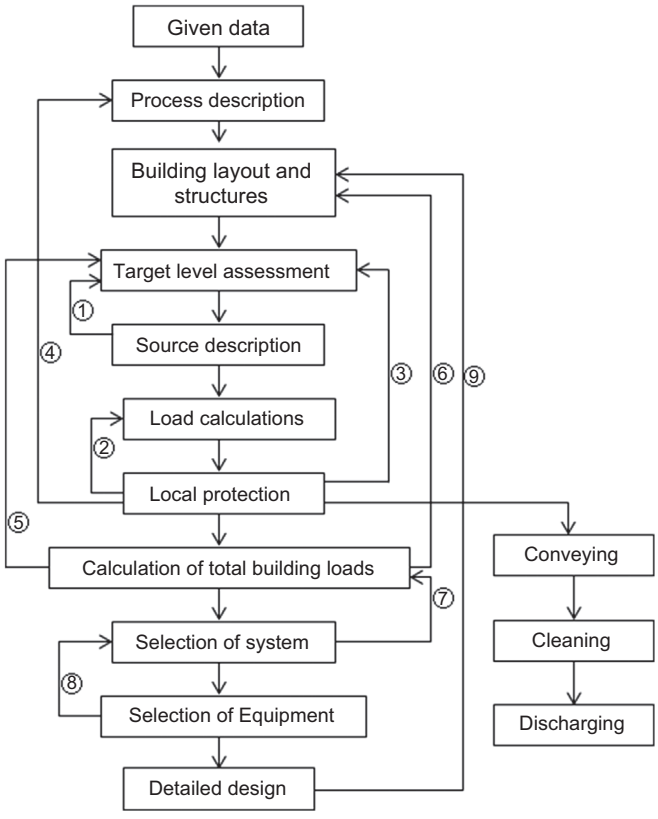


FIGURE 3.2 Decision tree of design process.

Step 5: Source description

Clarify characteristics of the sources and calculation methods for calculation of local loads. See Fig. 3.5.

The *tools* for this task include:

- standard tests,
- physical modeling,
- databases, and
- guidelines.

Step 6: Calculation of local loads

- Calculate loads from individual sources to the environment.

Step 7: Local protection

- Examine subprocesses (sources) in order to provide proper working conditions near them (local zones) or to reduce emissions to the environment.

Step 8: Calculation of total building loads

- Calculate total loads (heat, humidity, and contaminants) from different subprocesses and environment (building) to ventilated enclosure (zones).
- Take into account that loads are usually time dependent.

Step 9: Selection of the system

- Select acceptable systems based on target levels.
- Compare acceptable systems in order to choose the most desirable one.
- Use systems that allow maximum flexibility in airflow rates and control strategies when selection of systems is based on inaccurate (preliminary) data on production processes, volumes, and raw materials to be used in the building. Emission rates from these processes and total loads might be changed during the detailed design step.
- Consider constraints on the system selection, if some equipment has been already selected and installed in the earlier design period.

Step 10: Selection of equipment.

- Work out performance characteristics to the equipment.
- Select acceptable equipment based on performance characteristics.

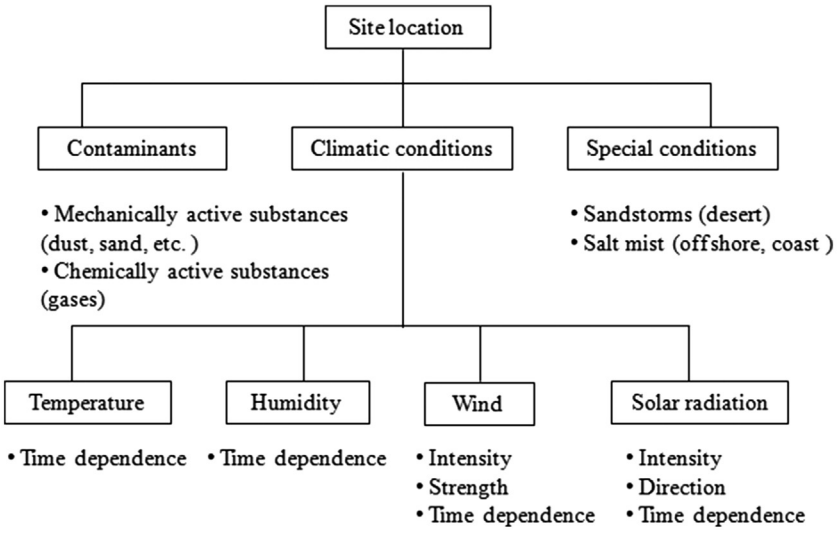


FIGURE 3.3 Given data.

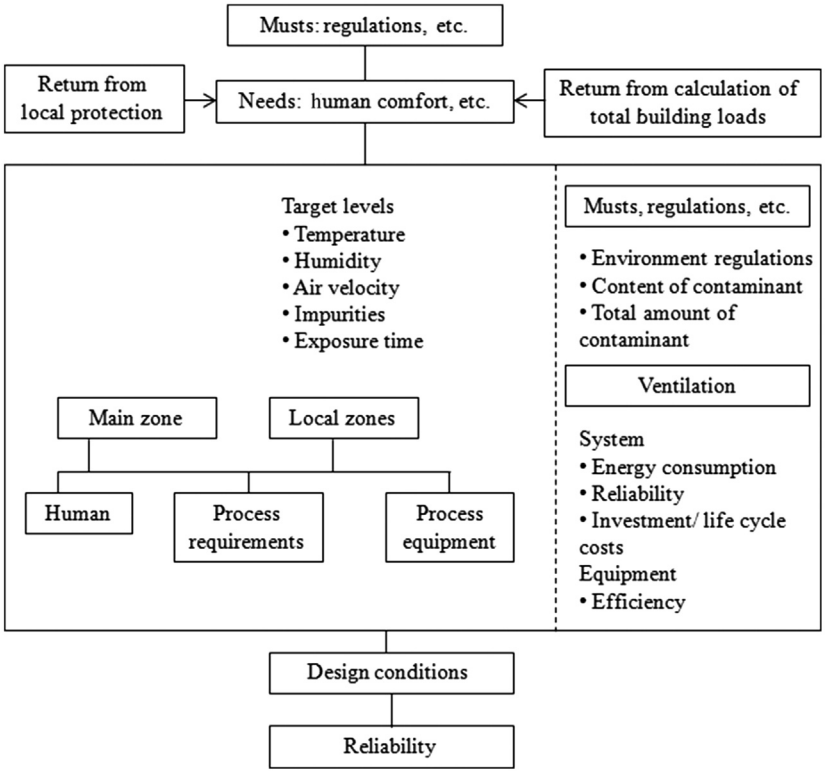


FIGURE 3.4 Target level assessment.

- Compare acceptable equipment in order to choose the most desirable one.
- Make a technical specification of selected equipment.

Step 11: Detailed design

- Do detailed layout and dimensioning design.
- Design adjustment and control system.
- Consider special issues such as thermal insulation, condensation risks, fire protection, and sound and vibration damping.
- Make commissioning plan.

Steps 12–14: Design of conveying, cleaning, and discharge of the pollutants.

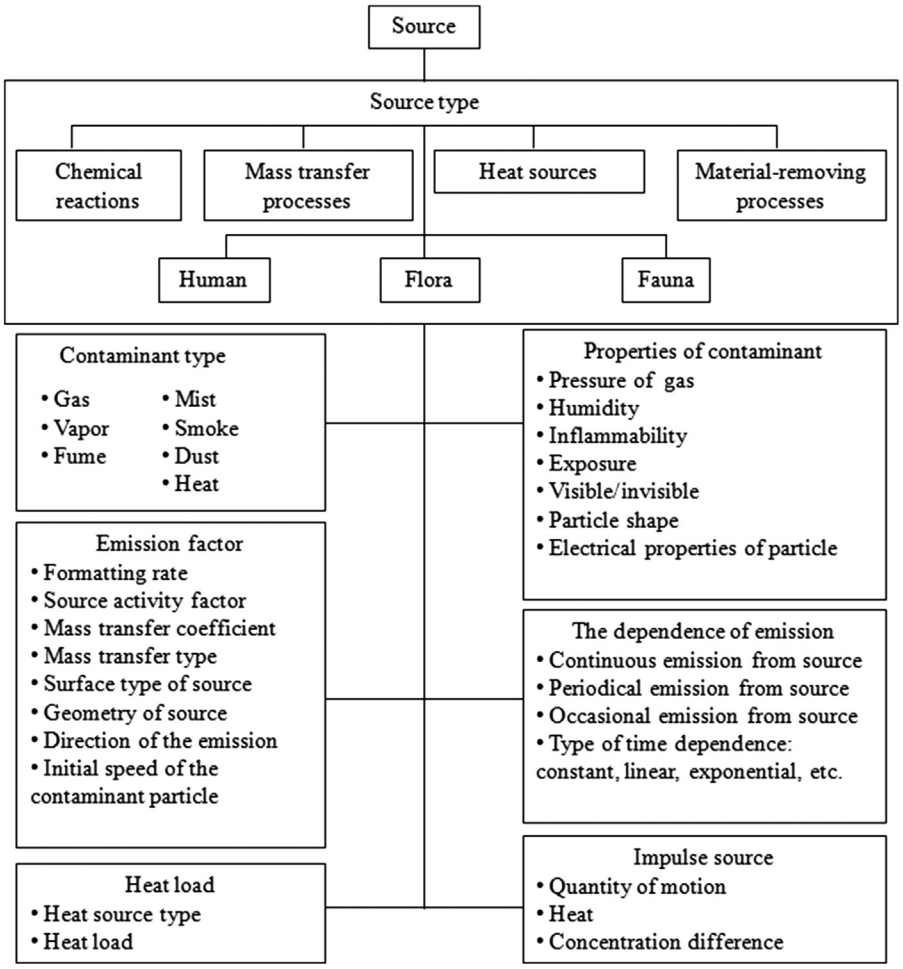


FIGURE 3.5 Source description, characteristics of the source, and calculation methods for load calculation.

3.2.2 Explanations of back couplings (BC) in the design process

BC 1: Source description → target level assessment

If some new agent is identified, the target level has to be defined for that agent too.

BC 2: Local protection → calculation of local loads

If the local protection has an effect on the exposure of the source, recalculate the load.

BC 3: Local protection → target level assessment

If defined target levels cannot be reached, reconsider target levels.

BC 4: Local protection → process description

Consider whether there is some process method to protect source/environment. In that case, return to process description. For example, if thermal insulation is needed to reduce loads, consider what influence that has on the process itself (insulation may, e.g., lead to a need to change material of equipment.).

BC 5: Calculation of total building loads → target level assessment

- Consider whether some source has governing role to total loads. At least, if returned from selection of system, choose one of the two following actions:
- If some source has governing role over total loads, reconsider the target level of that local zone in order to reduce loads.
- If there is no source that governs total loads, reconsider the target level of main zones in order to reduce loads.

BC 6: Calculation of total building loads → building layout and structures

If building loads have governing role over total loads, reconsider whether there is something that can be done with constructions (e.g., thermal insulation) to reduce loads.

BC 7: Selection of system → calculation of total building loads

If target levels cannot be achieved with any system or it is not economically possible, check whether something can be done with loads.

BC 8: Selection of equipment → selection of system

If no acceptable equipment exists, reconsider selection of system with available equipment.

BC 9: Detailed design → building layout and structures

- Identify openings needed in structures.
- Identify additional space and structure needs for ventilation installations.

3.3 Determination of ventilation airflow rate

3.3.1 Calculation of ventilation airflow rate

For general dilution ventilation the ventilation rate can be calculated in three states that are shown next[1].

1. *Ventilation airflow rate under unsteady state*

The ventilation rate under unsteady state can be calculated as the following equation:

$$L = \frac{x}{C_2 - C_0} - \frac{V_f}{t} \cdot \frac{C_2 - C_1}{C_2 - C_0} \tag{3.1}$$

where L is the ventilation rate, m^3/s; C_0 is the contaminant concentration of supply air, g/m^3; x is the release rate of pollutant, g/s; V_f is the volume of the room, m^3; t is the time for ventilation, s; C_1 is the initial contaminant concentration of indoor air, g/m^3; and C_2 is the contaminant concentration of indoor air after t seconds, g/m^3.

As seen in the above mentioned formula, when the initial concentration is zero and the time t tends to infinite, the concentration of the indoor harmful substance tends to be stable. Thus it reaches the stable state and there is the relation as shown in the following equation:

$$C_2 = C_0 + \frac{x}{L} \tag{3.2}$$

2. *Ventilation airflow rate under steady state*

a. Ventilation rate needed to eliminate waste heat

$$L_h = \frac{Q}{c\rho(t_e - t_0)} \tag{3.3}$$

where L_h is the ventilation airflow rate needed to eliminate waste heat, m^3/s;
c is the specific heat capacity of air, $kJ/(kg \cdot ^\circ C)$; Q is the waste heat in room, kW; t_e is the temperature of exhaust air, °C; t_0 is the temperature of supply air, °C; ρ is the density of air, kg/m^3.

b. Ventilation airflow rate needed to eliminate moisture load

$$L_m = \frac{G_m}{\rho(d_e - d_0)} \tag{3.4}$$

where L_m is the ventilation airflow rate needed to eliminate moisture load, m^3/s; G_m is the waste moisture, g/h; d_e is the moisture content of exhausted air, g/kg dry air; and d_0 is the moisture content of supply air, g/kg dry air.

c. Ventilation rate needed to eliminate pollutant

$$L_p = \frac{x}{C_m - C_0} \tag{3.5}$$

where L_p is the ventilation rate needed to eliminate pollutant, m^3/s, and C_m is the maximum permissible contaminant concentration for indoor air, g/m^3.

d. When waste heat, residual humidity, and pollutant released simultaneously in the room do not have superimposed harmful effects on human health, the ventilation airflow rate is defined as the maximum value calculated earlier.

e. If several indoor pollutants are released simultaneously and the effect of them on human body is superimposed, the ventilation rate should be calculated separately, and then the total ventilation rate should be taken as the sum of their parts.

The ventilation rate actually required should be greater than the calculated air rate, because the distribution of pollutant and ventilation airflow is not very uniform. In addition, it also needs some time for fresh air diluting pollutant. In the air near the harmful source, the concentration of harmful substance is higher than that of the average indoor air.

3. *Ventilation airflow rate calculated using air exchange rate*

When the pollutant diffused into the room cannot be calculated in detail, the total ventilation rate can be determined by the method of air exchange rate, as shown in the following equation:

$$L = nV_f \tag{3.6}$$

where n is the air exchange rate, and it can be found in relevant HVAC design manuals; V_f is the volume of the room, m^3.

3.3.2 Heat load

It is difficult to calculate the heat load in practical engineering theoretically for the complex site conditions. Therefore designers can refer to related reference for design calculation. The main sources of heat load can be seen as follows[1]:

- heat released from industrial furnace,
- heat released from electric furnace,
- heat released from metal cooling,
- heat released from electric equipment and welding equipment,
- heat released from generator unit and charging unit,
- heat released from lighting equipment,
- heat released from chemical reaction,

- heat released from surface of hot water tank,
- heat released from steam forging hammer,
- heat released from steam heating tank, and
- heat released from human body.

3.3.3 Moisture load

Moisture load of the occupied zone basically include[1]:

- moisture load from the open water surface or moist surface,
- evaporation moisture load from the hot water surface flowing along the ground,
- moisture load from the machine emulsified coolant,
- moisture load from gas combustion, and
- moisture load from gas combustion.

3.3.4 Emission rate of pollutants

In the occupied zone, the main sources of pollutant gases are as follows:

- pollutant gases emitted during combustion,
- fume leakage from furnace crevice,
- hazardous gases leaking from insecure places of equipment or pipeline,
- pollutant gases emitted from diesel engines, and
- evaporation of liquids (except of water).

For the complexity of the production process, the amount of dispersion of moisture and emission of polluted gases are generally determined by empirical data from field measurement and investigations.

3.3.5 Calculation of air balance and heat balance

1. *Calculation of air balance*

   Airflow rate balance is the balance of air quality in and out of buildings as expressed by the following equation[2]:

$$G_{nv} + G_{ms} = G_{ne} + G_{me} \tag{3.7}$$

where G_{nv} is the natural air intake rate, kg/s; G_{ne} is the natural exhaust rate, kg/s; G_{ms} is the mechanical supply air rate, kg/s; and Q is the mechanical exhaust rate, kg/s.

2. *Calculation of heat balance*

Heat balance means that the total heat gained in a ventilated room equals the total heat loss, so that the temperature of the ventilated room remains unchanged. The heat balance calculation is complicated by the variety of industrial plants, the complexity of the equipment, and the difference in ventilation systems. When both mechanical ventilation and circulating air are used, it can be calculated according to the equation[3]:

$$\left(\sum Q_n - \sum Q_s\right) + G_e c(t_n - t_w) = G_r c(t_{rs} - t_n) + G_{ms} c(t_{ms} - t_w) \tag{3.8}$$

where $\sum Q_n$ is the total heat loss of the heat absorption of envelope structure and material, kW; $\sum Q_s$ is the total heat release from indoor equipment and radiators, kW; G_e is the exhaust airflow rate, kg/s; G_r is the recycling airflow rate, kg/s; G_{ms} is the mechanical supply airflow rate, kg/s; t_n is the indoor air temperature, °C; t_w is the outdoor heating or ventilation design air temperature, °C; t_{rs} is the recirculating supply air temperature, °C; and t_{ms} is the mechanical supply air temperature, °C.

3.4 Design for ventilation system

3.4.1 Principle of ventilation design

The principles of dilution ventilation system design are as follows[4]:

- Locate the exhaust openings near the sources of contamination, if possible, in order to obtain the benefit of "spot ventilation."
- Locate the air supply and exhaust outlets to make sure that the air passes through the contaminated zone. People should remain between the air supply and the source of the contaminant.
- Replace exhausted air with supply air system. The supply or replacement air should be heated or possibly cooled to satisfy the temperature requirements of the space. Diluted ventilation systems usually handle large amount of airflow rates by means of low-pressure fans. Therefore adequate supply airflow rate must be provided if the system is to operate satisfactorily.
- Avoid reentry of the exhausted air by discharging the exhaust outlets high above the roof line or by assuring that no window, outdoor supply air intakes, or other such openings are located near the exhaust discharge.

3.4.2 Mixing ventilation

1. *Introduction*

   Mixing ventilation system combines both mechanical and natural ventilation aiming to dilute polluted and warm or cool room air with cleaner and cooler or warmer supply air.

   With a ventilation system based on the mixed principle, makeup air is supplied to the room with high initial mean velocity, and the established

velocity gradients generate high turbulence intensity aiming to promote good mixing for the room air and make the temperature and pollution concentration uniform[5].

2. *Air distribution of mixing ventilation*

Some typical air distribution schemes for applications in large enclosures with high ceilings are shown in Chapter 6 of the *REHVA Guidebook No. 19*[5], as seen in Fig. 3.6. By using high initial velocity and momentum flux, it is possible to guarantee the required mixing in the occupied zone of large enclosures.

In applications where the ceiling height is about 3 m or less, it is practical to utilize surfaces (ceiling and walls) for installation of air supply diffusers in order to guarantee good mixing and low air velocities in the occupied zone. Figs. 3.7–3.9 show three typical air distribution design methods for this situation[5].

Characteristics of the ceiling supply scheme are as follows:

- Suspended ceiling (exposed installation is also possible).
- High induction rate with short throw length in order to obtain high cooling capacity.
- Air distribution may be influenced by high heat gains such as warm windows.
- Throw pattern control is needed to ensure good performance in heating mode and to prevent temperature gradient.

Characteristics of the wall supply scheme are as follows:

- During warm periods, thermal plumes may affect the performance causing early jet detachment and draught.
- Not suitable in spaces with high cooling loads.

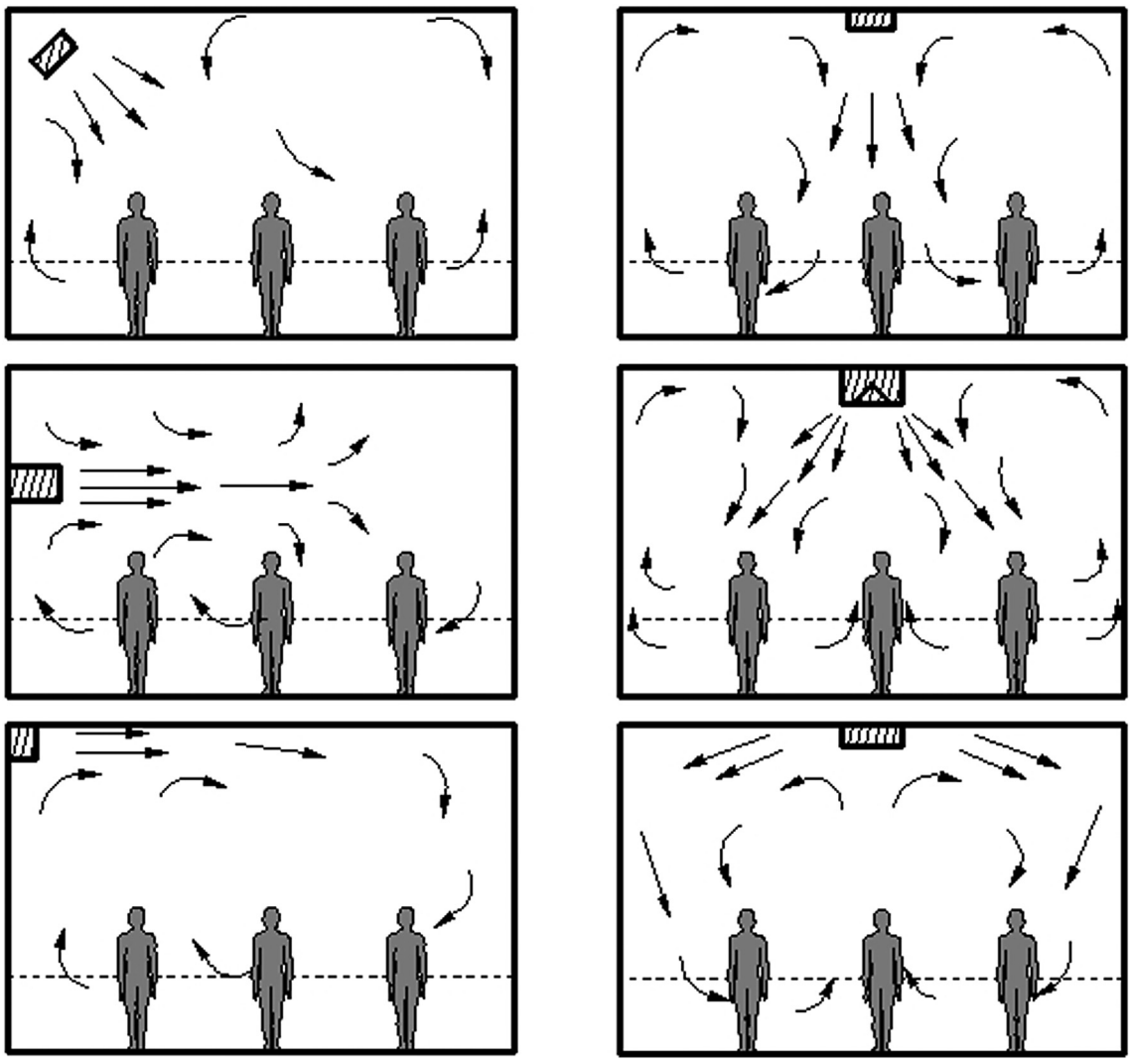


FIGURE 3.6 Typical air distribution schemes in large enclosures.

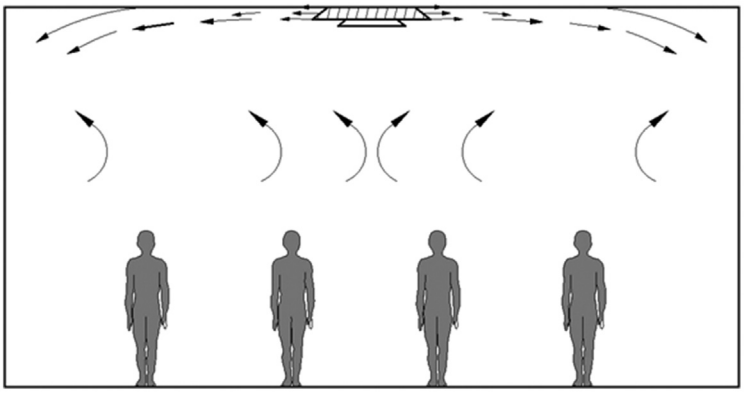


FIGURE 3.7 Ceiling supply air distribution method.

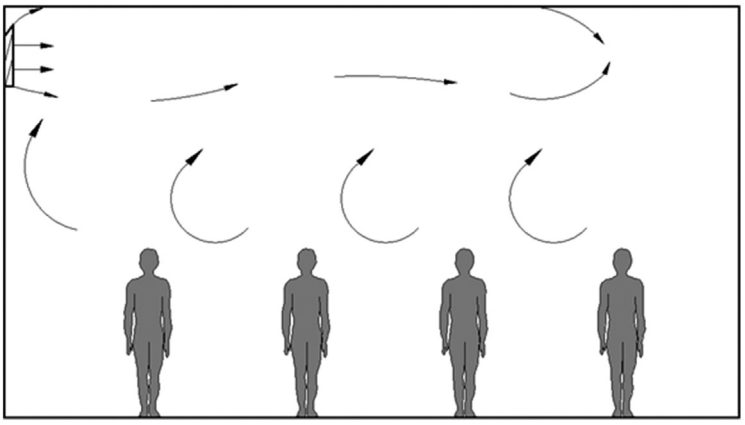


FIGURE 3.8 Wall supply air distribution method.

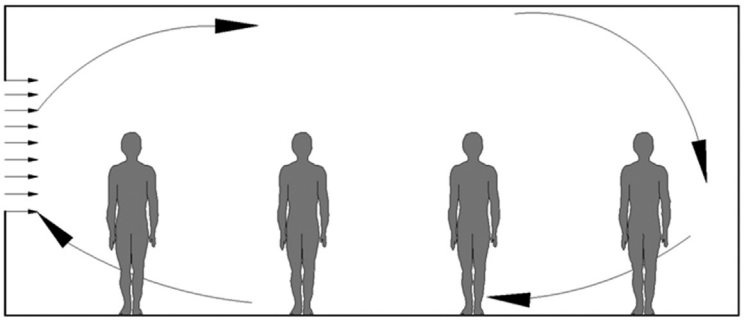


FIGURE 3.9 Window sill supply air distribution method.

- During cold periods, high velocities close to the floor can exist.
- In heating mode, continuous heating below window is required in order to avoid draught risk.

Characteristics of the window sill supply scheme are as follows:

- Initial velocity of supplied jet should be high to reach required throw length, l_{TH}, called throw for short, defined as the distance from the opening to the location where the maximum velocity in the jet, known as reference velocity, is equal to a given reference value.
- In cooling mode the supply air temperature cannot be much cooler than the room air temperature; the temperature difference has a significant effect on jet detachment
- Suitable in spaces, where depth is less than 6 m.

3.4.3 Displacement ventilation

1. *Introduction*

Displacement ventilation first appeared in Northern Europe has been used in industrial applications with high heat load for many years[6].

Compared with traditional mixing ventilation, the displacement ventilation system is popular with better air quality, ventilation efficiency, and thermal comfort under the combination of pollutant and heat source[7]. For large space buildings, such as concert halls and workshops, displacement ventilation system is much more applied, while for buildings with lower floor heights, displacement ventilation system is not the most suitable choice.

2. *Air distribution of displacement ventilation*

The dilution method of displacement ventilation is different from that of the traditional ventilation pattern. It is based on the principle of hot air rising and cold air dropping caused by air density difference, as shown in Fig. 3.10. The cold air with higher density is directly supplied into the occupied zone and sinks to floor forming an air reservoir. The indoor thermal pollutant source generates plume and constantly entrains the surrounding air, making the pollutant air flowing to the outlet upward under the combination of the air supply exhaust systems. In displacement ventilation system the lower flow rate cannot cause draught discomfort. Meanwhile, the clean air is directly supplied to the occupied zone making the body in a relatively clean environment, improving the air quality of the occupied zone as well.

3. *Design of displacement ventilation*

Skistad[8] had developed and introduced a design method of displacement ventilation systems, and it consists of five steps next.

Step 1: Determine the required airflow rate for removal of waste heat based on the cooling load and the air temperature differences between supply and exhaust openings.

Step 2: Find the required airflow rate for removal of pollutants according to ventilation standards.

Step 3: Choose the larger of the two flow rates determined at Steps 1 and 2 as the ventilation rate.

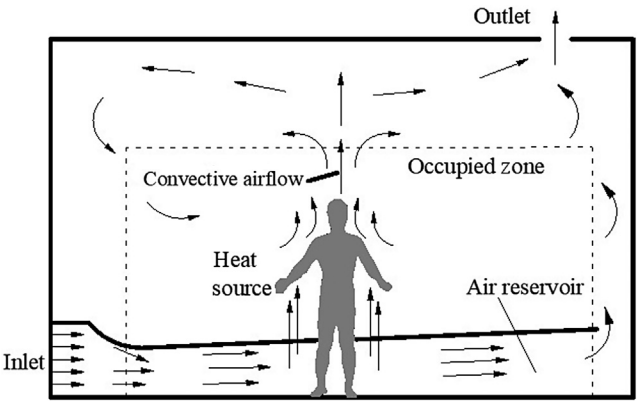


FIGURE 3.10 Principle of displacement ventilation.

Step 4: Determine supply air temperature.

Step 5: Choose supply diffusers according to the data provided by manufactures in order to avoid drafts.

4. *Performance evaluation of displacement ventilation system*

The effect of ventilation is directly related to the indoor air quality, which makes it necessary to assess the performance of ventilation system.

- Ventilation efficiency

Ventilation efficiency is an indicator of the ability of supply air to remove the pollutants, and it is defined in the following equation as:

$$\eta = \frac{C_r - C_0}{C_{oc} - C_0} \tag{3.9}$$

where η is ventilation efficiency; C_r is contaminant concentration at the air outlet, g/m^3; and C_{oc} is contaminant concentration at the occupied zone, g/m^3.

For displacement ventilation the pollutant concentration in the occupied zone is lower than that at the exhaust vent due to the thermal stratification that makes the ventilation efficiency greater than 1. However, for mixing ventilation, the maximum ventilation efficiency equals to 1.

3.4.4 Attachment ventilation

1. *Introduction*

Li[9,10] firstly proposed the design principle of attachment ventilation, and the concept of wall attached air supply can be traced back to last century[11,12]. He has developed a series of design methods including vertical wall-based attachment ventilation, pillar/column-based, deflector-based attachment ventilation used for "adjustable occupied zone"[13]. Attachment ventilation is a ventilation method based on Coanda Effect[14] and Extended Coanda Effect[15]. Usually, the air diffuser (slot) is set at the upper space of the room and is on or very close to vertical sidewall. A well designed attachment ventilation can create good room air distribution with high energy efficiency and saving occupied zone, which can improve the indoor air quality and achieve required indoor environment.

2. *Principle of attachment ventilation*

The principle of the attachment ventilation is shown in Fig. 3.11. When an isothermal airflow near to a vertical solid surface is a jet, the jet is deflected and attached to the surface (the original Coanda effect, region, see Fig. 3.11A). Based on the effect of inertia momentum, it moves along the original direction, reaches a separation point, and causes a

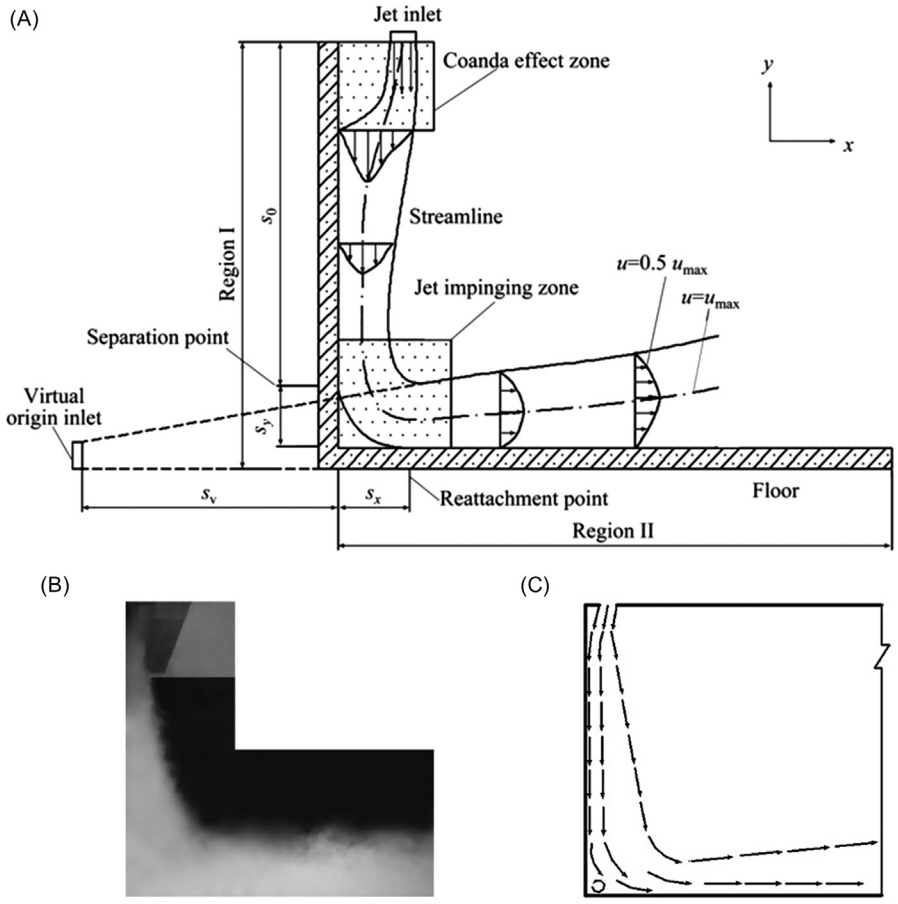


FIGURE 3.11 Principle of attachment ventilation: (A) an airflow structure of attachment ventilation by Extended Coanda Effect, (B) visualization of attachment ventilation and (C) airflow pattern of attachment ventilation.

stagnation phenomenon after collision. The pressure of the stagnation zone, between the separation point and the reattachment point, is close to the ambient pressure. In downstream region of the stagnation point, the dynamic pressure increases and reaches a maximum value. With the recovered dynamic pressure, fluid overcomes the flow resistance and moves along a horizontal surface (region), as illustrated in Fig. 3.11A. This is the fundamental principle for the attachment ventilation, which is called Extended Coanda Effect[15]. The similar phenomenon occurs for an air jet flowing along a horizontal surface, through collision, to a vertical surface.

3. *Boundary of the occupied/control zone*

The boundary of the control zone is defined by the European Heating, Ventilation and Air Conditioning Association (REHVA), as shown in Table 3.2. The boundary of the control zone for attachment ventilation is defined as follows:

1.0 m from the wall or pillar/column where the air inlet is located;
1.0 m from the exterior wall, door, and window;
0.5 m from the interior wall; and
0.1−2.0 m above the floor level.

Fig. 3.12 shows the specific occupied/control zone of attachment ventilation with vertical walls and pillars.

4. *Airflow parameters and layout of slot inlet in the control zone*

According to the provisions of various standards, such as GB/T 50155-2015[16], BS EN ISO 7730-2005[17], and ANSI/ASHRAE Standard 55-2017[18], on the design of air distribution parameters, the following control parameters for attachment ventilation[13] are proposed:

- Air temperature difference of the occupied zone: for sedentary posture, $t_{0.1}-t_{1.7} \leq 3.0^{\circ}C$; for standing posture, $t_{0.1}-t_{1.1} \leq 2.0^{\circ}C$.
- Minimum air temperature at 0.1 m above the floor in the occupied zone: in winter, $t_{0.1\ min} \geq 19^{\circ}C$; in summer, $t_{0.1\ min} \geq 21^{\circ}C$.
- Air velocity in occupied zone: for office and residential buildings, in winter, $u_n \leq 0.2$ m/s, and in summer, $u_n \leq 0.3$ m/s; for temporary stay places such as metro stations, subway stations, and airport waiting halls, $u_n \leq 0.3-0.8$ m/s; for industrial buildings such as hydropower stations, $u_n \leq 0.2-0.8$ m/s, or determined according to the requirements of production processes.

TABLE 3.2 Boundary of occupied/control zone for various air distributions.

	Distance between boundary of control zone and adjacent wall or pillar (m)		
Envelope or equipment	**Displacement ventilation**	**Attachment ventilation**	**Mixing ventilation**
Wall or pillar the air outlet located	0.5–1.5	1.0	1.0
Exterior wall, door, window	0.5–1.5	1.0	1.0
Interior wall, pillar without air outlet	0.25–0.75	0.5	0.5
Floor	0.0–0.2	0.1	0.0
Distance from floor to ceiling	1.1*–2.0**	2.0	1.8

Note: Value with "*" is for sedentary posture, and value with "**" is for standing posture.

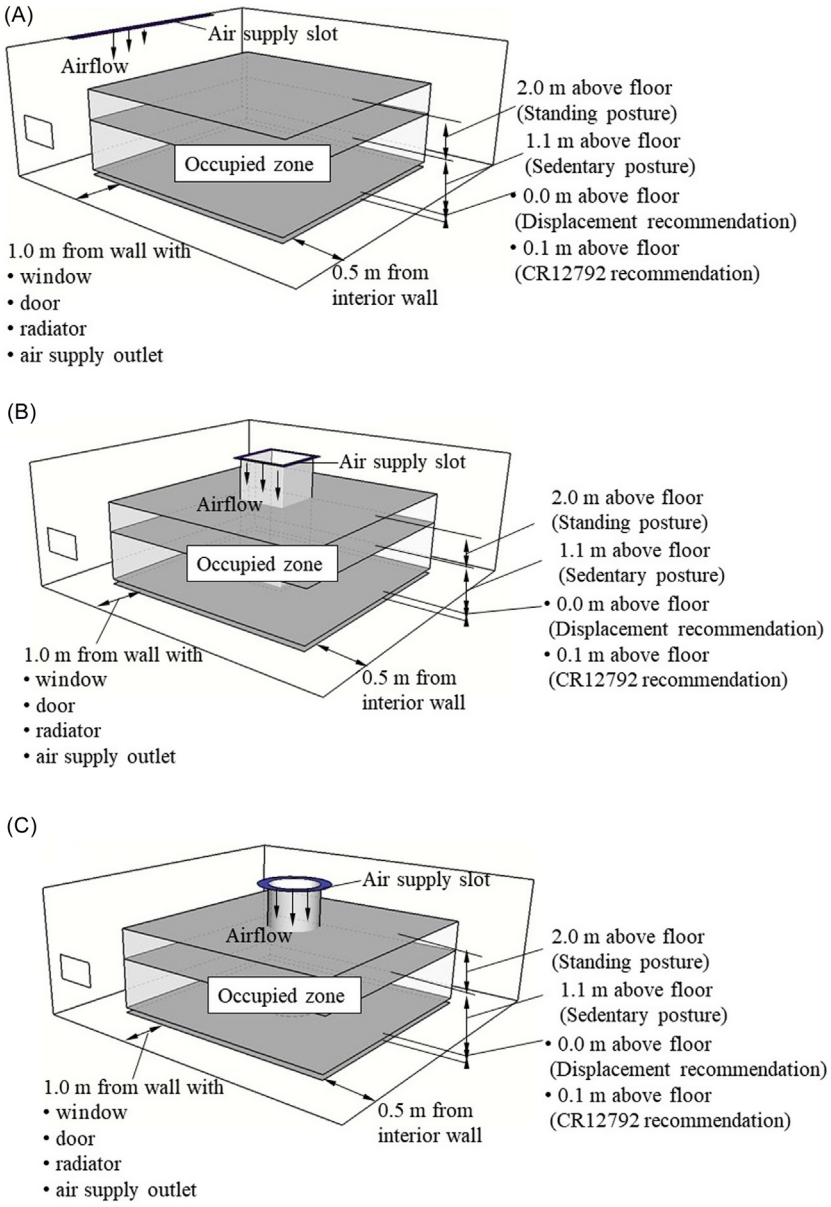


FIGURE 3.12 Definition of occupied/control zone of attachment ventilation: (A) wall attachment, (B) square or rectangular pillar attachment, and (C) circular pillar attachment.

- Boundary air velocity $u_{m,1.0}$ in control zone: for general office and residential buildings, $u_{m,1.0} \leq 0.5$ m/s; for temporary stay places, $u_{m,1.0} \leq 1.0$ m/s; for industrial buildings, it should be determined according to the specific production processes.
- The airflow of the exhaust and return outlet is similar to the confluence of the spherical space.

In addition, the following principles shall be followed for the layout of slot inlet of attachment ventilation.

- The slot inlet should not be set on the exterior wall or the exterior window.
- There should be no large number of obstructions on or near the impinging zone of the attached air distribution.
- When the air supply slots are arranged, the indoor personnel shall be outside the zone adjacent to the diffusion surface (1.0 m from the boundary of the control zone).
- The air exhaust outlet shall be set at the top or the highest place of the room as far as possible.

5. *Design of attachment ventilation*

A good design of attachment ventilation should meet the required distribution of air velocity and temperature for the occupied zone. The attachment ventilation in China has been used in subway stations, high-speed railway stations, hydropower stations, exhibition halls, and industrial applications with large spaces for many years[19]. The relevant design parameters are shown in Fig. 3.13. Based on the researches[20–23] on the design method of attachment ventilation, taking the summer conditions as an example, the engineering design steps of attachment ventilation presented by Li[13] are as follows:

Step 1: Determine basic indoor control parameters and air inlet size.

- According to the requirements of design, determine the target temperature, namely, the indoor control temperature at the height of 1.1 m from the floor, $t_{d,1.1}$.
- Define the vertical temperature gradient Δt_g of the occupied zone, and the value of Δt_g in attachment ventilation is generally 1.0–1.5°C/m.
- Define the size of the room, the pillars, the installation height h for air inlet and h_e for air outlet.

Step 2: Calculate indoor heat or cooling load Q_n.

For attachment ventilation design, in fact, the heat load Q_n in the room is the actual load of the occupied zone, which is calculated by the following equation:

$$Q_n = Q \times m \tag{3.10}$$

where Q is the total indoor heat load, m is the heat distribution factor and is defined by $m = (t_n - t_0)/(t_e - t_0)$, in which t_n is the temperature of room, t_e is the temperature of exhaust air, and t_0 is the temperature of supply air. The heat distribution factor m can be calculated by the thermal stratification height. Generally speaking, for large space buildings, it can be 0.50–0.85. However, in absence of adequate data, it can be assumed to be 0.70[3].

Step 3: Determine the temperature of exhaust air t_e.

According to the vertical temperature gradient Δt_g and the installation height of exhaust outlet

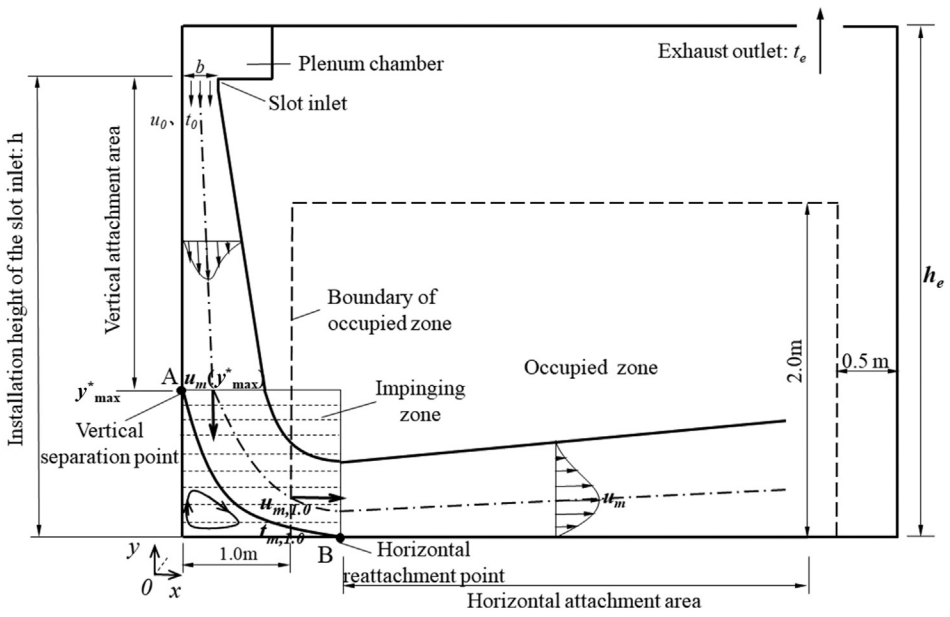


FIGURE 3.13 Design parameters for attachment ventilation.

h_e, the exhaust air temperature t_e can be calculated in the following equation:

$$t_e = t_{d,1.1} + \Delta t_g(h_e - 1.1) \tag{3.11}$$

Step 4: Determine the temperature of supply air t_0.

The dimensionless temperature rise κ near the ground (within 0.1 m above floor) is defined in the following equation:

$$\kappa = \frac{t_{0.1} - t_0}{t_e - t_0} \tag{3.12}$$

where $t_{0.1}$ is the air temperature in the height of 0.1 m above the floor. For vertical wall-based attachment ventilation, the value of κ is 0.55, then t_0 can be calculated by the following equation:

$$t_0 = t_{d,1.1} - \frac{1 + \kappa(h_e - 1.1)}{1 - \kappa}\Delta t_g \tag{3.13}$$

when $\kappa = 0.55$, $t_0 = t_{d,1.1} - (0.88 + 1.22h_e)\Delta t_g$.

Step 5: Calculate the supply air velocity u_0.

The width of the slot inlet is defined as b and its length is defined as l. According to the types of wall or pillar of the attachment ventilation, and their actual size, the area F of air inlet is preliminarily determined. Then the supply air velocity u_0 can be calculated according to the energy balance, and we have the following equation:

$$u_0 = \frac{Q_n}{\rho \cdot c_p(t_n - t_0) \cdot F} \tag{3.14}$$

Step 6: Check the air velocity at the controlled point of 1.0 m apart from the vertical wall in the horizontal zone.

The vertical distance between the separation point and air inlet is expressed as y^*_{max}, and the dimensionless centerline velocity is calculated as follows:

$$\frac{u_m(y^*_{max})}{u_0} = \frac{1}{0.012(y^*_{max}/b)^{1.11} + 0.90} \tag{3.15}$$

where y^*_{max} can be calculated by empirical correlation (3.16).

$$y^*_{max} = 0.92h - 0.43 \tag{3.16}$$

where h means the installment height of the air supply slot.

The research shows that there is a relationship between the dimensionless velocity $u_{m,1.0}/u_0$ and $(u_m(y^*_{max}))/u_0$, which is shown in Eq. (3.17). Therefore the controlled air velocity of the point that is 1.0 m apart from the attached wall in horizontal zone,can be calculated according to the following equation:

$$\frac{u_m(y^*_{max})}{u_0} = k_v\frac{u_{m,1.0}}{u_0} + C_v \tag{3.17}$$

The value of k_v and C_v depends on the type of attached wall surface. For vertical wall, $k_v = 1.808$, $C_v = -0.106$. And for pillars (both circular and square), $k_v = 1.374$, $C_v = -0.060$.

If the value of $u_{m,1.0}$ does not meet the required value mentioned earlier, namely, $u_{m,1.0} \le 0.5$ m/s (for office and residential building), and $u_{m,1.0} \le 1.0$ m/s (for the places of temporary stay), go back to Step 5 and reassume the value of b and l of the slot to recalculate.

Step 7: Check the centerline air velocity of the x point at the terminal zone of the air reservoir $u_{m,x}$. $u_{m,x}$ can be defined by the following correlation:

$$\frac{u_{m,x}}{u_0} = \frac{0.575}{C\left((x/b) + K_h\right)^{1.11} + 1} \tag{3.18}$$

where C is the shape factor, for vertical wall, $C = 0.0075$, for square pillar, $C = 0.0180$, and for circular pillar, $C = 0.0350$; K_h is the height correction factor, for vertical wall and square pillar, $K_h = (1/2)((h - 2.5)/b)$, and for circular pillar, $K_h = (1/6)((h - 2.5)/b)$.

If $u_{m,x}$ meet the requirement of the air velocity of the controlled zone, namely, $u_{m,x} \le 0.3$ m/s (for temporary stay places, $u_n \le 0.3$–0.8 m/s), then check whether the wall size of the room can meet the requirements of the total length l of the slot inlet. If the requirements can be met, the calculation process ends, otherwise, go back to Step 5 and reassume the value of b and l of the slot to recalculate.

3.5 Local ventilation

3.5.1 Introduction

Local ventilation is a kind of ventilation method, in which pollutants are collected at the source and handled centralized. Compared with other ventilation methods, local ventilation system requires minimum ventilation airflow rate and has better control effect.

3.5.2 Design principle of local exhaust system

1. For the dispersed polluted sources such as dust, harmful gases, waste heat, and moisture, the local exhaust system should be set up according to the technological conditions.

2. If there is a suddenly release of a large amount of harmful or explosive gas, an accident exhaust device should be installed.
3. When the local exhaust system is set, it should not disturb the normal operation.
4. Independent exhaust system should be set up when mixing may cause combustion, explosion, steam condensation and dust accumulation or form more toxic hazardous substances.
5. Avoid or weaken as much as possible the influence of disturbing airflow, such as cross-hall air, and supply air, on suction airflow of the exhaust system.

3.5.3 Composition of local exhaust system

The local exhaust system is mainly composed by the following five parts[24], as shown in Fig. 3.14.

1. *Hood*
   The hood is installed at the source of pollutants, and it can effectively capture the pollutants emitted from the productive process. The capture efficiency of the hood has an important effect on the economic performance of the local exhaust system.
2. *Duct system*
   Duct system is used to transport polluted air to air-cleaning device or exhaust stack. The economic performance of the entire exhaust system can be improved by reasonably determining the structure, size, and layout of the air duct.
3. *Air-cleaning device*
   When the collected waste gases cannot be emitted directly, air-cleaning device is needed for treatment. Different types of pollutants need to be treated with corresponding cleaning equipment. Recovery of some economically valuable pollutants can be considered.
4. *Air-moving device*
   Air-moving device provides the power for polluted air to overcome system resistance. In order to avoid the fan being worn and corroded by pollutants, the air-moving device is usually installed behind the purification equipment.
5. *Exhaust stack*
   The exhaust stack is to discharge the waste gases collected by the local exhaust system into the air pipe that meets the discharge standard. Special design of the exhaust pipe should be carried out according to the type of pollutants and the surrounding environment.

3.6 Industrial ventilation duct design

3.6.1 Duct losses

Pressure loss in duct is an irreversible loss caused by the conversion of mechanical energy into thermal energy, which includes two forms: friction pressure loss and local pressure loss.

Frictional pressure loss is caused by fluid viscosity, which is caused by the momentum change between molecules (laminar flow) or the momentum change of individual particles in adjacent fluid layers with different flow velocities.

Local pressure loss is pressure loss caused by fluid flow direction and area change when fluid flows through local components such as valve, bend pipe, and flow section change.

The detailed calculation method of duct system losses is shown in Chapter 21 of *2017 ASHRAE*[25].

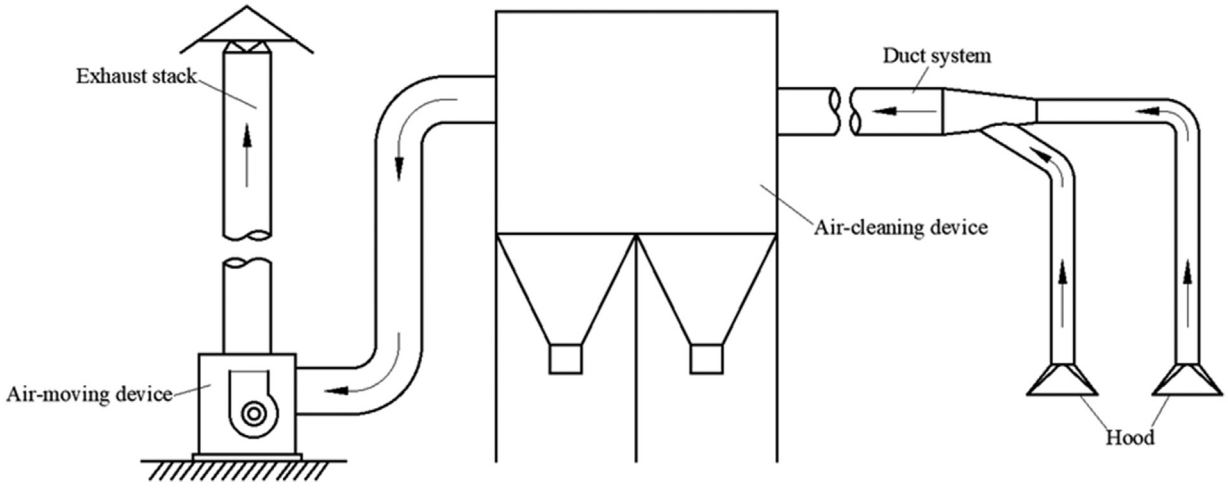


FIGURE 3.14 Schematic of local exhaust system.

3.6.2 Low resistance components

Lots of researches have been carried on drag reduction of duct by scholars. For instance, the local drag reduction effects of wedge-shaped components in elbow and T-junction close-coupled pipes[26] as shown in Fig. 3.15 were investigated. It reveals that the wedge-shaped drag reduction component with suitable height can reduce the resistance in elbow and T-junction close-coupled pipes. And that, in general, the largest height of wedge-shaped drag reduction components should not exceed 1/4 pipe inner diameter in HVAC field[27]. The performance of a novel low-resistance tee of ventilation and air-conditioning duct based on energy dissipation control[28] shown in Fig. 3.16 was studied. It demonstrates that the resistance of the novel tee can constantly be reduced by 42% under different flow ratios (5:1−1:3) and aspect ratios (4:1−1:4). The use of the novel tee reduces energy dissipation intensity, and the energy dissipation area is pushed away from the main flow area. A novel low-resistance tee with protrusion based on biomimicry of the branches of plants[29] is shown in Fig. 3.17. The resistance of the novel tee was compared with that of the five traditional types of tees, which reveals that the resistance reduction rates of the duct tee with protrusion structure in two flow directions are 36% and 21%, respectively. Another study[30] about the characteristics of a low-resistance tee based on an arc guide vane was presented under different flow velocities and aspect ratios of air duct in an air-conditioning system. The schematic of the tee is shown in Fig. 3.18. It reveals that the resistance reduction of the duct with proposed guide vane is more than 5% under different flow ratios (5:1−1:3) and different aspect ratios

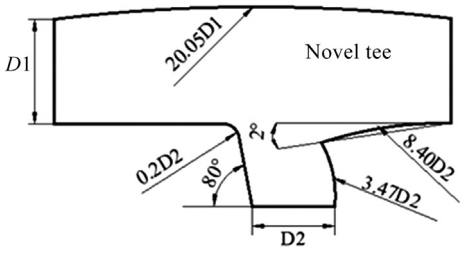


FIGURE 3.16 Schematic of a novel tee with cambered surfaces.

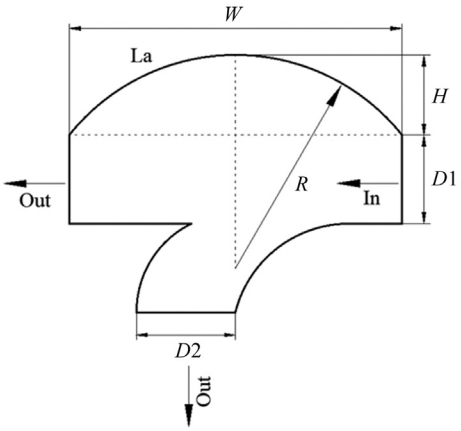


FIGURE 3.17 Structural schematic of a novel tee with protrusion: *La*—the arc length, *R*—the radius of the arc.

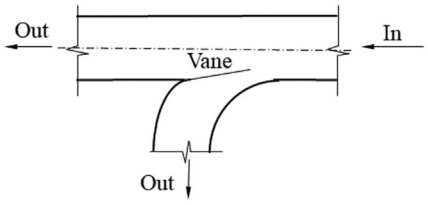


FIGURE 3.18 Structural schematic of the component.

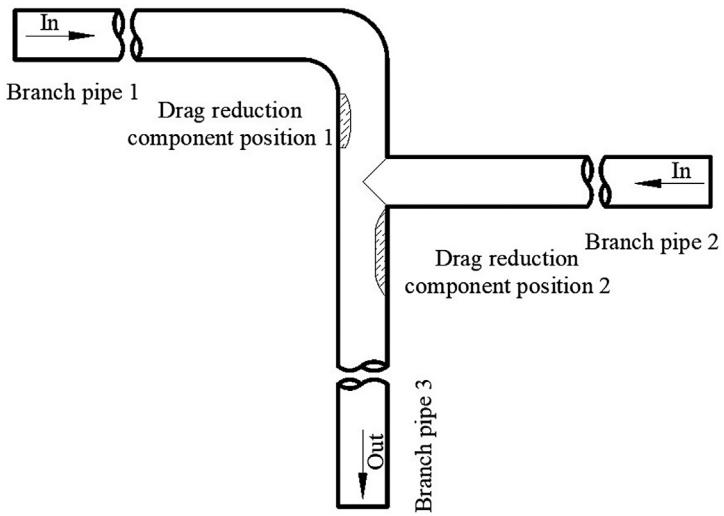


FIGURE 3.15 Schematic of drag reduction of wedge-shaped component in elbow and T-junction close-coupled pipes.

(4:1−1:4). For a single bend the following methods can be used to reduce its resistance:

(1) Install vane inside. For bends with plane side length larger than 500 mm, the standard JGJ141-2004-Technical specification for ventilation pipe shows that when the ratio of the interior arc radius to the plane side length of the bend is less than or equal to 0.25, the vane shall be set, and the radian of the vane shall be equal to that of the bend. And for bends with plane side length less than 500 mm, the vane should be installed at the position of 1/2 side length[31], as shown in Fig. 3.19.

(2) Modify the interior and exterior arc of the elbow. This can obviously reduce the resistance.

(3) Extended elbow is shown in Fig. 3.20B. Fig. 3.20A is the existing standard elbow, and Fig. 3.20B is the novel extended elbow. This kind of elbow has obvious directionality and better resistance reduction effect. According to the investigation conducted by Li's team[31], the resistance reduction rate of the optimized elbow structure can approach to 15%.

(4) U-shaped and S-shaped coupling elbow[32] is shown in Fig. 3.21. More details can be found in Ref. 32.

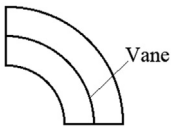


FIGURE 3.19 Bend with vane at the position of half the side length.

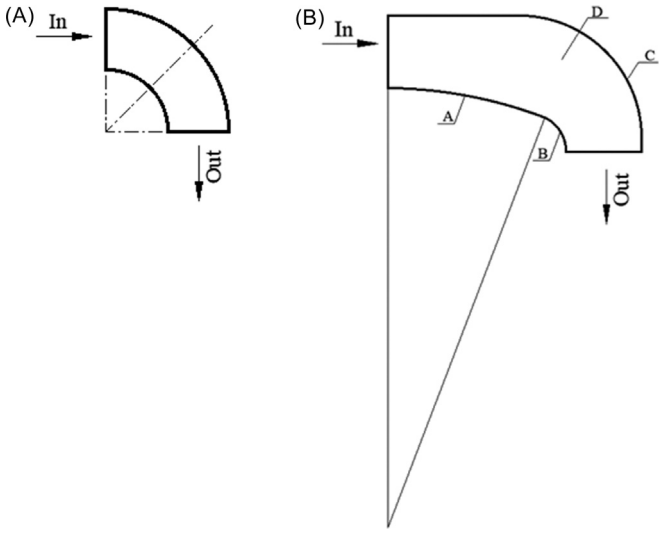


FIGURE 3.20 (A) The existing standard bend and (B) The novel extended bend.

3.6.3 Considerations about duct design

In addition, air duct or pipe system design should be considered. Designers should pay attention to:

- space availability,
- space air diffusion,
- noise levels,
- air distribution system (duct and equipment),
- duct heat gains and losses,
- balancing,
- fire and smoke control,
- initial investment cost,
- system operating cost, and
- air leakage.

3.6.4 Calculation of duct design

The purpose of the design calculation is to determine the pipe diameter (or section size) and pressure loss of each section, to ensure the required airflow rate distribution in the system and to provide basis for the selection and construction drawings of the fan system.

3.6.5 Duct design methods

Usually there are two design methods for sizing duct systems, one is the equal friction method and another is the static regain method. Taking the equal friction method as an example, when sizing the duct systems, we can use the target velocity to determine the size of the first duct section both downstream and upstream of the fan. From the size determined by the target velocity, the design friction rate is obtained to size all remaining duct sections. The whole duct systems can be finally calculated step by step. For more details, refer to Chapter 21 of the 2017 *ASHRAE Handbook-Fundamentals*[25].

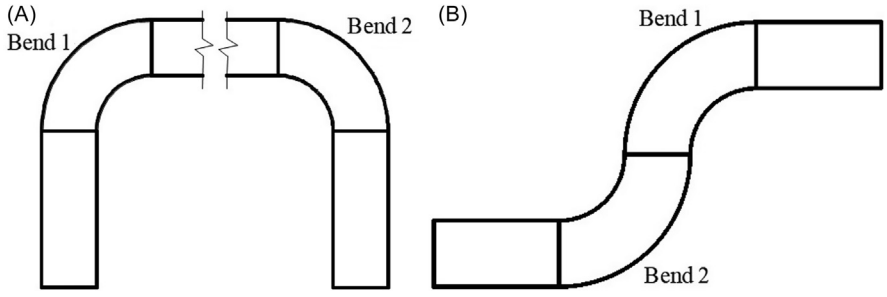


FIGURE 3.21 Coupling bend: (A) U-shaped coupling bend and (B) S-shaped coupling bend.

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CHAPTER

4

Physical fundamentals

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4.1 Fluid flow

It is essential that the engineers involved in industrial ventilation have a good foundation in the subject of fluid mechanics, which involves the study of fluids at rest or in motion.

The fields of application are wide involving computational fluid dynamics, flow in ducts and pipes, pumps, fans, collection devices, pollution dispersal, and many others.

4.1.1 Fluid properties

4.1.1.1 Fluid classification

Matter is considered to exist in three states:

- solid
- liquid
- gaseous

The term *fluid* applies to both liquids and gases, including liquid- and gas-containing particulate matters of various sizes.

When a shearing stress is imposed on a solid, deformation occurs, until a point is reached when the internal stresses produced balance the shearing stresses. Provided the elastic limit for the material is not exceeded, the solid will return to its original shape when the load is removed.

A fluid, on the other hand, flows under the action of a shearing stress, no matter how small this stress is. A fluid at rest has no shearing stresses, and all forces are at right angles to the surrounding surfaces. Materials such as glass and solid bitumen are fluids and, if stressed for a period of time, will tend to flow.

Ideal fluid

A theoretical ideal fluid situation, "a perfect fluid" having a constant density and no viscosity, is often used in a theoretical analysis.

Real fluid

A real fluid will have a velocity gradient when flowing due to the viscosity of the fluid.

Incompressible fluid

An incompressible fluid is a fluid, the density of which remains constant during flow. Liquids are normally treated as being incompressible, as a gas can be when only slight pressure variation occurs.

Compressible fluid

A compressible fluid is a fluid in which significant density variations that occur during its flow have to be considered, as is usually the case with vapors and gases.

Flow classification

Flows may be subdivided into steady and unsteady, uniform and nonuniform, laminar and turbulent, and rotational and irrotational flows.

Steady flow

A flow is steady when the conditions at any point remain constant with respect to time.

Unsteady flow

An unsteady flow is one in which the conditions at any point vary with time; such a flow is also called a transient flow.

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Uniform flow

A flow is uniform when the velocity of flow is the same at any given instant at every point in the fluid. This state of affairs can exist only with an ideal fluid. However, steady flow (uniform flow) is assumed to take place in a duct with the velocity constant along a streamline.

Nonuniform flow

In a nonuniform flow the velocity varies from point to point along a streamline.

Laminar flow

Laminar flow occurs at low flow rates, in which all particles of a fluid move parallel to the walls of the duct.

Transitional flow

The flow region between laminar and turbulent flow is called transitional flow. It is three dimensional and varies with time.

Turbulent flow

Turbulent flow occurs at higher flow rates. The particles of the fluid have velocity components perpendicular to the general direction of flow.

Rotational flow

Rotational flow occurs in an element of a fluid that rotates about its axis, in addition to having translational motion (e.g., water passing through a paddle wheel).

Irrotational flow

Irrotational flow occurs when the fluid motion rotates about its axis (e.g., water flowing in a bend in a pipe).

Other definitions to consider are as follows:

Path line

A path line is the path traced by a single particle of fluid over a period of time.

Streamline

A streamline shows the direction of a number of particles of fluid at the same instant in time. Flow cannot take place across a streamline. Path lines and streamlines will be identical for steady flow.

Stream tube

A number of streamlines form a stream tube. Flows can enter and leave a stream tube only through the ends.

Stream surface

A stream surface is the surface of a stream tube.

Streak line

When a dye is injected into a fluid, the resulting streak lines provide flow visualization of fluid particles that have passed the same density of the fluid.

One-, two-, or three-dimensional flow

Flow may be steady but has a variation of velocity, pressure, etc., with position. If one optional coordinate is used to describe the flow, it is one dimensional, a typical case being uniform flow in a constant-area duct.

Two-dimensional flow is in the x and y directions, while three-dimensional flow is in the x, y, and z directions.

A fluid can be considered as being liquid, which is incompressible, or a gas, which is easily compressible. When a force of sufficient magnitude is applied to a fluid, a motion will occur provided the frictional resistance within an open system is overcome.

A gas expands in an enclosure to fill up the entire space, while a liquid presents a free surface in contact with the gas boundary above it.

Once a fluid starts to move in a conduit, shearing forces are set up, the maximum being at the wall of the conduit. At this surface the velocity is at the lowest, while in adjacent layers above this surface, the velocity increases as the shearing stresses decrease.

It is the dynamic viscosity μ of the gas/fluid that determines its ability for free flow. Very viscous fluids require a large energy input to overcome the frictional forces.

4.1.1.2 Properties of fluids

Density

Density is the mass per unit volume kg/m^3. The density of a fluid depends on temperature and atmospheric pressure or a static imposed head. At standard conditions 20°C and 101.325 kPa (atmospheric pressure at sea level)

$$\rho_{\text{water}} = 998.2 \text{ kg}/m^3$$
$$\rho_{\text{air}} = 1.2 \text{ kg}/m^3$$

From these differences, it will be seen that water is 832 times as heavy per unit volume as air.

Water at 100°C at atmospheric pressure has a density of 958 kg/m^3. For data at other temperatures and pressures for water and other fluids, full use has to be made of various reference tables.

The relationship that exists between liquid density and temperature is expressed by

$$\Delta\rho = \frac{\Delta\theta^2}{A}. \tag{4.1}$$

where

$$\Delta\rho = \rho_0 - \rho$$
$$\Delta\theta = \theta - \theta_0$$

$$\rho = \rho_0 - \frac{(\theta-\theta_0)^2}{A}, \tag{4.2}$$

where ρ is the density at the temperature θ; ρ_0 is the density at the temperature θ_0; and A is a constant, specific to the fluid.

The relation of liquid density to pressure is

$$\frac{\Delta\rho}{\rho_0} = \frac{\Delta p}{E}, \tag{4.3}$$

where E is the modulus of elasticity.

The density of an ideal gas is dependent on the pressure and temperature as

$$p = \rho RT, \tag{4.4}$$

where R is the gas constant of the gas in question, J/(kg K). It is calculated by dividing the general gas constant $R = 8314.3$ J/(kmol K) by the molecular weight of the gas. If the composition of the ideal gas is unknown, but its pressure, temperature, and density are known, the value of the gas constant can be calculated from

$$R = \frac{p_0}{\rho_0 T_0}. \tag{4.5}$$

The equation can also be expressed as

$$\frac{\rho}{\rho_0} = \frac{pT_0}{p_0 T}. \tag{4.6}$$

The state equation of an ideal gas such as steam is

$$pv = ah + b, \tag{4.7}$$

where v is the specific volume, $v = 1/\rho$, h is enthalpy, and a and b are constants.

This equation is seldom used, because the tables of the thermodynamic properties of fluids (steam tables) allow the values of the fluid/gas vapor to be accurately obtained.

Specific weight

Specific weight is the weight per unit volume and is equal to $\rho \times g$, where g is the acceleration due to gravity. In the case of water of density 1000 kg/m^3, the specific weight is 9.81×10^3 N/m^3.

Specific gravity

Sometimes called relative density, specific gravity is the ratio of the fluid density with respect to a reference substance at a specified temperature.

Mercury has a density of 13,600 kg/m^3 and is 13.6 times as heavy as water, or 11,333 times as heavy as the same volume of air.

Water is taken to have a specific gravity of 1.0 at 4°C, where it has its maximum density, with other liquids having a value either greater or less than this.

In the case of air the specific gravity is taken as 1.0, with all other gases having specific gravity greater or less than this value.

Plastic fluids

Various types of fluids, known as plastic fluids, may be encountered, which do not start to flow until a certain minimum shear stress is reached. The relationship between shear stress and the rate of shear strain may or may not take a linear form.

If linear, the plastic is known as a Bingham plastic, a typical case being sewage sludge.

Pseudoplastic fluids

With this type of fluid the viscosity decreases as the shear strain increases, typical cases being mud and liquid cement.

Dilatant fluids

Quicksand is included in this category. The viscosity increases as the rate of shear strain increases.

Surface tension

Surface tension is the property of a fluid that produces capillary action, the rise and fall in a tube.

Water in a tube wets the glass, and the liquid rises, producing a cup. In the case of mercury, the glass is not wetted and the liquid falls, producing an inverted cup.

Viscosity

Viscosity is the shear resistance between adjacent fluid layers. Consider in Fig. 4.1, the shearing action between two parallel planes, each of area A, separated by a distance Y. The tangential force F for a given area required to slide one plate over the other at a velocity (v) parallel to each other is

$$F = \mu \frac{v}{Y} A. \tag{4.8}$$

The proportionality factor μ is the dynamic viscosity of the fluid, its units being force $\times$ time/length^2 and is expressed as N s/m^2 or Pa s.

Examination of the thermodynamic properties of fluid tables shows how the viscosity varies with temperature. In order to obtain a general impression of this, consider the data in the thermal properties of fluid tables and the various values at different temperatures.

Another viscosity unit is the kinematic viscosity v. This is the ratio of viscosity to density. Common units

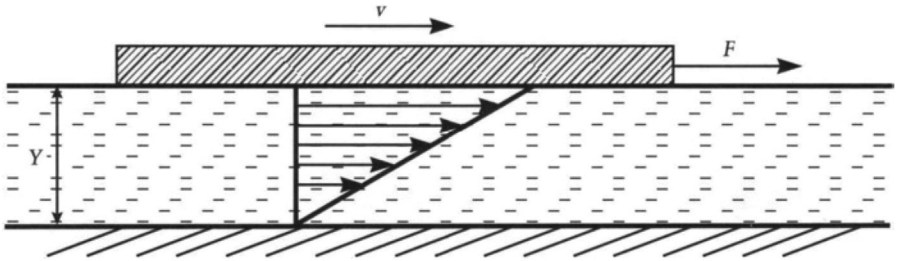


FIGURE 4.1 Viscosity.

used for this are the stoke (1 cm^2/s) and the centistoke (1 mm^2/s).

Because the velocity change in the y direction is linear, Eq. (4.8) can be written as

$$F = \mu \frac{dv}{dy} A. \tag{4.9}$$

When the shearing stress $\tau = F/A$,

$$\tau = \mu \frac{dv}{dy}. \tag{4.10}$$

With most fluids the shearing stress, τ is linearly proportional to the change of velocity; hence viscosity μ is not a function of dv/dy. A fluid having these characteristics is called a Newtonian fluid.

If the viscosity is a function of dv/dy, the fluid is classed as a non-Newtonian fluid. Fluids of this type are outside the scope of this chapter.

A special case of a Newtonian fluid is that of an ideal fluid, in which the viscosity $\mu = 0$. Ideal fluids do not exist; however, in many noncritical applications, the friction can be ignored to simplify calculations.

Thus viscosity is not a function of dv/dy, and it is independent of pressure. However, it is a function of the temperature.

The viscosity of noncompressible fluids depends on the temperature as

$$\frac{\mu}{\mu_0} = \exp\left(\frac{B}{C+T} + \frac{B}{C+T_0}\right), \tag{4.11}$$

where μ is the viscosity at any temperature T, μ_0 is the viscosity at any temperature T_0, and B and C are constants, depending on the nature of the fluid.

The viscosity of a gas depends on the temperature according to

$$\frac{\mu}{\mu_0} = \frac{S+T_0}{S+T}\left(\frac{T}{T_0}\right)^{2/3}, \tag{4.12}$$

where S is a constant specific for the gas. A simplified version sometimes used is

$$\frac{\mu}{\mu_0} = \left(\frac{T}{T_0}\right)^n. \tag{4.13}$$

The kinematic viscosity v is the ratio of the dynamic viscosity μ and density ρ

$$v = \frac{\mu}{\rho}. \tag{4.14}$$

The kinematic viscosity of a gas is a function of the pressure, and its dimension is the square of length divided by the time, its unit being m^2/s.

In old literature the cgs system is found, in which the dynamic viscosity is measured in centipoise = 0.1 poise = 0.001 dyn/cm^2

$$1\ \text{cP} = 10^{-2} g/(\text{cm}\ s) = 10^{-3} \text{kg}/(m\ s). \tag{4.15}$$

The non-SI unit of kinematic viscosity is the centistoke

$$1\ \text{cSt} = 10^{-2}\ \text{cm}^2/s = 10^{-6}\ \text{m}^2/s. \tag{4.16}$$

4.1.2 Constants for water

For water the values of the constants discussed in the previous section are given in Table 4.1. The value of the elasticity modulus increases as the pressure and temperature increase. At a pressure of 10 MPa and temperature 373K, the elasticity modulus E is 2.7×10^9 Pa.

The equations do not give exact results, but the error is small and in many cases can be ignored.

TABLE 4.1 Constants for water.

Pressure	$p_0 = 1$ kPa
Temperature	$T_0 = 273.15$K
Density	$\rho_0 = 999.80\ \text{kg/m}^3$
Constant A	$A = 225.30\ \text{m}^3\ \text{K}^2/\text{kg}$
Velocity of sound	$c_0 = 1400$ m/s
Viscosity	$\mu_0 = 1.791 \times 10^{-3}$ kg/(s m)
Kinematic viscosity	$v_0 = 1.791 \times 10^{-6}\ \text{m}^2/\text{s}$
Constant B	$B = 511.6$K
Constant C	$C = -149.4$K
Specific heat	$c_p = 4182.6$ J/(kg K)
Modulus of elasticity	$E_0 = 1.95 \times 10^9\ \text{N/m}^2$

4.1.3 Constants for gases

The constants for various gases are listed in Table 4.2, where ρ_0 is the density, c_0 is the velocity of sound, R is the gas constant, μ_0 is the dynamic viscosity, and κ is the adiabatic constant.

4.1.4 Properties of air and water vapor

Air can be considered as an ideal gas, which has a definition

$$pv = RT, \tag{4.17}$$

or

$$p = \rho RT. \tag{4.18}$$

This is the state equation of an ideal gas, where p is the pressure, v is the specific volume, ρ is the density, R is the gas constant, and T is absolute temperature. In an airflow, there is a transfer of heat from one layer to another. This change of state is adiabatic and reversible. Such an adiabatic reversible process is called an isentropic state change: one in which the entropy remains constant.

The thermodynamic equations to be considered at this stage are

$$T\,ds = dh - v\,dp, \tag{4.19}$$

where s is the entropy, kJ/(kg K) and h is the enthalpy, kJ/kg.

For isentropic process, we can write

$$dh = v\,dp. \tag{4.20}$$

The specific enthalpy change is defined as

$$dh = c_p dT. \tag{4.21}$$

The state equation gives

$$p\,dv + v\,dp = R\,dT. \tag{4.22}$$

When dT is eliminated from this equation, the following differential equation results

$$\frac{dv}{v}\left(\frac{1}{(R/c_p)-1}\right) = \frac{dp}{p}, \tag{4.23}$$

TABLE 4.2 Constants for gases.

	ρ_0 (kg/m^3)	c_0 (m/s)	R [J/(kg K)]	μ_0 [kg/(s m)]	τ
Air	1.275	332	287.04	1.717×10^{-5}	1.402
O_2	1.409	315	259.78	1.928×10^{-5}	1.399
N_2	1.234	37	296.75	1.625×10^{-5}	1.400
H_2	0.0887	1260	4124.0	8.350×10^{-5}	1.409
CO_2	1.949	259	188.88	1.370×10^{-5}	1.301

$p_0 = 100$ kPa, $T_0 = 273.15$K.

Solving,

$$pv^{c_p/(c_p-R)} = \text{constant}. \tag{4.24}$$

When c_p and R can be treated as constants, the equation is usually written as

$$pv^{\tau} = \text{constant}. \tag{4.25}$$

For a gas of one-atom molecules, $\kappa = 5/3 = 1.67$. For a gas of two-atom molecules, $\kappa = 7/5 = 1.4$. For gas of molecules containing three or more atoms, $\kappa = 9/7 = 1.3$.

For air (mostly a mixture of N_2 and O_2) the following is valid

$$pv^{1.4} = p_0 v_0^{1.4} = \text{constant}. \tag{4.26}$$

Water vapor is considered as an ideal gas and is defined by

$$pv = ah + b, \tag{4.27}$$

where a and b are constants. Converting,

$$p\,dv + v\,dp = a\,dh \tag{4.28}$$

and as

$$dh = v\,dp, \tag{4.29}$$

giving

$$pv^{1/(1-a)} = \text{constant}, \tag{4.30}$$

or

$$pv^{k} = \text{constant}, \tag{4.31}$$

where k is an empirically determined constant. For water vapor (H_2O), $k = 1.3$.

4.1.5 Liquid flow

In this section, incompressible liquid flow is dealt with, and the effect of compressibility is ignored.

4.1.5.1 *Energy equation*

The energy equation of a continuing system can be presented by means of the first law of thermodynamics and the energy balance of a flow system as

$$dh + g\,dz + d\left(\frac{v_m^2}{2}\right) = 0 \qquad dh = T\,ds + v\,dp = T\,ds + \frac{dp}{\rho}$$
$$\Rightarrow T\,ds + \frac{dp}{\rho} + g\,dz + d\frac{v_m^2}{2} = 0. \tag{4.32}$$

As the potential energy term has an essential meaning in hydromechanics, the static head is selected as a comparison quantity. When the energy Eq. (4.32) is

divided by g and integrated, it gives the Bernoulli flow tube equation

$$\frac{p_1}{\rho g} + z_1 + \frac{v_{m1}^2}{2g} = \frac{p_2}{\rho g} + z_2 + \frac{v_{m2}^2}{2g} + \int_1^2 \frac{T}{g} ds = H_1, \quad (4.33)$$

where H_1 is the hydraulic head, $v_m^2/2g$ is the velocity head, z is the elevation head, $p/\rho g$ is the pressure head, and $h_f = \int_1^2 (T/g) ds$ is the resistance head.

All flow losses are included in the term h_f, in which a flow system consists of two parts:

1. friction resistance and
2. local resistance.

4.1.5.2 *Viscous flow*

When liquid flows along a solid surface (see Fig. 4.2), a shearing stress is set up (friction power/surface), which is expressed by

$$\tau = f \frac{\rho v_m^2}{2} \text{ for a plate} \quad (4.34)$$

or

$$\tau = f \frac{\rho v_m^2}{2} = \xi \frac{\rho v_m^2}{8} \text{ for a tube,} \quad (4.35)$$

where v_m is mean velocity (velocity is zero at the surface), f is a dimensionless friction factor, $\xi = 4f$; this is the Blasius friction factor.

In some literature, λ or β is used instead of ξ; it is essential to use ξ, because λ is used for thermal conductivity and β is used for cubic expansion of air or an angle.

In a tube the pressure is constant in all radial directions perpendicular to the axis; it varies only in the flow direction x. The power balance of the element dx, denoting the sectional surface area as A and the periphery as C, gives

$$-rC\,dx - \frac{dp}{dx} dx\, A = 0 \text{ and } -\frac{dp}{dx} = \frac{\tau C}{A} = \frac{\Delta p}{l} = f \frac{\rho v_m^2}{2} \frac{C}{A}.$$

Denoting the hydraulic diameter as $d_h = 4\,A/C$, we have

$$\frac{\Delta p}{l} = \frac{4f}{d_h} \frac{\rho v_m^2}{2} = \xi \frac{\rho v_m^2}{2 d_h} \quad (4.36)$$

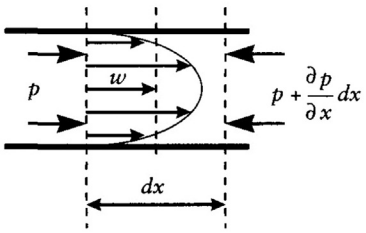


FIGURE 4.2 Tube flow velocity profile.

and the friction resistance head corresponding to the pressure difference is

$$h_f = \frac{\Delta p}{\rho g} = \xi \frac{l}{d_h} \frac{v_m^2}{2g}. \quad (4.37)$$

The factor ξ depends on the Reynolds number, which is a dimensionless variable that denotes the nature of flow

$$\mathrm{Re} = \frac{v_m d}{v} = \frac{\rho v_m d}{\eta} = \frac{4 q_m}{\pi d \eta}, \quad (4.38)$$

where v_m is the mean velocity; d is the characteristic length of a surface; in the case of flow in a tube, it is the tube diameter (note that d may be expressed by L in the case of a plate); v is the kinematic viscosity; η is the dynamic viscosity; q_m in the mass flow; and viscosity is defined by means of the equation.

$$\tau = \eta \frac{dv}{dy}, \quad (4.39)$$

where dv/dy is the velocity gradient, τ is the shearing stress between two flow layers, and η is the dynamic viscosity.

4.1.5.3 *Laminar and turbulent flow*

Flow phenomena can be divided into three main types:

- laminar (streamline)
- transitional
- turbulent

In laminar flow, there are no disturbances, and therefore all flow particles move in the same direction. Transitional flow is the flow regime that takes place during the change from streamline to turbulent flow. In the case of turbulent flow, the particles move in a given flow direction, but the flow is erratic and random.

Laminar tube flow

When the Reynolds number is under 2000, it is shown empirically that the flow in a smooth tube is laminar. This flow has a parabolic velocity profile, as shown in Fig. 4.3.

Now consider a cylindrical volume element in a flow stream. The radius of the element is r and its length is L. The force produced by the flow in this volume is due to the viscosity, which is

$$2\pi r L\, \tau = -2\pi r L \eta \frac{dv_m}{dr},\ r = -\eta \frac{dv_m}{dr}.$$

The pressure difference (drop) between the ends of the element produces a force $\Delta p \pi r^2$, and considering the force balance,

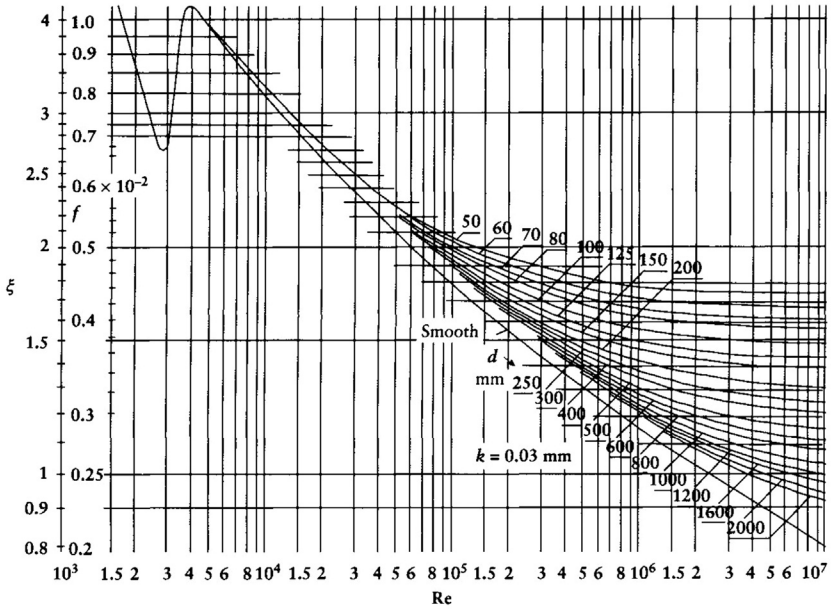


FIGURE 4.3 Cylindrical volume element.

$$\Delta p\,\pi r^2 = -2\,\pi r L\eta \frac{dv_m}{dr}.$$

Simplifying this gives

$$-\int_w^0 dv_m = \frac{\Delta p}{2\,\eta L}\int_0^R r dr.$$

Denoting w as v_{m1} at $r = 0$ and noting that $v_m = 0$ at $r = R$, the integration gives

$$w = v_{m1} = \frac{\Delta p}{2\eta L}R^2. \tag{4.40}$$

For a parabolic velocity profile the velocity expression is

$$v_m = \frac{\Delta p}{4\,\eta L}(R^2 - r^2). \tag{4.41}$$

Volume flow is calculated by integrating the expression for the velocity over the surface

$$q_v = \int_0^R 2\pi r v_m\,dr = \frac{\Delta p\pi}{2\,\eta L}\int_0^R (R^2 r - r^3)dr = \frac{\Delta p\,\pi}{2\,\eta L}\left(\frac{R^4}{2} - \frac{R^4}{4}\right)$$

Hence

$$q_v = \frac{\Delta p\,\pi R^4}{8\,\eta L}. \tag{4.42}$$

The mean velocity is

$$v_m = \frac{q_v}{\pi R^2} = \frac{\Delta p}{8\,\eta L}R^2. \tag{4.43}$$

The pressure difference is

$$\Delta p = \frac{8\,\eta\,v_m L}{R^2} = \frac{32\eta\,v_m L}{D^2}. \tag{4.44}$$

Eq. (4.44) is the Hagen–Poiseuille law that shows that the pressure loss during laminar flow is linearly proportional to the flow velocity.

The following equations show the relationship between the pressure loss and the friction factor

$$\Delta p = \xi\frac{L}{D}\frac{\rho\,v_m^2}{2} = \frac{32\,\eta v_m L}{D^2} \tag{4.45}$$

$$\xi = 64\frac{\eta}{\rho D v_m} = \frac{64}{\mathrm{Re}} \tag{4.46}$$

This connection is valid for laminar flow, as $\mathrm{Re} < 2000$.

Turbulent flow

Laminar flow after transition usually turns into turbulent flow when $\mathrm{Re} > 2000$. It has been shown that the pressure loss of a turbulent flow is caused by a friction factor with the magnitude of

$$f = 0.079\,\mathrm{Re}^{-1/4},\ \text{when}\ \mathrm{Re} < 10^5. \tag{4.47}$$

This equation is the Blasius equation.

The shearing stress τ_0 on the surface of the flow duct and the pressure loss can now be solved from Eq. (4.48), given next

$$\tau_0 = f\frac{1}{2}\rho v_m^2 = 0.0395\,\rho v^{1/4}D^{-1/4}v_m^{7/4}$$

$$\Delta p = 2f\frac{L}{D}\rho\,v_m^2 = 0.158\,L\rho\,v^{1/4}D^{-5/4}v_m^{7/4}. \tag{4.48}$$

Thus $\Delta p \sim v_m^{1.75}$.

When $\mathrm{Re} > 10^5$, the following equation, derived by means of the logarithmic velocity distribution by

Prandtl and the empirical research results of Nikuradse, is valid

$$\frac{1}{\sqrt{f}} = 4.01 \ln(\mathrm{Re}\sqrt{f}) - 0.4. \tag{4.49}$$

Surface roughness

In the previous section, it was assumed that the surface of the flow duct was smooth. In reality, duct surfaces are rough to varying degrees, which has an effect on the magnitude of friction. Thus Eqs. (4.47) and (4.49) represent the lowest possible levels of f; in other words, the effect of roughness is zero.

To allow for the effect of roughness, one can use the results of empirical tests in ducts that have been artificially roughened with particles glued on the surface. This approach allows roughness levels to be determined as a function of the particle diameter k. The following friction factor equation has been derived for large Reynolds numbers

$$\frac{1}{\sqrt{f}} = 4.0 \ln\frac{\mathrm{Re}}{k} + 3.48. \tag{4.50}$$

This is an ultimate case, when the friction factor is no longer a function of the Reynolds number and is a function of roughness; the pressure loss is now $\Delta p \sim w^2$, where w is the fluid velocity in the duct. The surface roughness of typical manufactured ductworks varies between the values of a theoretically fully smooth duct and an artificially roughened one. Accordingly, the pressure loss varies between $\Delta p \sim w^{1.75} - w^2$ and $\xi = f$ (Re, roughness).

With most forms of duct the roughness given by the following Colebrook and White equation can be used (Eq. 4.51). This equation has been determined by calculating an equivalent roughness, corresponding to the sand particle tests results, and taking into account that with large Reynolds numbers the friction factor's dependency on the Re value is minimal.

$$\frac{1}{\sqrt{f}} = -\ln\left[\frac{5.04}{Re\sqrt{f}} + \frac{k}{3.71\,d}\right] \tag{4.51}$$

This equation represents the change-over section between a smooth tube and a fully developed rough flow.

In practice, the friction factors are calculated either by integration of Eq. (4.51) or by reference to a Moody chart. This is based on Eq. (4.51) by using equivalent roughness values representing the sand particle roughness (see Table 4.3).

Fig. 4.4 shows the Moody chart for tubes when $k = 0.03$ mm, which is the case for steel tubes. Friction factors for other values of k can be attained by using the following ratio

$$\left(\frac{k}{d}\right)\mathrm{curve} = \left(\frac{k}{d}\right)\mathrm{case}$$

and determining the corresponding diameter from the Moody chart, which is derived from this equation.

TABLE 4.3 Equivalent roughness values for various materials.

Material	k_{equiv}
Commercial or follower steel	0.046
Asphalted cast iron	0.120
Galvanized steel	0.150
Cast iron	0.26
Wooden surface	0.18–9
Concrete	0.3–3

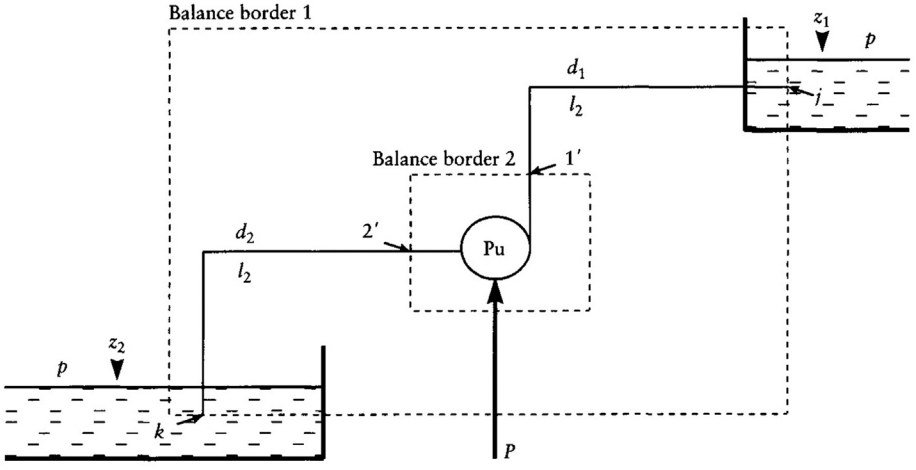


FIGURE 4.4 Moody chart, $k = 0.03$ mm.

4.1.5.4 Single resistances in a tube flow

In addition to the friction factor, individual resistances in ducts have to be taken into account. These resistances are created by the velocity, through bends, branches, valves, and other obstructions in the duct.

The single resistance factor ζ is defined as

$$\frac{\Delta p}{\rho g} = \zeta \frac{v_m^2}{2g} \quad \text{or} \quad \Delta p = \zeta \frac{\rho v_m^2}{2}. \tag{4.52}$$

The ζ values for different types of resistances are available in hydraulics textbooks and also from the literature of pipe and valve manufacturers.

The pressure loss and the corresponding resistance height in Eqs. (4.53) and (4.54), respectively, are the sum of the friction losses and individual resistances

$$\Delta p_{\text{loss}} = \left(\zeta \frac{1}{d_h} + \sum \zeta\right) \frac{\rho\, v_m^2}{2} \tag{4.53}$$

and

$$h_f = \frac{\Delta p_{\text{loss}}}{\rho g} = \left(\zeta \frac{1}{d_h} + \sum \zeta\right) \frac{v_m^2}{2g}. \tag{4.54}$$

Example 1

Find the power P required by the pump in the system shown in Fig. 4.5. The energy balance of a continuing system, when $q_m = q_v \rho$, gives q_m.

$$\frac{P}{q_v \rho g} = h_f + \frac{p_1 - p_2}{\rho g} + (z_1 - z_2) + \frac{v_{m1}^2 - v_{m2}^2}{2g}. \tag{4.55}$$

When $p_1 = p_2$ and $v_{m1} = v_{m2} = 0$, we obtain for the system defined by balance border 1

$$\frac{P}{q_v \rho g} = h_f + (z_1 - z_2), \tag{4.56}$$

where

$$h_f = \left(\xi_1 \frac{l_1}{d_1} + \sum\nolimits_1 \zeta\right) \frac{v_{m1}^2}{2g} + \left(\zeta_2 \frac{l_2}{d_2} + \sum\nolimits_2 \zeta\right) \frac{v_{m2}^2}{2g}.$$

Resistance factors ξ are taken from the Moody chart, when the Reynolds number and roughness are known.

$$\text{Re} = \frac{v_m d}{v}$$

and v_m is determined by the volume flow $q_v = A v_m$, where A is the cross-sectional area of the tube.

Single resistances ζ can be found from the literature. One of the single resistances is the outflow loss at point 1. The outflow loss is the kinetic energy representing v_{m1} and therefore $\zeta_{\text{outflow}} = 1$.

The form of Eq. (4.56) depends on the chosen balance border. The border can be chosen arbitrarily. When the balance space is chosen according to the balance border 2, the energy equation is

$$\frac{P}{q_v \rho g} = h_f' + \frac{P_1' - P_2'}{pg} + z_{1'} - z_{2'} + \frac{v_{m1'}^2 - v_{m2'}^2}{2g}. \tag{4.57}$$

Using the values corresponding to the states 1′ and 2′, h_f' is the resistance height between 1′ and 2′, and p_1 and p_2 are determined by the pressure loss and initial pressure. Other quantities are determined correspondingly. It is worth noting that there are no outflow losses in this case. Generally, it is wise to use energy balances in calculations.

The above is valid for a liquid flow, when the effect of compressibility can be ignored when calculating gas flows with small pressure differences. For instance, in ventilating ductwork, air is not compressed, so the density is considered as constant. In HVAC technology a unit of pressure frequently used for convenience is a millimeter of water column, 1 mm $H_2O \approx 10$ Pa.

4.1.5.5 Pressure loss in gas and steam pipes

When the gas compressibility no longer can be bypassed, the pressure loss equation is written in a differential form

$$dp = \left(\xi \frac{dl}{D} + \sum \zeta\right) \frac{\rho v_m^2}{2}, \tag{4.58}$$

where dl is the differential pipe length and D is the diameter.

On the basis that $\rho v_m A = q_m$ or $v_m = q_m / \rho A$,

$$dp = \left(\xi \frac{dl}{D} + \sum \zeta\right) \frac{q_{m^2}}{2\rho A^2} = \left(\xi \frac{dl}{D} + \sum \zeta\right) \frac{8 q_{m^2}}{\rho D^4 \pi^2}. \tag{4.59}$$

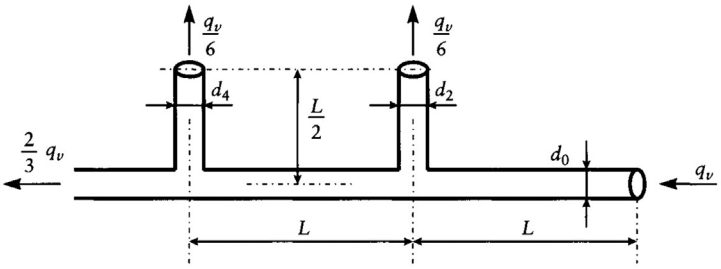


FIGURE 4.5 Pumping from one tank to another tank.

When both sides of the equation are multiplied by p, we have

$$p\,dp = \left(\xi dl + D\sum\zeta\right)\frac{8q_{m^2}}{D^5\pi^2}\cdot\frac{p}{\rho}. \tag{4.60}$$

For an ideal gas,

$$pV = NT \quad \text{or} \quad p\upsilon = p/\rho = T/M,$$

or when the gas follows the formula $p\upsilon = h^*/\zeta$; where ζ is the process factor and $h^* = h - h_0$, the deviation from the enthalpy of the reference state, Eq. (4.60), can easily be integrated, giving

$$\frac{p_1^2 - p_2^2}{2} = \frac{p}{\rho}\cdot\frac{8}{\pi^2}\cdot\frac{q_{m^2}}{D^5}\left(\xi L + D\sum\zeta\right), \tag{4.61}$$

where p/ρ is the constant.

If the pressures and densities are known, we can solve for either q_m or D from this equation.

If either p_1 or p_2 is known, simplification of the left side of Eq. (4.61) gives

$$\frac{p_1^2 - p_2^2}{2} = \frac{p_1 + p_2}{2}\cdot(p_1 - p_2) = p\Delta p = p_1\Delta p_1 = p_2\Delta p_2, \tag{4.62}$$

where Δp_1 and Δp_2 are auxiliary quantities, which can be solved from Eqs. (4.61) and (4.62). The real pressure loss can then be solved with the equations

$$\frac{\Delta p}{\Delta p_1} = \frac{2}{1 + \sqrt{1 - 2\Delta p_1/p_1}} \tag{4.63}$$

and

$$\frac{\Delta p}{\Delta p_1} = \frac{2}{1 + \sqrt{1 - 2\Delta p_2/p_2}}. \tag{4.64}$$

These equations can be used when the Mach number is small, and the acceleration effect is ignored.

Example 2

Calculate the final pressure of airflowing in a 200 m long pipe, when the initial pressure of air is 800 kPa and the mass flow is 2.5 kg/s. The pipe diameter is 100 mm, the roughness is 0.03 mm, and the sum of the single resistances is 6.5. The air temperature is 300K, the molar mass 28.96 kg/kmol, and the dynamic viscosity 1.85×10^{-5} kg/(s m).

First calculate

$$p\upsilon = \frac{TR}{M} = \frac{300\cdot8.314}{28.96} = 86.13\ \text{kJ/kg}.$$

Determination of friction factor ξ

$$\text{Re} = \frac{4\cdot q_m}{\pi\, d\eta} = \frac{4\cdot2.5}{\pi\cdot0.1\cdot1.85\cdot10^{-5}} = 1.72\cdot10^6$$

and

$k = 0.03\ \text{mm} \Rightarrow \xi = 0.0154.$

Eq. (4.61) gives

$$\frac{p_1^2 - p_2^2}{2} = p\upsilon\cdot\frac{8}{\pi^2}\cdot\frac{q_{m^2}}{D^5}\left(\xi L + D\sum\zeta\right)$$

$$= 86.13\times10^3\cdot0.81\cdot\frac{2.5^2}{0.1^5}(0.0154\cdot200 + 0.1\cdot6.5)\text{Pa}^2$$

$$= 1.626\times10^{11}\ \text{Pa}^2$$

Pressure loss from Eqs. (4.62) and (4.63)

$$\Delta p_1 = \frac{p_1^2 - p_2^2}{2\cdot p_1} = \frac{1.626\times10^{11}}{2\cdot800\times10^3} = 101.7\ \text{kPa}$$

$$\frac{\Delta p_1}{p_1} = \frac{101.7}{800} = 0.127$$

$$\frac{\Delta p_1}{p_1} = \frac{2}{1 + \sqrt{1 - 2\cdot\dfrac{\Delta p_1}{p_1}}} = \frac{2}{1 + \sqrt{1 - (2\cdot0.127)}} = 1.073$$

$$\Delta p = (1.073\cdot101.7)\ \text{kPa} = 109\ \text{kPa}$$

Final pressure is

$$p_2 = p_1 - \Delta p = (800 - 109) = 691\ \text{kPa}.$$

4.1.5.6 *Dimensioning of a duct with liquid flow*

This example demonstrates the dimensioning of a duct with a frictional incompressible fluid flow. Now the Bernoulli equation can be written as

$$p_1 + \rho g z_1 + \frac{1}{2}\rho w_1^2 = p_2 + \rho g z_2 + \frac{1}{2}\rho w_2^2 + \Delta p, \tag{4.65}$$

where $\Delta p = \left(\xi(L/D) + \sum\zeta\right)\rho(w^2/2)$ is the pressure loss term, $\xi(L/D)\rho(w^2/2)$ is the pressure loss due to the friction, and $\sum\zeta\rho(w^2/2)$ is the pressure loss due to the single resisitances.

To determine the pressure losses, we have to find out whether the flow is laminar or turbulent, because $\xi = f(\text{Re}, k/d)$. In practical dimensioning, Eq. (4.66) and the Moody chart are used.

$$\frac{1}{\sqrt{\xi}} = -2\ln\left[\frac{2.52}{\text{Re}\sqrt{\xi}} + \frac{k}{3.71d}\right]. \tag{4.66}$$

Empirical tables are used to determine the value of the single resistances.

The determination of the desired volume flow $\dot{V}$ and the pressure difference $p_2 - p_1$ leads to an iterative procedure in a turbulent case, due to terms ξ and $\Sigma\zeta$. In a laminar flow

$$\xi = \frac{64}{\text{Re}},$$

so no iteration is needed. This is due to the fact that the pipe diameter d is reduced from its value in the

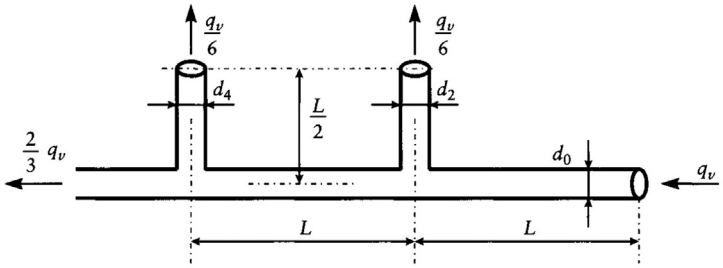


FIGURE 4.6 The system to be dimensioned.

formula for the pressure loss caused by friction, because $\mathrm{Re} = 4q_v/\pi\, dv$.

Example 3

A pipe of diameter d_0 conveys water at a volume flow rate q_v as shown in Fig. 4.6. The lateral branches 1–2 and 3–4 are each required to have volume flows of $q_v/6$. The pressures at points 0, 2, and 4 are known; the water leaves the system at points 2 and 4. Determine diameters d_4 and d_2.

The following values are known:

volume flow = 10 $\mathrm{m^3/s}$
diameter $d_0 = 79.8$ mm
pressure $p_0 = 600$ kPa
$p_2 = p_4 = 250$ kPa
temperature $T = 283$K
length $L = 50$ m
roughness $k = 0.03$ mm

Water properties can be found from tables, such as the VDI-Wasserdampftafeln, The NBS/NRC Steam Tables, the ASHRAE and CIBSE Guides, or *Thermodynamic and Transport Properties of Fluids* by Mayhew and Rogers.

First, we dimension d_4, and now Eq. (4.65) is

$$p_0 + \rho g z + \frac{1}{2}\rho w_0^2 = p_4 + \rho g z_4 + \frac{1}{2}\rho w_4^2 + \Delta p_4. \qquad (4.67)$$

Taking $z_0 = 0$, when $T = 283$K, $\rho = 999.7\ \mathrm{kg/m^3}$, $v = 1.3 \times 10^{-6}\ \mathrm{m^2 s}$, and $q_{v_0} = q_v = 10\ \mathrm{m^3/s}$. Thus the flow velocity $w_0 = 2$ m/s. From Eq. (4.67), we receive

$$106823 = 499.85\, w_4^2 + \Delta p_4. \qquad (4.68)$$

The formula for the pressure loss is now

$$\Delta p_4 = \left[\xi \frac{50}{79.8 \times 10^{-3}} + (-0.075)\right] \frac{999.7 \cdot 2^2}{2} + \xi_{13} \frac{50}{79.8 \times 10^{-3}} \times 999.7 \cdot \frac{1.666^2}{2} + \left[0.88 + \xi_{34} \frac{25}{d_4}\right] \frac{999.7\, w_4^2}{2} + \frac{999.7 w_4^2}{2} \qquad (4.69)$$

The Reynolds number is

$$\mathrm{Re}_{01} = \frac{4q_v}{\pi d_0 v} = 1.227 \times 10^5.$$

Hence the flow is turbulent.

When $k = 0.03$ mm, the ratio $k/d = 0.000\,375$, and therefore the friction factor $\xi_{01} = 0.0193$. Correspondingly, $\mathrm{Re}_{13} = 10 \times 10^5$ and $\xi_{13} = 0.0196$.

The formula for pressure loss is

$$\Delta p_4 = 41,080 + \left(989.7 + \xi_{34} \frac{12,496.3}{d_4}\right) w_4^2. \qquad (4.70)$$

The dimensioning of pipe 3–4 happens by means of Eqs. (4.68) and (4.70). The iteration equation is now

$$d_4 = \left(9.854 \times 10^{-9} + \frac{8.554 \times 10^{-7}}{d_4} \xi_{34}\right)^{1/4} \qquad (4.71)$$

Calculations are dealt with in the following stages:

1. d_4 is chosen.
2. Re and k/d_4 are calculated.
3. ξ_{34} is read from the Moody chart as a function of Re and k/d_4.
4. A new d_4 is calculated with Eq. (4.71) until the d_4 value is suitable.

Calculations are best carried out in a tabular form. The value of d_4 is 29.7 mm. In practice, the next larger standard pipe size would be selected.

The diameter d_2 is solved analogously. The Bernoulli equation at the interval 1–2 is

$$106\,823 = 499.85\, w_2^2 + \Delta p_2. \qquad (4.72)$$

The pressure loss is

$$\Delta p_2 = \frac{0.0193 \cdot 50 \cdot 999.7 \cdot 2^2}{79.8 \times 10^{-3} \cdot 2} + \left[0.88 + \xi_{12} \frac{25}{d_2} + 1\right] \frac{999.7\, w_2^2}{2} = 24\,178.2 + \left(939.7 + 12\,496.25 \frac{\xi_{12}}{d_2}\right) w_2^2 \qquad (4.73)$$

The iteration equation is

$$d_2 = \left(7.844 \times 10^{-8} + \frac{6.809 \times 10^{-7}}{d_2}\xi_{12}\right)^{1/4} \quad (4.74)$$

The diameter $d_2 = 28.8$ mm.

Example 4

A pump lifts water from a lake. At the pump suction entry a foot valve is fitted. Determine the maximum static delivery height the water can be raised without cavitation taking place. The saturation pressure of water is 1.23 kPa at 10°C, and the dynamic viscosity is 1.3×10^{-3} kg/(m s). The suction pipe water velocity is 2.0 m/s, the internal pipe diameter is 100 mm, and the pipe roughness is 0.03 mm. The resistance of the foot valve is 4.5.

Note: Cavitation occurs when the pressure at a point in a liquid flow field is equal or less than the vapor pressure of the liquid. At this point, bubbles of vapor are formed, this is cavitation. It has serious effects such as loss of duty, loss in efficiency, and serious erosion on the suction side of the pump impeller.

Pressure at the suction inlet p_t

$$p_t = p_i - \rho g h_s - \rho g h_f \geq 1.23 \text{ kPa}. \quad (4.75)$$

Resistance head

$$h_f = \left(\xi\frac{h_s}{d} + \sum\zeta\right)\frac{w^2}{2g}.$$

Friction factor

$$\text{Re} = \frac{wd\rho}{\eta} = \frac{2.0 \cdot 0.1 \cdot 1000}{1.3 \times 10^{-3}} = 153\,850$$

$$k = 0.03 \text{ mm},\ d = 100 \text{ mm} \Rightarrow \xi = 0.0185$$

By substituting the resistance head loss equation in Eq. (4.75), the suction head loss is determined from

$$p_i - \rho g h_s - \rho g\left(\xi\frac{h_s}{d} + \sum\zeta\right)\frac{w^2}{2g} = 1.23 \text{ kPa}$$

$$h_s = \frac{p_i - 1.23 - \rho g\sum\zeta(w^2/2g)}{\rho g(1 + \xi(1/d)\cdot(w^2/2g))}$$

$$= \frac{100 - 1.23 - 1000 \cdot 9.81 \cdot 4.5 \cdot (4.0/2 \cdot 9.81)}{1000 \cdot 9.81(1 + 0.0185 \cdot (1/0.1) \cdot (4.0/2 \cdot 9.81))} = 8.82 \text{ m}.$$

4.2 State values of humid air—Mollier diagrams and their applications

4.2.1 Properties of air and other gases

The analysis of dry atmospheric air varies with location, altitude, time of year, and other factors. Table 4.4 gives the molecular weights of the constituents and the volumetric and gravimetric analyses.

For general engineering work for altitudes up to 1500 m, it is sufficiently accurate to use the following:

By volume		By weight	
$O_2 = 21\%$,	$N_2 = 79\%$	$O_2 = 23\%$,	$N_2 = 77\%$

The abovementioned values are based on the assumption that argon is combined with nitrogen, adjusting the molecular weight to 28.16. Other gases present in the atmosphere air are normally ignored, as these represent less than 0.003% (by volume, 27.99 ppm). Table 4.5 provides some basic information on these trace gases.

The gases also have other constituents mixed with them, typical ones being: dusts, pollens, bacteria, viruses, mold spores, smoke particles, and the products of industrial activity such as SO_2, H_2, and S. Volcanic activity also adds various gases and dusts to the atmosphere.

TABLE 4.4 Analysis of air at sea level.

Constituent	Symbol	Molecular weight (M)	Volumetric analysis (%)	Gravimetric analysis (%)
Nitrogen	N_2	28.016	78.09	75.55
Oxygen	O_2	32.00	20.95	23.13
Argon	Ar	39.944	0.93	1.27
Carbon dioxide	CO_2	44.01	0.03	0.05
Total			100	100

Keey RB. Drying principles and practice. Oxford: Pergamon Press; 1972.[1]

TABLE 4.5 Minor constituents of dry air.

			Parts per million	
Gas	Symbol	Molecular weight	Volume	Weight
Neon	Ne	20.183	18	12.9
Helium	He	4.003	5.2	0.74
Methane	CH_4	16.04	2.2	1.3
Krypton	Kr	83.8	1	3
Nitrous oxide	N_2O	44.01	1	1.6
Hydrogen	H_2	2.016	0.5	0.03
Xenon	Xe	131.3	0.08	0.37
Ozone	O_3	48.00	0.01	0.02
Radon	Rn	222	0.06×10^{-12}	–

4.2.2 Fundamentals

Air is seldom dry; it normally contains varying amounts of moisture. Humid air is a mixture of dry air and water vapor. The term *dry air* denotes the mixture of all gases present in the air (nitrogen, oxygen, carbon monoxide, and inert gases), except water vapor.

The molar mass of dry air is dependent on the consistency of air, but for standard air, it is $M_i = 0.028964$ kg/mol. In practical calculations, we may use the rough value of 0.0290 kg/mol. The molar mass of water vapor is $M_h = 0.0180153$ kg/mol, and the rough value used in practical calculations is 0.0180 kg/mol.

According to the state equation of ideal gas, the partial density of dry air in humid air is

$$p_i = \frac{p_i M_i}{R\,T}, \tag{4.76}$$

where p_i is the partial pressure of dry air and R is the gas constant. According to present knowledge, the best value for the gas constant is

$R = (8.31441 \pm 0.00026)$

In practical calculations, we use the value of $R = 8.314$ J/(mol K).

The partial density of water vapor in humid air is

$$\rho_h = \frac{p_h M_h}{R\,T}, \tag{4.77}$$

where p_h is the partial pressure of water vapor.

The density of humid air is the sum of the partial densities of dry air and water vapor

$$\rho = \rho_i + \rho_h \tag{4.78}$$

and the total pressure of humid air is a sum of the partial pressures of dry air and water vapor

$$p = p_i + p_h \tag{4.79}$$

The mass of dry air in a volume V is denoted as m_i ($\rho_i = m_i/V_i$), and the mass of water vapor in V is m_h ($\rho_h = m_h/V_h$). Humidity of air, meaning the ratio of water vapor mass to dry air mass, is defined as

$$p_h = m_h/V_h \tag{4.80}$$

$$x \equiv \frac{m_h}{m_i}. \tag{4.81}$$

Using partial pressures, this definition can be written as

$$x \equiv \frac{\rho_h}{\rho_i}. \tag{4.82}$$

Humidity x is thus a dimensionless number, but sometimes a "dimension" is added to it, as a reminder of its definition. We can, for instance, write $x = 0.05$ or $x = 0.05$ kg H_2O/kg d.a., where d.a. stands for dry air.

According to Eqs. (4.76), (4.78), and (4.80), humidity can be written as

$$x = \frac{M_h}{M_i}\frac{p_h}{p_i} = 0.6220\frac{p_h}{p_i} = 0.6220\frac{p_h}{p - p_h}. \tag{4.83}$$

By solving the partial pressure of water vapor from the earlier equation, we receive

$$p_h = \frac{xp}{0.6220 + x}. \tag{4.84}$$

We denote again the mass of dry air in a volume V as m_i and the mass of water vapor as m_h. When humid air is treated as an ideal mixture of two components, dry air and water vapor, the enthalpy of this mixture is

$$H = m_i h_i + m_h h_h, \tag{4.85}$$

where h_i is the specific enthalpy of dry air (J/kg), and h_h is the specific enthalpy of water vapor.

Technical calculations dealing with humid air are reasonable to solve with dry air mass flow rates, because these remain constant in spite of changes in the amount of water vapor in the air. For that reason a definition for enthalpy,

$$h_k = \frac{H}{m_i}, \tag{4.86}$$

which is the enthalpy of humid air divided by the dry air mass, is made. The dimension of enthalpy h_k is J/kg, but it is often written as J/kg d.a. as a reminder that the total enthalpy of the mixture is calculated in terms of a kilogram of dry air.

Combining Eqs. (4.85) and (4.86), we get

$$m_i h_k = m_i h_i + m_h h_h,$$

and using Eq. (4.81), we have

$$h_k = h_i + x h_h. \tag{4.87}$$

In calculations with humid air, when the pressure is not high (usually the atmospheric pressure of 1 bar), water vapor and dry air can be handled as an ideal gas, as we have already done in Eqs. (4.76) and (4.78). For ideal gases the specific enthalpy is just a function of temperature

$$h_i = h_i(T)$$

and

$$h_h = h_h(T).$$

When 0°C dry air is chosen as the zero point of dry air enthalpy, and 0°C water as the zero point of water vapor enthalpy, the enthalpies of dry air

and water vapor can be calculated from the equations

$$h_i(T) = \int_{273.15\text{K}}^{T} c_{pi}(T)\, dT \tag{4.88}$$

$$h_h(T) = l_{h\,o} + \int_{273.15\text{K}}^{T} c_{ph}(T)\, dT \tag{4.89}$$

where $c_{pi}(T)$ is the specific heat of dry air [J/(kg K)], $c_{ph}(T)$ is the specific heat of water vapor, and l_{ho} is the heat of vaporization at 0°C. Its numerical value is

$$l_{h\,o} = 2501 \text{ kJ/kg}$$

Specific heats c_{pi} and c_{ph} are somewhat dependent on temperature. In the temperature range of −10°C to +40°C, their average values are

$$c_{pi} = 1.006 \text{ kJ/(kg °C)}$$

$$c_{ph} = 1.85 \text{ kJ/(kg °C)}.$$

At the temperature of +50°C, their values are $c_{pi} = 1.008$ kJ/(kg °C) and $c_{ph} = 1.87$ kJ/(kg °C).

Using numerical values mentioned earlier, the enthalpy of humid air h_k (Eq. 4.87) can be written as

$$h_k = 1.006\theta + x(2501 + 1.85\theta)\text{kJ/kg}, \tag{4.90a}$$

where θ is the temperature in Celsius. Eq. (4.90a) can also be written as

$$h = c_{pi}\theta + x\left(l_{ho} + c_{ph}\theta\right) \tag{4.90b}$$

We denoted the mass of dry air in a volume V as m_i, that is, $\rho_i = m_i/V_i$, and the mass of water vapor in V as m_h, that is, $\rho_h = m_h/V_h$. In practical calculations, we usually handle volume flow $q_v(\text{m}^3/\text{s})$ instead of volume V. For instance, the value of volume flow is known in the suction inlet of a fan when the operating point of the fan is defined. Volume flow q_v, expressing the total airflow or the combined volume flow of water vapor and dry air, is not constant in various parts of the duct, because the pressure and temperature can vary. Therefore in technical calculations dealing with humid air, material flows are treated as mass flows. Also, while the humidity can vary, the basic quantity is dry air mass flow $\dot{m}_i$(kg d.a./s). If, for instance, we know the volume flow q_v of a fan, the dry air mass flow through the fan is

$$\dot{m}_i = p_i q_v \tag{4.91}$$

where p_i is the partial pressure of dry air in the suction inlet of the fan, in the same place where the total volume flow $\dot{V}$ is defined. Accordingly, the water vapor flow $\dot{m}_h$ (kg H_2O/s) along the volume flow is

$$\dot{m}_h = p_h q_v, \tag{4.92}$$

where p_h is the partial pressure of water vapor.

Due to the definition of humidity (4.87), on the basis of the Eqs. (4.91) and (4.92),

$$\dot{m}_h = x\dot{m}_i, \tag{4.93}$$

When an energy balance is written, an enthalpy flow of humid air $\dot{H}$ is needed. This can be written according to the Eqs. (4.85) and (4.93) as

$$\dot{H} = \dot{m}_i h_i + \dot{m}_h h_h = \dot{m}_i(h_i + x h_h)$$

or briefly, using definition (4.86),

$$\dot{H} = \dot{m}_i h_k. \tag{4.94}$$

In energy balance calculations, which we will handle later on, we use Eq. (4.94).

Example 1

A fan takes +15°C humid air at 0.5 m^3/s. What is the dry air mass flow $\dot{m}_i$ taken by the fan when the outdoor humidity is $x = 0.009$ and the outdoor pressure is $p = 1.0$ bar?

Partial pressure of water vapor is calculated from Eq. (4.84)

$$p_h = \frac{0.009}{0.6220 + 0.009} \cdot 1.0 = 0.01426 \text{ bar}.$$

Dry air partial pressure is

$$p_i = p - p_h = 1.0 - 0.01426 = 0.986 \text{ bar} = 0.986\ 10^5 \text{ Pa}.$$

Partial density of dry air is calculated from Eq. (4.76)

$$p_i = \frac{0.986 \cdot 10^5 \cdot 0.0290}{8.314 \cdot (273.15 + 15)} = 1.194 \text{ kg d.a.}/m^3.$$

Dry air mass flow is calculated from Eq. (4.91)

$$\dot{m}_i = 1.194 \cdot 0.5 = 0.597 \text{ kg d.a.}/s.$$

Water vapor flow through the fan is

$$\dot{m}_h = x\dot{m}_i = 0.009 \cdot 0.597 = 0.005373 \text{ kg } H_2O/s.$$

4.2.3 Water vapor pressure in the presence of air

The equilibrium between water and water vapor in the case of humid air is illustrated in Fig. 4.7.

The state of equilibrium differs from the equilibrium between water and pure water vapor in that, in a gas phase, there is also inert gas (dry air) present. This means that the water pressure is equal to the total gas pressure, $p = p_i + p_h$, not to the water vapor pressure p_h.

In a state of equilibrium the chemical potentials of water and water vapor are equal

$$\mu_v(T,\ p) = \mu_h(T,\ p_h), \tag{4.95}$$

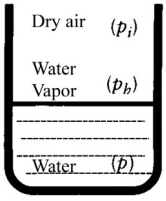


FIGURE 4.7 Equilibrium between water and vapor in the presence of air.

where the subscript v refers to water and h to water vapor. Notice that $p = p_i + p_h$.

From Eq. (4.95) the partial pressure of water vapor p_h can be solved for, and we see that it is dependent on temperature and the partial pressure of dry air

$$p_h = p_h(T,\ p_i). \tag{4.96}$$

Next we will show that the dependence of water vapor pressure on the partial pressure of dry air is very small, and consequently, a good approximation is

$$p_h = p_h(T). \tag{4.97}$$

To show this, Eq. (4.95) is differentiated, and we get

$$\frac{\partial \mu_v}{\partial T}dT + \frac{\partial \mu_v}{\partial p}dp = \frac{\partial \mu_h}{\partial T}dT + \frac{\partial \mu_h}{\partial p_h}dp_h. \tag{4.98}$$

On the other hand,

$$\begin{aligned} \frac{\partial \mu_v}{\partial T} &= -s_v, \quad & \frac{\partial \mu_h}{\partial T} &= -s_h, \\ \frac{\partial \mu_v}{\partial p} &= v_v, \quad & \frac{\partial \mu_h}{\partial p_h} &= v_v, \end{aligned} \tag{4.99}$$

where s_v is the specific entropy of liquid water, s_h is the specific entropy of water vapor, v_v is the specific volume of water, and v_h is the specific volume of water vapor. Notice that $v_h = 1/\rho_h$. Substituting Eq. (4.99) in Eq. (4.98), we obtain

$$-s_v dT + v_v dp = -s_h dT + v_h dp_h,$$

and it follows that

$$dp_h = \frac{s_h - s_v}{v}dT + \frac{v_v}{v_h}dp. \tag{4.100}$$

On the other hand, while $\mu = h - Ts$, in accordance with the balance clause (4.95),

$$h_v - Ts_v = h_h - Ts_h,$$

and it follows that

$$s_h - s_v = \frac{h_h - h_v}{T}.$$

Substituting this equation in Eq. (4.100), we obtain

$$dp_h = \frac{h_h - h_v}{T v_h}dT + \frac{v_v}{v_h}dp. \tag{4.101}$$

On the other hand, the differential of the total pressure is

$$dp = dp_i + dp_h,$$

which, substituted in Eq. (4.101), gives

$$dp_h = \frac{h_h - h_v}{T(v_h - v_v)}dT + \frac{v_v}{v_h - v_v}dp_i. \tag{4.102}$$

This differs from the water pressure equation of Clapeyron that lacks the last term. If $p_i = 0$ or $dp_i = 0$, then Eq. (4.102) is identical to the Clapeyron equation, as it should be.

Considering that $v_h,\ >> v_v$ and $v_h = 1/\rho_h$ and using Eq. (4.78), an approximation to Eq. (4.102) is obtained

$$\frac{dp_h}{p_h} = \frac{M_h(h_h - h_v)}{R\,T^2}dT + \frac{M_h v_v}{R\,T}dp_i. \tag{4.103}$$

On the other hand,

$$\frac{dp_h}{p_h} = d\left(\ln p_h\right),$$

so according to Eq. (4.103), we can write

$$\left(\frac{\partial(\ln p_h)}{\partial T}\right)_{p_i} = \frac{M_h(h_h - h_v)}{R\,T^2} \tag{4.104}$$

$$\left(\frac{\partial(\ln p_h)}{\partial p_i}\right)_{T} = \frac{M_h v_v}{R\,T} \tag{4.105}$$

The specific volume of water is approximately $v_v = 10^{-3}\,\mathrm{m^3/kg}$, and an estimate of Eq. (4.105) at a temperature of 50°C is

$$\left(\frac{\partial(\ln p_h)}{\partial p_i}\right)_{T} = \frac{0.0180\cdot 10^{-3}}{8.314\cdot 323.15}\frac{1}{\mathrm{Pa}} = 6.70\cdot 10^{9}\,\frac{1}{\mathrm{Pa}}.$$

Integrating Eq. (4.105) with the help of this value, we can examine the effect of air on the vapor pressure. When the partial pressure of air $p_i = 0$ and when the partial pressure of air is 10^5 Pa, a ratio of the vapor pressures corresponding to these situations at the temperature of 50°C is obtained

$$\ln\frac{p_h(50^\circ\mathrm{C},\ p_i = 10^5\ \mathrm{Pa})}{p_h(50^\circ\mathrm{C},\ p_i = 0)} = 6.70\cdot 10^{-9}\,\frac{1}{\mathrm{Pa}}\cdot(10^5 - 0)\ \mathrm{Pa} = 6.70\cdot 10^{-4}$$

or

$$\frac{p_h(50^\circ\mathrm{C},\ p_i = 10^5\ \mathrm{Pa})}{p_h(50^\circ\mathrm{C},\ p_i = 0)} = e^{6.70\cdot 10^{-4}} = 1.0006702.$$

When $p_i = 0$, a situation is described where water and water vapor are in an equilibrium without the presence of dry air. The corresponding vapor pressure can be found in tables for vapor

$$p_h(50^\circ\mathrm{C},\ P_i = 0) = p'h(50^\circ\mathrm{C}) = 0.12335\ \mathrm{bar}.$$

The difference between vapor pressures is $p_h(50°C, p_i = 10^5\ \text{Pa}) - p_h(50°C, p_i = 0) = 8\ \text{Pa}$.

This shows that the presence of air in the gas phase has a very small influence on the vapor pressure of water. Repeating the same calculation procedure for other temperatures, we can show that the vapor pressure of water can with good accuracy be taken from the vapor pressure tables for saturated water (water has the same pressure as water vapor when they are in equilibrium), as though there was no air in the gas phase. So the vapor pressure of water is with good accuracy also in this case just a function of temperature, and Eq. (4.97) is valid. New vapor pressure tables will not be needed for calculations with humid air.

4.2.4 Vapor pressure of water and ice and calculation of humid air state values

The partial pressure of water vapor in air cannot be higher than the vapor pressure of saturated water $p'_h(T)$ corresponding to air temperature T. If it were higher, condensation of water vapor would occur until the equilibrium state corresponding to the saturated vapor pressure was achieved.

Saturated water vapor pressure is most accurately found from vapor tables or can be approximated with the following equation

$$\log p'_h(\theta) = 28.59051 - 8.2\log(\theta + 273.16) + 0.0024804(\theta + 273.16) - \frac{3142.31}{\theta + 273.16} \quad (4.106)$$

The logarithm in Eq. (4.106) is Briggsian (a logarithm with 10 as the base), pressure is in units of bar, and the temperature is in Celsius.

A simpler approximation for the pressure of saturated water vapor is

$$ph'(T) = p_o \exp\big(11.78(T - 372.79)/(T - 43.15)\big), \quad (4.107)$$

where the constant $p_o = 10^5$ Pa, and the temperature T is in degrees Kelvin.

When the temperature is under 0°C, the saturation pressure, p_h, is calculated using the vapor pressure of ice (ice turns into vapor directly, i.e., sublimates), and we can use the following empirical formula

$$\log p'_h(\theta) = 10.5380997 - \frac{2663.91}{\theta + 273.16} \quad (4.108)$$

The logarithm in Eq. (4.108) is Briggsian, pressure has units of mbar, and the temperature is in Celsius.

For the vapor pressure of ice the equation of Clapeyron can be obtained in the same way as for water

$$dp_h = \frac{h_h - h_j}{T(v_h - v_j)}dT, \quad (4.109)$$

where h_j is the enthalpy of ice, and v_j is the specific volume of ice.

The relative vapor pressure of air or the relative humidity is defined by the following equation

$$\phi = \frac{p_h}{p'_h(T)}, \quad (4.110)$$

where p_h is the partial pressure of water vapor in air, and $p'_h(T)$ is the saturated water vapor pressure at temperature T. When φ is 1 or 100%, we say that the air is saturated.

Example 2a

Find the properties of saturated air and $\varphi = 50\%$ air, when the total pressure of air is $p = 1.0$ bar, and the temperature of air is 20°C.

1. *Saturated air, $\varphi = 100\%$*

a. Pressure of saturated vapor $\rho'_h(20°C)$ from Eq. (4.106)

$$\log p'_h(20°C) = 28.59051 - 8.2\log(20 + 273.16) + 0.0024804(20 + 273.16) - \frac{3142.31}{20 + 273.16} = 1.631$$

$$p'_h(20°C) = 10^{-1.631}\ \text{bar} = 0.0234\ \text{bar}$$

b. Humidity x from Eq. (4.83)

$$x = 0.6220 \cdot \frac{0.0234}{1.0 - 0.0234} = 0.01490\ \text{kg}\ H_2O/\text{kg d.a.}$$

c. Densities ρ_i, ρ_h, and ρ from Eqs. (4.76), (4.78), and (4.79)

$$\rho_h = \frac{0.0234 \cdot 10^5 \cdot 0.018053}{8.314 \cdot 293.15} = 0.01731\ \text{kg}/m^3$$

$$\rho_i = \frac{(1.0 - 0.0234) \cdot 10^5 \cdot 0.028964}{8.314 \cdot 293.15} = 1.1606\ \text{kg}/m^3 \left(= \frac{\rho_h}{x} \right)$$

$$\rho = 1.1606 + 0.01731 = 1.178\ \text{kg}/m^3$$

$$h_k = 1.006 \cdot 20 + 0.01490 \cdot (2501 + 1.85 \cdot 20) = 57.9\ \text{kJ/kg d.a.}$$

2. *Humid air, $\varphi = 50\%$*

$$p'_h(20°C) = 0.0234\ \text{bar}$$

$$p'_h = 0.5 \cdot 0.0234 = 0.0117\ \text{bar}$$

$$x = 0.6220 \cdot \frac{0.0117}{1.0 - 0.0117} = 0.00736\ \text{kg}\ H_2O/k\text{.g.d.a.}$$

$$\rho_h = \frac{0.0117 \cdot 10^5 \cdot 0.018053}{8.314 \cdot 293.15} = 0.00865\ \text{kg}/m^3$$

$$\rho_i = \frac{(1.0 - 0.0117) \cdot 10^5 \cdot 0.028964}{8.314 \cdot 293.15}$$

$$= 1.174\ \text{kg}/m^3 \left(= \frac{\rho h}{x} = \frac{0.00863}{0.00736} \right)$$

$$\rho = 1.174 + 0.00865 = 1.183\ \text{kg}/m^3$$

$$h_k = 1.006 \cdot 20 + 0.00736 \cdot (2501 + 1.85 \cdot 20) = 38.8\ \text{kJ/kg d.a.}$$

Example 2b

Find the properties of saturated air and $\varphi = 50\%$ air, when the total pressure of air is $\pi = 0.825$ bar, and the temperature of air is 20°C.

1. *Saturated air, $\varphi = 100\%$*

$$\rho_h'(20^\circ C) = 0.0234 \text{ bar}$$

$$x = 0.6220 \cdot \frac{0.0234}{0.825 - 0.0234} = 0.01816 \text{ kg } H_2O/\text{kg d.a.}$$

$$\rho_h = 0.01731 \text{ kg}/m^3$$

$$\rho_i = \frac{(0.825 - 0.0234) \cdot 10^5 \cdot 0.028964}{8.314 \cdot 293.15} = 0.953 \text{ kg}/m^3 \left(= \frac{\rho_h}{x} \right)$$

$$\rho = 0.970 \text{ kg}/m^3$$

$$h_k = 1.006 \cdot 20 + 0.01816 \cdot (2501 + 1.85 \cdot 20) = 66.2 \text{ kJ/kg d.a.}$$

2. *Humid air, $\varphi = 50\%$*

$$\rho_h'(20^\circ C) = 0.0234 \text{ bar}$$

$$\rho_h = 0.0117 \text{ har}$$

$$x = 0.6220 \cdot \frac{0.0117}{0.825 - 0.0117} = 0.00895 \text{ kg } H_2O/\text{kg d.a.}$$

$$\rho_h = 0.0865 \text{ kg}/m^3$$

$$\rho i = \frac{(0.825 - 0.0117) \cdot 10^5 \cdot 0.028964}{8.314 \cdot 293.15}$$

$$= 0.967 \text{ kg}/m^3 \left(= \frac{\rho_h}{x} = \frac{0.00863}{0.00736} \right)$$

$$h_k = 1.006 \cdot 20 + 0.00895 \cdot (2501 + 1.85 \cdot 20) = 42.8 \text{ kJ/kg d.a.}$$

Comparing Examples 2a and 2b, we notice that the total air pressure has effects on the humidity x, partial density of dry air p_i, total pressure or pressure of humid air, and enthalpy h_k. Knowing the total pressure is therefore essential in calculations of the thermodynamic properties of humid air.

Pressure and humidity have also an effect on the mass flows. We continue Examples 2a and 2b by calculating the dry air mass flow in a fan when the humid air volume flow in the fan is 0.8 m^3/s. According to Eq. (4.91) and the calculations earlier, we obtain

$$\dot{m}_i = \rho_i q_v = 1.1606\ 0.8 = 0.928 \text{ kg d.a./s}$$

$$\dot{m}_h = 0.01490\ 0.928 = 0.01383 \text{ kg/s} \qquad \text{(Ex.1a)}$$

$$\dot{m}_i = 1.174\ 0.8 = 0.939 \text{ kg d.a./s}$$

$$\dot{m}_h = 0.00736\ 0.939 = 0.00691 \text{ kg/s} \qquad \text{(Ex.1b)}$$

$$\dot{m}_i = 0.953\ 0.8 = 0.762 \text{ kg d.a./s}$$

$$\dot{m}_h = 0.01816\ 0.762 = 0.01384 \text{ kg/s} \qquad \text{(Ex.2a)}$$

$$\dot{m}_i = 0.967\ 0.8 = 0.774 \text{ kg d.a./s}$$

$$\dot{m}_h = 0.00895\ 0.774 = 0.00693 \text{ kg/s} \qquad \text{(Ex.2b)}$$

Thus the total mass flows $\dot{m} = (\dot{m}_i + \dot{m}_h)$ differ in different cases. Water vapor flow $\dot{m}_h$ is obtained by multiplying the dry air mass flow by the corresponding humidity x (Eq. 4.93). As a basic quantity in humid air mass and energy balance calculations, we use dry air mass flow $\dot{m}_i$, and the effect of humidity on the energy balance is noted in the enthalpy h_k (Eq. 4.87).

4.2.5 Construction of a Mollier diagram

Properties of humid air are usually described by means of the Mollier diagram. The Mollier diagram is constructed for a certain air pressure, normally for the value

$$p = 1.013 \text{ bar} = 760 \text{ mmHg} = 1 \text{ atm},$$

which is the so-called standard atmosphere pressure. As suggested in Examples 1 and 2, a Mollier diagram can be used only when the total pressure is the same or almost the same as the pressure for which the diagram was constructed.

The abscissa of the Mollier diagram is humidity x. The axis is provided with a pitched scale. A straight line is drawn with a 45 degrees angle to the abscissa, and it is provided with an enthalpy scale (h_k) according to the equation $h_k = l_{ho} \times x = 2501.6 \times x$, kJ/kg (Fig. 4.8). Consequently, the enthalpic scale (h_k) is also pitched.

Now we construct this oblique-angled coordinate system with isotherms.

$$\theta = 0^\circ\text{C},\ h_k = c_{pi}\theta + x\left(c_{ph}\theta + l_{ho}\right) = x l_{ho} = 2501x,\ \text{kJ/kg}.$$

While the h_k scale was constructed with the equation $h_k = 2501x$, it is noticed that the isotherm $\theta = 0$°C joins the abscissa.

$$\text{For } \theta = \theta_{1g}\ h_k = c_{pi}\theta_1 + x\left(c_{ph}\theta_1 + l_{ho}\right)$$
$$= 1.006\theta_1 + x(1.85\theta_1 + 2501).$$

The isotherm $\theta = \theta_1$ is a straight line in the h_k–x coordinate system. The isotherms are not exactly paral-

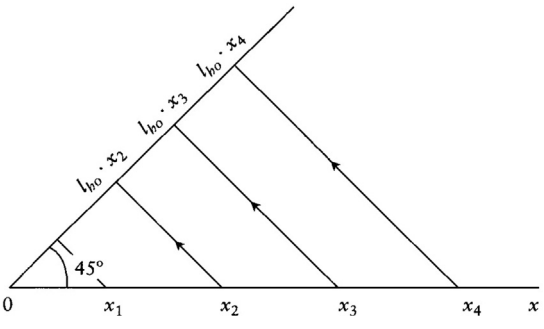


FIGURE 4.8 Constructing the enthalpy scale.

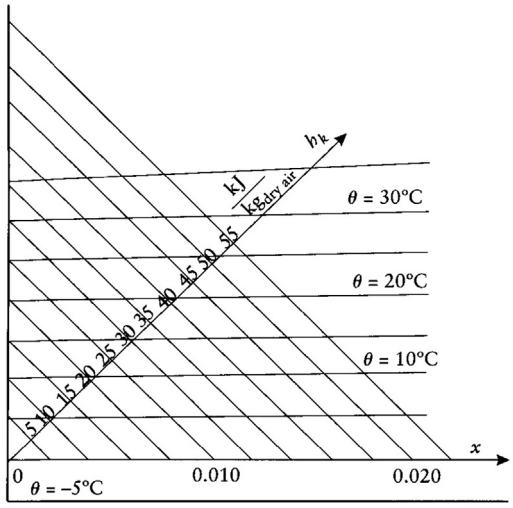


FIGURE 4.9 Fundamental scales of the Mollier diagram.

TABLE 4.6 Values calculated for the construction of the Mollier diagram when the total pressure is $p = 0.875$ bar.

		$\varphi = 100\%$		$\varphi = 50\%$	
θ (°C)	$p'_h(bar)$	x'	$h'_k(kJ/kg)$	$x_{50\%}$	$h'_k(kJ/kg)$
−10	0.00260	0.00185	− 5.47	0.000925	
−5	0.00402	0.00287	+ 2.12	0.001432	
0	0.00611	0.00437	10.93	0.00218	
+5	0.00872	0.00626	20.7	0.00311	
10	0.01227	0.00885	32.4	0.00439	
15	0.01704	0.01235	46.3	0.00612	
20	0.0234	0.01709	63.5	0.00843	41.5
25	0.0317	0.0234	84.8	0.01147	54.4
30	0.0424	0.0317	111.2	0.01544	69.7
35	0.0562	0.0427	144.8	0.0206	88.1
40	0.0738	0.0573	187.8	0.0274	110.8

lel due to the term $c_{ph}\theta_1$. When $\theta_1 < 0°C$, the isotherms go downward (relative to the abscissa), and when $\theta_1 > 0°C$, the isotherms go upward.

The isotherms cut the y-axis at $x = 0$ or $h_k = c_{pi}\theta$. When c_{pi} is held constant, it follows that the temperature scale is pitched.

When the saturation curve ($\varphi = 100\%$) and other curves of relative humidity (φ) are added, the Mollier diagram is complete (Fig. 4.9).

Example 3

Construct a Mollier diagram for air, when the total pressure of air is 875 mbar.

First, we construct the h_k–x coordinate system according to the instructions given earlier.

We provide the y-axis $x = 0$ with temperatures with the help of the equation

$$h_k = c_{pi}\theta + x\left(c_{ph}\theta + l_{ho}\right) = c_{pi}\theta = 1006\theta,\ \text{kJ/kg}.$$

When $\theta = -5°C$, $h_k = -5.03$ kJ/kg, and when $\theta = +5°C$, $h_k = +5.03$ kJ/kg, etc. Points where the isotherms cut the y-axis are located pitched.

Next we draw the saturation curve in the h_k–x coordinate system. Vapor pressures can be calculated with Eqs. (4.106) and (4.108) or taken directly from the tables. The humidity x' corresponding to the saturation pressure $p'_h(t)$ is calculated with Eq. (4.83) noting that $p = 0.875$ bar. The enthalpy of humid saturated air is calculated with Eq. (4.94)

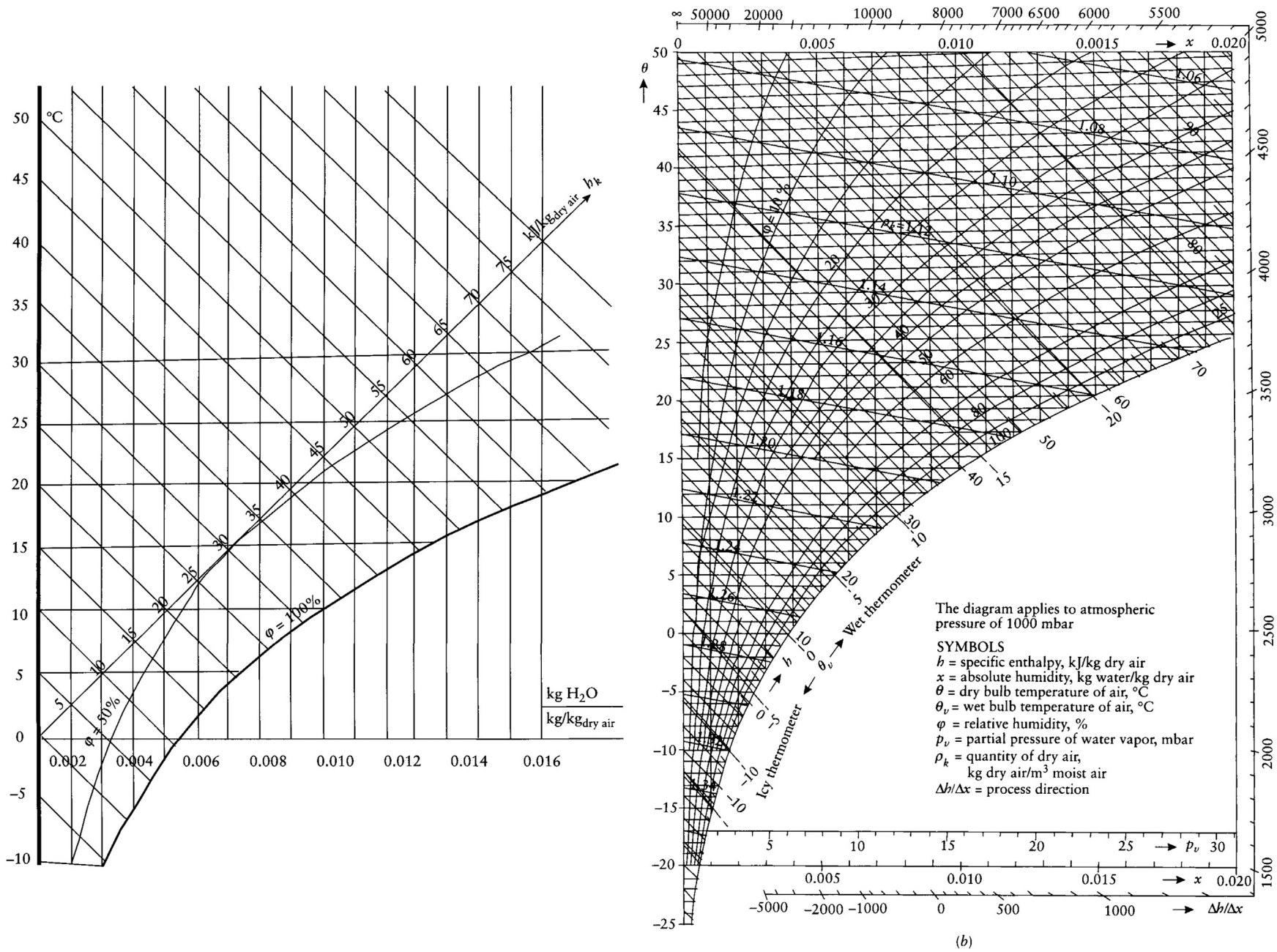
$$h'_k = 1.006\,\theta + x'(1.85\,\theta + 2501),\ \text{kJ/kg}.$$

The saturation curve is drawn through points (x', h'_k), calculated for different temperatures. At the same time the other ends of isotherms are determined, and because they are straight lines, they can now be drawn.

The curves of relative humidity φ_1, φ_2, ... can now be easily drawn with the help of the isotherms by just calculating the humidity corresponding to φ_1, φ_2, ... and using the already constructed isotherms.

With high temperatures the x' values will not fit into the diagram. Then the h_k values have to be calculated with smaller x values in order to draw the isotherms. In Table 4.6, these values are calculated with values $x = x_{50\%}$ at various temperatures. Drawing the fundamental axes and isotherms with the instructions given earlier and the saturation curve with the help of Table 4.6 leads to the Mollier diagram in Fig. 4.10A.

In Fig. 4.10B–D, some commonly used Mollier diagrams are presented. The diagram in Fig. 4.10B is valid for the air pressure $p = 1$ bar and is used in conventional calculations of air conditioning technology. Fig. 4.10C is an American version of Fig. 4.10B. It is a mirror image, and the direction of the scales is reversed. A diagram that covers a very wide temperature range and is therefore excellently suited to applications in the field of process technology is presented in Fig. 4.10D. This diagram is used, for example, in the technical design of the drying part of a paper machine. In Fig. 4.10D the enthalpy scale is on the abscissa and the curves of constant enthalpy are straight lines. The curves give the humidity relation, which is defined as $f = x/x'(\theta)$, where $x'(\theta)$ is the humidity of saturated air at temperature θ. Humidity relation f and relative humidity φ are different figures and should not be mixed.


°C
kJ/kg dry air
h_k
φ = 100%
φ = 50%
kg H_2O
kg/kg dry air
0.002
0.004
0.006
0.008
0.010
0.012
0.014
0.016
θ
x
p_v
$\Delta h/\Delta x$
Wet thermometer
Icy thermometer
The diagram applies to atmospheric pressure of 1000 mbar
SYMBOLS
h = specific enthalpy, kJ/kg dry air
x = absolute humidity, kg water/kg dry air
θ = dry bulb temperature of air, °C
θ_v = wet bulb temperature of air, °C
φ = relative humidity, %
p_v = partial pressure of water vapor, mbar
ρ_k = quantity of dry air, kg dry air/m³ moist air
$\Delta h/\Delta x$ = process direction

(*b*)

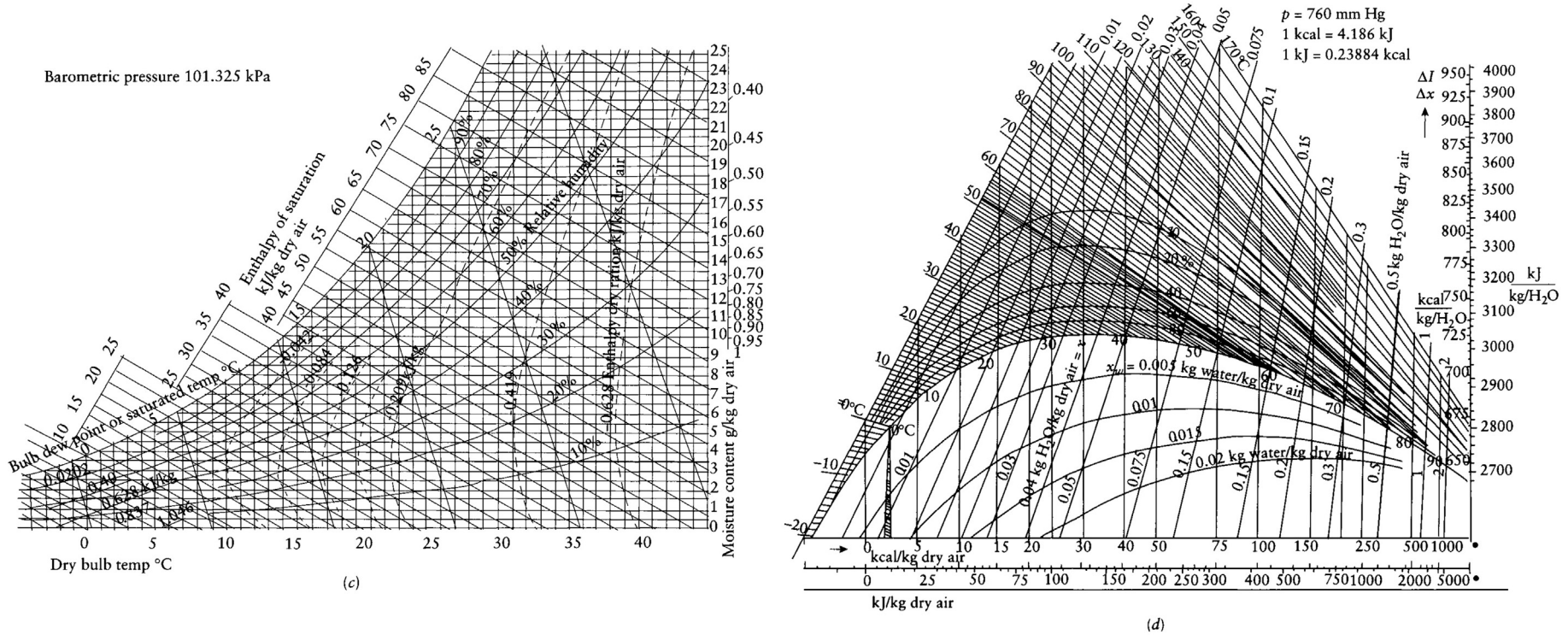


FIGURE 4.10 **(***A***)** Mollier diagram, $p = 0.875$ bar; (B) Mollier diagram; (C) Mollier diagram; and **(**D**)** Mollier diagram for humid air with the perspective modification of Salin–Soininen.

4.2.6 Determination of air humidity

The humidity of air can be measured by either the dew point of the air or its wet-bulb temperature.

Dew point means the temperature of saturated water vapor that has the same vapor pressure as the humid air in question. When the total pressure is constant, the constant vapor pressure means the same as the humidity x. In other words, dew point is the temperature of saturated air that has the same humidity as the air being considered.

By cooling a certain surface so cold that water starts condensing on it and measuring that temperature, the dew point can be measured. Combining this with the measurement of the dry-bulb temperature, the state of air can be defined.

Example 4

The dry-bulb temperature of air is 20°C, and the dew point is 8°C. What is the relative humidity?

$$h'_p(20°C) = 0.0234 \text{ bar } h'_p(8°C) = 0.01072 \text{ bar}$$
$$p_h\phi = p'_h(20°C) = 0.01072/0.0234 = 0.458 = 45.8\%.$$

Example 5

The air pressure in a room is 950 mbar, the temperature is 20°C, and the relative humidity $\varphi = 40\%$. Define the dew point of the room.

$$h'_p(20°C) = 0.0234 \text{ bar } p_h = 0.4 \cdot 0.0234 \text{ bar} = 0.00936 \text{ bar}$$

θ_1 is found by using the tables; $h'_p(\theta_1) = 0.00936$ bar, and thus $\theta_1 = 6.0°C$.

The total pressure thus has no importance. If this result is sought from a Mollier diagram by finding the intersection of the humidity line (x = humidity of air = constant) and the saturation curve, which gives the dew point temperature, a diagram constructed for a pressure of 950 mbar should be used. A decent approximation can be found from a diagram constructed for pressure $p = 1$ bar.

When a damp cloth is laid in an airflow, it settles after a certain time to an equilibrium temperature, the so-called wet-bulb temperature (θ_M), which is determined through heat and mass transfer. Negotiating the heat flow obtained by radiation and conduction, the heat balance of the wet cloth in a stationary situation can be expressed as

$$\alpha(\theta - \theta_M) = m''_h \, l(\theta_M), \tag{4.111}$$

where θ is the temperature of the airflow (°C), $\dot{m}''_h$ is the water flow vaporizing from the damp cloth [kg/(m^2 s)], $l(\theta_M)$ is the vaporization heat of water at temperature θ_M (J/kg), and α is the convective heat transfer coefficient [W/(m^2 °C)].

An equation for the water flow vaporizing from the surface of the cloth can be obtained as

$$m''_h = M_h \frac{p}{RT} k \ln \frac{p - p_h}{p - h'_p(\theta_M)}, \tag{4.112}$$

where k is the mass transfer coefficient (m/s). According to the analogy between heat and mass transfer coefficient,

$$k = \frac{\alpha}{\rho c_p} \mathrm{Le}^{1-n}, \tag{4.113}$$

where the power n varies between 0.33 and 0.5 and

$$\rho c_p = \rho_h c_{ph} + \rho_i c_{pi} \tag{4.114}$$

Also,

$$\mathrm{Le} = \frac{\rho c_p D}{\lambda}. \tag{4.115}$$

The dimensionless number Le is called the Lewis number (in Russian literature it is called the Luikov number). The Lewis number incorporates the specific heat capacity of humid air ρc_p [J/(m^3 °C)], the diffusion factor of water vapor in air D (m^2/s), and the heat conductivity of humid air λ [W/(m °C)]. In Table 4.7 the thermodynamic properties of saturated air, including the diffusion factor and the heat conductivity are presented.

Substituting Eqs. (4.112) and (4.113) into Eq. (4.111), we obtain

$$\theta - \theta_M = \frac{M_h}{\rho c_p} \frac{p}{RT} \mathrm{Le}^{1-n} l(\theta_M) \ln \frac{(p - p_h)}{p - h'_p(\theta_M)}, \tag{4.116}$$

from which we observe that the heat transfer factor α has been reduced. The only factor in Eq. (4.116) depending on the airflow conditions of the measurement is the power n in the Lewis number. Because the value of the Lewis number is very close to 1, the effect of n is very small.

The wet-bulb temperature θ_M can be solved from Eq. (4.116) when the state of the air, the temperature t, and the partial pressure of water vapor p_h are known. Inversely, if the temperature t and the wet-bulb temperature θ_M are known, the partial pressure and consequently the humidity of air can be found from Eq. (4.116).

Example 6

Given the temperature of air θ = °C and the wet-bulb temperature $\theta_M = 10°C$, calculate the humidity x, when the pressure of air is (1) $p = 1$ bar and (2) $p = 0.90$ bar.

By solving for the steam pressure ρ_h from Eq. (4.116), we obtain

TABLE 4.7 Thermodynamic characteristics of saturated air for a total pressure of 100 kN/m^2.

Temperature (°C)	Humidity (kg H_2O/ kg dry air)	Water vapor partial pressure (kPa)	Water vapor partial density (kg/m^3)	Water vaporization heat (kJ/kg)	Mixture enthalpy (kJ/kg dry air)	Dry air partial density ($kg_{dry\ air}$/ m^3)	Kinematic viscosity (10^4 m^3/s)	Specific heat (kJ/K kg)	Heat conductivity [W/(m K)]	Diffusion factor water-air (10^4 m^3/s)	Temperature (°C)
0	0.003 821	0.6108	0.004 846	2 500.8	9.55	1.285	13.25	1.0108	0.023 80	22.2	0
2	0.004 418	0.705 4	0.005 557	2 495.9	13.06	1.275	13.43	1.012 0	0.024 13	22.4	2
4	0.005 100	0.812 9	0.006 358	2 491.3	16.39	1.264	13.61	1.013 4	0.024 27	22.6	4
6	0.005 868	0.934 6	0.007 257	2 486.6	20.77	1.254	13.79	1.014 9	0.024 40	22.8	6
8	0.006 749	1.072 1	0.008 267	2 481.9	25.00	1.243	13.97	1.016 7	0.024 54	23.1	8
10	0.007 733	1.227 1	0.009 396	2 477.2	29.52	1.232	14.15	1.018 6	0.024 66	23.3	10
12	0.008 849	1.401 5	0.010 66	2 472.5	34.37	1.221	14.34	1.020 8	0.024 78	23.6	12
14	0.010 105	1.597 4	0.012 06	2 467.8	39.57	1.211	14.52	1.023 3	0.024 90	23.9	14
16	0.011 513	1.816 8	0.013 63	2 463.1	45.18	1.199	14.71	1.026 0	0.025 00	24.2	16
18	0.013 108	2.062	0.015 36	2 458.4	51.29	1.188	14.89	1.029 1	0.025 11	24.5	18
20	0.014 895	2.337	0.017 29	2 453.1	57.86	1.177	15.08	1.032 5	0.025 20	24.8	20
22	0.016 892	2.642	0.019 42	2 449.0	65.02	1.175	15.27	1.036 4	0.025 29	25.2	22
24	0.019 131	2.982	0.021 77	2 442.0	72.60	1.154	15.46	1.040 7	0.025 37	25.5	24
26	0.021 635	3.360	0.024 37	2 439.5	81.22	1.141	15.65	1.045 5	0.025 44	25.9	26
28	0.024 435	3.775	0.027 23	2 434.8	90.48	1.129	15.84	1.050 9	0.025 08	26.3	28
30	0.027 558	4.241	0.030 36	2 430.0	100.57	1.116	16.03	1.056 9	0.025 56	26.6	30
32	0.031 050	4.753	0.033 80	2 425.3	111.58	1.103	16.22	1.063 5	0.025 61	27.0	32
34	0.034 950	5.318	0.037 58	2 420.5	123.72	1.090	16.41	1.071 0	0.025 65	27.4	34
36	0.039 289	5.940	0.041 71	2 415.8	136.99	1.076	16.61	1.079 3	0.025 67	27.8	36
38	0.044 136	6.624	0.046 22	2 411.0	151.60	1.061	16.80	1.088 5	0.025 69	28.3	38
40	0.049 532	7.375	0.051 14	2 406.2	167.64	1.046	17.00	1.098 9	0.025 69	29.7	40
42	0.055 560	8.198	0.056 50	2 401.4	185.40	1.030	17.20	1.110 3	0.025 68	29.1	42
44	0.062 278	9.010	0.062 33	2 396.6	204.94	1.014	17.39	1.123 2	0.026 66	29.6	44
46	0.069 778	10.085	0.068 67	2 391.8	226.55	0.9979	16.59	1.137 5	0.025 63	30.0	46
48	0.078 146	11.161	0.075 53	2 387.0	250.45	0.9791	17.79	1.153 4	0.025 58	30.5	48
50	0.087 516	12.335	0.082 98	2 382.1	277.04	0.9606	17.99	1.171 3	0.025 52	30.9	50
52	0.098 018	13.613	0.091 03	2 377.3	306.64	0.9411	18.19	1.191 3	0.025 45	31.4	52

54	0.109 76	15.002	0.099 74	2 372.4	339.51	0.9207	18.39	1.213 7	0.025 36	31.9	54
56	0.122 97	16.509	0.109 1	2 367.6	373.31	0.8999	18.59	1.238 9	0.025 26	32.4	56
58	0.137 90	18.146	0.119 3	2 362.7	417.72	0.8768	18.79	1.267 3	0.025 14	32.9	58
60	0.154 72	19.92	0.130 2	2 357.9	464.11	0.8532	18.99	1.299 4	0.025 01	33.4	60
62	0.173 80	21.84	0.141 9	2 353.0	516.57	0.8283	19.19	1.335 7	0.024 87	34.0	62
64	0.195 41	23.91	0.154 5	2 348.1	575.77	0.8021	19.38	1.377 0	0.024 71	34.5	64
66	0.220 21	26.14	0.168 0	2 343.1	643.51	0.7746	19.57	1.424 1	0.024 55	35.1	66
68	0.248 66	28.55	0.182 6	2 338.2	721.01	0.7456	19.76	1.478 2	0.024 37	35.7	68
70	0.281 54	31.16	0.198 1	2 333.3	810.36	0.7150	19.94	1.541 8	0.024 18	36.3	70
72	0.319 66	33.96	0.214 6	2 328.3	915.57	0.6829	20.01	1.613 2	0.023 99	36.9	72
74	0.364 68	36.96	0.232 4	2 323.3	1 035.60	0.6489	20.28	1.698 6	0.023 79	37.6	74
76	0.417 90	40.19	0.251 4	2 318.3	1 179.42	0.6132	20.44	1.799 4	0.023 60	38.3	76
78	0.480 48	43.65	0.271 7	2 313.3	1 348.40	0.5755	20.58	1.919 9	0.023 41	39.0	78
80	0.559 31	47.36	0.293 3	2 308.3	1 560.80	0.5358	20.71	2.066 4	0.023 23	39.8	80
82	0.655 73	51.33	0.316 2	2 303.2	1 820.46	0.4939	20.81	2.247 7	0.023 07	40.7	82
84	0.777 81	55.57	0.340 6	2 298.1	2 148.92	0.4497	20.90	2.476 7	0.022 94	41.5	84
86	0.937 68	60.50	0.366 6	2 293.0	2 578.73	0.4031	20.96	2.773 9	0.022 85	42.5	86
88	1.152 44	64.95	0.394 2	2 287.9	3 155.67	0.3542	20.99	3.170 8	0.022 81	43.6	88
90	1.458 73	70.11	0.423 5	2 282.8	3 978.42	0.3026	20.99	3.730 4	0.02283	44.7	90
92	1.927 18	75.61	0.454 5	2 277.6	5 236.61	0.2482	20.94	4.574	0.022 95	46.0	92
94	2.731 70	81.46	0.487 3	2 272.4	7 395.49	0.1909	20.84	5.987	0.023 18	47.4	94
96	4.426 70	87.69	0.522 1	2 267.1	11 944.39	0.1305	20.69	8.820	0.023 55	49.0	96
98	10.303 06	94.30	0.558 8	2 261.9	27 711.34	0.06694	20.47	17.338	0.024 09	50.8	98
100	•	101.325	0.597 7	2 256.7	•	0	20.08	•	0.024 86	52.8	100

$$p - p_h = (p - h'_p(\theta_M))\exp\left[(\theta - \theta_M)\left(\frac{\rho c_p}{M_h}\frac{RT}{p}\frac{1}{\mathrm{Le}^{1-n}}\frac{1}{1(\theta_M)}\right)\right]. \quad (4.117)$$

The diffusion factor D, which is contained in the Lewis number, is inversely related to the total pressure

$$D = \frac{1}{p}f(T) \quad (4.118)$$

From Table 4.7, we obtain the diffusion factor at the temperature of 10°C and pressure $p = 1.0$ bar: $D(10°\mathrm{C}, p = 1.0\ \mathrm{bar}) = 23.3 \times 10^{-6}\mathrm{m}^2/\mathrm{s}$. The diffusion factor at the same temperature but a pressure $p = 0.9$ bar is, according to Eq. (4.118), $D(10°\mathrm{C}, p(0.9\ \mathrm{bar})) = 25.9 \times 10^{-6}\ \mathrm{m}^2/\mathrm{s}$.

For heat conductance, there is a dependency on pressure almost like that of Eq. (4.118), so for good accuracy it is a valid approximation that $D/\lambda = g(T)$. From Table 4.7 we have $\lambda(10°\mathrm{C}, p(1.0\ \mathrm{bar})) = 0.02466\ \mathrm{W/(m\ K)}$, and so $D/\lambda = 9.45 \times 10^{-4}\ \mathrm{m^3\ K/J}$. Vaporization heat and the pressure of saturated steam are, from Table 4.7,

$$l(\theta_M) = 1(10°\mathrm{C}) = 2477 \cdot 10^3 J/\mathrm{kg},$$

$$p'_h(\theta_M) = p'_h(10°\mathrm{C}) = 0.012271\ \mathrm{bar}.$$

Temperature T is taken as a mean boundary layer temperature

$$T = (20 + 10)/2 + 273.15 = 288.15\ \mathrm{K}$$

Heat capacity,

$$\rho c_p = \rho_i c_{pi} + \rho_h c_{ph} = \rho_i(c_{pi} + x c_{ph}), \quad (4.119)$$

has to be iterated because steam pressure p_h and therefore humidity x are at this stage unknown. When humidity is low, as in this example, an approximation is $\rho c_p \cong \rho_i c_{pi}$. The calculation can be repeated when needed using the steam pressure from Eq. (4.117). In this example, we do not do that. For calculating the density and heat capacity, we will use an approximation $\rho_{cp} \cong \rho_i$, so $p \cong p_i$ and $cp \cong c_{pi}$. From this approximation, it follows that

$$\frac{\rho c_p}{M_h}\frac{RT}{p} \cong \frac{M_i}{M_h}c_p \cong \frac{M_i}{M_h}c_{pi}, \quad (4.120)$$

the value of which is

$$\frac{M_i}{M_h}c_{pi} = 1617\ J/(\mathrm{kg}\ °\mathrm{C}).$$

From Eq. (4.76), we get the partial pressure of dry air:

1. $\rho_i = \frac{p_i M_i}{RT} \cong \frac{pM_i}{RT} = \frac{10^5 \cdot 0.029}{8.314 \cdot 288.15} = 1.211\ \mathrm{kg/m^3}$,
2. $\rho_i = \frac{p_i M_i}{RT} \cong \frac{pM_i}{RT} = \frac{0.9 \cdot 10^5 \cdot 0.029}{8.314 \cdot 288.15} = 1.089\ \mathrm{kg/m^3}$,

The Lewis number in cases a and b is as follows:

1. $\mathrm{Le} = \frac{\rho c p D}{\lambda} \cong \rho_i c_{pi}\frac{D}{\lambda} = 1.211 \cdot 1006 \cdot 9.45 \cdot 10^{-4} = 1.151$
   a. It is assumed that $n = 0.5$, so $\mathrm{Le}^{1-n} = \mathrm{Le}^{1-0.5} = 1.073$
2. $\mathrm{Le} = \frac{\rho c p D}{\lambda} \cong \rho_i c_{pi}\frac{D}{\lambda} = 1.089 \cdot 1006 \cdot 9.45 \cdot 10^{-4} = 1.035$

Thus $\mathrm{Le}^{1-n} = 1.017$. Substituting all values into Eq. (4.117), we obtain the following:

1. $p - p_h = (1.0 - 0.012271)$

$$\exp\left[(20 - 10) \cdot 1617 \cdot \frac{1}{1.073} \cdot \frac{1}{2477.2 \cdot 10^3}\right]\mathrm{bar} = 0.99376\ \mathrm{bar}$$

$$p_h = (1.0 - 0.99376)\mathrm{bar} = 0.00624\ \mathrm{bar} = 624\ \mathrm{Pa}$$

$$x = 0.6220\frac{p_h}{p - p_h} = 0.6220 \cdot \frac{624}{10^5 - 624} = 0.00391$$

2. $p - p_h = (0.9 - 0.012271)$

$$\exp\left[(20 - 10) \cdot 1617 \cdot \frac{1}{1.017} \cdot \frac{1}{2477.2 \cdot 10^3}\right]\mathrm{bar} = 0.89345\ \mathrm{bar}$$

$$p_h = (0.9 - 0.89345)\mathrm{bar} = 0.00655\ \mathrm{bar} = 655\ \mathrm{Pa}$$

$$x = 0.6220\frac{p_h}{p - p_h} = 0.6220 \cdot \frac{655}{10^5 - 655} = 0.00456$$

Comparing the results for (1) and (2), we see that pressure has a considerable effect on the result. This is important to remember, especially in industrial ventilation and process measurements with notable underpressures and overpressures.

We will now derive an approximation for Eq. (4.116) that can be used when the partial pressure of water vapor in air is low compared with the total pressure.

First, we note that with fairly good accuracy it is valid that

$$\ln\frac{p - p_h}{p - p'_h(\theta_M)} = \ln\left[1 + \frac{p'_h(\theta_M) - p_h}{p - p'_h(\theta_M)}\right]$$

$$\cong \frac{p'_h(\theta_M) - p_h}{p - p'_h(\theta_M)} \cong \frac{p'_h(\theta_M)}{p - p'_h(\theta_M)} - \frac{p_h}{p - p_h}$$

where using Eq. (4.83) leads to the approximation

$$\ln\frac{p - p_h}{p - p'_h(\theta_M)} \cong \frac{M_i}{M_h}(x'(\theta_M) - x), \quad (4.121)$$

where $x'(\theta_M)$ the humidity of saturated air at temperature θ_M when the total pressure of air is p.

Substituting approximations (4.120) and (4.121) to Eq. (4.116), we obtain

$$\frac{x'(\theta_M) - x}{t - \theta_M} = \frac{c_p}{1(\theta_M)} \cdot \frac{1}{\mathrm{Le}^{1-n}} \tag{4.122}$$

The Lewis number for air is approximately 1 (see Example 6), so with good accuracy $\mathrm{Le}^{1-n} \cong 1$, and we get an approximation from Eq. (4.122)

$$\frac{x'(\theta_M) - x}{t - \theta_M} = \frac{c_p}{1(\theta_M)}. \tag{4.123}$$

We considered the abovementioned question of which temperature the damp cloth settles to when it is thermally insulated against all surroundings but the airflow, and when it can be assumed that there is no radiation heat transfer between the cloth and the airflow. In this consideration the state of the air has been constant.

If, instead, the air is damped adiabatically with the wet cloth so that the state of the air varies, the cloth will settle to a slightly different temperature. Each state of air (θ, x) is represented by a certain wet-bulb temperature θ_M, which can be calculated from Eq. (4.116) or its approximation (4.123), when the partial pressures of water vapor are low compared with the total pressure. When the state of air reaches the saturation curve, we have an interesting special case. Now the temperatures of the airflow and the cloth are identical. This equilibrium temperature is called the *adiabatic cooling border* or the *thermodynamic wet-bulb temperature* (θ_{ad}).

When air is humidified with water flow $\dot{m}_v$ and when the incoming and outgoing humid airflows are denoted $\dot{m}_1$ and $\dot{m}_2$, the energy balance of the conditioning chamber can be written as

$$\dot{m}_2 h_2 - \dot{m}_1 h_1 = \dot{m}_v h_v. \tag{4.124}$$

Eq. (4.124) is illustrated in Fig. 4.11.

In Eq. (4.124) the incoming enthalpy flow of humid air is

$$\dot{m}_1 h_1 = \dot{m}_{i1} h_{i1} + \dot{m}_{h1} h_{h1} \tag{4.125a}$$

and the outgoing enthalpy flow is

$$\dot{m}_2 h_2 = \dot{m}_{i2} h_{i2} + \dot{m}_{h2} h_{h2}. \tag{4.125b}$$

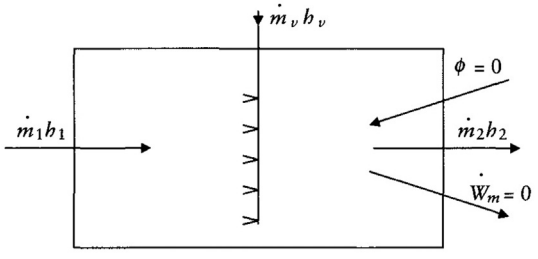


FIGURE 4.11 Energy balance for an adiabatic-conditioning chamber.

While the dry airflow stays constant, it can be written that

$$\dot{m}_i = \dot{m}_{i1} = \dot{m}_{i2} \tag{4.126}$$

and using Eq. (4.126), the enthalpy flows (4.125a) and (4.125b) can be written as

$$\dot{m}_1 h_1 = \dot{m}_i h_{kl} \tag{4.127a}$$

$$\dot{m}_2 h_2 = \dot{m}_i h_{k2}, \tag{4.127b}$$

where according to Eq. (4.87)

$$h_{k1} = h_{i1} + x_1 h_{h1} \tag{4.128a}$$

$$h_{k2} = h_{i2} + x_2 h_{h2}. \tag{4.128b}$$

When all the water fed to the conditioning chamber vaporizes, the following humidity balance is valid

$$\dot{m}_v = \dot{m}_{h2} - \dot{m}_{h1} = \dot{m}_i (x_2 - x_1). \tag{4.129}$$

Substituting (4.127a), (4.127b), and (4.129) into the energy balance (4.124), we obtain

$$\frac{\Delta h_k}{\Delta x} = h_v, \tag{4.130}$$

where $\Delta h_k = h_{k2} - h_{k1}$ and $\Delta x = x_2 - x_1$. When air is humidified adiabatically with water with temperature θ_{ad}, the enthalpy of water is now

$$h_v = c_{pv} \theta_{ad} \tag{4.131}$$

If, in addition, the air is humidified so that it reaches the saturation point, with the corresponding temperature θ_{ad}, we will now use the notations

$$\begin{aligned} h_{k2} &= h_{k\ ad} \\ x_2 &= x_{ad} \end{aligned} \tag{4.132}$$

Using the notations in (4.132), we can without danger of mix-up leave out the subscript 1 of the incoming point and write $x_1 = x$ and $h_{k1} = h_k$. Using these notations and Eq. (4.131), Eq. (4.130) can be written as

$$\frac{h_k - h_{k\ ad}}{x - x_{ad}} = c_{pv} \theta_{ad}. \tag{4.133}$$

So the state point of air in the Mollier diagram is shifted in a direction where the dependency between the enthalpy and the humidity change according to Eq. (4.133) is valid. This result is illustrated in Fig. 4.12.

Eq. (4.133) can be formally written in a form resembling Eq. (4.123). To demonstrate this, we first write Eq. (4.133) as

$$h_k - h_{ad} - c_{pv} \theta_{ad} (x - x_{ad}) = 0$$

On the other hand,

$$h_k = c_{pi} \theta + x (c_{ph} \theta + l_{ho}) = 0$$

and

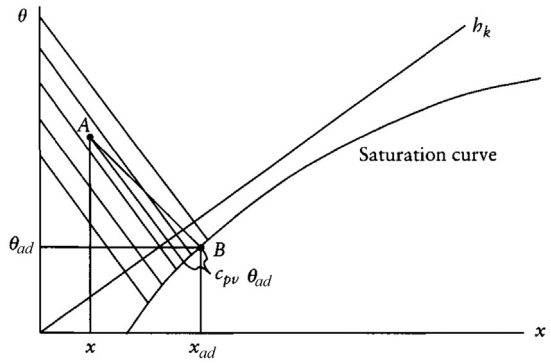


FIGURE 4.12 The state change of air in adiabatic humidification. *A* is the initial state, and *B* is the saturated final 1 state.

$$h_{kad} = c_{pi}\theta_{ad} + x_{ad}(c_{ph}\theta_{ad} + l_{ho})$$

Substituting these into the previous equation, we obtain by grouping terms appropriately

$$c_{pi}(\theta - \theta_{ad}) + xc_{ph}(\theta - \theta_{ad}) + x(l_{ho} + c_{ph}\theta_{ad} - c_{pv}\theta_{ad}) - x_{ad}(l_{ho} + c_{ph}\theta_{ad} - c_{pv}\theta_{ad}) = 0 \tag{4.134}$$

On the other hand, the vaporization heat of water at temperature θ_{ad} is

$$l(\theta_{ad}) = l_{ho} + c_{ph}\theta_{ad} - c_{pv}\theta_{ad} \tag{4.135}$$

and the specific heat of humid air per kilogram of dry air is

$$c_{pk} = c_{pi} + xc_{ph} \tag{4.136}$$

Substituting Eqs. (4.135) and (4.136) into Eq. (4.134), we get

$$\frac{x_{ad} - x}{\theta - \theta_{ad}} = \frac{c_{pk}}{l(\theta_{ad})}, \tag{4.137}$$

which is equivalent to Eq. (4.133).

Eq. (4.137) is almost exactly the same as the approximation Eq. (4.123) derived for wet-bulb temperature. When the partial pressure of water vapor is low compared with the total pressure—in other words, when the humidity x is low—the specific heat of humid air per kilogram of humid air, c_p, and the specific heat of humid air per kilogram of dry air, c_{pk}, are almost the same: $c_p \cong c_{pk}$. Therefore in a situation where the humidity is low and Le$\cong$1, the thermodynamic wet-bulb temperature is very nearly the same as the technical wet-bulb temperature θ_M.

Example 7

Draw in the Mollier diagram at the 14°C point of the saturation curve (1) the state change line of the adiabatic humidification and (2) an auxiliary line, associated with the wet-bulb temperature measurement, by means of which the state can be defined. The pressure of air is $p = 1$ bar (Table 4.7).

$p'_h(14°\text{C}) = 0.01597$ bar (Table 4.7)

$x_{\text{ad}} = 0.6220 \cdot \frac{0.01597}{1.0 - 0.01597} = 0.01009 = x'(\theta_M)$

$l(14°\text{C}) = 24678$ kJ/kg (Table 4.7)

$\rho_i = 1.211$ kg/m^3 (Table 4.7)

$p_h = 0.01206$ kg/m^3 (Table 4.7)

$\rho = \rho_i + \rho_h = 1.223$ kg/m^3

$\rho c_p = \rho_i c_{pi} + \rho$
$_h c_{ph} = 1.211 \times 1.006 + 0.01206 \times 1.85 = 1.241$ kJ/(m^3 °C)

$c_p = 1.015$ kJ/(kg °C)

$D = 23.9 \times 10^{-6}$ m^2/s (Table 4.7)

$\lambda = 0.0249$ W m/°C

$$\text{Le} = \frac{D\rho c_p}{\lambda} = \frac{23.9 \cdot 10^{-6} \cdot 1241}{0.0249} = 1.191$$

$$n \cong 0.5$$

1. We choose an auxiliary point on the isotherm $\theta = 25°\text{C}$. According to Eq. (4.137),

$$x_{\text{ad}} - x = \frac{1.025}{2467.8} \cdot (25 - 14) = 0.00457,$$

so it follows that the place of the auxiliary point is $x = 0.00552$.

2. For the auxiliary point on the isotherm $\theta = 25°\text{C}$, according to Eq. (4.122),

$$x'(\theta_M) - x = \frac{1.015}{2467.8} \cdot \frac{1}{1.191^{0.5}} \cdot (25 - 14) = 0.00415,$$

and it follows that the place of the auxiliary point is $x = 0.00594$. The results are illustrated in Fig. 4.13.

The enthalpy of humid air responding to the point 14°C on the saturation curve is

$$h_{kad} = 1.0016 \cdot 14 + 0.01009 \cdot (1.85 \cdot 14 + 2501) = 39.58\ \text{kJ/kg}.$$

The humidity at the temperature of 25°C corresponding to this enthalpy is determined from the equation

$$39.58 = 1.006 \cdot 25 + x(1.85 \cdot 25 + 2501),$$

and so $x = 0.00567$.

Comparing this value with the (1) and (2) point results of Example 7, we discover that the line of

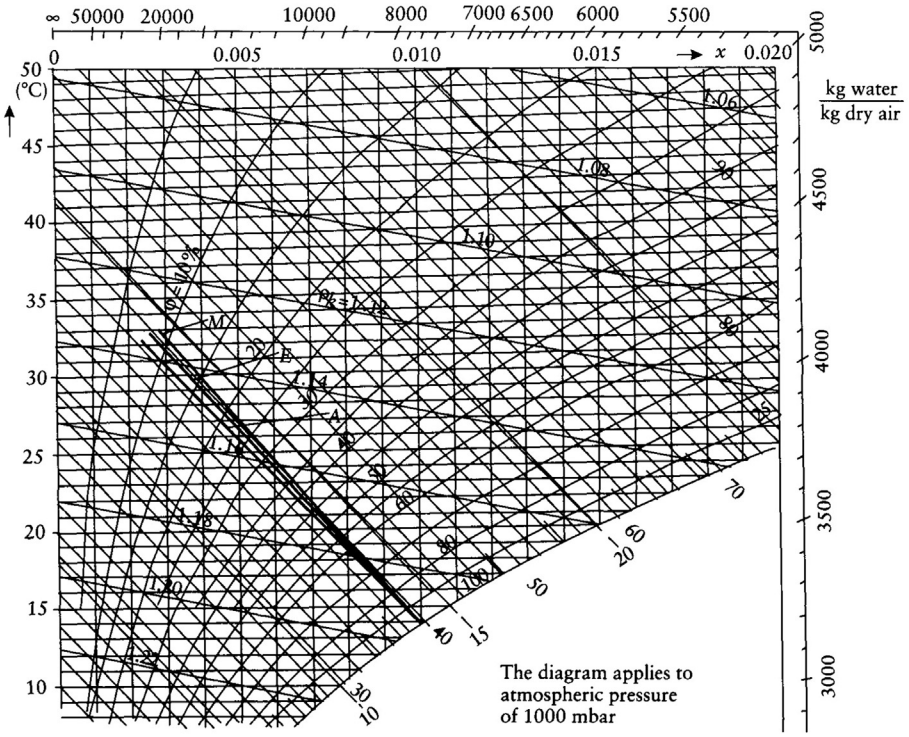


FIGURE 4.13 Determination lines for the state of air when $\theta_M = \theta_{ad}$ 14°C: M is the determination line for wet-bulb temperature, E is the constant enthalpy, and A is the adiabatic humidification line.

constant enthalpy lies between the determination line of wet-bulb temperature and the adiabatic humidification line. The nearer the Lewis number is to 1, the nearer the wet-bulb temperature is to the adiabatic humidification temperature.

In practical calculations the Mollier diagram's constant enthalpy line can be used as the auxiliary line for the wet-bulb temperature line to a satisfactory accuracy. The intersection of the constant enthalpy line with the isotherm responding to the temperature of air gives the humidity of air. For more accurate calculations, Eq. (4.116) should be used or its approximation (4.122) when the steam pressures (p_h and p'_h) are low compared to the total pressure of air (p).

As an example of using a Mollier diagram in defining the state of air, we can take a typical measurement from the local exhaust hood of a paper machine. The temperature of the exhaust air is 82°C and its wet-bulb temperature 60°C. In Fig. 4.10D, we move from the saturation curve at the point 60°C straight up along the constant enthalpy line ($h_k = 460$ kJ/kg d.a.) until we reach the isotherm $\theta = 82$°C. The intersection represents the state of air, and from Fig. 4.10D, we see that to the accuracy of the diagram $x = 0.14$ and the corresponding humidity relation $f \equiv x/x'(82°C) = 0.20$. Based on the values $x = 0.14$ and $p = 1.0$ bar, the relative vapor pressure φ can be calculated. From Eq. (4.84), we have $p_h = 0.183$ bar and from Table 4.7 p'_h (82°C) = 0.5133 bar; then on the basis of definition (4.110) $\varphi = (p_h/p'_h) = 0.356 = 35.6\%$. We see that the values of f and φ clearly differ from each other.

According to Eq. (4.122) when Le ≅ 1, $1(\theta_M) \cong$ 2450 kJ/kg and $c_p \cong 1.0$ kJ/(kg °C)

$$p_h = p'_h(\theta_M) - 6.6 \cdot 10^{-4} \cdot p \cdot (\theta - \theta_M) \frac{1}{°C} \qquad (4.138)$$

and by means of this the state of air can be approximately calculated. Often, we call the temperature of air the dry-bulb temperature to distinguish it from the wet-bulb temperature.

It is important to emphasize that, especially in process measurements, radiation can have an essential influence on the wet-bulb temperature, and therefore generally the wet-bulb temperature is dependent on the measurement device and the method of measurement. If the airflow is very low, the radiation can have a remarkable contribution in addition to the convective heat transfer. Basically, an equation analogous to Eq. (4.138) can be empirically determined for each wet-bulb temperature and method of measurement.

4.2.7 State changes of humid air

Now we will consider a balance borderline of the system presented in Fig. 4.14. The system can be any part of the air surrounding the process device. If an air-handling application is considered, the balance can

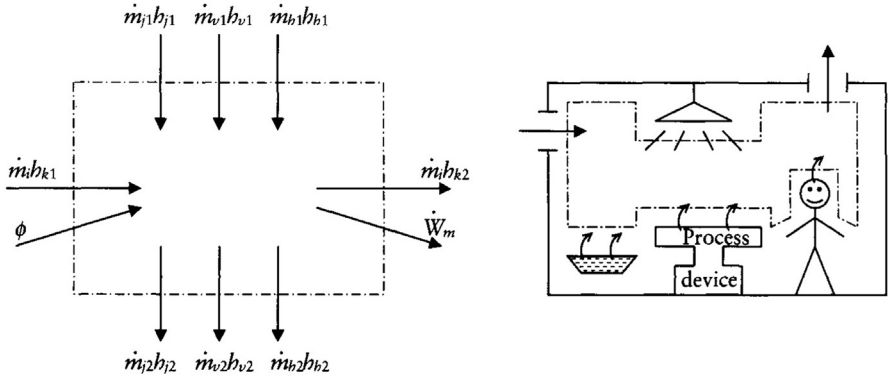


FIGURE 4.14 The energy-humidity borderline.

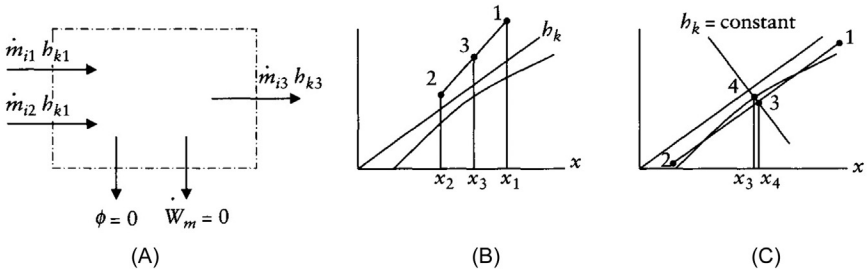


FIGURE 4.15 The mixing point in a Mollier diagram. Energy balance (A); mixing point (B); if the supersaturated area (C) is considered, the state of the air is driven to point 4 and $(x_3 - x_4)$ kg H_2O/kg d.a. of water is condensed in the mixing chamber.

be calculated over the inner air of a room, an office, or an industrial hall, for example.

The energy balance of the system, consisting of the area inside the balance borderline, is in a stationary situation

$$\varphi - \dot{W}_m = (\dot{m}_i h_{k2} - \dot{m}_i h_{k1}) + \left[(\dot{m}_{j2} h_{j2} - \dot{m}_{j1} h_{j1}) + (\dot{m}_{v2} h_{v2} - \dot{m}_{v1} h_{v1}) + (\dot{m}_{h2} h_{h2} - \dot{m}_{h1} h_{h1})\right]$$

where ϕ is the net heat power received by the system, $\dot{W}_m$ is the net work power to the environment preferred by the system, $\dot{m}_v$ is the water flow (1 = inflow and 2 = outflow), $\dot{m}_j$ is the ice flow, and $\dot{m}_h$ is the separate clean steam flow not included in the airflows. The steam flows included in the humid airflow are $\dot{m}l_1$ and $\dot{m}_i x_2$, and they have an effect on the energy balance through terms h_{k1} and h_{k2}.

Correspondingly, the humidity or water balance is

$$\dot{m}_i(x_2 - x_1) = (\dot{m}_{j1} - \dot{m}_{j2}) + (\dot{m}_{v1} - \dot{m}_{v2}) + (\dot{m}_{h1} - \dot{m}_{h2}) \tag{4.140}$$

In many cases the incoming and outgoing airflows can consist of various airflows in different states (temperature, humidity), which must be treated separately. This means that the air enthalpy flow must be divided into corresponding parts.

Example 8

Mixing of two airflows. As illustrated in Fig. 4.15A, the energy balance is

$$\dot{m}_{i1} h_{k1} + \dot{m}_{i2} h_{k2} = \dot{m}_{i3} h_{k3} \tag{4.141}$$

and the humidity balance is

$$\dot{m}_{i1} x_1 + \dot{m}_{i2} x_2 = \dot{m}_{i3} x_3 \tag{4.142}$$

The dry air balance is

$$\dot{m}_{i1} + \dot{m}_{i2} = \dot{m}_{i3} \tag{4.143}$$

From Eqs. (4.141)–(4.143), it follows that

$$\frac{h_{k3} - h_{k2}}{h_{k1} - h_{k3}} = \frac{x_3 - x_2}{x_1 - x_3} = \frac{\dot{m}_{i1}}{\dot{m}_{i2}}, \tag{4.144}$$

which shows that the mixing point is on the line connecting points 1 and 2 in Fig. 4.15B.

Example 9

Heating of an airflow. From Eq. (4.139), it follows that

$$\varphi = \dot{m}_{i1}(h_{k2} - h_{k1}) \tag{4.145}$$

and from Eq. (4.140),

$$\dot{m}_i(x_2 - x_1) = 0,$$

so $x_2 = x_1$. When air is heated, the state point moves up along the constant humidity line.

Example 10

Cooling of an airflow. From Eq. (4.139), it follows that

$$\varphi = \dot{m}_i(h_{k2} - h_{k1}) + \dot{m}_{v2} h_{v2}, \tag{4.146}$$

where now $\phi < 0$. When air is cooled, some water can be condensed. This depends on the surface temperature of the cooling coil, and therefore we have the term $\dot{m}_{v2}h_{v2}$ in the previous equation. From Eq. (4.140), we obtain the water balance

$$\dot{m}_i(x_2 - x_1) = -\dot{m}_{v2},$$

so the final humidity $x_2 \leq x_1$. The air cooling process is illustrated in Fig. 4.16.

When the airflow meets a surface whose temperature is lower than the dew point, water vapor from the air condenses on the surface of the cooling coil. If all air comes into contact with the cold surface, the state of the air after the process will be at point 3. Some air always escapes the cold surface, and therefore the state of air after contact with the coil is a mixture of saturated air (3) and escaped air (1). The mixing point 2 lies on the line connecting points 1 and 3, as shown in Example 8. The nearer point 2 is to point 3, the more effective is the cooling coil.

Example 11

Adding steam to the air. From Eqs. (4.139) and (4.140), it follows that

$$\dot{m}_h h_h = \dot{m}_i(h_{k2} - h_{k1})$$
$$\dot{m}_h = \dot{m}_i(x_2 - x_1),$$

where $\dot{m}_h$ is the added steam flow and h_k its enthalpy. From these equations, it follows that

$$\frac{h_{k2} - h_{k1}}{x_2 - x_1} = h_h. \tag{4.147}$$

In differential form, Eq. (4.147) is

$$\frac{dh_k}{dx} = h_h. \tag{4.148}$$

On the other hand, differentiating Eq. (4.90b), $h_k = c_{pi}\theta + x(c_{ph}\theta + l_{ho})$, with respect to the variables θ and x, we obtain

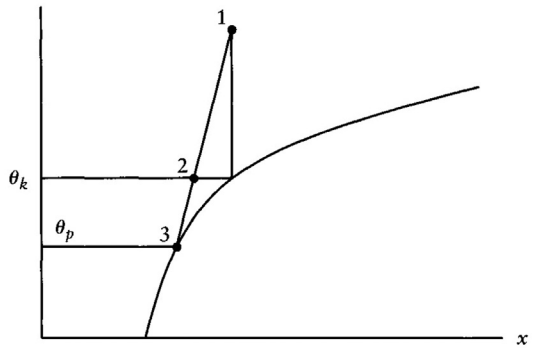


FIGURE 4.16 Air state change in a cooling coil. If the surface temperature θ_p is under the dew point θ_k, there will be condensation. If $\theta_p > \theta_k$ cooling takes place along the constant humidity line $x_1 = x_2$.

$$dh_k = (c_{pi} + xc_{ph})d\theta + (\theta c_{ph} + l_{ho})dx,$$

and substituting this in Eq. (4.148), we get

$$\frac{d\theta}{dx} = \frac{h_h - (l_{ho} + \theta c_{ph})}{c_{pi} + xc_{ph}} = \frac{h_h - h_h(\theta)}{c_{pi} + xc_{ph}}, \tag{4.149}$$

where $h_h(\theta) = l_{ho} + c_{ph}\theta$ is the steam enthalpy at air temperature θ.

From Eq. (4.149), we notice that if the temperature of the steam added to air is below the temperature of the air, the air will cool down and $d\theta/dx < 0$. If the steam temperature is higher than the air temperature, the temperature of air will rise ($(d\theta)/dx > 0$).

Example 12

The room temperature is required to be 20°C and the relative humidity $\varphi = 50\%$. The net heat load developing in the room is 2.45 kW and the net steam flow 1.5×10^{-3} kg/s. What should the inlet air temperature and humidity be when the inlet air is (1) $\dot{m}_i = 0.3$ kg/s and (2) $\dot{m}_i = 0.6$ kg/s

$$p'_h(20^\circ C) = 0.02337\ p_h = 0.5 \cdot 0.02337\ \text{bar} = 0.01169\ \text{bar},$$
$$\text{Room air humidity } x = 0.6220 \cdot \frac{0.01169}{1.0 - 0.01169} = 0.00736.$$

When the inlet air is well mixed with the room air, the humidity of the outlet air (2) is the same as the room air humidity, so $x_2 = 0.00736$, and its temperature is the same as the temperature of the room air, $\theta = 20^\circ$C.

The enthalpy of the outlet air $h_{k2} = 1.006 \times 20 + 0.00736 \times 2501 + 1.85 \times 20 = 38.8$ kJ/kg. The enthalpy of inlet air h_{k1} and its humidity x_1 are determined by the energy and humidity balances

$$\dot{m}_i(h_{k2} - h_{k1}) = \varphi + \dot{m}_h h_h$$
$$\dot{m}_i(x_2 - x_1) = \dot{m}_h$$

The net heat power also includes the enthalpy flow, $\dot{m}_h h_h$. Then

$$\varphi + \dot{m}_h h_h = 2.45\ \text{kW}$$

and therefore

1. $h_{k2} - h_{k1} = \dfrac{2.45}{0.3} = 8.2\ \text{kJ/kg} \rightarrow h_{k1} = 38.8 - 8.2 = 30.6\ \text{kJ/kg}$

2. $h_{k2} - h_{k1} = \dfrac{2.45}{0.6} = 4.1\ \text{kJ/kg} \rightarrow h_{k1} = 38.8 - 4.1 = 34.7\ \text{kJ/kg}$

The arising steam flow is $\dot{m}_h = 1.53 \times 10^{-3}$ kg/s, and thus

1. $x_2 - x_1 = \dfrac{1.53 \cdot 10^{-3}}{0.3} = 0.0051$

$x_1 = 0.00736 - 0.00255 = 0.00481$

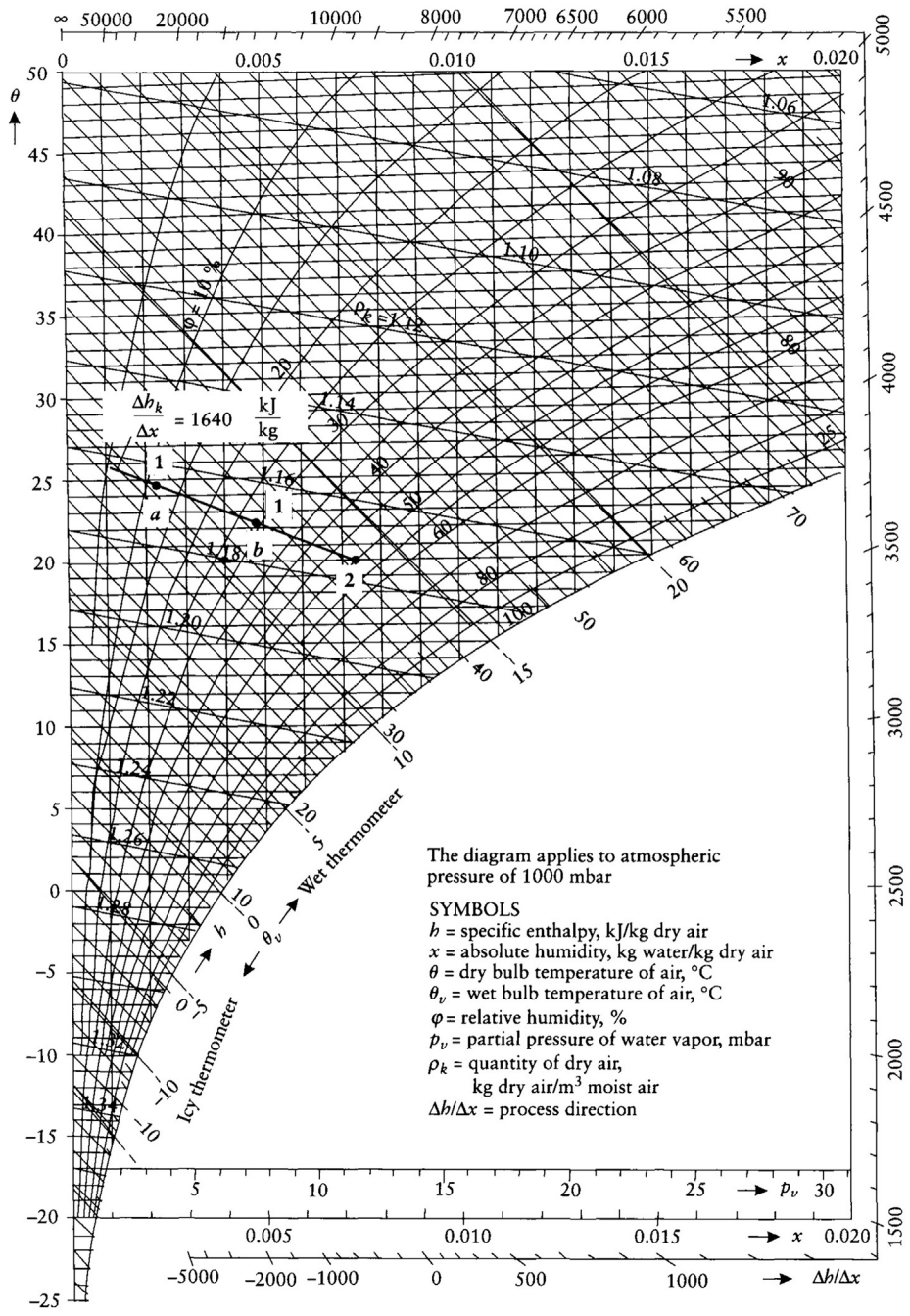


FIGURE 4.17 The state determination line for inlet air.

2. $x_2 - x_1 = \dfrac{1.53 \cdot 10^{-3}}{0.6} = 0.00255$

$x_1 = 0.00736 - 0.00255 = 0.00481$

The corresponding air temperatures can be calculated with the equation

$$\theta = \frac{h_k - x l_{ho}}{c_{pi} + x c_{ph}}.$$

$x = 0.00226 \quad h_k = 30.6$ kJ/kg

1. $\theta = \dfrac{30.6 - 0.00226 \cdot 2501}{1.006 + 0.00226 \cdot 1.85} = 24.4°\text{C}$

2. $x = 0.00481$ $h_k = 34.7$ kJ/kg $\theta = 22.2°\text{C}$

The result of Example 12 is illustrated in Fig. 4.17.

4.2.8 Example of cooling tower dimensioning

Cooling towers are commonly used for water cooling, but they can also be used for heat recovery from outlet air. If the water temperature is higher than the dew point of the air, water will cool in the tower. Cooling is caused by vaporization on the

surface of the waterdrops. The vaporization energy comes from the inner energy of the water and in a certain phase, when the water temperature is lower than the dry-bulb temperature of the air, also from the airflow. When the water temperature drops to near the air wet-bulb temperature at the observation point, water will not cool further even though there is still water vaporization from the drop surface. This is due to the fact that the temperature difference between the air dry-bulb temperature and drop surface is so large that the energy needed for vaporization comes convectively from the air. This is illustrated in Fig. 4.18.

If the air dew point is higher than the water temperature (or more accurately, the surface temperature of the drops), water vapor condenses from the air on the surface of the waterdrops. Now the water warms up and the air cools down and at the same time dries up; in other words, the cooling tower recovers heat from the outlet air. We will now consider the operation of a cooling tower more closely with the notations of Fig. 4.19.

The energy balance for a distance dL is

$$-\dot{m}_v c_{pv}\, d\theta_v = \dot{m}''_h\, l(\theta_v) dA - \alpha(\theta - \theta_v) dA.$$

From Eqs. (4.112), (4.113), and (4.121), we have, when $\mathrm{Le} \cong 1$, for vaporization $\dot{m}''_h = (\alpha/c_p)(x'(\theta_v) - x)$, and substituting this in the previous equation, we get

$$-\dot{m}_v c_{pv} \frac{c_p}{\alpha} d\theta_v = -\left[c_p(\theta - \theta_v) + (x - x'(\theta_v))l(\theta_v)\right] dA.$$

On the other hand, while (Eq. 4.119) $c_p = (\rho_i/\rho)(c_{pi} + x c_{ph}) \cong c_{pi} + x c_{ph}$ and $l_0 \cong l(\theta_v)$, we obtain, according to the definition of h_k (Eq. 4.90b), an approximate value of

$$c_p(\theta - \theta_v) + (x - x'(\theta_v))l(\theta_v) = h_k - h'_k(\theta_v),$$

and thus we have

$$d\varphi = -\dot{m}_v c_{pv} d\theta_v = \frac{\alpha}{c_p}(h'_k(\theta_v) - h_k) dA. \tag{4.150}$$

Alternatively, we can write the energy balance with the help of the enthalpy flows as

$$-\dot{m}_i dh_k = \dot{m}_v c_{pv}\, dt_v - d\dot{m}_v c_{pv} t_v,$$

where $dh_v = \dot{m}_i\, dx$, where dx is the humidity change of the airflow. Then we can write

$$-\dot{m}_v c_{pv} d\theta_v = -\dot{m}_i (dh_k - c_{pv}\theta_v\, dx).$$

While $dh_k = (c_{pi} + c_{ph})d\theta + l_0\ dx$ and $c_{pv}\theta_v \ll l_0$, to a good accuracy $dh_k - c_{pv}\theta_v\, dx \cong dh_k$ and therefore

$$d\varphi = -m_v c_{pv}\, d\theta_v = -m_i\, dh_k$$

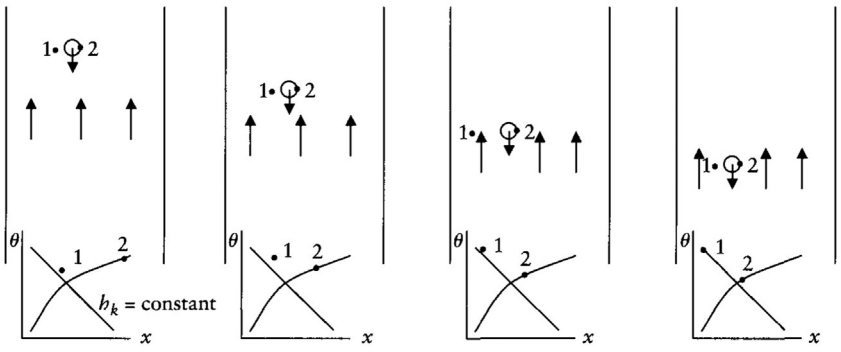


FIGURE 4.18 Waterdrop cooling in the airflow. Point 1 represents the air state surrounding the waterdrop (2).

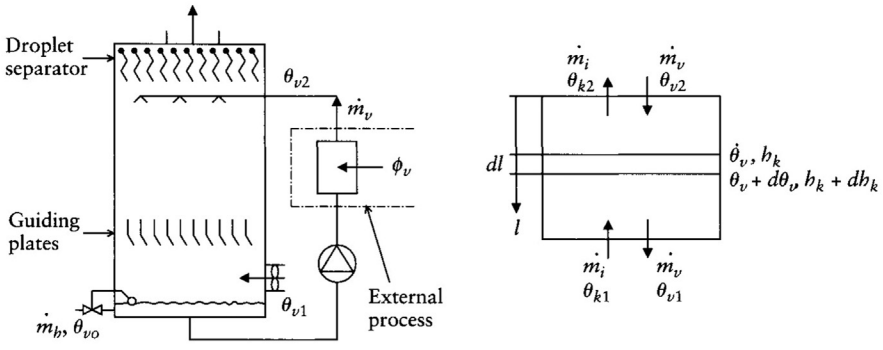


FIGURE 4.19 Mathematical approach for a cooling tower.

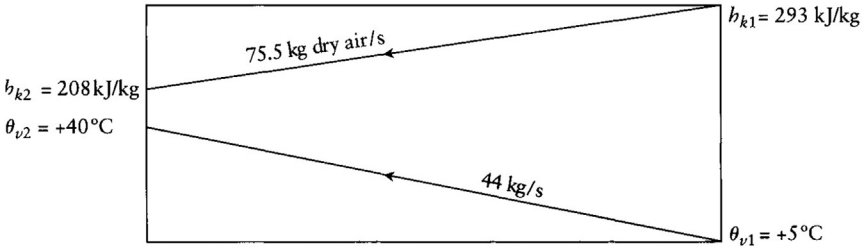


FIGURE 4.20 Heat transfer according to the parallel-flow principle.

Considering Eq. (4.150), we have

$$\dot{m}_i\, dh_k = \frac{\alpha}{c_p}(h_k - h'_k(\theta_v))dA. \tag{4.151}$$

From Eq. (4.151), we see that in a dampening process the state of air tends to change toward the saturated air state corresponding to the surface temperature of the water. We will now solve Eq. (4.151) approximately. First, we state that

$$dh_k = d(h_k - h'_k(\theta_v)) + dh'_k(\theta_v) = d(h_k - h'_k(\theta_v)) + \frac{dh'_k(\theta_v)}{dh_k}dh_k$$

Using the approximation

$$\frac{dh'_k(\theta_v)}{dh_k} = \frac{h'_k(\theta_{v1}) - h'_k(\theta_{v2})}{h_{k1} - h_{k2}}$$

we get

$$dh_k = \frac{d(h_k - h'_k(\theta_v))}{1 - \left(\left(h'_k(\theta_{v1}) - h'_k(\theta_{v2})\right)/(h_{k1} - h_{k2})\right)} = \frac{(h_{k1} - h_{k2})d(h_k - h'_k(\theta_v))}{(h_{k1} - h'_k(\theta_{v1})) - (h_{k2} - h'_k(\theta_{v2}))},$$

and substituting this in Eq. (4.151), we have

$$\frac{d(h_k - h'_k(\theta_v))}{h_k - h'_k(\theta_v))} = \frac{\alpha}{c_p m_i(h_{k1} - h_{k2})} \cdot \left[(h_{k1} - h'_k(\theta_{v1})) - (h_{k2} - h'_k(\theta_{v2}))\right] \cdot dA.$$

Integrating the previous equation, we obtain

$$\dot{m}_i(h_{k1} - h_{k2}) = \frac{\alpha A}{c_p}\Delta h_{k\,\ln}, \tag{4.152}$$

where the definition $\Delta h_{k\,\ln}$ is

$$\Delta h_{k\,\ln} = \frac{(h_{k1} - h'_k(\theta_{v1})) - (h_{k2} - h'_k(\theta_{v2}))}{\ln\left(\left(h_{k1} - h'_k(\theta_{v1})\right)/\left(h_{k2} - h'_k(\theta_{v2})\right)\right)}$$

and

$$A = A_k A_p L$$

The quantity αA_p is defined separately for each type of cooling tower. It depends on many variables: jet pressure, jet division, airflow velocity, and others. The total energy balance for a cooling tower is (see Fig. 4.19)

$$\dot{m}_i(h_{k2} - h_{k1}) = \dot{m}_h c_{pv}\theta_{vo} + \dot{m}_h c_{pv}(\theta_{v2} - \theta_{v1}),$$

where θ_{vo} is the temperature of the excess feedwater. The need for excess feedwater represents the rate of vaporization $\dot{m}_h$. Usually, the term $\dot{m}_h c_{pv}\theta_{vo}$ has minor significance (vaporization rate $\dot{m}_h$ corresponds usually to just a small percentage of the water flow $\dot{m}_v$), so on the basis of Eq. (4.152), we get an equation for the cooling power ϕ_v

$$\varphi_v = \dot{m}_v c_{pv}(\theta_{v2} - \theta_{v1}) = \frac{\alpha A}{c_p}\Delta h_{k\,\ln} \tag{4.153}$$

We can dimension a cooling tower according to the previous equation.

Example 13

A paper industry's cooling tower recovers heat from the outlet air. This situation is represented by the following values:

- inlet air enthalpy $h_{k1} = 293$ kJ/kg
- outlet air enthalpy $h_{k2} = 208$ kJ/kg (saturated 44.3°C air)
- outlet water temperature $\theta_{v1} = 40°C$
- inlet water temperature $\theta_{v2} = 5.0°C$
- water flow $\dot{m}_v = 44$ kg/s
- airflow $\dot{m}_i = 44 \times 4.186 \times (40 - 5)/(293 - 208) = 75.8$ kg/s
- cross-sectional area of the cooling tower $A_k = 31$ m^2 and height $L = 3$ m

It is discovered that in the cooling tower the water moving downward from the jets changes its direction to upward after drop formation. There is an effective heat transfer process when the drops move upward: heat transfers from the outlet air to the drops through convection and condensation.

Drops collide with the drop separator and drain down to the lower part of the tower. These drops are large, so their total surface area is small and insignificant. The effective heat transfer process takes place when the drops move with the airflow, so this arrangement has to be treated as a parallel-flow heat transfer.

1. Calculate αAp. According to the parallel-flow principle, the situation is as shown in Fig. 4.20.

First, we calculate the logarithmic enthalpy difference

$$h'_k(\theta_{v1}) = h'_k(5^\circ C) = 18 \text{ kJ/kg}$$
$$h'_k(\theta_{v2}) = h'_k(40^\circ C) = 168 \text{ kJ/kg}$$

and

$$\Delta k_{\text{kln}} = \frac{(293-18)-(208-168)}{\ln\big((293-18)/(208-168)\big)} = 121.9 \text{ kJ/kg}$$

The specific heat capacity of humid air calculated per kilogram of dry air is

$$c_{p\,K} = c_{pi} + x c_{ph} = 1.006 + 08 \cdot 1.85 = 1.154 \text{ kJ/(kg }^\circ C)$$
$$c_p = \frac{\rho_i}{\rho} c_{p\,K} = 1.154 \big[\text{kJ/(kg }^\circ C)\big]$$

From Eq. (4.153),

$$\alpha A = \frac{c_p}{\Delta h_{\text{kln}}} \dot{m}_v c_{pv}(t_{v1} - t_{v2}) = -\frac{1.154}{121.9}.44.0 \cdot 4.186 \cdot (5-40)$$
$$= 61.0 \text{ kW/}^\circ\text{C}$$

On the other hand, $AA_p\ A_k\ L$, and therefore

$$\alpha A p = \frac{61.0 \text{ kW/}^\circ\text{C}}{A_k - L} = \frac{61.0}{31.3} = 0.656 \text{ kW/}^\circ\text{C m}^3$$

2. The same cooling tower is to function as a water cooler. Let the outdoor air be 24°C, $\varphi = 50\%$, and the airflow 100 kg/s (dry air). The water inlet temperature is 24°C and the water flow 30 kg/s. What is the cooling capacity if we assume that $\alpha - \sqrt{v_i}$ and $A_p \sim \dot{m}_v$, or $\alpha A_p = k\sqrt{v_i}\dot{m}_v$ and also that the active cooling process is parallel-flow heat transfer?

For case (1) was 44 kg/s and the airflow velocity = (75.5/1.0)/31 = 2.44 m/s. Therefore the heating capacity (or cooling capacity, depending on the sign) can be presented as

$$\varphi = \frac{\alpha A_p A_k L}{c_p} \Delta h_{\text{kln}} = \frac{k\sqrt{v_i}\dot{m}_v}{c_p} A_k L \Delta h_{\text{kln}},$$

and solving factor k out of this equation,

$$k = \frac{c_p \varphi}{\sqrt{v_i}\dot{m}_v A_k L \Delta h_{\text{kln}}} = \frac{1.154 \cdot (44.0 \cdot 4.186 \cdot (40-5))}{\sqrt{2.44} \cdot 44.0 \cdot 31.0 \cdot 3 \cdot 121.9}$$
$$= 9.55 \cdot 10^{-3}.$$

Then the capacity ϕ is

$$\varphi = \frac{k}{c_p} \cdot A_k L \cdot \sqrt{v_i}\dot{m}_v \Delta h_{\text{kln}} = \frac{9.55 \cdot 10^{-3}}{1.154} \cdot 31.3 \cdot \sqrt{v_i}\dot{m}_v \Delta h_{\text{kln}}$$

and after calculations,

$$\varphi = 0.770 \cdot \sqrt{v_i} \cdot m_v \cdot \Delta h_{\text{kln}}$$

where v_i is in m/s, $\dot{m}_v$ in kg/s, Δh_{kln} in kJ/kg, and ϕ is in kW. It was found that $\dot{m}_v = 30$ kg/s and $\dot{m}_i = 100$ kg/s, and now $v_i \cong (100/1.2)/31 = 2.69$ m/s. Thus

$$\varphi = 0.770 \cdot \sqrt{2.69} \cdot 30.0 \cdot \Delta h_{\text{kln}} = 37.9 \cdot \Delta h_{\text{kln}}$$

The cooling tower functioning in an air cooling situation is illustrated in Fig. 4.21.

Because of the logarithmic enthalpy difference, the solution must be iterated:

1. Guess: $\theta_{v2} = 20^\circ C$; $h'_k(20^\circ C) = 57.9$ kJ/kg

$$\varphi = 30 \cdot 4.186 \cdot (24-20) = 502 \text{ kW}$$
$$h_{k2} - h_{k1} = 502/100 = 5.02 \text{ kJ/kg};\ h_{k2} = 53.3 \text{ kJ/kg}$$
$$\Delta h_{\text{k ln}} = \frac{24.1 - (57.9 - 53.3)}{\ln\big(24.1/(57.9-53.3)\big)} = 11.8 \text{ kJ/kg}$$
$$\varphi = 37.9 \cdot 11.8 = 446 \text{ kW}$$
$$k_{k2} = 24 - 446/30 \cdot 4.186 = 20.4^\circ C$$

2. We choose $\theta_{v2} = (20.0 + 20.4)/2 = 20.2^\circ C$; $h'_k(20.2^\circ C) = 58.6$ kJ/kg.

$$\varphi = 30 \cdot 4.186 \cdot (24 - 20.2) = 477 \text{ kW}$$
$$_{hk2} = 48.3 + 477/100 = 53.1 \text{ kJ/kg};$$
$$\Delta h_{\text{k ln}} = \frac{24.1 - (58.6 - 53.1)}{\ln\big(24.1/(58.6-53.1)\big)} = 12.6 \text{ kJ/kg}$$
$$\varphi = 37.9 \cdot 12.6 = 477 \text{ kW}$$

Now $\theta_{v2} = 20.2^\circ C$ and cooling capacity $\phi = 477$ kW.

When the cooling tower was operating as a heat recovery device, its capacity was considerably higher because of the high temperatures and humidities. In case (1), we had

$$\phi = 44 \cdot (40-5) \cdot 4.186 = 6450 \text{ kW}.$$

If we assume that the outlet air is saturated, the air state change process is as presented in Fig. 4.22. The

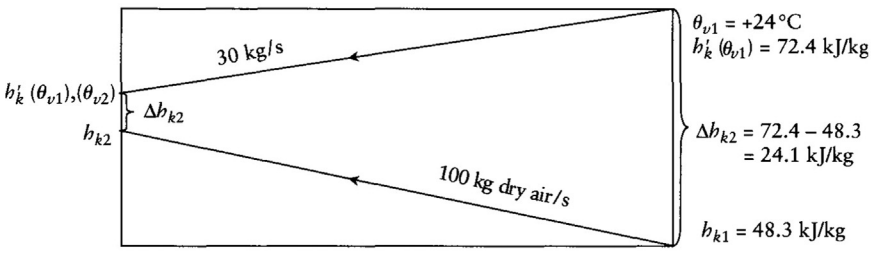


FIGURE 4.21 Water cooling in a cooling tower, which operates according to the parallel-flow principle; in other words, the water spray turns in the direction of the airflow.

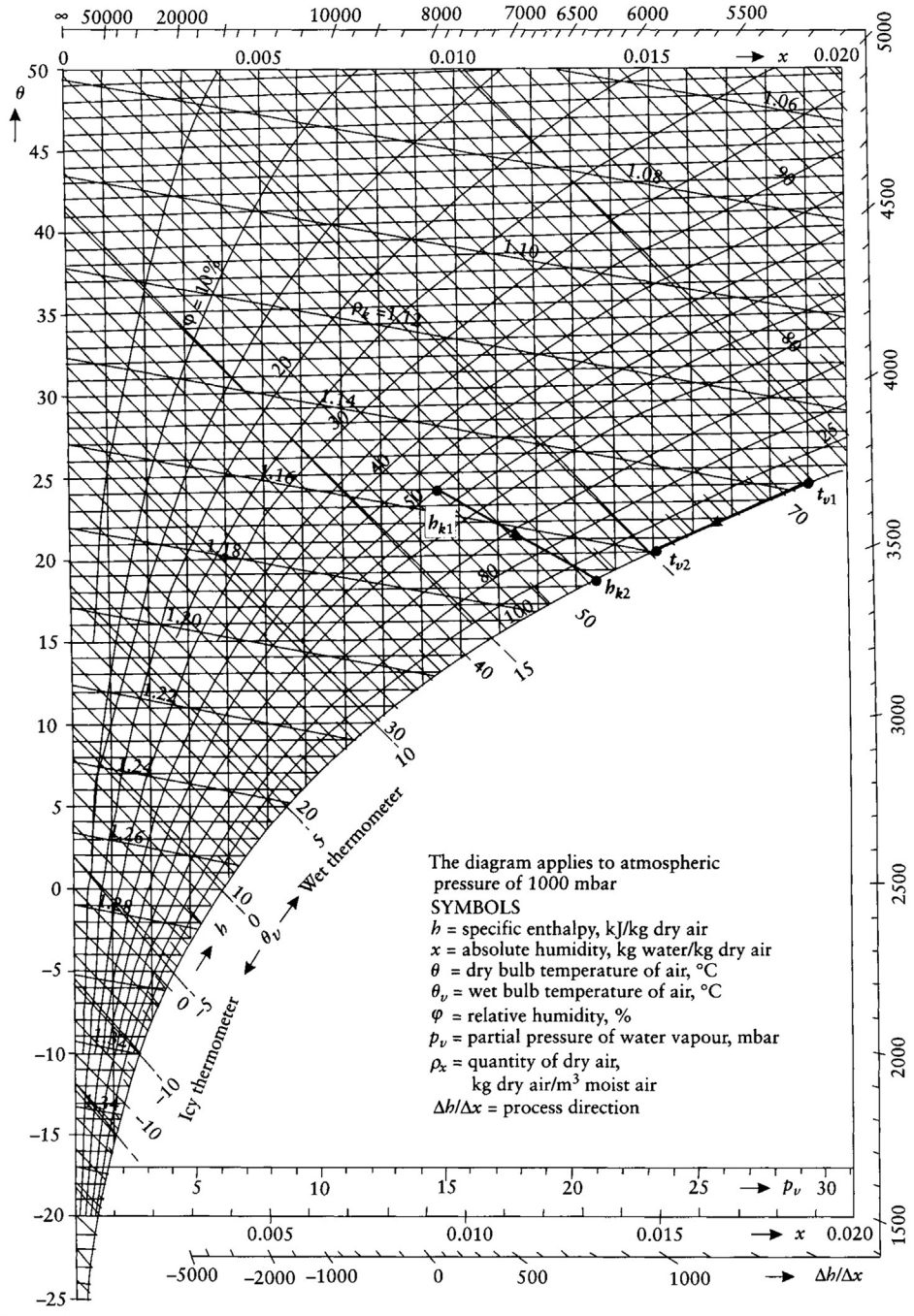


FIGURE 4.22 Water cooling in a cooling tower.

exact determination of the air humidity at the end of the process would demand separate mass and heat transfer examinations.

4.3 Heat and mass transfer

This section introduces the important subject of heat and mass transfer to serve as a reference work to both the beginning engineer and the practicing industrial ventilation engineer.

4.3.1 Different forms of heat transfer

For any method of heat transfer to take place, a temperature difference is necessary between two faces of a solid body or at the boundaries of a gas or vapor. Heat transfer will take place only from a high-temperature

source to a lower temperature sink and is an irreversible process unless acted upon by another agency, as is the case with the refrigeration process.

Heat transfer may occur by one or more of three different modes:

- conduction, which may be one-, two-, or three-dimensional or internally generated;
- convection, which may be natural or forced; and
- radiation, which may be symmetrical or nonsymmetrical.

4.3.1.1 Conduction

Conduction takes place at a solid, liquid, or vapor boundary through the collisions of molecules, without mass transfer taking place. The process of heat conduction is analogous to that of electrical conduction, and similar concepts and calculation methods apply. The thermal conductivity of matter is a physical property and is its ability to conduct heat. Thermal conduction is a function of both the temperature and the properties of the material. The system is often considered as being homogeneous, and the thermal conductivity is considered constant.

Thermal conductivity, λ, W/(m °C), is defined using Fourier's law,

$$q_x = -\lambda \frac{\partial T}{\partial x} = \frac{\Phi_x}{A}, \tag{4.154}$$

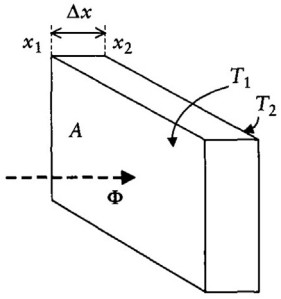


FIGURE 4.23 One-dimensional heat flow.

where $q = \Phi/A = \Phi''$ is the heat flow (W/m^2), Φ_x is the heat flow in the x direction, and $\partial T/\partial x$ is the temperature gradient.

The minus sign in the equation denotes that the heat flow is positive in the direction of decreasing temperature.

Fig. 4.23 represents a simple one-dimensional system with constant heat flow Φ through the plate. The plate thickness is Δx (m), and the area of the plate is A (m^2).

Integration of Eq. (4.154) with constant heat conductivity gives

$$\Phi \int_{x_1}^{x_2} dx = -\lambda A \int_{T_1}^{T_2} dT$$

$$\Phi = -\lambda A \frac{\Delta T}{\Delta x}. \tag{4.155}$$

One-dimensional heat conduction means that the heat flow is in one direction only, and one coordinate is required to represent the case. For example, in the case of a cylinder, it is parallel with the radius.

4.3.1.2 Convection

Convection occurs in a moving fluid, generally from the fluid to a solid surface or vice versa. Although heat transfer between single particles is by conduction, it is the energy transfer with the matter that governs the heat transfer. The basic laws of heat and mass transfer have to be considered in order to describe convection mathematically.

Natural convection is self-induced and is created by the density differences that are temperature related; the boiling of water in a kettle is an example of free convection. Forced convection is caused by an external force being applied by mechanical means such as a fan or pump; the cooling of a warm bottle in cool flowing water is an example of forced convection.

Convection is influenced by the fluid flow adjacent to the solid surface. To appreciate the mechanics of this mode of heat transfer, the nature of the fluid flow in relation to the particular flow process must be known. Consideration of the flow structure created by the passage of a turbulent fluid over a smooth solid surface shows (see Fig. 4.24) the following:

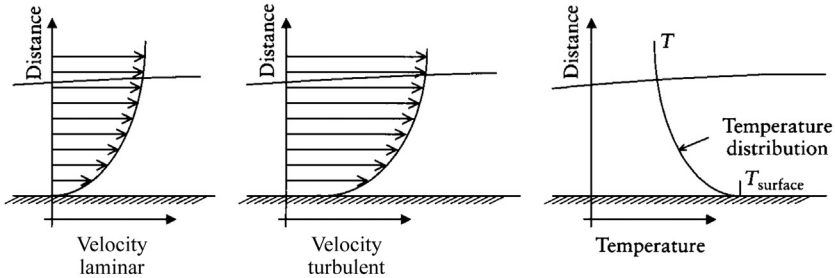


FIGURE 4.24 Laminar and turbulent boundary layers and temperature distribution inside the boundary layer.

1. The determining factor in convection is the flow boundary layer. Outside the boundary layer, the fluid is considered to have achieved a maximum velocity at an infinite distance from the surface.
2. In a laminar boundary layer, no mixing takes place and the flow is parallel. In this case the heat transfer occurs mainly by conduction through the boundary layer.
3. In a turbulent boundary layer, flow takes place in the direction perpendicular to the surface over which the flow occurs.

A heat transfer factor (α) between the fluid and surface is defined as

$$q = \alpha\,\Theta = \frac{\Phi}{A}\Phi'', \tag{4.156}$$

where is the temperature difference between the surface and the fluid at a long distance from the surface.

When heat transfer occurs by conduction through the boundary layer,

$$\alpha \sim \frac{\lambda}{\delta}, \tag{4.157}$$

where δ is the thickness of the boundary layer, and the unit of α is W/(m^2 K). The heat transfer factor α thus decreases as the boundary layer thickness increases. The following discussion gives some indication of the range of the heat transfer values obtained due to the different modes of convective heat transfer.

Next we give some values of α to give an idea of the magnitude of the heat transfer:

	α [W/(m^2 K)]
Free convection	3.5–50
Forced convection, air	10–500
Forced convection, liquid	100–5000

A liquid has a higher rate of conductivity than a gas.

In boiling convection, liquid motion is created by steam bubbles breaking loose from the surface.

If steam condenses on a surface, there is no boundary layer; the resistance to heat flow is due to scale, metal thickness, and the condensed liquid layer, resulting in a high heat transfer factor. A thin layer of air or other noncondensing gas forms at the surface through which the steam diffuses. The heat transfer factor diminishes rapidly but is considerably higher than in dry convection.

4.3.1.3 Radiation

Heat radiation is electromagnetic radiation that all bodies emit due to their temperature. The wavelength of electromagnetic radiation is between 0.3 and 50 μm. This mode of heat transfer does not depend on an intermediate agent. When radiation falls on a body, part of the energy is absorbed, part is reflected, and the remainder is transmitted through the body. These components of the incoming radiation are the absorption ratio α, reflectance ratio ρ, and transmission ratio τ. When a body is in a state of equilibrium, the incoming and outgoing radiation are equal. Hence, $\alpha + \rho + \tau = 1$.

A body having good electrical conductivity will absorb the incoming radiation on a distance of one wavelength. Now $\tau = 0$ and $\alpha + \rho = 1$.

A planar polished surface reflects heat radiation in a similar manner with which it reflects light. Rough surfaces reflect energy in a diffuse manner; hence radiation is reflected in all directions. A blackbody absorbs all incoming radiation and therefore has no reflection. A perfect blackbody does not exist; a near-perfect blackbody surface such as soot reflects 5% of the radiation, making it the standard for an ideal radiator.

The radiant emittance of a blackbody is

$$M_m = \sigma T^4, \tag{4.158}$$

where σ is the Stefan–Boltzmann constant, 11.865 W/m^2/(MJ/kmol)^4.

The radiation emitted by a body due to its temperature is defined by the factor ϵ, the emissivity,

$$\epsilon M_m = \epsilon\sigma T^4. \tag{4.159}$$

This leads to Kirchhoff's law,

$$\epsilon(T,\ \vartheta,\ v) = \alpha(T,\ \vartheta,\ v), \tag{4.160}$$

for a given temperature, angle, and frequency. For approximate calculations the emissivity can be assumed constant over the whole frequency spectrum. In this case the body is classified as a gray body.

The net heat transfer between two surfaces according to Eq. (4.159) is proportional to the first or second power of the temperature difference; hence the radiation heat transfer dominates at a high temperature or for large temperature differences. When the temperature difference is small, a heat transfer factor is used similar to that used for convective heat transfer

$$q = \alpha_s \Delta T. \tag{4.161}$$

4.3.2 Analogy with the theory of electricity

Eq. (4.154) gives conduction for the one-dimensional case with constant thermal conductivity

$$\Phi = \frac{\lambda A}{\delta}\Theta, \tag{4.162}$$

where δ is the distance corresponding to the temperature difference. For the following three heat transfer forms,

- conduction, Eq. (4.162)
- convection, Eq. (4.156)
- radiation, Eq. (4.161)

we have, respectively,

$$\Theta = \frac{\delta}{\lambda A}\Phi \tag{4.163}$$

$$\Theta = \frac{1}{\alpha_k A}\Phi \tag{4.164}$$

$$\Theta = \frac{1}{\alpha_s A}\Phi. \tag{4.165}$$

Following from Ohm's electrical law (theory of electricity), a heat resistance can be defined:

potential difference = resistance × current
temperature difference = heat resistance × heat flow

$$\Theta = R \cdot \Phi \tag{4.166}$$

The conductance or the coefficient of heat transfer $U = 1/R$ or

$$\Phi = U\Theta \cdot \tag{4.167}$$

For conduction the heat resistance is the distance divided by the heat conductivity, $R = \delta/\lambda A$, and the heat conductance is heat conductivity divided by distance, $U = \lambda A/\delta$. For convection and radiation the heat resistance is 1 divided by the heat transfer factor, $1/\alpha A$, and the heat conductance is the same as the heat transfer factor, $U = \alpha A$. A coefficient of heat flow is also used, the K value, which is the total conductance

$$K = \frac{U}{A}\left[\frac{\mathrm{W}}{\mathrm{m^2\,K}}\right] \tag{4.168}$$

The following connecting rules are based on the abovementioned analogy: heat resistance R in series connection

$$R = R_1 + R_2$$

and in parallel connection

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2}$$

heat conductance series connection

$$\frac{1}{U} = \frac{1}{U_1} + \frac{1}{U_2}$$

and in parallel connection

$$U = U_1 + U_2$$

The heat resistance between the fluids on the two sides of the pipe wall in Fig. 4.25 is

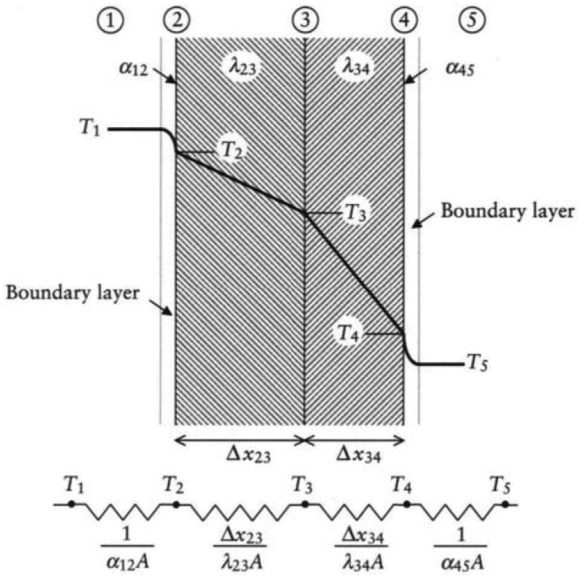


FIGURE 4.25 Heat transfer through a wall.

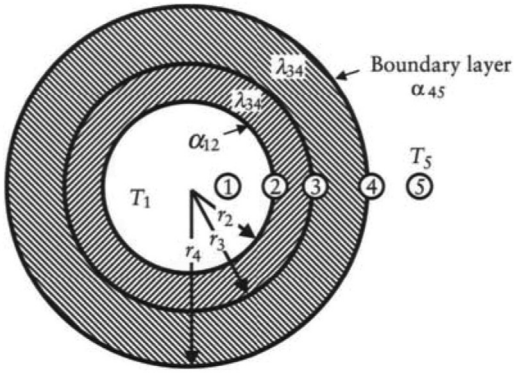


FIGURE 4.26 Heat flow through a pipe wall.

$$R \cdot A = \frac{A}{U} = \frac{1}{K} = \frac{1}{\alpha_1} + \frac{1}{\alpha_2} + \sum \frac{\delta}{\lambda}, \tag{4.169}$$

where $\alpha = \alpha_{\mathrm{conv}} + \alpha_{\mathrm{rad}}$, and δ and λ are the thickness and heat conductivity of consecutive layers.

The resistance between the fluids on the inside and outside of the pipe is obtained by integrating with respect to the radius (Fig. 4.26)

$$R \cdot 1 = \frac{1}{U} = \frac{1}{\pi d_u \alpha_u} + \frac{1}{\pi d_s \alpha_s} + \sum \frac{\ln\left(d_{ui}/d_{si}\right)}{2\pi\lambda_i} \tag{4.170}$$

The sum includes concentric cylinder layers, such as the layer between the outer and inner diameters of the pipe or a possible thermal insulation layer. For each layer the corresponding heat conductivity λ_I is used. The outer heat transfer factor is the sum of the proportions of convection and radiation. (*Note*: Very thin pipes or wires should not be insulated. Because the

outer diameter of the insulation is smaller than λ/α_u, the resistance is less than that without the insulation.)

The resistance between fluids separated by two coaxial spherical surfaces is

$$R = \frac{1}{\pi d_u^2 \alpha_u} + \frac{1}{\pi d_s^2 \alpha_s} + \frac{\delta}{(1/2\pi\lambda)\left((1/d_u) - (1/d_s)\right)}. \tag{4.171}$$

Example 1

Heat transfer through a pipe wall. A pipeline part 15 m long carries water. Its internal diameter d_i is 34 mm, and its external diameter d_o is 42 mm. The thermal conductivity of the pipe λ is 40 W/(m K). The pipeline is located outdoors, where the outdoor temperature θ_{ao} is $-8°C$. Determine the minimum flow velocity necessary in the pipe to prevent the pipe from freezing. The heat transfer coefficient inside the pipe is $\alpha_s = 1000$ W/(m^2 K) and outside the pipe $\alpha_{u,\ conv} = 5$ W/(m^2 K) and $\alpha_{u,\ rad} = 4$ W/(m^2 K). The specific heat capacity of water is $c_p = 4.2$ kJ/(kg K) and the initial temperature of water $\theta_i = 4°C$.

$$\alpha_u = \alpha_{u,\ conv} + \alpha_{u,\ rad} = (5+4)W/(m^2\ K) = 9\ W/(m^2\ K)$$

$$R' = \frac{1}{\pi d_u \alpha_u} + \frac{1}{\pi d_s \alpha_s} + \sum \frac{\ln^{d_u/d_s}}{2\pi\lambda_i}$$

$$= \frac{1}{\pi \cdot 0.042\ \mathrm{m} \cdot 9(W/m)^2\mathrm{K}} + \frac{1}{\pi \cdot 0.034\ \mathrm{m} \cdot 1(W/m)^2\ \mathrm{K}} + \frac{\ln\frac{42}{34}}{2\pi \cdot 40 W/\mathrm{m\ K}} = 0.852(1)\mathrm{m}^2\ \mathrm{K}/W$$

$$d\Phi = \dot{m} c_p\, d\Theta = -\frac{1}{R'}\Theta\, dx \Rightarrow \int_{\Theta_1}^{\Theta_2} \frac{d\Theta}{\Theta} = -\frac{1}{R'\dot{m}c_p}\int_0^1 dx$$

$$\Rightarrow \ln\frac{\Theta_2}{\Theta_1} = \frac{L}{R'\dot{m}c_p}$$

$$\dot{m} = \frac{L}{R' c_p \ln(\Theta_1/\Theta_2)}$$

$$w = \frac{q_v}{A} = \frac{\dot{m}/\rho}{\pi d_s^2/4} = \frac{4L}{R' c_p \ln(\Theta_1/\Theta_2)\rho\pi d_s^2}$$

$$= \frac{\mathbf{4} \cdot 15\mathrm{m}}{0.852 \cdot 4200 \cdot \ln(12/8) \cdot 1000 \cdot \pi \cdot 0.034^2}. = 0.011\ m/s$$

4.3.3 Heat conduction

The heat flow density q of a material depends on the local temperature gradient, according to Fourier's law

$$q = -\lambda\frac{\partial T}{\partial x}. \tag{4.172}$$

In simple one-dimensional cases, it is easy to determine the temperature gradient and calculate the heat flow from Fourier's law.

The general case is that of steady-state flow, and the thermal conductivity factor is a function of the temperature. In the unsteady state the temperature of the system changes with time, and energy is stored in the system or released from the system reduced. The storage capacity is

$$\frac{dU}{dt} = \Phi = mc_p\frac{\partial T}{\partial t} = \rho c_p V\frac{\partial T}{\partial t}. \tag{4.173}$$

4.3.3.1 General heat conduction equation

Consider a small control volume $V = \delta x\delta y\delta z$ (Fig. 4.27), where the inner heat generation is $Q'''_g(T)$ (heat production/volume), and the heat conductivity is $\lambda(T)$. The material is assumed to be homogeneous and isotropic, and the internal heat generation and thermal conductivity are functions of temperature.

The heat flow to the control volume through area $\delta y\delta z$ at x is

$$\delta Q_x = -\delta y\delta z\lambda(T)\left(\frac{\partial T}{\partial x}\right)\delta t. \tag{4.174}$$

The outgoing heat at the point $x + \delta x$ is

$$\delta Q_x = -\delta y\delta z\left[\lambda(T)\frac{\partial T}{\partial x} + \frac{\partial}{\partial x}\left(\lambda(T)\frac{\partial T}{\partial x}\delta x\right)\delta t\right]. \tag{4.175}$$

Similar formulas can be derived for the other directions. The change of internal energy inside the control volume during time δt is

$$\delta U = \rho c_p \delta x\delta y\delta z\frac{\partial T}{\partial t}\partial t \tag{4.176}$$

and the heat generation inside the control volume is

$$\delta Q_g = Q'''_g(T)\delta x\delta y\delta z\delta_t. \tag{4.177}$$

From the first law of thermodynamics,

$$\delta Q_x + \delta Q_y + \delta Q_z + \delta Q_g = \delta Q_{x+\delta x} + \delta Q_{y+\delta y} + \delta Q_{z+\delta z} + \delta U. \tag{4.178}$$

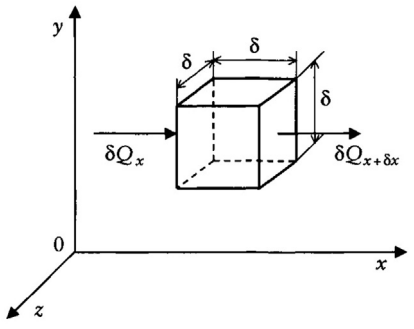


FIGURE 4.27 Control volume.

Substituting Eq. (4.175) and the formulas for other directions into Eq. (4.178) gives

$$\frac{\partial}{\partial x}\left(\lambda(T)\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(\lambda(T)\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(\lambda(T)\frac{\partial T}{\partial z}\right)+Q'''_g(T)=\rho c_p\frac{\partial T}{\partial t}. \tag{4.179}$$

It is normal to assume that the thermal conductivity is constant; hence Eq. (4.179) gives

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}+\frac{Q'''_g}{\lambda}=\frac{1}{a}\frac{\partial T}{\partial t}, \tag{4.180}$$

or

$$\frac{\partial T}{\partial t}=a\nabla^2 T+H, \tag{4.181}$$

where $\partial T/\partial t$ is the derivative of temperature as a function of time; in a steady-state case, it is equal to zero; $a=\lambda/\rho c_p$, the heat conductivity or thermal diffusivity; $H=\Phi'''/C'''=$ heat generation inside a material; for example, for Joule's heat or a nuclear reaction, $\Phi'''=$ heat generation/volume and $C'''=c_p\rho=$ heat capacity/volume.

For Cartesian coordinates

$$\nabla^2 T=\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2},$$

for cylindrical coordinates

$$\nabla^2 T=\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial^2 T}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T}{\partial \varphi^2}+\frac{\partial^2 T}{\partial z^2},$$

and for spherical coordinates

$$\nabla^2 T=\frac{1}{r}\frac{\partial^2 rT}{\partial r^2}+\frac{1}{r^2\sin\psi}\frac{\partial}{\partial\psi}\left(\sin\psi\frac{\partial T}{\partial\psi}\right)+\frac{1}{r^2\sin^2\psi}\frac{\partial^2 T}{\partial\varphi^2}.$$

See Fig. 4.28.

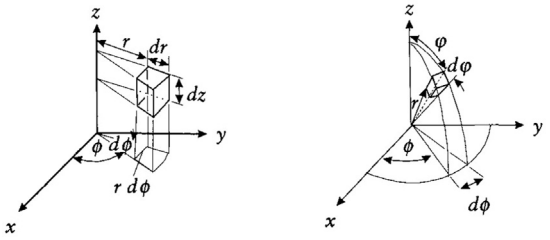


FIGURE 4.28 Cylindrical and spherical coordinates.

4.3.3.2 *One-dimensional steady-state heat conduction*

Infinite plate

A simple case of heat conduction is a plate of finite thickness but infinite in other directions. If the temperature is constant around the plate, the material is assumed to have a constant thermal conductivity. In this case the linear temperature distribution and the heat flow through the plate are easy to determine from Fourier's law (Eq. 4.154).

In a case similar to Fig. 4.23, the heat conduction equation (Eq. 4.180) becomes

$$\frac{\partial^2 T}{\partial x^2}=0. \tag{4.182}$$

In steady-state conditions the right side of Eq. (4.180) is zero, and no heat generation takes place; the thermal conductivity in the one-dimensional case is constant.

The solution of Eq. (4.182) is

$$T=C_1x+C_2, \tag{4.183}$$

with boundary conditions

$T=T_1$ when $x=x_1$
$T=T_2$ when $x=x_2$

This gives the linear temperature distribution

$$T=T_1+(T_2-T_1)\frac{x-x_1}{x_2-x_1} \tag{4.184}$$

Substituting the abovementioned equation into Eq. (4.154) gives the heat flow through the plate

$$q=-\lambda\frac{\partial \mathrm{T}}{\partial \mathrm{x}}=\lambda\frac{T_1-T_2}{x_2-x_1}=\lambda\frac{T_1-T_2}{\Delta x}. \tag{4.185}$$

Axial-symmetric case

For the axial-symmetric case the equation is

$$\nabla^2 T=\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{1}{r^2}\frac{\partial^2 T}{\partial\varphi^2}+\frac{\partial^2 T}{\partial z^2}. \tag{4.186}$$

In the one-dimensional case

$$\frac{d^2T}{dr^2}+\frac{1}{r}\frac{dT}{dr}=0. \tag{4.187}$$

The solution of Eq. (4.187) is

$$T=C_1\ln r+C_2, \tag{4.188}$$

with boundary conditions

$T=T_1$ when $r=r_1$
$T=T_2$ when $r=r_2$

The logarithmic temperature distribution is

$$T=T_1+(T_2-T_1)\frac{\ln(r/r_1)}{\ln(r_2/r_1)}. \tag{4.189}$$

Thus the heat flow per a unit of length Φ' is

$$\Phi' = -\lambda 2\pi r \frac{dT}{dr} = \lambda 2\pi \frac{T_1 - T_2}{\ln(r_2/r_1)}. \quad (4.190)$$

4.3.4 Heat convection

The mathematical principles of convective heat transfer are complex and outside the scope of this section. The problems are often so complicated that theoretical handling is difficult, and full use is made of empirical correlation formulas. These formulas often use different variables depending on the research methods. Inaccuracy in defining material characteristics, experimental errors, and geometric deviations produces noticeable deviations between correlation formulas and practice. Near the validity boundaries of the equations, or in certain unfavorable cases, the errors can be excessive.

The general forms of the convection equations are given next in a simple form. More accurate equations can be found from the latest research results presented in technical journals.

The general equation for the case of forced convection is Nu = *f*(Re, Pr). In the case of free convection, it is Nu = *f*(Gr, Pr).

Nu = Nusselt number = $\frac{\alpha L}{\lambda}$

Gr = Grashof number = $\frac{g\vartheta L^2}{v^2}$

$$\vartheta = \frac{\Delta v}{v_\infty} = \alpha_v \Theta \quad \Theta = T_p - T_\infty$$

where T_p is the surface temperature.

$$\alpha_v = \frac{1}{v}\left(\frac{\partial V}{\partial T}\right)_p$$

Pr = Prandtl number = Pr = Prandtl number = $\frac{v}{a}\frac{vC_p\rho}{\lambda}$

Re = Reynolds

number = Re = Reynolds number = $\frac{wL}{v}$

The characteristic length L denotes the pipe diameter or the hydraulic diameter $d_{hyd} = 4\ A/P$ (A is the cross-sectional area and P is the wet periphery). If the cross section is not circular, or in the case of a plane, the length is measured in the flow direction.

The temperature changes taking place through the surface of an exothermic body depend on the material characteristics and changes in the parameters. In formulas involving convection, either the solid surface temperature or the heat flow from the surface is assumed to be constant. The temperature θ defines the material characteristics (c, ρ, v, etc.). Normally, this temperature is the mixed temperature of the flowing fluid. The mean temperature of the boundary layer is the average temperature of the surface temperature and the undisturbed flow ($T_m = (T_p + T_\infty)/2$). Sometimes the boundary layer temperature, which is the average of the mixing temperature and the surface temperature, is used.

4.3.4.1 Calculation using the correlation formulas

First, the dimensionless characteristics such as Re and Pr in forced convection, or Gr and Pr in free convection, have to be determined. Depending on the range of validity of the equations, an appropriate correlation is chosen and the Nu value calculated. The equation defining the Nusselt number is

$$\mathrm{Nu} = \frac{\alpha L}{\lambda} \quad (4.191)$$

given the heat transfer factor α. Using the equation

$$q = \frac{\Phi}{A} = \alpha\Theta, \quad (4.192)$$

the heat flow density is determined, which represents a certain temperature difference or the temperature difference between the wall and the fluid for a certain heat flow rate.

As an example, for free convective heat transfer from a vertical wall,

$$\mathrm{Nu} = 0.13 \cdot (\mathrm{Gr\ Pr})^{1/3} = 0.13 \cdot \left(\frac{g\vartheta L^3}{v^2}\mathrm{Pr}\right)^{1/3}. \quad (4.193)$$

Eqs. (4.191), (4.192), and (4.193) give

$$q = 0.13\lambda\left(\frac{g\vartheta}{v^2}\mathrm{Pr}\right)^{1/3}\Theta. \quad (4.194)$$

This equation does not incorporate the characteristic length L; hence the wall height has no influence on heat transfer.

In problems of forced convection, it is usually the cooling mass flow that has to be found to determine the temperature difference between the cooling substance and the wall for a given heat flow. In turbulent pipe flow the following equation is valid

$$\mathrm{Nu} = 0.0395\ \mathrm{Re}_d^{3/4}\mathrm{Pr}^{1/3} = \frac{qd}{\lambda\Theta}. \quad (4.195)$$

The mass flow is found using the continuity equation $\dot{m} = \rho w \pi d^2/4$ and the Reynolds number formula $\mathrm{Re} = 4\dot{m}/(\pi\rho\, dv)$

$$\dot{m} = \frac{\pi\rho\, dv}{4}\left(\frac{qd}{0.0395\ \mathrm{Pr}^{1/3}\Theta\lambda}\right)^{4/3}. \quad (4.196)$$

In some convection equations, such as for turbulent pipe flow, a special correction factor is used. This factor relates to the heat transfer conditions at the flow inlet, where the flow has not reached its final velocity

distribution, and the boundary layer is not fully developed. In this region the heat transfer rate is better than at the region of fully developed flow.

The reason the heat transfer is improved can be seen from the equations $\mathrm{Nu} = \alpha s/\lambda$ (where s is the thickness of the boundary layer) and $q = \alpha$, giving

$$q = \frac{\mathrm{Nu}\lambda\Theta}{s} \tag{4.197}$$

Thin boundary layers provide the highest values of heat flow density. Because the boundary layer gradually develops upstream from the inlet point, the heat flow density is the highest at the inlet point. Heat flow density decreases and achieves its final value in the region of fully developed flow. The correction is noted in the equations by means of the quotients d/L and d/x.

For some fluids, such as oils, the viscosity is temperature dependent. Here the correction factor $(\eta_F/\eta_w)^{0.14}$ is used, where η_F is the viscosity at the mean fluid temperature and η_w is the viscosity at the wall temperature.

4.3.4.2 *Forced convection*

In this section the correlations used to determine the heat and mass transfer rates are presented. The convection process may be either free or forced convection. In free convection fluid motion is created by buoyancy forces within the fluid. In most industrial processes, forced convection is necessary in order to achieve the most economic heat exchange. The heat transfer correlations for forced convection in external and internal flows are given in Tables 4.8 and 4.9, respectively, for different conditions and geometries.

The mass transfer correlations are obtained by replacing Nu by Sh and Pr by Sc according to the heat and mass transfer analogy.

4.3.4.3 *Free convection*

Flow up a vertical wall

$$\mathrm{Nu}_l = \left\{ 0.825 + \frac{0.387(\mathrm{Gr}\ \mathrm{Pr})^{1/6}}{\left(1 + (0.492/\mathrm{Pr})^{9/16}\right)^{8/27}} \right\}^2, \tag{4.198}$$

valid for $\mathrm{Gr}_l\mathrm{Pr} < 10^{12}$

$T_{\mathrm{st}} = 0.5(T_\infty + T_p)$

TABLE 4.8 Heat transfer correlations for external flow.

Geometry	Conditions	Correlation
Flat plate	Laminar, local, T_{av} $\mathrm{Pr} \geq 0.6$, $\mathrm{Re}_x < 10^5$	$\mathrm{Nu}_x = 0.332\ \mathrm{Re}_x^{1/2}\mathrm{Pr}^{1/3}$
	Laminar, average, T_{av} $\mathrm{Pr} \geq 0.6$, $\mathrm{Re}_x < 10^5$	$\overline{\mathrm{Nu}}_x = 0.664\ \mathrm{Re}_x^{1/2}\mathrm{Pr}^{1/3}$
	Turbulent, local, T_{av} $60 \geq \mathrm{Pr} \geq 0.6$ $\mathrm{Re}_x \leq 10^8$	$\mathrm{Nu}_x = 0.0296\ \mathrm{Re}_x^{1/2}\mathrm{Pr}^{1/3}$
	Mixed, average, T_{av} $60 > \mathrm{Pr} > 0.6$, $\mathrm{Re}_x \leq 10^8$	$\overline{\mathrm{Nu}}_x = (0.037\ \mathrm{Re}_x^{4/5} - 871)\mathrm{Pr}^{1/3}$
	Fully turbulent, average, T_{av} $5 \times 10^5 < \mathrm{Re}_x < 10^8$	$\overline{\mathrm{Nu}}_x = 0.037\ \mathrm{Re}_x^{4/5}\mathrm{Pr}^{1/3}$
Cylinder	Average, T_∞, $\mathrm{Pr} > 0.7$ $0.4 < \mathrm{Re}_d < 4 \times 10^5$	$\overline{\mathrm{Nu}}_d = C\mathrm{Re}_d^m\mathrm{Pr}^{1/3}$ C and m are given in Table 4.10
	Average, T_∞, $500 > \mathrm{Pr} > 0.7$ $1 < \mathrm{Re}_d < 10^6$	$\overline{\mathrm{Nu}}_d = C\mathrm{Re}_d^m\mathrm{Pr}^n\left(\frac{\mathrm{Pr}_\infty}{\mathrm{Pr}_s}\right)^{1/4}$ C and m are given in Table 4.11
	Average, T_{av} $\mathrm{Re}_d\ \mathrm{Pr} > 0.2$	$\overline{\mathrm{Nu}}_d = 0.3 + \frac{0.62\mathrm{Re}_d^{1/2}\mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}} \times \left[1 + \left(\frac{\mathrm{Re}_d}{28\ \ 200}\right)^{5/8}\right]^{4/5}$
Sphere	Average, T_∞, $180 > \mathrm{Pr} > 0.71$ $3.5 < \mathrm{Re}_d < 7.6 \times 10^4$ $1 < \frac{\mu_\infty}{\mu_S} < 3.2$	$\overline{\mathrm{Nu}}_d = 2 + \left(0.4\mathrm{Re}_d{}^{1/2} + 0.06\mathrm{Re}_d{}^{2/3}\right) \times \mathrm{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$
Falling drop	Average, T_∞	$\overline{\mathrm{Nu}}_d = 2 + 0.6\mathrm{Re}_d{}^{1/2}\mathrm{Pr}^{1/3}\left[25\left(\frac{x}{d}\right)^{-0.7}\right]^{1/4}$

Note: $T_{av} = 0.5\ (T_\infty + T_s)$, where T_∞ is the free stream temperature and T_s is the surface temperature.

TABLE 4.9 Heat transfer correlations for internal flow.

Conditions	Correlation
Laminar, fully developed, T_m, $q_s'' = cst$, $\mathrm{Pr} > 0.6$, $\mathrm{Re}_d < 2300$	$\mathrm{Nu}_d = 4.36$
Laminar, fully developed, T_m, $T_s = cst$, $\mathrm{Pr} > 0.6$, $\mathrm{Re}_d < 2300$	$\mathrm{Nu}_d = 3.66$
Laminar, thermal entry length $\overline{T}_m$, $T_s = cst$, $\mathrm{Pr} >> 1$ or unheated starting length	$\overline{\mathrm{Nu}}_d = 3.66 \frac{0.0668(d/L)\mathrm{Re}_d\mathrm{Pr}}{1+0.04[(d/L)\mathrm{Re}_d\mathrm{Pr}]^{2/3}}$
Turbulent, fully developed, T_m, $160 > \mathrm{Pr} > 0.6$, $\mathrm{Re}_d > 2300$, $\frac{L}{d}$ 10	$\mathrm{Nu}_d = 0.023\ \mathrm{Re}_d^{4/5}\ \mathrm{Pr}^{1/3}$
Turbulent, fully developed, T_m, $16{,}700 > \mathrm{Pr} > 0.7$, $\mathrm{Re}_d > 10^4$, $\frac{L}{d} > 10$	$\overline{\mathrm{Nu}}_d = 0.027\mathrm{Re}_d^{4/5}\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}$
Liquid metals, turbulent, fully developed, T_m, $q_s'' = cst$, $3.6\times10^3 < \mathrm{Re}_d < 9.05\times10^5$, $10^2 < \mathrm{Pe}_d < 10^4$	$\overline{\mathrm{Nu}}_d = 4.82 + 0.0185(\mathrm{Re}_d\ \mathrm{Pr})^{0.827}$
Liquid metals, turbulent, fully developed, T_m, $T_s = cst > \mathrm{Pe}_d > 100$	$\overline{\mathrm{Nu}}_d = 5 + 0.025(\mathrm{Re}_d\ \mathrm{Pr})^{0.8}$

Note: T_m is the mean bulk temperature and $\overline{T}_m = 0.5\ (T_{m,\ \mathrm{in}} + T_{m,\ \mathrm{out}})$.

TABLE 4.10 Constants for external flow correlation.

Re_d	C	M
0.4–4	0.989	0.33
4–400	0.911	0.385
40–4000	0.683	0.466
4000–40,000	0.193	0.618
40,000–400,000	0.027	0.805

TABLE 4.11 Constants for external flow correlation.

Re_d	C	m
1–400	0.75	0.4
400–1000	0.51	0.5
10^3–2×10^5	0.26	0.6
2×10^5–10^6	0.076	0.7

Flow upward on a horizontal plane

$$\mathrm{Nu}_l = 0.70(\mathrm{Gr}\ \mathrm{Pr})^{1/4} \quad \mathrm{Gr}\ \mathrm{Pr} < 4\times10^7$$

$$\mathrm{Nu}_l = 0.155(\mathrm{Gr}\ \mathrm{Pr})^{1/3} \quad \mathrm{Gr}\ \mathrm{Pr} > 4\times10^7 \qquad (4.199)$$

$$T_{\mathrm{st}} = 0.5(T_\infty + T_p)$$

Flow past a horizontal pipe

$$\mathrm{Nu}_d = \left\{0.825 + \frac{0.387(\mathrm{Gr}\ \mathrm{Pr})^{1/6}}{\left(1+(0.492/\mathrm{Pr})^{9/16}\right)^{8/27}}\right\}^2, \qquad (4.200)$$

valid for $\mathrm{Gr}_d\mathrm{Pr} < 10^{12}$

$$T_{\mathrm{st}} = 0.5(T_\infty + T_p)$$

Example 2

Heat transfer coefficient between a pipe and a wall. Water flows in a pipe ($d_s = 15$ mm) with a velocity of $v = 1.0$ m/s. The mean temperature of water is $\theta_m = 15°\mathrm{C}$ and the wall temperature is $\theta_s = 50°\mathrm{C}$. Calculate the heat transfer coefficient away from the pipe inlet. For water the properties are $\eta_{15°\mathrm{C}} = 1.14\times10^{-3}$ kg/(m s), $\eta_{50°\mathrm{C}} = 0.54\times10^{-3}$ kg/(m s), $c_{p15°\mathrm{C}} = 4.2$ kJ/(kg K), and $\lambda_{15°\mathrm{C}} = 0.60$ W/(m K), with turbulent flow. The Nusselt number equation is

$$\mathrm{Nu}_d = 0.037(\mathrm{Re}^{0.75} - 180)\mathrm{Pr}^{0.42}\left[1 + \left(\frac{d}{L}\right)^{2/3}\right]\left(\frac{\eta_F}{\eta_W}\right)^{0.14} \qquad (4.201)$$

$$\mathrm{Re} = \frac{vd}{\upsilon} = \frac{vd\rho}{\eta} = \frac{1.0\ m/s\times0.015\ \mathrm{m}\times1000\ \mathrm{kg}/m^3}{1.14\cdot10^{-3}\ \mathrm{kg/ms}} = 1.32\times10^4$$

$$\mathrm{Pr} = \frac{\eta c_p}{\lambda} = \frac{1.14\times10^{-3}\cdot4200}{0.6\ W/(\mathrm{m}\ K)} = 7.98$$

The flow is turbulent, $\mathrm{Re} > 2300$, and thus the part of Eq. (4.201) that considers the inlet flow region ≈ 1 can be ignored.

$$\mathrm{Nu}_d = 0.037(\mathrm{Re}^{0.75} - 180)\mathrm{Pr}^{0.42}\left(\frac{\eta_F}{\eta_W}\right)^{0.14}$$
$$= 0.037((1.32\times10^4) - 180)\times 7.98^{0.42}\left(\frac{1.14}{0.54}\right)^{0.14} = 103.3$$
$$\mathrm{Nu}_d = \frac{\alpha d}{\lambda} \Rightarrow \alpha = \frac{\mathrm{Nu}_d\lambda}{d} = \frac{103.3\times 0.60\ W/(\mathrm{m\,K})}{0.015}$$
$$= 4100\ W/(\mathrm{m^2\,K})$$

4.3.5 Thermal radiation

4.3.5.1 *Planck's law of radiation*

Total heat transfer consists of radiation at different frequencies. The distribution of radiation energy in a spectrum and its dependency on temperature is determined from Planck's law of radiation. M_{mv} and $M_{m\lambda}$ are the spectral radiation intensities for a blackbody

$$M_{mv} = \frac{2\pi h v^3}{c^2\left(\exp\left(hv/\lambda T\right) - 1\right)} = f(v, T) \tag{4.202}$$

$$M_{m\lambda} = \frac{2\pi hc^2}{\left(\exp\left(hc/\lambda T\right) - 1\right)} = \frac{c_1/\lambda^S}{\exp\left(c_2/\lambda T\right) - 1} = f(\lambda, T) \tag{4.203}$$

where h is the Planck's constant $= 3.99028\times10^{-7}$ J s/kmol $= 6.6252\times10^{-34}$ J s, c is the velocity of light $= 2.9979\times10^8$ m/s, c_1 is the first radiation constant $= 2\pi hc^2 = 3.7415\times10^{-16}$ W m^2, and c_2 is the second radiation constant $= hc = 119.626$ μm(MJ/kmol) $= 14{,}387.9$ μm K.

When these are derived with respect to the wavelength, and the wavelength value, with the maximum value of radiation intensity, is solved for, the result is Wien's law

$$\lambda_{\max}\cdot T = \text{constant} = 2898\ \mu m\ \mathrm{K} = 24{,}093\ \mu m\ \mathrm{kJ/kmol} \tag{4.204}$$

According to Wien's law, the wavelength representing the maximum point decreases with increasing temperature (Fig. 4.29).

The visible region of the spectrum lies between the wavelengths of 0.4 and 0.7 μm. When the temperature of a body is increased, its color changes toward smaller wavelengths—in other words, from the red region of the spectrum to the blue region.

4.3.5.2 *Emissivity and absorption*

Suppose two objects are in a hollow (Fig. 4.30): object A, which is black, and object B, which is gray (a body that does not absorb all the incoming radiation). The energy and mass are in balance when the temperatures of A, B, and C are equal. In the balanced state the radiation emitted by the bodies is equal to the radiation received.

The radiation density in C is in constant balance at all points and in all directions (for a given frequency). The radiation density is

$$L = \frac{d\Phi}{dA\,d\omega} \tag{4.205}$$

when the radiant energy $d\Phi$ passes through a surface element dA in the direction of its normal vector, in a space angle element $d\omega$.

For a black object,

$$L_v d\,v = L_{m\,v} dv \Rightarrow L_v = L_{m\,v}. \tag{4.206}$$

A gray object absorbs only part of the incoming radiation: incoming $= \alpha L_v\, d_v =$ outgoing $= \epsilon L_{mv}\, dv$.

From Kirchhoff's law,

$$\alpha(v,\ T,\ \vartheta) = \epsilon(v,\ T,\ \vartheta) \tag{4.207}$$

absorption ratio = emissivity

ϑ = direction angle to the surface normal

The radiation intensity from a surface to a semispace is temperature dependent on

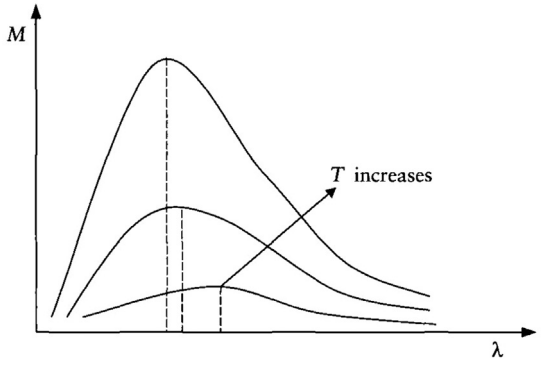


FIGURE 4.29 Radiation intensity of a blackbody as a function of wavelength (temperature parameter).

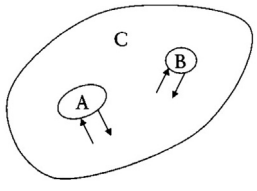


FIGURE 4.30 Radiative bodies.

$$\phi_{\text{emis}} = \epsilon M_m = \int e(v) M_{m\,v}\, dv \qquad M_m = \sigma T^4 \tag{4.208}$$

$\sigma = 5.67032 \times 10^{-8}$ W/(m^2 K^4), the Stefan–Boltzmann constant.

The radiation intensity absorbed by the surface is

$$\phi_{\text{abs}} = \alpha E = \int \alpha(v) E_v\, dv \tag{4.209}$$

E = radiation intensity = incoming Φ/A.

Emissivity is strongly dependent on the surface quality. The emissivity of a rough surface is greater than that of a smooth surface, increasing the rate of absorption. Emissivity values are found in textbooks. Care must be taken when using these values, as they usually denote total emissivities. The emissivity is considered constant in the spectrum, and this may be a poor approximation.

For example, the emissivity of white paper is high (≈ 0.93) at room temperature, and according to Kirchhoff's law, the absorption factor at the same temperature is also high. At relatively low temperature, as in this case, the radiation is concentrated in the long wavelengths, according to Wien's law. However, when the same paper receives radiation from the sun, at a radiation temperature of 6000K, when the absorption factor is small, this radiation has a short wavelength. A white object is a good reflector; this is why white clothing is used in the tropics.

When a low-temperature heating radiator is painted, the color is selected according to heat radiation. At this relatively low temperature, the radiation lies almost outside the visible region. The color may be deceptive. Snow is a good reflector of visible radiation; still, its total emissivity and therefore its absorption in normal conditions is as high as 0.98. Visible radiation passes through glass; its emissivity at the temperature 20°C is 0.98. Glass radiation lies within the 300–2800 nm range. An important feature of glass is that it is opaque to longwave radiation, which is produced by low-temperature emitters. It is this phenomenon that is termed the greenhouse effect.

The radiation emitted by the sun, due to its high temperature, has a short wavelength. Glass is transparent at this wavelength, allowing the radiation to pass through into the interior of the building. This energy is absorbed by the room surfaces, causing them to rise in temperature and to become low-temperature emitters. The radiation from these low-temperature surfaces is longwave, to which glass is opaque, and thus the radiation cannot escape through the glass to the outdoors, resulting in a rise in the space temperature. The transmission of radiation through the glass depends on the spectral characteristics of the nature of the glass.

Example 3

Silica glass transmits 92% of the radiation in the wavelength region of 0.3–2.7 μm, and it is impervious to other radiation. Determine the wavelength of the sun's radiation that the glass transmits when the sun is treated as a blackbody, $T = 5600$K. What happens for a blackbody at a temperature of 295K?

According to Eq. (4.204), at the median point of the spectral energy, λ_{50}, $T = 4107\ \mu$m K, and therefore the median point of the spectral energy of the sun is

$$\lambda_{50} = \frac{4107\ \mu m\ \text{K}}{5600\ \text{K}} = 0.733\ \mu m$$

Calculations using Planck's radiation law show which part of the radiation energy remains in the wavelength range

$$\frac{\lambda_{50}}{\lambda_1} = \frac{0.733}{0.35} = 2.095 \rightarrow \frac{1}{M}\int_0^{\lambda} M_\lambda d\lambda = 0.06$$

$$\frac{\lambda_{50}}{\lambda_2} = \frac{0.733}{2.7} = 0.2716 \rightarrow \frac{1}{M}\int_0^{\lambda} M_\lambda d\lambda = 0.97$$

Thus $97\% - 6\% = 91\%$ of the radiation energy lies in the range, and the silica glass transmits $91\% \times 92\% = 83.7\%$ of the radiation energy.

4.3.5.3 *Lambert's cosine law*

The radiation power in the direction of the normal vector to the surface dA in a space angle $d\omega$ (see Fig. 4.31) is

$$d\Phi = L dA\, d\omega \tag{4.210}$$

and in the direction ϑ,

$$d\Phi = L\, dA \cos\vartheta d\omega, \tag{4.211}$$

where $dA \cos\vartheta$ is the projection surface of the opening in the direction concerned. Thus the radiation power is distributed to different directions at a ratio of $\cos\vartheta$. This is the cosine law of Lambert. It is valid for a blackbody and approximately valid for insulating materials, but it cannot be used for bright metal surfaces.

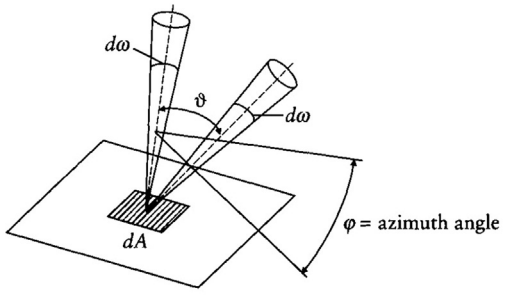


FIGURE 4.31 Direction and nature of incident radiation.

Substituting in Eq. (4.211) gives

$$d\omega = d\,\varphi \sin\vartheta\, d\vartheta = d\,\varphi\, d(\cos\vartheta) \tag{4.212}$$

and integrating for the radiation intensity to the semispace by a blackbody gives

$$d\Phi = M_m d\,A = \pi L_m dA, \tag{4.213}$$

where $M_m = \pi L_m = \sigma T^4$.

4.3.5.4 *Thermal radiation inside a vacuum (without gas)*

Surface element dA_j is located on the spatial surface A_j sending radiation to the surface A_i. The radiation power from the surface element dA_j to the element dA_j according to Eq. (4.211) is

$$d\Phi_{ji} = L_j(\omega_j) d\,A_j \cos\vartheta_j\, d\,\omega j. \tag{4.214}$$

The solid angle $d_{\omega j} = dA_i \cos\vartheta_i / r^2$. Eq. (4.211) gives

$$\frac{d\,\Phi_{ji}}{d\,A_i} = L(\omega_j)\frac{\cos\vartheta_i\,\vartheta_j}{r^2} = d\,A_i = d\,E_{ij}, \tag{4.215}$$

where E_{ij} is the proportion of the radiation from dA_j relative to the radiation intensity of dA_i. If Lambert's cosine law is valid, L is not dependent on direction and

$$\pi L_j = M_j. \tag{4.216}$$

Thus

$$d\,E_{ij} = M_j \frac{\cos\vartheta_i \cos\vartheta_j}{\pi r^2} d\,A_i \tag{4.217}$$

$$d\,E_{ij} = M_j \phi_{ij} dA_j. \tag{4.218}$$

The visibility factor of the surface element φ_{ij} depends on the geometry and gives that part of the radiation intensity of dA_j that falls directly on the surface dA_j or vice versa.

The radiation intensity of a surface element is the sum of emission and reflection

$$M_i = \epsilon_i M_{mi} + \rho_i E_i, \tag{4.219}$$

where ρ is the reflectance and

$$E_i = \int_{A_i} \phi_{ij} M_j dA_j, \tag{4.220}$$

giving an integral formula for the radiation intensity function,

$$M_i = \epsilon_i M_{mi} + \rho_i \int_{A_i} \phi_{ij} M_j dA_j. \tag{4.221}$$

Eq. (4.221) is difficult to solve, and for practical cases, approximate methods are used.

Usually, the surface is divided into zones, and with sufficient accuracy, M is considered constant over this area, giving

$$M_i = \epsilon_i M_{mi} + \rho_i \sum M_k f_{ik}, \tag{4.222}$$

where

$$f_{ik} = \int_{A_k} \varphi_{ik} dA_k \tag{4.223}$$

Also, by integration,

$$M_l A_l = \int_{A_l} M_l dA_l = \epsilon_i A_l M_{mi} + \rho_i \sum M_k A_l F_{kl}, \tag{4.224}$$

where

$$F_{kl} = \frac{1}{A_l}\int_{A_l} f_{lk} d\,A_l = \frac{1}{A_l}\int_{A_l}\int_{A_k} \phi_{lk} d\,A_k dA_1 \tag{4.225}$$

F_{kl} is the visibility factor between two finite surfaces, and $A_l F_{kl} = A_{kl}$ is the geometrical radiation surface. Eq. (4.225) shows that the geometrical radiation surface is symmetric and therefore

$$A_{kl} = A_{lk}. \tag{4.226}$$

Integrated over the semispace, the integrals (4.223) are 1. Thus in the hollow,

$$\sum F_{kl} = 1 \text{ and } \sum A_{kl} = A_k. \tag{4.227}$$

This sum includes all the hollow surfaces and also the surface k, if it is concave, in which case the portion F_{kk} of the received radiation is reflected back.

The net radiation power falling the surface is the difference between the incoming and outgoing radiation and the difference between absorption and emission

$$\frac{\Phi_{\text{net}}}{A} = q = E - M = \alpha E - \epsilon M_m. \tag{4.228}$$

Eliminating E when $\epsilon = \alpha$,

$$q = \left(M - \frac{M_m \epsilon}{\alpha}\right)\frac{\alpha}{\rho} = \frac{\epsilon}{\rho}(M - M_m). \tag{4.229}$$

For direct net radiation between two blackbodies (from Eq. 4.220),

$$\Phi_{12\text{net}} = (M_{m1} - M_{m2})A_{12} = A_{12}\sigma(T_1^4 - T_2^4). \tag{4.230}$$

Radiation heat transfer in a hollow can be represented by an electrical analogy as

$$\Phi_{\text{net}} = q\,A = \frac{\alpha}{\rho} A\left(M - \frac{\epsilon}{\alpha} M_m\right) = G(M - U) \tag{4.231}$$

$$\text{current} = \text{conductance} \times \text{potential difference}$$

where $(\epsilon/\alpha)M_m = U$ is the radiation potential, which is dependent on the temperature; ϵ is dependent on the radiation properties of the surface and the temperature; and α is dependent on the spectrum of the incoming radiation. $(\alpha/\rho)A = G$ is the radiation conductance between the potentials U and M.

For a gray body

$$\Phi = G(M - U)$$

When the surface is black, $G = \infty$ or $R = 0$, while $\alpha = 1$ and $\sigma = 0$; points U and M unite, and the potential is M_m.

When the surface is thermally insulated, $\Phi_{\text{net}} = 0$. Points U and M unite, and the potential is $M = (\epsilon/\alpha) M_m = UE = M$.

For two surfaces, $\Phi_{ij\ \text{net}} = E_{ij}A_i - E_{ji}A_j = A_{ji}(M_j - M_i)$, so by analogy A_{ij} is the radiation conductance between potentials M_i and M_j (see Fig. 4.32).

When there are only two surfaces in the hollow, the net thermal radiation is

$$\Phi_{12} = G(U_1 - U_2) = \frac{U_1 - U_2}{(1/G_1) + (1/A_{12}) + (1/G_2)} = \frac{(\epsilon_1/\alpha_1)M_{m1} - (\epsilon_2/\alpha_2)M_{m2}}{(\rho_1/\alpha_1 A_1) + (1/A_{12}) + (\rho_2/\alpha_2 A_2)}. \tag{4.232}$$

If surface 1 is convex or planar, all the incoming radiation is from surface 2, and $F_{12} = 1$, while the visibility factor expresses that part of radiation coming from this surface. If surface 2 is concave, a part of the radiation is also from this surface.

$$A_1F_{12} = A_{12} = A_{21} = A_2F_{21} = A_1$$

$$q_1 = \frac{\Phi_{12}}{A_1} = \frac{(\epsilon_1/\alpha_1)M_{m1} - (\epsilon_2/\alpha_2)M_{m2}}{(\rho_1/\alpha_1) + 1 + (\rho_2/\alpha_2)(A_1/A_2)}. \tag{4.233}$$

When $\rho_1 = 1 - \alpha_1$,

$$q_1 = \frac{(\epsilon_1/\alpha_1)M_{m1} - (\epsilon_2/\alpha_2)M_{m2}}{(1/\alpha_1) + (\rho_2/\alpha_2)(A_1/A_2)}. \tag{4.234}$$

Eq. (4.234) is valid for two coaxial cylinders and spheres. If $A_2 >> A_1$, $\alpha_2 \approx \epsilon_2$, as almost all the radiation from surface 2 is reflected back to it.

$$q_1 = M_{m1}\epsilon_1 - M_{m2}\alpha_1. \tag{4.235}$$

For two planes the dimensions of which are large compared with the distance between them,

$$F_{12} = F_{21} = 1A_{12} = A_1 = A_2$$

$$q = \frac{(\epsilon_1/\alpha_1)M_{m\,1} - (\epsilon_2/\alpha_2)M_{m2}}{(\rho_1/\alpha_1) + 1 + (\rho_2/\alpha_2)} = \frac{(\epsilon_1/\alpha_1)M_{m\,1} - (\epsilon_2/\alpha_2)M_{m2}}{((1-\alpha_1)/\alpha_1) + 1 + ((1-\alpha_2)/\alpha_2)} = \frac{(\epsilon_1/\alpha_1)M_{m\,1} - (\epsilon_2/\alpha_2)M_{m2}}{(1/\alpha_1) + (1/\alpha_2) - 1}. \tag{4.236}$$

It is difficult to estimate the absorption ratio. Approximately $\alpha_1 = \epsilon_1(T_2)$ and $\alpha_2 = \epsilon_2(T_1)$.

When the absorption relations are not dependent on temperature, the following approximations can be used ($\alpha = \epsilon = \text{constant} = 1 - \rho$). For two coaxial cylinders and spheres,

$$q = \frac{M_{m1} - M_{m2}}{((1-\epsilon_1)/\epsilon_1) + 1 + ((1-\epsilon_2)/\epsilon_2)(A_1/A_2)} = \frac{M_{m1} - M_{m2}}{(1/\epsilon_1) + (A_1/A_2)((1/\epsilon_2) - 1)}. \tag{4.237}$$

For two parallel planes, $A_1 \approx A_2$,

$$q = \frac{M_{m1} - M_{m2}}{(1/\epsilon_1) + (1/\epsilon_2) - 1}. \tag{4.238}$$

When $A_2 >> A_1$,

$$q = \epsilon_1(M_{m1} - M_{m2}) \tag{4.239}$$

In HVAC technology the following formula is used for small temperature differences with sufficient accuracy (see Fig. 4.33)

$$q_1 = \alpha_s \Delta T \approx \epsilon_1 \alpha_{mu} \Delta T. \tag{4.240}$$

Example 4

Radiation heat transfer. The radiation heat transfer between two parallel planes is reduced by placing a parallel aluminum sheet in the middle of the gap. The surface temperatures are $\theta_1 = 40°C$ and $\theta_2 = 5°C$, respectively; the emissivities are $\epsilon_1 = \epsilon_2 = 0.85$. The emissivity of both sides of the aluminum is $\epsilon_a = 0.05$. Calculate by how much the radiation heat transfer is reduced due to

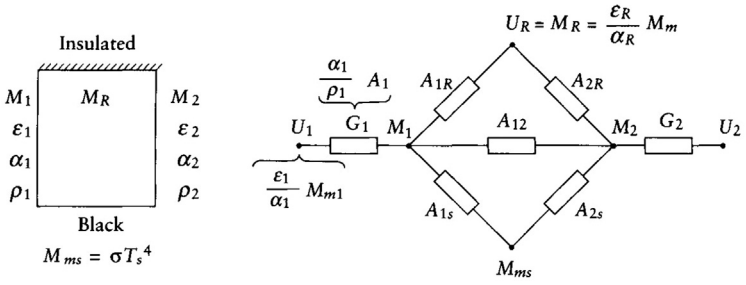


FIGURE 4.32 Radiation net for a hollow with four surfaces.

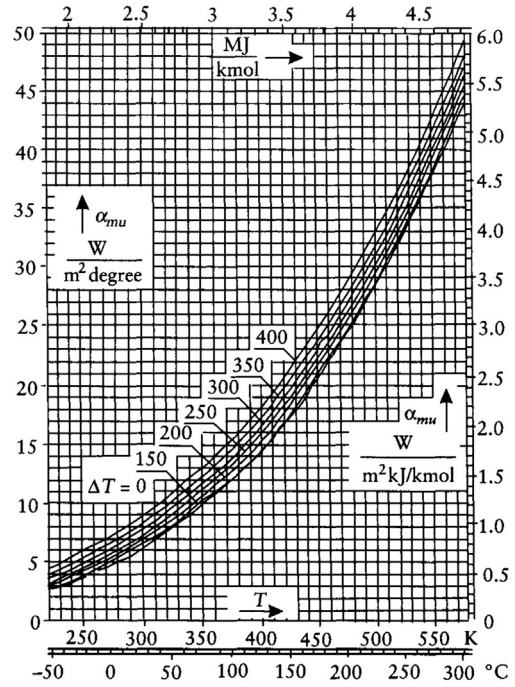


FIGURE 4.33 Heat transfer factor representing blackbody radiation for various mean temperatures and temperature differences.

the aluminum sheet; surface temperatures remain constant, and the surfaces are assumed to be gray.

Without the aluminum sheet,

$$q_{12} = \frac{M_{m1} - M_{m2}}{(1/\epsilon_1) + (1/\epsilon_2) - 1} = \frac{\alpha(T_1^4 - T_2^4)}{(2/\epsilon) - 1}$$

$$= \frac{5.27\ W/(m^2(100\ K)^4)(3.13^4 - 2.78^4)(100\ K)^4}{(2/0.85) - 1}$$

$$= 151.9\ W/m^2$$

$$A_{1a} = A_{a2} = \frac{1}{(1/\epsilon_1) + (1/\epsilon_a) - 1}.$$

With the aluminum,

$$q_{1a} = A_{1a}\sigma(T_1^4 - T_a^4) = q_{a2} = A_{\alpha 2}\sigma(T_a^4 - T_2^4) \Rightarrow$$
$$T_a^4 = \frac{T_1^4 + T_2^4}{2} = \frac{313^4 + 278^4}{2} = 7.785 \times 10^9\ K^4.$$

Thus

$$q_{1a} = \frac{1}{(1/0.85) + (1/0.05) - 1} 5.67\ W/(m^2(100\ K)^4)$$
$$(3.13^4 - 77.85)(100\ K)^4 = 5.1\ W/m^2.$$

Radiation heat transfer decreases by

$$1 - \frac{5.1}{151.9} \times 100\% = 96.6\%.$$

4.3.6 Mass transfer coefficient

Consider a binary mixture consisting of components A and B. If component A moves with a velocity of v_A and the component B with a velocity of v_B, there is a force against the motion of component A that is proportional to the velocity difference $(v_A - v_B)$. This is the physical content of Fick's law in the steady-state condition

$$j_A = -D_{AB}\frac{\partial c_A}{\partial_z}, \tag{4.241}$$

where j_A is the molar flux density (mol/m^2 s), D_{AB} is the diffusion factor (m^2/s), c_A is the concentration of component A (mol/m^3), and z is a coordinate parallel to the flux (m). Note that $j_A = c_A v_A$. On the basis of the force and counterforce, $D_{AB} = D_{BA}$.

Note: Eq. (4.241) characterizes diffusion when the mixture element is in steady state with no turbulence. Diffusion in a pipe can be represented by Eq. (4.241) in convective mass transfer; the flow and turbulence are important.

An important convective flow is created from vaporization alone, if no other component is absorbed from the gas and is replacing the vaporizing component. In drying technology, for example, the diffusion process is considered to be diffusion between water vapor (A) and dry air (B) (a mixture of nitrogen and oxygen), and only a small amount of dry air replaces the vaporized water, if the volume of the water in the form of liquid is very small. With good accuracy $j_B = 0$, and the diffusion caused by the concentration gradient $\partial c_B/\partial z$ is fulfilled with convective flow (Stefan flow) according to

$$j_B = -D_{BA}\frac{\partial c_B}{\partial z} + c_B v,$$

where $c_B v$ represents the convective flow that cancels the diffusion. Therefore the Stefan flow is

$$v = \frac{D_{BA}}{c_B}\frac{\partial c_B}{\partial z}.$$

The net flow of component A with Stefan flow taken into consideration is

$$j_A = -D_{AB}\frac{\partial c_A}{\partial z} + c_A v = -D_{AB}\frac{\partial c_A}{\partial z} + \frac{c_A}{c_B}D_{BA}\frac{\partial c_B}{\partial z}.$$

With constant temperature ($c = c_A + c_B = \text{constant}$),

$$\frac{\partial c_A}{\partial z} + \frac{\partial c_B}{\partial z} = 0 \text{ and } D_{AB} = D_{BA}$$

giving

$$j_A = -D_{AB}\frac{c}{c - c_A}\frac{\partial c_A}{\partial z}. \tag{4.242}$$

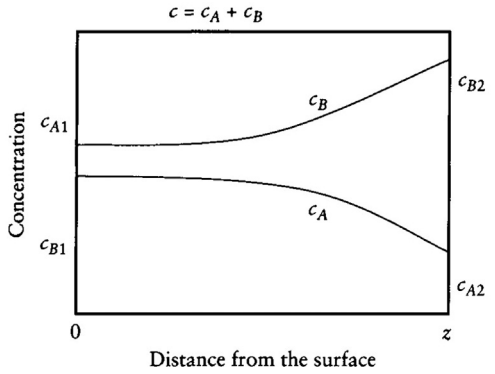


FIGURE 4.34 Diffusion of A through B when $j_B = 0$: semipermeable surface.

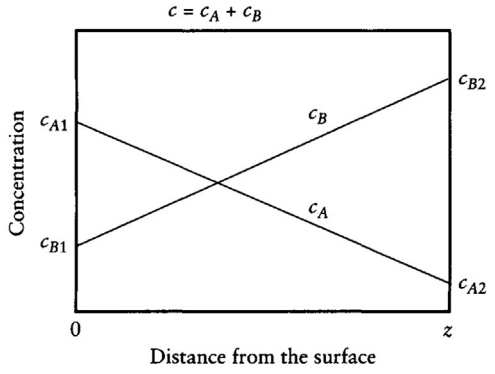


FIGURE 4.35 Diffusion with equal measures, $j_A = j_B$: fully permeable surface.

Integrating Eq. (4.242) gives

$$j_A = \frac{D_{AB} \cdot c}{z} \ln \frac{c - c_{A2}}{c - c_{A1}}. \tag{4.243}$$

where z is the thickness of the diffusion layer, $c_{A2} = c_A(z = z)$, and $c_{A1} = c_A(z = 0)$.

By giving j_A a constant value, $c_A(z)$ can be calculated from Eq. (4.243) for different z values. The concentration c_B can then be calculated as $c_B(z) = c - c_A(z)$. The result is shown in Fig. 4.34.

Component A diffuses due to the concentration gradient $-\partial c_A/\partial_z$. Component B diffuses due to the mean molar velocity v, $v = (c_A v_A + c_B v_B)/c$, like a fish swimming upstream with the same velocity as the flowing water, $j_B = 0$, with regard to a fixed point.

In a distillation process the diffusion is nearer to the case $j_A = -j_B = \text{constant}$ or component B absorbs in place of the vaporizing component A, and now $j_B \neq 0$. If $j_A = -j_B$, the concentrations are similar to those presented in Fig. 4.35.

An integral equation consistent for this case is the integrated Eq. (4.21)

$$j_A = \frac{D_{AB}}{z}(c_{A1} - c_{A2}). \tag{4.244}$$

Figs. 4.34 and 4.35 represent two extreme cases. Drying processes represent the case shown in Fig. 4.34, and distillation processes represent the case shown in Fig. 4.35. Neither case represents a convective mass transfer case; while the gas flow is in the boundary layer, other flows are Stefan flow and turbulence. Thus Eqs. (4.243) and (4.244) can seldom be used in practice, but their forms are used in determining the mass transfer factor for different cases.

Considering the case of Eq. (4.244), it is normal to describe a real mass transfer case by taking into consideration the boundary layer flows and the turbulence by using a mass transfer factor k'_c, which is defined by

$$j_A = k'_c(c_{A1} - c_{A2}). \tag{4.245}$$

For Eq. (4.243), which can be written similar to Eq. (4.244) as

$$j_A = \frac{D_{AB}c}{z} \ln \frac{c_{B2}}{c_{B1}} = \frac{D_{AB}c \ln(c_{B2}/c_{B1})}{z(c_{A1} - c_{A2})}(c_{A1} - c_{A2})$$
$$= \frac{D_{AB}c}{z} \frac{\ln(c_{B2}/c_{B1})}{(c_{A1} - c_{A2})}(c_{A1} - c_{A2}),$$

it is seen that by defining a logarithmic concentration difference

$$c_{BM} = \frac{c_{B2} - c_{B1}}{\ln(c_{B2}/c_{B1})}, \tag{4.246}$$

Eq. (4.243) can be written in an identical form

$$j_A = \frac{D_{AB}c}{z c_{BM}}(c_{A1} - c_{A2}). \tag{4.247}$$

Based on this, it is normal to define a mass transfer factor consistent with this case, analogous with Eq. (4.245)

$$j_A = k_c(c_{A1} - c_{A2}) \tag{4.248}$$

Assuming that the relation of Eqs. (4.243) and (4.244) represents correctly the ratio of the real mass transfer flows, if it is valid that

$$\frac{j_A(j_B = 0)}{j_A(j_A = -j_B)} = \frac{j_A(\text{Eq. (4.243)})}{j_A(\text{Eq. (4.244)})},$$

with Eq. (4.237) and the equations defining the mass transfer factors, Eqs. (4.245) and (4.248) give

$$k_c = \frac{c}{c_{BM}} k'_c. \tag{4.249}$$

If the ideal gas law is used for the gases, the concentrations can be shown by using partial pressures

$$\frac{c_A}{c} = \frac{p_a}{p} = y_A, \tag{4.250}$$

where $p = p_A + p_B$ is the total pressure, and y_A is the molar fraction of component A in the gas. The total concentration $c = c_A + c_B$ can be expressed in terms of pressure

$$c = \frac{n}{V} = \frac{p}{RT}, \tag{4.251}$$

where $R = 8.314$ J/kmol and T is the temperature (K). Partial pressure ρ_A can be calculated from

$$\rho_A = \frac{p_A M_A}{RT} = M_A c_A, \tag{4.252}$$

where M_A and M_B are the molar masses of components A and B. Eq. (4.246) can be expressed in a form using partial pressures

$$p_{BM} = \frac{p_{B2} - p_{B1}}{\ln(p_{B2}/p_{B1})} = \frac{p}{c} c_{BM}. \tag{4.253}$$

Eq. (4.247) can then be written as

$$j_A = \frac{D_{AB} p}{RTz p_{BM}} (p_{A1} - p_{A2}). \tag{4.254}$$

By using different potential differences,

$$c_{A1} - c_{A2} = \frac{c}{p}(p_{A1} - p_{A2}) = c(y_{A1} - y_{A2}) = \frac{M_B}{M_A} c(\rho_{A1} - \rho_{A2}), \tag{4.255}$$

single-material flow can be written in various ways

$$j_A = k_c(c_{A1} - c_{A2}) = k_G(p_{A1} - p_{A2}) = k_y(y_{A1} - y_{A2}). \tag{4.256a}$$

Instead of the molar flow, the mass flow can be used

$$m''_A = M_A j_A = k_\rho(\rho_{A1} - \rho_{A2}). \tag{4.256b}$$

With the use of Eqs. (4.255) and (4.256a), the following relationships between the mass transfer factors are obtained

$$k_c = \frac{1}{c} k_y = \frac{RT}{p} k_y = \frac{p}{c} k_G = RT k_G. \tag{4.257a}$$

Correspondingly, using Eqs. (4.255) and (4.256b), the following is obtained

$$k_c = \frac{k_\rho}{M_B c} = \frac{RT}{M_B p} k_\rho. \tag{4.257b}$$

The equations are valid for the case $j_A = -j_B$

$$k'_c = \frac{RT}{p} k'_y = RT k'_G = \frac{RT}{M_B p} k'_\rho. \tag{4.258}$$

By using Eq. (4.253), the approximation (4.249) becomes

$$k_c = \frac{p}{p_{BM}} k'_c. \tag{4.259}$$

In practice, the mass transfer factors are often presented without stating the experimental assumptions by which $j_A = -j_B$ or $j_B = 0$ has been obtained. The designer has to decide on the suitability of the experiments from which the quantity k'_c or k_c, is measured.

An idea of the approximate nature of Eq. (4.249) or the equivalent Eq. (4.259) can be gained by comparing a pure diffusion case (Fig. 4.36A) with the case involving a diffusion boundary layer (Fig. 4.36B).

4.3.7 Heat and mass transfer differential equations in the boundary layer and the corresponding analogy

We will consider flow through a solid element. Introducing the notations for molar flow density, partial density, and the reaction rate gives an equation for the mass balance

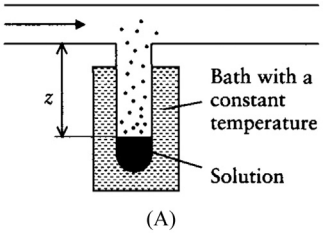
$$M_A\left(\frac{\partial I_{Ax}}{\partial x} + \frac{\partial I_{Ay}}{\partial y} + \frac{\partial I_{Az}}{\partial z}\right) + \frac{\partial \rho_A}{\partial t} = M_A r_A$$

(outflow − inflow) + accumulation = generation (4.260)

where I_{Ax} is the molar flow density of component A (mol/m^2 s) in the x direction, M_A is the molar mass of component A (kg/mol), ρ_A is the partial density of component A (kg/m^3), and r_A represents the formation rate of component A by chemical reactions (mol/m^3 s).

The corresponding equation for component B is

$$M_B\left(\frac{\partial I_{Bx}}{\partial x} + \frac{\partial I_{By}}{\partial y} + \frac{\partial I_{Bz}}{\partial z}\right) + \frac{\partial \rho_B}{\partial t} = M_B r_B. \tag{4.261}$$



(A)

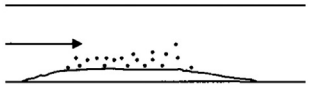

(B)

FIGURE 4.36 Comparison of diffusion cases.

The total mass balance is the sum of Eqs. (4.260) and (4.261)

$$\frac{\partial(M_A I_{Ax} + M_B I_{Bx})}{\partial x} + \frac{\partial(M_A I_{Ay} + M_B I_{By})}{\partial y} + \frac{\partial(M_A I_{Az} + M_B I_{Bz})}{\partial z} + \frac{\partial \rho}{\partial t} = 0, \tag{4.262}$$

where $\rho = \rho_A + \rho_B$. For a two-component or binary mixture, $M_A r_A + M_B r_B = 0$ is valid.

The mass flow density of the mixture in the x direction is defined by

$$\rho u_x = M_A I_{Ax} + M_B I_{Bx}, \tag{4.263}$$

with corresponding equations for other directions or velocities u_y and u_z. Using these notations, the mass balance of the mixture (Eq. 4.262) is written as

$$\frac{\partial(\rho u_x)}{\partial x} + \frac{\partial(\rho u_y)}{\partial y} + \frac{\partial(\rho u_z)}{\partial z} + \frac{\partial \rho}{\partial t} = 0. \tag{4.264}$$

For the mass balance of component A, diffusion velocity and the corresponding diffusion factor are defined with regard to the mean molar velocity v, defined by the following equation

$$v_x = \frac{c_A}{c} v_{Ax} + \frac{c_B}{c} v_{Bx}, \tag{4.265}$$

where v_{Ax} is the velocity of component A in the x direction, assuming a stable coordinate system. According to this, the mass flow density is

$$M_A I_{Ax} = \rho_A v_{Ax} = M_A c_A v_{Ax}. \tag{4.266}$$

The velocity of the mass center of the system, u_x (Eq. 4.263), can be written as

$$\rho u_{Ax} = \rho_A v_{Ax} + \rho_B v_{Bx} \tag{4.267}$$

using velocities v_{Ax} and v_{Bx}. The mass flow density may be written in the following ways

$$\rho v_{Ax} = \rho_A (v_{Ax} - v_x) + \rho_A v_x \tag{4.268a}$$

$$= \rho_A (v_{Ax} - u_x) + \rho_A u_x \tag{4.268b}$$

$$= \rho_A \frac{v_{Ax} - u_x}{v_{Ax} - v_x} (v_{Ax} - u_x) + \rho_A u_x. \tag{4.268c}$$

In a boundary layer equation the mass center is considered with the help of the velocity (u_x, u_y, u_z), and therefore a distribution of the velocity of the mass center is desirable. The diffusion velocity and diffusion factor are determined with regard to velocity v_x, giving a formula for $v_{Ax} - v_x$, but not for $v_{Ax} - u_x$. A useful approach is offered by Eq. (4.268c), using the artificial multiplication factor $(v_{Ax} - u_x)/(v_{Ax} - v_x)$.

From Eq. (4.267) and $\rho = \rho_A + \rho_B$, it is seen that the following equation is valid

$$\rho_A (v_{Ax} - u_x) = -\rho_B (v_{Bx} - u_x) = \frac{\rho_A \rho_B}{\rho} (v_{Ax} - v_{Bx}). \tag{4.269}$$

The last term is best understood by noting that

$$(v_{Ax} - v_{Bx}) = (v_{Ax} - u_x) - (v_{Bx} - u_x)$$

and then using the first part of Eq. (4.269).

Using Eq. (4.265) and $c = c_A + c_B$, the following connections result

$$c_A (v_{Ax} - v_x) = -c_B (v_{Bx} - v_x) = \frac{c_A c_B}{c} (v_{Ax} - v_{Bx}). \tag{4.270}$$

According to Eqs. (4.269) and (4.270),

$$\frac{v_{Ax} - u_x}{v_{Ax} - v_x} \frac{(\rho_B/\rho)}{c_B/c} \frac{(v_{Ax} - v_{Bx})}{(v_{Ax} - v_{Bx})} = \frac{\rho_B c}{\rho c_B}$$

Because $\rho_A = M_A c_A$, $\rho_B = M_B c_B$, and $\rho = Mc$, where $M = (c_A/c) M_A + (c_B/c) M_B$,

$$\frac{v_{Ax} - u_x}{v_{Ax} - v_x} = \frac{M_B}{M}. \tag{4.271}$$

Substituting Eq. (4.271) in Eq. (4.268c) gives

$$\rho_A v_{Ax} = \frac{M_B \rho_A}{M} (v_{Ax} - v_x) + \rho_A u_x. \tag{4.272}$$

For diffusion flow, $\rho_A (v_{Ax} - v_x)$, Fick's model (Eq. 4.241) gives

$$\rho_A (v_{Ax} - v_x) = M_A j_{Ax} = -M_A D_{AB} \frac{\partial c_A}{\partial x}. \tag{4.273}$$

Using important connections,

$$M_A I_{Ax} = \rho_A v_{Ax} = \frac{M_B}{M} M_A j_A + \rho_A u_x = -\frac{M_A M_B}{M} D_{AB} \frac{\partial c_A}{\partial x} + \rho_A u_x. \tag{4.274}$$

To shorten the notations the following definition is used

$$D'_{AB} = \frac{M_B}{M} D_{AB}. \tag{4.275}$$

Using this, the mass flow density is represented as

$$M_A I_{Ax} = -M_A D'_{AB} \frac{\partial c_A}{\partial x} + \rho u_x, \tag{4.276}$$

and correspondingly for directions y and z. Substituting Eq. (4.276) into Eq. (4.260) and keeping D'_{AB} constant give

$$\frac{\partial(\rho_A u_x)}{\partial x} + \frac{\partial(\rho_A u_y)}{\partial y} + \frac{\partial(\rho_A u_z)}{\partial z} - M_A D'_{AB} \left(\frac{\partial^2 c_A}{\partial x^2} + \frac{\partial^2 c_A}{\partial y^2} + \frac{\partial^2 c_A}{\partial z^2} \right) + \frac{\partial \rho_A}{\partial t} = M_A r_A \tag{4.277}$$

If the mixture density $\rho = \rho_A + \rho_B$ is held constant, Eq. (4.264) gives

$$\frac{\partial u_x}{\partial_x} + \frac{\partial u_y}{\partial y} + \frac{\partial u_z}{\partial_z} = 0. \tag{4.278}$$

Dividing Eq. (4.277) by M_A and using Eq. (4.278) give

$$u_x\frac{\partial c_A}{\partial x} + u_y\frac{\partial c_A}{\partial y} + u_z\frac{\partial c_A}{\partial z} + \frac{\partial c_A}{\partial t} = D'_{AB}\left(\frac{\partial^2 c_A}{\partial x^2} + \frac{\partial^2 c_A}{\partial y^2} + \frac{\partial^2 c_A}{\partial z^2}\right) + r_A, \tag{4.279}$$

because $c_A = (\rho_A/M_A)$.

In a similar manner the energy balance equation can be determined

$$u_x\frac{\partial T}{\partial x} + u_y\frac{\partial T}{\partial y} + u_z\frac{\partial T}{\partial z} + \frac{\partial T}{\partial t} = a\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) + \frac{\dot{Q}}{\rho c_p}. \tag{4.280}$$

where $a = \lambda/\rho c_p$ and $\dot{Q}$ is the heat generation per unit volume due to the chemical reactions (W/m^3), or $\dot{Q} = r_A\Delta H$, where ΔH is the reaction heat (J/mol). The thermal conductivity of the mixture is λ [W/(m K)], and c_p is the specific heat [J/(kg K)], or $\rho c_p = \rho_A c_{pA} + \rho_B c_{pB}$.

Eqs. (4.279) and (4.280) are similar. Fig. 4.37 shows a two-dimensional boundary layer flow over a plane. Ignoring any chemical reactions and considering steady-state conditions, Eqs. (4.279) and (4.280) give

$$u_x\frac{\partial c_A}{\partial x} + u_z\frac{\partial c_A}{\partial z} = D'_{AB}\left(\frac{\partial^2 c_A}{\partial x^2} + \frac{\partial^2 c_A}{\partial z^2}\right) \tag{4.281}$$

$$u_x\frac{\partial T}{\partial x} + u_z\frac{\partial T}{\partial z} = a\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial z^2}\right). \tag{4.282}$$

Assuming laminar flow for a linear momentum equation in the x direction (an approximation from the Navier–Stokes equations) gives

$$u_x\frac{\partial u_x}{\partial z} + u_z\frac{\partial u_x}{\partial x} = v\left(\frac{\partial^2 u_x}{\partial x^2} + \frac{\partial^2 u_x}{\partial z^2}\right) \tag{4.283}$$

where v is the kinematic viscosity (m^2/s).

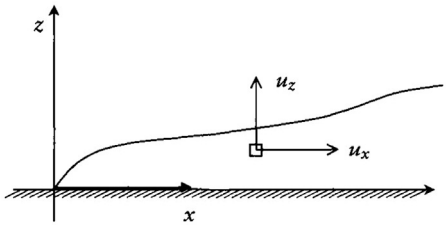


FIGURE 4.37 Boundary layer flow.

Eqs. (4.281)–(4.283) have to be solved at the same time as the continuity Eq. (4.278). The following dimensionless variables are used

$$w_x = \frac{u_x}{u_o} \tag{4.284a}$$

$$w_z = \frac{u_z}{u_o} \tag{4.284b}$$

$$z_A = \frac{c_A - c_{As}}{c_{Ao} - c_{As}} \tag{4.284c}$$

$$\theta = \frac{T - T_s}{T_o - T_s} \tag{4.284d}$$

$$\xi = \frac{x}{L} \tag{4.284e}$$

$$\eta = \frac{z}{L}, \tag{4.284f}$$

where u_o is the velocity outside the boundary layer or formally, $c_{As} = c_A(z = 0)$; c_{Ao} is the concentration of component A outside the boundary layer, $T_s = T(z = 0)$, and T_o is the temperature outside the boundary layer. The dimension L is the characteristic length. All dimensionless variables range between 0 and 1.

For example, $u_x(x, z) = u_o w_x(\xi(x), \eta(z))$. Using the chain rule,

$$\frac{\partial^2 u_x}{\partial x^2} = \frac{\partial}{\partial x}\left(\frac{\partial u_x}{\partial x}\right) = \frac{\partial}{\partial_x}\left(u_o\frac{\partial w_x}{\partial \xi}\frac{d\xi}{dx}\right) = \frac{u_o}{L}\frac{\partial}{\partial x}\left(\frac{\partial w_x}{\partial \xi}\right) = \frac{u_o}{L}\frac{\partial}{\partial \xi}\left(\frac{\partial w_x}{\partial \xi}\frac{d\xi}{dx}\right) = \frac{u_o}{L^2}\frac{\partial^2 w_x}{\partial \xi^2}$$

Treating the other terms in a similar manner, the linear momentum equation in a dimensionless form is obtained

$$w_x\frac{\partial w_x}{\partial_\eta} + w_z\frac{\partial w_x}{\partial \xi} = \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 w_x}{\partial \xi^2} + \frac{\partial^2 w_x}{\partial \eta^2}\right), \tag{4.285}$$

where $\mathrm{Re} = (u_o L)/v$ is the Reynolds number.

The dimensionless form of the continuity Eq. (4.278) ($u_y = 0$) in two-dimensional boundary layer flow is

$$\frac{\partial w_x}{\partial \xi} + \frac{\partial w_z}{\partial \eta} = 0. \tag{4.286}$$

For the two Eqs. (4.285) and (4.286) and two unknown variables $w_x(\xi, \eta)$, $w_z(\xi, \eta)$, boundary conditions are $\eta = 0$; $w_x = 0$, $\eta = \infty$; $w_x = 1$, $w_z = 0$. The boundary condition $w_z(\eta = 0)$ is not given in a mass transfer case, as it depends on the vaporization.

The dimensionless form of Eq. (4.282) is

$$w_x\frac{\partial \theta}{\partial \xi} + w_z\frac{\partial \theta}{\partial \eta} = \frac{a}{Lu_o}\left(\frac{\partial^2 \theta}{\partial \xi^2} + \frac{\partial^2 \theta}{\partial \eta^2}\right)$$

$$\frac{a}{Lu_o} = \frac{\lambda/\rho c_\mathrm{p}}{Lu_o} = \frac{1}{\mathrm{Re\ Pr}},$$

where $\mathrm{Pr} = (\mu c_p)/\lambda$ is the Prandtl number, and μ is the dynamic viscosity, giving

$$w_x \frac{\partial \theta}{\partial \xi} + w_z \frac{\partial \theta}{\partial \eta} = \frac{1}{\mathrm{Re\,Pr}} \left(\frac{\partial^2 \theta}{\partial \xi^2} + \frac{\partial^2 \theta}{\partial \eta^2} \right). \tag{4.287}$$

Boundary conditions for the dimensionless temperature are

$$\begin{aligned} &\theta = 0 \text{ at } \eta = 0 \\ &\eta = \infty \text{ at } \theta = 1 \end{aligned} \tag{4.288}$$

From Eqs. (4.285), (4.286) and (4.287), (4.288),

$$\theta = F(\xi,\ \eta,\ \mathrm{Re},\ \mathrm{Pr}). \tag{4.289}$$

Eq. (4.289) is an approximation in the mass transfer case, as the boundary conditions cannot always set $w_z(z = 0) = 0$. For the case $j_A = -j_B$, we nearly have $w_z(z = 0) = 0$, and the analogy equation is based on this situation.

The dimensionless form of Eq. (4.281) is

$$w_x \frac{\partial z_A}{\partial \xi} + w_z \frac{\partial z_A}{\partial \eta} = \frac{D'_{AB}}{Lu_o} \left(\frac{\partial^2 z_A}{\partial \xi^2} + \frac{\partial^2 z_A}{\partial \eta^2} \right)$$

$$\frac{D'_{AB}}{Lu_o} = \frac{D'_{AB}/v}{Lu_o/v} = \frac{1}{\mathrm{Re\,Sc}}$$

where $\mathrm{Sc} = v/D'_{AB}$ is the Schmidt number. We thus have

$$w_x \frac{\partial z_A}{\partial \xi} + w_z \frac{\partial z_A}{\partial \eta} + \frac{1}{\mathrm{Re\,Sc}} \left(\frac{\partial^2 z_A}{\partial \xi^2} + \frac{\partial^2 z_A}{\partial \eta^2} \right) \tag{4.290}$$

Boundary conditions for the dimensionless concentration are

$$\begin{aligned} &\text{at } \eta = 0,\ z_A = 0 \\ &\text{at } \eta = \infty,\ z_A = 1. \end{aligned} \tag{4.291}$$

Eq. (4.287) is in exactly the same form as Eq. (4.290), and the boundary conditions (4.288) and (4.291) are also similar.

If the solution to Eq. (4.289) is known, it is also valid for (4.290) and (4.291); hence

$$z_A = F(\xi,\ \eta,\ \mathrm{Re},\ \mathrm{Sc}). \tag{4.292}$$

The function F is then the same in Eqs. (4.289) and (4.292). This is not strictly correct, however; see the comments after Eq. (4.289).

We can apply this result to determine the analogy between mass and heat transfer factors. Mass flow density j_A (mol/m^2 s) can be given as

$$j_A = -D_{AB} \left(\frac{\partial c_A}{\partial z} \right)_{z=0} = k'_c (c_{As} - c_{Ao}). \tag{4.293}$$

The mass transfer factor k'_c is used because Eqs. (4.289) and (4.292) demand the boundary condition $w_z(\eta = 0) = 0$, which represents the case $j_A = -j_B$. Strictly speaking, a new mass transfer factor should be defined that represents the situation $M_A j_A = -M_B j_B$ or $w_z = 0$.

Using the dimensionless quantities z_A and η, Eqs. (4.284c) and (4.284f), Eq. (4.293) can be written as

$$j_A = -D_{AB}(c_{Ao} - c_{As}) \frac{1}{L} \left(\frac{\partial z_A}{\partial \eta} \right)_{\eta=0} = k'_c (c_{As} - c_{Ao}),$$

in which

$$\mathrm{Sh} = \frac{k'_c L}{D_{AB}} = \left(\frac{\partial z_A}{\partial \eta} \right)_{n=0}. \tag{4.294}$$

The dimensionless quantity Sh is called the Sherwood number.

The heat transfer factor α is defined by

$$q = -\lambda \left(\frac{\partial T}{\partial z} \right)_{z=0}, \tag{4.295}$$

where q is the heat flow density from the surface to the surroundings. Using dimensionless variables and η from Eqs. (4.284d) and (4.284f), Eq. (4.295) gives

$$\mathrm{Nu} = \frac{\alpha L}{\lambda} = \left(\frac{\partial \theta}{\partial \eta} \right)_{\eta=0}, \tag{4.296}$$

where the dimensionless quantity Nu is the Nusselt number.

According to Eqs. (4.289) and (4.292), it is seen that with constant ξ or x

$$\left(\frac{\partial z_A}{\partial \eta} \right)_{\eta=0} = \frac{\partial F}{\partial \eta}(\xi,\ 0,\ \mathrm{Re},\ \mathrm{Sc}) = G(\mathrm{Re},\ \mathrm{Sc})$$

$$\left(\frac{\partial \theta}{\partial \eta} \right)_{\eta=0} = \frac{\partial F}{\partial \eta}(\xi,\ 0,\ \mathrm{Re},\ \mathrm{Pr}) = G(\mathrm{Re},\ \mathrm{Pr}),$$

which leads to the important results

$$\mathrm{Sh} = \mathrm{G}(\mathrm{Re},\ \mathrm{Sc}) \tag{4.297}$$

$$\mathrm{Nu} = \mathrm{G}(\mathrm{Re},\ \mathrm{Pr}) \tag{4.298}$$

The above shows how the dimensionless numbers are used to provide the most accurate solution. Collecting these definitions together,

$$\mathrm{Sh} = \frac{k'_c L}{D_{AB}},\ \mathrm{Nu} = \frac{\alpha L}{\lambda},\ \mathrm{Re} = \frac{u_o L}{v},\ \mathrm{Pr} = \frac{\mu c_p}{\lambda},$$

$$\text{and } \mathrm{Sc} = \frac{v}{D'_{AB}}$$

Note the diffusion factor appearing in the Schmidt number, $D'_{AB} = (M_B/M) D_{AB}$ (Eq. 4.275).

The preceding discussion has attempted to formulate the situation for laminar boundary layer flow as accurately as possible and to obtain precise correlation between the heat transfer and mass transfer factors.

It is not possible to translate the above reasoning to turbulent flow, as turbulent flow equations are not reliable. However, in practice, it is typical to assume that the same analogy is also valid for turbulent flow. Because of this hypothesis level, it is quite futile to use the diffusion factor D'_{AB} in the Schmidt number; instead, we will directly use the number D_{AB} as in the Sherwood number. Hence in practical calculations, $\mathrm{Sc} = v/D_{AB}$.

Example 5

Heat transfer is defined by $\mathrm{Nu} = A\mathrm{R}^m\mathrm{Pr}^n$. The function $G(\cdot)$ is given as

$$G(\mathrm{Re},\ \mathrm{Pr}) = A\ \mathrm{Re}^m\mathrm{Pr}^n.$$

According to the analogy model, it is valid that

$$\mathrm{Sh} = G(\mathrm{Re},\ \mathrm{Sc}) = A\ \mathrm{Re}^m\mathrm{Sc}^n \tag{4.299}$$

This allows the mass transfer factor to be calculated. The abovementioned equation can be refined to

$$\mathrm{Sh} = G(\mathrm{Re},\ \mathrm{Sc}) = A\ \mathrm{Re}^m\mathrm{Pr}^n\left(\frac{\mathrm{Sc}}{\mathrm{Pr}}\right)^n.$$

It follows that

$$\frac{k'_c L}{D_{AB}} = \frac{\alpha\, L}{\lambda}\left(\frac{v/D_{AB}}{\rho c_p/\lambda}\right)^n,$$

simplifying to

$$k'_c = \frac{\alpha}{\rho c_p}\mathrm{Le}^{1-n}, \tag{4.300}$$

where $\mathrm{Le} = (D_{AB}\rho c_p)/\lambda = \mathrm{Pr}/\mathrm{Sc}$ is the Lewis number (or Luikov's number in the Russian literature).

4.3.8 Diffusion through a porous material

In a steady-state situation when gas flows through a porous material at a low velocity (laminar flow), the following empirical formula, Darcy's model, is valid

$$\rho u_x = -\frac{k}{v}\frac{\partial p}{\partial x}, \tag{4.301}$$

where k represents the permeability of the matter (m^2). The kinematic viscosity of the gas is denoted by v (m^2/s), and ρ is the density of the gas for the total volume—or if the real density of the gas is d, $\rho = \phi$, where ϕ is the volume percentage of the gas in the porous material. It is also seen that ϕ gives the percentage of the free cross-sectional area of the gas in the material

$$\varphi = \frac{\rho}{d} = \frac{m(g)/V}{m(g)/V(g)} = \frac{V(g)}{V} = \frac{A(g)L}{AL} = \frac{A(g)}{A}.$$

In Eq. (4.301), velocity u_x is the real velocity of the gas in the pores: $u_x = q_v/A(g) = q_v/(\phi)A$, where q_v is the volume flow.

From Darcy's equation, we can determine a formula for the counterforce produced by the porous material to the flowing or diffusing component A. If this counterforce is found, it can be added to the diffusion resistance force caused by component B to component A; hence the sum of these two forces represents the total diffusion resistance.

For a porous material the linear momentum equation can be written as

$$\rho\frac{\partial u_x}{\partial t} = -\varphi\frac{\partial p}{\partial x} + f_{mx}, \tag{4.302}$$

where f_{mx} represents the resistance force between the gas and the material, the flow friction. The term ϕ is important in Eq. (4.302). It comes from the fact that while ρ appears on the left side of Eq. (4.302), the balance is constructed for the mixture of the material and the gas, and therefore the pressure must be calculated for the surface area of the total material ϕ. When ϕ is held constant, independent of x, Eq. (4.302) is obtained.

In a steady-state case, Eq. (4.302) is simplified to

$$f_{mx} = \varphi\frac{\partial p}{\partial x},$$

and with Eq. (4.301),

$$f_{mx} = -\frac{\eta}{k}\varphi^2 u_x, \tag{4.303}$$

with $v = \eta/d = \eta\ \ dh\ \ 3$. If the flow velocity is zero, Eq. (4.303) can be interpreted as saying that the resistance force is linearly proportional to the velocity difference between the gas and the material and also linearly proportional to the dynamic viscosity of the gas.

Eq. (4.303) is valid but it is lacking something. The resistance force $f_{mx}^{(A)}$ that applies to the component A has to be found, and not that for the whole mixture. The force applying to the whole mixture f_{mx} is the sum of the partial forces $f_{mx}^{(A)}$ and $f_{mx}^{(B)}$

$$f_{mx} = f_{mx}^{(A)} + f_{mx}^{(B)}. \tag{4.304}$$

Assuming that the force is divided along the ratio of the mass flows, Eq. (4.303) gives

$$f_{mx}^{(A)} = -\frac{\rho_A}{\rho}\frac{\eta}{k}\varphi^2 \mathrm{v}_{Ax} \tag{4.305a}$$

$$f_{mx}^{(B)} = -\frac{\rho_B}{\rho}\frac{\eta}{k}\varphi^2 \mathrm{v}_{Bx} \tag{4.305b}$$

Summing (4.305a) and (4.305b), f_{mx} is obtained for Eq. (4.303). This is due to the fact that $\rho_A v_{Ax} + \rho_B v_{Bx} = \rho u_x$.

We now consider the resistance force $f_{mx}^{(A)}$ caused by the diffusion. This force resists the diffusion flow in a porous material together with $f_{mx}^{(A)}$. Writing the linear momentum equation for component A in accordance with Eq. (4.302),

$$\rho_A \frac{d\mathrm{v}_{Ax}}{dt} = -\varphi \frac{dp_A}{dx} + f_{dx}^{(A)} + f_{dx}^{(B)}. \tag{4.306}$$

This gives a model for $f_{mx}^{(A)}$, Eq. (4.305b), but not a model for force $f_{dx}^{(A)}$. While force f_{mx} gives the flow force caused by the material, it is normal to represent this fact so that $f_{dx}^{(A)}$ gives the pure diffusion resistance force that is not caused by the material. This requires treating $f_{dx}^{(A)}$ independently from the material or porosity.

For $\phi = 1$ or $k = \infty$, where $f_{mx}^{(A)} =$ Eq. (4.306) gives

$$\rho_A \frac{d\mathrm{v}_{Ax}}{dt} = -\frac{\partial p_A}{\partial x} + f_{dx}^{(A)}. \tag{4.307}$$

In a steady-state case the results are

$$f_{dx}^{(A)} = \frac{\partial p_A}{\partial x}. \tag{4.308}$$

In a steady-state case at constant pressure ($p = p_A + p_B =$ constant), Fick's law (Eq. 4.273) is valid

$$\rho_A(\mathrm{v}_{Ax} - \mathrm{v}_x) = -M_A D_{AB} \frac{1}{RT} \frac{\partial p_A}{\partial x},$$

with $c_A = p_A/(RT)$.

Instead of the mean velocity v_x weighted with the molar fractions, a velocity weighted with the mass fractions—the mass center velocity u_x—can be used. Using Eq. (4.271), the abovementioned equation becomes

$$\rho_A(v_{Ax} - u_x) = -\frac{M_A M_B}{M} \frac{D_{AB}}{RT} \frac{\partial p_A}{\partial x}, \tag{4.309}$$

Eqs. (4.308) and (4.309) give a formula for the diffusion resistance force $f_{dx}^{(A)}$

$$f_{dx}^{(A)} = -\frac{M}{M_A M_B} \frac{RT}{D_{AB}} \rho_A(\mathrm{v}_{Ax} - u_x). \tag{4.310}$$

Substituting the formulas for forces $f_{dx}^{(A)}$ and $f_{mx}^{(A)}$ (Eqs. 4.310 and 4.305a) in Eq. (4.306) gives

$$\rho_A \frac{dv_{Ax}}{dt} = -\varphi \frac{\partial p_A}{\partial x} - \frac{M}{M_A M_B} \frac{RT}{D_{AB}} \rho_A(v_{Ax} - u_x) - \frac{\eta_A \varphi^2}{k} v_{Ax}. \tag{4.311}$$

In the steady-state case, Eq. (4.311) is simplified to

$$\frac{M}{M_A M_B} \frac{RT}{D_{AB}} \rho_A(v_{Ax} - u_x) + \frac{\rho_A}{\rho} \frac{\eta \varphi^2}{k} v_{Ax} = -\varphi \frac{\partial p_A}{\partial x}. \tag{4.312}$$

The aim is to solve this equation for the term v_{Ax}, or actually $\rho_A v_{Ax}$, which is the diffusion flow density of component A.

An important case is

$$v_{Bx} = 0. \tag{4.313}$$

This case applies to drying processes ($B =$ dry air), condensation, and absorption, such as the diffusion of sulfur dioxide gas through a calcium oxide.

When Eq. (4.313) is valid, $\rho u_x = \rho_A v_{Ax} + \rho_B v_{Bx} = \rho_A v_{Ax}$, and $u_x = (\rho_A/\rho) v_{Ax}$ ($\rho = \rho_A + \rho_B$). Therefore in this case,

$$v_{Ax} - u_x = \left(1 - \frac{\rho_A}{\rho}\right) v_{Ax} = \frac{p_B}{p} v_{Ax.} \tag{4.314}$$

Substituting Eq. (4.314) in Eq. (4.312) and solving for the mass flow density give

$$\rho_A v_{Ax} = -\frac{\varphi}{(M/M_A M_B)(RT/D_{AB})(\rho_B/\rho) + (\eta\varphi^2/\rho k)} \frac{\partial p_A}{\partial x}. \tag{4.315}$$

This can be written as

$$\frac{\rho_B}{\rho} = \frac{p_B M_B}{pM} = \frac{p - p_A}{p} \frac{M_B}{M}$$

and

$$\frac{\eta}{\rho} = \frac{\eta}{\varphi d} = \frac{v}{\varphi}.$$

Substituting these into Eq. (4.315) after grouping terms,

$$\rho_A v_{Ax} = -\frac{\varphi}{1 + (p/p - p_A)(M_A/RT) D_{AB} v(\varphi/k)} \frac{p}{p - p_A} \frac{M_A}{RT} D_{AB} \frac{\partial p_A}{\partial x}. \tag{4.316}$$

The term

$$\epsilon = \frac{\varphi}{1 + (p/(p - p_A))(M_A/RT) D_{AB} v(\varphi/k)} \tag{4.317}$$

is the resistance factor caused by the material. Using Eq. (4.317), the diffusion flow is

$$\rho_A v_{Ax} = -\epsilon \frac{p}{p - p_A} \frac{M_A}{RT} D_{AB} \frac{\partial p_A}{\partial x}. \tag{4.318}$$

The term $p/(p - p_A)$ derives from assumption (4.313) representing the effects of Stefan flow. If $\epsilon = 1$, Eq. (4.318) gives the diffusion flow in a free space.

Eq. (4.318) indicates that if $k \to \infty$, the diffusion resistance remains under 1, namely, $\epsilon_d = \varphi$. This is easy to understand, as φ represents that part of the material cross-sectional surface through which the vapor diffuses.

4.3.9 Example of drying process calculation

The problems experienced in drying process calculations can be divided into two categories: the boundary layer factors outside the material and humidity conditions and the heat transfer problem inside the material. The latter is more difficult to solve mathematically, due mostly to the moving liquid by capillary flow. Capillary flow tends to balance the moisture differences inside the material during the drying process. The mathematical discussion of capillary flow requires consideration of the linear momentum equation for water and requires knowledge of the water pressure, its dependency on moisture content and temperature, and the flow resistance force between water and the material. Due to the complex nature of this, it is not considered here.

We will cover a simple drying model to examine the radiation drier of coated paper. We assume there are no major temperature or humidity variations in the direction of the paper web thickness, and that temperature T and humidity u are constant in the direction of thickness. This assumption requires that the capillary action be ignored, and the pressure gradient of water is zero on the assumption $\partial u/\partial x = \partial T/\partial x = 0$. How is it possible that the humidity distribution remains uniform?

The only approach is to ignore the capillary flow and to assume water vaporization takes place evenly in the thickness of the paper web. With a radiation drier, this approach is reasonable if the radiation energy is absorbed evenly inside the web. Assuming the boundary layers on both sides of the web are similar, the vapor flow is distributed symmetrically to the center of the web. The model of vaporization and water drift is shown in Fig. 4.38.

To derive formula (4.318) for vapor flow in a porous material, we approximate the pressure gradient in Eq. (4.318) with

$$\frac{\partial p_A}{\partial x} = \frac{p_A(s) - p_A(T,\ u)}{\Delta x}, \tag{4.319}$$

where $\Delta x = s/4$ and s is the thickness of the paper web (Fig. 4.38). The vapor flow through the boundary layer can be represented as

$$\rho_A v_{Ax} = k_c' \frac{M_A p}{RT} \ln \frac{p - p_A(y)}{p - p_A(s)}, \tag{4.320}$$

where $k_c' = (\alpha/\rho c_p)\mathrm{Le}^{1-n}$ (Eq. (4.300)).

Eq. (4.314) follows from Eqs. (4.250), (4.253), (4.256a), and (4.259)

$$\begin{aligned} j_A &= k_c\big(c_A(s) - c_A(y)\big) \text{ (Eq. 4.256a)} \\ &= k_c' \tfrac{p}{p_{BM}}(c_A(s) - c_A(y)) \text{ (Eq. 4.259)} \\ &= k_c' \tfrac{c}{p_{BM}}(p_A(s) - p_A(y)) \text{ (Eq. 4.250)} \\ &= k_c' \tfrac{c \ln(p_B(y)/p_B(s))}{p_B(y) - p_B(s)}(p_A(s) - p_A(y)) \text{ (Eq. (4.253))} \\ &= k_c' c \ln \tfrac{p - p_A(y)}{p - p_A(s)} \qquad (p = p_A + p_B) \end{aligned}$$

with reference to Eq. (4.251),

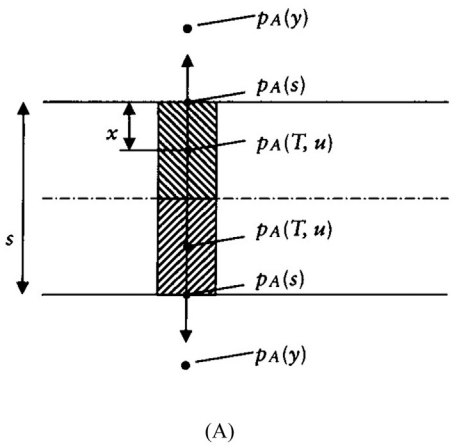
$$\rho_A v_{Ax} = M_A j_A = k_c' \frac{M_A p}{RT} \ln \frac{p - p_A(y)}{p - p_A(s)},$$

or Eq. (4.320).

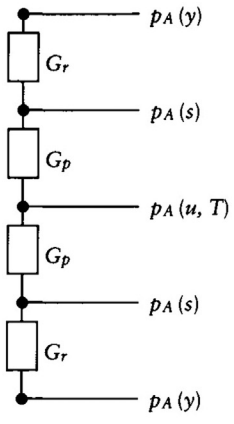
The logarithmic function in Eq. (4.320) can be written as

$$\ln \frac{p - p_A(y)}{p - p_A(s)} = \ln\left(1 + \frac{p_A(s) - p_A(y)}{p - p_A(s)}\right).$$

When $(p_A(s) - p_A(y))/(p - p_A(s)) < 1$, a logarithmic function to a series $(\ln(1 + x) = x - x^2/s + \cdots)$ can be



(A)



(B)

FIGURE 4.38 (A) The uniform vaporization of water in paper. (B) Resistance web analogy for steam flow; $1/G_p$ = resistance due to the paper, $1/G_r$ = resistance due to the boundary layer.

developed and, using the first term, gives the approximation

$$\ln\frac{p-p_A(y)}{p-p_A(s)}=\frac{p_A(s)-p_A(y)}{p-p_A(T,\,u)}. \tag{4.321}$$

In this approximation, $p_A(s)$ in the denominator is replaced by $p_A(T, u)$.

We use the following notations

$$G_r=k_c'\frac{M_A p}{R\,T}\frac{1}{p-p_A(T,\,u)} \tag{4.322}$$

$$G_p=\epsilon\frac{p}{p-p_A(T,\,u)}\frac{M_A}{R\,T}D_{AB}\frac{1}{\Delta x}. \tag{4.323}$$

Using approximation (4.319) in Eq. (4.318) and Eq. (4.317) in Eq. (4.320) leads to the steam flow formula

$$\rho_A v_{Ax}=G_p(p_A(T,u)-p_A(s))=G_r(p_A(s)-p_A(y)).$$

Eliminating $p_A(s)$ gives

$$\rho_A v_{Ax}=G(p_A(T,u)-p_A(y)), \tag{4.324}$$

The total conductance is determined from

$$\frac{1}{G}=\frac{1}{G_r}+\frac{1}{G_p}. \tag{4.325}$$

The total vaporization on both sides of the web is determined by Eq. (4.324) multiplied by 2. If the radiation power absorbed in the web is $\dot{Q}''$ (W/m^2), the energy balance can be written as

$$\dot{Q}''=C''\frac{dT}{dt}+2\,G(p_A(T,u)-p_A(y))l(T,\,u), \tag{4.326}$$

where $l(T, u)$ is the vaporization heat of water, and C'' is the heat capacity of the web [J/(m^2 K)]. This can be calculated from

$$C''=m''\left(c_{p1}+uc_{p2}\right), \tag{4.327}$$

where m'' is the square mass of the dry substance of the paper web (kg/m^2), c_{p1} is the specific heat of the dry substance, and c_{p2} is the specific heat of water.

The humidity change due to vaporization is

$$-m''\frac{du}{dt}=2G(p_A(T,\,u)-p_A(y)). \tag{4.328}$$

The negative sign in Eq. (4.328) is due to the fact that $(du/dt)<0$ represents the net vaporization to the surroundings.

Calculating the temperature rise at each time t with Eqs. (4.326) and (4.328) gives the corresponding change in humidity. As the function G and the pressure $p(T, u)$ are complex, numerical solutions are used.

Example 6

Calculate the humidity change and the temperature rise in a paper web at the time when the web humidity $u=0.20$ and temperature $\theta=70°$C. Assume the heat transfer factor on the web surface is $\alpha=40$ W/(m^2 K), and the humidity of the surrounding air is $x=0.05$ kg H_2O/kg dry air. The radiation power density absorbed by the web surface is 250 kW/m^2.

The vaporization heat of water, which depends on the humidity, is accurately determined by

$$l(T,\,u)=l_o(T)+r(T,\,u), \tag{4.329}$$

where $l_o(T)$ is the vaporization heat of free water, and $r(T, u)$ is the required auxiliary heat (sorption heat).

The sorption heat for the newsprint is calculated as

$$r=A_6u^{A_2}\exp(A_4u)T^2\left(\frac{T_{cr}-T}{A_7}\right)^{A_3-1}, \tag{4.330}$$

where $A_2=-1.3820$, $A_3=7.557$, $A_4=-3.372$, $A_6=8.633\times10^{-3}$ kJ/kg/K^2, $A_7=696.0$K, and $T_{cr}=647.3$K. Substituting $u=0.20$ and $T=343$ K in this equation gives $r=21.1$ kJ/kg. Table 4.7 gives $l_o(\theta=70°\text{C})=2333.3$ kJ/kg; hence, $l(T, u)=2354\times10^3$ J/kg.

The partial pressure of the surrounding air $p_A(y)$ is calculated using the humidity $x=0.05$ kg H_2O/kg dry air

$$p_A(y)=\frac{x}{0.622+x}p=\frac{0.05}{0.622+0.05}10^5=7440\text{ Pa}$$

The steam pressure inside the paper web $p_A(T, u)$ calculated in the previous example was $p_A(T,\,u)=27.7\times10^3$ Pa and the diffusion resistance factor $\varepsilon=0.50$. If the thickness of the paper web $s=0.09$ mm, the mean diffusion distance inside the paper is $\Delta x=s/4=0.0225$ mm.

Substituting the numerical values in Eq. (4.323) to determine the conductance

$$G_p=0.5\left(\frac{10^5}{10^5-27.7\times10^3}\right)\left(\frac{18\times10^{-3}}{8.314\times343}\right)\times36.3\times3\times10^{-6}\frac{1}{0.0225\times10^{-3}}$$
$$=7.04\times10^{-6}\text{ kg/(}m^2\text{ s Pa).}$$

To calculate the conductance of the boundary layer, we first calculate the mass transfer factor using Eq. (4.300)

$$k_c'=\frac{\alpha}{\rho c_p}\text{Le}^{1-n}.$$

Assume that $n=0.4$. Table 4.7 gives ($\theta=70°$C)

$$\rho_A = 0.1981\ \text{kg}/m^3, \qquad \rho_B = 0.715\ \text{kg}/m^3$$
$$D_{AB} = 36.3 \times 10^{-6}\ \text{m}^2/s, \quad \lambda = 0.02418\ W/(m\ K)$$
$$\rho c_p = \rho_A c_{pA} + \rho_B c_{pB} = 0.1981 \times 1850 + 0.715 \times 1006 = 1086\ J/(m^3\ K)$$
$$\text{Le} = \frac{D_{AB}\rho c_p}{\lambda} = 1.63$$
$$k_c' = \frac{40}{1086} \times 1.63^{1-0.4} = 4.93 \times 10^{-2} m/s$$

Boundary layer conductance is, from Eq. (4.322),

$$G_r = 4.93 \times 10^{-2} \frac{18 \times 10^3 \times 10^5}{8.314 \times 343} \frac{1}{10^5 - 27.7 \times 10^3} = 0.432 \times 10^{-6}\ \text{kg}/(m^2\ \text{s Pa}).$$

Total conductance is, from Eq. (4.325),

$$G = \left[\frac{1}{0.432 \times 10^{-6}} + \frac{1}{7.04 \times 10^{-6}}\right]^{-1} = 0.407 \times 10^{-6}\ \text{kg}/(m^2\ \text{s Pa}).$$

Vaporization at time $t = 2 \times 0.407 \times 10^{-6}(27.7 \times 10^3 - 7.44 \times 10^3) = 16.49 \times 10^{-3}\ \text{kg}/(\text{m}^2\ \text{s})$. The humidity change rate is calculated from Eq. (4.328) when the square mass of the dry substance of the paper web is $m'' = 40.5 \times 10^{-3}\ \text{kg}/\text{m}^2$.

$$\frac{du}{dt} = -\frac{16.49 \times 10^{-3}}{40.5 \times 10^{-3}} = -0.41\ \text{s}^{-1}.$$

Heat capacity $C'' = 40.5 \times 10^{-3}(1400 + 0.2 \times 4186) = 90.6\ \text{J}/(\text{m}^2\ \text{K})$.

The temperature rise, Eq. (4.326), is

$$\frac{dT}{dt} = \frac{1}{90.6}(250 \times 10^3 - 16.49 \times 10^{-3} \times 2354 \times 10^3) = 2330\ K/s.$$

The velocity of a coated paper web is 17 m/s, and the width of the IR drier is 0.4 m. Thus the delay time in one drier is $0.4/17 = 0.0235$ s. This yields an indication of the processes inside the drier, using the above calculated values

$$|\Delta u| \approx 0.41 \times 0.0235 = 0.01$$
$$\Delta T \approx 2330 \times 0.0235 = 55\ \text{K}$$

An accurate indication is achieved by carrying out the calculations in small time steps, such as $\Delta t = 0.004$ s, and then by calculating the vaporization, humidity change, and corresponding temperature rise at each time step. This is the numerical solution of differential Eqs. (4.326) and (4.328). The results of a calculation of this type are shown in Table 4.12.

4.3.10 Evaporation from a multicomponent liquid system

In many industrial processes the many components contained in the liquid evaporate simultaneously. Evaporation of individual components is easy to determine. For multicomponent liquid systems, the individual evaporation rates are summed to obtain the total evaporation rate.

Applying mass transfer theory to a component i in the liquid, assuming good mixing, and neglecting atmospheric concentrations, the evaporation molar rate of a single component can be expressed as

$$\frac{dn_i}{dt} = \frac{k_{G,i}A}{n_t} p_i^s n_i, \tag{4.331}$$

where $k_{G,i}$ is the mass transfer coefficient, A is the surface area of the liquid, n_t is the total moles of liquid, and p_i^s is the saturation vapor pressure of pure component i.

Integrating Eq. (4.331) and knowing that n_i^o is the initial number of moles of component i yield

$$n_i = n_i^o \exp(-K_i p_i^s t), \tag{4.332}$$

where

$$K_i = \frac{k_{G,i}A}{n_t}. \tag{4.333}$$

Eq. (4.333) assumes a constant ratio.

The total mass is

$$m_t = \sum_1^N n_i M_i. \tag{4.334}$$

The total evaporation mass flow rate is obtained as

$$-\dot{m}_t = \frac{dm_t}{dt} = \sum_1^N M_i \frac{dn_i}{dt} = \sum_1^N -K_i M_i p_i^s n_i^o \exp(-K_i p_i^s t). \tag{4.335}$$

Defining

$$m_t^o = n_t^o M_{av} \tag{4.336}$$

$$M_{av} = \sum_1^N x_i M_i \tag{4.337}$$

$$x_i^o = \frac{n_i^o}{n_t^o}, \tag{4.338}$$

we obtain the following

$$\frac{dm_t}{dt} = m_t^o \frac{\sum_1^N K_i M_i p_i^s x_i^o \exp(-K_i p_i^s t)}{\sum_1^N x_i^o M_i}, \tag{4.339}$$

where x_i is the mole fraction of component i in the liquid phase, and M is the molecular weight.

The partial pressure of the component i is obtained from

$$p_i = \frac{x_i^o p_i^s \exp(-K_i p_i^s t)}{\sum_1^N x_i^o \exp(-K_i p_i^s t)}. \quad (4.340)$$

For a nonideal liquid solution, multiplying Eq. (4.331) by the activity coefficient γ gives

$$\frac{dn_i}{dt} = \frac{k_{G,\ i} A}{n_t} p_i^s n_i \gamma_i, \quad (4.341)$$

For an ideal solution the activity coefficient is $\gamma_i = 1$.

$$k_{G,\ i} = \frac{h_{m,\ i}}{R\,T}, \quad (4.342)$$

where $h_{m,i}$ is the mass transfer coefficient in m/s, which is calculated from the heat and mass transfer analogy correlations, R is the universal gas constant, and T is the absolute temperature (K).

TABLE 4.12 Calculations for infrared drier.

Infra power = 250 kW/m^2		
Time (s)	**Temperature (°C)**	**Humidity (kg water/kg dry air)**
0.0040	47.58	0.2317
0.0080	57.32	0.2310
0.0120	66.05	0.2294
0.0160	73.57	0.2265
0.0200	79.59	0.2220
0.0240	83.83	0.2157
0.0280	77.01	0.2091
0.0320	72.21	0.2046
0.0360	68.43	0.2010
0.0400	65.31	0.1981
0.0440	62.64	0.1957
0.0480	60.34	0.1936
0.0520	58.32	0.1918
0.0560	56.53	0.1902
0.0600	54.93	0.1887
0.0640	53.50	0.1875
0.0680	52.22	0.1863
0.0720	51.05	0.1853
0.0760	49.99	0.1844
0.0800	49.03	0.1835
0.0840	48.16	0.1827
0.0880	47.36	0.1821
0.0920	46.63	0.1814
0.0960	45.96	0.1808
0.1000	45.34	0.1803
0.1040	44.78	0.1798
0.1080	44.26	0.1794
0.1120	43.78	0.1789
0.1160	43.33	0.1786
0.1200	42.93	0.1782

Note: Initial web temperature is 37°C and the initial humidity is 0.23. After the infrared drier ($t \geq 0.024$ s), there is a free draw, where the water vaporizes from the web to the surroundings ($x = 0.03$ kg H_2O/kg dry air), resulting in the cooling down of the web. The heat transfer factor in the drier and after the drier is $\alpha = 40$ W/m^2 K.

4.4 Water properties and treatment

4.4.1 Introduction

It is essential that the industrial ventilating engineer have a basic understanding of the properties of water and its treatment. This is to ensure an efficiently running and trouble-free plant. Additional to these issues are the problems relating to the discharge of contaminated water to the surrounding environment.

Water treatment over the past 100 years has grown into a complex science. It is of interest to note that in the 1880s a steamship left the port of Liverpool in the United Kingdom with instructions that the boiler water was to be treated with a mixture of cow dung and peat. A short time after leaving Liverpool, the ship's boiler exploded and the ship sank. It was not reported whether the explosion was due to the unusual method of water treatment.

4.4.2 Common water impurities

The following factors indicate the problems that poor-quality water may cause to the engineering plant and to human health:

- metal corrosion
- scale formation on the heat transfer surface
- dezincification
- plumbosolvency
- biological health hazards

Water supplies should never be assumed to be chemically pure. Groundwater from wells and springs contains dissolved impurities. Its properties depend on the nature of the ground over which it flows or passes through. Surface water from lakes and rivers contains

silt, dissolved impurities, and organic matter; its quality varies widely depending on flow rate.

Table 4.13 lists and briefly describes some of the impurities found in typical water supplies. As can be seen, a wide range of problems have been considered, and no one method of treatment is suitable for all cases. The pharmaceutical industry and silicon chip manufacturers require the greatest purity in the water used.

4.4.2.1 *Heavy metals*

Table 4.13 lists the most commonly encountered pollutants in water; however, in many industrial applications, various heavy metals are frequently combined in the process discharge.

Some heavy metals are essential to life at low concentrations but are dangerous to animal and plant life in higher concentrations. Generally, it is the free metal ion that is the most toxic; however, with Hg and Sn, certain organic forms have a greater toxicity.

The WHO, the CEN, and the Environmental Protection Agency in the United States specify the maximum permissible concentrations of these metals and other pollutants in the environment.

Due to the many variables involved, no attempt is made at this stage to cover the various methods used to remove these pollutants before the water is released into the environment. Table 4.14 lists the common heavy metals in water.

4.4.3 Cooling water systems

Cooling water systems used for process heat rejection can be classified under one of the following headings:

- open recirculation
- closed recirculation
- once-through system

4.4.3.1 *Open recirculation*

Open recirculation includes the standard cooling tower, spray pond, or evaporator condenser as shown in Fig. 4.39. An arrangement of this type provides an efficient cooling system. Its main disadvantage is the growth of microorganisms such as the *Legionella* species. To protect people from these bugs, the biological water treatment represents a very high cost in the operation of the plant. This arrangement is losing favor with many engineers and is being replaced by the less efficient closed systems.

Typical applications include heat rejection from the refrigeration plant. The highest proportion of cooling takes place by evaporation.

Advantages

1. It can cool water down to 2°C above the wet-bulb temperature.
2. Average temperature drop through tower in 10°C–18°C range depends on wet-bulb temperature.

Disadvantages

1. Corrosion due to absorption from the atmosphere of pollutants as the water droplets pass through the tower.
2. Fouling of surfaces, resulting in decreased heat transfer efficiency.
3. Scale buildup, resulting in a reduction of fluid flow through the heat exchanger and loss of effectiveness.
4. Microbiology problems (such as 2 and above 3) together with corrosion of materials and health hazards.
5. Decay problems in wooden cooling towers.
6. Spray water loss, resulting in costly additional water treatment for the makeup water
7. Spray drift may cause annoyance to people in its path, as well as corrosion of adjacent metals and concrete breakdown; improved design of drift eliminators available (in PVC) for critical control of drift.

The temperature difference between the recooled water temperature and the inlet air wet-bulb temperature is called the *approach*. The lower the approach, the more complex the tower's design becomes. The normally used minimum approach temperature is 2°C.

4.4.3.2 *Closed recirculation*

The arrangement for closed recirculation is shown in Fig. 4.40. It overcomes many of the problems encountered with the open system. Typical applications include engine cooling and heat rejection from refrigeration plant.

Advantages

1. Compared with the open system, the average cooling water temperature drop is small, only in the 6°C–8°C range.
2. Provided no water losses occur due to pipeline leaks, the cost of water treatment is at a minimum.

Disadvantages

1. Corrosion and fouling will occur, though normally considerably less than that experienced in the open system.

TABLE 4.13 Common water impurities.

Constituent	Formula	Problems caused	Methods of treatment
Alkalinity	Bicarbonates (HCO_3) Carbonates (CO_3) Hydroxyl (OH) as $CaCO_3$	Steam systems: foaming and solid carryover Steel embrittlement (HCO_3) and (CO_3) Corrosion	Distillation Demineralization Lime and lime soda Dealkalization (ion exchange) Acid treatment hydrogen zeolite
Ammonia	NH_3	Corrosion of copper and zinc alloys	Cation exchanger (hydrogen zeolite) Chlorinating Deaeration
Carbon dioxide	CO_2	Severe corrosion in condensate lines	Aeration Deaeration Alkalies (neutralization) Filming and neutralization amines
Chloride	Cl	Increases solid contents Produces a corrosive solution	Distillation Demineralization
Color	–	Boiler foaming Presents problems with iron removal Discoloration of manufactured product	Adsorption (activated carbon) Coagulation Filtration Chlorination
Conductance	μS	Due to ionizing solids in solution; an increase in conductivity occurs resulting in corrosive water	Reduce dissolved solids by lime softening or demineralization
Dissolved solids	–	Caused by evaporation, in steam generation resulting in blockage and foaming Frequent blowdown Loss of treated water Loss of heat	Lime softening Distillation Cation exchange. (hydrogen zeolite) Demineralization
Fluoride	F	Few major industrial water problems Reduces dental decay	Alum coagulation Magnesium Hydroxide reaction Anion exchange Membrane separation
Free mineral acids	HCl H_2SO_4	Corrosion	Any process using alkalines to neutralize
Hydrogen sulfide	H_2S	Corrosion (rotten egg smell)	Aeration Chlorination Ozone
Iron	Fe(II) (ferrous) Fe(III) (ferric)	Discolored water Deposits formed will foul surfaces	Coagulation and filtration Catalytic filtration Lime softening Aeration
Manganese	Mn(II)	See iron	See iron
Nitrate	NO_3	Increased solid content assists in the reduction of metal embrittlement Health problems with infants if used in foods	Distillation Demineralization
Oil		Scale Sludge Foaming Fouling of pipework and heat exchangers	Baffle separators Strainers Coagulation Diatomaceous earth
Oxygen	O_2	Severe metallic corrosion	Addition of corrosion inhibitors, sodium sulfite Automatic air vents, deaeration
pH	$pH = \log \frac{1}{H^+}$	Graded into acidic or alkaline water Scale 0–14:	Increased by alkaline addition Decreased by the addition of acid

(*Continued*)

TABLE 4.13 (Continued)

Constituent	Formula	Problems caused	Methods of treatment
		0 = highly acidic 7 = neutral 14 = highly alkaline	
Silica	SiO_2	Scale buildup on surrounding surfaces, reducing flow and heat transfer	Removed by applying hot magnesium salts Demineralization processes
Sulfate	SO_4	Increased solid content Combines with Ca to form calcium sulfate salt	Distillation Demineralization
Suspended solids	–	Clogs pipelines Fouls heat exchanger Surfaces	Settling Filtration
Total solids	–	Sum of dissolved and suspended solids	See dissolved and suspended solids

TABLE 4.14 Heavy metals.

Constituent	Chemical formula	Problems caused
Antimony	Sb	Moderately toxic
Arsenic	As	Highly toxic, corrosive, carcinogen
Beryllium	Be	High toxicity, long-term effects
Cadmium	Cd	Highly toxic, carcinogen
Chromium	Cr	Compounds may be highly toxic
Cobalt	Co	Moderate toxic
Copper	Cu	Highly toxic
Lead	Pb	A cumulative poison
Mercury	Hg	Very highly toxic
Molybdenum	Mo	Compounds are highly toxic
Nickel	Ni	Highly toxic, carcinogenic
Selenium	Se	Highly toxic
Silver	Ag	Low toxicity
Tellurium	Te	Highly toxic
Thallium	Tl	Sulfates highly toxic
Tin	Sn	Irritation of skin, eyes, lungs, and stomach
Titanium	Ti	Chlorides are moderately toxic
Uranium	U	Toxic, insoluble in water
Vanadium	Va	Oxides and chlorides have a high toxicity
Zinc	Zn	Moderately toxic, carcinogenic

4.4.3.3 *Once-through system*

This is the simplest of the three systems (see Fig. 4.41). Typical applications include:

- process water,
- potable water, and
- general service supplies.

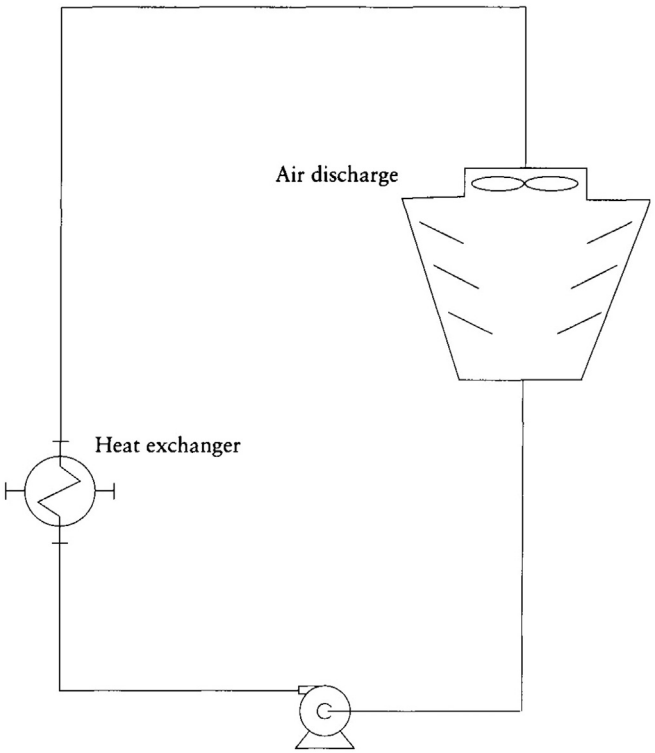


FIGURE 4.39 Cooling water system (open recirculation).

Advantages

1. low initial cost
2. space savings, small footprints

Disadvantages

1. low average temperature change, only in the 3°C–6°C range

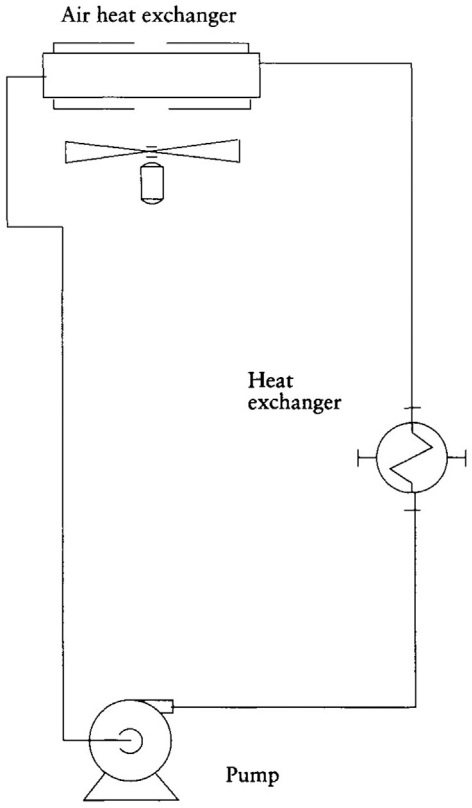


FIGURE 4.40 Cooling water system (closed recirculation).

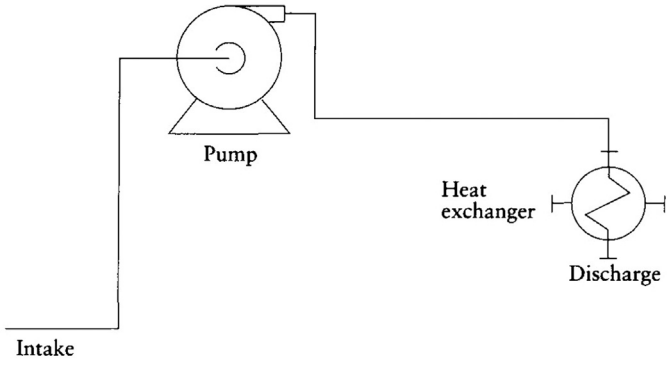


FIGURE 4.41 Cooling water system (once-through system).

2. expensive in water consumption, with large volume flow rates
3. sewer discharge costs
4. corrosion
5. fouling and scale problems
6. microbiological problems

4.4.4 Water treatment

In most engineering applications the supply water is not suitable for immediate use without treatment. It is essential that the method of water treatment selected be the one most suited to the application. If steam is used as the working medium for a process, it is essential that water treatment be used to prevent the precipitation of substances in the water from fouling pipework and heat exchangers; otherwise, costly plant damage will result.

The method of treatment selected depends on many factors, such as the nature of the salts and the pH of the water, and the assistance of a reputable specialist company is necessary to carry out regular testing and an analysis report.

4.4.4.1 Methods of feedwater treatment

The treatment may be internal, which involves the addition of chemicals to the water to make the salts causing scale and sludge less likely to bond to pipework and heat exchanger surfaces. In the case of boilers and cooling towers, blowdown of the water on a continuous or regular basis is required. Care must be taken in the case of blowdown to ensure that the exiting water temperature does not damage the drains or that the chemicals pollute waterways. The consent of the water authorities is required in order to determine what levels can be discharged into waterways. In certain cases, further treatment may be necessary before discharge.

In the case of boilers operating at low pressure, organic materials such as natural and modified tannins, starches, or alginates are added to aid blowdown. For boilers operating at high pressure, synthetic materials such as polyacrylates and polymethacrylates have been developed. The most commonly used chemicals for boiler feedwater treatment are phosphates and hydrazine.

External treatment involves the removal of impurities from the water by various methods before it enters the plant; this is the most effective method of water treatment. This category of treatment involves one or more of the following processes.

Sedimentation

In sedimentation the water to be treated flows slowly through a tank, allowing the suspended material in the water to fall to the base of the tank. The use of coagulating compounds, such as aluminum and ferric sulfate, increases the efficiency.

Oxidation

It is during oxidation that iron and manganese in suspension are removed from the water. Oxidizing agents (chlorine, ozone, hydrogen peroxide, potassium permanganate, etc.) or direct aeration is used; the metals in solution are converted to insoluble oxides that are removed by filtration. The use of gaseous

chlorine is not recommended if the water contains organics, or else carcinogenic by-products can be formed (trihalomethanes). Gaseous chlorine also presents health problems in the case of leaks and is corrosive.

Filtration

If the solution is allowed to flow through a granular bed such as sand, the larger particulate matter remains on the surface, while the smaller material is collected in the thickness of the granular bed. Pressurization of the filter accelerates the process. Besides sand, other materials used as filtering media are anthracites, manganese dioxide, and activated carbon.

Softening

The sense of touch allows one to determine if water is hard or soft. For a domestic application in a hard-water area, more soap is required to produce lather than is required in a soft-water area.

The temporary hardness salts can be removed by boiling; these salts may be classified as alkaline or carbonate hardness salts. These salts in solution are calcium carbonate, $CaCO_3$, calcium hydrogen carbonate, $Ca(HCO_3)_2$, and magnesium hydrogen carbonate, $Mg(HCO_3)_2$. Heating $Ca(HCO_3)_2$ produces water, carbon dioxide, and calcium carbonate, and this compound is deposited on heat exchangers.

The salts that cause permanent hardness are calcium sulfate, $CaSO_4$, calcium chloride, $CaCl_2$, magnesium sulfate, $MgSO_4$, and magnesium chloride, $MgCl_2$. These are known as nonalkaline or noncarbonate hardness salts and cannot be removed by boiling; they must be removed by chemical treatment.

The internal process complements the external process by taking care of any contamination that may enter the water from the process.

The following is a brief introduction to the various types of water softening plants encountered.

Lime soda. If carbon dioxide is in solution in water, and calcium hydroxide is added, the resulting precipitation product is $CaCO_3$; this can be removed by sedimentation.

If the water is temporarily hard due to the presence of $Ca(HCO_3)_2$, and calcium hydroxide is added, the resulting products will be a precipitate.

If the water is permanently hard due to $MgSO_4$, and lime is added, the precipitates calcium sulfate, $CaSO_4$, and magnesium hydroxide, $Mg(OH)_4$, result, which are removed by sedimentation.

Permanent hardness can also be due to the presence of $CaSO_4$, in which case the addition of soda (sodium carbonate), Na_2CO_3, produces sodium sulfate, Na_2SO_4, and calcium carbonate, $CaCO_3$; this precipitate once again is removed by sedimentation.

Ion exchange. When water flows through a resin ion-exchange material bed, some of the undesirable ions are adsorbed and replaced with less objectionable ones. The process may be either:

- base exchange,
- dealkalization, or
- demineralization.

The base-exchange process removes both the temporary and permanent hardness salts from the water by allowing the water to flow through resin beads containing sodium zeolite, Na_2Z.

When the permanent hardness salt $CaSO_4$ passes through the bed, calcium zeolite (CaZ) and sodium sulfate (Na_2SO_4) are formed, which are then flushed away.

A temporary hardness salt such as calcium carbonate ($CaCO_3$) passing through a sodium zeolite bed will produce calcium zeolite (CaZ) and sodium carbonate (Na_2CO_3). This solution is flushed away. But the temporary hardness salt calcium hydrogen carbonate, $Ca(HCO_3)_2$, passing through a sodium zeolite bed, will produce calcium zeolite and sodium hydrogen carbonate, $NaHCO_3$. The latter increases the alkalinity of water, causing foaming of the boiler water due to the formation of sodium hydroxide, NaOH. Similar reactions are possible involving magnesium chloride and sodium zeolite.

After a time, depending on the concentration of salts and the flow rate, the remaining sodium zeolite is converted to either calcium or magnesium zeolite. When the zeolite becomes saturated, the resin bed must be regenerated. The regeneration process is achieved by backwashing (flushing) the bed with fresh water to remove some of the remaining solids, followed by passing a solution of salt through the resin bed. This flushing removes the calcium chloride (CaCl) and the sodium zeolite; a final rinse removes any salt remaining, allowing the process to continue.

The dealkalization process removes the temporary hardness in water. This uses an acid resin bed for regeneration—in this case sulfuric acid (H_2SO_4).

To remove the majority of the salts from water, a mixture of resins is used; the process in this case is called demineralization.

Basically, the hardness salts of calcium and magnesium ions are exchanged for sodium ions in the dealkalization process; the carbonate and bicarbonate salts, which cause high levels of alkalinity, are replaced with chloride ions. Reverse osmosis can also be used to produce demineralized water.

Precipitation softening. This process depends on sufficient holdup time within a vessel to allow sedimentation and clarification to occur. A coagulation chemical such as alum or iron salts added to the solution will improve the process efficiency.

Evaporation. The process of evaporation or distillation in the past was carried out in submerged-tube evaporators. These have been superseded by flash-type evaporators that are more economical to run and reduce scale problems. The process is suitable for brackish water, where the cost of chemical methods is excessive. The resulting distilled water is not palatable and requires aeration to make it potable.

The operating cost is high due mainly to the energy used for the heat source, cooling water, and the necessary chemical treatment. To reduce the running costs, waste process heat or solar collectors are used.

Falling film vertical evaporators, direct expansion systems, and vacuum freezing techniques may also be used.

Reverse osmosis. The process of osmosis is used by plants to obtain food and moisture from the soil. The density of the sap in the roots of the plant is greater than that of the soil water surrounding it. The root wall provide a semipermeable membrane, and the difference in suction across it is the osmotic pressure.

In reverse osmosis the osmotic pressure is increased manually to get the water to flow from a high-density area through a semipermeable membrane to the lower density weaker solution. The water will pass through the membrane and leave the solids behind. A pressure of about 2.76 MPa will extract 90% or more of the dissolved absorbed solids; further refinement may be achieved through a base-exchange process.

Magnetic water treatment. This method is used in marine engineering and district heating networks in Russia. The hard water to be treated, either hot or cold, flows first through a filter and then at high velocity through permanent magnets. The magnetic field influences the nature of the crystallization of the hardness salts. This results in numerous nuclei being formed in the solution, creating sludge instead of a hard scale, which is easily removed by blowdown.

Deaeration

The removal of all gases in the water by means of traps or chambers will improve the pumping characteristics and reduce corrosion and noise.

In the case of hot water the oxygen in the water becomes about twice as corrosive for every 20°C increases in temperature; hence, the removal of the oxygen is of prime importance. Oxygen is extremely corrosive in hot water systems containing demineralized water.

In groundwater, gases such as carbon dioxide, hydrogen sulfide, and radon may be dissolved under pressure. For efficient removal an intensive degassing process (GDT, or Gas-Degas Technology) has been developed by the GDT Corporation (United States). It consists of groundwater and air (or ozone) being intensively mixed in a venturi injector, followed by optimum residence time in a reactor vessel, and finally the efficient removal of unwanted stripped gases in a centrifugal separator.

Oxygen scavenging

By removing oxygen completely, corrosion by this gas is eliminated. It can be achieved by the addition of sodium sulfite or hydrazine that reacts with oxygen. The reaction product will not normally cause any problems.

Scale control

Chemicals such as disodium or the polyphosphates are used to precipitate scale-forming solids in the water. If alkalinity control is required, caustic soda or soda ash is used in controlled amounts. For some boiler water, treatment-chelating agents are used to full advantage.

For a high-pressure boiler plant with a high evaporation rate, demineralized feedwater is classified as having an electrical conductivity of less than 0.2 μS/cm; for less critical plant conditions, an electrical conductivity greater than 0.2 μS/cm may be acceptable.

The water chemistry relating to power plants operating at high temperatures and pressures is a complex issue. To determine if water is corrosive or scale forming, use is made of the Langelier or Ryznar index. For further information, refer to the VGB guidelines for plants operating at pressures above 68 bar (VGB-450L) and the Scandinavian recommendations (DENA).

Sludge

Either straightforward drainage or blowdown can readily remove sludge from the plant. It is, however, necessary in some cases to ensure that the residual solids are free flowing; this is achieved by the use of tannin, lignin, seaweed derivatives, and starch organics.

Foam

Water level control and the use of organic antifoam chemicals are essential in steam plants in order to break down the bubbles at the water surface in steam systems, which cause foaming.

Condensate

Condensate formed in steam systems may require treatment to remove the carbon dioxide in suspension or free flowing in the condensate. Due to the nature of the water source, carbon dioxide and oxygen as dissolved gases are always present to some degree in water supplies, and in some instances hydrogen sulfide (H_2S) and ammonia (NH_3) may be present, producing a weak carbonic acid gas and causing

elevated-temperature corrosion of metals. The treatment in this case is achieved by the addition of chemicals, pH control, oxygen removal, etc.

When dealing with a large steam plant, the chemistry and the methods of water treatment required are complex.

The efficiency of water separation varies considerably from boiler to boiler. The purity of the steam supplied to a steam turbine should be checked. On the basis of the results, the maximum allowable salt concentration in the boiler water can be determined. This concentration may be much lower than the values given in the table.

When the heat load even locally exceeds 230 kW/m^2, the target values for drum pressure, 160 bar (except for SiO_2), should be used for all boiler pressures. For feedwater the recommended values for >67 bar should be used as follows:

1. The maximum *P*-value is independent of feedwater treatment.
2. The gauge pressure when using phosphates to reduce the residual hardness and when using a coordinated phosphate method for pH control falls in the pressure range of 35–90 bar PO_4 between 10 and 20 mg/kg and in the range 67–125 bar between 7 and 15 m/g.
3. γ_{25} = Conductivity of boiler water from a neutralized sample at 25°C.

Although the table is outside the scope of most industrial ventilating engineering requirements, it does indicate the many problems to be considered in the measurement techniques.

To finish this section, a typical flow diagram has been included (Fig. 4.42).

Biological factors

It is essential that the engineer not lose sight of the numerous potential problems related to microbiological concentration, which include the following:

- microbiological fouling in heat exchanger pipelines, cooling towers, etc.;
- microbiological corrosion in pipe work; and
- the effects of contaminated water on human health.

In the case of a closed water system, once the correct water treatment is provided, the incidence of microbiological fouling or corrosion is virtually eliminated, provided that the addition of fresh water is not a frequent occurrence. It is, however, essential to have water tests carried out at regular intervals by a water laboratory.

In the case of an open water system, the problem is compounded due to the addition of microorganisms from the atmosphere. Water temperature control is critical to stop the water from becoming a breeding soup culture for the microorganisms.

The aerosols formed in an open system, if inhaled, can cause various forms of *Legionella*. No one biocide is adequate to control these, as there are some 30 known groups, the most virulent being *Legionella pneumophila*.

It is essential to practice good design of all open systems by adhering to set guidelines. A well-planned and effective maintenance program is of prime importance.

The use of ozone for water treatment is now well established and has the following advantages:

- an efficient biocide;
- low owning and operating costs compared with other methods;
- no chemical handling, storage, or discharge problems; and
- simple methods of automatic control.

Ozone is more effective than chlorine in deactivating poliovirus, *Cryptosporidium parvum*, *Giardia lamblia*, and other protozoa. It also improves the color, taste, and odor of water dramatically. However, since no residual amount remains, it is always necessary to add a small amount of a more stable disinfectant as well (sodium hypochlorite, chlorine dioxide, etc.).

The disadvantages are as follows:

- High or medium initial cost. The energy cost for ozone synthesis is about 12–15 kW h/(kg O_3) consumed (1999), while 10–15 years ago, it was in the 25–30 kW h/(kg O_3) range.
- Limitations on water temperature and quality.

4.4.4.2 Separation techniques

Up to now, only the water treatment aspects relating to the efficient running of a plant have been covered. It is necessary to consider the discharge of the water from any cleaning process into the waterways or drainage systems, in order to ensure that statutory regulations are not violated. The following techniques are briefly considered:

- separation of a liquid from a solid
- separation of a solid from a liquid

The selection of the method used depends on many complex factors that are not covered in this chapter. The basics of each of the two methods are briefly covered. Only a few of these techniques would be used in the water treatment procedure encountered in the industrial ventilation field; however, for the sake of completeness, others are also covered.

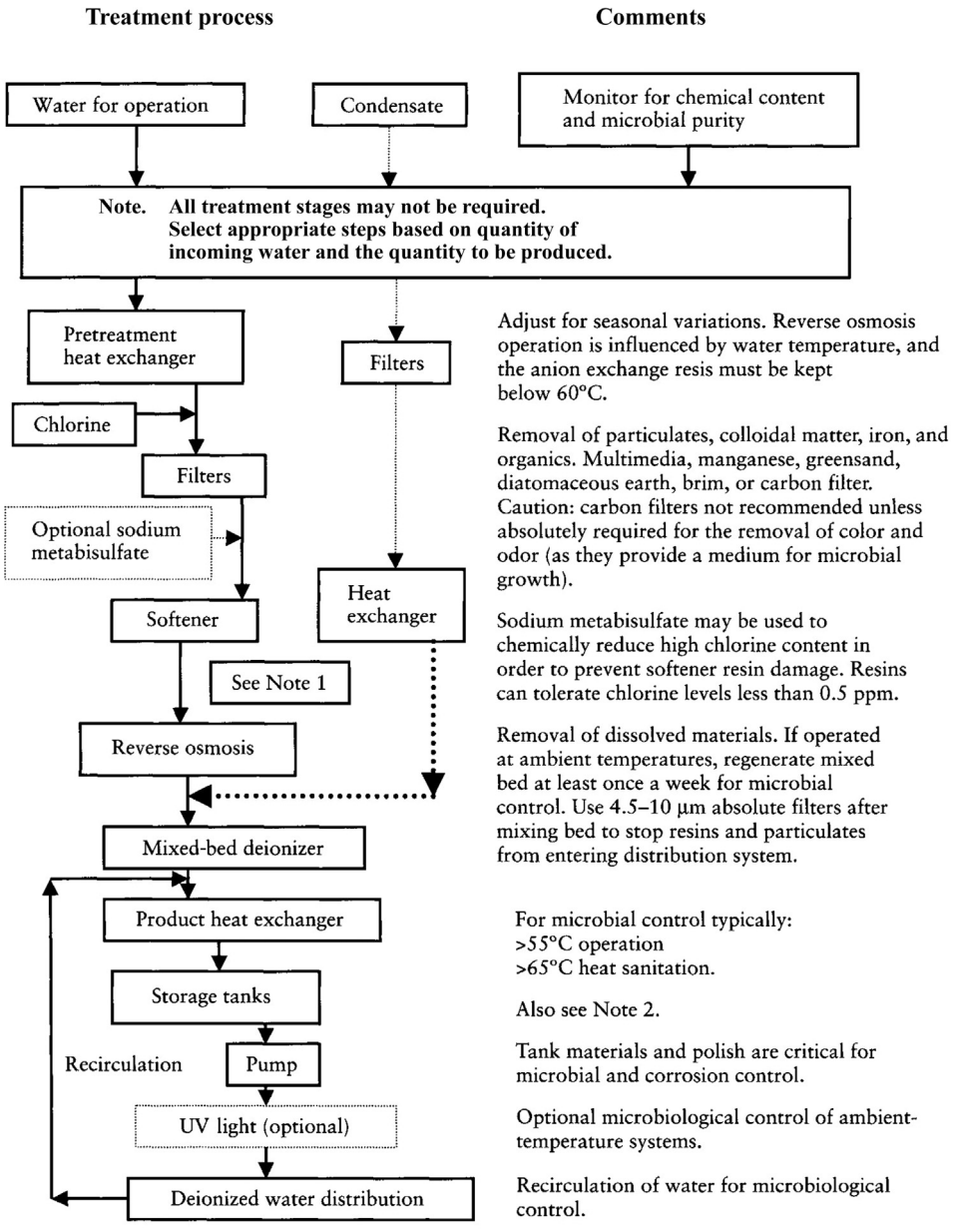


FIGURE 4.42 Typical flow diagram of water treatment.

Separation of a liquid from a solid

The removal of the various solid particulates from a gas stream may be achieved by washing either with chemicals or with water. Once this process has been carried out, the problem is to remove the solids from the liquid in order to:

- reuse the liquid,
- collect any valuable particulate matter for recycling, and
- ensure that the drainage system is not contaminated.

Numerous techniques may be used, each with disadvantages and advantages. The methods used are as follows:

- gravitational force
- centrifugal force
- application of a vacuum
- application of mechanical force

- action of a solvent
- displacement
- vaporization

In considering each of these techniques, the properties and principles on which they depend are related to the liquid only.

The solid in the liquid mixture must be considered as:

- nonvolatile,
- insoluble,
- unaffected by the treatment to which it is subjected,
- capable of being reused with a minimum of treatment,
- not causing any problems when discharged into the drains, and
- discharging into drains less than 40°C with pH in the 6–11 range.

Removal by gravitational force. This method simply involves a settling chamber in which the fact that the liquid–solid mixture–solid pore interface has little holding power. Thus given sufficient time, the solids will settle into the base of the chamber due to gravitational forces.

The efficiency of this operation can be improved by the use of finely perforated vee troughs, which will contain the particulate matter and allow the liquid to descend to the base of the container, where it then drains away either for further treatment or for reuse.

Removal by centrifugal force. This method is more efficient in extracting the particulate matter from a liquid due to the fact that the forces developed are many times greater than the force of gravity.

The machine used is normally the basket-type centrifuge that is a rotating perforated drum with a vertical axis. The solids remain in the drum and the liquid passes out through perforated holes. The smaller the holes, the greater the collection efficiency. However, there is the risk of hole clogging, causing a rapid fall in operating efficiency. The fluid viscosity and the particulate size are of prime importance. Plant arrangement in series using different-size perforations tends to overcome the clogging problems.

Vacuum removal. This approach is used in the paper industry for denaturing the paper, in which a vacuum is applied under the paper stock.

Mechanical force. Liquid can be readily expelled from a sponge-like particulate mass of solid by using various pressing techniques. With this method, mechanical energy is used to force the liquid containing the particulate matter through a porous bed. The particulate matter is held in the pores in the bed. When the pressure drop reaches a certain level, replacement or backwashing takes place. This process may be either intermittent or continuous.

Solvent action. Materials that tend to respond well to extraction by pressing will be more effective in solids removal when solvents are used. The complication is that it becomes necessary to separate not only the solids and the containing liquid from the finished process, but the solvent as well.

Displacement. This approach, which displaces one liquid from a solid mass by the introduction of another, is seldom used.

Vaporization methods. The abovementioned methods may be unacceptable on certain counts, and if complete removal of the liquid from the solid is required, vaporization methods are used. A nonvolatile solid can be removed from a volatile liquid by the application of heat, a vacuum, or both.

The various techniques can be classified as:

- heat applied at atmospheric pressure,
- heat applied at reduced pressure, and
- vapor distillation.

The dryers make use of warm air, flue gases, and direct radiant heat to the liquid-particle mixture. This method allows complete extraction of the solid through removal of the liquid by vaporization. Due to the energy input required with this method, it is the most costly.

Separation of a solid from a liquid

A solid in a liquid medium may be in either of the conditions:

1. The solid may be dissolved into the liquid medium.
2. It may be insoluble and remain suspended in the liquid.

The liquid is assumed to be saturated if undissolved material is present, with solids both in suspension and in solution in the liquid phase.

The efficiency of separation will obviously affect the purity of the liquid, and it may be necessary to provide a series of separate stages to meet the standard required by the specification.

The most important consideration is the actual condition of the solid in the liquid. Is it in solution, or is it in suspension? Other considerations are as follows:

1. The relationship of the solid to the liquid
   - **a.** Solids in solution in the liquid;
     - **i.** the solid concentration in the solution
     - **ii.** degree of solubility
     - **iii.** the relationship between temperature and solubility
     - **iv.** the viscosity of the liquid;
   - **b.** solid in suspension in the liquid
     - **i.** the relative amounts of solids in suspension
     - **ii.** the size of solid particulate matter

iii. the density of the particulate matter
iv. the compacting properties of the solid
v. the viscosity of the liquid medium
xvi. the nature of the solid—sponge-like, gritty, etc.

2. The degree of separation required
3. The heat-sensitive properties of the solid or liquid
4. Corrosive nature of solids and liquids
5. The degree of separation required for both the liquid and the solid
6. The value of the recovered solids or liquids
7. The volume of materials to be treated

The following list provides an indication of the various techniques on which the separating methods are based:

- relative vapor pressure of the solid and liquid
- the phase relationship between the solid and liquid
- the relative solubility of immiscible solvents
- reduction of solubility
- chemical precipitation
- ion exchange
- electrolytic deposition
- adsorptive properties

The separation of solids in suspension in liquids can be achieved by either of the following techniques:

- density difference,
- the cross-section of the solid particulate matter, or
- the electrostatic properties of the solid.

Next we briefly consider each of the above in turn.

Relative vapor pressure of solid and liquid. If the dissolved solids in a liquid have a low vapor pressure relative to the liquid in which they are dissolved, provided the solid is not affected by the liquid boiling point, it is an easy matter to vaporize the liquid, leaving a dry residue.

Phase relationship between the solid and liquid. A phase relationship may involve a number of crystalline forms from which materials can be separated. When a solid material is precipitated as a result of the solution becoming supersaturated, crystallization occurs. Crystallization may be achieved by:

- cooling alone,
- concentration,
- concentration followed by cooling, and
- simultaneous concentration and cooling.

Relative solubility of immiscible solvents. Many solid materials in solution can be removed by transferring them to a second solvent; it is essential that the solvents be mutually insoluble.

This approach will not produce a solid; it can, however, be used to remove a solid from one solution or solvent and transfer it to another, from which it can be readily removed.

Reduction of solubility. It is possible to remove a solid from a solution by changing the condition of a solvent. One method is the addition of a second solvent miscible with the first, in which the solid in solution is relatively insoluble. Another method depends on the fact that if the substance can be ionized, its solubility can be suppressed. This is achieved by adding a highly ionized second substance having one ion in common with the original dissolved solid.

Chemical precipitation. If physical separation techniques do not work, separation may be achieved by chemical conversion to a soluble precipitate.

Ion exchange. Certain solid substances have the property of exchanging one ion for another if placed in a solution containing the ions. Typical substances with this property are the zeolites and certain synthetic resins.

Electrolytic deposition. The separation of a metal from a solution can be achieved by electrolysis.

Adsorptive properties. Substances such as silica gel and activated charcoal can be used to collect (adsorb) certain solids from solution. The adsorber bed may be discarded when depleted or recycled by washing and heating.

Separation of solids in suspension in liquids. This can be achieved using:

1. density difference,
2. the difference in cross section between the solid particulate matter and the liquid molecule, and
3. the electrostatic properties of the solid.

Density techniques. Many methods are in use, with the selection depending on:

- particulate size,
- settling velocity, and
- quantity of solids in suspension.

In the case of gravitational settling, the unit design depends on the method of solid removal after settling. The methods in use are as follows:

- removal of solids from the separation unit in which they are contained,
- intermittent or continuous discharge of the solids from the base of the unit,
- continuous discharge over the rim of the settling tank, and
- separation by centrifugal force.

Gravity filters used in the density removal are subdivided into:

- sand gravity and
- continuous gravity.

Vacuum filters. If, due to the nature of the liquid, the gravity filter becomes unsuitable, a vacuum filter is used to create a substantial pressure difference. Vacuum filters can be divided into the following types:

- intermittent filters
- leaf filters
- continuous (e.g., rotary vacuum) filters
- vacuum pressure filters, as used in desulfurization plant
- pressure filters

In the true gravity case, pumps are not used. If, however, the liquid is highly viscous, to achieve efficient operation, pumps are required to force the fluid through the pressure filters. The pump can be considered essentially as a press with a plate-and-frame filter. The plate-and-frame filter consists of a series of frames over which the filter medium is stretched. A centrifugal basket of fine mesh is another method of particulate removal.

Dialysis. If a solution containing colloidal particle is placed on one side of a dialysis membrane, the water on the other side will allow the solution to be reduced in concentration as it passes through the membrane.

Electrostatic properties of solids in suspension. Some solids in suspension will migrate from one pole to another when placed between direct current electrodes. The phenomenon of solids moving toward an electrode is known as cataphoresis.

To close this section on treatment, two more methods that depend on bacteriological action are considered:

- aerobic treatment
- anaerobic treatment

Aerobic treatment. The activated sludge process depends on aerobic biological action. In this case the microorganisms, in searching for food, break down the complex organic substances into simple stable substances. This process results in the removal of soluble and suspended organic matter from wastewater.

The growth of microorganisms in the presence of dissolved oxygen removes the majority of pollutant matter; in turn, protozoa grow and feed on these organisms. The resulting balance is of a living culture in suspended form in the activated sludge floe. This process is ideally suited for the removal of carbonaceous matter and nitrification from wastewater.

The principal elements of the system include an aeration tank in which the wastewater is thoroughly mixed with continuously activated sludge and oxygen. From this part of the process, it passes into a clarifier tank, where the settled sludge is removed from the purified water to be recycled by the return activated sludge pumps.

For this system to work, two exacting requirements must be met. The aeration device must be capable of both transferring oxygen from the atmosphere to the liquid and distributing this oxygen throughout the wastewater to the suspended living microorganism. This type of system is suited for low-strength waste, typically on the order of 50–200 mg/L BOD.

To enhance the purification process and increase the degree of purification, powdered activated carbon (PAC) may be added directly to the aeration tank, or the biologically treated wastewater may be filtered through granulated activated carbon (GAC) for posttreatment.

Pre- or posttreatment with ozone of wastewater may also be applied. Pretreatment with ozone takes place in the presence of biorefractory compounds, as ozone increases the BOD/COD ratio.

Anaerobic treatment. Typical of this method is the upflow anaerobic sludge blanket. This consists of a corrosion-resistant tank complete with separators. The flow network enters the reactor base without short-circuiting, ensuring the proper formation of the granular sludge. New bacterial cells are formed in the reactor and aggregate into tiny granules, which have good settling characteristics.

Biogas is produced by the bacteria in the form of small bubbles; these float upward through the sludge bed/blanket, providing a good mixing action. When the biogas reaches the top of the reactor, it is collected and used as a fuel.

Design considerations

In determining the best method of treatment, the following factors have to be considered:

Owning and operating costs

- initial cost
- maintenance
- energy costs
- water treatment
- corrosion costs
- odor treatment
- abrasion problems
- slurry pumping problems
- the maximum temperature the drains and the water sinks can accept (bearing in mind thermal stresses and corrosion in discharge pipelines, and algae growth and oxygen depletion in the watercourses)

Properties of liquid used

- vapor pressure
- temperature of decomposition
- viscosity
- density

Properties of solid being extracted

- temperature of decomposition

- solubility
- state of subdivision
- surface absorptive properties
- elasticity

In some cases, fragility related to the dryness of the resultant solid, which will influence the removal technique:

Relationship between liquid and solid

- mechanically held liquid
- liquid absorbed on solid surfaces

Removal efficiency required. This depends on design requirements and current legislation.

Requirements for regeneration

- of liquid
- of solid
- of both liquid and solid

Equipment availability. In many cases the equipment is not available "off the shelf," and the delivery time may be lengthy; hence, adequate planning is necessary to ensure that the commissioning date can be met with the plant selected.

4.4.4.3 *Heat transfer fluids*

It is prudent at this stage to briefly consider the problems that can be experienced in either refrigeration or heat recovery systems when water treatment is required to prevent freezing. The antifreeze treatment of pure water may be achieved by various means, typical ones being various brines, ethylene glycol, and propylene glycol.

In the treating of water by any of these methods, it must be remembered that due to property changes, they can cause problems on both the heat transfer characteristics and fluid flow characteristics compared with pure water.

Many proprietary trade heat transfer fluids are in common use. Depending on operating temperatures, typical characteristics are as follows:

TABLE 4.15 Typical properties of some fluids.

Temperature (°C)	Solution	Solution by weight	Density (kg/m^3)	Specific heat [kJ/(kg °C)]	Thermal conductivity W/(m °C)	Viscosity (Pa s × 10^{-3})	Freezing point (°C)	Boiling point (°C)
−1	Sodium chloride	12	1092.6	3.6	0.485	2.2	−8.5	102.0
	Calcium chloride	12	1109.0	3.47	0.57	2.4	−7.2	101.0
	Methanol water	15	985.3	4.187	0.485	3.2	−10.3	86.0
	Ethylene water	20	977.2	4.35	0.47	5.3	−11.1	87.2
	Ethylene glycol	25	1036.5	3.85	0.52	3.7	−10.7	102.8
	Propylene glycol	30	1033.3	3.93	0.45	8.0	−10.6	102.2
−10	Sodium chloride	21	1166.3	3.35	0.43	4.2	−17.2	102.2
	Calcium chloride	20	1198.3	3.0	0.54	4.8	−17.2	101.1
	Methanol water	22	967.6	4.06	0.45	5.3	−15.5	83.3
	Ethylene water	25	977.2	4.27	0.43	8.3	−15.5	86.1
	Ethylene glycol	35	1057.3	3.60	0.48	6.8	−17.8	103.9
	Propylene glycol	40	1046.0	3.73	0.41	20.0	−4.2	103.3
Water at (0°C)			999.87	4.2174	0.569	1.792	0	100

- density from 600 to 1100 kg/m^3
- specific heat capacity from 1.12 to 2.75 kJ/(kg K)
- thermal conductivity from 0.1 to 0.29 W/(m K).
- boiling points up to 340°C at atmospheric pressure
- freezing points down to −15°C
- dynamic viscosity of 0.9×10^{-3} to 1.2×10^{-3} Pa s
- flash points from 108°C to 263°C

The influences of these factors have on fluid flow in the tubes are as follows:

- What may be turbulent flow in the heat exchanger for water will reduce to transitional or laminar flow for the heat transfer fluid, reducing the coefficient of heat transfer to a value 70% or more of that for water.
- Viscosity changes make conventional water pipe sizing tables useless; these must be upgraded by the application of appropriate correction factors.
- Density changes.
- Specific heat capacity changes.
- Thermal conductivity changes.

Table 4.15 gives typical properties of some fluids. For other fluids, contact the manufacturer for exact data. With heat transfer oils, care must be taken that chemical changes such as carbonization do not take place.

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CHAPTER

5

Physiological and toxicological considerations

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5.1 Thermal comfort

5.1.1 Introduction

Humans seek and want thermal comfort, even at work in industrial settings. Clothing, activities, posture, location, and shelter are chosen, adjusted, altered, and sought consciously and unconsciously to reduce discomforts and enable us to focus more on the other tasks of life. Discomfort can contribute to mistakes, productivity decreases, and industrial accidents.[1–3] Thermal discomfort results from the physiological strain of thermoregulation. The strain can be in the form of altered body temperatures, sweating and excessive skin moisture, muscle tension and stiffness, shivering, performance degradation, and loss of dexterity. A small amount of discomfort can sometimes enhance concentration and productivity by heightening arousal but too much discomfort is clearly detrimental.

Thus thermal comfort is clearly desirable and important to the well-being and productivity, and thereby the financial health, of industry. An understanding of the principles of thermal comfort and discomfort can help guide a designer's efforts in creating and operating industrial environments that are both energy-efficient and thermally acceptable to the occupants.

A commonly expressed definition[4] is "Thermal comfort is that condition of mind that expresses satisfaction with the thermal environment." The definition implies that the judgment of comfort is a mental process that results from physical, physiological, and psychological factors and processes. Dissatisfaction can lead to complaints and other undesirable side effects.

Manufacturing engineers, operators, and owners, of course, want to minimize complaints. A goal in the design process should be to recognize this objective and work to minimize discomfort from the outset. In general, designs that provide satisfying or acceptable environments will be financially more successful for the designer. That is, individual productivity will not be impaired by the environment, resulting in fewer accidents and lost time, fewer complaints, reduced employee turnover, and lower insurance costs.

5.1.1.1 Why one is comfortable? What affects our comfort?

Both primary factors and lesser secondary factors affect our sense of satisfaction with the thermal environment. The primary factors have significant reproducible effects and directly affect heat transfer and the

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occupant's thermal state. Secondary factors such as gender and age may affect ones's sense of thermal satisfaction. Individual differences in thermal comfort requirements between females and males as well as between young and elderly have widely been studied. It is reported that females and the elderly are more critical of indoor thermal environment and more sensitive to deviations of temperatures than males and the young. However, when the clothing and thermal state of the individuals are controlled, differences in thermal comfort diminish.[5] On the other hand, indoor climate standards for thermal comfort models, such as classical predictive mean vote[4,6] are based on an average male and his metabolic rate and therefore overestimates, for example, female metabolic rate.[7] This may cause either low thermal comfort for some individual occupants or nonenergy-efficient buildings. Other secondary factors such as circadian rhythm, physical disabilities, fitness, color and ambiance, local climate, sound, and food have been found to have impact on thermal comfort. These secondary factors have smaller to negligible effects on one's thermal state and will not be discussed here, but such information is available.[5,8]

5.1.2 Primary factors

Humans and the other warm-blooded animals have developed thermoregulatory systems to carefully control body temperature to levels that enable them to function and survive effectively. In general, thermal comfort occurs when the physiological effort to control body temperature is minimized for the activity. Table 5.1 illustrates that as conditions deviate from neutral the body activates mechanisms to stabilize body temperature. These efforts all result in small but noticeable and measurable increases in metabolism and physiological effort.

5.1.2.1 Body temperature

To maintain proper internal core temperature (T_c) near to 37°C, metabolic energy (M) must be continuously transferred to the environment.

TABLE 5.1 Thermal environment and physiological responses of thermoregulation.

Thermal environment	Physiological responses
Hot	↑ blood flow to skin (vasodilation), heart rate ↑, sweating ↑, skin moisture ↑, body temperatures ↑, and metabolism ↑
Neutral	Comfort, minimized effort, and T_{mb} (mean body temperature) ~ 36.2°C
Cold	↓ blood flow to skin (vasoconstriction), muscle tension and shivering ↑, body temperatures ↓, and metabolism ↑

Energy balance: metabolism − energy losses − mechanical work = rate of energy storage in body (S)

$$M - L - W = S\,dT/dt$$

where energy losses (L) = dry heat loss + evaporative heat loss. If $M - W > L$, then body T ↑—feel warmer. If $M - W < L$, then body T ↓—feel cooler.

The body temperature limits for health in terms of internal or core temperature is fairly limited. The limits are basically related to the function of nervous tissue. Hypothermic body temperatures around 28°C or less can result in cardiac fibrillation and arrest. In hyperthermia temperature of 43°C and greater can result in heat stroke, brain damage, and death. Often, too high temperature causes irreversible shape changes to the protein molecules of nervous tissue. That is, cooling overheated tissue to normal temperatures may not restore its original function.

5.1.2.2 Metabolism

Metabolism is often characterized by a convenient, relative, and dimensionless quantity called the metabolic equivalent, met unit (the ratio of the work metabolic rate to the resting metabolic rate). 1 met is defined as 58.2 W/m^2, which is equal to the rate of energy produced per unit surface area of an average person seated at rest. The surface area of an average person is 1.8 m^2 (ANSI/ASHRAE Standard 55).[4] Some met levels of various activities are listed in Table 5.2.

In some activities metabolic energy may be converted to useful work (force × distance). At steady state the rate of doing work P = force × distance/time and the thermal losses must balance with metabolism:

$$M = P + L \tag{5.1}$$

TABLE 5.2 Met level of various activities.

	Met
Reclining	~0.8
Seated and quiet	~1.0
Standing	~1.2
Standing and light activity (shopping, laboratory, and light industry)	~1.6
Standing and medium activity (house work and machine work)	~2
Walking 5 km/h	~3
Standing and heavy activity (heavy work, garage work)	~3
Basketball	~5–8
Max	~10–12

and if the rate of work is expressed as a thermal efficiency, $\eta = P/M$, then Eq. (5.2) simplifies to

$$M(1-\eta) = L \tag{5.2}$$

Example

Determine the met level of a person who bicycles up a 150-m high hill in 10 minutes. The person weighs 75 kg and is 182 cm tall. The bicycle weighs 10 kg.

$$\text{Work of cycling up the hill} = \text{force} \times \text{distance} = (75+10) \times 9.8 \times 150 = 124\,950 \text{ N m}.$$

The work is accomplished over a period of 10 minutes, so

$$P = 124,950/(10 \times 60) = 208 \text{ Nm/s} = 208 \text{ W}.$$

Cycling with the legs is rather efficient and it can be reasonably assumed that the thermal efficiency (η) is about 20%. Thus

$$M = P/\eta \cong 208/0.2 \cong 1040 \text{ W}.$$

This energy, normalized per unit of body surface area (M/A_D) where

$$A_D, = 0.202 m^{0.425} h^{0.725} = 0.202 \times 75^{0.425} \times 1.82^{0.725} = 1.95 \text{ m}^2,$$

is

$$M/A_D = 1040/1.95 = 533.3 \text{ W/m}^2.$$

Expressed in terms of met:

$$M/A_D = 533.3/58.2 = 9.2\text{met}.$$

Since this activity is greater than about 7 met, the effort of breathing may make it difficult to talk during the climb.

5.1.2.3 *Physiological temperature regulation*

For most situations and conditions in daily life, the human can be represented adequately by a simple model that is helpful for understanding human thermal regulation.[9] The model has two thermal compartments (Fig. 5.1). The compartments are characterized as having relatively uniform temperatures throughout. The bigger compartment (85%–95% of body weight) represents the body's core and contains all of the muscles and other significant heat- and energy-generating tissue. Blood profusion of the muscles and internal organs distributes the heat fairly well so the core can be represented as having an approximately uniform temperature (T_c). The smaller compartment represents the skin with uniform temperature T_{sk}. The temperature uniformity of this simple lumped parameter model is reasonable for people at sedentary to medium activities (0.7–5 met) in conditions where healthy people feel slightly cool to very hot.

Essentially all the energy produced in the body by the various metabolic activities is generated in the core. The skin functions as a protective and heat transfer surface for the core. As such, the skin, which is about 1.6 mm thick on average, has tissue with very small oxygen needs and heat-producing capabilities. The energy (M) produced by the core includes the extra heat generated by muscles in tensioning and shivering (Table 5.3) under active control for thermoregulation. In humans, shivering can increase metabolic heat production from around 1 to 3 met.

The metabolic energy production (M) of the body is lost by (1) doing work (energy released in terms of heat), (2) respiration, (3) passive heat conduction to the skin, and (4) active blood flow to the skin. Any heat not transferred from the core is stored, with a resulting increase in core temperature. Work is energy that leaves the body as in the raising of a weight or other thermodynamic work (force × distance) activities. Respiratory heat loss occurs from bringing ambient air into the core, raising its temperature to near core temperature, humidifying it to near saturation at core temperature, and exhaling it. The resulting heat loss is proportional to breathing rate and to the temperature and humidity differences. The breathing rate or air flow through the lungs is regulated mainly by CO_2 levels in the blood and as a result is proportional to metabolic rate.

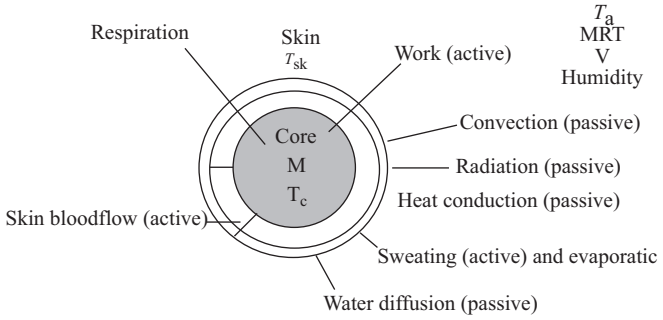


FIGURE 5.1 Simple representation of physiological temperature regulation in man.

TABLE 5.3 Active physiological controls: shivering, sweating, and skin blood flow.

Shivering = $K_{shiv} \times (T_{skset} - T_{sk}) \times (T_{cset} - T_c)$ W/m^2, $T_{skset} \cong 33.7°C$, $T_{cset} \cong 36.8°C$
Skin blood flow = BFN + $C_{dil} - (T_c - T_{cset})/(1 + \text{Str} - (T_{skser} - T_{sk}))$ L/(h m^2), where BFN is normal blood flow to skin for its metabolic needs. It is small 6.3 L(h m^2). SKBL↑ as T_c↑ > 36.8°C, SKBL↓ as T_{sk}↓ <33.7°C
Sweat = $K_{sw} \times (T_{mb} - T_{mbset})e - (T_{sk} - T_{skset})/10.7$ g/min/m^2, where $T_{bm} = \alpha T_{sk} + [1-\alpha]T_c$ and $\alpha \approx 0.1$.

The skin receives heat from the core by passive conduction and active skin blood flow (Table 5.3). It transfers this heat to the surroundings by convection, radiation, and evaporative (perspiration and diffusion) mechanisms. All of these mechanisms are unregulated or passive except evaporation from sweating. The sweating process is actively controlled by the human's thermoregulatory center where the rate of sweat secretion is proportional to elevations in core and skin temperature from respective set point temperatures (Table 5.3).

The physiologically active elements in body temperature regulation, summarized in Table 5.3, function and regulate in part on deviations in body temperatures from set points. In humans thermogenesis by shivering is small and inefficient in comparison to other animals. Thus the very precise regulation of body temperature in man is primarily due to only two active mechanisms associated with the skin: blood flow and sweating. Under normal comfort conditions, blood flow to skin is about 6 L/(h m^2) of skin. Of this about 1.5 L/(h m^2) is for the relatively constant minimal metabolic needs of the skin. In hot environments and during exercise skin blood flow can be increased by 15 times to about 90 L/(h m^2).[9] When necessary to reduce heat loss in cold environments, the vessels can restrict blood flow to as little as 1 L/(h m^2). With continued heat exposure, the thermoregulatory system increases its sensitivity so that blood flow increases with smaller and smaller changes in body temperature as the body acclimates to the hot environment.

Sweating, the other powerful heat loss mechanism actively regulated by the thermoregulatory center, is mostly developed in humans. With about 2.6 million sweat glands distributed over the skin and neutrally controlled, sweat secretion can vary from 0 to 1 L/(h m^2). The other, lesser, passive evaporative process of the skin is from the diffusion of water. The primary resistance to this flow is the stratum corneum or outermost 15 μm of the skin. The diffusion resistance of the skin is high in comparison to that of clothing and the boundary layer resistance and as a result makes water loss by diffusion fairly stable at about 500 g/day.

When the energy flows in and out of a compartment do not balance, the energy difference accumulates and the temperature increases or decreases. The changes in core and skin temperature then in turn alter the physiological control signals to restore balance and thermal stability.

5.1.3 Body control temperatures

Body temperatures are primarily sensed by temperature sensors in the hypothalamus near the center of the brain. Arterial blood flowing over and near the hypothalamus gives it information about the average thermal condition of the body. In addition, there is evidence that temperature sensors in the spinal cord and gut also give the hypothalamus core temperature information.[10] The skin has abundant numbers of warm and cold sensors that also communicate to the hypothalamus (Fig. 5.2).

5.1.3.1 Thermal sensation

The temperatures monitored in Fig. 5.2 are used by the brain to regulate shivering, blood flow to the skin, and sweating. The sensed temperatures also contribute to our overall feelings of warmth and other thermal sensations (TSs). TS can be predicted over a wide range of activities (0.8–4 met) from simple deviations in the mean body temperature (T_{mb}) and from the mean body temperature when the person feels neither warm nor cool but neutral (T_{mbn}) (Fig. 5.2).

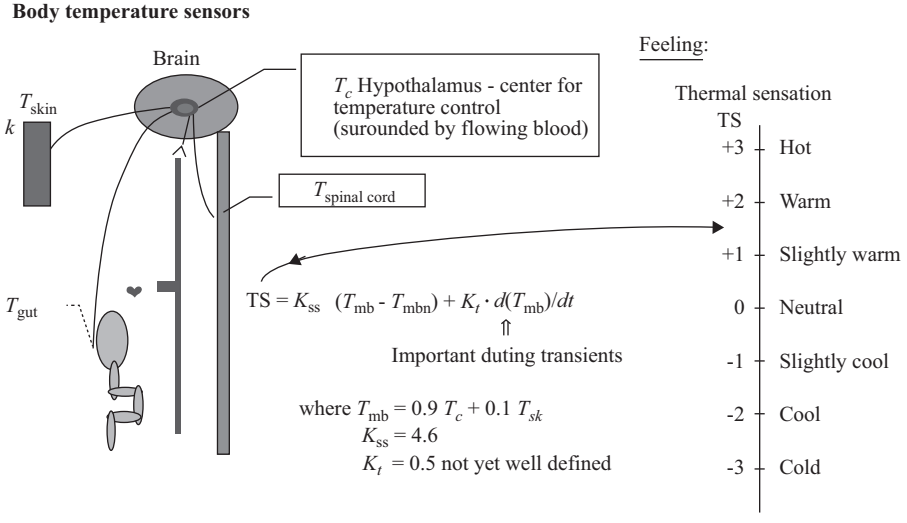


FIGURE 5.2 Temperature sensors for temperature regulation and thermal sensation.

The mean body temperature is a weighted average of core and skin temperatures, with core temperature being much more important:

$$T_{mb} = (1 - \alpha)T_c + \alpha T_{sk} \quad (5.3)$$

where α is weighting factor that depends on the skin blood flow. Estimates of α vary from 0.1 to 0.3 for vasodilated and vasoconstricted skin, respectively.[11]

That is as it should be as the purpose of the regulation system's operations is to maintain core temperature for the brain and other vital organs. The mean body temperature for a neutral TS is about 36.2°C. At temperatures above or below that, one feels progressively warmer or cooler, which further protects the individual by stimulating conscious behavioral actions to reduce physiological strain and restore neutral sensations. During transients the rate of change of mean body temperature can have a strong effect on TS.

5.1.3.2 *Body temperature sensors*

In most transient environmental situations, it is rapid changes in skin temperature that affect our feelings of warmth; rapid changes in core temperature only occur during rapid changes in metabolism and possibly during transient radar or other microwave exposures. Diving into cold water after a hot sauna is pleasant rather than cold because core temperature remains high and changes of T_{sk} reduces the hot TS.

In summary, core temperature is much more important than skin temperature in determining how warm we feel. Core temperature is affected by metabolic activity and heat storage. It is relatively isolated from the environment except through whole-body heat balance and resulting heat storage. Feet and hands have little metabolic heat generation themselves and depend on warm blood from the core for their temperature. The feeling of cold feet then means that the whole body heat balance has caused the core to lose temperature and the hypothalamus is restricting heat flow to the feet to stabilize the core temperature.

The consequence of the relationships of Table 5.3 and Fig. 5.2 is that for a neutral TS, at steady state, the core temperature increases while the skin temperature decreases with increased metabolic activity (Fig. 5.3). The increase in metabolism causes sweating which decreases skin temperature.

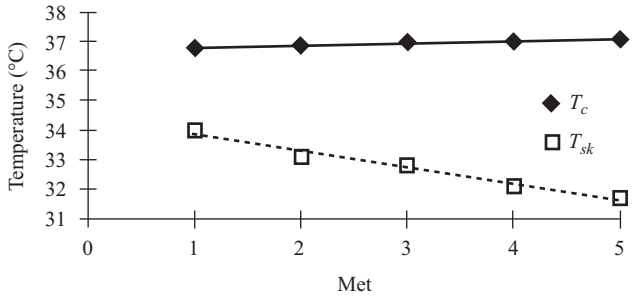


FIGURE 5.3 Schematic of skin (T_{sk}) and core (T_c) temperatures for a neutral thermal sensation.

5.1.4 Clothing

5.1.4.1 *Heat and moisture transfer in clothing*

Clothing hinders heat and moisture transfer between human body and environment. Thermal and moisture transfer occurs due to dry heat transportation by convection, conduction, and radiation and due to moisture transportation.[12,13] Higher thickness or number of layers of clothing increases insulating capability of the clothing and reduces body heat loss. Heat and moisture transfer occurs through pores of textile, fiber interior and surface, capillaries between fibers and yarns, and air between fabrics and yarns.[14] The mechanisms of heat and moisture transfer from the skin to environment are illustrated in Fig. 5.4.[11,15]

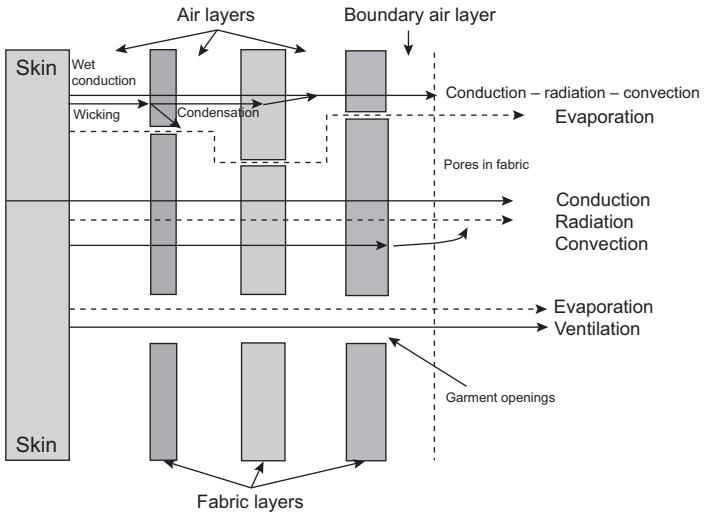


FIGURE 5.4 Heat and moisture transfer mechanisms from skin to environment. Source: *Modified from Parsons K. Human environments. In: The effects of hot, moderate, and cold environments on human health, comfort, and performance. 3rd ed. Boca Baton, London, New York: CRC Press, Taylor & Francis Group; 2002.*

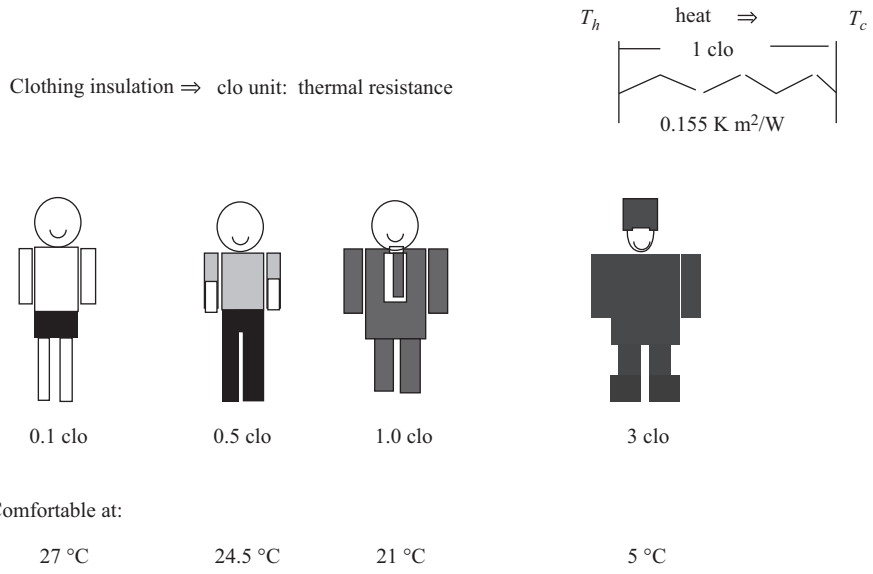


FIGURE 5.5 Some clothing ensembles with associated clo values and comfort temperatures.

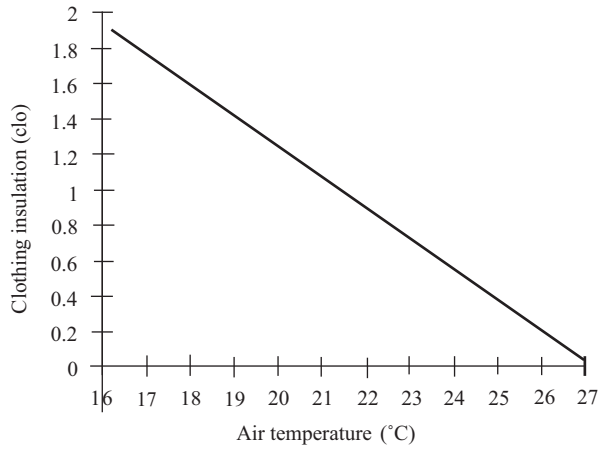


FIGURE 5.6 Clothing insulation necessary for neutral thermal sensation of sedentary persons (1 met) in a thermally uniform still-air environment with 50% relative humidity.[4] For higher activity levels the temperature at a clo level can be reduced about 1.4 °C per met increase.

5.1.4.2 *Thermal insulation*

Clothing insulation is usually described with the clo unit. Originally, 1 clo was defined as the thermal resistance necessary for comfort while sedentary in a uniform still air environment of 21°C. In conventional SI nomenclature 1 clo has a thermal resistance of 0.155 m^2K/W. Some ensembles' clo values and associated comfort temperatures are shown in Fig. 5.5.

The clothing insulation necessary for comfort or a neutral TS (TS = 0) in a thermally uniform 50% relative humidity (RH) still air environment is graphed in Fig. 5.6.[4] The slope of the graph is such that comfort temperature is decreased about 0.6°C for each 0.1 clo increase in clothing insulation. The graph is for 1 met but the line can be shifted to cooler temperatures for increased metabolism at the rate of −1.4K/met.

From Fig. 5.6, comfort is possible in still air from 18°C to 27°C by adjusting clothing insulation from 1.5 to 0 clo. This has significant building energy reducing potential with buildings only heated to 18°C and cooled to 27°C. However, personal, societal, and institutional preferences, norms, and codes usually limit the possible clo variation to a narrower range. For sedentary long-term comfort and because the hands are usually uncovered, the minimum practical temperature is about 20°C.[16]

Clothing insulation values are usually measured on heated manikins in specialized laboratories. The skin temperature of the manikin is controlled to about 33°C and the power input is measured (Fig. 5.7). Clothing insulation can also be similarly measured on humans with instruments to measure T_{sk}, ambient temperature (T_a), and dry heat flux from skin.

A very useful way to estimate clothing insulation is by summing the insulation values of the individual items worn (EN ISO 9920)[17]:

$$\text{Clothing insulation } (I_{cl}) = \sum I_{clu}. \qquad (5.4)$$

Some clothing item insulation values are listed in Table 5.4.[4,17] For example, the insulation value of a person wearing a thin shirt, thin trousers, underwear, shoes, and socks estimated by this method would be: 0.17 + 0.25 + 0.05 + 0.05 = 0.52 clo. If the person were to add a T-shirt under the shirt, the clothing insulation would be expected to increase to 0.6 clo.

Garment size, fit and thus, volume of the dry, and still air content in the clothing system increase the thermal insulation.[18–20] On the other hand, loose clothing allows air movement inside the clothing and thus to remove excessive heat due to convective heat loss.

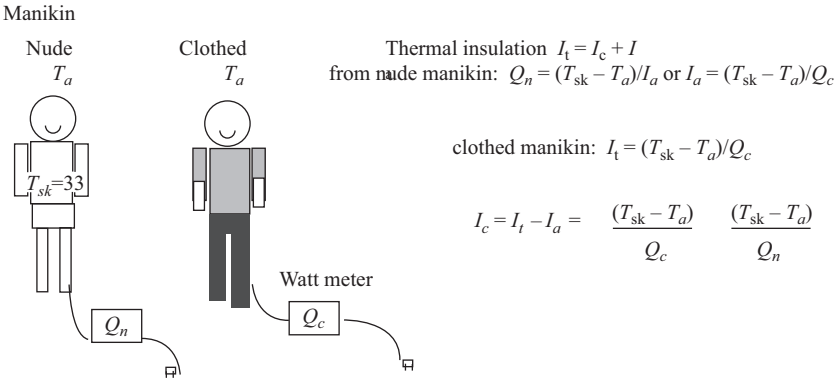


FIGURE 5.7 Clo value measured on heated manikins in a controlled environment.

TABLE 5.4 Insulation values of some individual clothing items.

Item	I_{clu}	Item	$I_{clu\ i}$
Trousers (thin)	0.15	Sweater (thin)	0.25
Trousers (thick)	0.24	Sweater (thick)	0.36
Sweat pants	0.28	Jacket (thin)	0.4
Overalls	0.30	Jacket (thick)	0.7
Coveralls	0.49	Sleeveless vest (thin)	0.13
Walking shorts	0.08	Sleeveless vest (thick)	0.22
Short-sleeved knit sport shirt	0.17	Sandals	0.02
Short-sleeved dress shirt	0.19	Shoes	0.03
Long-sleeved dress shirt	0.25	Boots	0.1
Long-sleeved flannel shirt	0.34	Ankle-length athletic socks	0.02
Long-sleeved sweatshirt	0.34	Calf-length socks	0.03
T-shirt	0.08	Long underwear (top)	0.2
Underwear	0.05	Long underwear (bottom)	0.15

5.1.4.3 *Effects of moisture on clothing*

Heat loss from wet clothing happens simultaneously due to moisture evaporation and dry heat loss. Moisture condensation into clothing depends on ambient temperature and saturation vapor pressure distribution within the clothing. Accumulated moisture in clothing increases conductivity of the materials and dry heat transportation through clothing.[21]

Moisture in textile material decreases thermal insulation in proportion to moisture retention by replacing air with water in the material, compressing garments and increasing material thermal conductivity.[12] Water (0.58 W/mK) has 24 times higher thermal conductivity than air (0.024 W/mK) at the same temperature (+25°C). Decrease in thermal insulation is also caused by enhanced heat conductivity.[22] When moisture content reaches about 15%, thermal insulation is about 50% of the dry thermal insulation value.[23] This phenomenon does not differ significantly between fiber materials; however, fiber materials have major differences in moisture absorption properties.

Water vapor resistance (R_{et}) describes material resistance to moist heat transfer through fabric. The R_{et} varies depending on fabric thickness and construction density, and both chemical and physical properties of fibers. The R_{et} of conventional clothing fabrics is about between 4 and 9 m^2Pa/W, and corresponding value of fabrics with semipermeable membranes is between about 9–20 m^2Pa/W.[24] Water vapor transfers from the inner side to the fabric surface due to fabric construction or garment holes (diffusion), fiber absorption and fiber surface.

5.1.4.4 *Effect of chairs on clothing insulation*

When a person is sitting, the chair generally has the effect of increasing clothing insulation (ΔI_{cls}) by up to 0.15 clo depending on the contact area (CSAC) between the chair and body. Specifically,

$$\Delta I_{cls} = 7.48 \times 10^5 \times \mathrm{CSAC} - 0.1 \ (\mathrm{clo}) \qquad (5.5)$$

where CSAC is the chair surface area contact in cm^2 or the surface area of the chair in contact with the human.[8,25]

For example, a desk chair with a body contact area of 2700 cm^2 has a ΔI_{cl} of 0.1 clo. This amount should be added to the insulation of the standing clothing ensemble (I_{cl}) to obtain the insulation of the ensemble when a person is sitting (I_{cls}) in the desk chair,

$$I_{cls} = I_{cl} + \Delta I_{cls}. \qquad (5.6)$$

5.1.4.5 *Effect of walking and air movement on clothing insulation*

Body motion and air movement generally increases the ventilation of garments, and thereby carries away heat and decreases the clothing ensemble's effective insulation. The increased airflow between the garment and the skin is due to a combination of increased air speed and the pumping action of the garment as it flexes during movement. As a result, walking

decreases clothing insulation. The change in clothing insulation (ΔI_{clw}) can be estimated from the standing intrinsic insulation of the ensemble (I_{cl}) and the walking speed (w) in steps per minute[8,26]:

$$\Delta I_{clw} = 0.504 \times I_{cl} + 0.0281 \times 10^{-3} \times w - 0.24 \text{ (clo)} \quad (5.7)$$

Thus the insulation of the walking person (I_{clw}) is found by subtracting the walking effect from the insulation of the standing clothing ensemble:

$$I_{clw} = I_{cl} + \Delta I_{clw} \quad \text{(clo)} \quad (5.8)$$

For example, the clothing insulation of a person wearing a winter business suit with a standing intrinsic insulation of 1 clo would decrease by 0.52 clo when the person walks at 90 steps per minute (about 3.7 km/h). Thus the ensemble's intrinsic insulation when walking would be 0.48 clo. More complete clothing tables and figures are available in the literature, for example, Chapter 8, Room Air Conditioning, of the *ASHRAE Handbook of Fundamentals*.[8]

5.1.5 Comfort zones

In general, when a person is thermally comfortable, the person's TS for the whole body is at or near neutral as depicted in Fig. 5.8A. As we have seen, the thermal conditions necessary for comfort are affected by clothing insulation. Fig. 5.8B shows the range of temperatures and humidities that are considered comfortable by ASHRAE Standard 55[4] for the typical summer and winter clothing levels of Fig. 5.8A.

In Fig. 5.8B, the comfort zone at 50% RH for the 0.9 clo winter clothing is from 20°C to 23.5°C and for the 0.5 clo summer clothing is from 22.5°C to 26°C. The temperature boundaries on the right and left sides of the comfort zones have constant ASHRAE Effective Temperature (ET*) levels. An ET* line is the locus of conditions that are calculated to have the same heat loss from the skin, skin temperature, and skin moisture levels. Since the physiology of the skin is the same for a constant ET* line, the TS and comfort judgments are also generally constant along the line. The temperature value of this line is the temperature where the RH is 50%. Along an ET* line the environment feels the same as it feels for air at temperature T at 50% RH. The ET* lines are not vertical but are affected by humidity and the human's physiological responses to the environment. Thus on the warm side of a comfort zone the humidity has more of an effect than on the cool side of the zone. But the differences in the slopes are not large and both indicate that temperature has a much stronger effect than humidity on the human thermal response. That is, the ET* lines show that for the same TS at a higher humidity the temperature must be lower. On average, for an 11°C increase in dew point

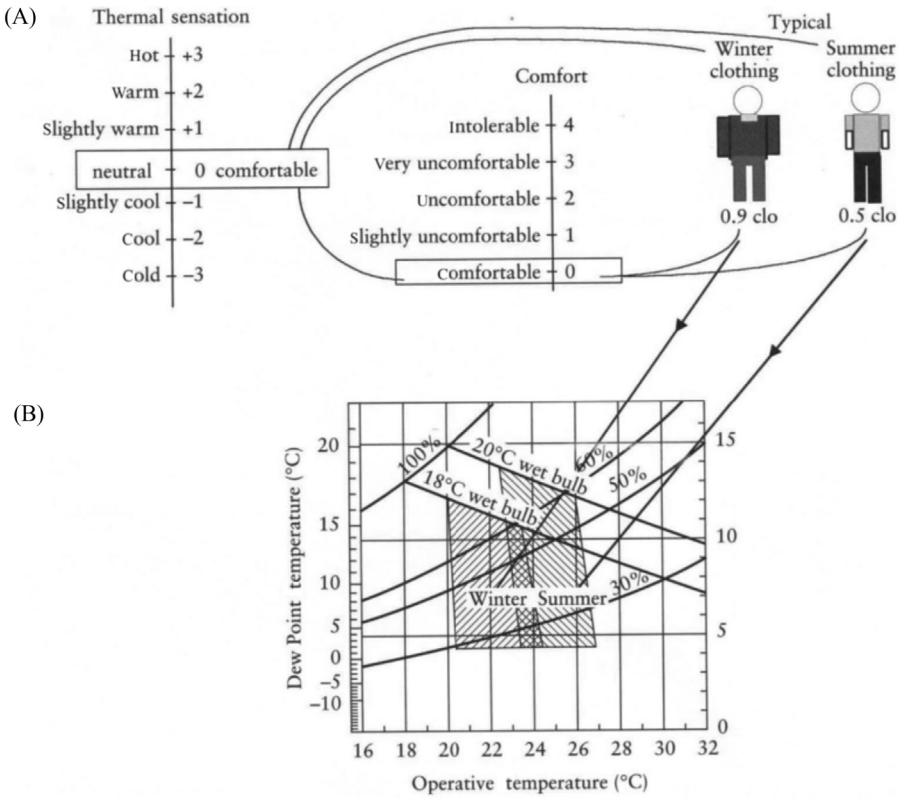


FIGURE 5.8 (*a*) Comfort and thermal sensations of a comfort zone. (*b*) Conditions for comfort on pyschometric chart for sedentary persons ($\leq$ 1.2 met).[4]

the temperature would need to be 1°C lower to have the same TS. In terms of human response the boundaries are not hard and sharp as indicated in Fig. 5.8B but instead are more soft and fuzzy in nature.

Optimum comfort would be in the center of each zone. Moving away from the center, some people would be expected to have TSs approaching −0.5 and +0.5 at the cooler and warmer ET* borders. The zones of Fig. 5.8B are for sedentary or slightly active ($M \leq 1.2$ met) people. If the activity level is higher than that, then the ET* line borders can be shifted about 1.4K lower per met of increased activity. Similarly, if the clothing is different than the 0.9 and 0.5 clo vales of Fig. 5.8A, the temperature boundaries can be decreased about 0.6K for each 0.1 clo increase in clothing insulation. Another, similar way to adjust the comfort zone for both different activity levels and clothing insulation values (I_{cl}) is to shift the zone centered on the optimum temperature ($T_{\text{sedentary}}$) at 50% RH as

$$T_{\text{active}} = T_{\text{sedentary}} - 3(1 + I_{cl}) \times (\text{met} - 1.2)\ (^{\circ}\text{C}) \quad (5.9)$$

Conditions that are warmer than the applicable still air comfort zone of Fig. 5.8B can often be made comfortable by increasing the air speed. If the conditions are 1°C–6°C warmer than the still air comfort zone of Fig. 5.8B, the necessary air speed (v) to restore thermal balance and comfort can be estimated from Fig. 5.9, where $T - T_{\text{comf}}$ is the temperature difference between the environment and the still air comfort temperature. It is stated that when air velocity is at least 0.15 m/s, increase of every 0.075 m/s of air movement is sensed by 1°C drop on the body.[27] Though the increased air speed will bring the whole-body TS to the comfort level, air motions above 0.8 m/s or so may cause other kinds of discomfort from moving papers, hair, and other light objects, and the pressure of the air speed itself may affect some people.

ANSI/ASHRAE Standard 55[4] has incorporated a model of adaptive thermal comfort [adaptive comfort standard (ACS)] for occupant-controlled naturally ventilated buildings (Fig. 5.10). The ACS prescribes a mean comfort zone band of 5K for 90% acceptance and another of 7K for 80% acceptance, both centered around the optimum comfort temperature (T_{comf}).

$$T_{\text{comf}} = 17.8 + 0.31\ T_{\text{air,out}} \quad (5.10)$$

Hypothesis of the adaptive comfort comprises that occupants of naturally ventilated buildings achieve thermal comfort in wider range of indoor temperatures than occupants with centrally controlled HVAC buildings.[3] Occupants' thermal responses in such spaces

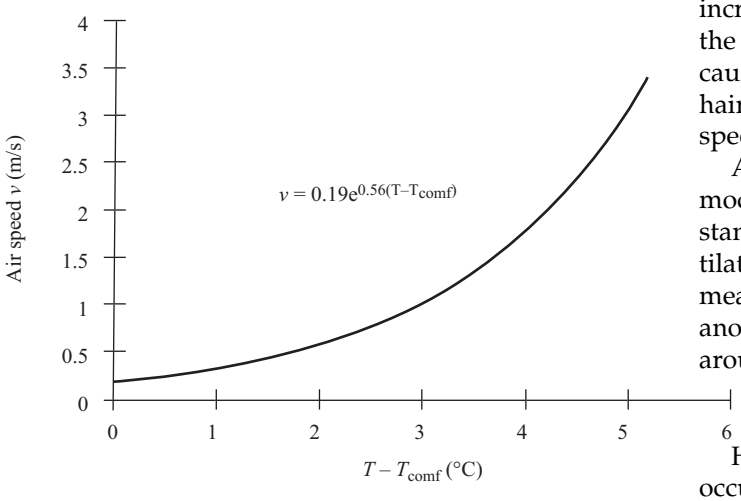


FIGURE 5.9 Air speed necessary at temperature T for the same thermal response as T_{comf} in a still-air environment (≤ 0.2 m/s).[4]

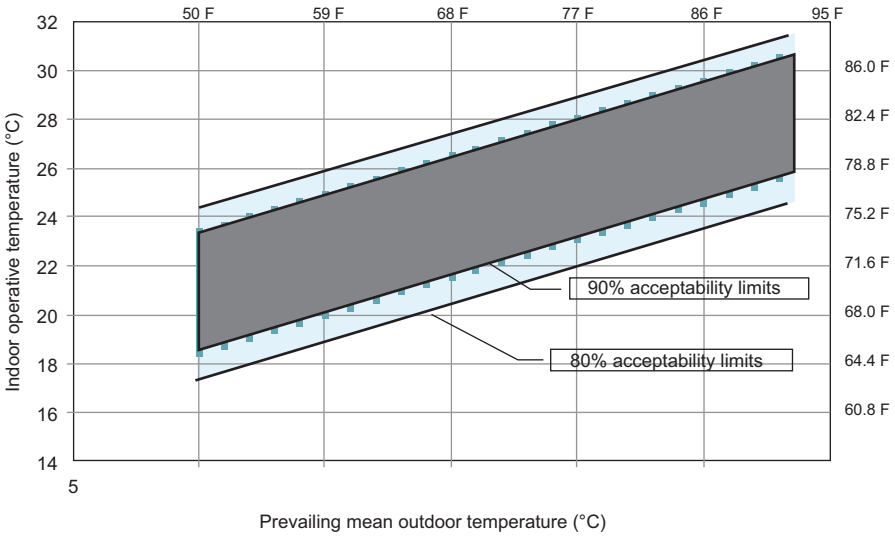


FIGURE 5.10 Acceptable operative temperature ranges for naturally conditioned space. Source: *Copied from ANSI/ASHRAE 55–2010.*

depend in part on the outdoor temperature (Fig. 5.10) and they can control their thermal environment through opening or closing windows. Occupants of naturally conditioned buildings seem to be more active in thermoregulatory adaptation by changing the activity level and clothing (behavioral adaptation) and appear more tolerant for a wider range of temperatures (psychological adaptation).[28] Occupants in air-conditioned buildings tend to adapt less and therefore their TS is more sensitive to changes in temperature.

5.1.5.1 Warm discomfort and skin moisture

In warm environments or situations with prolonged activities above about 1.2 met there is sweating. The sweat glands put water on the skin for evaporative cooling. Since the latent heat of evaporation of water is so high very little water is consumed in this cooling process. In the process the skin gets wet. If the conditions are very good for evaporation the skin can remain nearly dry while sweating occurs, as, for example, in windy desert conditions. In humid still air conditions a larger surface of water is necessary to evaporate the sweat and the skin becomes wetter. The fraction of the surface of the skin that is covered with water for evaporation is called skin wettedness (w_{sw}).[29,30] It is a measure of the physiological strain or effort of evaporative cooling and has long been associated with warm discomfort (Fig. 5.11).[31–34] It is rare that a person feels comfortable with a skin wettedness above 20%–25%.

Some of the discomfort of warm environments, the perception of skin moisture, and the interactions of clothing fabrics with the skin may be due to the moisture itself. The skin's outer layer of dead squamous cells of the stratum corneum can readily absorb or lose water. With moisture addition, the cells swell and soften. With drying, they shrink and become hard. In this setting the skin's moisture may be better indicated or characterized by the RH of the skin (RH_{sk}) rather than skin wettedness,[35]

$$RH_{sk} = P_m / P_{s,sk}, \tag{5.11}$$

where P_m is the average vapor pressure of the skin and $P_{s,sk}$ is the saturated vapor pressure of water at skin temperature. Typically, the water content (water/dry skin) of the stratum corneum is about 10% but it can absorb as much as four times its dry weight.

Skin moisture may be detected by mechanoreceptors of the skin and hair follicles or some other neural mechanism that senses the skin's swelling and shrinking. At high levels of skin moisture the swelling is sufficient to close or reduce the lumen of sweat glands and reduce sweating (called hydromeiosis). Hydromeiosis occurs at $RH_{sk} \geq 0.9$.[36] Conversely, under good drying conditions the skin can shrink to the extent that lesions form.

As mentioned previously, the other term for characterizing skin moisture is skin wettedness (w) or the size of the water film as a fraction of total skin area that is necessary to account for the observed evaporative heat loss from the skin (E_{sk}),

$$E_{sk} = w \times A_{du} \times h_e \times (P_{s,sk} - P_a), \tag{5.12}$$

where A_{du} is total skin area, h_e is evaporative heat transfer coefficient, and P_a is the ambient vapor pressure.

Skin wettedness and skin RH are related by

$$RH_{sk} = w_{sw} + (1 - w)(P_a / P_{s,sk}). \tag{5.13}$$

From Eq. (5.13) it is clear that RH_{sk} will be greater than w except when $w_{sw} = 1$. It is also evident that with a constant w_{sw}, RH_{sk} increases with ambient absolute humidity. Thus though the ET* temperature boundaries have constant skin wettedness levels, the RH_{sk},

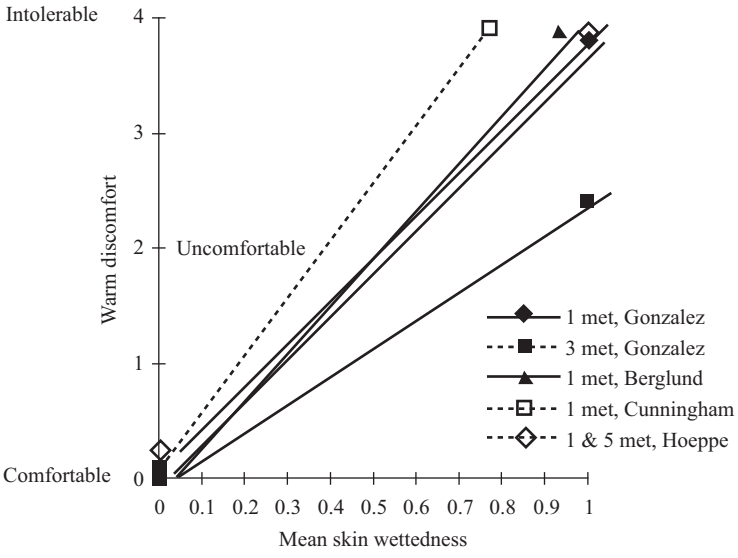


FIGURE 5.11 Warm discomfort related to skin wettedness from various studies. Refs. 31-34.

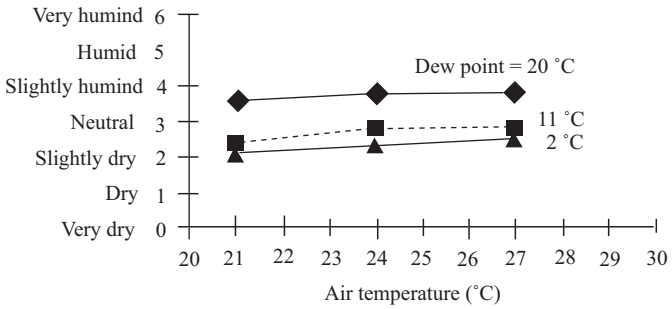


FIGURE 5.12 Perceived ambient humidity by sedentary subjects.

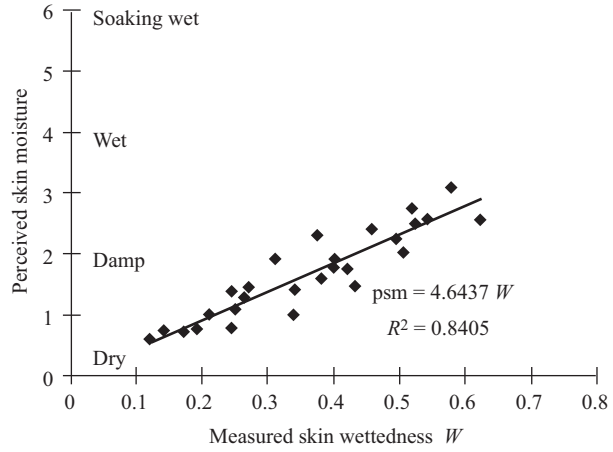


FIGURE 5.13 Perceived skin moisture correlated to measure skin wettedness for activities from 1 to 3 met.

swelling, and softening of the skin increase with increasing ambient absolute humidity.

Humans are sensitive to moisture and can reliably describe the humidity of the environment using word scales as demonstrated in Fig. 5.12.[37] The subject's humidity judgments appear to be functions of the air's dew point, a measure of absolute humidity, and are relatively unaffected by the ambient temperature. Furthermore people are also good at perceiving skin moisture, as illustrated in Fig. 5.13, where perceived skin wettedness is seen to correlate well with measured skin wettedness.

In situations of prolonged sweating, skin wettedness slowly increases with time because of accumulating salt on the skin. The increasing salt occurs because the water in perspiration evaporates while the dissolved materials, principally sodium chloride, remain on the surface. The salt lowers the vapor pressure of the sweat film, decreasing its rate of evaporation per unit area. The area of the film then naturally increases in the order that evaporation will equal the rate of sweat secretion. It is thought that part of the relief that bathing brings after a warm day or strenuous activity is that by cleaning the skin, perspiration can then evaporate more efficiently with reduced skin wettedness.

Clothing can be one of the detractors from acceptability in humid environments. Measurements by Gwosdow[38] reveal that the friction between skin and clothing increases abruptly for skin wettedness levels above 25%. Furthermore fabrics are perceived to be rougher or to have a coarser texture and to be less pleasant with increasing skin moisture. This may be one of the reasons that, in the comfort studies cited earlier, the people have rarely indicated they were comfortable when they had skin wettedness levels near and above 25%.

5.1.5.2 *Indoor humidity*

In general, a RH level between 30% and 60% is ideal for thermal comfort for human. Low indoor humidity level affects comfort and health. Comfort complaints about dry nose, throat, eyes, and skin occur in low-humidity conditions, typically when the dew point is less than 0°C. Too low humidity can lead to drying of the skin and mucous surfaces.[39] On respiratory surfaces, drying can concentrate mucus to the extent that ciliary clearance and phagocytic activities are reduced, increasing susceptibility to respiratory disease as well as discomfort. Green[40] quantified that respiratory illness and absenteeism increase in winter with decreasing humidity. He found that any increase in humidity from the low winter levels decreased absenteeism. Excessive drying of the skin can lead to lesions, skin roughness, and discomfort and impair the skin's protective functions. Dusty environments can further exacerbate low-humidity dry skin conditions.[41] Liviana et al.[42] found that drying from low humidity can contribute to eye irritation. Eye discomfort increased with time in low-humidity environments $T_{dp} < 2°C$.

Comfort is reduced by elevated humidity levels. It is recommended[43] that on the warm side of the comfort zone the RH should not exceed 60% to prevent warm discomfort. On the cool side of the comfort zone, high humidity is less important because there is no sweating to increase skin moisture. For these reasons the upper boundaries of comfort zones in Fig. 5.8B are wet bulb temperatures of 18°C and 20°C for the winter and summer comfort zones, respectively.

5.1.6 Spatial and temporal nonuniformity

The thermal parameters for comfort should be relatively uniform both spatially and temporally. Variations in heat flow from the body make the physiological temperature regulation more difficult. Nonuniform thermal conditions can lead to nonuniform skin temperatures. The active elements of the regulatory system may need to make more adjustments and work harder to keep thermal skin and body temperatures stable. To avoid discomfort from

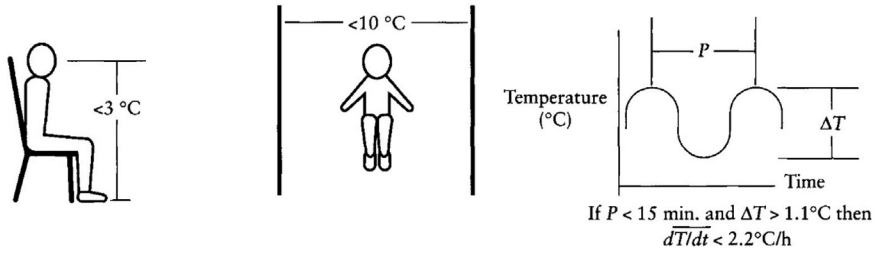


FIGURE 5.14 Nonuniformity limits to avoid discomfort.

environmental nonuniformities, the temperature difference between feet and head should be less than about 3°C (Fig. 5.14), the mean surface temperature or radiant difference from one side of the body to the other should not be greater than about 10°C and from a warm (heated) ceiling less than 5°C.[6]

Similarly, with cycling temperatures, large fast cycles can cause discomfort. To avoid this, if the time to complete one cycle is less than 15 minutes and the peak-to-peak temperature variation is greater than 1.1°C, the average rate of temperature change should be less than 2.2°C/h (Fig. 5.14). Very slow rates of temperature change ($dT/dt < 0.5$°C/h) are much less difficult to adjust to and the change can go unnoticed until the temperature is beyond the comfort zone temperature.

Local air motion is another thermal nonuniformity that can cause a local cooling of the skin and the feeling of a draft. Draft discomfort from local air motion ($\overline{v}$) increases as the air temperature (T_a) decreases below skin temperature. Draught rating (DR) can be estimated according to ISO 7730:2005[6] as

$$\mathrm{DR} = (34 - T_a)(\overline{v} - 0.05)^{0.62}(0.37\,\overline{v}\,\mathrm{Tu} + 3.14) \qquad (5.14)$$

Fluctuations in the local air motion increase the perception of drafts and should be avoided. The unsteadiness of air motion is often described in terms of its turbulence intensity (Tu):

$$\mathrm{Tu} = \mathrm{SD}_v/\overline{v}, \qquad (5.15)$$

where $\overline{v}$ is the average air speed of the draft and SD_v is its standard deviation. In spaces with forced air systems, the turbulence intensity is typically between 0.3 and 0.6.[44] That level of turbulent intensity generally limits maximum air speeds to <0.2 m/s for occupants in cool environments.[45] However, in warm environments turbulence intensity is desirable as it increases the cooling effectiveness of the air motion.[46]

5.1.7 Thermal radiation and operative temperature

In buildings away from outside perimeter walls, air and surface temperatures are usually approximately equal. The heat losses from a person by radiation (q_r) and convection (q_c) are then flowing to the same temperature level. In such uniform spaces, the radiant and convective losses are about equal and together account for about 80%–90% of the total heat loss of a sedentary comfortable individual. In the presence of hot or cold surfaces, as may occur in perimeter or other locations in a building, the average surface temperature of the surroundings [called mean radiant temperature (MRT)] as seen by the person's body may be substantially different from air temperature. If the MRT is greater or less than air temperature (T_a) the person will feel warmer or colder than in a thermally uniform space where MRT = T_a.

To simplify the effects of radiation and convection on dry heat transfer, the concept of operative temperature is often used. By definition operative temperature is the temperature of a uniform environment (T_a = MRT) that has the same total dry heat loss (convection + radiation) as the actual environment where $T_a \neq$ MRT.

Dry heat losses (q_{dry}) from the person's surface at temperature T_s can be expressed as

$$q_{\mathrm{dry}} = q_c + q_r = h_c \times (T_s - T_a) + h_r \times (T_s - \mathrm{MRT}), \qquad (5.16)$$

where the convective (h_c) and linearized radiation (h_r) heat transfer coefficients are

$$h_c = 8.5\nu^{0.5}\ \ \mathrm{W/m^2K\ with\ } \nu \mathrm{\ in\ m/s} \qquad (5.17)$$

and

$$h_r = 4e\sigma(A_r/A_D)[273.2 + (T_s + \mathrm{MRT})/2]^3\ \ (\mathrm{W/m^2K}), \qquad (5.18)$$

where e is the emissivity of clothing-body surface ≈ 0.9, σ is the Stefan–Boltzmann constant, 5.67×10^{-8} W/m^2 K, and A_r is the effective radiation area of body, m^2 $(A_r/A_D) \approx 0.7$.

A_r is less than the skin area A_D because some of the skin of fingers, arms, legs, and feet radiates to other skin and is not as effective for radiant heat loss. Eq. (5.16) can be rearranged to

$$q_{\mathrm{dry}} = (h_c + h_r) \times [T_s + T_o] \qquad (5.19)$$

where T_o is the operative temperature,[8] evaluated as $T_o = [h_c \times T_a + h_r \times T_r]/(h_c + h_r)$. The equation shows that operative temperature is the average of air and MRTs weighted by their respective heat transfer

coefficients. It is the temperature of a uniform environment that physically and mathematically represents the actual environment. Fortunately, for the low air speeds ($v < 0.25$ m/s) of most indoor environments $h_c \cong h_r$ and operative temperature becomes the simple average of the air and MRTs,

$$T_o \cong (T_a + T_r)/2. \tag{5.20}$$

At higher air speeds $h_c > h_r$, convective heat loss becomes greater than radiation and T_o approaches T_a. For such conditions Eq. (5.21) is recommended[4]:

$$T_o = AT_a + (1 - A)\text{MRT}, \tag{5.21}$$

where A depends on air speed (v):

v (m/s)	0–0.2	0.2–0.6	0.6–1.0
A	0.5	0.6	0.7

The above equation indicates that to maintain a constant level of comfort when MRT decreases, T_a must be increased an equal amount. This is the difficulty of perimeter zones. In many such environments the air and surface temperatures differ and operative temperature is a convenient way to characterize the environment.

How is MRT determined? One could calculate or measure the surface temperatures of the room and calculate MRT from

$$\text{MRT} = (F_{p-w} \times T_{rw}^4 + F_{p-f} \times T_{rf}^4 + F_{p-c} \times T_{rc}^4) + ++ [F_{p-n} \times T_{rn}^4]^{1/4}, \tag{5.22}$$

where T_{rn} is the absolute temperature (K) of the radiating surface n and F_{p-n} is the angle factor from the person to surface n,[4,47] and F_{p-n} is the fraction of radiation leaving p that strikes n.

If the surface temperatures are not widely different, Eq. (5.22) can be simplified to

$$\text{MRT} = (F_{p-w} \times T_{rw} + F_{p-f} \times T_{rf} + F_{p-c} \times T_{rc}) + ++ F_{p-n} + T_{rn}. \tag{5.23}$$

At a location MRT and T_o are often measured with a sphere or ellipsoid representing the person, as shown in Fig. 5.15. In the diagram the energy balance on the globe at steady state is $q_c = q_r$, or

$$h_c \times (T_g - T_a) = h_r \times (\text{MKT} - T_g), \tag{5.24}$$

and after rearranging:

$$\text{MRT} = T_g + [h_c/h_r] \times (T_g - T_a] \tag{5.25}$$

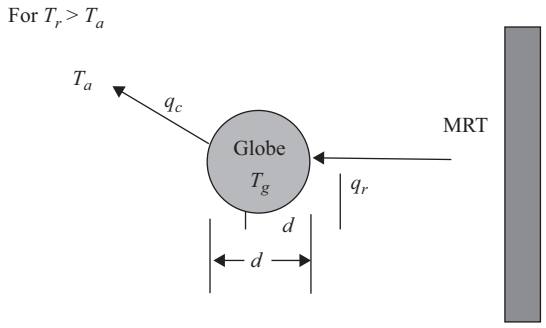


FIGURE 5.15 The determination of MRT.

Substituting numerical values for h_c and h_r with $d = 15$ cm and v in m/s,

$$\text{MRT} = T_g + 0.247\, v(T_g - T_a). \tag{5.26}$$

Furthermore from the definition of operative temperature (T_o),

$$T_o = (h_c \times T_a + h_r \times \text{MRT})/(h_c + h_r). \tag{5.27}$$

Substituting Eq. (5.25) into Eq. (5.27) and rearranging,

$$T_o = \frac{T_a + [h_r + h_c]_h \times T_g + [h_r/h_c]_h \times [h_c + h_r]_g \times (T_g - T_a)}{1 + [h_r/h_c]_h}, \tag{5.28}$$

where subscripts h and g designate the human and globe. For 15–20 cm diameter globes $[h_r/h_c]_g \cong [h_r/h_c]_h$, which after substituting and rearranging Eq. (5.28) simplifies to

$$T_o = T_g. \tag{5.29}$$

Globes can be made of any opaque material. A globe of low mass is helpful to provide a short-time constant for transient conditions. Globes are typically gray or black, but color is not important if they do not receive high temperature radiation from the sun or other glowing objects. If significant high-temperature radiation is present, then they should have a color similar to that of the occupant. The comfort zones of Fig. 5.8B should be entered with T_o when it is known that MRT$\neq T_a$ because T_o is the temperature that the environment feels like to the occupant of the space.

5.1.8 Future perspectives

Global warming should be recognized when assessing human thermal comfort. It is likely that global warming affects the indoor temperature and challenges individual to achieve thermal comfort and thermal balance. According to the Intergovernmental Panel on Climate Change (IPCC) 2018 report[48] human-induced global warming reached approximately 1°C

(range of 0.8°C−1.2°C) above preindustrial levels in 2017, increasing at 0.2°C per decade. Global warming is likely to reach 1.5°C between 2030 and 2052 if it continues to increase at the current rate. Global warming of 1.5°C or more is predicted to increase mean temperature in most land and ocean regions, hot extremes, and heavy precipitation and probability drought in most or several regions. Any increase in global temperature (e.g., +0.5°C) is projected to affect human health, with primarily negative consequences, and heat-related morbidity and mortality. Urban heat islands often amplify the impacts of heatwaves in cities.

Modern technology, such as wearable devices and artificial intelligence, will provide opportunities to shift from centralized to individualized air conditioning strategies in the built environment. The individualized air conditioning is most beneficial for particularly sensitive occupants because their requirements for thermal comfort will also be able to achieve.

5.2 Human respiratory tract physiology

5.2.1 Introduction

Industrial environments expose individuals to a plethora of airborne chemical compounds in the form of vapors, aerosols, or biphasic mixtures of both. These atmospheric contaminants primarily interface with two body surfaces: the respiratory tract and the skin. Between these two routes of systemic exposure to airborne chemicals (inhalation and transdermal absorption) the respiratory tract has the larger surface area and a much greater percentage of this surface exposed to the ambient environment. Ordinary work clothing generally restricts skin exposures to the arms, neck, and head, and special protective clothing ensembles further limit or totally eliminate skin exposures, but breathing exposes much of the airway to contaminants.

Inhaling potentially noxious airborne mixtures exposes respiratory tissue and the supporting vasculature to disease and injury. In addition, other organs can be injured due to transepithelial transport along the airway to the bloodstream and subsequent bulk transport throughout the body. Consequently, understanding the relationship between industrial ventilation and human health requires knowledge of how the respiratory tract interacts with the surrounding environment. It is the goal of this chapter to lay the groundwork for understanding how the human airway deals with potential airborne threats.

5.2.2 Anatomical overview

The human respiratory tract serves to deliver oxygen to the bloodstream and remove carbon dioxide. It accomplishes this by utilizing two large air bags (lungs) with extremely large internal surface areas to transport these gases between the pulmonary airstream and capillaries. The lungs are situated inside a semirigid bony structure (rib cage), which is joined together by intercostal muscles and supported from below by a large sheet of muscle tissue (diaphragm). These structures serve to physically protect the lungs and generate the forces required for inspiration and exhalation. There are thin membranes immediately surrounding the lungs (visceral pleura) as well as lining the rib cage (parietal pleura), between which forms the pleural cavity with a small layer of pleural fluid with low protein concentration making the pleural leaves slippery for the lungs to move freely. Pleura is penetrated by numerous blood vessels, and in infectious disorders or heart failure there may be an excess leakage of fluid into the pleural space. However, normally pleural leaves are close to each other, and they couple the lungs to the chest wall and act in transferring the force generated by the diaphragm and intercostal muscles to the lungs.

The respiratory tract can be theoretically subdivided into distinct functional regions (Fig. 5.16). Dividing the respiratory tract into conducting and respiratory airways is perhaps the simplest division. Framed in this way, the respiratory tract consists of two airway regions: a series of tubes (nasal and oral cavities, pharynx, larynx, trachea, bronchi, and nonalveolated bronchioles) leading to a terminal region of essentially bag-like structures (respiratory bronchioles and alveoli), where gas is exchanged between the airway lumen and the surrounding capillaries.

Extrathoracic airways (upper airways) comprise all airway structures proximal to the larynx. Fig. 5.16A shows this to include the nasal passages, nasopharynx, oral cavity, oropharynx, pharynx, and larynx. These structures have the functions of removing gross contaminants from the inspired airstream, humidifying and warming inspired air, and primary recovery of whatever heat and humidity can be retained from expired air. Nasal and other extrathoracic deposition of inhaled substances may further lead to their lymphatic or gastrointestinal deposits as function of the extrathoracic clearance of airways.[49]

The intrathoracic tracheobronchial tree (lower airways) consists of a straight tube (trachea) terminating in a series of bifurcating tubes, which subsequently terminate at the pulmonary airways. The trachea, bronchi, and nonrespiratory bronchioles have the functions of removing fine particulates from the inspired airstream and completing the conditioning (raising to body temperature and complete saturation) of inspired air. Distal to the terminal bronchioles (the most distal nonrespiratory bronchioles) is the lung parenchyma,

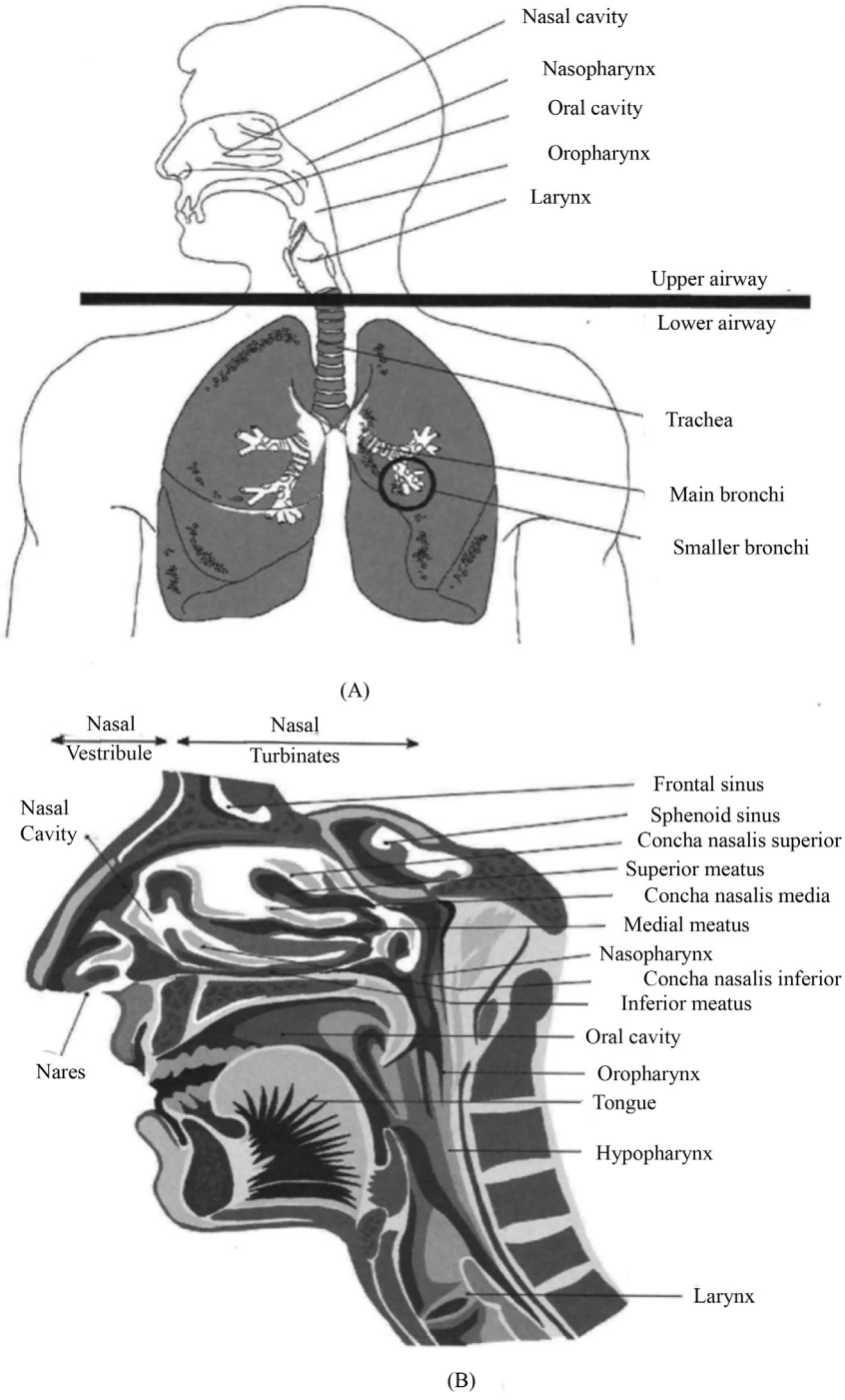


FIGURE 5.16 (A) Anatomical overview of the human respiratory tract. The larynx generally serves as the boundary between the upper (extrathoracic) and lower airways. (B) Anatomy of the upper airway.

where gas exchange occurs in the respiratory bronchioles and alveoli.

5.2.2.1 *Extrathoracic airway anatomy*

The most proximal regions of the extrathoracic airways are the nasal and oral cavities, which act as portals to and from the ambient environment. Fig. 5.16B shows how, during nasal breathing, inspired air enters at the two nares, passes through the nasal vestibules and turbinates, and exits at the nasopharynx. Total distance along the nasal passageway from the nares to the nasopharynx is approximately 10–14 cm. This narrow conduit (1–3 cm in width) divides into two paths by a septum extending from the nares to the distal edge of the turbinates. Though relatively short and narrow, the nasal passageways have a large surface area

($\approx$160 cm^2 compared with $\approx$69 cm^2 for the trachea) because of the highly convoluted turbinate structure.

Inspired air enters the nasal passages via two nares (nostrils), whose cross-sectional area can be enlarged by circular muscles (dilator naris muscles). Immediately distal to the nares are the nasal vestibules, pyramidal openings lined by squamous epithelium with nasal hairs projecting from the epithelium. These hairs achieve coarse filtration of the inspired airstream. Inspired air passes out of the vestibules via the nasal valves, slit-like openings at the back of the vestibules (each valve having a cross-sectional area of $\approx$30 mm^2), and enters the turbinates.

The turbinate regions are 5–8 cm long and defined by bony projections (superior, middle, and inferior conchae) forming convoluted passages through this region of the nasal cavity. Corresponding openings (superior, middle, and inferior meatus) define three airway passages. Ciliated epithelia and mucus-secreting goblet cells generally line the luminal surfaces of the turbinate region, though olfactory tissues are found in the superior meatus. Fig. 5.17 shows how air traveling within the turbinates can easily pass between the different meatus. The tortuous passageways promote deposition of inspired particles as well as the exchange of heat and water vapor between the airway wall and the inspiratory or expiratory airstreams. Meatus cross-sectional areas correlate to airflow, the greatest quantity of air passing through the inferior meatus. Slower airflow within the superior meatus allows for greater residence times along these airway surfaces. Increased residence times enhance olfaction occurring at the olfactory bulbs located along the superior surface of the superior meatus. Airstream mixing caused by eddy currents within the superior meatus further enhances olfaction.

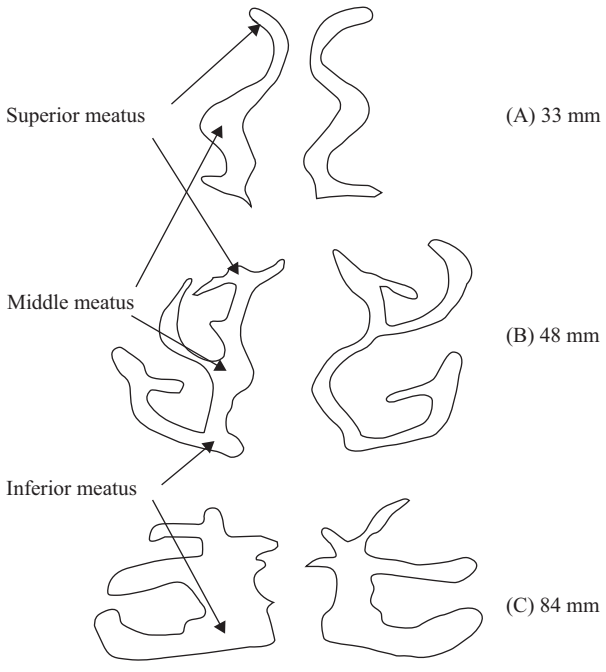


FIGURE 5.17 Cross-section of human nasal turbinates at various positions along the airway. Distances indicated are from the nares. The medial surface in each cross-section represents the nasal septum. (Modified from Guilmette et al.[2])

The two (left and right) inspiratory nasal airstreams merge in the distal end of the turbinates before experiencing a 90-degrees bend in the airway upon entering the nasopharynx. The nasopharynx is roughly 5 cm long, has a volume of 12 cm^2, and is lined with squamous epithelium, which appears to protect underlying tissue from gross mechanical injury. Any relatively large particles (>3 μm) successfully navigating the nasal passages will likely impinge upon the nasopharyngeal wall because of inertia. Ciliated columnar epithelium interspersed with mucus-secreting goblet cells appearing distal to the nasopharynx marks the start of the oropharynx.

Ambient air entering the oral cavity during oral breathing confronts a variety of surface structures. Inspired air initially passes between highly vascular lips and across the teeth, which can be viewed as a series of heat transfer fins. The tongue and buccal surfaces (both rough, highly vascular surfaces) and the hard palate border the cavernous opening beyond the teeth. The soft palate defines the distal limit to the oral cavity, beyond which the airstream bends 90 degrees to enter the oropharynx. Oral cavity dimensions vary greatly depending on tongue position and extension of the buccal surfaces but a simple cylindrical model (8 cm length and 1.8 cm diameter) has been used to characterize the oral cavity.[50] Inspired air passing out of the oral cavity enters the oropharynx.

The pharynx (nasopharynx, oropharynx, and hypopharynx) serves to pass air between the airway portals (nasal and oral cavities) and the thoracic airways (tracheobronchial tree and alveoli). It terminates at the epiglottis, a valve that prevents swallowed food and liquids from entering the lower airways. Beyond the epiglottis lies the larynx, which serves as a conduit for air passing in and out of the lower airways and as a tone-producing structure. Both pharyngeal and laryngeal surfaces are lined with columnar ciliated epithelium and goblet cells, except for the squamous epithelium lining the nasopharynx and a small area on the vocal folds of the larynx.

5.2.2.2 *Central and pulmonary airway anatomy*

Inspired air passing out of the larynx forms a jet as it enters the trachea, the largest conducting tube in the airway. The most proximal tube in the tracheobronchial tree (generation zero in the Weibel "A" model),[51] the trachea has an approximate diameter of 1.8 cm and

extends in adults roughly 12 cm from the distal edge of the larynx to the carina. Columnar ciliated epithelium and goblet cells are the primary cell types lining the tracheal lumen. Negative pressures within the tracheal lumen during strenuous inspiration can produce significant radial pressure gradients that would, if possible, collapse the trachea. Tracheal patency during strenuous breathing is ensured by a series of incomplete cartilaginous rings supported by fibroelastic and smooth muscle tissues extending along the length of the trachea.

The trachea terminates at the carina, the site at which the main bronchi bifurcate. Bronchial tube diameters and generally lengths decrease distally from the carina with successive bifurcations. The right and left main bronchi have diameters of approximately 1.2 cm and lengths of 4.76 cm, decreasing to diameters of approximately 0.13 cm and lengths of 0.46 cm in the smallest bronchi (generation 10). Cartilage occurring in airway walls down to the tenth generation of bifurcations assists bronchial smooth muscle in maintaining bronchial patency during strenuous breathing. Bronchioles (generations 11–18) lack cartilage and rely entirely on smooth muscle for maintaining luminal patency during breathing. Alveolar ducts (generations 19–23) and alveoli (generation 24) lack any cartilage or smooth muscle and maintain patency by a balance between tensile forces generated by gases present within the alveolar lumen and alveolar fluid surface tension. Surfactants present in alveolar fluid prevent surface tension from collapsing the alveoli (atelectasis).

The estimated number of tubes in each airway generation depends on the bifurcation model used in describing the tracheobronchial tree. Though bronchial bifurcations are asymmetric, symmetric models, exemplified by Weibel,[52] or asymmetric models, such as one suggested by Horsfield,[53] can each serve to represent airway branching. Later studies have also suggested a fractal or functional patterns to the bifurcations.[54,55] Whatever the overall bifurcation pattern, the general structure can be most easily summarized by the Weibel "A" model,[51] in which successive bronchial generations are more numerous, shorter, and have smaller individual cross-sectional areas than more proximal generations (Fig. 5.18). According to the Weibel "A" model, the number of branches in generation z is

$$N(z) = 2^z \tag{5.30}$$

and the mean diameter of airways in generation z, $d(z)$, is given by

$$d(z) = d_0 2^{-z/3} \tag{5.31}$$

where d_0 is the tracheal diameter (Table 5.5). Consequently, overall cross-sectional area increases exponentially as a function of distance from the nares, producing a predicted alveolar surface area of

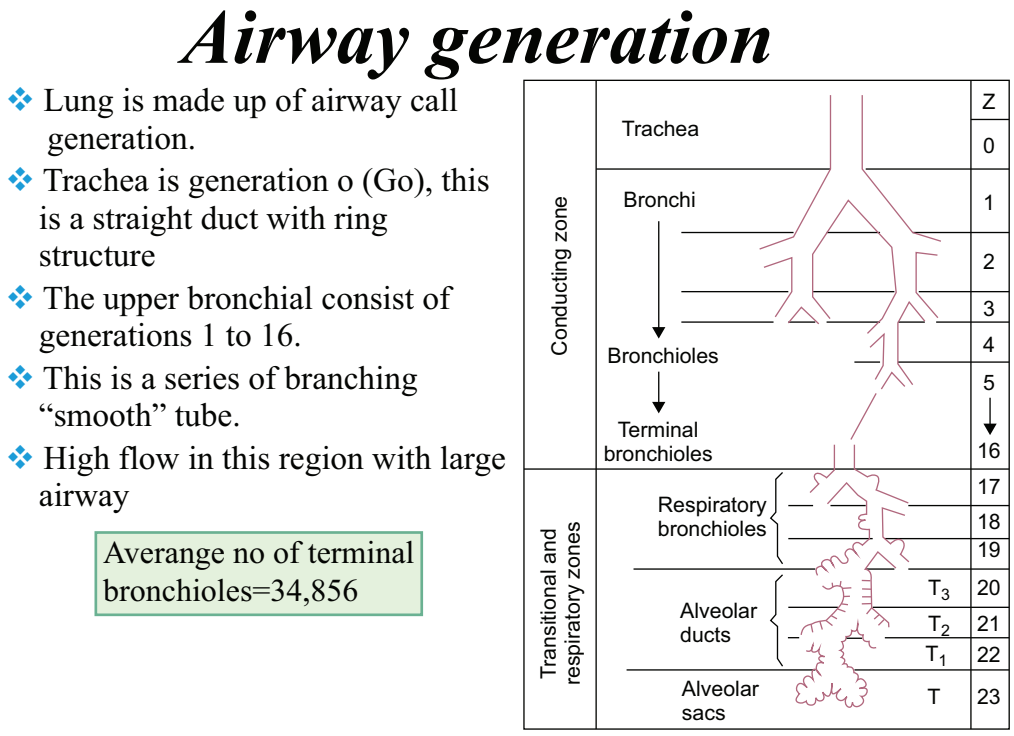


FIGURE 5.18 Schematic picture of airway generation.

TABLE 5.5 Representative conducting airway dimensions based on the Weibel "A" model.

Airway region	Cross-sectional area (cm)	Equivalent diameter (cm)	Airway segment length (cm)	Reynolds number		
				Minute ventilation		
				8 L/min	16 L/min	30 L/min
Nasal vestibule	1.3	67	0.9	1515	3029	5679
Nasal cavity	2.4	0.61	1.8	1664	3327	6238
Nasal turbinates	2.3	0.44	1.3	2306	4612	8648
Nasal turbinates	2.6	0.36	4.4	2819	5637	10,570
Nasal turbinates	3	1.18	0.6	860	1720	3225
Proximal nasopharynx	3.9	2.03	2	500	1000	1875
Distal nasopharynx	2.9	1.45	3	700	1400	2624
Proximal oropharynx	2.8	1.26	1.7	805	1611	3020
Distal oropharynx	3	1.3	1.3	781	1561	2927
Proximal hypopharynx	2.3	1.5	2.4	676	1353	2537
Distal hypopharynx	1.9	1.3	1.3	781	1561	2927
Larynx	1.8	1.5	1.1	676	1353	2537
Proximal trachea	2.1	1.6	2.7	634	1268	2378
Distal trachea	2.8	1.9	9.3	534	1068	2003
Bronchii gen. 1	3.4	1.47	4.8	345	689	1292
Bronchii gen. 2	3.9	1.11	1.9	227	454	851
Bronchii gen. 3	3.9	0.79	0.8	161	323	605
Bronchii gen. 4	3.9	0.56	1.3	114	229	429
Bronchii gen. 5	4	0.4	1.1	80	159	299
Bronchii gen. 6	4.3	0.29	0.9	54	107	202
Bronchii gen. 7	4.7	0.22	0.8	37	75	140
Bronchii gen. 8	5.4	0.16	0.6	24	47	89
Bronchii gen. 9	6.7	0.13	0.5	15	31	58
Bronchii gen. 10	8	0.1	0.5	10	20	37
Bronchii gen. 11	19.6	0.11	0.4	4.4	8.9	17
Bronchii gen. 12	28.8	0.1	0.3	2.8	5.5	10
Bronchii gen. 13	44.5	0.08	0.3	1.4	2.9	5.4
Bronchii gen. 14	69.4	0.07	0.2	0.8	1.6	3
Bronchii gen. 15	113	0.07	0.2	0.5	1	1.9
Bronchii gen. 16	180	0.06	0.2	0.3	0.5	1
Bronchii gen. 17	300	0.05	0.1	0.1	0.3	0.5

Turbulent flow (Reynolds number >3000) is predicted only in the extrathoracic airways at flow rates <30 L/min.

43–80 m^2. Respiration (exchanging O_2 for CO_2) depends on this large exchange surface to provide sufficient gas exchange capacity during strenuous activity to accommodate demands by active muscles for greater volumes of O_2 and the need to remove excess CO_2.

5.2.2.3 *Airway wall anatomy*

Airway cross-sections have the nominal anatomy shown in Fig. 5.19. Airway surface liquid (ASL), primarily composed of mucus gel and water, surrounds the airway lumen with a thickness thought to vary

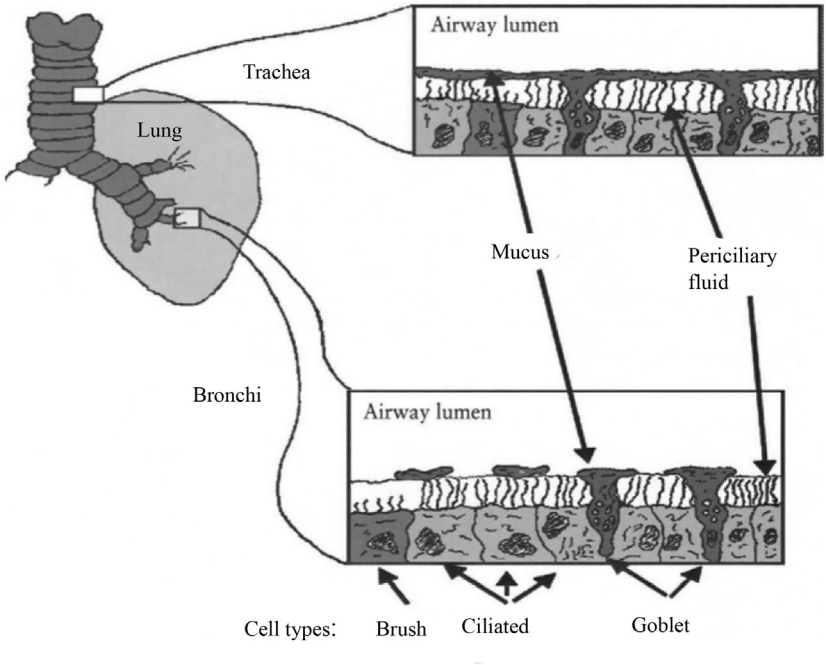


FIGURE 5.19 Depiction of representative airway cross-section at various points (trachea, bronchi, and pulmonary airway) along the respiratory tract showing common cell types. Note how mucus gel Is generally presumed to form sheets in the more proximal airways. The pulmonary airway depiction includes both a section of respiratory bronchi and an alveolus.

from 5 to 10 mm. ASL lies on the apical surface of airway epithelial cells (mostly columnar ciliated epithelium). This layer of cells, roughly two to three cells thick in proximal airways and eventually thinning to a single cell thickness in distal airways, rests along a basement membrane on its basal surface. Connective tissue (collagen fibers, basement membranes, elastin, and water) lies between the basement membrane and airway smooth muscle. Edema occurs when the volume of water within the connective tissue increases considerably. Interspersed within the smooth muscle are respiratory supply vessels (capillaries and arteriovenous anastomoses), nerves, and lymphatic vessels.

Certain respiratory diseases [e.g., asthma and chronic obstructive pulmonary disease (COPD)] alter airway dimensions, and thereby modifying airflow patterns and adversely affecting particle deposition. Emphysema which develops usually by effect of oxygen radicals of tobacco smoke or other environmental exposure is characterized by destruction of alveolar walls which enlarges alveolar sac volume. However, simultaneously the alveolar surface area suitable for gas exchange is reduced. Reduction of alveolar surface area decreases gas exchange and diminishes the body's ability to obtain oxygen from the inspired airstream. In addition, destruction of alveoli causes disbalance of ventilation related to perfusion, and poorly ventilated regions, for example, emphysematic bullae may develop. Diseases that reduce bronchial diameter (asthma, chronic bronchitis, and cystic fibrosis) increase airstream velocity in occluded airway regions, increasing heat and water vapor exchange and particle impaction while reducing sedimentation in the affected bronchi. This results in a greater volume of fine inhaled particles [$< 3.0\,\mu m$ mean mass aerodynamic diameter (MMAD)] passing to more distal (and potentially more vulnerable) regions, permitting more of these fine particulates to settle in distal airways by sedimentation (see Section 5.2.7).

Airway surface liquid

The airways' luminal surfaces are lined with ASL along all airway surfaces except portions of the extrathoracic and respiratory airways (respiratory bronchi and alveoli). ASL serves to protect airway epithelium against airborne pathogens and toxins, desiccation and changes in pH, and its volume, pH, ionic and nutrient content are important in regulating antimicrobial activity, ciliary function, and mucociliary transport. This fluid is secreted ASL consists of a periciliary layer composed mainly of water and various ions approximately 5–7 μm in depth[56,57] and the epiphase, an overlying gel layer of hydrated mucins in the form of droplets, sheets, or blankets[58–60] (Fig. 5.19). Epithelial cells control periciliary fluid water and ion concentration by chloride secretion and sodium absorption. Solids constitute approximately 5% of periciliary fluid mass, with water comprising the remaining 95%, though disease can raise solids concentration above 10%. Periciliary fluid solids include glycoproteins, proteins, peptides, glycosaminoglycans, immunoglobins, and lipids in addition to materials deposited from the passing airstream. The epiphase is thought to be a hydrogel consisting of various complex glycoproteins,

with hydration controlled by a Donnan effect.[61,62] (Negatively charged big protein molecules do not pass through a semipermeable membrane but attract small size cations to move to their side of the membrane.) Control of periciliary fluid hydration is a complex interaction of evaporation,[57,63] osmotic pressure differentials regulated by ion transport,[58,64,65] and hydrostatic pressure.[66] Estimates of daily mucus production range from 7 to 12 mL/day in healthy individuals to >100 mL/day in cystic fibrosis patients.

Airway epithelial cell types

Conducting airway passages are generally composed of ciliated pseudostratified cuboidal columnar epithelial cells interspersed with basal, brush, and secretory cells (goblet, serous, and Clara). Cilia found on ciliated epithelial cell apical surfaces (along the lumen) provide motive force for propelling mucus gel along the airway. Extrathoracic airway surfaces are lined with ciliated epithelium, except for squamous epithelium covering the nasal vestibule, nasopharynx, oral cavity, oropharynx, and portions of the larynx. Squamous epithelium protects airway surfaces against mechanical impact or shear in areas where relatively large inspired particles usually impact. Also nasal olfactory surfaces are not lined with ciliated epithelium but covered instead by special sensory cells. These specialized olfactory receptor cells react with inhaled odorant molecules, generating neural signals sent to the olfactory bulbs of the brain that produce a sense of smell. All tracheobronchial surfaces are lined with ciliated epithelium down to the pulmonary airways. Proximal airway epithelium is thickest and progressively flattens and thins toward the lung parenchyma, gradually transitioning into alveolar endothelium.

Secreting cells found along the conducting airways include nonciliated goblet and serous cells. Goblet cells produce glycoproteins that form droplets or sheets of mucus gel floating on periciliary fluid. Serous cell exudates are believed to include periciliary fluid, various proteins, and peptides (including lysozyme and lactoferrin), and protease inhibitors. Periciliary fluid also derives from interstitial fluid transudate. Glycosaminoglycans, lipids, serum proteins, and ions found in ASL appear to originate from all surface epithelial cells and submucosal glands (serous and mucous). The quantity of submucosal glands decreases in more distal airways and are absent from pulmonary airways.

Microvilli, approximately 2 μm long, give a "brush-like" appearance as they project from the apical surface of brush cells. These cells contribute to fluid regulation along the luminal surface by absorbing excess periciliary fluid either secreted by neighboring serous cells or transported from distal airways by the mucociliary elevator. Basal cells are progenitors of the other epithelial cells and are the most actively mitotic epithelial cells. Lymphocytes also appear in ASL as either migratory or basal cells.

Pulmonary airways are lined with specialized cells generally not found in the conducting airways. Alveolar epithelium, composed of thin sheet-like cells separated from pulmonary capillaries by only a basement membrane, permits easy exchange of gases between alveolar sacs and blood (Fig. 5.20). Secretory clara and type II pneumonocyte cells produce surfactant, lipids, and protease inhibitors within the

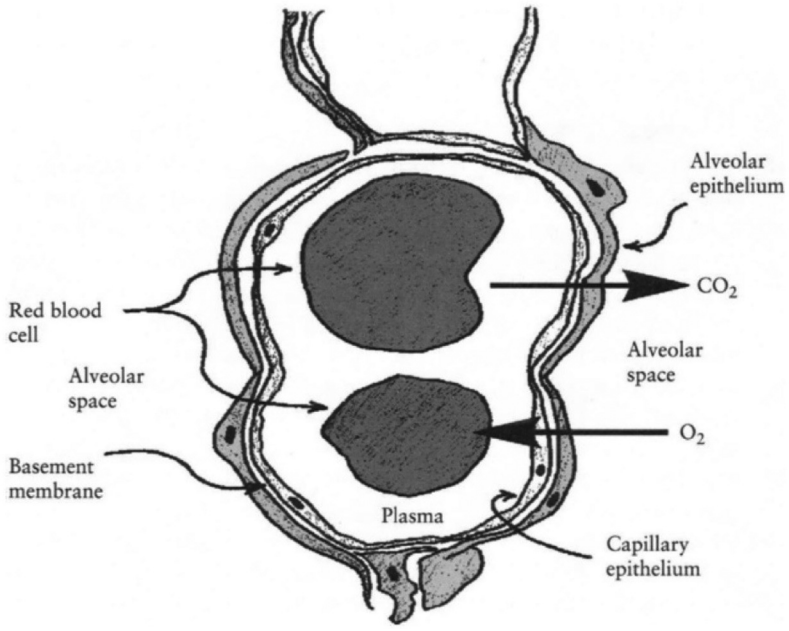


FIGURE 5.20 Gas exchange between alveolar and capillary compartments.

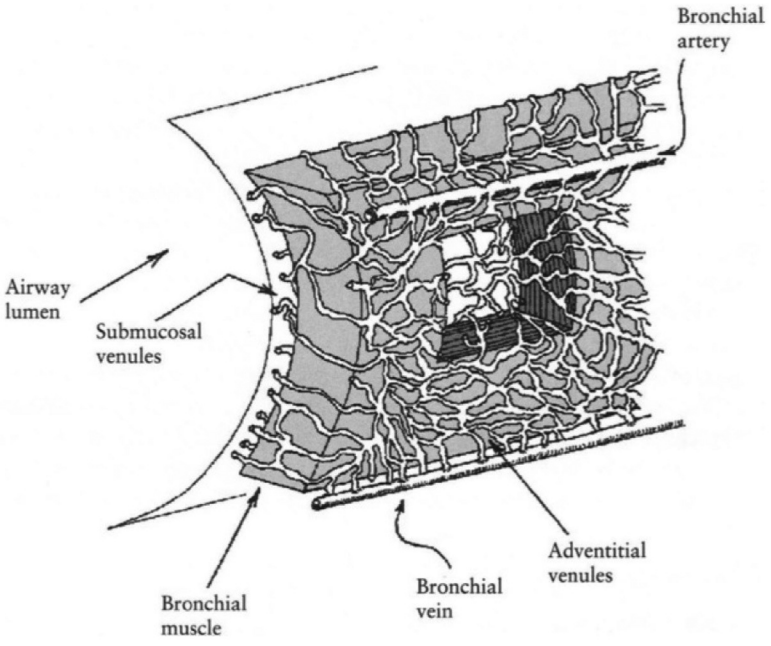


FIGURE 5.21 Vasculature structure along a portion of bronchial muscle. Airway epithelia are not shown in this figure but lie between the submucosal venules and the airway lumen.

pulmonary airways. Macrophages are scavenger cells that remove microorganisms and particulates depositing along alveolar surfaces.

5.2.2.4 *Airway vasculature*

Pulmonary gas exchange is intimately connected to cardiovascular function. Deoxygenated blood from the right ventricle of the heart passes through the pulmonary artery to the lungs, exchanges carbon dioxide and oxygen across the alveolar wall, and returns to the left atrium of the heart via the pulmonary veins. The heart propels this oxygenated blood from the left ventricle through the aorta and hence throughout the body via a high-pressure system of thick-walled vessels known as arteries branching out from the aorta. Further branching gradually reduces the cross-section of arteries until, at a diameter of approximately 30 μm, they are termed arterioles. Total vascular surface area increases as arterioles continue to branch and diminish in diameter until they terminate at a capillary bed or connect directly with venuoles in an anastomosis, a dense network of interconnected vessels.

Arteriole wall smooth muscle controls vascular diameter and regulates blood flow by modulating the pressure drop along the length of the vessel. Enlisting groups of arterioles regulates local or regional vascular resistance by modulating capillary flow in response to temperature changes or other stimuli. Active control of arteriole wall smooth muscle tone due to a variety of internal and external stimuli also regulates blood flow through anastomoses and consequently peripheral blood volume and pressure. Anastomoses control peripheral blood flow by allowing a portion of total blood flow to bypass capillary beds.

Intravascular blood pressure drops as a function of arterial and arteriole diameter to such an extent that capillary walls can consist of a single layer of endothelial cells. Capillaries, with diameters of 6–8 μm, transport blood close enough (roughly 20–30 μm) to cells throughout the body to allow gas (O_2 and CO_2), heat, nutrient and waste, and water exchange between blood and cells via diffusion. Interstitial tissue containing collagen fibers, basement membranes, elastin, and water supports capillary endothelial cells and provides additional tensile strength. Capillaries merge to form venules, which in turn merge to form veins. These low-pressure components of the cardiovascular system—capillaries, venules, and veins—transport deoxygenated blood from the capillaries to the right atrium of the heart via the largest vein, the inferior vena cava.

Blood supplying conducting airway tissues derives from large bronchial arteries branching off either the aorta or intercostal arteries. These vessels also supply blood to the visceral pleura, regional nerves and lymph nodes, and vascular walls of the pulmonary arteries and veins. Bronchial artery branches follow the conducting airways and provide blood to the bronchial walls down to the respiratory bronchioles. Smaller arterial branches form anastomoses along the peribronchial surface (Fig. 5.21). Arterioles originating from the peribronchial anastomoses penetrate the bronchial smooth muscle and form relatively straight, thin bronchial capillaries, and submucosal anastomoses. Conducting airway luminal cells (ciliated epithelium, serous, and goblet cells) are supplied with

nutrients, oxygen, water, and heat via these submucosal anastomoses.

Respiratory bronchioles and alveoli are supplied with deoxygenated blood from the right ventricle of the heart by the pulmonary arteries. Five lobar arterial branches follow the bronchi, and subsequent bronchopulmonary arterial branches run adjacent to smaller airways to the level of the respiratory bronchioles. Dense coiled capillary networks beyond this point distribute deoxygenated blood to capillaries and return oxygenated blood to the venules arising from the respiratory bronchiolar, alveolar, and alveolar duct capillary beds. Pulmonary capillaries directly attach to lung connective tissue, reducing diffusive resistance to gas exchange.

Vessels linking bronchial arteries directly with pulmonary alveolar microvessels are commonly found in neonates but apparently decrease in frequency with age. There is also evidence of direct communication between bronchial arteries and pulmonary veins. Venous blood originating from extrapulmonary airways (proximal to approximately generation 3 bronchi) drains into the right atrium via the azygos and hemiazygos veins. Intrapulmonary bronchial venous flow, returning blood to the heart from bronchi distal to the third generation, drains into the pulmonary circulation, which subsequently drains into the left atrium either directly or via the pulmonary vein.

Airway surfaces, like skin, are continually exposed to the ambient environment. In contrast to skin submucosal vessels, however, which shed excess heat by vasodilating when heated and conserve heat by vasoconstricting when chilled, it is unclear how the airway vasculature responds to temperature extremes. Inspiring cold air poses two challenges to conducting airway tissues: the risk of tissue injury should inadequate heat reach the airway surface and excessive body heat loss due to increasing the radial temperature gradient. Vasodilation would protect airway tissue but increase heat loss, while vasoconstriction would produce the opposite effect.

Nasal vasculature may offer some insight into this question, though research to date has been equivocal. Nasal turbinate vessels can be classified as either capacitance vessels or resistive vessels. Capacitance vessels appear to vasodilate in response to infection[67] while resistance vessels appear to respond to cold stimuli by vasoconstriction.[68] Buccal vascular structures also respond to thermal stimuli but appear to respond principally to cutaneous stimuli.[69] How pharyngeal and tracheobronchial submucosal vessels react to thermal stimuli is not known, though cold-induced asthma is believed to result from bronchospasms caused by susceptible bronchial smooth muscle responding to exposure to cold dry air.[70,71] This asthmatic response suggests an inadequate vascular response to surface cooling.

5.2.3 Ventilation patterns

5.2.3.1 Breathing mechanics

Breathing consists of the cyclic action of the lungs to inspire and expire atmospheric gases. Based on the work of the inspiratory muscles, in the pleural spaces, a negative (subatmospheric) pressure develops being about −2 mmHg in quiet breathing at the start of inspiration decreasing to about −6 mmHg, when the lungs expand slightly, the pressure in the airway becomes slightly negative. Air enters the respiratory tract from the surrounding atmosphere when the inspiratory pressure exceeds airway resistance. At the end of inspiration, the lung recoil pulls the chest back to the expiratory position which is a balance between the lung and chest recoil pressures. During expiration, the airway pressure becomes slightly positive, but the quiet expiration is mainly a passive phenomenon. When lung viscoelastic forces overcome pleural pressure, the lungs constrict and air is expelled.

Movement of the chest wall in relationship to lung volume can be represented on a pressure−volume diagram (Fig. 5.22). The pressure term refers to pleural pressure, a measure of pressure within the space between the pleural membranes surrounding the lungs. The volume term represents changes in percent vital capacity (%VC), changes in lung volume, or other convenient measures of lung volume. Hysteresis, that is, the failure of the chest wall and lungs to follow identical pressure−volume paths during inspiratory and expiratory loads, is caused by viscoelastic and plastic properties of the lung and chest wall. Specifically, mechanical differences between lung surfactant properties and alveolar recruitment, on the one hand, and chest wall skeletal muscle and elastic fiber properties, on the other, are believed to account for most of the observed hysteresis. It is worth noting that posture also affects pressure−volume relationships by shifting gravitational forces within the abdomen.

Gases entering the airways first fill the volume of the anatomical dead space (conducting airways) before filling the pulmonary airway. Alveolar ventilation ($\dot{V}_A$) defines the volumetric rate of gas passing through pulmonary airways that participate in O_2 and CO_2 exchange, with O_2 uptake ($\dot{V}_{O_2}$) and CO_2 production ($\dot{V}_{CO_2}$) determined by metabolic demands.

Air entering alveolar spaces but not partaking in gas exchange due to poor perfusion of individual alveoli is not part of V_A but adds to the total dead space volume (V_D). This additional dead space leads to the concept of a physiological dead space that includes

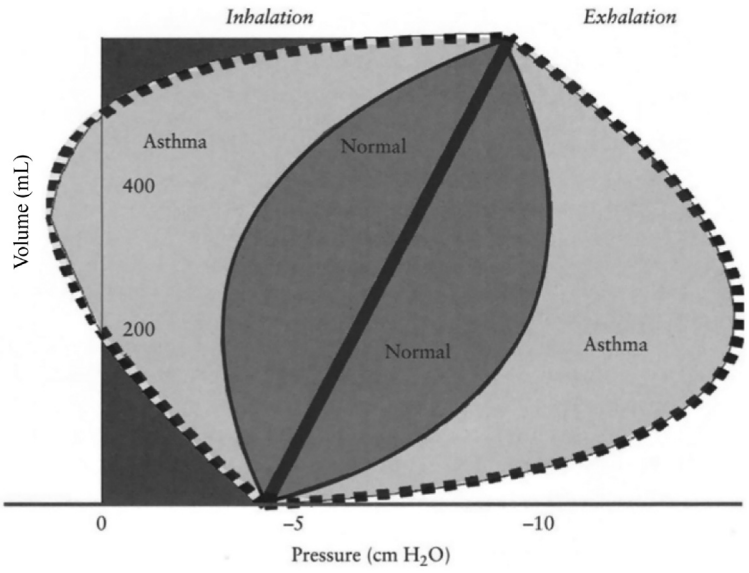


FIGURE 5.22 Relationship of transpleural pressure to volume in normal and asthmatic individuals.

not only an anatomical component (conducting airways) but also a functional component (poorly perfused or nonperfused alveoli). Diseases affecting either conducting airway geometry or pulmonary perfusion can thus alter V_D. The total volume of air ($\approx$500 mL) inspired (or expired) during each breath is known as the tidal volume, V_T and can be described by

$$V_T = V_D + \frac{\dot{V}_A}{f}, \tag{5.32}$$

where f is the breathing frequency (breaths/min) so that

$$\dot{V}_A = f(V_T - V_D). \tag{5.33}$$

Expired minute ventilation, $\dot{V}_E$, defines the gas volume inspired or expired in 1 minute and is given by

$$\dot{V}_E = V_T f, \tag{5.34}$$

Typical $\dot{V}_E$ for normal quiet breathing is approximately 6–8 L/min. In extreme circumstances, individuals can live for brief periods with minute ventilation rates as low as 1–2 L/min or as high as 300 L/min. Table 5.6 shows the dependence of $\dot{V}_A$ on V_D, V_T, and f for a given $\dot{V}_E$.

Alveolar ventilation supplies O_2 to the bloodstream while alveolar capillary perfusion provides alveolar gas with CO_2. Resting individuals consume approximately 250 mL O_2/min and produce approximately 200 mL CO_2/min because, stoichiometrically, metabolic processes require a greater supply of O_2 than the quantity of CO_2 produced. Defining the respiratory exchange ratio, R, as

$$R = \frac{\dot{V}_{CO_2}}{\dot{V}_{O_2}}, \tag{5.35}$$

TABLE 5.6 Effect of dead space volume, tidal volume, and breathing frequency on alveolar ventilation at a fixed minute ventilation ($\dot{V}_E = 58.0$ L/min).

$\dot{V}_A$ (L/min)	V_D (m/L)	V_T (mL)	f (/min)
3.2	150	250	32
4.0	250	500	16
4.8	200	500	16
5.6	150	500	16
6.8	100	1000	8

$\dot{V}_A$, alveolar gas volume; V_D, dead space volume; V_T, tidal volume; f, breathing frequency.
Modified from Cherniak RM. Pulmonary function testing. 2nd ed. Philadelphia, PA: W.B. Sanuders; 1992.[72]

then $R = 0.8$ during normal resting breathing and $\dot{V}_A = 4L$/min is required to lower the arterial CO_2 partial pressure to 40 torr and raise arterial O_2 partial pressure to 100 torr to maintain blood hemoglobin saturation levels (97.5%) in the venous end of pulmonary capillaries. A corresponding pulmonary perfusion rate, $\dot{Q}$, equal to 5 L/min of arterial blood is necessary when both ventilation and flow are uniform. The subsequent ventilation–perfusion ratio, $\dot{V}_A/\dot{Q}$, provides a quantitative measure of gas exchange efficiency. $\dot{V}_A/\dot{Q} = 0.8$ in this ideal case but generally ranges from 1.0 at rest to 3.0 or greater during heavy exercise.

5.2.3.2 Measurement of pulmonary gas exchange

Pulmonary gas exchange is usually measured with diffusing capacity measurement measuring the diffusion of carbon monoxide (CO) from alveoli to pulmonary capillaries. In diffusing capacity measurement, the subject inhales about one VC of gas mixture

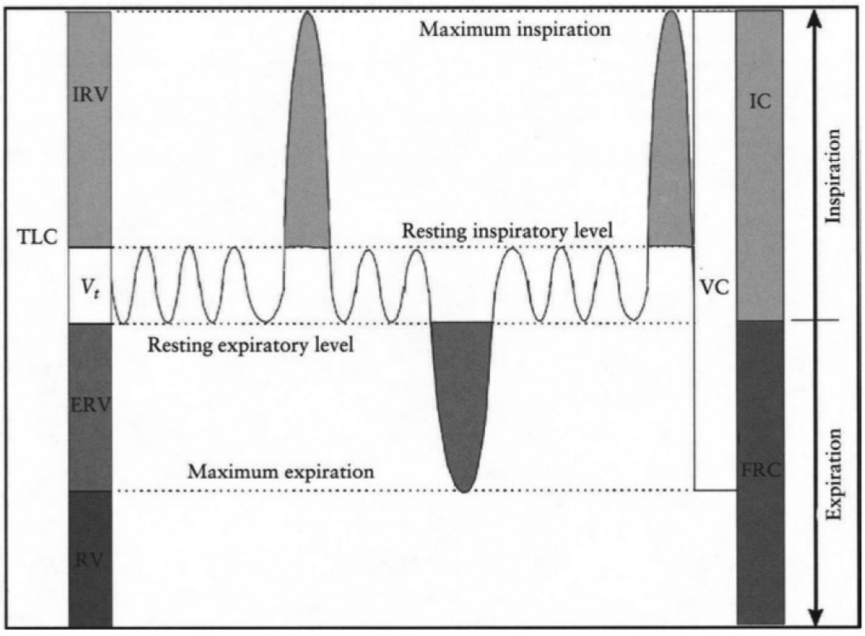


FIGURE 5.23 Graphical representation depicting relationship between airway volume measurements. The curve represents both tidal and forced breathing patterns.

containing CO and helium or methane (CH_4), holds breath for 10 seconds and exhales. From the inhaled and exhaled concentrations of CH_4 are measured the alveolar volume, which is the lung volume during breath holding. From the inhaled and exhaled concentrations of CO are measured the single breath diffusing capacity for CO, and diffusing capacity is the amount of CO that is diffusing from alveoli (gas flow, V' mmol) during a certain time and during a certain pressure difference between alveolar air (P_A) and pulmonary capillaries (P_C).

$$DL = V'/(P_A - P_C)/t$$

There are several structural factors influencing the diffusing capacity (pulmonary gas volume, the diffusing distance, thickness and area of the membrane of alveolar wall and capillaries, and amount of hemoglobin) as well as functional factors (e.g., relation and distribution of ventilation, the content of alveolar gas, the diffusing properties of alveolocapillary membrane, the rapidity of CO binding to hemoglobin, and the partial pressures of CO and O_2 in the capillary blood).[73]

5.2.3.3 *Static and dynamic lung volumes*

Summing V_T, the inspiratory reserve volume (IRV), the expiratory reserve capacity (ERV), and the residual volume (RV) gives the total lung capacity (TLC). IRV is the maximum additional volume one can inspire from end-tidal inspiration. ERV measures the maximum additional volume one can expire from an end-tidal expiration level. RV measures the gas remaining in the respiratory tract after the maximum possible exhalation and reflects the minimum noncollapsible volume (under normal circumstances) within the airway. In contrast, the functional residual capacity (FRC) measures the gas volume remaining in the airway at an end-tidal exhalation. The deepest possible breath (TLC-RV) is defined as the VC. Fig. 5.23 graphically depicts the various components of airway volume. Values for TLC, VC, and RV depend on health, body size, gender, and age. Table 5.7 lists predictive equations for healthy individuals. In general, females have 10%–25% smaller volumes than men of the same age and size. Age has its greatest effect on RV, which increases by 50% or more from ages 20 to 60.

Forced expiration is commonly used to assess pulmonary function in both healthy and impaired individuals. Static measures of lung volumes (TLC, V_T, and FRC) fail to detect dynamic changes in pulmonary function that are attributable to disease (e.g., asthmatic airway constriction). Obtaining maximum expiratory flow-volume (MEFV) curves (Fig. 5.24) permits derivation of key parameters in detecting changes in lung function.

Forced vital capacity (FVC) quantifies the maximum air volume expired following a maximal inspiration and it primarily reflects the elastic properties of the respiratory tract, and is reduced in loss of lung volume. To measure FVC, an individual inhales maximally and then exhales as rapidly and completely as possible. The gas volume forcibly expired within a given time interval, FEV_t (where t is typically one second, $FEV_{1.0}$) is also commonly used for diagnostic purposes and represents expiratory flow resistive

TABLE 5.7 Predictive equations for static lung volumes and dynamic pulmonary function.[72]

Parameter	Gender	Prediction equation
Vital capacity	Female	$0.0404H - 0.022A - 2.35 - \frac{147.1}{H} - 0.00828A + 0.5673 \log A - \frac{0.5509}{W} + 0.0242$
	Male	$0.0481H - 0.020A - 2.81 - \frac{1.4892}{H} - 0.0069A + 0.5191 \log A - \frac{0.6347}{W} + 1.0206$
Residual volume	Female	$0.032H - 0.009A - 0.390 - \frac{2.1684}{H} - 0.00374A + 0.0185 \log A - \frac{9.2457}{W} + 1.2934$
	Male	$0.027H - 0.017A - 3.447 - \frac{2.3637}{H} - 0.00338 + 0.6387 \log A - \frac{10.5711}{W} + 0.694$
Total lung capacity	Female	$0.079H - 0.008 - 7.49 - \frac{171.34}{H} - 0.00380A + 0.3481 \log A - \frac{3.0601}{W} + 1.3667$
	Male	$0.094H - 0.015A - 9.167 - \frac{173.61}{H} - 0.00561A + 0.5292 \log A - \frac{3.5461}{W} + 1.2155$
Forced expired volume, 1 s[a]	Female	$3.95H - 0.025A - 2.60$
	Male	$4.30H - 0.029A - 2.49$
Forced vital capacity[a]	Female	$4.43H - 0.026A - 2.89$
	Male	$5.76H - 0.026A - 4.34$
Peak expired flow[a]	Female	$5.50H - 0.030A - 1.11$
	Male	$6.14H - 0.043A + 0.15$

[a]*Health Survey for England (1996).*
A, Age (years); *H*, height (cm); *W*, weight (kg).

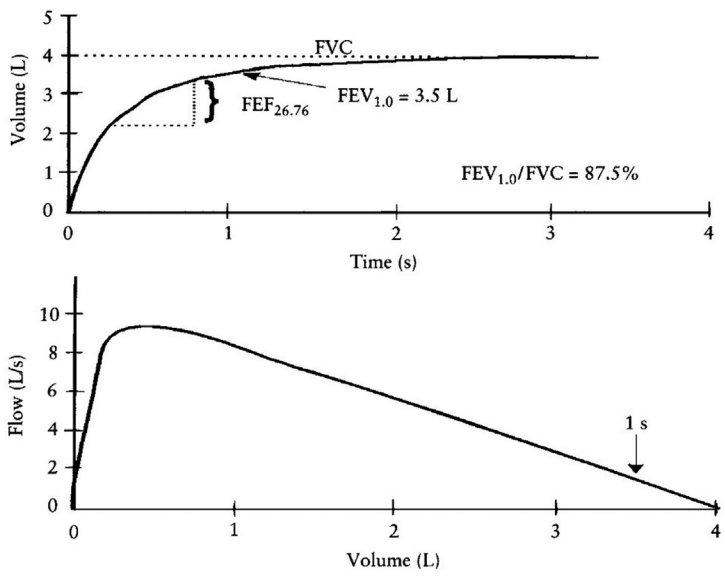


FIGURE 5.24 Representative spirogram (top) and flow-volume curve (bottom) during forced expiration. $FEV_{1.0}$ shown in the spirogram corresponds to the arrow in the flow-volume curve indicating forced expired volume in one second.

properties of the respiratory tract. $FEV_{1.0}$ has the advantage of being relatively independent of effort. Despite diagnosing obstruction, its reduction also may reflect the reduction of TLC or loss of lung recoil. Other timed expiratory intervals are also used, for example, ($FEV_{0.5}$) is recommended to be used in assessment for pediatric lung function and FEV6 has been suggested to be used instead of FVC, but its use has not become routine. In bronchodilation tests (usually performed with 0.4 mg salbutamol inhalations) changes of 12% and 200 mL of FEV1 compared with baseline during a single testing session suggest a significant bronchodilation diagnostic for asthma.[74]

FEV1.0 related to FVC (the FEV1/FVC ratio) is the most important parameter to detect pulmonary obstruction, for example, in assessment of COPD. Low values of FEV1/FVC ratio and especially its value <0.7 in bronchodilation phase has long been used in epidemiological studies to be as simple variable for bronchial obstruction.[75] In pulmonary restriction, the values of FEV1/FVC ratio are usually normal or (e.g., pulmonary fibrosis and obesity). If the instantaneous flow at certain volumes of FVC are measured, the maximal instantaneous flow when X% of the FVC remains to be expired (MEFX%) or has been exhaled (FEFX%) can be measured representing peripheral airways

function but being rather effort dependent they have often regarded to be of restrictive use.[74] Another common measure of lung function derived from the MEFV curve is the peak expiratory flow, PEF, which is used as a simple method to predict airway conductance. Unfortunately, PEF is sensitive to effort during testing, depends much more on extrathoracic and tracheal conductance rather than pulmonary conductance, and is insensitive to lesser airway obstruction.

5.2.3.4 Bronchial hyperresponsiveness (hyperreactivity)

Nonspecific bronchial hyperresponsivenees (BHR) is a state of the airways with an increased tendency to bronchospasm (contraction of the smooth muscles of bronchi) when the subject is challenged to nonspecific stimuli. Increased BHR is typical for asthma, but also in COPD it may be present. BHR may also develop in allergen exposure, infection or during high level or long lasting irritant exposure of the airways. Increased BHR is detected by decrease of FEV1 or PEF in inhalation exposure to nonspecific stimuli as exercise, cold air or irritants or inhalation of allergens, and it can be also be verified in histamine or methacholine challenge testing.

5.2.3.5 Intraairway airflow patterns

Transporting inspired and expired gases through the airway, depositing particulates onto mucosal surfaces, and exchanging heat and water vapor between the airstream and airway surfaces depends on a number of factors, one of the more important being airway flow characteristics. Airway geometry, airstream velocity, and gas density determine the flow regime prevailing in each airway region. Turbulence in fluid flow through a conduit is generally associated with fluid inertial forces greatly exceeding fluid viscous forces such that Poiseuille flow (parabolic laminar flow) is not established and eddy currents develop. The Reynolds number, Re, quantifies this relationship between inertial and viscous forces and is given by

$$\mathrm{Re} = \frac{\mu D u}{\rho}, \tag{5.36}$$

where μ is the fluid viscosity, D is the tube diameter, u is the mean fluid velocity, and ρ is the fluid density. The equivalent diameter, $D_e = 4A/P$, where A is the conduit cross-sectional area and P is the wetted perimeter, replaces D for noncircular conduits. In circular straight tubes, $\mathrm{Re} > 2300$ typically indicates the presence of turbulence. Convection caused by eddy currents enhances deposition of buoyant airborne particles by bringing more of these particles into contact with the mucosal surface. Airway heat and water vapor exchange is also enhanced by turbulent airflow.

Turbulence in nasal cavity airflow is a consequence of both high airstream velocities, caused by small nasal cross-sectional areas, and very irregular nasal airway geometry, which induces flow distortions. Nasal turbinate Re exceeds 2300 even during normal quiet breathing and nasal cavity airflow is apparently turbulent at most $\dot{V}_E$. Flow in the pharynx, larynx, and trachea is also generally turbulent at most $\dot{V}_E$ despite $\mathrm{Re} > 2300$ only at higher $\dot{V}_E$ (30 L/min and greater). This airstream mixing enhances convective heat and mass transfer in extrathoracic airways and plays a major role in airway defense mechanisms.

Humans preferentially breathe nasally, possibly because of the highly efficient filtration, humidification, and warming performed on the inspiratory airstream, but inspiratory flow passing through the convoluted passageways incurs a substantial pressure drop. Filtration by the oral cavity is much less effective, but the pressure drop is also lower. As a result, humans normally breathe nasally until $\dot{V}_E$ reaches approximately 30 L/min, when oronasal breathing begins. This shift in breathing pattern occurs because, at lower flow rates, the pressure gradient between the atmosphere and pulmonary airways generated by inspiratory negative pressure in the lungs can overcome nasal resistance, but as flow rates increase, the nasal cavity pressure drop increases proportionally with Re. Consequently, oral breathing must supplement nasal breathing above roughly 30 L/min to maintain respiratory airflow. Since flow through the oral cavity has a lower pressure drop than flow through the nasal cavity, a greater proportion of airflow during oronasal breathing passes through the oral cavity. It is unclear whether oronasal breathing produces laminar or turbulent airflow in the oral cavity, though $\mathrm{Re} < 2100$ at $\dot{V}_E \leq 30$ L/min.

Pharyngeal turbulence results from the 90 degrees bend at the nasopharynx and irregular surfaces at the oropharynx and larynx. Vocal cords constrict the passageway and cause significant flow distortions and turbulence within the larynx. The passageway abruptly expands from the laryngeal orifice into the trachea. Rapid expansion produces a jet in proximal tracheal airflow and turbulence all along the trachea. Turbulent tracheal airflow occurs because the abrupt expansion causes reverse flow in the boundary layer, causing flow separation in the proximal trachea (Fig. 5.25). Under these conditions, turbulence forms at Re much less than 2300 in an abrupt expansion (often at $\mathrm{Re} \approx 300$).

Studies of airflow through models of bifurcating airways[76,77] show that turbulence generated in the

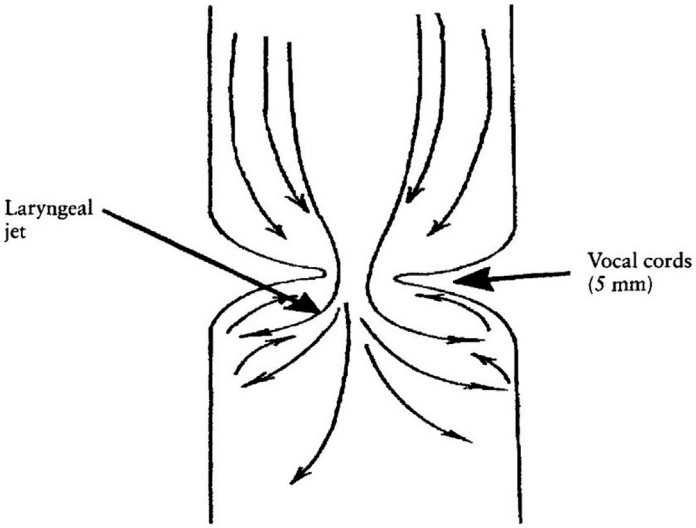


FIGURE 5.25 Schematic depiction of airflow pattern through the larynx. Note how eddies form downstream as air passes through the tracheal jet created by the vocal cords. This effect varies according to vocal chord position.

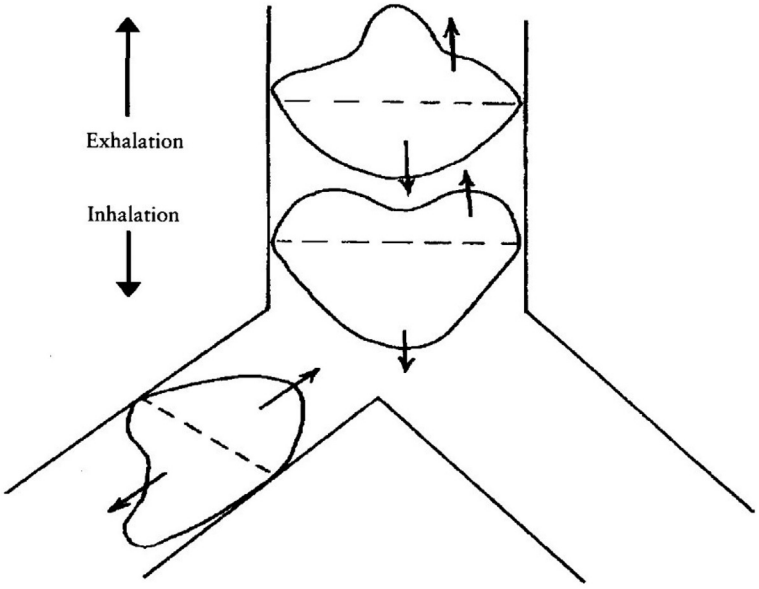


FIGURE 5.26 Airstream velocity profiles through a bronchial bifurcation. Shear forces along the medial bronchial wall cause flow distortions in the daughter tubes during inspiration. Bimodal velocity profiles generated in the parent tube during expiration are also caused by shear along the daughter tube medial walls. Modified from Scherer and Haselton.[77].

trachea does not sufficiently decay in the largest bronchi to produce laminar flow. Despite Re diminishing to below 1000, flow through at least the fourth generation bronchi is believed to be turbulent at all but the lowest $\dot{V}_E$. Eventually, however, flow disturbances dampen along the bronchial tree and flow becomes laminar. Entrance flow predominates because bronchi are typically only three to four diameters long. In addition, bifurcations modify velocity profiles because of asymmetric shear forces along inner and outer walls possibly caused by flow separation along the outer wall near the bifurcation (Fig. 5.26). Consequently, disturbed laminar flow appears to exist during both inspiration and exhalation in most bronchi.

Mean airstream velocity diminishes as inspiratory flow moves toward the lung parenchyma because of the rapid increase in total cross-sectional area. The largest increases in area occur in the distal bronchioles and pulmonary airways, causing u to approach zero because

$$u = \frac{Q}{A}, \tag{5.37}$$

where Q = volumetric flow rate ($\dot{V}_E$). Although flow is laminar, Poiseuille flow does not occur despite Re < 1.0 because of the complex geometry of these airways. Axial diffusion (also known as Taylor dispersion) accounts for mass transport within distal bronchioles

and combines convection and diffusion in an oscillating fluid with a low Re such that

$$\frac{\partial c}{\partial x} = K\left[\frac{\partial^2 c}{\partial y^2} + \frac{\partial^2 c}{\partial z^2}\right], \tag{5.38}$$

where c = solute concentration; x = direction of airflow; y, z = transverse directions to flow; and K = dispersion constant. K depends on the molecular diffusion coefficient, D_{ab}, where Fick's law defines mass transport by molecular diffusion as

$$m = -D_{ab}A\frac{dc}{dx}, \tag{5.39}$$

so that

$$K = f\left(D_{ab,}\frac{1}{U}\right), \tag{5.40}$$

where U = convective velocity. Pulmonary airways rely solely on molecular diffusion for mass transport.

5.2.4 Mucociliary clearance

Mucus gel floating on airway periciliary fluid becomes contaminated by atmospheric contaminants deposited onto the air–mucus interface during respiration. Deposition generally traps these materials, especially particulates, in the mucus gel and prevents them from being transported further by the airstream. Merely trapping these materials, however, serves little purpose because they would diffuse through the periciliary fluid to enter the epithelia and bloodstream. Cilia projecting from the apical surface of ciliated columnar epithelial cells, however, continuously propel mucus toward the epiglottis. Given sufficient mucus velocity, trapped contaminants will reach the epiglottis before they can diffuse through the periciliary fluid in sufficient quantity to cause injury or disease. Swallowing passes the contaminated mucus into the esophagus and eliminates the threat to the respiratory tract.

5.2.4.1 Ciliary location

Cilia are present along most extrathoracic airway surfaces except for the nasal vestibule, olfactory surfaces, nasopharynx, oropharynx, oral cavity, and portions of the larynx. Extrathoracic airway cilia gradually push mucus distally toward the epiglottis. In nonciliated regions, mucus moves by mechanical force (coughing, sneezing, and swallowing) or by gravity. Cilia line all tracheobronchial surfaces down to the pulmonary airways, propelling mucus proximally toward the epiglottis. Respiratory airway surfaces (respiratory bronchi, alveoli) are devoid of cilia.

5.2.4.2 Ciliary structure

Cilia are thin cylindrical hair-like structures with a cross-sectional radius of 0.1 μm projecting from the apical epithelial surface of ciliated columnar cells. Ciliary length is thought to correspond to periciliary fluid depth and range from approximately 7 μm in proximal airways to roughly 5 μm in more distal airways.[78] Each ciliated epithelial cell supports approximately 200 cilia at a density of eight cilia/μm^2. Short microvilli, possibly associated with secretory functions, are interspersed among the cilia.

Nonciliated cells separate fields of ciliated epithelial cells from each other. Synchronized ciliary movement, with a beat frequency in human proximal airways under normal conditions of 8–15 Hz,[79–83] propels mucus along the mucociliary escalator at a rate of up to 25 mm/min.[84,85] Beat frequencies appear to slow to roughly 7 Hz in more distal airways. Cilia move in the same direction and in phase within each field but cilia in adjacent fields move in slightly different directions and are phase shifted. These beat patterns result in metachronal waves that steadily move mucus at higher velocities (≃12–18 mm/min) than would be achievable by summing the motion of individual cilia.

5.2.4.3 Relationship of ciliary motion to mucus movement

Mucus gel is propelled toward the epiglottis by a two-phase ciliary beat cycle. Forward mucus movement occurs during the effective or power phase of the cycle, when cilia fully extend and traverse an arc perpendicular to the epithelial surface (Fig. 5.27). Claw-like structures, 25–35 nm long, project from each cilia tip and appear to assist in the mechanical transfer of momentum from cilia to mucus gel. Maximum mucus velocity depends on the extent cilia penetrate the epiphase during the power phase, periciliary and mucus gel viscosity, and cilia density.

During the recovery or preparatory phase, cilia bend over, swing back to start position generally parallel to the epithelial surface, and stiffen in anticipation of the next power phase. Ciliary bending and axial movement parallel to the cell surface significantly reduce retrograde momentum exerted on the surrounding fluid during the ciliary recovery phase because periciliary fluid viscosity is much lower than that of mucus gel. In addition, the no-slip condition along the epithelial surface also retards retrograde movement. Mucus viscosity and the presence of surrounding cilia further retard any retrograde mucus movement such that gravity has little effect on tracheobronchial mucociliary transport.

Derangement of metachronal motion impedes mucosal movement and increases the risk of disease or

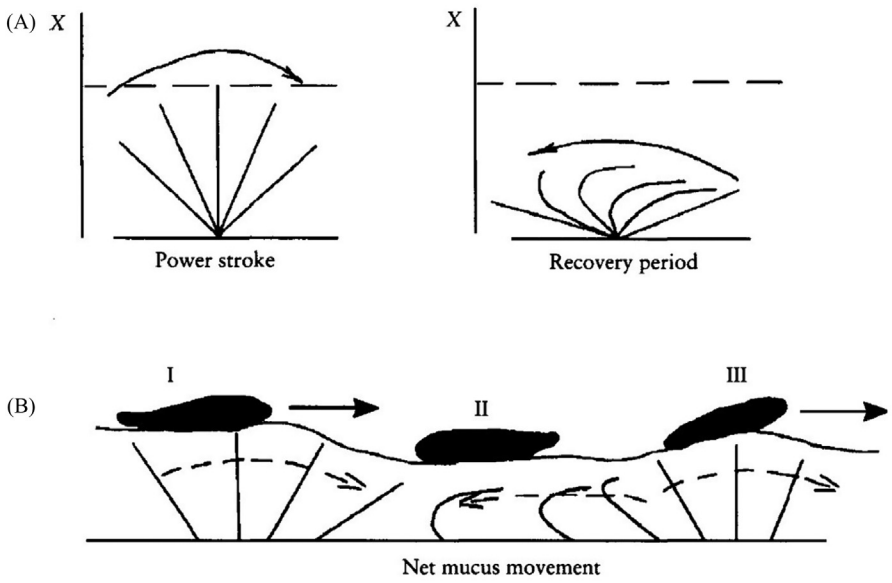


FIGURE 5.27 Components of ciliary movement, (A) Power and recovery phases of ciliary movement Arrows indicate the direction of ciliary travel, (B) Net mucociliary transport. Dotted arrows show the direction of cilia while the solid arrows show mucus transport. Note that net gel movement is forward in I and III while no gel movement occurs in II during the cilia recovery phase. Modified from Fulford and Blake.[78].

injury. Slowing mucosal velocity increases residence times in the affected airway region, permitting greater diffusion of deposited pathogens and toxins through periciliary fluid and increasing the risk of direct injury to airway epithelium and systemic injury via the bloodstream. Reducing the number, activity, or coordination of adjacent cilia or ciliary fields, hypersecretion of serous fluid or mucus gel, increased periciliary or mucus viscosity, and excessive periciliary fluid evaporation can each adversely affect mucociliary transport. Inspiring dry or cold air or cigarette smoke decreases periciliary fluid depth (altering cilia penetration into mucus gel) and cilia beat frequencies, which slows mucociliary transport. In addition, changes in periciliary fluid pH, ion concentration, or viscosity due to deposited chemicals, microorganisms (e.g., influenza, mycoplasmas), or systemic disease (e.g., asthma, cystic fibrosis) also inhibit ciliary beat frequency.

5.2.5 Airway heat and water vapor transport

5.2.5.1 Longitudinal and radial temperature/humidity gradients

Air passing through the respiratory tract must be properly conditioned (warmed and humidified) to optimize alveolar O_2 and CO_2 transport and minimize heat and water losses from the body. Respiratory air conditioning, as shown in Fig. 5.28, occurs as the airstream passes over the airway mucosal surfaces and results in both spatial and temporal humidity and temperature changes during each phase of respiration.[86,87] Submucosal blood temperatures (T_{blood}) are thought to be cooler in the extrathoracic airways and gradually warm along the length of the conducting airways until body core temperature (T_{core}, roughly 37°C) is achieved within the bronchi. This longitudinal temperature gradient exists because ambient air temperatures are generally lower than T_{core}. Inspired air extracts heat and water from airway walls as it passes through the lumen, warming the airstream and cooling the wall. The radial temperature gradient lessens as the ever-warmer airstream moves along the airway, causing gradually less heat extraction from the wall until eventually no heat is exchanged. The process is reversed during expiration. At the onset of inspiration, the walls of the airway are their warmest and approach end-expiratory airstream temperatures throughout most of the respiratory tract.[87,88] Typically, inspired air is cooler than this, creating a radial temperature gradient, such that the air within the airway lumen is cooler than the walls (Fig. 5.25). A radial water vapor concentration gradient also exists because air at the air–mucus interface (airway wall) is fully saturated,[89] while inspired air has a lower absolute humidity due to its being at a lower temperature than the airway wall.

Under most circumstances, passage of relatively cool inspiratory air along the airway results in convective and evaporative cooling of the mucosa while warming and humidifying the inspired air.[87,88] Airflow patterns caused by convoluted upper airway morphology augment heat and water vapor transport.[90–93] Radial temperature gradients can persist at least as far as the carina during oral breathing of room air,[94,95] causing heat and water vapor exchange to

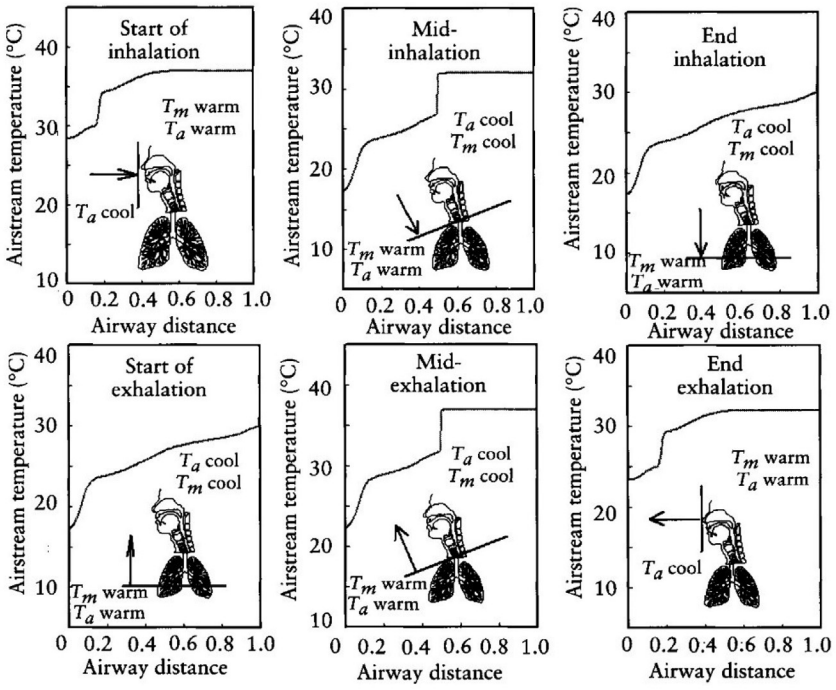


FIGURE 5.28 The relationship between the inspiratory and expiratory airstream front boundary and air, T_a, and wall, T_m, temperatures as a function of nondimensional distance from the nares. Solid lines shown on the airway picture indicate respiratory fronts and arrows depict the direction of airflow. Qualitative temperatures indicate relative temperature gradients across front boundaries. The graph depicts airstream temperatures as a function of nondimensional distance along the airway while breathing normal room air. Nondimensional distance is distance from the nares (x) divided by overall airway length (L).

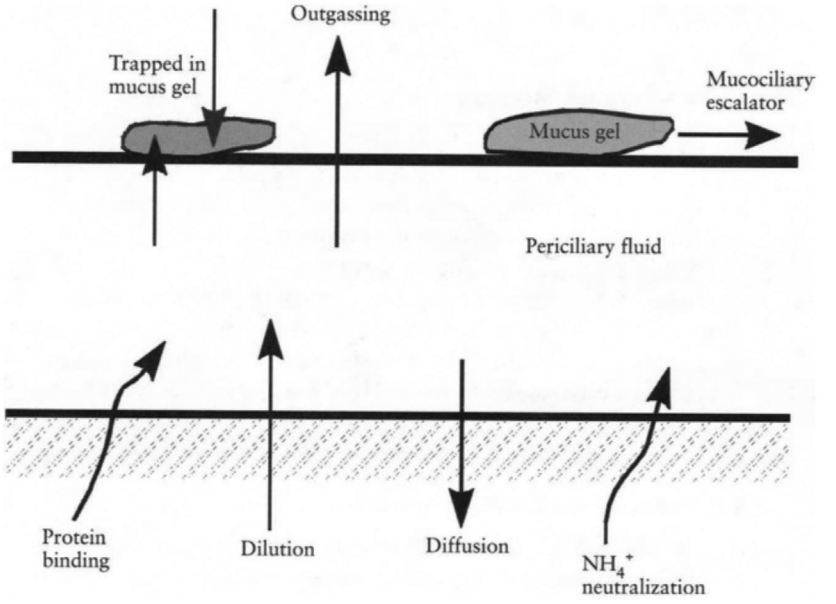


FIGURE 5.29 Various chemical and physical mechanisms which can affect ASL chemical concentration during breathing. Dilution due to transepithelial water exchange depends on the osmotic pressure gradient between periciliary and interstitial fluid.

occur for much of the length of the upper airway. At the end of inhalation, longitudinal temperature and water vapor concentration gradients exist along the airway (Fig. 5.29). Air reaching the most distal airway regions (beyond about the 14th bronchial generation) is believed to be fully conditioned (37°C, 100% humidity) during normal breathing. Exhalation causes warm air originating in the distal airway to pass over airway walls in proximal airway segments, which had been cooled during inspiration and are normally maintained below body core temperature.[96] The resulting temperature gradient causes airstream water vapor to condense and the airstream to lose heat to the airway walls. This acts to minimize net heat and water losses.[97]

Effectiveness of the conditioning process is dependent upon respiratory tract geometry, ambient air temperature (T_{amb}) and humidity (C_{amb}), inspiratory and

expiratory flow rates and volumes,[97] mucus temperature (T_m), airway wall blood temperature (T_{blood}), and flow rate in the submucosal capillary bed.[98–101] These variables interact and their effects are interdependent. For example, airway geometry plays a major role in conditioning since the rate of heat exchange between the wall and airstream, $\dot{q}$ is given by

$$\dot{q} = hA_s\Delta T, \tag{5.41}$$

where h = heat transfer coefficient, A_s = airway wall surface area, and ΔT = temperature difference between the airstream and wall.[102] Water vapor exchange is analogous to heat exchange and thus is also a function of A_s.[103] In addition, both mean gas velocity, $\overline{u}$ and residence time, t_w, are dependent on airway geometry since $t_w = f(\overline{u})$, $u = f(A)$, and conduit geometry directly affects the generation of flow disturbances and subsequent development of turbulent flow.

5.2.5.2 Role of airway heat and water vapor exchange in disease and injury

Exchange of heat and water vapor in the respiratory tract can significantly influence airway patency, alveolar gas transport, and whole body homeostasis, such as seen with cold- or exercise-induced bronchospasms. Maintaining airway patency is important in reducing airway resistance, maximizing inspiratory volume, and minimizing the work of breathing. The mechanism by which heat and water vapor exchange influences airway resistance has been widely debated[104–106] but probably depends on both airway mucosa heat and water losses.[96,107] It has been suggested that alterations in the conditioning of inspired and expired air can lead to increased total airway resistance[88,96,98,104] by causing increased nasal blood flow,[108,109] altering vascular tone and permeability in the bronchial circulation,[110] and increasing airway smooth muscle tension.[111–114] Under pathological conditions, diminished conditioning may also increase mucus thickness,[64,115] which in extreme cases causes increased airway resistance by reducing airway cross-sectional area and increasing shear stress at the air/mucus interface.[116] In addition to effects on the conducting airways, alveolar O_2 and CO_2 transport could be hampered if air has not been warmed to body temperature (37°C) and fully humidified by the time it reaches the alveoli.

The importance of respiratory heat and water losses is not confined to the respiratory structures. Inspiration of cold, hot, or dry air poses the potential threats of thermal injury or desiccation to the airway epithelium[95,101,115,117,118] and is a challenge to whole-body thermoregulation. Under certain conditions, such as hyperbaria,[119,120] airway heat losses can account for a considerable percentage of total body heat production (in some cases >100%).[120] Normally these threats are ameliorated by rapid moderation of inspired air temperature and humidity by exchanging heat and water vapor between the mucus and airstream in the upper airway.[121,122] Recovering much of the heat and water vapor contained in expired air minimizes heat and water losses to the ambient environment[123] and aids in whole-body thermoregulation.

Heat and water vapor transport can also lead to respiratory impairment, infection, and injury through thermal and osmotic stresses occurring at the mucosal epithelium.[115,123] These stresses cause changes in mucus osmolarity, pH, ciliary activity, and cellular transport,[124,125] resulting in altered mucosal thickness[64,115] and impaired airway defenses. Normal breathing allows microorganisms, pollutant gases, and particulate matter to contact the mucus coating (comprised of mucus gel and periciliary fluid) atop the apical surface of respiratory epithelium. A complex system of chemical, immunological, and mechanical defense mechanisms protects the respiratory epithelium and alveoli from potential diseases or injury caused by noxious airstream components.[126] Aside from chemical neutralization of pollutants in the airstream[127,128] and physical defenses such as bronchoconstriction, coughing, and particle impaction caused by airway morphology,[126,129–131] the defense of the airway depends on the physical and chemical properties of airway mucus (e.g., chemical detoxification reactions with proteins[132]) and the ciliary mechanism which moves it toward the epiglottis (mucociliary escalator). Inspiring cold dry air can impede mucociliary transport, reducing mucus velocity and increasing the risk of airway disease or injury.

Airway deposition patterns of inspired hygroscopic particles are also affected by airway heat and water vapor exchange. Inspired particles passing from a relatively dry ambient environment into the fully saturated airway quickly adsorb water from the surrounding airstream. Water vapor adsorption at the particle surface increases hygroscopic particle mass m_p as a function of particle diameter d_g and the water vapor concentration gradient between the bulk fluid, c_∞, and particle surface, c_0, according to

$$\frac{dm_p}{dt} = 2\pi d_g D_w C_w (c_\infty - c_0), \tag{5.42}$$

where D_w is the diffusion coefficient of water vapor in air and C_w is the slip correction factor = $f(d_g, D_w, T)$.[133] In addition, d_g can be determined from m_p and particle density, ρ,

$$d_g = 2\left[\frac{3m_p}{4\pi\rho(X)}\right]^{1/3}, \tag{5.43}$$

where X = particle composition at time t.[134] Particle growth continues until equilibrium is reached between particle surface and bulk airstream water vapor pressure. Given sufficient growth, extremely fine particles that might otherwise pass entirely through the airway during the breathing cycle deposit along the airway because of the increase in mass. Compromised extrathoracic submucosal blood flow due to injury or disease will change water vapor exchange between the mucosal surface and inspired airstream. This in turn will alter the growth patterns of inspired hygroscopic particles. This process may play a role in lower airway injury caused by inspired toxins (e.g., acid aerosols) or succumbing to diseases normally present in the ambient environment (e.g., pneumonia) that seem to affect weakened individuals.

5.2.6 Endogenous ammonia production

Evidence suggests that the highest airway NH_3 concentrations occur in the oral cavity,[127] the only segment of the respiratory system that is normally colonized by bacteria, and that the remainder of the airway, including the nasal passages, have significantly lower levels. Diffusion of NH_3 from the bloodstream into the airway lumen is probably the primary source of NH_3 for the entire airway except the oral cavity.[127] Blood ammonium concentration, $[NH_{4+}]_B$, is normally the consequence of protein deamination during dietary protein digestion,[135] though deamination of AMP in muscle tissue during strenuous exercise can significantly increase $[NH_{4+}]_B$.[135–137] Ureolysis by gastrointestinal bacteria can also contribute to $[NH_{4+}]_B$.[135] It is theorized that airstream NH_3 concentration, $[NH_3]_A$, is in equilibrium with $[NH_{4+}]_B$ throughout most of the respiratory tract,[127] though this has not been demonstrated. Airway mucus may impede diffusion of blood ammonia into the airway lumen because of its net negative charge.[132] The effect this Donnan exclusion phenomenon may exert on airway NH_3 diffusion has not been demonstrated, since $[NH_{4+}]_B$ has not yet been correlated with $[NH_3]_A$ in humans.

Bacterial catabolism of oral food residue is probably responsible for a higher $[NH_3]_A$ in the oral cavity than in the rest of the respiratory tract.[127] Ammonia, the byproduct of oral bacterial protein catabolism and subsequent ureolysis, desorbs from the fluid lining the oral cavity to the airstream.[138,139] Saliva, gingival crevicular fluids, and dental plaque supply urea to oral bacteria [139] and may themselves be sites of bacterial NH_3 production, based on the presence of urease in each of these materials.[138,140] Consequently, oral cavity $[NH_3]_A$ is controlled by factors that influence bacterial protein catabolism and ureolysis. Such factors may include the pH of the surface lining fluid, bacterial nutrient sources (food residue on teeth or on buccal surfaces), saliva production, saliva pH, and the effects of oral surface temperature on bacterial metabolism and wall blood flow. The role of teeth, as structures that facilitate bacterial colonization and food entrapment, in augmenting $[NH_3]_A$ is unknown.

The significance of pH is particularly interesting since pH may either augment or diminish NH_3 production. The possible mechanisms by which pH affects NH_3 production are (1) inhibition of bacterial metabolism, (2) pH-dependent changes in urea metabolic pathways, (3) pH-dependent bacterial utilization of glucose and urea as energy sources, and (4) increased bacterial utilization of NH_3 in amino acid synthesis. Ureolysis appears to be very sensitive to pH; NH_3 production increases as salivary pH is reduced from 7.0 to 6.0[141] but decreases significantly when pH is lowered to approximately pH 2.5.[142] A salivary pH of 2.5, however, only temporarily depresses NH_3 production[142] since NH_3 diffusing from the bloodstream may neutralize acids responsible for reduced oral cavity pH and slowly increase oral pH. Ureolysis may increase rapidly at some pH threshold, perhaps near pH 5.5,[143] because of the steady supply of salivary urea.[139] Oral pH continues to increase as NH_3 is generated,[143] with peak NH_3 production thought to occur near an oral pH of 6.0.[141] Salivary HCO_3^- may act to buffer increases in oral pH and thus maintain NH_3 production rates.[144] Therefore an increase in salivary flow will not only increase the availability of urea to oral bacteria but also help maintain oral conditions advantageous for NH_3 production. Theories regarding in vivo regulation of oral NH_3 production are speculative since the bulk of data was obtained from in vitro studies of salivary sediments and dental plaque samples; greater knowledge of in vivo interaction between oral cavity NH_3 production, pH, and saliva is needed.

Fasting combined with poor oral hygiene results in an elevated dental plaque pH ($\sim$ 7.6),[145] suggestive of active ureolysis. Whether fasting or poor oral hygiene is responsible for the higher pH is unclear. Carbohydrates in the mouth lower dental plaque pH,[145] while glucose, in particular, buffers oral pH,[146] and thereby inhibiting NH_3 production.[141] The formation of NH_3 appears to be inhibited by glucose for two other reasons: (1) it is preferentially used for bacterial energy production in place of proteins and peptides, and (2) its presence favors acid-producing bacteria that scavenge NH_3.[141] Oral food residues with a high protein content should serve as a rich substrate for oral NH_3 production through bacterial deamination.

5.2.7 Respiratory defense mechanisms

Breathing exposes a large body surface area to attack by noxious materials or pathogens present in the ambient atmosphere. A complex series of defense mechanisms, including physical removal, chemical neutralization, and immunological response, protect against biological, chemical, or mechanical injury. Physically removing toxins or pathogens from the airstream by coughing, sneezing, movement along the mucociliary escalator, or phagocytosis in the alveoli reduces exposure concentrations in vulnerable airway regions. Airborne toxins can be neutralized by endogenous NH_3, while deposited materials are diluted, buffered, and neutralized by periciliary fluid and mucus gel. Immunoglobins and enzymes present in periciliary fluid and along alveolar surfaces can eliminate deposited pathogens that have not been physically removed.

5.2.7.1 Vapor-phase neutralization

In addition to typical atmospheric or metabolic constituents (N_2, O_2, H_2O, and CO_2), breathing transports chemical species in the form of vapors and particulates along the respiratory tract. While O_2 and CO_2 gas exchange occurs solely in the lung parenchyma, air/blood exchange of other species can and does occur throughout the airway. The principal airway absorption sites of gases such as O_3 and SO_2 are largely determined by their water:air partition coefficient (the concentration ratio at equilibrium) and water solubility.[147] Highly water-soluble chemical species with high water:air partition coefficients are generally absorbed in the extrathoracic airways, while less soluble species pass beyond the extrathoracic airways in relatively high concentrations (see Section 5.3).

Concentration gradients provide the driving force for gaseous chemical species diffusion between the luminal gas mixture and ASL. Factors that alter this gradient, such as local airstream concentration, chemical reactivity, lipid solubility, and ASL metabolism, modulate local absorption or reentrainment into the airstream. Local airstream/ASL concentration gradients drive diffusion into or out of ASL along a given airway length.[148,149] As the inspiratory air passes along the airway and comes into contact with previously unexposed ASL, chemical species follow the concentration gradient and diffuse into ASL. The leading edge of the inspiratory wave becomes increasingly depleted of the diffusing species, increasing proximal ASL concentrations while reducing airstream concentrations downstream. Consequently, ASL absorption decreases in more distal airways. Reentrainment can occur during expiration if concentration gradients are reversed, that is, tracheobronchial and extrathoracic ASL concentrations exceed those found in gases flowing outward from the lung parenchyma.

The rate at which an absorbed chemical species is removed from the ASL determines whether reentrainment occurs during a breathing cycle.[147] Slow removal rates relative to the breathing cycle allow the concentrations in the ASL to be higher than in the expiratory airstream. Fig. 5.29 shows processes that diminish the ASL concentration of absorbed chemical species. Metabolic processes or interactions with ions and other chemically reactive substances found in ASL can eliminate absorbed chemicals. Diffusion through airway epithelium into the submucosal bloodstream is an alternative removal pathway (Fig. 5.30) that depends on lipid solubility or facilitated transport across cell membranes.

5.2.7.2 Aerosol defense

Particle deposition

Particles entrained in the airstream deposit along the airway as a function of size, density, airstream velocity, and breathing frequency. Sizes of roughly spherical or irregularly shaped particles are commonly characterized by relating the settling velocity of the particle to that of an idealized spherical particle.[150] For example, an irregular particle which settles at the same rate as a 5 μm spherical particle has a MMAD of 5 μm. Since spherical particle mass, m_p, is a function of particle diameter, d

$$m_p = \frac{\pi}{6} d^3 \rho, \tag{5.44}$$

where ρ = particle density, MMAD can be viewed as representing the mass and buoyancy of a spherical particle equivalent to the randomly shaped airborne particle.[150] Three basic deposition mechanisms, impaction, sedimentation, and diffusion, act on all entrained particles, but each mechanism predominantly affects specifically sized particles within a given airway region.[150]

The nasal passages form an efficient filtration mechanism for inspired air, removing larger particulates (>3 μm MMAD) before they can enter the thoracic airways. The very largest inspired particles (roughly 10 μm MMAD and larger) impinge on nasal hairs (vibrissae) and are mechanically removed from the nasal cavity (e.g., by blowing one's nose). Particle inertia generally causes the remaining larger particles to deposit along the nasal cavity surfaces by impaction because of convoluted nasal geometry. A particle impacts an airway wall when the path length to the wall equals the lateral displacement, L, occurring while the particle moves at a velocity u along a streamline altering direction by an angle θ, which is given by

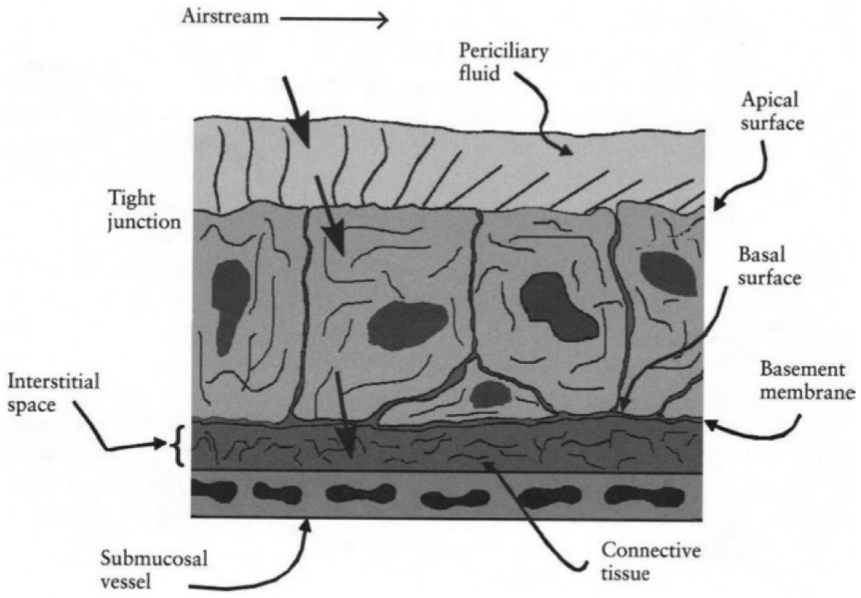


FIGURE 5.30 Diffusional pathway of deposited materials from the airstream to interstitial space. Large arrows depict diffusion across the ASL, apical, and basal call membranes. Na^+ and Cl^- passively diffuse across the cellular apical surface while K^+ diffuses and a Na^+-K^+-2CI^- co-transporter exchanges these ions across the cellular basal surface. Active transport (Na + -K + pump) also transports Na^+ out of and K^+ into the cell across the cellular basal surface. Water diffuses across the different cell membrane surfaces depending upon the existing osmotic pressure gradient. Diffusion of water and salts through the paracellular spaces between cells can also occur.

$$L = \frac{u u_t \sin\theta}{g}, \tag{5.45}$$

where terminal velocity, u_t, is the particle velocity at which particle inertia is balanced by drag forces. For $1.0\ \mu m \le d \le 40\ \mu m$,

$$u_t = \frac{g d^2}{18\mu_a}(\rho_p - \rho_a), \tag{5.46}$$

where g is the gravitational constant, μ_α is the air viscosity, and ρ_p and ρ_a are the density of the particle and air, respectively. Larger particles that successfully traverse the nasal passages typically impact the nasopharyngeal wall at the 90 degrees turn beyond the distal edge of the nasal cavity.

Finer particles ($<3\ \mu m$), termed respirable particles, pass beyond the extrathoracic airways and enter the tracheobronchial tree. Impaction plays a significant role near the tracheal jet, but sedimentation predominates as the effects of rapid conduit expansion dampen in the distal trachea and beyond. Sedimentation occurs when gravitational forces exerted on a particle equal drag forces, that is, when particle velocity falls to u_t. As mean inspiratory airstream velocity gradually declines along the tracheobronchial tree, particle momentum diminishes and 0.5–3 μm MMAD particles settle out of the airflow and onto mucosal surfaces.

Mean airflow velocities approach zero as the inspired airstream enters the lung parenchyma, so particle momentum also approaches zero. Most of the particles reaching the parenchyma, however, are extremely fine ($<0.5\ \mu m$ MMAD), and particle buoyancy counteracts gravitational forces. Temperature gradients do not exist between the airstream and airway wall because the inspired airstream has been warmed to body temperature and fully saturated before reaching the parenchyma.[99,100] Consequently, diffusion driven by Brownian motion is the only deposition mechanism remaining for airborne particles. Diffusivity, D_c, can be described under these conditions by

$$D_c = \frac{kT}{3\pi\mu d}, \tag{5.47}$$

where k is the Boltzmann constant, T is the absolute temperature, μ is the air viscosity, and d is the particle diameter. Particle displacement, δ, is a function of residence time, t, and D_c such that

$$\delta = (6 D_c t)^{0.5}. \tag{5.48}$$

Consequently, any breathing pattern which increases pulmonary residence times, such as breath-holding, increases fine particle deposition throughout the airway.

Where along the airway inspired particles deposit depends on particle mass, since the deposition mechanism depends on particle MMAD. Passage through the airway has no effect on nonhygroscopic particle mass (e.g., fly ash), and initial MMAD determines the deposition pattern (Fig. 5.31). In contrast, hygroscopic particles (e.g., acid droplets) increase in mass when exposed to humid environments like the respiratory tract. Particle properties (e.g., chemical composition, ionic concentration, and particle surface area) and airstream conditions (e.g., temperature, RH, and $\dot{V}_E$) which affect hygroscopic growth consequently play major roles in determining particle mass and deposition patterns (Fig. 5.32).

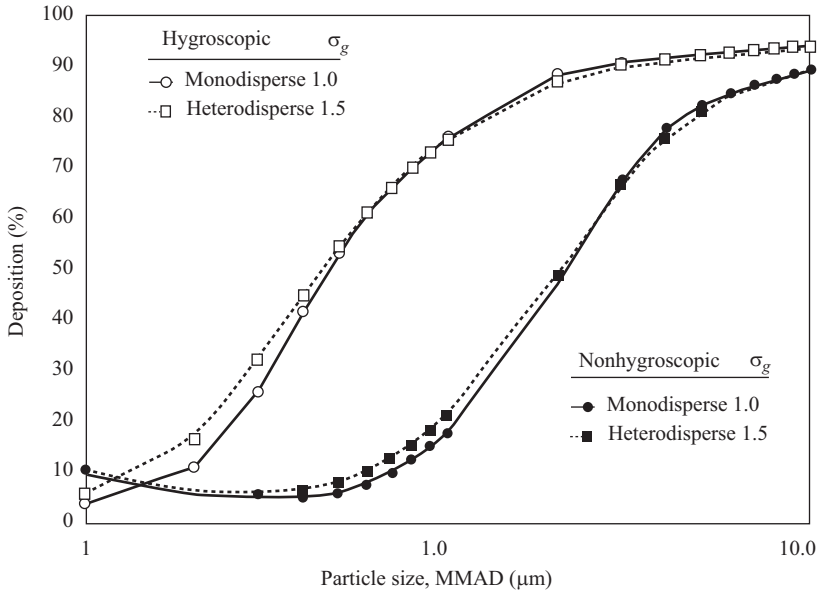


FIGURE 5.31 Estimated overall airway deposition as a function of initial particle size and particle hygroscopicity for particles with mass median aerodynamic diameters (MMAD) between 0.1 and 10 μm.[102] Geometric dispersion, a measure of particle size distribution, principally affects only smaller MMAD.

Acid aerosol neutralization

Sulfuric acid (H_2SO_4) and ammonium bisulfate (NH_4HSO_4) contribute importantly to ambient acid aerosols, particularly in geographic locations where sulfur-rich coal is used for power plant fuel, such as the eastern United States.[151] Studies on animals and human subjects have shown that H_2SO_4 and NH_4HSO_4 alter mucociliary transport in a dose-dependent fashion[152–154] and can adversely affect pulmonary function in humans.[154] While this effect on clearance has generally been attributed to hydrogen ion concentration, $[H^+]$, the work of Schlesinger et al.[155] suggests that, for equivalent inhaled $[H^+]$, H_2SO_4 elicits a greater change than NH_4HSO_4. If this observation is confirmed, it would appear that the molecular form of the inhaled acid may play a significant role, perhaps through differences in hygroscopic growth and neutralization rate between H_2SO_4 and NH_4HSO_4 particles.

For a given ambient concentration of acid aerosol, the dose of acid delivered to the respiratory tract is in large measure determined by the pH and particle size of the aerosol. Due to the efficiency of the upper airways (particularly the nasal passages) in filtering coarse (>3 μm) particles, submicrometric acid aerosol particles pose the greatest risk to the lower airways. Ambient acid aerosols are overwhelmingly submicrometric in size distribution at most relative humidities.[156] Submicron acid particles therefore merit special attention in the attempt to understand the action of acid aerosols on airway health, particularly as they comprise a large proportion of acidic environments.[156]

Airstream neutralization of acid aerosols by NH_3 present in the airway lumen reduces the health risk associated with acid particles by reducing the acid concentration prior to particle deposition.[127,157] In addition, the liquid lining of the respiratory tract probably acts as a chemical buffer,[158] further reducing the health hazard posed by inspired acid particles. Principal factors controlling airstream neutralization of acid aerosols, which is considered to be a diffusion-limited process, are particle surface area, $[NH_3]_A$, and particle residence time in the airstream.

Since NH_3 is highly water-soluble and neutralization within the droplet occurs rapidly,[159] the rate-limiting step in acid neutralization is normally NH_3 transport to the air/droplet interface, which is dependent on $[NH_3]_A$ and particle surface area. At high $[NH_3]_A$, the rate of NH_3 uptake across the air/droplet interface is given by

$$\frac{dC_s}{dt} = \frac{3D_g}{r^2}[NH_3]_A - C_S H q_{NS}, \qquad (5.49)$$

where C_s is the NH_3 concentration in the acid droplet, D_g, is the airstream NH_3 diffusion coefficient, r is the droplet radius, H is the Henry's law coefficient, and q_{NS} is the activity coefficient of neutral undissociated species in solution in the droplet.[160] Particle size of inhaled liquid aerosols does not remain constant within the airways, however. Water will condense on the surface of particles as they move distally along the airway because of local increases in RH. [134,161,162] The resulting increase in particle radii due to hygroscopic growth will reduce NH_3 concentration according to Eq. (5.47), while the increase in particle size will increase particle deposition.[163,164] However, increasing the particle radius results in greater particle surface area, which should increase NH_3 uptake, and thereby opposing the reduction in particle $[NH_3]$.

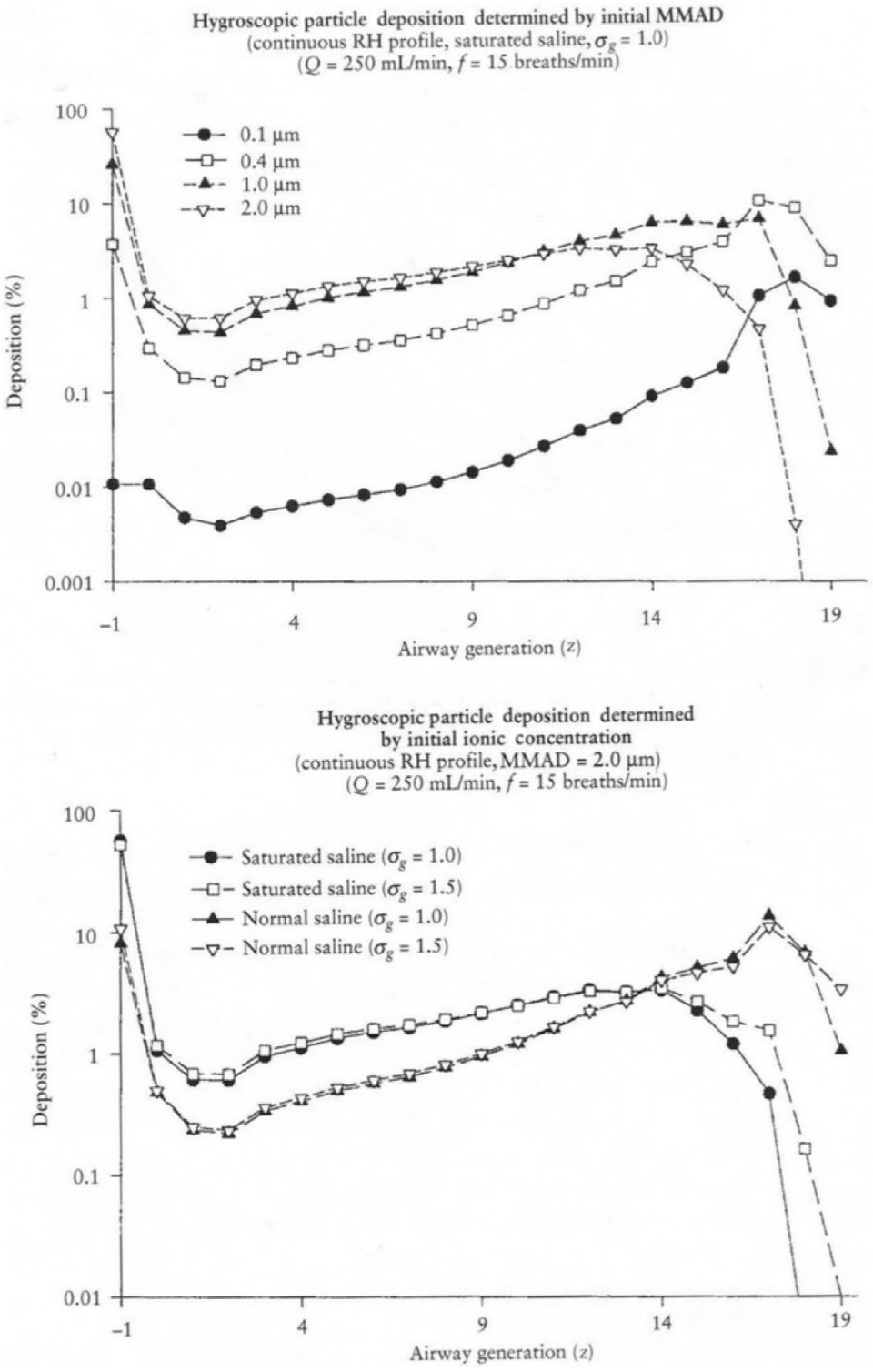


FIGURE 5.32 Predicted effects of initial hygroscopic particle properties (initial diameter, ionic concentration) on the airway deposition profile.[102] Larger initial particles are predicted (left) to deposit in the more proximal airways while fine particles reach pulmonary airways in much greater concentrations. Initially high ionic concentrations (saturated saline) are predicted (right) to deposit primarily in upper airways, probably due to their rapid growth during transit through the upper airway. More moderate growth rates represented by normal saline aerosol result in greater predicted deposition in the pulmonary airways. Airway generation −I represents the upper airway while the trachea is given as generation 0.

Elevated inspiratory flow rates reduce airway $[NH_3]$ by diluting the endogenous NH_3. The effect of flow rate on $[NH_3]_A$ should be most apparent in the upper airway due to the large NH_3 concentration difference between ambient air (as the diluent) and the upper airway (containing endogenous NH_3). The diluting effects of ventilation should be less evident in the large central airways and diminish steadily as the inspiratory wavefront moves distally along the airway because air velocity declines as airway volume increases and NH_3 is a highly soluble gas and most likely equilibrates rapidly with blood ammonia. As a result, flow-rate effects on $[NH_3]_A$ should be negligible after approximately the eighth bronchial generation because the rapid increase in airway volume causes air velocity to decline and the airstream to become more diluted. Since the expiratory wavefront is anticipated to encounter uniform $[NH_3]_A$ throughout the lower conducting airways, increased expiratory flow rates should have no effect on $[NH_3]_A$ until the wavefront reaches the upper airway. With nasal expiration, there may be no longitudinal NH_3 concentration gradient except at the nares, unless NH_3 diffuses from the oral cavity into the oropharynx.

Despite our limited knowledge of $[NH_3]_A$ distribution and control, there are at least two mathematical models[129,160] that attempt to predict the neutralization of inhaled acid aerosols. Cocks and McElroy[160] base their model on acid particle growth by predicting equilibrium particle size as a function of initial particle diameter and RH. Molecular diffusion is a major determinant of particle growth in the Cocks and McElroy[160] model, particularly for submicrometric particles, because their size approaches the mean free path of water vapor. Neutralization of acid particles was determined as a function of time and constant $[NH_3]_A$ at parenchymal conditions. Cocks and McElroy[160] did not account for higher levels of ammonia in the upper airways, which suggests that the bulk of neutralization will occur in the upper airway, at lower RH and temperature than in the parenchyma. The effect of a longitudinal intraairway $[NH_3]_A$ gradient on neutralization was also not considered.

Larson[129] developed a model of acid aerosol neutralization that accounts for RH and temperature gradients along the airway. The longitudinal gradients used in the model were taken from the model of Martonen and Miller,[165] which did not account for airway geometry or ventilation. Two fixed intraairway $[NH_3]_A$ gradients, reflecting oral and nasal breathing, were modeled and both assumed linear concentration gradients along the airway (with a step change at the oropharynx during nasal breathing). Dilution due to increased flow rate was not modeled, nor is it clear whether the $[NH_3]_A$ gradients changed during exhalation. Neither Cocks and McElroy[160] nor Larson[129] accounted for gas-phase NH_3 transport (except at the particle surface) or the possible effect a reduction in oral wall pH, caused by exposure to acid aerosol, would have on segmental control of $[NH_3]_A$.

Both models predict that two factors would decrease $[H^+]$ in the particle: (1) hygroscopic growth of the particle, which is thought to be capable of reducing particle $[H_2SO_4]$ from 15.3 to 0.22 M and (2) particle neutralization due to $[NH_3]_A$, which is potentially more significant but likely to be more variable within an exposed population. Without neutralization, highly acidic submicrometric particles (pH = 0.66) were predicted to be deposited onto distal airway tissues.[129] To refine our understanding of the potential for acid neutralization to mitigate adverse health effects, the assumptions regarding NH_3 concentration appear to be critical for any mathematical description of acid aerosol effects. Cocks and McElroy[160] demonstrated the importance of NH_3 concentration estimates, with complete neutralization of submicron droplets at 500 $\mu g/m^3$ NH_3 but less than 15% neutralization at 50 $\mu g/m^3$ NH_3. Since measured oral NH_3 concentration vary over a wide range, 144−1536 $\mu g/m^3$,[127,142] model predictions would improve if the factors controlling NH_3 production and $[NH_3]_A$ were known.

Mucociliary escalator

Bacterial and viral inoculants deposited onto airway mucus are normally inactivated by immunoglobins and macrophages[166] while being physically removed by the mucociliary escalator (mucociliary clearance).[64,126] Deposition and adherence of particulates onto airway mucus also prevents aspirated pollutants,[156,167] viral particles,[126,168] and infected epithelial cells shed from the airway wall[126] from reaching the alveoli.[64,68,168] Airway mucus also plays an important role in buffering and chemically neutralizing inhaled pollutant gases.[169] In addition, mucus serves to protect the airway epithelium against injury caused by rapid fluctuations in airstream temperature, T_a, and humidity, C_a.[115]

Disruption of these defense mechanisms can lead to bacterial colonization or viral infection. Mucus temperature is important in controlling respiratory infections because decreasing T_m below central body core temperature not only impairs ciliary movement,[124,125] but also enhances viral replication,[67] greatly increasing the likelihood of respiratory infection. Drying of airway mucus also increases the possibility of respiratory infection by reducing mucus thickness and impairing mucociliary clearance.[170,171]

Exhaled nitric oxide

Nitric oxide (NO) is produced by airway epithelial cells, airway and vascular endothelial cells, as well as wandering inflammatory cells, and have different physiologic and regulatory functions for the airways, and the function may both beneficial or unfavorable. Also in nasal mucosa production of NO is abundant,

and bacterial function in the mouth also produces NO ([172,173]). NO is produced when nitric oxide synthase (NOS) enzymes convert L-arginine to L-citrulline. There are three isomers of NOS; the constitutive NOS (cNOS) isoforms are neuronal (nNOS or NOS1) and endothelial (eNOS or NOS3), both activated by calcium. Inducible NOS (iNOS or NOS2) is induced by inflammation or infectious stimuli. In respiratory mucosa all these isofoms of NOS can be present. NO and its metabolites are suggested to have several physiological functions, for example, regulation of smooth muscle and cilia function, as well as antimicrobial effect.

NOS1 is localized in cholinergic nerves in the airways and they could act as a neural antagonist (nonadrenergic and noncholinergic) to asetylcholine and cause bronchodilation, and conversely, reduction of NOS1 has been reported to result in increased neuronal bronchoconstriction. NOS1 is expressed also in epithelial cells of airways and type 1 pneumocytes. There is evidence that the expression and activity of NOS1 would be increased in COPD patients' peripheral lung regions as a result of oxidative stress and the role of NOS1 might here be unfavorable for the lungs[174,175]

NOS3 is expressed in endothelial cells of the bronchial and pulmonary circulation as well as in alveolar endothelial cells and airway epithelial cells. NOS3 is involved in regulating vascular flow and is supposed to have a role in reducing plasma exudation in the airways. Epithelial NOS3 has been suggested be involved in mucociliary clearance by regulating ciliary beating. NOS3's effect for the lungs is probably beneficial, because defective NOS3 has suggested to lead to bronchial hyperreactivity, and in COPD and emphysema its expression has been reported to be low.[174]

NOS2 (iNOS), the iNOS increases NO production associated with asthma with eosinophilic inflammation when several cytokines associated with asthmatic inflammation, activate iNOS. Also changes of acid−base balance on airway mucosa, for example, in acute asthma may lead to deliberation of fraction of exhaled NO (FENO) from nitrite. As a gas and being a small molecule, FENO may directly pass cell membrane, and its increased production can be measured from exhaled air (ATS/ERS 2005),[172] and FENO level has been measured in diagnosis of eosinophilic inflammation in asthma diagnostics as well as in follow-up of asthma. Nose and nasopharynx produce FENO markedly more than the lower airways in healthy subjects, However, in asthmatics, the Feno levels have been at similar level as measured from trachea, lower airways or expiratory air indicating that increased FENO depends of increased NO production in the lower airways.

FENO measurement is used mostly as an indicator on eosinophilic inflammation and indication to treat with inhaled corticosteroids. However, if NO has some effect on the bronchi remains obscure. Anyway, as the lungs are exposed to oxidative stress, NO may be involved in production of highly reactive species as peroxynitrite, which may be applied in the development of emphysema.[176] The levels of FENO in COPD are usually normal, but may be increased in severe COPD and indicates that the patient might benefit of inhaled streroids.

5.3 Toxicity and risks induced by occupational exposure to chemical compounds

5.3.1 Introduction and background

5.3.1.1 Health hazards of occupational exposure

Workers exposed to chemicals often experience discomfort and adverse health effects which may progress to occupational diseases. Although working conditions have improved markedly during recent decades, in general the number of individuals suffering from occupational diseases has declined rather slowly. In addition, the number of new cases of registered occupational health diseases depends on employment circumstances; there is a natural decline during times of economic recession. This gradual change is due to several factors: the diagnostic criteria have become less stringent, physicians have learned to recognize occupational diseases better, and many occupational diseases develop slowly and thus the present situation reflects, at least to some extent, past exposures. Furthermore the significance of occupational allergies has increased and allergic reactions can be caused even by low exposure levels.

Awareness of toxicity of chemicals has led to safer use and better protections and thus occupational exposure to harmful chemicals, such as benzene and carbon disulfide has decreased. Most of the current occupational diseases are caused by exposures which are not particularly acutely toxic, but cause irritancy and allergies. Typical exposures causing allergies are animal and flour dusts. If one considers actual chemicals, then isocyanates have become a major problem, principally via their ability to cause sensitization. Historical exposure to asbestos and other mining dusts still leads to numerous new diseases, many of which are very serious, even fatal. Solvents and pesticides are the groups of chemicals probably causing the largest amount of acute poisoning-type occupational diseases.[177,178]

Traditionally, the greatest risk due to chemicals has been considered to occur via inhalation. Chemicals may also penetrate through the skin. Water-based products are increasingly replacing solvent-based products in many applications, such as painting, printing, and gluing. The water-based products may, however,

contain glycol derivatives which penetrate through the skin with ease. Many chemicals also irritate or sensitize the skin. Chromium, nickel, and epoxy resins are examples of common occupational skin allergens.

Ventilation engineers and occupational hygienists must be aware of the risks of chemicals with a high acute toxicity. Chemicals which are odorless (e.g., CO), paralyze the sense of smell (e.g., hydrogen sulfide), or cause pulmonary edema as a delayed effect (e.g., nitrogen dioxide and ozone) are especially insidious. Often these gases are produced as unwanted by-products. For example, nitrogen dioxide and ozone may be formed due to oxidation of air during welding. Welding near sources of chlorinated solvents, such as perchloroethylene, may cause pyrolysis and the formation of phosgene.

Occupational exposure limits (OELs) have been set in most industrial countries to prevent excessive exposures. The limits for the most common exposures are based on experimental animal and epidemiological studies. Most novel agents have now generally gone through extensive toxicological testing. For the older chemicals, usually a plethora of epidemiological data is available.

When the incidence of occupational diseases was compared with the frequency of OEL violations in Finland, a rather good correlation was observed. This indicates that these OELs are reasonably well defined. This is also natural because they are based on long-term exposure history of a large number of people. However, the OELs for many chemicals are given still by limited scientific evidence, and when the new information about toxicity is available, it may lead to large scale updates of OELs. In addition, most chemicals still have no OEL. Only about 2000 chemicals have an OEL in some country.[179]

A particularly strict exposure-control policy is applied for carcinogenic chemicals. The OELs are usually lowered considerably even when a chemical is only suspected of being a carcinogen. When the evidence becomes stronger, the OELs are usually tightened further. Vinyl chloride provides a good example; its OEL was first lowered to 20 ppm from 500 ppm and then further to 3 ppm in Sweden in 1974–5 when its ability to cause a very rare type of cancer, angiosarcoma of the liver, was detected. The rarity of the disease made it possible to locate the association; on the other hand, the practical impact of this carcinogenic potency also remains rather low. It has been estimated that less than 400 angiosarcoma cases will appear worldwide due to vinyl chloride exposure (in comparison with the number of occupational cancers caused by asbestos which is already about 1000-fold higher).[180] Internationally, there is an ongoing vigorous discussion on whether there are possible thresholds for genotoxic carcinogens. In many instances these compounds are considered to have no safe dose. If one assumes that there is some threshold also for genotoxic carcinogens, this would have major consequences for the assessment of risks of carcinogenic compounds.[181–183]

Since the OELs provide the basis for ventilation requirements, an astute designer tries to find out how secure the OELs of the chemicals which will be used in the plant he or she is planning. Some of the chemicals used may totally lack OELs. Therefore it is advisable to become familiar with the relevant literature, preferably together with a specialist. It is clear that the ventilation engineer needs to be aware of the possible significance of toxicology for industrial ventilation construction.

The epidemiological data have the advantage of being based on human exposures. However, the results of epidemiological studies often remain inconclusive because of various confounding factors and poor exposure assessments. In addition, epidemiological data are available for only a small number of agents. The target level approach, presented in Chapter 6, Target Levels (and Design Methodology), uses inherently large safety margins in relation to OELs. Unfortunately, it is also applicable only for the most common exposures. Since zero exposure is the best, the as low as reasonably achievable principle, adopted in radiation protection, is, in principle, also a good approach for other exposures.[184,185] However, even then the question, how low is low enough, may remain unanswered. This chapter has been written with the intention of lowering the threshold for a ventilation engineer to seek a toxicological consultation and to provide the fundamental background information needed to utilize the available toxicological literature. Occupational hygienists may also find the text to be a useful compact overview of the essential concepts of toxicology.

5.3.1.2 *Epidemiology*

Epidemiological studies usually consist of the knowledge obtained from human exposures supplementing data derived from experimental studies. Epidemiological data often provide the ultimate proof of the deleterious effects of a chemical compound on humans, and form an important component of the assessment of the risks of some chemical compounds. In the future, the role of epidemiological data should be confirmatory rather than decisive in the risk assessment of existing and, especially, of new chemicals, since toxicology is becoming more and more a preventive rather than an observational science in protecting the health of workers exposed to chemicals and mixtures of chemicals in occupational environments.

The purpose of epidemiological studies is to try to identify whether there are causal relationships between the occurrence of diseases or other biological effects and exposures to various agents. There are

three main types of epidemiological studies: cross-sectional, cohort, and case–control studies. The working population is, on the average, healthier than the general population. Due to this "healthy worker effect," comparisons should be made with another worker group instead of the general population. The reason for the healthy worker effect is the fact that it is difficult for sick or disabled people to stay in employment due to the limitation caused by their diseases. Poor health may also prevent a person from getting a job in the first place.

Cross-sectional studies

In a cross-sectional study, exposure and effect are studied simultaneously. This approach contains an inherent problem because exposure must precede the effect. However, it can be used to investigate acute effects and also mild chronic effects (which do not force people to leave their jobs) if exposure has remained rather stable for a long time. When the prevalence of the effects studied are compared with the prevalence in other worker groups (controls or references) which correspond otherwise with the study group but are not exposed to the agent investigated, indicative evidence of possible causality may be obtained. For example, cross-sectional studies have been applied successfully to reveal the associations between mild neurotoxic effects and exposure to organic solvents.[186]

Cohort Studies

In a cohort or follow-up study, a group of workers exposed to the same agent is followed for a certain period, which can be either retrospective (starts at some time in the past and continues to the present) or prospective (starts in the present and continues for a certain time into the future). A cohort of controls should be formed with the same selection criteria as used for the study groups, except that they lack the exposure. Thus exposure to one agent only can be studied, whereas several health outcomes can be included. A cohort study is the only possible study method when the exposure studied is rare. The results of the cohort study are expressed as relative risks (risk ratios, RR) for various diseases (see Table 5.8 for results of different types of epidemiological studies on cancers in printing workers and epidemiological terms),[187]

$$\text{Relative Risk RR} = \frac{(\text{exposed with disease})/(\text{all exposed})}{(\text{controls with disease})/(\text{all controls})}$$

$$\text{Odds Ratio OR} = \frac{(\text{exposed with disease})/(\text{exposed healthy})}{(\text{nonexposed with disease})/(\text{nonexposed healthy})}$$

The benefit of a prospective cohort study is the possibility for accurate exposure assessment. However, these are not common, because many occupational diseases (including cancers) require long exposure times to develop. It is not practical or ethical to wait for decades before one obtains the result.

The problems often encountered in retrospective cohort studies include poor exposure data and incomplete follow-up of all individuals. The accuracy of health outcome data may also be low.

Case–control studies

In case–control studies, only one disease can be investigated. The cases include all patients with a certain disease observed in a hospital, city, or a larger area in a given period of time. Their exposure histories are compared with those of the controls. Thus several exposures can be investigated. The exposure data are not very accurate because they are obtained by interview. Especially in cases of serious diseases, patients are often desperate to seek some reason for their disease. Therefore patients of some other disease are usually employed as controls to avoid this information bias. The selection of controls is a crucial but extremely difficult task. Since factors such as age, sex, smoking, living habits, and place of abode are known to be risk factors for several diseases, the effects of these confounding factors are eliminated by matching. However, overmatching should also be avoided. Odds ratio (see Tables 5.9 and 5.10) is used to express how often the cases have been exposed to various exposures compared to controls. Case–control studies are common because they are inexpensive and relatively easy to perform. If the disease studied is rare, this approach is also the only practical alternative.[187]

5.3.1.3 Classifications of toxicology

The word toxicology originates from the Greek word toxicon, which means arrow poison. In ancient times, arrows were dipped into plant poisons to increase their lethality in hunting. Today, toxicology refers to that scientific discipline that explores the deleterious effects of chemicals or of physical or biological factors on living organisms. Toxicology also explores the mechanisms whereby chemicals or physical or biological factors induce their harmful effects in the organism.

There are several definitions and classifications of toxicology. One classification is based on the target organs which are harmfully affected by chemicals. Hence, there are terms such as neurotoxicology, liver or hepatic toxicology, kidney or renal toxicology, and toxicology of the eye (ocular toxicology). Inhalational toxicology emphasizes the importance of the lungs as the target organ of chemicals. In addition to these descriptive classifications, toxicology can be divided into mechanistic toxicology, conducted mainly in university and governmental research institutions, and

TABLE 5.8 Record-linkage studies among workers in the printing industry.[187]

Reference, country	Study subjects	Period of follow-up	Occupation/exposure	Cancer site/cause of death	No. obs.	RR	95% CI	Comments
Malker and Gemne,[188] Sweden	24,652 men and 6450 women registered at 1960 census as printing workers	1961–73	Printing workers (M)	Lung	190	1.5	1.3–1.8	Morbidity
			Blue–collar workers (M) in printing enterprises (newspaper, journal/book printing, and others)	Lung	149	1.6	1.4–1.9	
			Birth cohort around 1990 (M)	Lung	45	1.9	1.4–2.5	
				Urinary bladder	76	1.3	NG	$P>.01$
				Kidney	48	1.1	NG	$P>.01$
				Skin melanoma	27	1.2	NG	$P>.01$
			Printing workers (F)	Lung	9	1.3	NG	$P>.01$
				Urinary bladder	5	0.8	NG	$P>.01$
				Kidney	7	1.1	NG	$P>.01$
				Skin melanoma	8	1.2	NG	$P>.01$
				Cervix/uterus	162	1.3	1.1–1.5	
McLaughlin et al.,[189] Sweden	Male printing workers at 1960 census; 91 melanomas	1961–79	Printing industry	Skin melanoma	91	1.4	1.1–1.7	Morbidity
			Newspaper printing industry		39	1.9	1.1–1.7	
			Newspaper publishing industry		7	3.1	1.2–6.4	
			Typographers in newspaper printing industry		19	2.0	1.2–3.1	
			Machine repairers in newspaper printing industry		2	14.5	1.6–52.3	
			Journalists/editors in newspaper printing industry		16	2.4	1.4–3.9	
			Business/executives in newspaper printing industry		5	9.1	2.9–21.2	
Aronson and Howe,[190] Canada	242,196 women identified through employment survey	1965–79	Printing and publishing industry	Breast	11	2.2	1.1–3.9	Mortality; other sites not significantly elevated Mortality
Costa et al.,[191] Italy	1981 population census of Turin, Italy, residents; 10,798 deaths among persons employed	1981–89	Printing and publishing industry (M)	Pleura	2	6.0	0.7–22	
				Colon	7	2.1	0.9–4.4	
				Lung	22	1.1	0.7–1.7	
				Urinary bladder	2	1.0	0.1–3.6	
				Hematopoietic	7	1.6	0.6–3.3	
			Printing and publishing industry (F)	Lung	3	2.6	0.5–7.6	
				Colon	2	2.7	0.3–9.7	

(*Continued*)

TABLE 5.8 (Continued)

Reference, country	Study subjects	Period of follow-up	Occupation/exposure	Cancer site/cause of death	No. obs.	RR	95% CI	Comments
				Ovarian	3	3.2	0.6–9.3	
				Hematopoietic	2	2.0	0.2–7.2	
			Printers (M)	Liver	3	1.7	0.3–5.0	
				Colon	3	1.8	0.4–5.3	
				Multiple myeloma	2	[9.7]	1.1–33.1	
				Lung	12	1.2	0.6–2.1	
				Urinary bladder	0	–	–	
Costa et al.,[191] Italy	1981 population census of Italian residents; 15,734 deaths among persons employed	1981–82	Printing and publishing industry (M)	Kidney	3	4.8	1.7–13.4	Mortality
				Lung	11	1.1	0.8–1.4	
				Urinary bladder	2	2.9	0.8–11.1	
Pukkala,[192] Finland	1970 population census 47,178 men, 46,853 women	1971–85	Printing occupations (M)	Colon	16	2.2	1.2–35.	Morbidity
				Lung	50	0.8	0.6–1.1	
			Printing occupations (F)	Colon	10	1.4	0.7–2.5	
				Lung	12	1.8	0.9–3.2	
			Printers (F)	Breast	74	1.4	1.1–1.8	
				Ovarian	30	2.2	1.5–3.1	
				Skin melanoma	5	1.1	0.3–2.5	
			Printers (M)	Skin melanoma	7	1.1	0.5–2.4	
				Urinary bladder	9	1.1	0.5–2.0	
				Leukemia	2	0.4	0.1–1.4	
				Lung	19	1.1	0.7–1.7	
			Lithographers	Skin basal-cell carcinoma	4	4.4	1.2–11.2	

RR estimated by SMR (for mortality) or SIR (for morbidity). *F*, Female; *M*, male; *NG*, not given; *RR*, relative risk.
Modified from International Agency for Research on Cancer. Printing processes and printing inks, carbon black and some nitro compounds. In: IARC monographs on the evaluation of carcinogenic risks to humans, *vol. 65. Lyon, France: International Agency for Research on Cancer; 1996. p. 67–70.*

descriptive or regulatory toxicology, which is required for classification and labeling and risk assessment of chemicals for registration or marketing authorization purposes. Descriptive toxicology is important in characterizing the properties of a chemical compound, but improving the accuracy and quality of risk assessment is increasingly dependent on mechanistic information.

Toxicology can also be divided into different classes based on the goals it serves. Clinical toxicology explores ways of treating intoxicated patients, and also aims to develop quick methods to diagnose poisonings. Forensic toxicology is the science involved in use of toxicology for them purposes of law, such as detecting the role of poisons in fatalities. Environmental

TABLE 5.9 Case–control studies of urinary bladder cancer among workers in the printing industry.[187]

Reference, country	Type of controls	Exposure	Sex	No. of exposed cases/ controls	Odds ratio	95% CI	Comments
Najem et al.,[193] United States	Hospital-based, tobacco-related heart diseases and neoplasms excluded	Printing industry (≥1 year)	M + F	7/5	2.7	0.8–9.6	Crude odds ratio
Cartwright,[194] United Kingdom	Hospital-based, nonmalignant diseases	Printers (exposed to ink-fly from high-speed presses)	M + F	18/NG	3.1	1.4–6.8	Adjusted for type of case (incident or prevalent) and sex
Silverman et al.,[195] United States	Population based	Printing industry (ever)	M	50/45	1.1	0.7–1.7	Crude odds ratio
		Printers (ever)	M	6/2	3.0	0.6–14.8	
Schoenberg et al.,[196] United States	Population based	Printing workers (ever)	M	20/38	0.9	0.5–1.5	Adjusted for age, smoking, and other employments
		Printing ink (self-reporting)	M	42/53	1.6	1.0–2.5	
Baxter and McDowall,[197] United Kingdom	Other cancers	Printers (stated on death certificates)	M	21/NG	1.5	*P*.05	Against other cancers; matched on residence, year of death, and age
	All causes of death				1.2	*P*.05	
Brownson et al.[198]	Population-based, other nonsmoking related	Printing machine operators (longest held job)	M	7/8	3.1	1.1–8.9	All controlsProstate cancer excluded
			M	7/6	2.3	0.8–7.2	
Silverman et al.,[199,200] United States	Population based	Printer (ever)	M (white)	37/77	0.8	0.5–1.2	Adjusted for smoking; frequency matching for age and geographic
			F	1/10	0.2	<0.1–1.4	
Kunze et al.,[201] France	Hospital-based, nonneoplastic diseases of the lower urinary tract	Printing industry (ever)	M	11/3	5.0	1.3–19.6	Crude odds ratio
		Printing worker (ever)	M	7/3	3.0	0.7–13.8	
Cordier et al.,[202] France	Hospital-based, neoplastic, respiratory, and urological conditions excluded	Printing and publishing industry (ever)	M	26/28	0.9	0.5–1.5	Adjusted for age, hospital residence, and smoking
		Printers	M	14/9	1.5	0.6–3.5	
Siemiatycki et al.,[203] Canada	Population and hospital-based, other cancers, excluding lung and kidney sites	Printing and publishing industry <10 years	M	2/NG	0.3	0.1–1.2	Adjusted for age, family income, smoking, coffee consumption, ethnicity, and respondent status
				11/NG	1.9	0.9–3.9	
		≥10 years photographic products (substantial exposure)		4/NG	3.0	0.9–10.1	

F, Female; *M*, male; *NG*, not given; *RR*, relative risk.

toxicology assesses the importance of environmental pollution and the effects of exposure to environmental risk factors on human health. Ecotoxicology is interested in adverse effects of chemicals on all organisms especially at the population and ecosystem level. Industrial or occupational toxicology aims to study the effects of chemicals on workers exposed in an occupational environment (see Table 5.11).

Toxicology often provides the basis for a number of regulations aimed at protecting workers from potentially harmful effects. Toxicology has primarily a preventive function in providing information on safe use

TABLE 5.10 Case–control studies of lung cancer among workers in the printing industry.

Reference, country	Type of Controls	Exposure	Sex	No. of exposed cases/controls	Odds ratio	95% CI	Comments
Coggon et al.,[204] United Kingdom	Deaths from other causes	Printing inks	M	28/36	1.6	0.9–2.7	Job exposure matrix applied to occupations recorded on death certificates; age <40 years, cases and controls
		Printing inks (high exposure)	M	9/9	2.0	0.8–5.0	
Schoenberg et al.,[205] United States	Population-based	Printing workers ≥ 10 years	M	20/11	2.5	1.0–6.1	Adjusted for smoking (p. 20.05, crude)
			M	7/1	8.4	NG	
		Printing industry	M	37/31	1.3	0.8–2.3	Adjusted for smoking
Benhamou et al.[206] France	Hospital-based, nontobacco-related diseases	Printers and related workers	M	32/51	1.2	0.7–1.9	Matched for sex, age at diagnosis, hospital, interviewer; adjusted for smoking
Hoar Zahm et al.,[207] United States	Selected cancer sites	Printing occupations	M	21/41	1.1	0.6–1.9	Adjusted for age, smoking
				7/[4] (adenocarcinoma)	1.8	0.7–4.2	Occupations unknown for about half of cases and controls
Siemiatycki,[208] Canada	Hospital-based, other cancers	Printing and publishing industry	M	35/NG	2.0	1.2–3.5	Smoking-adjusted
		Printers	M	26/NG	2.1	1.1–4.1	Smoking-adjusted
		Printers (>10 years)	M	13/NG	1.7	0.7–4.1	Smoking-adjusted
		Printing process workers	M	15/NG	3.1	1.1–8.7	Smoking-adjusted
			M	6/NG (adenocarcinoma)	7.0	1.8–27.9	Smoking-adjusted
		Inks (any)	M		1.6	1.0–2.7	Smoking-adjusted
		Inks (substantial)	M	37/NG	1.5	0.7–3.1	Smoking-adjusted

TABLE 5.11 Classifications of toxicology.

Area of toxicology	Scope
Mechanistic	Understanding of cellular and molecular mechanisms
Regulatory	Drafting regulations and legislation
Organ specific	Defining organ-specific effects and defining chemically induced critical effects
Forensic	Diagnosis and fatalities
Occupational	Delineating occupational hazards and risks and prevention
Environmental	Identification of chemical hazards in the environment, and their effects on humans and wildlife species
Clinical	Diagnosis and treatment of poisoning

of chemicals. It is difficult to imagine occupational or other safety regulations without a major input from toxicology.

The main role of toxicology in the industrial setting originates from its ability to identify harmful chemicals and other hazards in advance. After toxicological research has identified exposure–effects relationships for different chemicals, OELs for various industrial chemicals can be established. Subsequently, workers can be protected against excessive exposures by measuring the exposure and ensuring that the OELs are not violated; ventilation engineers and occupational hygienists are the key persons in this field. Careful planning and design ensure that most workers are protected, nevertheless the most sensitive individuals may still react to exposure levels that are below the acceptable exposure limits. These relationships also indicate the close relationship between industrial

toxicology and industrial hygiene. Without a broad knowledge of the toxicological characteristics of chemicals, industrial hygiene is irrelevant. On the other hand, without industrial hygiene, toxicology would be helpless in protecting the workers against chemical hazards.

5.3.1.4 Industrial toxicology, hygiene, and occupational medicine

Industrial toxicology, industrial hygiene, and occupational medicine all have a common goal: to protect workers from occupational hazards in the workplace. The goal of industrial toxicological research is to characterize the biological effects of chemicals, whereas the goal of occupational hygiene is to protect workers by improving working environment to minimize exposure. The goal of occupational medicine, in turn, is to protect workers' health by identifying early signs of harmful effects, and to diagnose and treat occupation-related diseases. In many cases, reduction of exposure will suffice to prevent many occupation-related diseases after the first symptoms. In some cases the exposure may need to be stopped completely, if severe health outcomes are clearly demonstrated to occupational exposure to the chemical. Therefore occupational medicine relies on toxicological and occupational hygienic knowledge in solving occupational health problems. However, the scope of occupational medicine is much wider than simply examining chemical-induced toxicity, as it covers a wide area of interests such as occupational ergonomics and psychophysiological factors in the occupational setting.[209]

Poisoning incidents in the workplace

The hazards of chemicals are commonly detected in the workplace first, because exposure levels there are higher than in the general environment. In addition, the exposed population is well known, which allows early detection of the association between deleterious health effects and the exposure. The toxic effects of some chemicals, such as mercury compounds and soot, have been known already for centuries. Already at the end of the 18th century, small boys who were employed to climb up the inside of chimneys to clean them suffered from a cancer of the scrotum due to exposure to soot. This was the first occupational cancer ever identified. In the viscose industry, exposure to carbon disulfide was already known to cause psychoses among exposed workers during the 19th century. As late as the 1970s, vinyl chloride was found to induce angiosarcoma of the liver, a tumor that was practically unknown in other instances.

Even in the Nordic countries, exposure to carbon disulfide still caused severe central nervous effects among exposed workers during the late 1960s and early 1970s, and exposure to lead caused several lead poisonings at the same time. Exposure to asbestos remained a major health hazard until the 1970s. The use of asbestos is nowadays strictly controlled and it has been banned in many countries. Nevertheless, it continues to be an important occupational health problem because of the long latency period of asbestos for causing lung cancer and mesothelioma, a time period of 20–40 years. In addition, there are large amounts of asbestos remaining in buildings, and renovation of old buildings will pose a health risk to workers for a long time to come.[210]

Many very hazardous solvents, such as benzene and carbon tetrachloride, were widely used until the 1970s. The situation was very similar for the use of pesticides. Among the toxic pesticides that were still in wide use 20 years ago were chlorophenols, DDT, lindane, and arsenic salts, all of which are classified as human carcinogens as well as being acutely toxic.[180,211] Fortunately, use of these kinds of very toxic chemicals is now limited in the industrialized world. However, because the number of chemicals used in various industries continues to increase, the risks of long-term health hazards due to long-term exposure to low concentrations of chemicals continues to be a problem in the workplace.

5.3.1.5 Concept of risks

The term risk has wide implications. It is used to characterize difficulties in predicting changes in the currency markets and to indicate the probability of potential financial losses due to such changes. A surgeon prior to a major operation also needs to evaluate the risks to the patient, not only due to the disease but also risks associated with the operation itself and the anesthesia. Car drivers seldom consider the risk of a traffic accident when starting a car even though the risk of a fatal car accident is many times greater than the calculated risks associated with exposure to chemicals. Another example, widely discussed in the media, is the comparison of risks from energy production by fossil fuels and nuclear energy. This comparison has proven to be extremely difficult due to a number of philosophical aspects. We can calculate with some degree of certainty the risks involved in the production of energy with fossil fuels. There are major risks in mining or oil drilling, during the transportation of the fuel, and due to the extensive emissions emanating from the combustion of the fuel. In Europe, annual loss of life due to energy production utilizing fossil fuels, and due to traffic exhaust, is close to 300,000. The verifiable health hazards due to nuclear energy are only a small fraction of the losses due to the use of fossil fuels. However, the potential risk due to a nuclear accident raises alarm in individuals. Even if

the accident probability is small, the losses due to even one incident may be catastrophic. This is well illustrated by the accident in Chernobyl in 1986.[212]

With regard to chemicals risk is the likelihood of an adverse effect due to exposure. It is the product of hazardous property of the chemical (including potency) and exposure. Therefore risks of hazardous chemicals can be limited by limiting exposure, for example, by the use of personal protection equipment. On the other hand, moderate exposure to a chemical that has hazardous properties only at very high doses does not constitute a risk.

Safety evaluation of chemicals prior to their release to the market is based on analysis of physical and chemical properties as well as characterization of toxic effect using experimental models, typically a combination of in vitro methods and animal experiments. When the results of animal experiments or in vitro studies are applied to humans, several assumptions have to be made, including (1) that animals or in vitro systems are a good model to predict toxic effects in humans and (2) that high doses or concentrations of chemicals used in studies cause similar effects to what would be seen in humans though at a lower frequency or with a milder change in functions of target organs. Some toxic effects of chemicals may, however, different in rodents from those in humans. For example, guinea pigs tolerate the effects of strychnine rather well, in contrast to humans. For organ toxicity (neurotoxicity, liver toxicity, and kidney toxicity) end points, safety factors can be used for assessing[213] safe levels for humans (see below). Dose–responses are regularly used to delineate the potency and toxicological characteristics of chemicals, and to make comparisons among species.[177] In most cases animal studies are used to define the no-observable-adverse-effect level (NOAEL), that is, the lowest dose that does not cause an adverse effect in animals. This dose is then divided by uncertainty (or safety) factors, for example, the commonly used "default" factor of 100 (consisting of 10 for interspecies differences in sensitivity between rodents and humans and 10 for intraspecies variability among humans Fig. 5.33) to estimate the dose (mg/kg/day) which is considered safe for humans. This approach assumes that there is a safe dose below which a chemical does not cause harmful effects on humans. The assumption of a safe threshold dose is used for most end points of deterministic toxicology, that is, organ toxicology. However, whether in fact there can be any safe dose for carcinogens, especially for genotoxic carcinogens, has been challenged, and the linear extrapolation models that have been widely used in carcinogenic risk assessment do not utilize safety factors. However, this approach has also been challenged, because throughout biology it seems to be impossible to find effects without any threshold, and because it neglects biological defense mechanisms present within cells.

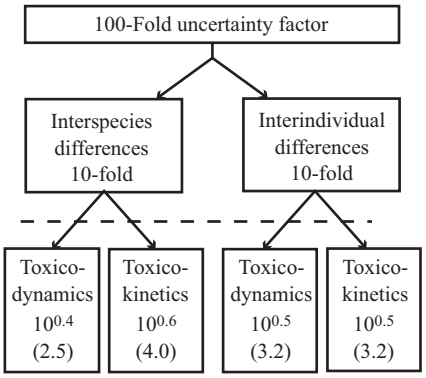


FIGURE 5.33 Subdivision of the 100-fold "default" uncertainty factor showing the relationship between the use of uncertainty factors (above the dashed line) and proposed subdivisions based on toxicokinetics and toxicodynamics. Actual data should be used to replace the default values if available.[214]

Usually risk assessment procedure, discussed in more detail later [see Chapter 6: Target Levels (and Design Methodology)], is divided into four different stages or steps (Fig. 5.34):

1. hazard identification based on in vitro or animal experiments, epidemiological studies or structure–activity analyses;
2. hazard characterization, or dose–response assessment, by using mainly data from animal experiments to reveal target organs and toxic doses, and the shape of the dose–response curve;
3. exposure assessment to reveal the extent, frequency and duration of exposure of different groups of people, and to identify groups with special characteristics of exposure; and
4. risk characterization, a synthesis of the preceding three steps, which aims to assess both qualitatively and quantitatively the risks induced by a chemical at a given or at different exposure levels.

Based on the results of risk assessment, decision makers have to attempt to manage risks, for example, by determining various exposure limits to protect individuals against deleterious effects of chemicals. This kind of procedure is commonly used for determining acceptable daily intake values for contaminants in foods and acceptable operator exposure level for pesticides. Even though the results obtained in experimental studies are part of the basic data on which the OELs have been based, the levels result from consideration of many other aspects, especially epidemiological data. In addition, these decisions take into consideration economic and political consequences of the

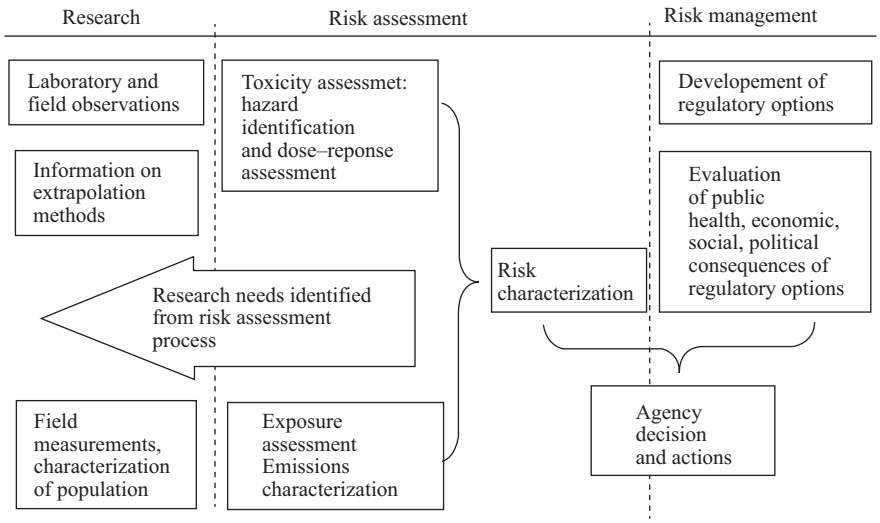


FIGURE 5.34 Elements of risk assessment and risk management.

decisions, as well as perception of various risks by the general public. Furthermore properties such as strong odor or irritation influence the levels of OELs. It needs to be kept in mind that even though risk assessment of exposures to single chemicals is still far from complete, much greater difficulties are encountered in assessing the risks of multiple exposures.

5.3.2 Exposure to chemical substances

Among the 10 million known chemical compounds, there are some 50,000 which are in common use. Workers are usually exposed to several agents simultaneously (their interactions are considered in Section 5.3.4.2). In addition, many impurities in workplace air are inherently complex mixtures, which may consist of hundreds of different compounds. Mineral oils and wood and bakery dusts are examples of common complex mixtures.

5.3.2.1 Characterization of exposures

Indoor and outdoor exposure to pollutants

Occupational and environmental exposure to chemicals can take place both indoors and outdoors. Occupational exposure is caused by the chemicals that are used and produced indoors in industrial plants, whereas nonoccupational (and occupational nonindustrial) indoor exposure is mainly caused by products. Toluene in printing plants and styrene in the reinforced plastic industry are typical examples of the two types of industrial occupational exposures. Products containing styrene polymers may release the styrene monomer into indoor air in the nonindustrial environment for a long time. Formaldehyde is another typical indoor pollutant. The source of formaldehyde is the resins used in the production process. During accidents, occupational and environmental exposures may occur simultaneously. Years ago, dioxin was formed as a by-product of production of phenoxy acid herbicides. An explosion in a factory in Seveso, Italy, caused wide-spread pollution of the industrial site as well as its surroundings. Serious effects of dioxin were detected both in domestic animals, such as cows and sheep, and in humans, the most serious early effects being a serious skin disease, chloracne, and alterations in the function of the immune system. Follow-up studies have demonstrated that this accident also increased the cancer risk in exposed individuals.[215]

Outdoor inhalation exposure is mainly due to traffic, energy production, heating, and natural factors such as pollen, mineral dusts, forest wires and volcanic eruptions. These outdoor sources of pollution also affect indoor air quality. The indoor concentration is typically 20%–70% of the corresponding outdoor concentration. Occasionally the indoor concentrations of an external pollutant (especially radon) may even exceed the concentrations outdoors.[212]

In densely populated areas, traffic is responsible for massive exhausts of nitrous oxides, soot, polyaromatic hydrocarbons, and CO. Traffic emissions also markedly contribute to the formation of ozone in the lower parts of the atmosphere. In large cities, fine particle exposure causes excess mortality which varies between one and five percent in the general population.[212] Contamination of the ground water reservoirs with organic solvents has caused concern in many countries due to the persistent nature of the pollution.

A total exposure assessment that takes into consideration all exposures via all routes is the first step of a toxicological risk assessment.[216,217]

Characteristics of industrial processes

The in-plant emissions can be divided into process and manual emissions. In the process industry, the emission sources can usually be enclosed and the workers do not need to stay for long periods close to the emissions. The emissions are minor, and even if they do occur, they are generally released far from the areas where workers have their accommodation. Often, workers can spend most of their time in clean control rooms. In certain process industries, such as the petrochemical industry, the processes are located largely outdoors, and the emissions are mainly fugitive emissions from leaking seals of flanges, valves, and pump shafts. Manual emissions occur in the immediate vicinity of the worker due to the task he or she is performing. Typical examples include welding, painting, gluing, sawing, grinding, haircutting and baking. It is natural that the exposure control is much easier for process emissions than for manual ones. Even very toxic substances can be used safely in the process industry, whereas even moderately noxious chemicals may cause major problems in manual tasks. Thus avoidance of those manual tasks with chemicals known to cause adverse health effects is important. If the automation of these tasks becomes very expensive, it may be possible to use subcontractors who specialize in this kind of work and have adequate control arrangements in their production facilities.

The best way to control exposure is to replace dangerous agents with safer ones. Today, highly toxic solvents, such as benzene, bromobenzene, carbon tetrachloride, and chloroform, are no longer extensively used. Benzene remains, however, an important chemical in the petrochemical industry, but the processes where it is used are closed.

The use of other highly toxic substances, such as lead and carbon disulfide, which have in the past caused many occupational diseases, is also rare in manual tasks nowadays. Thus relatively few possibilities for substitution are left in individual workplaces. One rather common exception does exist; very fine powders can often be replaced with granular or liquid products. All possibilities to replace solvent-based products with water-based alternatives have not yet been utilized. However, one must be aware of the possible novel risks involved with the use of the new products; for example, when acid-cured furniture paints and lacquers, which released formaldehyde, have been replaced with acrylic resins, skin sensitization has become more common among furniture painters.

Since process disturbances do take place, and accidental releases are possible, even from processes closed under normal conditions, the plants where highly toxic or sensitizing substances are in use or may be generated should be provided with continuous monitoring and alarm systems in the critical areas. An example is strong odor compounds added in liquefied petroleum gases.

5.3.2.2 Exposure routes

The exposure routes include the lungs, that is, inhalational exposure, the skin, that is, dermal exposure, and the mouth, that is, oral exposure.[216,218] Inhalation is usually considered to be the most important route for occupational exposure. Some chemicals are also absorbed via the skin or damage (irritation or sensitization) to the skin, and thereby amplify their own absorptions. Poor personal hygiene may result in oral exposure from eating or smoking with dirty hands. Toxic effects also often depend on the exposure route, because more extensive first pass metabolism is taking place for substances via oral than via inhalation or dermal exposure. The effects of irritating agents occur at the contact site. On the other hand, many compounds are distributed widely in the body and the target organ may be situated far from the entry site. Compounds may become concentrated in certain organs. The organ with the highest concentration is, however, not necessarily the target organ; for example, lead is accumulated in the bones but its most severe effects appear in the central nervous system (CNS). Many lipophilic carcinogens are accumulated in the adipose tissue but the cancer does not usually develop there but rather in the target organs, such as the liver, the kidneys, or the lungs.[219–221]

Inhalational exposure

Gases, vapors, mists, and dusts are mainly absorbed into the body through the lungs. Lipid-soluble vapors, especially those of solvents, and gases reach the alveolar space without any difficulty from where they pass through the respiratory tract, and diffuse readily across alveolar lining to reach the systemic circulation. Passive diffusion is based on a concentration gradient between alveolar air and the blood. The rapidity of the saturation of the blood with gaseous compounds largely depends on the blood and lipid solubility of the gas. Highly blood- and lipid-soluble compounds reach saturation slowly, whereas vapors and gases with low blood and lipid solubility rapidly become saturated in the blood.[222] Also water solubility and reactivity greatly affect penetration through the lung. Very water soluble and reactive compounds tend to dissolve in the mucus in the upper respiratory airways, or react with proteins in the mucus, and only a small portion

of the dose of such compounds ever reaches the alveolar region of the lungs. Examples include sulfur dioxide and formaldehyde. Sulfur dioxide reacts readily with water and forms sulfurous acid. Formaldehyde reacts with proteins in the mucus and in the cells of the epithelium of the upper respiratory tract. Therefore they both reach the alveolar region only at high concentrations. As a consequence of reactivity, high concentrations of formaldehyde cause serious lung injury and lung edema upon reaching the alveolar region.[223,224]

Aerosols reach the alveolar space depending on their particle size and physicochemical characteristics. Small particles that reach the alveolar region (see Sections 2.3.7 and 3.1.1) may reach the circulation through the lymphatic drainage of the alveolar region.

Dermal exposure

Skin is also important as an occupational exposure route. Lipid-soluble solvents often penetrate the skin, especially as a liquid. Not only solvents, but also many pesticides are, in fact, preferentially absorbed into the body through the skin. The ease of penetration depends on the molecular size of the compound, and the characteristics of the skin, in addition to the lipid solubility and polarity of the compounds. Absorption of chemicals is especially effective in such areas of the skin as the face and scrotum. Even though solid materials do not usually readily penetrate the skin, there are exceptions (e.g., benzo(*a*)pyrene and chlorophenols) to this rule.[210,222,225]

Oral exposure

In the occupational setting, oral exposure is of minor significance, being mainly due to poor personal hygiene. In addition, gases that dissolve or are otherwise trapped in the upper respiratory tract, usually are swallowed and enter the gastrointestinal tract. Particles that are removed as such or are captured by macrophages by the mucociliary escalator from the respiratory tract are also ultimately swallowed and enter the gastrointestinal tract.

5.3.2.3 Physicochemical determinants of exposure

Physicochemical characteristics greatly determine the entry of chemicals into the body, and also their behavior in the body (distribution, biotransformation, and excretion). Therefore the physicochemical characteristics of a compound affect its internal dose and its subsequent effects by determining how quickly and extensively a chemical reaches the target organs. In the following section, some of these important physical–chemical characteristics of chemicals will be discussed.

Water solubility

The site and the severity of the effect of respiratory irritant gases depend largely on their water solubility. Very water-soluble gases and vapors, such as ammonia, hydrogen chloride, sulfur dioxide, and formaldehyde dissolve in the mucus of the nose and upper airway and cause inflammation. Poorly water-soluble gases, such as nitrogen dioxide and ozone, are able to reach the deep lung area. Inflammation results from damage to cellular membranes of bronchiolar and alveolar cells and subsequent accumulation of liquid in the lungs (edema). Because the alveoli have no receptors for irritation, the effects are generally noticed only several hours after the exposure when the amount of liquid accumulating has become so large that it impairs gas exchange. In addition to water solubility, the reactivity of the gas with airway proteins is important. Thus water soluble and reactive sulfur dioxide is removed effectively by the nose while less slowly water reacting ethanol is partially absorbed. If a soluble gas is adsorbed on fine particles, it can be transported deep to the lungs.[223,224]

The solubility coefficient S is used as a measure of water solubility. It is the ratio between the concentrations in water and air phases at equilibrium. Ethanol, a very soluble gas, has a solubility coefficient of 1100 at 37°C while the coefficient for nitrous oxide, a poorly soluble gas, is 0.15.

The importance of pH and pK_a

Under physiological conditions, pK_a (negative common logarithm of the acid dissociation constant) of weak acids or bases largely determines its ionization at varying pH. This is important because the dissolution of polar molecules in lipid bilayers is a difficult and slow process, and from a practical toxicokinetic point of view, most polar compounds fail to penetrate biological membranes to any significant extent. Ionization of most weak acids and bases depends on their dissociation constant and pH according to the Henderson–Hasselbalch equation[226]:

$$\log\left[A^-/HA\right] = \mathrm{pH} - \mathrm{p}K_a \quad \text{(for weak acids)}$$

$$\log\left[B/BH^+\right] = \mathrm{pH} - \mathrm{p}K_a \quad \text{(for weak bases)}$$

The proportion of ionized and unionized forms of acidic and basic chemical compound can be readily calculated according to the above equation. pK_a is the log value of acid dissociation constant and indicates pH value at which 50% of the compound exists in ionized form. The ionization of weak acids increases as the pH increases, whereas the ionization of weak bases increases when the pH decreases. As the proportion of an ionized chemical increases, the diffusion of the

chemical through the biological membranes is greatly impaired, and slows down permeation. For example, the common drug acetosalicylic acid (aspirin), a weak acid, is readily absorbed from the stomach because most of its dose is in an unionized form at the acidic pH of the stomach.[226]

Lipid solubility

Cell membranes are composed of lipid bilayers to which contain large protein molecules such as receptors, transporters, and glycoproteins are incorporated. To be able to penetrate through the cell membrane, a compound has to dissolve in the lipid bilayer, where it diffuses according to the concentration gradient across both sides of the membrane, and after passing through the membrane, dissolve once more in the water phase within the cell. Lipid-soluble compounds can reach high concentrations in lipid-rich organs, such as the adipose tissue, brain, bone marrow, and spleen. Lipid solubility is often characterized by an octanol/water coefficient (Ko/w) which indicates the concentration ratio of the compound between these two phases. For example, xylene, a nonpolar lipid-soluble organic solvent, has an octanol/water coefficient of 3200 and many inorganic gases have low octanol/water coefficients.[226]

Blood solubility

Absorption of a gaseous compound from the lungs depends on its blood solubility. For most compounds, blood solubility is similar to water solubility. However, the blood solubility coefficient may become much higher than the water solubility coefficient if the blood proteins have a high affinity for the compound. CO, for which hemoglobin has a high affinity, is a good example (see Section 4.3.3). Blood solubility is decisive for the rapidity of the action of the compound, especially on the CNS, but also on other organs. Often lipid-soluble vapors such as diethyl ether or organic solvents such as xylenes also have a high blood solubility.[226]

The toxic effect depends both on lipid and blood solubility. This will be illustrated with an example of anesthetic gases. The solubility of dinitrous oxide (N_2O) in blood is very small; therefore, it very quickly saturates in the blood, and its effect on the CNS is quick, but because N_2O is not highly lipid soluble, it does not cause deep anesthesia. Halothane and diethyl ether, in contrast, are very lipid soluble, and their solubility in the blood is also high. Thus their saturation in the blood takes place slowly. For the same reason, the increase of tissue concentration is a slow process. On the other hand, the depression of the CNS may become deep, and may even cause death. During the elimination phase, the same processes occur in reverse order. N_2O is rapidly eliminated, whereas the elimination of halothane and diethyl ether is slow. In addition, only a small part of halothane and diethyl ether are eliminated via the lungs. Their main elimination takes place by metabolism, which stops the anesthesia. The metabolites are excreted through the kidneys into the urine.[226]

Partition coefficients

Other important determinants of the effects of compounds, especially solvents, are their partition coefficients, for example, blood-tissue partition coefficients, which determine the distribution of the compound in the body. The air-blood partition coefficient is also important for the absorption of a compound because it determines how quickly the compound can be absorbed from the air-space of the lungs into the circulation. An example of a compound that has a high air-blood partition coefficient is trichloroethane (low blood solubility), whereas most organic solvents (e.g., benzene analogues) have low air-blood partition coefficients (high blood solubility).[226]

Vapor pressure

Vapor pressure is important simply because a compound that is easily vaporized can also readily cause a marked exposure through the lungs. Organic solvents are good examples of volatile compounds, and known to cause marked exposure via the lungs, in addition to exposure via the skin.[227]

Particle size

The size of inhaled particles varies markedly. The size distribution approximates a log-normal distribution that can be described by the median or the geometric mean, and by the geometric standard deviation. For fibers, both fiber diameter and length are important determinants of their behavior in the airways. The effect of particle size on the fate of particles is discussed in more detail in Sections 3.1 and 5.2.[228]

5.3.2.4 Physiological determinants of exposure

Anthropologic features of humans, their physical activities, ventilation capacities, and the state of their circulation all affect exposure to chemical compounds. Some of the physiological determinants of exposure will be dealt with below. Exercise typically increases cardiac output, facilitates circulation, increases the minute volume of ventilation, is associated with vasodilation of the skin circulation, and increases perspiration and secretory activity of the sweat glands. All of these changes tend to facilitate the absorption of chemicals through multiple routes.

Inhalational exposure

During exercise, both minute ventilation and cardiac output increase dramatically. Although minute ventilation averages 7–10 L/min at rest for an average person of about 70 kg, it can increase to 160 L or more/min during intense exercise, and be 25–40 L/min with moderate exercise. This has a considerable direct effect on exposure through the lungs. For example, when young persons were exposed to *m*-xylene at a concentration of 100 ppm, the concentration of *m*-xylene in their venous blood reached a level of 19 μmol/L, whereas after a moderate exercise at 100 W, a concentration of 100 μmol/L was reached in their blood. Thus the exercise caused about a fivefold increase in the concentration of *m*-xylene in the blood compared to values in sedentary subjects even though the ambient air xylene concentration was the same.[229] The increase was approximately equivalent to the change in minute ventilation (which was four- to sixfold). Increased cardiac output and thereby increased circulation helped in maintaining the concentration gradient between the alveolar space and the blood and thereby facilitated pulmonary absorption of *m*-xylene.[230,231]

Dermal exposure

Exercise also increases skin circulation and perspiration, which both enhance dermal penetration of compounds into the body. Furthermore skin lesions, such as wounds and dermatitis, can increase the permeability of the skin to chemicals. Also exposure of the skin to solvents and removal of skin fat increase dermal penetration of a number of compounds. Compounds penetrate the skin more readily in places where the skin is thin, like the face, hands, and scrotum. Increased dermal blood flow due to exercise facilitates the penetration of the skin by chemicals.[229–231]

Considerable protection against dermal exposure can be achieved by using the appropriate protective clothing, such as overalls, rubber gloves, and boots. For example, protective clothing provided 80%–95% protection when workers manually handled ethylenebisdithiocarbamate fungicides in agriculture. [216,218,221] Not all gloves protect against lipid soluble compounds, therefore it is important to use right material, for example, nitrile gloves. It would seem that a similar protection against dermal exposure can be achieved in agriculture and industry in general. Fig. 5.35 shows that urinary excretion of ethylenethiourea mainly depends on dermal absorption of the parent compound, maneb (a dithiocarbamate) because a delay can be seen before the start of urinary elimination of ethylenethiourea (Fig. 5.35B). Fig. 5.35A shows that the urinary elimination of ethylenethiourea has several elimination phases due to the distribution of the compound in different body compartments.

5.3.3 Kinetics of chemical compounds

The kinetic properties of chemical compounds include their absorption from environment to the blood, distribution in the body, biotransformation to more soluble forms through metabolic processes in the liver and other metabolic organs, and the excretion of them and their metabolites in the urine, the bile, the exhaled air, and in the saliva (Fig. 5.38). In drug development absorption, distribution, metabolism, and excretion are known as ADME. Total elimination of compounds is the sum of metabolism and excretion. An important issue in toxicokinetics deals with the formation of reactive toxic intermediates during biotransformation reactions (see Section 5.3.3).

5.3.3.1 Absorption

As stated earlier, inhalation is the main route of absorption for occupational exposure to chemicals. Absorption of gaseous substances depends on solubility in blood and tissues (as presented in Sections 2.3.3–2.3.5), blood flow, and pulmonary ventilation. Particle size has an important influence on the absorption of aerosols (see Sections 2.3.7 and 3.1.1).

Absorption via the skin depends on the lipid and water solubility of the compound, its polarity, and the molecular size. Dermal absorption is also markedly affected by the size of the exposed skin area.[216,218,229,231]

Chemicals have to pass through either the skin or mucous membranes lining the respiratory airways and gastrointestinal tract to enter the circulation, which is called absorption. Then they are distributed and reach their site of action. Different mechanisms of entry into the body also greatly affect the absorption of a compound. Passive diffusion is the most important transfer mechanism. According to Fick's law,[221] diffusion velocity v depends on the diffusion constant (D), the surface area of the membrane (A), concentration difference across the membrane (Δc), and thickness of the membrane (L)

$$v = \frac{DA\Delta c}{L} \tag{5.50}$$

The diffusion constant depends on the lipid solubility, molecular weight, and structure of the compound. Lipid-soluble compounds with a molecular weight less than about 500 diffuse readily through the membranes. Polar compounds are poorly absorbed. However, active transport systems play a major role in the absorption of a number of amino acids, sugars, ions, and other nutrients and compounds resembling them. The blood–brain

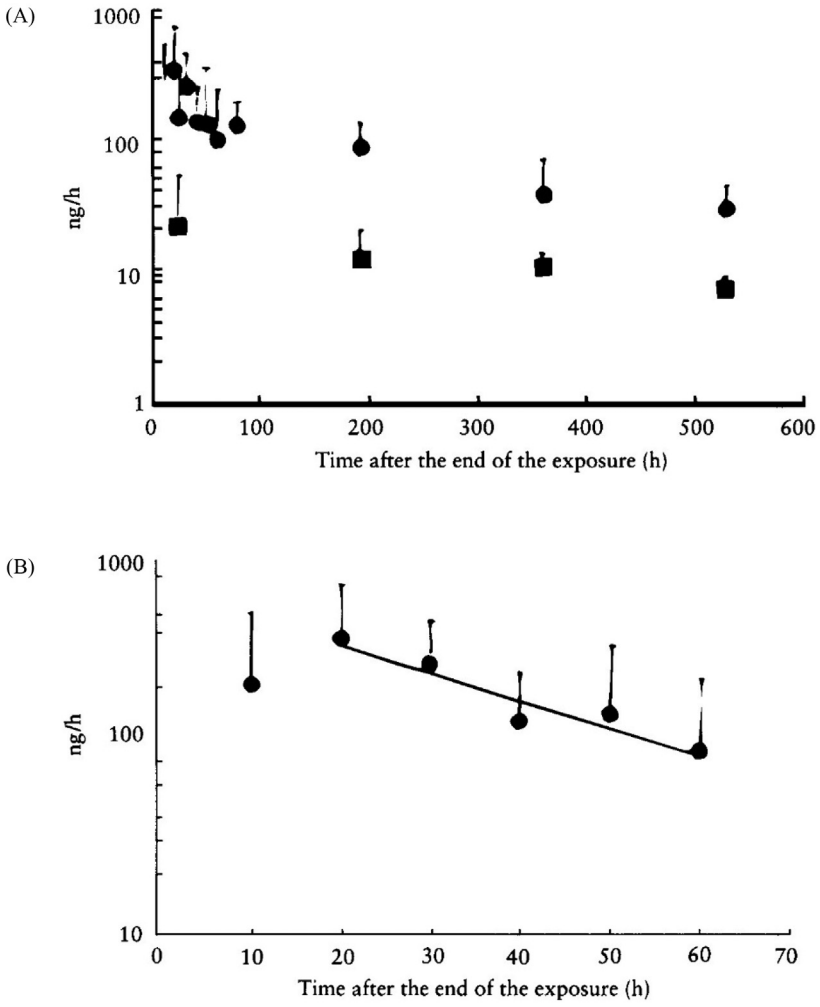


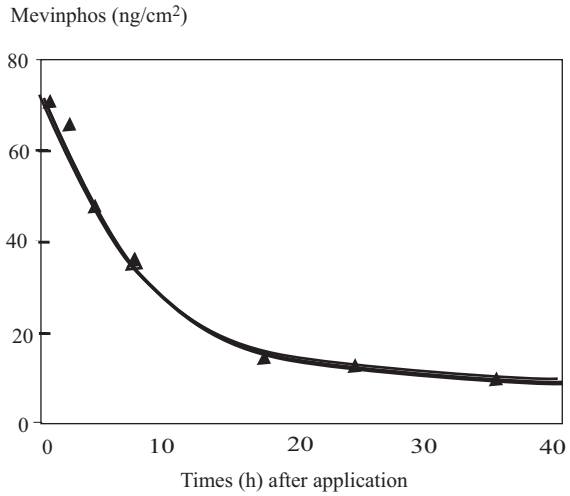
FIGURE 5.35 (A) Excretion rate (mean ± SD) of ETU in the urine (ng/h) of potato field applicators (circles) (groups I and II) and pine nursery weeders (squares) (group IV) after exposure to ethylene bisdithiocarbamates during pesticide application (groups I and II) and the weeding of the sprayed vegetation (group IV). The ETU concentrations were at the detection limit in group IV after 2 weeks of follow-up (two last time points). (B) Excretion rate of ETU (mean ± SD) in the urine (ng/h) of potato field applicators (group I) during 60 h after the cessation of exposure to ethylene bisdithiocarbamate fungicides. The first time point at 10 h after the cessation of the application was omitted from the analysis because of possible continuous exposure for a few hours after the application and because of the effect of dermal absorption. The regression equation is $y = -6x + 455$, where y is the excretion rate of ETU (ng/h), x is the time (h), and the correlation coefficient squared (r^2) is 0.86. *ETU*, Ethylenethiourea. Source*: With permission from Kurttio P, Savolainen K. Ethylenethiourea in air and in urine: implications to exposure to ethylenebisdithiocarbamate fungicides.* Scand J Work Environ Health *1990;**16**:203.*

barrier, a functional structure that protects the CNS against foreign substances, prevents the entry of most compounds to the CNS. In fact, only lipid-soluble compounds, and polar compounds which have an influx transporters, can readily enter the CNS. Examples of such polar compounds include amino acids and sugars. Influx transporter also plays an important role in the testicles, which are protected by a testicular–blood barrier which has a role similar to the blood–brain barrier.[221] Fig. 5.36A shows how mevinphos, a greenhouse organophosphorus insecticide which mainly gains access to the body via the skin, is absorbed. Fig. 5.36B shows that exposure through lungs was negligible, because there was an excellent correlation between mevinphos on the foliage, the source of the compound, and mevinphos level on the skin.[218]

Entry of particles into the body

The aerodynamic particle diameter determines the fate of particles in the respiratory system. Coarse particles are deposited in the nose and nasopharynx. Smaller particles that pass the upper airway can be deposited in the bronchial region and lower airway. A size-selective deposition model and sampling of particles has been standardized both in Europe[232] and internationally.[233] The Standard includes size definitions for three mass fractions. The inhalable fraction consists of particles that can enter the upper airway. Its upper size limit is 100 μm (the diameter of human hair is 50–100 μm). The thoracic mass fraction consists of particles that can penetrate past the larynx. Its upper size limit is about 30 μm and median cut point 10 μm. The respirable mass fraction consists of particles that can enter the alveolar region. Its upper size limit is about 10 μm and median cut point 4 μm (see Fig. 5.37).

The particle size is the most important factor that contributes to the clearance of particles. For particles deposited in the anterior parts of the nose, wiping and blowing are important mechanisms, whereas particles on the other areas of the nose are removed with



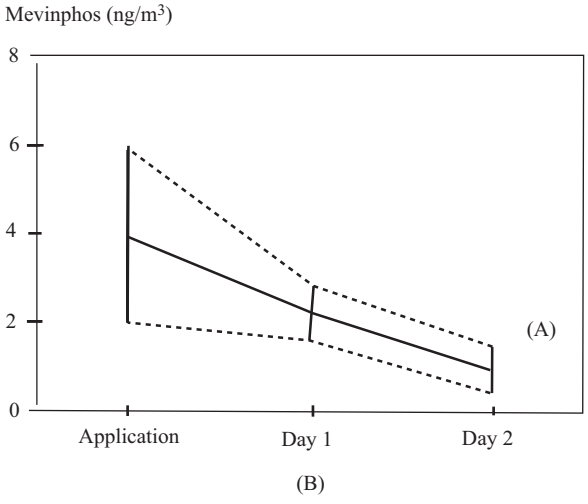


FIGURE 5.36 (A) Correlations ($y = 7.2x \lesseqgtr 3.5$; $r = 0.97$) between the amount of mevinphos on the foliage and the dermal exposure rate to mevinphos via the hands.[218] (B) Mean ($\pm$SD) concentrations of mevinphos in the breathing zone of the workers immediately after application and on the morning of the two first working days after the application.[218]

mucus. The cilia move the mucus toward the glottis where the mucus and the particles are swallowed. In the tracheobronchial area, the mucus covering the tracheobronchial tree is moved upward by the cilia beating under the mucus. This mucociliary escalator transports deposited particles and particle-filled macrophages to the pharynx, where they are also swallowed. Mucociliary clearance is rapid in healthy adults and is complete within 1–2 days for particles in the lower airways. Infection and inflammation due to irritation or allergic reaction can markedly impair this form of clearance.

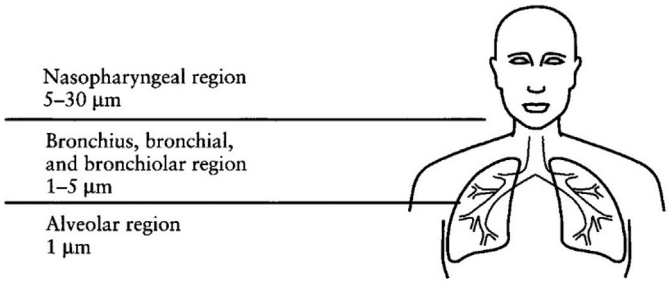


FIGURE 5.37 Regions of pulmonary pathways and size of particles that can reach different regions of the lungs.[180]

Particles deposited in the alveoli are phagocytized by alveolar macrophages and cleared either through the mucociliary escalator or through the lymphatic drainage system. Fibers may be too long to become phagocytized by single macrophages. In such a case, several macrophages can participate in the phagocytosis in a cooperative manner (see Fig. 5.37). Macrophages are able to dissolve synthetic miner fibers to some extent, but asbestos (especially amfiboles) fibers remain mostly unaffected. This leads to the production of oxygen radicals and inflammation mediators, which induce macrophages to kill themselves. Another macrophage will then phagocytize the asbestos fiber and it too will die. This vicious cycle will continue and it may ultimately lead to lung fibrosis and cancer. Small particles may also directly penetrate the epithelial membrane and enter the circulation.

5.3.3.2 *Distribution*

After absorption, a chemical compound enters the circulation, which transfers it to all parts of the body. After this phase, the most important factor affecting the distribution is the passage of the compound through biological membranes. From the point of view of the distribution of a chemical compound, the organism can be divided into three different compartments: (1) the plasma compartment; (2) the intercellular compartment; and (3) the intracellular compartment. In all these compartments, a chemical compound can be solved to water and bound to biological macromolecules. The proportion of bound and unbound (free) chemical compound depends on the characteristics of both the chemical and the binding macromolecules. Binding to macromolecules increases distribution to the compartment.[227]

In the plasma, most chemicals are bound to plasma proteins. Albumin is quantitatively the most important binding protein but beta globulin and acidic glycoprotein also bind chemicals. The number of binding sites is limited, and, therefore, high doses of chemicals may

cause saturation of protein binding. In most cases an adverse effect does not require saturation of protein binding sites because free and bound chemical are in equilibrium in the plasma, and the free chemical is available for toxic action in the target tissues. The circulation is extremely important for distribution of chemicals. Heavily perfused organs, that is, the brain, liver, and kidneys, receive most of the cardiac output, and in these organs the concentration of a chemical increases more rapidly than in the other organs. Organs whose perfusion is small, for example, resting muscles and adipose tissue, receive only a small portion of the cardiac output, and therefore concentration of a chemical in these organs increases much more slowly than in the heavily perfused organs.[226,227]

Adipose tissue and bones function as storage sites for many substances. Most chemicals have some tissue specificity with regard to their tissue binding. In many cases, this property of a chemical is not important, but, especially for lipid-soluble chemicals, adipose tissue often becomes an important storage depot from which they are released. Both accumulation and release of compounds from the adipose tissue are slow processes, partly because adipose tissue receives only 2% of the cardiac output. The accumulated compound may be released if the size of the fat depot decreases. For example, lipid-soluble insecticides, such as chlordane, may even cause acute intoxication due to dieting, and dieting also causes release of the supertoxic compound dioxin into the circulation. Lipid-soluble compounds can also be released from their depots in the adipose tissue during breast feeding of infants, and this may cause excessive exposure.[234] The features of absorption, distribution, and excretion have been depicted in Fig. 5.37.

Another important storage depot for toxic compounds is the skeleton. In particular, fluoride, cadmium and lead bind and accumulate in the bone tissue from which they are released very slowly. The half-life of cadmium is several years, the half-life of lead is several months.

Theoretical volume of distribution (V_d) of a chemical is the volume in which the chemical would be distributed if its concentration were equal to a theoretical steady-state plasma concentration (C_0) at time zero. The volume of distribution quantifies the distribution of a compound between plasma and rest of the body. It is thus obtained quite similarly as the steady state concentration of a compound in the workroom air:

$$V_d = \frac{m}{C_0}, \tag{5.51}$$

where m is the dose of a compound and C_0 is its theoretical plasma concentration at time zero. Even though the volume of distribution is not the real volume, it provides valuable information on the behavior of the compound in the body that can be used in a number of pharmaco/toxicokinetic models.[226]

Special considerations

Chemical compounds may also be distributed to the placenta and through the placenta to the fetus and thereby cause exposure of the offspring. Even though the placental wall consists of several layers, it is a biological membrane, and the same principles apply to the placenta as to any other biological membrane, that is, penetration depends on lipid solubility and ionization of chemical compounds. The absorbed compounds penetrate the placenta, are transferred from mother's to the fetus's circulation and an equilibrium will be reached between the mother and the fetus. This is the main reason chemical exposure during pregnancy is strictly controlled in most developed countries.

5.3.3.3 Metabolism

The purpose of metabolism or biotransformation of xenobiotics (foreign compounds) is to transform them into a water-soluble form so that they can be excreted either in the urine or in the bile. These processes are catalyzed by a number of enzymes. Biotransformation reactions are divided into phase I and phase II reactions and catalyzed by the corresponding enzymes. In phase I reactions, functional groups, such as the hydroxyl group, are linked to the xenobiotic (Fig. 5.38). This is why phase I reactions are also called functionalization reactions, which are oxidation, reduction, or hydrolysis reactions. In phase II, the functional group is conjugated with a small endogenous compounds such as glucuronic acid, glutathione (GSH), sulfone, methyl, acetyl, or glycine. Most xenobiotics undergo both phase I and phase II reactions, but some compounds undergo only one of the phases. It is noteworthy that rarely whole dose of the absorbed compound will be metabolized; in most cases small amounts of unchanged parent compound can also be found in the urine. This can also be utilized as a specific biological monitoring test. The enzymes responsible for biotransformation of xenobiotics also catalyze the metabolism of endogenous compounds, such as hormones and neurotransmitters. For example, steroid hormones undergo phase I oxidation catalyzed by CYP enzymes and then conjugation reactions of the functional groups catalyzed by glucuronyltransferase, sulfotransferase, or other conjugating enzyme. The number of possible metabolites of various chemicals is often very large because of the multiple of reactions of several phase I and phase II enzymes in the cells.[235]

The highest number and amounts of biotransformation enzymes are found in the liver, and this organ

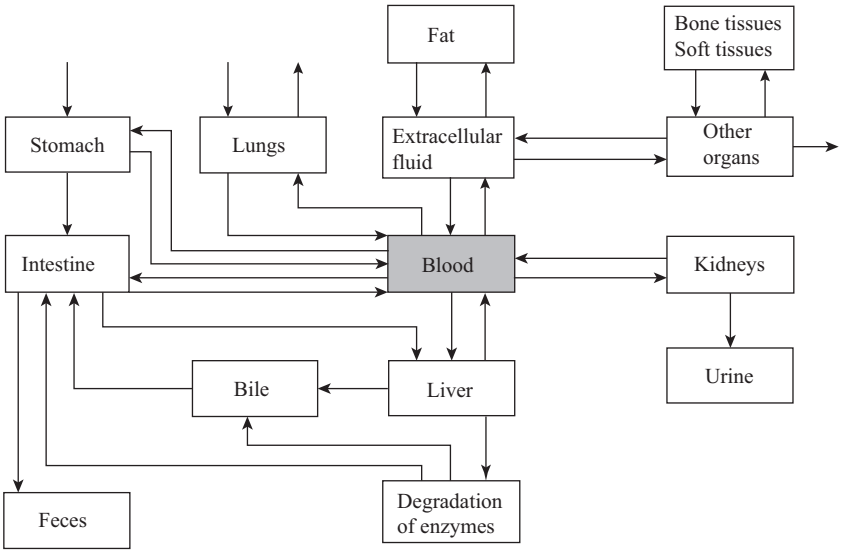


FIGURE 5.38 Schematic representation of absorption, distribution, and excretion of xenobiotics.[178]

plays a key role in the metabolism of both endogenous and foreign compounds. However, these enzymes can be found in many other organs, and one can hypothesize that enzymes expressed at the entries to the body, that is, the skin and the mucosa of the gastrointestinal tract and airways, have developed during evolution to protect the organism against foreign compounds. In fact, the liver and kidneys are also direct or indirect sites of entry of foreign compounds into the body. The liver is an important port of entry because of the portal vein that carries most foreign compounds directly from the intestine to the liver. The kidneys can also be a port for chemical compounds into the body because a number of compounds excreted in the urine may be reabsorbed in the proximal tubules of the kidney. Such compounds include those with an active transport system, many lipid-soluble compounds, and metabolites that have been hydrolyzed in the urine. On the other hand, ionic metabolites such as glucuronide, sulfate, and amino acid conjugates of these parent compounds are readily excreted in the urine or bile. GSH conjugates are metabolized to mercapturic acids in kidney and then excreted to urine, while these conjugates can excreted without further metabolism to the bile.[235]

Biologically active compounds are often inactivated during biotransformation. However, in some instances, biological activity of chemical compounds may be increased. Especially CYP enzymes catalyze activation of protoxin compounds to toxic metabolites. These reactions may yield two types of active metabolites. First active metabolites can be ligands for target molecules, with which the interaction is selective. Example of this type of activation is desulfuration of phosphorothioate organophosphates. The first pass metabolism can increase this type of compounds toxicity as demonstrated by orally exposed parathion. It is desulfurated to potent paraxon in the first pass metabolism and it is more potent than given intravenously. Secondly, metabolic activation can produce reactive electrophilic metabolites, which react more randomly with the nucleophilic groups of biological macromolecules, such as nucleic acids and proteins. Toxic consequences of too high level of electrophilic metabolites can be cell death, mutagenesis, malignant transformation of cells, or teratogenesis. For example, activation of carbon tetrachloride, bromobenzene, and acetaminophen (paracetamol) after high doses cause liver necrosis. At lower doses, they may cause genotoxic alterations in the cells and subsequent malignant transformation of the exposed cells. Active metabolites of mycotoxin aflatoxin B1, combustion product benzo(a) pyrene, and plastic monomer vinyl chloride, induce cancer subsequent to their binding with bases in the DNA. Since all of the compounds that are absorbed in the gastrointestinal tract enter the liver directly through the portal vein, their biotransformation can take place in the liver hepatocytes. The anatomical structure of the liver further promotes effective biotransformation of xenobiotics in this organ (Fig. 5.39). A number of factors affect the metabolizing capacity of the liver. These include blood flow through the liver, the uptake of the compounds by the liver cells, the concentration of xenobiotic metabolizing enzymes, affinity of the compounds to these enzymes, structure of the compounds, for example, highly chlorinated compounds are slowly metabolized, and different pathological processes such as collagen formation due to cirrhosis and hepatitis.[219,220]

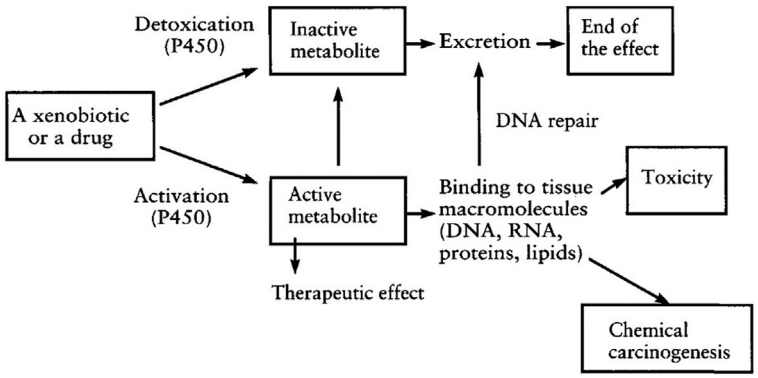


FIGURE 5.39 Janus faces of the biotransformation of xenobiotics. On one hand metabolism leads to inactivation and elimination of xenobiotics, but on the other hand many metabolites are reactive and may cause deleterious effects by binding to DNA, proteins, and other macromolecules.

5.3.3.4 *Excretion*

Water solubility (polarity) is essential for excretion. Even though lipid-soluble compounds may also be excreted to primary urine, they are usually at least partially reabsorbed. The ionic metabolites formed in the liver and extrahepatic tissues remain free (i.e., not bound to proteins) and are, therefore, readily excreted in urine and bile. Methylated, acetylated and often reduced metabolites are less polar and slower excreted than the parent xenobiotic.

Cadmium is effectively accumulated in the kidneys. When the cadmium concentration exceeds 200 μg/g in the kidney cortex, tubular damage will occur in 10% of the population, and proteins begin to leak into urine (proteinuria). When the concentration of cadmium in the kidney cortex exceeds 300 μg/g, the effect is seen in 50% of the exposed population. Typically, excretion of low-molecular-weight proteins, such as beta-microglobulin, is increased, due to dysfunction of proximal tubular cells of the kidney. The existence of albumin or other high-molecular-weight proteins in the urine indicates that a glomerular injury has also taken place. The excretion of protein-bound cadmium will also be increased.[226,237]

Pulmonary excretion takes place for volatile compounds. Alveolar air is at equilibrium with capillary blood. Thus pulmonary excretion depends on the vapor pressure of the compound and its blood solubility. If blood solubility is low, the compound will be rapidly excreted (see Section 2.3.9). The determination of alveolar exhaled air concentration can be used as biological exposure test for organic solvents. This test is also widely applied to control for drunken driving due to precise alcometers. The concentration of a solvent in the blood is obtained by multiplying the alveolar air concentration by the blood solubility coefficient, which is 2300.[226,238]

Lungs also secrete nonvolatile compounds. Lipid-soluble compounds may thus be transported with the alveobronchotracheal mucus to the pharynx, where they are swallowed. They may then be excreted or reabsorbed. Particles are also removed by this mucociliary escalator.

5.3.3.5 *Movements of chemical compounds in the body*

Absorption, distribution, biotransformation, and excretion of chemical compounds have been discussed as separate phenomena[239]. In reality all these processes occur simultaneously, and are integrated processes, that is, they all affect internal dose of a compound. To understand the whole outcome of these actions in the body, concentrations of the compound itself and its metabolites need to be determined at different time points from the initial exposure to the total elimination. Analysis of these data produces toxicokinetic description of the compound. Toxicokinetic determines the duration of action of a substance in the organism. It is essential to quantify the concentrations of the compound and its metabolites for presenting its toxicokinetics. For presenting this informatione various models are used, of which the most widely utilized are the one-compartment, two-compartment, and various physiologically based pharmacokinetic models. These models resemble models used in ventilation engineering to characterize air exchange.

One-compartment model

The simplest toxicokinetic analysis involves measurement of the plasma concentrations of a chemical at several time points after the administration of a single intravenous injection. If the kinetic data obtained yield a straight line when plotted as the logarithm of plasma concentrations versus time, the kinetics of the compound can be described by a one-compartment model, in which the whole body is treated as one single space or compartment. This type of plasma concentration dependency is called also the first-order kinetics. Even though the one-compartment model is an extreme simplification of the compound in the organism in the

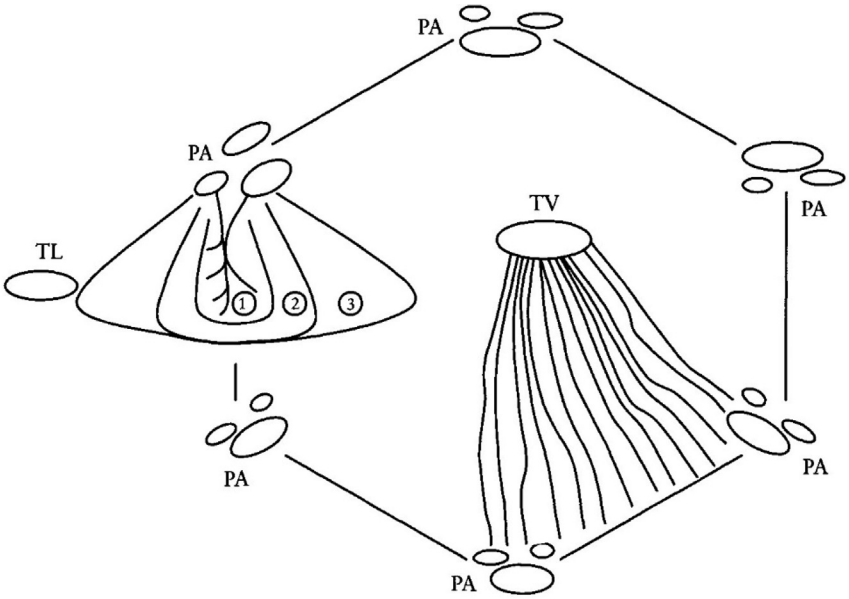


FIGURE 5.40 Pictorial presentation of the microscopic structure of the liver. The picture shows the classical liver lobulus. The functional acinus and its three zones are at the left. The acinal zones are marked by numbering them 1–3. These zones correspond to the direction of blood flow from the HA to the TV. Zone I corresponds to the periportal area in classical liver pathology, zone 2, the interlobular region (midzone), and zone 3, centrelobular region.[236] *HA*, Hepatic arteries; *TV*, terminal veins.

physiological and toxicological sense, the behavior of several chemical compounds can be well described and understood by using this model. The kinetics of compounds whose distribution is homogenous and distribution is rapid in the body can be described with the one-compartment model. As described below (Fig. 5.40), the theoretical initial concentration C_0 can then be calculated.

The rate of elimination of a chemical compound from the body is proportional to the amount of the chemical in the body. Elimination processes include biotransformation, exhalation, and excretion in the urine, bile, saliva, and sweat, and even in the hair and nails. The first-order elimination rate constant k_{el} is the slope of the line describing the decrease of concentration (5.54). Its unit is reciprocal time (e.g., min^{-1} or h^{-1}). For example, if the elimination rate constant is $0.5\,h^{-1}$ the percentage of the dose excreted after the first, second or third hour is the same, regardless of the given dose. In this case, the percentage of the dose excreted is 39%, even though the rate constant is $0.5\,h^{-1}$, because the dose remaining in the body (C) decreases continuously with time. The elimination rate decreases when the dose remaining in the body (C) decreases. The first-order elimination of the compound is mathematically expressed as an exponential equation $C = C_0 \times exp(-k_{el}t)$ where C is the plasma concentration, C_0 is plasma concentration at zero time point, k_{el} the first-order elimination rate constant, and t the time of blood sampling. With logarithmic transformation a straight line is obtained:

$$\ln C_t = \ln C_0 - k_{el} \cdot t \qquad (5.52)$$

where ln C_0 represents the intercept and $-k_{el}$ represents the slope of the line. Therefore the first-order elimination rate constants can be determined by utilizing the slope of the ln C versus time plot.

In addition to the elimination rate constant, the half-life ($t_{1/2}$) is another important parameter that characterizes the time-course of chemical compounds in the body. The elimination half-life ($t_{1/2}$) is the time to reduce the concentration of a chemical in plasma to half of its original level. The relationship of half-life to the elimination rate constant is $t_{1/2} = 0.693/k_{el}$ and, therefore, the half-life of a chemical compound can be determined after the determination of k_{el} from the slope of the line. The half-life can also be determined through visual inspection from the log C versus time plot (Fig. 5.41). For compounds that are eliminated through first-order kinetics, the time required for the plasma concentration to be decreased by one half is constant. It is important to understand that the half-life of chemicals that are eliminated by first-order kinetics is independent of dose.[239,240] If the dose and bioavailability are known, the distribution volume (Vd) can be calculated with the equation $Vd = \text{Dose}/C_0$. C_0 is derived from Eq. (5.54) at time point zero.

The independent toxicokinetic parameter is clearance (Cl). It indicates the blood volume, which is cleaned of the compound per unit time, that is, mL/min or L/h. The most important clearance processes are hepatic (Clh), renal (Clr), or lung (Cll), which are additive and the sum of which determines the total clearance. Hepatic, renal, and lung clearance are always smaller than the blood flow of the organ; for example, normal flow of human liver 90 L/h. Hepatic

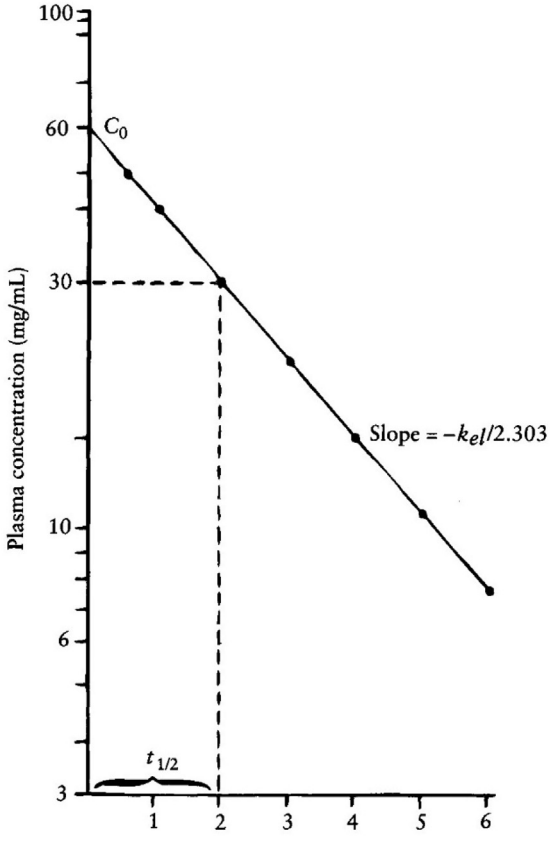


FIGURE 5.41 Schematic representation of the concentration of a chemical in the plasma as a function of time after an intravenous injection if the body acts as a one-compartment system and elimination of the chemical obeys first-order kinetics with a rate constant (k_{el}).

clearance of 60 L/h means that two-third of the compound is cleaned by the liver in one pass.

Two-compartment model

If the plotting of the logarithm of the plasma concentration against time does not result in a straight line but rather in a curve, the use of multicompartment models is required. Multicompartment models are required for compounds that distribute to different organs or are eliminated by different organs at different rates. Such compounds are usually either distributed quickly and then eliminated slower or slow distribution relative to elimination. This results in multiexponential elimination. In the simplest case, this type of curve can be resolved into two exponential terms (a two-compartment model). Concentration can be expressed as $C = Ae\alpha^*t + B\beta^*t$ where A and B are proportionality constants and α and β are rate constants of distribution and elimination, respectively. If during the distribution alpha phase, concentrations of the chemical in plasma decrease more rapidly than during the beta phase, then half-life of this terminal phase is an appropriate estimation of the elimination. The length of the phases may vary from minutes to hours to days. Whether the distribution phase becomes apparent depends on the time of the sampling after the cessation of the exposure. Since most chemicals in the occupational environment follow two- or other multicompartmental kinetics, the correct timing of blood sampling for biological monitoring is essential.[239,240]

In a two-compartment model, β is equivalent to k_{el} in the one-compartment model. Therefore the terminal half-life for the elimination of a chemical compound following two-compartment model elimination can be calculated from the equation $\beta = 0.693/t_{1/2}$.

Saturation of Elimination

Saturation kinetics is also called zero-order kinetics or Michaelis–Menten kinetics. The Michaelis–Menten equation is mainly used to characterize the enzymatic rate at different substrate concentrations, but it is also widely applied to characterize the elimination of chemical (the first-order kinetics) compounds from the body. The substrate concentration that produces half-maximal velocity of an enzymatic reaction, termed K_m value or Michaelis–Menten constant, can be determined experimentally by graphing v_i as a function of substrate concentration, [S].

The Michaelis–Menten equation is written

$$v_i = \frac{v_{\max}[S]}{K_m + [S]} \tag{5.53}$$

where v_i is the measured initial velocity of an enzymatic reaction, $v_{\max}$ is the maximal velocity of the enzymatic reaction, and K_m is the Michaelis–Menten constant. Note that when [S] far exceeds the K_m, the initial velocity, v_i, is close to the maximal velocity, $v_{\max}$.

In zero-order kinetics, a constant amount of a chemical compound is excreted per unit of time. In most cases this phenomenon is caused by the saturation of a rate-limiting enzyme, and the enzyme commonly functions at its maximal rate, that is, a constant amount of a chemical compound is metabolized per unit time. A good example is ethyl alcohol; alcohol dehydrogenase becomes saturated with normal doses of alcohol beverages; for example, 0.5‰ blood concentration is more than 20 times higher than the K_m-value of ethanol for alcohol dehydrogenase. Because of this saturation, ethyl alcohol is eliminated at a constant rate about 0.1 g/h/kg or 7 g/h in human. However, the reason is not always an enzyme; any system that becomes saturated follows zero-order kinetics. When the concentration of a chemical compound decreases below the saturation concentration, it returns to the first-order kinetics.

From a practical point of view, saturation of elimination has important consequences. If the metabolism becomes saturated, the duration of the action of the compound is prolonged and half-life cannot be determined. In such a case, correct timing for collection of biological monitoring samples also becomes difficult to assess. Furthermore saturation of metabolism may also have qualitative effects. For example, it has been argued (but not yet proved) that arsenic compounds cause cancer at high doses at which methylation of inorganic arsenic becomes saturated.

Physiologically based toxicokinetic models

Physiologically based toxicokinetic (PBTK) models are nowadays used increasingly for toxicological risk assessment. These models are based on parameters of human or animal physiology, anatomy and biochemistry and thus take into consideration the actual toxicokinetic processes more accurately than the one- or two-compartment models. In these models, all of the relevant information regarding absorption, distribution, biotransformation, and elimination of a compound is utilized. It means that to the PBTK models there are selected minimum number of key physiological actions to describe the behavior of compound in the body. The principles of physiologically based pharmaco/toxicokinetic models are depicted in Fig. 5.42A and B. The main difficulty in using these models is that in most cases not enough information is currently available about the compound under study.[239]

Advantages of these models are that they can simulate the concentration of a compound also in target tissues and predict toxicokinetics across species and populations and predict effects of changes of physiological processes

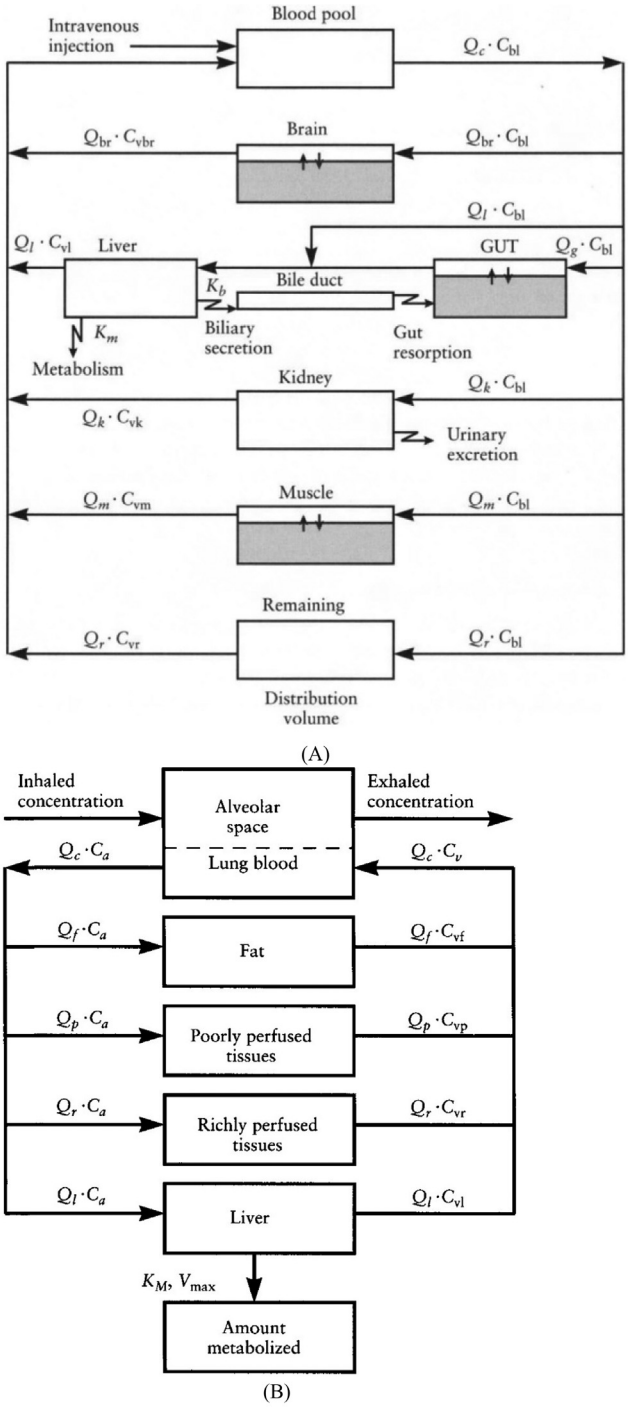


FIGURE 5.42 (A) Physiological model for phenobarbital. (B) Physiological model for the volatile organic chemical benzene.

5.3.4 Toxic effects of chemicals

5.3.4.1 The nature of toxic effects

A toxic reaction may take place during or soon after exposure (defined as acute toxicity), or it may only appear after a latency period (defined as chronic toxicity). Chronic toxicity requires exposure of several years for a toxic effect to occur in humans. Acute toxic reactions that occur immediately are easy to associate with the exposure and the exposure–effect relationship can readily be demonstrated. The longer the time interval between exposure and effect, the more difficult it is to delineate the relationship between exposure and effect.

Toxic effects can occur at the site of exposure as exposure to acetic acid causes irritation of the skin or irritation of bronchial cells following exposure to NO. If chemical has been absorbed into the circulation, through the skin, lungs, or gastrointestinal tract, it can cause systemic effects in any organ or tissue.

Toxic effects often disappear after cessation of exposure, but they can also be permanent. The ability of the tissue to regenerate is one of the most important factors that determines the nature of toxic effects. For

example, liver has a remarkable capacity to regenerate, and therefore liver injury is often reversible. On the other hand, neuronal cells do not regenerate at all, thus neuronal injury is irreversible. It is true that neuronal cells can compensate for possible losses, but only to a minor degree. In particular, chronic effects tend to be irreversible.[227]

There are some basic differences between toxic and allergic reactions. The most important differences are (1) an allergic reaction always requires a prior exposure to the compound, and this reaction only occurs in sensitized individuals and (2) a dose–response relationship is characteristic to a toxic reaction, whereas such a relationship is less clear for an allergic reaction. Even minute doses can elicit an allergic reaction in a sensitized individual (see Fig. 5.43).[227]

5.3.4.2 *Joint effects of chemicals*

Industrial workers are almost always exposed to several agents simultaneously. The possible interactions of these multiple exposures are (because the possible combinations are almost infinite) an area of great uncertainty. The situation is further complicated by the simultaneous presence of many lifestyle factors, especially smoking and the use of alcohol and drugs. Other exposures may enhance the toxic effect of an agent. Most commonly the increased combined effect is additive $(1+1=2)$, but it can also be synergistic $(1+1=>2)$. The neurotoxic effects of most organic solvents are usually additive; therefore, industrial hygienists use the combined exposure level to assess the conditions. It is obtained by dividing the concentration of each solvent by its OEL and by adding the quotients. If the sum exceeds one, the exposure is considered excessive. There are cases of synergism, where the toxic effects of individual exposures become greatly potentiated. A well-known example is the combination of asbestos exposure and smoking. Various constituents of a mixture may have no mutual interactions. In such a case, the effects of different agents can be considered individually. Since in most cases we are ignorant of these potential interactions, the no-interaction assumption is the most common premise. Finally, it is also possible that some constituents reduce the effects of other exposures; however, there are no well-demonstrated examples of this kind of antagonistic action between occupational exposures.

The interactions may be physicochemical without the participation of biological mechanisms; for example, deep lung exposure to highly soluble irritative gases, such as sulfur dioxide, may become enhanced due to adsorption of the gas onto fine particles. Biological interactions may occur at all stages and body sites. For example, toxicity is increased when adverse effects are due to some reactive metabolic intermediate and exposure to another agent stimulates its metabolic activation (enzyme induction).

5.3.4.3 *Mechanisms of toxicity*

Paracelsus, a Swiss physician of the 16th century, stated that everything is toxic, it is just the dose that matters.[241] This statement still holds true 500 years after Paracelsus developed it to defend the use of toxic compounds such as lead and mercury in the treatment of serious diseases such as syphilis. Chemical compounds cause their toxic effects by inducing changes

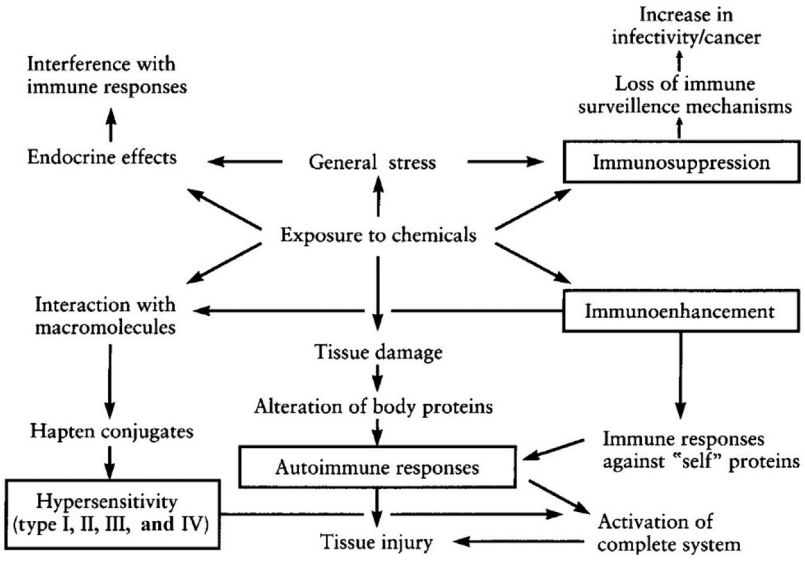


FIGURE 5.43 Responses of the immune system to exposure to some chemicals.[229] Source: *Used with permission.*

in cell structure, physiology and biochemistry, and an understanding of cellular biology is a prerequisite if one wishes to understand the nature of toxic reactions.

Toxic reactions occur by several mechanisms: activation of metabolism, production of reactive intermediates, and subsequent reactions with cell macromolecules (proteins, RNA, and DNA), changing receptor responses, disrupting function of enzymes, and transporters or through abnormal defense reactions. Several compounds cause toxicity by mimicking the organism's own hormones or neurotransmitters, or activating the body's endogenous receptors in some nonphysiological way.[227]

Cells are capable of repairing minor damage, but extensive damage leads to cell death. There are various forms of cell death, including apoptosis, necrosis, oncosis, pyroptosis, and autophagy. One of the main forms is necrosis, which is a chemical-driven chaotic and passive process, and the other is programmed cell death, apoptosis, which is a genetically controlled and energy consuming process. Apoptosis is also a part of normal cell physiology in organogenesis during the development of the embryo before superfluous cells commit a form of cellular suicide by activating their apoptotic programs. However, many chemicals, for example, several quinone oxidants, and heavy metals may overtly augment apoptosis in adults when it can turn into a pathological process. Thus cell death is a crucial toxic injury which is affected by the rapidity of the cell injury as well as the target organ. It is noteworthy that the dose of a compound may determine whether cells will proliferate in a tissue or undergo apoptosis or necrosis (see Fig. 5.44 for necrosis and apoptosis).[227,243]

Cells in various tissues such as liver, kidney, or gastrointestinal tract, have a remarkable capacity to repair injuries inflicted by chemicals. Furthermore the ability of most organs to fulfill their functions usually exceeds requirements that they need to perform. For example, humans can live with one lung, one kidney, and only part of their liver. In this regard, the CNS is an exception because neuronal cells do not regenerate. However, even neuronal cells are capable of compensating for an injury. This does not take place through the replacement of dead cells but through the outgrowth of new extensions of existing neurons and through the formation of new synapses, that is, contacts between neurons that allow chemical neurotransmission between neurons. Even though many toxic effects are due to cell death, toxicity may occur with functional consequences without there being any visible morphological alterations in cells or tissues.[227]

Chemically induced changes in DNA, that is, mutations, chromosomal damage, and epigenetic modifications, are also an important toxicity mechanism. The bases in DNA, like bases in general, are nucleophilic (electron donors) and react with electrophiles (electron acceptors). Strong electrophiles, such as carbonium ions and epoxides, are formed during the metabolism of many known potent carcinogens. Thus the formation of DNA adducts may cause malignant transformation of cells and lead to initiation of cancer. In addition,

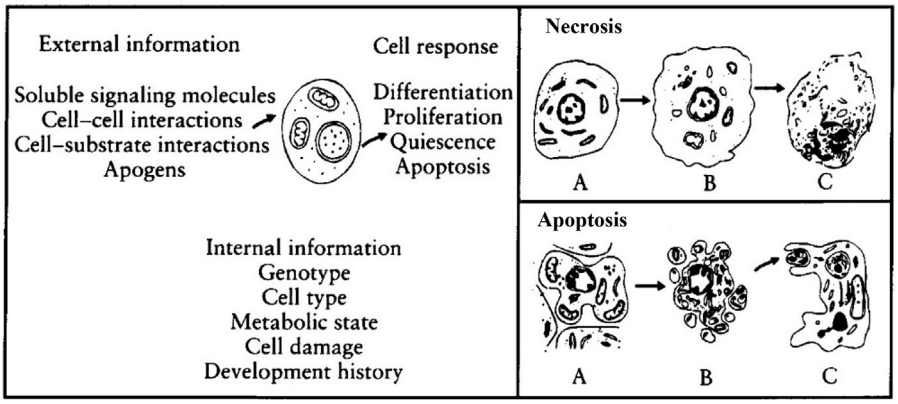


FIGURE 5.44 *Left:* External and internal stimuli triggering various cellular responses including apoptosis. *Right:* Comparison of morphologic characteristics of necrosis (*top*) with apoptosis (*bottom*). A normal cell (*top, A*) usually begins the process of necrosis with an initial phase of generalized swelling (*top, B*), which progresses to a dissolution of organelles and rupture of plasma membranes (*top, C*). The earliest phase of apoptosis (*bottom, A*) involves retraction from adjacent cells, loss of specialized surface structures, shrinkage with condensation of cytoplasm, margination of compacted nuclear chromatin, and localized protrusions of the cell surface. Nuclear fragmentation may occur at this time. In the next phase, the protuberances of the cell surfaces separate into multiple membrane-bound bodies (apoptotic bodies) that contain nuclear remnants and intact organelles. The apoptotic bodies are then engulfed and degraded by resident tissue cells (*bottom, C*) or phagocytes. Note that the light microscopic appearances of nuclear rupture and chromatin disintegration (karyorrhexis) may be seen in both late necrosis (*top, C*) and apoptosis (*bottom, B*). Source: *Modified from Loikkanen J, Naarala J, Savolainen KM. Modification of glutamate-induced oxidative stress by lead: the role of extracellular calcium. Free Rad Biol Med 1998;24:377–84.*[242]

chemicals can cause epigenetic modifications to DNA, that is, heritable changes in genome without a change in DNA sequence. These epigenetic changes include DNA methylation, covalent modifications of histone tails, and regulation by noncoding RNAs. In the following section, mechanisms whereby chemical compounds induce their toxicity will be discussed.[210,227]

Receptor-mediated toxicity

Several chemical compounds induce their toxic and other effects through stimulating specific receptors and events occurring after receptor activation, a process called signal transduction. Receptors themselves are protein molecules sitting in the lipid bilayer of the cell membrane or located in the cytosol in the case of nuclear receptors. They have the ability to recognize physiological intercellular transmitters such as hormones, neurotransmitters or growth factors (also called first messengers). Normally a very small amount of a transmitter is sufficient to activate the receptor. Many of the receptors are ion channels and their activation leads to the influx of ions into the cell. There are specific receptor-coupled receptors for sodium, potassium, and calcium. Increased influx of these ions usually leads to increased enzymatic activity, and activation of the cell. Some receptors are intimately associated with enzymes such as tyrosine kinase, adenylate cyclase, or phospholipase C. In addition, nuclear receptors are ligand-regulated transcription factors that regulate transcription of the target genes in response to small lipophilic compounds.[227,244,245]

Some of the cell membrane receptors are coupled to an amplifier, called the G-protein. Activation of a G-protein leads either to activation or inhibition of an effector enzyme on the internal side of the cell membrane (see Fig. 5.45).[227] These effector enzymes are responsible for the generation of second messengers that are essential for cellular signal transduction. Although first messengers, described above, are responsible for chemical intercellular communication, second messengers are responsible for transducing the information that has reached the cell surface receptor to all parts of the cell interior. There is also a specific enzyme machinery for inactivating the second messengers to terminate the action that was initiated by the first messenger. Typical effects of a second messenger are elevation of free intracellular calcium associated with cellular activation, activation of specific enzymes such as protein kinase C, or production of a tertiary cellular messenger such as NO. NO is a gaseous cellular messenger that can act as both an intra- and intercellular signal transduction factor. Being lipid soluble, NO easily diffuses in the cell as well as penetrating through the cell membrane and thereby also reaching other cells.[227,246]

Nuclear receptors are activated by a variety of endogenous and exogenous ligands including steroid and thyroid hormones, vitamin D, and environmental contaminants. Some nuclear receptors are known to become activated only due to interaction with a synthetic chemical, and no endogenous ligand for such a receptor has been identified. An example of a receptor for which no high-affinity endogenous ligands have been identified is the aryl hydrocarbon receptor (AHR), a nuclear receptor that becomes activated subsequent to its exposure to 2,3,7,8-tetrachlorodibenzo-*p*-dioxin (TCDD). Activation of the AHR by TCDD leads to increased expression of multiple genes that encode

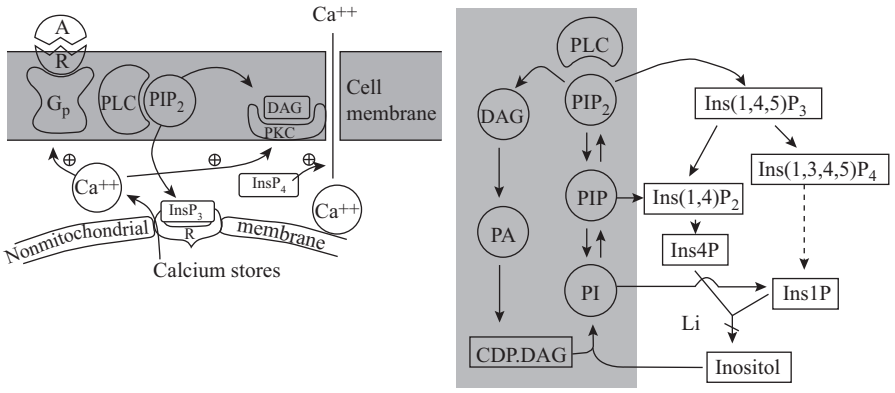


FIGURE 5.45 Acetylcholine (A) binds to a receptor (R) coupled to a G-protein (G_p), and stimulates PLC. PLC hydrolyzes PIP_2 to Ins(1,4,5)P_3 and DAG. DAG stimulates PKC, and Ins(1,4,5)P_3 binds to its receptor (R) in the intracellular Ca^{2+} store, releases Ca^{2+}, and elevates the levels of free intracellular calcium ($[Ca^{2+}]_i$). Ins(1,3,4,5)P_4 is formed from Ins(1,4,5)P_3 by phosphorylation, and it, together with Ins(1,4,5)P_3 controls influx of Ca^{2+}. PKC and Ca^{2+} cause neuronal stimulation.[246] *DAG*, Diacylglycerol; *Ins(1,4,5)P_3*, inositol-(1,4,5)-triphosphate; *Ins(1,3,4,5)P_4*, Inositol-(l,3,4,5)-tetracisphosphate; *PLC*, phospholipase C; *PIP_2*, phosphatidylinositol-(4,5)-bisphosphate; *PKC*, protein kinase C. Source: *Used with permission.*

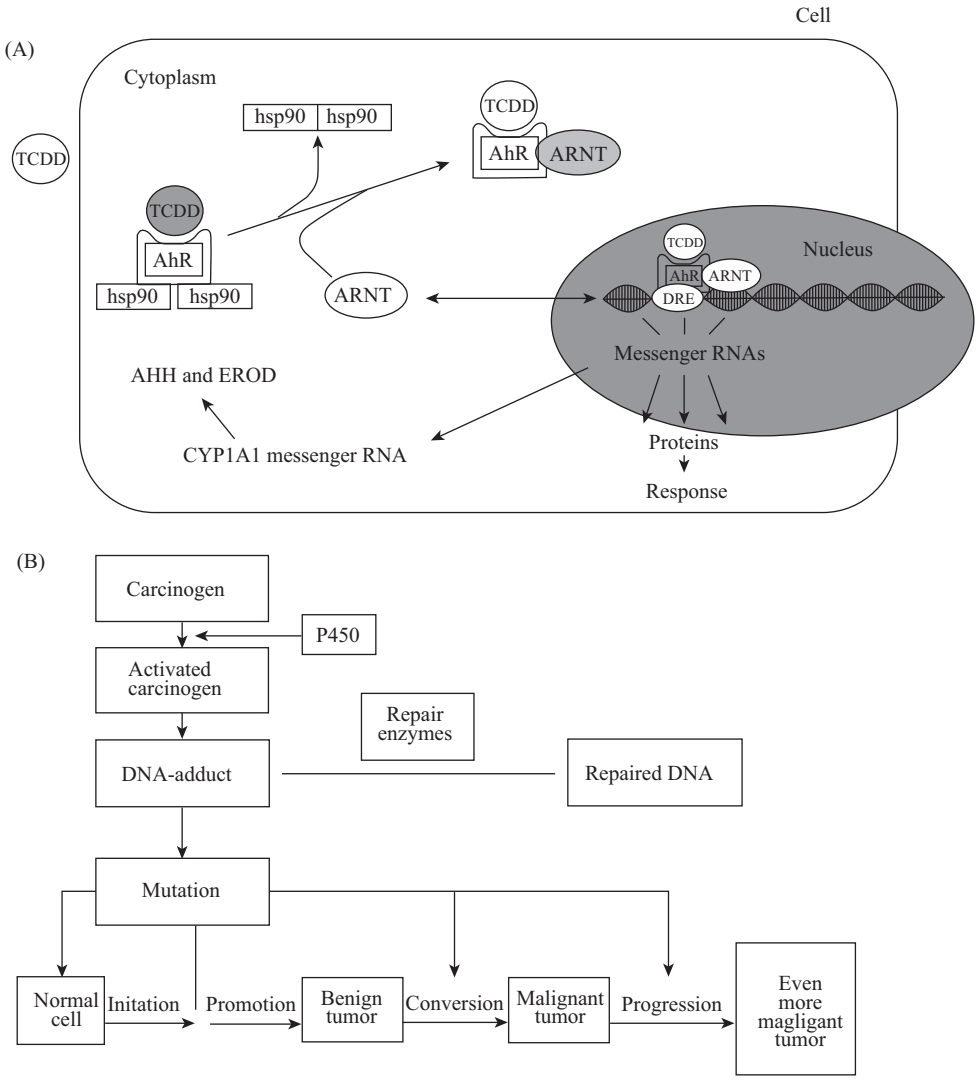


FIGURE 5.46 (*a*) Mechanism of induction of xenobiotic metabolism caused by TCDD and polycyclic aromatic hydrocarbons via the AhR. In the cytosol, soluble Ah-receptor is associated with two hsp. When TCDD molecule binds to the Ah-receptor, the stress proteins are released, and the TCDD-Ah-receptor complex dimerizes with an Ah-receptor nuclear translocater protein. In the nucleus, this dimer binds to the DRE and causes an induction of the synthesis of messenger RNA, subsequent translocation to ribosomes, and formation of proteins, for example, P450 enzymes as well as induction of AHH and EROD activities. (B) Mutations according to our current understanding are an essential part of every stage of chemical carcinogenesis. *AHH*, Aryl hydrocarbon hydroxylase; *AhR*, Ah-receptor; *DRE*, dioxin response element; *EROD*, ethoxyresorufin-O-deethylase; *hsp*, heat shock protein.

for many proteins, such as the xenobiotic metabolizing phase I and II CYP family (see Fig. 5.46).[227,247]

In the CNS, acetylcholine (ACh) is one of the key excitatory neurotransmitters (first messengers). In neuronal cells, it binds preferentially to muscarinic receptors. Stimulation of muscarinic receptors increases levels of neuronal free intracellular calcium leading to neuronal activation. If present in excess, ACh leads to epileptiformic seizures and tonic–clonic convulsions which may be associated with neuronal injury.[227] Glutamate (Glu) is the most ubiquitous excitatory neurotransmitter in the CNS which binds to several subtypes of glutamatergic receptors. Glutamate is released from nerve endings of glutamatergic neurons subsequent to neuronal stimulation, and also during hypoxia and neuronal injury. Glu also elevates neuronal levels of free intracellular calcium, activates cells, and, when in excess, can cause neuronal injury.[227,247] Thus both of these molecules are endogenous neurotransmitters which in excess are harmful to the CNS. Occupational and environmental contaminants such as lead may amplify the effects of Glu, and thereby cause severe neurotoxic risks to exposed individuals.[242,248]

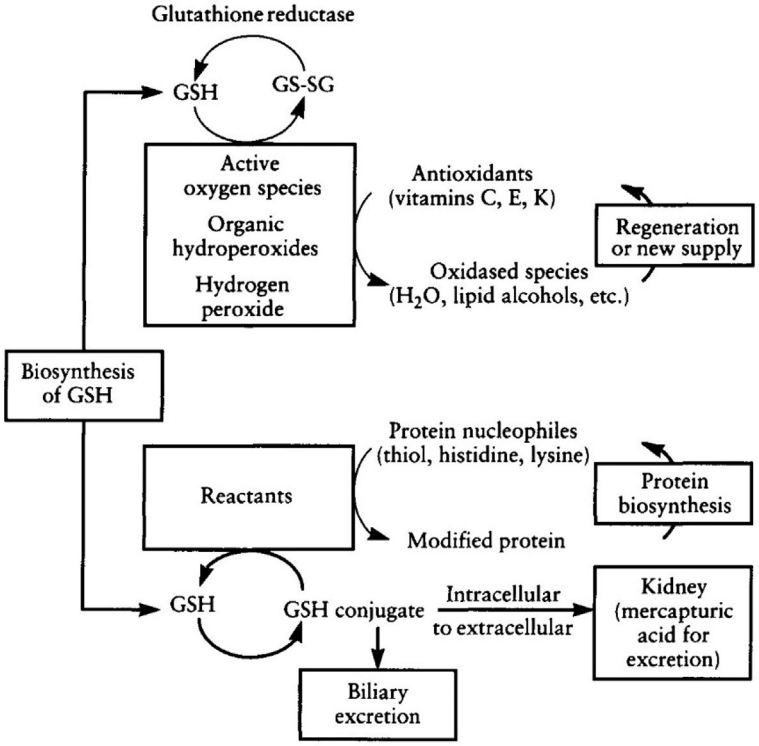


FIGURE 5.47 The role of glutathione and metabolic pathways Involved in the protection of tissues against intoxication by electrophiles, oxidants and active oxygen species.[229] Source: *Used with permission.*

One of the important consequences of neuronal stimulation is increased neuronal aerobic metabolism which produces reactive oxygen species (ROS). ROS can oxidize several biomolecules (carbohydrates, DNA, lipids, and proteins). Thus even oxygen, which is essential for aerobic life, may be potentially toxic to cells. Addition of one electron to molecular oxygen (O_2) generates a free radical (O_2^-), the superoxide anion. This is converted through activation of an enzyme, superoxide dismutase (SOD), to hydrogen peroxide (H_2O_2), which is, in turn, the source of the hydroxyl radical (OH). Usually catalase further metabolizes hydrogen peroxide to molecular oxygen and water. Oxygen may also be activated to the highly reactive singlet oxygen. It is important to note that the formation of ROS is a part of normal cell respiration, that is, in electron transfer during metabolism of oxygen by the CYP enzyme system. During this process, a part of the ROS formed leaks into the cell (Fig. 5.47).[227,249]

Production of ROS is not only a detrimental process; several cells carry out their functions in the body by generating ROS. For example, neutrophils and macrophages produce ROS upon activation. This is one of their ways of destroying invading microorganisms. However, other exposures, such as mineral fibers, and inorganic and biological particles are also able to activate phagocytes to produce ROS. Excessive ROS production may be harmful to the host cell and surrounding cells.[227,249]

Cells have defense systems to protect themselves against these radical species. The defense systems constitute intracellular thiols, such as GSH, a molecule rich in SH groups, and thus capable of scavenging the reactive species through oxidation of the SH groups. This oxidation leads to the formation of disulfide bridges; oxidized GSH is unable to scavenge oxygen radicals. GSH has to be regenerated and this reduction is performed by a specific enzyme, GSH reductase.[249] Cells also contain water- and lipid-soluble molecules that remove ROS. The most important of these molecules are a water-soluble vitamin, vitamin C, that acts in the cytoplasm, and a lipid-soluble vitamin, vitamin E, that functions in the cell membrane. Cellular defense mechanisms against excessive production of ROS also include enzymes that metabolize these reactive species; SOD metabolizes superoxide anion to H_2O_2, and catalase breaks down H_2O_2 to molecular oxygen and water. Oxidative stress results when activation of cells leads to such a high production of ROS that it overwhelms the capacity of the defense mechanisms. The initial phases of stress are associated with depletion of cellular GSH. Then the depletion of defense vitamins C and E occurs. This means that vital biological macromolecules, notably DNA, proteins, carbohydrates, and lipids, can be attacked by the reactive species. This cascade of events may lead to cell death through necrosis or apoptosis.[227,242,243,246–250]

Effects on excitable membranes

Maintenance of electrical potential between the cell membrane exterior and interior is a necessity for the proper functioning of excitable neuronal and muscle cells. Chemical compounds can disturb ion fluxes that are essential for the maintenance of the membrane potentials. Fluxes of ions into the cells or out of the cells can be blocked by ion channel blockers (e.g., some marine toxins).[227]

Insecticides, such as DDT and lindane, cause their neurotoxic effects by affecting the functions of ion channels in the neuronal cell membrane, thereby altering depolarization of the cell membrane. Organic solvents also modify the normal functioning of excitable neuronal membranes. It was originally assumed that organic solvents non-specifically altered the fluidity of the cell membrane. Current knowledge is that the effects of organic solvents are more specifically directed toward cell membrane proteins such as ion channels, other receptors, and specific enzymes.[227]

Effects on cellular energy metabolism

Several toxic compounds act by inhibiting the oxidation of carbohydrates or by inhibiting the formation of adenosine triphosphate (ATP), a molecule that is an essential energy source of the cells. Cellular energy metabolism can be prevented by inducing anoxia, for example, by exposure to CO.[227] CO reacts with hemoglobin and forms carboxyhemoglobin, which is unable to bind oxygen. Nitrite-induced oxidation of heme iron causes formation of methemoglobin from hemoglobin. An increased amount of methemoglobin also prevents oxidation of cells and tissues because methemoglobin does not bind oxygen. However, the treatment of methemoglobinemia is much easier than the treatment of CO-induced carboxyhemoglobinemia. In fact, induction of slight methemoglobinemia with nitrite can be used as an antidote in cyanide poisonings, because the ferric (trivalent iron) form present in methemoglobin acts as a sink by binding free cyanide.[227,251]

Cyanide, hydrogen sulfide, and azides prevent cells and tissues from utilizing oxygen by binding to cytochrome oxidase and thereby preventing mitochondrial energy production. The release of hydrogen cyanide may take place if cyanides make contact with acids, and, for example, sewage workers may be exposed to hydrogen sulfide if anaerobic conditions occur. Formation of hydrogen sulfide also takes place in many industries. This gas is insidious because it is very unpleasant odor virtually disappears at high concentrations.[252] The inhibition of ATP formation can also take place through other mechanisms. Dinitrophenol (a herbicide) blocks the citric acid cycle by uncoupling it from mitochondrial oxidative metabolism. Fluoro-acetic acid, in turn, blocks the citric acid cycle by inhibiting several key enzymes in the cycle.[227,253]

Depletion of ATP in the cells prevents maintenance of the membrane potential, inhibits the functioning of ion pumps, and attenuates cellular signal transduction (e.g., formation of second messengers such as inositol phosphates or cyclic AMP). A marked ATP depletion ultimately impairs the activity of the cell and leads to cell death. The agents that impair mitochondrial ATP synthesis are listed in Table 5.12.

Disturbances in cellular calcium metabolism

As stated above, calcium is an extremely important cellular ion for several cellular functions. The concentration of calcium in human extracellular fluid is about 2.5 mM, while the intracellular concentration is only 100–200 nM depending on the cell type. Thus there is 10,000–20,000-fold concentration difference between the cell interior and exterior that has to be maintained by cellular pumping mechanisms. This requires a large amount of energy.[254]

The behavior of calcium in the cells can be considered as a metabolic process. There is uptake, distribution, and excretion of calcium in the cells. The uptake of calcium occurs via activation of calcium channels. The end result is elevation of intracellular calcium levels and subsequent activation. Because calcium is a powerful cell-activating ion, increased calcium levels in the cell have to be controlled carefully. There are a number of calcium pumps that are responsible for pumping of calcium out of the cells. This again requires a large amount of energy. The agents causing sustained elevation of cytosolic Ca^{2+} are listed in Table 5.12. [227,254]

Nitric oxide

NO is a gaseous cellular messenger that transmits information between cells and within cells. In spite of its physiological role, NO is also a reactive species which is capable of reacting with biological molecules, and therefore in some instances tissue damage may ensue. NO is produced by an enzyme, NOS which acts on arginine, transforming it into citrulline and NO. This enzyme has both inducible and constitutive forms. iNOS is expressed in immunological cells, mainly phagocytes such as macrophages and neutrophiles, and in epithelial cells of the airways as well as endothelial cells of the circulatory system.[255] eNOS (also known as cNOS) is expressed in many cells, for example, neuronal cells. It is characteristic of iNOS that NO is produced only subsequent to persistent induction of the enzyme. Upon stimulation, the induction of iNOS (stimulated synthesis of the enzyme

TABLE 5.12 Agents causing sustained elevation of cytosolic Ca^{2+} and/or impaired synthesis of mitochondrial ATP.

A. Agents inducing Ca^{2+} influx into the cytoplasm
- **I.** Via ligand-gated channels in neurons:
  - **1.** glutamate receptor agonists ("excitotoxins"): glutamate, kainate, and domoate
  - **2.** TRPV1 receptor ("capsaicin receptor") agonists: capsaicin and resiniferatoxin
  - **3.** TRPV2 receptor agonists: SH-reactive electrophiles, such as lacrimators (e.g., chlorobenzalmalonitrile), acrolein, methyl isocyanate, phosgene, and chloropicrin
- **II.** Via voltage-gated channels: maitotoxin (?) HO*
- **III.** Via "newly formed pores": maitotoxin, amphotericin B, chlordecone, and methylmercury alkyltins
- **IV.** Across disrupted cell membrane:
  - **1.** Detergents: exogenous detergents, lysophospholipids, and free fatty acids
  - **2.** Hydrolytic enzymes: phospholipases in snake venoms and endogenous phospholipase A_2
  - **3.** Lipid peroxidants: carbon tetrachloride
  - **4.** Cytoskeletal toxins (by inducing membrane blebbing): cytochalasins and phalloidin
- **V.** From mitochondria:
  - **1.** Oxidants of intramitochondrial NADH: alloxan, t-BHP, NAPBQI, divicine, fatty acid hydroperoxides, menadione, and MPP +
  - **2.** Others: phenylarsine oxide, gliotoxin
    - •NO, and ONOO
- **XVI.** From the endoplastic reticulum
  - **1.** IP3 receptor activators: γ-HCH (lindane) and IP3, formed during "excitotoxicity"
  - **2.** Ryanodine receptor activators: δ-HCH

B. Agents inhibiting Ca^{2+} export from the cytoplasm (inhibitors of Ca^{2+}-ATPase in cell membrane and/or endoplasmic reticulum)
- **I.** Covalent binders: acetaminophen, bromobenzene, CCl_4, chloroform, and DCE
- **II.** Thiol oxidants: cystamine (mixed disulfide formation), diamide, *t*-BHP, O
  - •2 and HOOH generators (e.g., menadione, diquat)
- **III.** Others: vanadate, Ca^{2+}, and thapsigargin (specific SERCA inhibitor)

C. Agents impairing mitochondrial ATP synthesis:
- **I.** Inhibitors of hydrogen delivery to the electron transport chain
  - **1.** Glycolysis (critical in neurons): hypoglycemia, iodoacetate, koningic acid, and NO^+
  - **2.** Gluconeogenesis (critical in renal tubular cells): coenzyme A depletors (see below)
  - **3.** Fatty acid oxidation (critical in cardiac muscle): hypoglycin, 4-pentenoic acid, and 4-ene-valproic acid
  - **4.** Pyruvate dehydrogenase: arsenite, DCVC, and *p*-benzoquinone
  - **5.** Citrate cycle
    - **i.** Aconitase; fluoroacetate, ONOO −
    - **ii.** Isocitrate dehydrogenase: DCVC
    - **iii.** Succinate dehydrogenase: malonate, DCVC, PCBD.Cys, 2-bromohydroquinone, 3-nitropropionic acid, and cis-crotonalide fungicides
  - **6.** Depletors of TPP (inhibit TPP-dependent PDH and α-KGHD): ethanol (chronic consumption)
  - **7.** Compounds that deplete CoA
    - **i.** Thiol-reactive electrophiles: 4-(dimethylamino)phenol and *p*-benzoquinone
    - **ii.** Drugs enzymatically conjugated with CoA: salicylic acid and valproic acid
  - **8.** Compounds that deplete NADH
    - **i.** Alloxan and t-butylhydroperoxide
    - **ii.** Activators of PARP: agents causing DNA damage (e.g., MNNG, hydrogen peroxide, and $ONOO^-$)
- **II.** Inhibitors of electron transport
  - **1.** Inhibitors of electron transport complexes
    - **i.** NADH-coenzyme Q reductase (complex I): antimycin-A, MPP + , and paraquat
    - **ii.** Coenzyme Q-cytochrome c reductase (complex III): antimycin-A and myxothiazole
    - **iii.** Cytochrome oxidase (complex IV): cyanide, hydrogen sulfide, azide, formate,
      - •NO, and PH_3
    - **iv.** Multisite inhibitors: dinitroaniline and diphenylether herbicides, and $ONOO^-$
  - **2.** Electron acceptors: CCl_4, doxorubicin, menadione, and MPP +
- **III.** Inhibitors of oxygen delivery to the electron transport chain
  - **1.** Chemicals causing respiratory paralysis: CNS depressants (e.g., opioids) and convulsants
  - **2.** Chemicals impairing pulmonary gas exchange: CO_2, NO_2, phosgene, and perfluoroisobutene
  - **3.** Chemicals inhibiting oxygenation of Hb: carbon monoxide and methemoglobin-forming chemicals
  - **4.** Chemicals causing ischemia: ergot alkaloids and cocaine
- **IV.** Inhibitors of ADP phosphorylation
  - **1.** ATP synthase: oligomycin, cyhexatin, DDT, and chlordecone
  - **2.** Adenine nucleotide translocator: atractyloside, DDT, free fatty acids, and lysophopholipids
  - **3.** Phosphate-transporter: *N*-ethylmaleimide, mersalyl, and *p*-benzoquinone
  - **4.** Chemicals dissipating the mitochondrial membrane potential (uncouplers)

(*Continued*)

TABLE 5.12 (Continued)

- i. Cationophores: pentachlorophenol, dinitrophenol, benzonitrile, thiadiazole herbicides, salicylate, CCCP, cationic amphiphilic drugs (bupivacaine and perhexiline), valinomycin, gramicidin, and calcimycin.
- ii. Chemicals permeabilizing the mitochondrial inner membrane: PCBD-Cys, and chlordecone

5. Multisite inhibitor drugs: phenformin, propofol, and salicylic acid (overdose)

V. Chemicals causing mitochondrial DNA damage

1. Antiviral drugs: zidovudine, zalcitabine, didanosine, and fialuridine
2. Antibiotics: chloramphenicol (overdose) and linezolid
3. Ethanol (chronic consumption)

ATP, Adenosine triphosphate; *CCCP*, carbonyl cyanide m-chlorophenylhydrazone; *CNS*, central nervous system; *CoA*, coenzyme A; *DCE*, 1,1-dichloroethylene; *DCVC*, dichlorovinyl-cysteine; *MNNG*, *N*-methyl-*N*′-nitro-*N*-nitrosoguanide; *MPP* + , l-methyl-4-phenylpyridinium; *NAPBQI*, *N*-acetyl-*p*-benzoquinoneimine; *PARP*, poly(ADP-ribose) polymerase; *PCBD-Cys*, pentacholorobutadienylcysteine; *PDH*, pyruvate dehydrogenase; PH_3, phosphine; *t-BHP*, *t*-butyl hydroperoxide; *TPP*, thiaminee pyrophosphate; *TRPV*, transient receptor potential cation channel of the vanilloid subtype; *α-KGHD*, α-ketoglutarate dehydrogenase.
Modified from Lehman-McKeeman LD, editor. Casarett and Doull's toxicology: the basic science of poisons. New York: McGraw-Hill; 2019.[227]

protein) may take several hours, but after this time period, the cell can produce large amounts of NO. When airway epithelial cells and circulatory endothelial cells produce NO, they contribute to the control of the tone of the smooth muscle in these systems and thus modify airway resistance and blood pressure.[244,255,256] NO production is associated with asthma and airway infections; in both situations, an increased concentration of NO can be measured in the exhaled air.[244,256]

On the other hand, eNOS is continuously expressed in the cells, and upon stimulation of the cell, the formation of NO begins immediately. However, the amounts of NO produced are minute. The nature of NO in cells expressing eNOS is only to act as a messenger molecule, whereas NO has also other functions in cells expressing iNOS. For example, NO has bacteria and cell killing properties in immunological cells, such as phagocytes.[227,255]

NO may induce deleterious effects when airway epithelial or immunological cells are exposed to mineral particles (asbestos, quartz). These particles also stimulate cells to produce NO in large quantities, but pulmonary cells are unable to destroy these particles, and a no-physiologically excess production of NO results, perhaps causing tissue damage due to a reaction of NO with cellular macromolecules.[227,256]

Immunological responses and sensitization

A number of chemical compounds are potent sensitizers that can lead to serious immunological reactions. Immunotoxicology explores interactions between chemical compounds and the immune system. Chemicals can amplify, attenuate, or otherwise modify immunological reactions subsequent to exposure. [227]

The basic function of the immunological system is to detect and destroy foreign material that may be harmful to the organism. Cells that belong to the immunological system include macrophages, monocytes, granulocytes, and T- and B-lymphocytes. All cells that belong to the immune system have differentiated from the same multipotent hematopoietic stem cell. In harmful immunological reactions, the response of an organism to an exposure changes. The environmental factor does not act directly, but alters the reaction of the person exposed. The most important forms of this kind of immunological reactions are (1) immunosuppression; (2) uncontrolled cell growth, for example, leukemia and lymphoma; (3) disturbances of immunological defense mechanisms against infectious agents and malignant cells; (4) allergies; and (5) autoimmunity. Allergies will be dealt with in more detail later in this chapter. [227]

Necrotic and apoptotic cell death

The main types of cellular injury induced by chemical compounds are necrotic and apoptotic (programmed) cell death. Necrosis implies chaotic ending of cellular functions, and it always represents an unwanted effect on the cell by a chemical. Apoptosis is a physiological phenomenon that is required during development of the embryo in shaping the developing organs into their final size and form, and it is also functionally important in the development of organs and even body parts (e.g., fingers and toes). Apoptosis is also important in maintaining the integrity and renewal of mucous membranes and the skin. In direct contrast to necrosis which is a passive, nonenergy-requiring phenomenon, apoptosis requires gene expression and synthesis of new proteins, and it is an energy-expensive process. [227,243]

Necrotic cell death is often due to binding of reactive species to biologically important cellular macromolecules, such as proteins, lipids, and DNA. Biotransformation of a number of chemicals such as carbon tetrachloride or styrene leads to formation of epoxides that bind to nucleophilic sites on proteins and DNA. Many of these compounds are also carcinogens.

Furthermore several compounds also cause increased production of ROS. These phenomena may also damage the cell membrane, leading to its leakage and rupture. Necrosis is characterized by cell swelling and leakage of cell constituents into the surroundings of the cell.

In apoptotic cell death, several factors such as growth factors, NO, the tumor suppressor gene p53, and the protein encoded by this gene contribute to the process that leads to cell death. One of the functions of p53 protein is the activation of apoptosis if a cell is transformed to a malignant cell. Apoptosis typically leads to the formation of smaller membrane-encapsulated particles within the cell. Apoptotic cell death begins in the nucleus and proceeds to other parts of the cell. The death process may be quite advanced before it can be observed from outside the cell. Ultimately, the cellular particles are phagocytized by the surrounding cells without any inflammatory process. This is one of the characteristic morphological differences between necrotic and apoptotic cell death: whereas inflammation is typical for necrosis, lack of inflammation is the hallmark of apoptosis.[227,243] Exposure to chemical compounds such as some heavy metals (e.g., lead) may activate apoptosis in a nonphysiological way, leading to organ injury and reduced functional capacity of the organ. It is noteworthy that effects of various oxidants, such as quinones, can vary as a function of dose: at low doses they may induce cellular proliferation, at moderate doses apoptosis, and at high doses they induce necrosis. Thus again dose is the ultimate determinant of the effect, even when very basic cellular responses such as death or survival are involved.[227,243]

Binding to cellular macromolecules

Many chemical compounds induce their toxic effect by binding to the active site of an enzyme, transporter or to other proteins that are vital for cellular functions. As described earlier, hydrogen sulfide and cyanide bind to the Fe^{3+} of cytochrome oxidase, whereas CO binds to the Fe^{2+} of hemoglobin.[227] Consequently, cyanide prevents a cell from utilizing oxygen even if it would be available and carboxyhemoglobin formation during CO exposure inhibits the access of cells to oxygen and thereby terminates oxidative metabolism inside the cells. Lead, mercury, and cadmium bind to SH-groups of proteins and thereby inhibit their functions.[227] A classic example of fatal enzyme inhibition is the covalent binding of organophosphate insecticides, such as the activated form of parathion, paraoxon, to the acetycholinesterase enzyme. This leads to accumulation of ACh in the CNS, endocrine glands, smooth muscle, and other organs. This, in turn, leads to clinical signs such as breathing difficulties, excessive salivation, tremors, convulsions, and even death. [245] The mechanism of this enzyme inhibition is illustrated in Fig. 5.48.

Covalent binding of chemicals to biological macromolecules can also cause toxicity. During biotransformation and metabolic activation, chemical compounds can be changed to free radicals, which have an unpaired electron. These are extremely reactive, and readily react with cellular lipids, causing lipid peroxidation, where polyunsaturated fatty acids are converted to lipid peroxyradicals that are further changed to lipid hydroxy-peroxides. These are then the source for lipid peroxides. This is a typical chain reaction that continues until it is stopped by antioxidants. If there is a shortage of antioxidants in the cell, for example, due to oxidative stress that has depleted GSH, the end result may be cell death. Thus intracellular thiols, especially GSH, are extremely important in preventing radical-induced cellular injuries. Fig. 5.48 depicts the

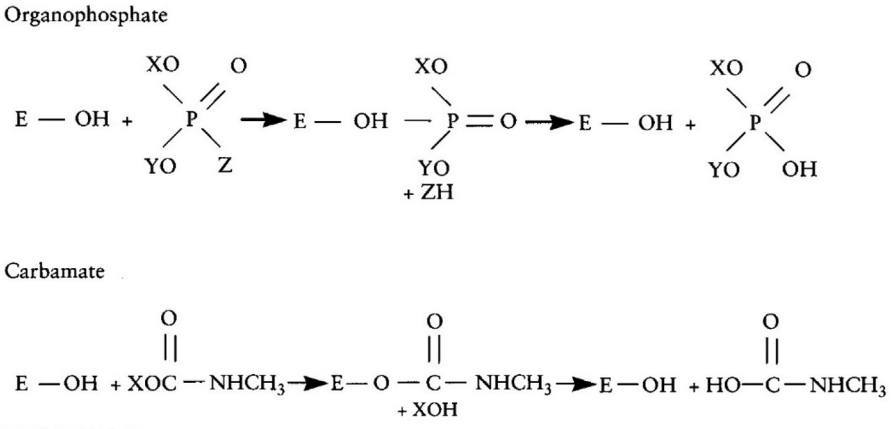


FIGURE 5.48 Interaction of the serine hydroxyl residue in the catalytically active site of acetylcholinesterase enzyme with esters of organophosphates or carbamates. The interaction leads to binding of the chemical with the enzyme, inhibition of the enzyme, inhibition of acetylcholine hydrolysis, and thus accumulation of acetylcholine in the synapses.

role of GSH in the protection of cells against attack by electrophiles, oxidants, and ROS.[249]

Nucleic acids in the DNA contain a high number of nucleophilic sites that can be attacked by electrophilic intermediates (metabolites) of chemical compounds. DNA adducts formed may cause alterations in the expression of a critical gene in the cell and thus lead to cell death. For example, modification of p53 tumor suppressor gene may inactivate the functions of the p53 protein and render cells sensitive to malignant transformation. Also formation of RNA adducts may inhibit key cellular events because RNA is essential for protein synthesis.

In addition, the chemicals can cause alterations in gene expression by affecting, for example, mRNAs and microRNAs. Altered gene expression may also be affected directly at transcription and signal transduction pathways. [227]

Genotoxicity

DNA damage, mutations, chromosomal aberrations, and micronuclei are genotoxic effects of substances. DNA damage such DNA adducts and DNA breaks leads to the induction of DNA repair, mutations, dysfunction, or death of cells. There are endless possibilities for such interactions between DNA and chemical compounds because each human cell contains about 20,000 genes. Genetic mutations lead to an inheritable toxicity only when it occurs in a germ cell that is involved in fertilization and development of a new organism. Genetic mutations in a somatic cell may lead to a deleterious effect in an individual since it can ultimately lead to a toxic end result such as cancer. In addition, epigenetic modification can cause changes to genome without modification of DNA structure.

Most of the compounds that induce alterations in genetic material, that is, genotoxic mutagens, also act as potential inducers of cancer, that is, they are also carcinogens. For this reason, mutagenicity tests are widely used to predict carcinogenicity. Mutagenicity of substances can be studied in bacteria, eukaryotic cells and in vivo in animals. Bacterial mutagenicity tests are also used for biological monitoring of exposed workers by testing mutagenicity of urine. A positive result in a single mutagenic test indicates inherent genotoxic effect of a substance. It alone can never be considered as the indication of carcinogenicity, as several factors affect development of a cancer. A combination of other mutagenicity and genotoxicity tests are performed to clarify potential concern of cancer.[257]

Genotoxic compounds can induce substitution or frameshift mutations. In substitution mutation a base pair is changed. In frameshift mutation a base pair or few base pairs is deleted (deletion) or added to a genetic code (insertion). A gene may be also amplified (amplification). Those mutation, which take place in the coding region of a gene, can cause alteration in the structure of protein or even lead to inactivation of a protein. The result of the frame-shift mutations is more serious as they cause upset of the regular arrangement of the three nucleotide code. This kind of change alters the amino acids throughout the protein because each amino acid has its own code consisting of three nucleotides. As mentioned earlier number of chemical compounds bind to DNA, and may cause point mutations. Ionizing radiation-induced DNA damage typically causes deletions. Table 5.13 lists the principal assays used in genetic toxicology.[227]

Chromosomal aberrations induced by chemicals can be either structural (clastogenic) or numerical (aneugenic) changes. Aneuploidy is an excess or a shortage of a single chromosome. Polyploidy is an excess of a whole set of chromosomes in the cell. Mixoploidy means different sets of chromosomes between cells in the same tissue.

Chromosomal aberrations represent damage to the chromosomal structure that can be detected microscopically when cell cycle is stopped at metaphase by colchemid or colchicine. The most frequent chromosomal aberrations are deletion (lack of a chromosome or its part), duplication (part of a chromosome has been

TABLE 5.13 Principal assays in genetic toxicology.

I.	Pivotal assays
	A. An assay for gene mutations
	Salmonella/mammalian microsome assay (Ames test)
	B. A mammalian assay for chromosome damage in vivo
	Metaphase analysis or micronucleus assay in rodent bone marrow
II.	Other assays offering an extensive database or unique end point
	A. Assays for gene mutations
	E. coli WP2 trytophan reversion assay
	TK or HPRT forward mutation assays in cultured mammalian cells
	Drosophila sex-linked recessive lethal assay
	B. Cytogenetic analysis in cultured Chinese hamster or human cells
	Assays for chromosome aberrations and micronuclei
	Assays for aneuploidy
	C. Other indicators of genetic damage
	Assay for mitotic recombination in yeast and *Drosophila*
	Assay for unscheduled DNA synthesis in cultured hepatocytes and rodents
III.	Mammalian germ cell assays
	Mouse visible or electrophoretic specific-locus tests
	Assays for skeletal and cataract mutations
	Cytogenetic analysis and heritable translocation assays
	DNA damage and repair in rodent germ cells
	Dominant lethal assay

Modified from Klaassen CD.[177]

duplicated), inversion (parts of a chromosome have changed place within that particular chromosome), and translocation (parts of chromosomes have changed their position between two chromosomes). Many of these chromosomal changes are transferred to sister cells when the cell divides, and become, therefore, stable chromosomal aberrations. Cytostatic drugs and cigarette smoke are examples of chemical exposures known to induce chromosomal aberrations. Chromosomal aberrations themselves do not, however, give any clue of the causative agents for the changes.[257]

5.3.4.4 *Target organs*

Organs as targets of chemical compounds

Circulation acts as the transport system for distribution of absorbed substances throughout the body. The distribution is often uneven. Adverse responses may occur if the concentration exceeds a critical concentration in the target organ. As stated previously, the target organ is not necessarily the same as the organ with the largest accumulation of the substance. Many compounds are stored in the skeleton and fatty tissue but critical effects usually occur in other organs. Lipophilic organic materials are deposited in fatty tissue, whereas some inorganic materials accumulate in the bones due to their resemblance to calcium (e.g., lead) or their ability to bind with calcium (e.g., fluoride).[227]

Water-soluble compounds are easily transported in the blood. Nonsoluble compounds are usually transported bound to plasma proteins (albumins). This binding is reversible in most cases but may vary remarkably. The degree of protein binding may vary between 50% and 99%. The proportion of the free (unbound) compound in the circulation is the amount of the compound that can reach the tissues and thus the target organs. Very lipid-soluble compounds are also easily transported in the blood, mainly bound to lipoproteins. They move freely from the circulation to the organs depending on the lipid content of various organs. Thus at equilibrium, organs such as the brain and other lipid-containing organs have the highest concentration of the agent at equilibrium. Typical examples of very lipid-soluble components are aromatic solvents such as benzene, xylenes, toluene, styrene, and ethylbenzene. Also chlorinated hydrocarbons such as tri- and tetrachloroethylene belong to this category.[178,227]

The reactivity of a compound greatly affects its distribution and, therefore, the potential target organs. For example, formaldehyde is a very reactive and irritating gas. Because of its reactivity, inhaled formaldehyde binds with mucus and proteins in the nasal and oral cavities and in the upper respiratory tract, but it does not reach the alveolar region or the systemic circulation through inhalational exposure. For this reason, the most serious health effects of formaldehyde, notably cancer, are only seen in the upper respiratory tract. In fact, a considerable amount of formaldehyde is being formed endogenously in normal metabolism. However, it does not cause any harm under these conditions, because it is tightly bound to serum proteins. Thus harmful reactions of formaldehyde with macromolecules, such as DNA, only occur in very limited areas in the body. Due to its reactivity, formaldehyde also readily forms protein adducts, which in some cases can be used for biomonitoring of formaldehyde exposure.[257]

Toxicity to the central and peripheral nervous systems

The nervous system consists of two main categories of cells: neurons and glial cells. Neurons are the actual nerve cells, which are responsible for transmitting information. There are fewer nerve cells than glial cells present in the brain. Glial cells play a variety of supportive functions. The brain and spinal cord form the CNS. Most parts of CNS are isolated from other parts of the body by the blood–brain barrier, which is a functional rather than a morphological entity that consists of tightly connected cell membranes. Some substances, however, pass through the blood–brain barrier due to their lipophilicity. In addition, there are active transport mechanisms for hydrophilic nutrients and minerals that are vital for CNS function. Some toxic compounds can use these mechanisms to cross the barrier. The remaining parts of the nervous system are called the peripheral nervous system (PNS). PNS can be considered, in fact, as an extension of CNS.[258]

Structural parts of neurons are the cell body, dendrites, axon, and axon terminals. The cell body contains the nucleus and the organelles needed for metabolism, growth, and repair. Dendrites are branched extensions of the cell body membrane. The axon is a long, thin structure which transfers electrical impulses down to the terminals. The axon divides into numerous axon terminals and it is in this specialized region that neurotransmitters are released to transmit information from one neuron to its neighbors. The synapse is the space between two subsequent interrelated neurons.[227]

Glial cells support the neurons physically. Certain glial cells (oligodendroglial cells) synthesize myelin, a fatty insulation layer wrapped around axons. Myelin is necessary for the so-called saltatory conduction of electrical impulses. The myelin layer is not continuous but has breaks called the nodes of Ranvier. Action potentials occur only at those nonprotected nodes where they "jump" (the Latin verb *saltare* means "jump") from one node to the next. The glial cells also have

various maintenance functions, for example, maintaining ionic equilibrium.[259,260]

The nervous system is vulnerable to attacks from several directions. Neurons do not divide, and therefore death of a neuron always causes a permanent loss of a cell. The brain has a high demand for oxygen. Lack of oxygen (hypoxia) rapidly causes brain damage. This manifests itself both on neurons and oligodendroglial cells. Anoxic brain damage may result from acute CO, cyanide, or hydrogen sulfide poisonings. CO may also be formed in situ in the metabolism of dichloromethylene.[227,261]

Organic solvents have acute narcotic effects. Aromatic and chlorinated hydrocarbons seem to be especially effective. The symptoms caused by organic solvents, often called prenarcotic symptoms, resemble those caused by the use of alcohol. A decrease in reaction time and impairment in various psychological performances can be observed. Acute neurotoxicity can also be detected as abnormalities in the electroencephalogram, which records the electrical activity of the brain.[258]

Chronic neurotoxic effects can be divided into four groups. Neuronopathy, where the whole neuron is destroyed, is the most dramatic of them. In axonopathy, the axon partly degenerates. The damage usually begins from a certain site and progresses towards the terminal. One can imagine it as a break in the axon. Carbon disulfide and n-hexane are examples of chemicals causing axonopathy. The toxic effect results in reactions with the amino groups of proteins. In the case of n-hexane, the toxic metabolite (2,5-hexadione) is formed via oxidative metabolism. The reactions cause precipitations of neurofilaments in axons which hinder its transport capabilities. The sensomotoric neuropathy caused by n-hexane exposure appears in extremities as numbness, weakness, and muscle pain. Myelinopathy slows the velocity of nerve conduction. The damage cannot be easily rectified in the CNS. Demyelination may also occur in the PNS. Lead is the most common agent causing myelinopathy. ADD the fourth class: conduction disorders (altered function of neurotransmitters).[227]

Eye toxicity

Vision is vital for human activities, and eyes are very sensitive to a number of toxic insults of chemicals. The most serious outcome is permanent eye damage, which may lead to loss of vision. The eye consists of cornea and conjunctiva, choroid, iris, and ciliary body. It also contains retina, which is of neural origin, and the optic nerve. The retina is a highly specific light-sensitive type of neural tissue that contains photoreceptors. The eye also contains lens and a small cerebrospinal fluid system, the aqueous humor system, that is important for the maintenance of the steady state of hydration of the lens and thus the transparency of the eye.[262]

The cornea must be transparent to allow normal function of the eye. Therefore even a minor scar formation, commonly induced by exposure to acids or alkalies, may seriously damage the visual system. Acid and alkali burns of the eye have to be washed immediately for at least 30 minutes with water.

A special feature of the iris is its autonomic innervation. Sympathetic activation widens the aperture of the iris, whereas impulses from the parasympathetic nerves decrease it. Therefore adrenergic agonists and anticholinergic compounds both increase the aperture of the iris, that is, cause mydriasis, and antiadrenergic and cholinergic agonists decrease it, that is, cause miosis. The iris can thus be considered an excellent mirror reflecting the balance of the autonomic nervous system in the body.[262]

The eye has its own hydraulic system, and disturbances in it may cause serious eye damage. The normal eye pressure is 22 mm Hg, but when the pressure increases to 28–30 mm Hg, the optic nerve is squeezed and becomes hypoxic. This increase in the eye pressure may be due to acids or alkali causing inflammation in the anterior chamber of the eye, blocking the outflow of aqueous humor back into the systemic circulation.

The lens is an avascular transparent tissue surrounded by an elastic, collagenous capsule. Disturbances in the normal metabolism of the lens and rupture of the lens alter its optical characteristics, and may cause cataract, that is, reduced transparency of the lens. For example, exposure to the herbicide 2,4-dichlorophenol may cause cataract.[262]

The retina is the part of the eye that belongs to the nervous system. Rods and cones are the photoreceptors of the retina that synapse with the cells in the bipolar layer in the retina, and these cells, in turn, make connections with ganglion cells. The metabolism of retina is very active and therefore the retina is sensitive to toxic insults. For example, natural retinols that were used in skin therapies provoke retinal damage by replacing the retinoids of the photoreceptors. Hyperbaric oxygen can also cause serious retinal damage in immature newborn children who have had respiratory difficulties.[262]

Methanol intoxication can cause blindness due to damage to ganglion cells in the retina. The blindness results from the accumulation of formaldehyde and formic acid, which are metabolites of methanol. Chemicals can also damage the visual cortex, for example, visual damage was observed among the victims of organic mercury intoxication in Japan who were fishermen of Minamata Bay.[262]

Pulmonary toxicity

The lungs are an important port of entry for toxic compounds into the body and also an important target organ for chemical compounds. Gas exchange is the most important function of the lungs. Oxygen enters the circulation through the lungs, and carbon dioxide and other products of metabolism are exhaled. In addition, the lungs are an important metabolizing organ. Lungs possess a number of nonspecific defense systems, such as sneezing and coughing, active movement of the cilia of the pulmonary epithelial cells and secretion of mucus in the airways. In addition, there are a number of specialized phagocytic cells, such as neutrophils, eosinophils, and macrophages, that destroy foreign particles through phagocytosis, that is, they first engulf the particles and then destroy them by proteolytic enzymes. The immunological system is responsible for providing specific responses against specific antigens.[180,240]

Inhaled gaseous compounds are absorbed in all parts of the respiratory system, whereas particle size determines how deep into the airways the particles will be transported in the airstream. Shortness of breath is a typical sign of a chemical exposure that has affected the lungs, and it may be evoked through immunological mechanisms (e.g., formaldehyde and ethyleneoxide), or through toxic irritation (formaldehyde, isocyanates, sulfur dioxide, nitrogen dioxide, and ozone). Frequently the mechanism depends on the concentration of the compound in the inhaled air. The industrial accident in Bhopal, India, is an example of a poisoning epidemic that caused serious lung injuries. An explosion of a large container led to poisoning of thousands of individuals by methylisocyanate, and subsequently to blindness, serious lung injuries, and deaths in the exposed population.[240]

Acute lung toxicity. Toxic compounds can induce acute deleterious effects in various parts of the airway. Irritating compounds may cause bronchoconstriction within the bronchial tree, edema of its mucous membranes, and increased secretion of mucus. In addition, ciliary activity may decrease in the bronchial and bronchiolar regions and thereby prevent the clearance of mucus and foreign particles from the airway.[223,224,240]

Bronchoconstriction may take place without any cellular injury. For example, low concentrations of sulfur dioxide induce bronchoconstriction. Asthmatics are especially sensitive; a concentration of sulfur dioxide as low as 0.4 ppm may induce bronchoconstriction.[223,224] Cholinergic activation mediated via the vagal nerve is responsible for this effect, because it can be prevented with anticholinergic compounds.[240] An inflammatory reaction may also cause bronchoconstriction. Inflammatory mediators, such as metabolites of arachidonic acid released from the epithelial cells of the airways, may increase the extent of the bronchoconstriction. Epithelial cells also produce relaxing compounds that antagonize bronchoconstriction (e.g., prostaglandin E2), but in inflammation, there is reduced production of these compounds. Also exposure to inorganic particles may induce a dramatic acute inflammation in the lungs, leading to the excretion of a number of bioactive molecules from pulmonary phagocytic cells.

Compounds that induce bronchoconstriction include tobacco smoke, formaldehyde, and diethyl ether. Several other compounds, such as acidic fumes (e.g., sulfuric acid) and gases, such as ozone and nitrogen dioxide, as well as isocyanates, can cause bronchoconstriction. Also cellular damage in the airways induces bronchoconstriction because of the release of vasoactive compounds. Frequently, different mechanisms work at the same time, provoking bronchoconstriction and increased secretion of mucus, both of which interfere with respiration.[223,224] The alveolar surface is predominantly covered by alveolar type I cells. These cells are the primary targets of chemical compounds causing alveolar damage. Typically, alveolar type I cells are replaced by alveolar type II cells subsequent to alveolar damage induced by deep lung irritants (e.g., nitrogen dioxide and ozone).[240] On the other hand, when small particles reach the alveolar region macrophages phagocytize the particles and are then removed from the lungs by the mucociliary escalator in the trachea or by the lymphatic system. Alternatively they may persist in the lungs.[223,224] When macrophages are phagocytizing the particles, they become activated and secrete large amounts of oxygen radicals. While the radicals may have no effect on the particles, they may well damage the surrounding cells and tissues. It has been suggested that the mechanisms by which asbestos particles induce lung cancer and mesothelioma (a fatal cancer type in the pleura) may be associated with excessive production of ROS by specialized phagocytes.[210] An important consequence of alveolar level damage is that it may sensitize the lungs to inflammation. Serious air pollution episodes are associated with increased incidence of lung inflammations, especially in the elderly.

Chronic pulmonary toxicity. Chronic damage to the lungs may be due to several subsequent exposures or due to one large dose that markedly exceeds the capacity of pulmonary defense, clearance, and repair mechanisms. Chronic pulmonary toxicity includes emphysema, chronic bronchitis, asthma, lung fibrosis, and lung cancer. The single most important reason for chronic pulmonary toxicity is tobacco smoke, which induces all types of chronic pulmonary toxicity with the exception of fibrosis.[263]

In developed countries, where the prevalence of chronic obstructive lung diseases has increased rapidly, the finger of suspicion is often pointed at the air quality, especially that in large cities. In emphysema, the walls separating alveoli from each other disappear, and this reduces the surface area for gas exchange. Chronic bronchitis is characterized by persistent cough and increased mucus secretion. In asthma, the lungs become sensitive to bronchoconstriction induced by environmental agents. In addition, asthma also involves inflammation of the airways.[223,224]

Lung cancer. Lung cancer is one of the most common cancers. In many countries, lung cancer is the most common cancer among the male population, and its incidence among females has shown a dramatic and alarming increase. The incidence of lung cancer has always carried a strong association with smoking in the past, that is, the latency period of lung cancer is about 20 years after the beginning of the exposure to tobacco smoke. In addition to tobacco smoke, many chemicals can increase the risk of lung cancer. These include asbestos, radon, nickel, chromium, and beryllium. Asbestos and radon are considered to be the next most important factors after tobacco smoke causing lung cancer. Both also have a synergistic effect with smoking. The incidence of asbestos-related diseases has remained high even though the use of asbestos has dramatically decreased and is now totally banned. This is due to the long latency period of asbestos-induced diseases.[223,224,240] Fig. 5.49 provides epidemiological data on the relationship between smoking and lung cancer.

Cardiovascular toxicity

Several chemical compounds can have an adverse effect on the heart and the vascular system. The effect may first appear as a transient change in the cardiac function. However, prolonged exposure increases the risk of permanent effects. Occasionally, functional effects such as cardiac arrhythmias may even lead to death. Furthermore in many cases the effects of chemicals on the cardiovascular system are secondary; that is, a compound may affect lipid metabolism and thereby amplify atherosclerotic alterations in the circulatory system. Eventually, these changes can lead to heart failure which means that heart is unable to pump sufficiently to maintain blood flow to meet the needs of the body.[267]

Mechanisms of cardiotoxicity

Chemical compounds often affect the cardiac conducting system and thereby change cardiac rhythm and force of contraction. These effects are seen as alterations in the heart rate, excitability, conduction velocity of impulses within the heart, and contractivity. For example, alterations of pH and changes in

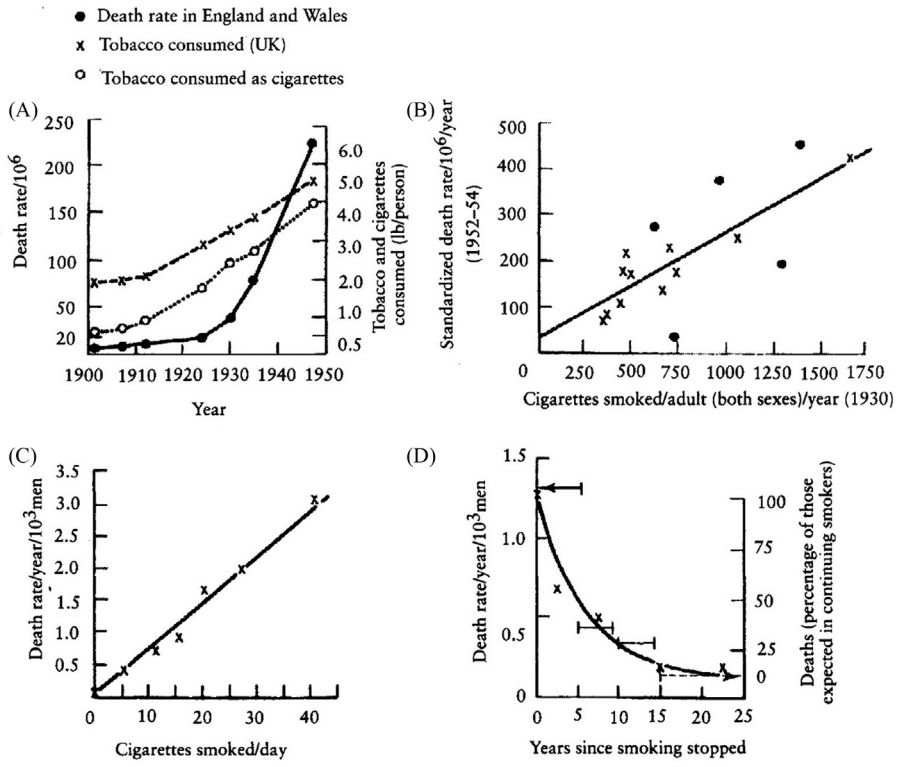


FIGURE 5.49 Epidemiological data defining the relationships between smoking of cigarettes and carcinoma of the lung. (A) Death rate from cancer of the lung and the rate of consumption of tobacco in the UK. The rates are based on three-year averages for all years except 1947. (B) Relationship between lung cancer mortality and previous cigarette consumption in sixteen countries. From left to right the solid dots below the line (lower incidence) are from Japan and United States and above the line (higher incidence) are from the Netherlands, Austria and England/Wales. (C) Death rate from lung cancer, standardized for age among doctors smoking different daily numbers of cigarettes. (D) Death from lung cancer among doctors who had given up smoking cigarettes for different periods. |—| indicates data for <5, 5–9, 10–14, and >15 years since stopping smoking.[227,264–266] Source: *Used with permission*.

ionic balance affect these cardiac functions. In principle, cardiac toxicity can be expressed in three different ways: (1) pharmacological actions become amplified in a nonphysiological way; (2) reactive metabolites of chemical compounds react covalently with vital macromolecules in myocytes (cardiac muscle cells) causing permanent functional and morphological alterations; and (3) the reaction is mediated through immunological mechanisms. Mechanistically, heart is sensitive to interference with ion homeostasis. Cell organelle dysfunction, for example, sarcolemmal injury may cause disturbances in calcium homeostasis, whereas mitochondrial injury affects to energy metabolism. Oxidative stress plays a major role in myocardial apoptosis, necrosis and cell organelle dysfunction. Myocytes have a limited regenerating potential, which emphasizes the importance of myocyte cell death as a cellular mechanism of heart failure. Coronary vasoconstriction due to the atherosclerosis alters coronary blood flow and predisposes heart muscle to ischemic injury.[267]

Mean arterial pressure and cardiac output, an expression of the amount of blood that the heart pumps each minute, are the key indicators of the normal functioning of the cardiovascular system. Mean arterial pressure is strictly controlled, but by changing the cardiac output, a person can adapt, for example, to increased oxygen requirement due to increased workload. Blood flow in vital organs may vary for many reasons, but is usually due to decreased cardiac output. However, there can be very dramatic changes in blood pressure, for example, blood pressure plummets during an anaphylactic allergic reaction. Also cytotoxic chemicals, such as heavy metals, may decrease the blood pressure. In addition, chemicals can elevate blood pressure, for example, increasing vasoconstriction. If prolonged, increased blood pressure can lead to structural changes in heart muscle, that is, cardiac hypertrophy.[267]

Compounds causing cardiovascular toxicity. Alcohols are among the most important compounds causing vascular toxicity. Ethanol causes cardiac dysfunction by attenuating its contractivity when the concentration of ethanol in the blood exceeds 0.75 mg/100 mL. Ethanol also causes arrhythmias and acetaldehyde, a metabolite of ethanol, can disturb mitochondrial oxidative phosphorylation system by formation and accumulation of protein-aldehyde adducts. Furthermore high concentrations of acetaldehyde cause cardiac arrhythmias.[268] Halogenated hydrocarbons depress cardiac contractility, decrease heart rate, and inhibit conductivity. The cardiac toxicity of these compounds is related to the number of halogen atoms; it increases first as the number of halogen atoms increases, but decreases after achieving the maximum toxicity when four halogen atoms are present. Some of these compounds, for example, chloroform, carbon tetrachloride, and trichloroethylene, sensitize the heart to catecholamines (adrenaline and noradrenaline) and thus increase the risk of cardiac arrhythmia.

Some metals, such as cadmium, cobalt, and lead, are selectively cardiotoxic. They decrease contractivity and slow down conduction in the cardiac system. They may also cause morphological alterations, for example, cobalt, which was once used to prevent excessive foam formation in beers, caused cardiomyopathy among heavy beer drinkers. Some of the metals also block ion channels in myocytes. Manganese and nickel block calcium channels, whereas barium is a strong inducer of cardiac arrhythmia. [268]

Several chemicals may cause inflammation or constriction of the blood vessel wall (vasoconstriction). Ergot alkaloids at high doses cause constriction and thickening of the vessel wall. Allylamine may also induce constriction of coronary arteries, thickening of their smooth muscle walls, and a disease state that corresponds to coronary heart disease. The culprit is a toxic reactive metabolite of allylamine, acrolein, that binds covalently to nucleophilic groups of proteins and nucleic acids in the cardiac myocytes.[269]

Atherosclerosis is a degenerative disease of the arteries, which is characterized by cholesterol-containing thickening of arterial walls. Saturated fatty acids, high levels of cholesterol, elevated blood pressure, and elevated serum lipoprotein are well-known risk factors for atherosclerosis. Exposure to some chemicals, such as carbon disulfide (CS_2) and CO may promote the development of the disease.[269]

Liver toxicity

The liver is the most important metabolizing organ in the body. It is largely responsible for the biotransformation of chemicals and drugs to water-soluble forms that can be excreted in urine or bile. The functional unit of the liver is the triangular-shaped acinus, the tip of which is located between the terminal vein and adjacent portal arteries (see Fig. 5.50).[271] Liver damage may cause dramatic changes in the biotransformation of chemicals, and lead to alterations in metabolic pathways. Severe liver damage is characterized by fibrosis and scar formation and the loss of functional capacity of the organ. There are many chemical compounds capable of inducing liver damage.[272]

Yellow phosphorus was the first identified liver toxin. It causes accumulation of lipids in the liver. Several liver toxins such as chloroform, carbon tetrachloride, and bromobenzene have since been identified. The forms of acute liver toxicity are accumulation of lipids in the liver, hepatocellular necrosis, intrahepatic cholestasis, and a disease state that resembles viral

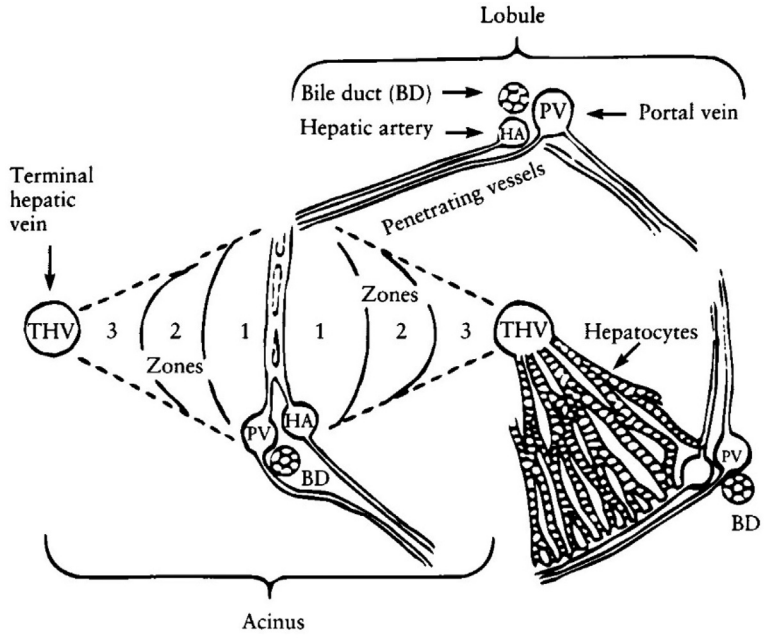


FIGURE 5.50 Schematic of liver operational units: the classic lobule and the acinus.[270]

hepatitis. The types of chronic hepatotoxicity are cirrhosis and liver cancer.

Acute liver damage. Several compounds (e.g., dimethyl nitrosoamine, carbon tetrachloride, and thioacetamide) cause necrosis of hepatocytes by inhibiting protein synthesis at the translational level, that is, by inhibiting the addition of new amino acids into the protein chain being synthetized. This is not, however, the only mechanism. Ethionine is a compound which inhibits protein synthesis but does not induce liver necrosis. Carbon tetrachloride, tetrachloroethylene, and yellow phosphorus induce lipid peroxidation, one common mechanism of liver necrosis. There are, however, a number of compounds (e.g., dimethyl nitrosoamine) that cause liver necrosis without causing lipid peroxidation. Recent findings suggest that neutrophil-mediated cytotoxicity may play a role in some forms of liver toxicity due to inflammatory mediators or ROS excreted by these inflammatory cells.[272]

Accumulation of lipids in the liver (steatosis) is one possible mechanism for liver toxicity. Several compounds causing necrosis of hepatocytes also cause steatosis. There are, however, some doubts that steatosis would be the primary cause of liver injury. Several compounds cause steatosis (e.g., puromycin and cycloheximide) without causing liver injury. Most of the accumulated lipids are triglycerides. In steatosis, the balance between the synthesis and excretion of these lipids has been disturbed (see Table 5.14).[272]

Chronic liver damage. Cirrhosis is one the main forms of chronic liver damage. Formation of a collagen network that destroys the typical liver structure is characteristic of cirrhosis. The underlying mechanism is typically necrosis of individual hepatocytes leading to scar formation. In cirrhosis, the circulation to the liver is severely disturbed because of altered liver morphology. The same compounds that induce liver cancer also induce liver cirrhosis. In humans, the most important compound causing liver cirrhosis is ethyl alcohol.[271,275]Table 5.15 lists chemical compounds that can induce acute liver damage.

Liver cancer can also be a consequence of exposure to hepatotoxic chemicals. Natural hepatocarcinogens include fungal aflatoxins. Synthetic hepatocarcinogens include nitrosoamines, certain chlorinated hydrocarbons, polychlorinated biphenyls (PCBs), chloroform, carbon tetrachloride, dimethylbenzanthracene, and vinyl chloride.[271] Table 5.16 lists the chemical compounds that induce liver cancer or cirrhosis in experimental animals or humans. Humans are exposed to aflatoxins in hot and humid regions in Africa and Asia where peanuts and grain have to be stored in inappropriate conditions which favor the growth of fungi. In these regions, hepatitis is also common, and these two factors act synergistically to promote the formation of liver cancer. Historically, workers involved in the production of polyvinylchloride polymers (plastics and elastomers) were exposed to high concentrations of vinyl chloride. In these workers, the incidence of liver angiosarcoma increased dramatically, and the incidence of brain tumors has also been reported to be higher than the incidence in control workers. Ethyl alcohol can increase the risk of liver carcinoma. It is not primarily considered a very potent liver carcinogen, but nonetheless is important because the doses of ethyl alcohol to which humans are exposed are so high.

TABLE 5.14 Examples of drugs that induce intrahepatic cholestasis or liver damage resembling that induced by viral hepatitis.

Intrahepatic cholestasis	Viral hepatitis-like liver damage
Amitriptyline	Ethacrynic acid
Azathioprine	Halothane
Carbamazepine	Indomethacin
Chlorodiazepoxide	Imipramine
Chlorpromazine	Iproniazid
Chlorthiazide	Carbamazepine
Diazepam	Alphamethyldopa
Erythromycin estholate	Nialamide
Estradiole	Phenylbutazone
Ethacrynic acid	Pyrazinamide
Fluphenazine	Sulphamethoxazole
Haloperidol	Isoniazid
Imipramine	
Mestranole	
17-Methylnortestosterone	
Methyltestosterone	
Nitrofurantoin	
Noretandrolone	
Oxacillin	
Oxandrolone	
Penicillamine	
Perphenazine	
Perchloroperazine	
Promazine	
Thioridazine	
Tolbutamide	

Modified from Savolainen and Vähäkangas.[273]

TABLE 5.15 Chemical compounds that induce chronic liver damage.

Chemical compound	Cirrhosis	Cancer[a]
Natural compounds		
Aflatoxin[b]	x	x
Ethanol[b]	x	?
Pyrrolizidine alkaloids	?	x
Saffrole	?	x
Synthetic compounds		
Anabolic androgens[b]	–	x
Dialkylnitroamines	?	x
Organochlorine pesticides	?	x
Polychlorinated hydrocarbons	?	x
Carbon tetrachloride	x	x
Chloroform	?	x
Vinyl chloride[b]	?	x
Dimethylaminobenzene	?	x
Acetylaminofluorene	?	x
Thioacetamide	?	x
Urethane	?	x
Ethiomine	?	x
Dimethylbenzanthrazene	?	x
Galactosamine	?	x

[a] *In experimental animals.*
[b] *Is also a human carcinogen.*
?, Unknown; –, does not cause any effect.
Modified from Savolainen and Vähäkangas.[273]

Kidney toxicity

The integrity of mammalian kidneys is vital to body homeostasis, because the kidneys play the principal role in the excretion of metabolic wastes and the regulation of extracellular fluid volume, electrolyte balance, and acid–base balance. In addition, the kidney is responsible for the synthesis of a number of hormones that regulate several systemic metabolic events. These include 1,25-dihydroxyvitamin D_3, erythropoietin, renin, and several vasoactive prostanoids and kinins. In addition to these physiologically important functions, the kidneys are also metabolically active organs that contribute to the biotransformation of xenobiotics.

The sensitivity of the kidneys to various toxic insults is due to large blood flow,[274,276,277] ability to concentrate compounds to be excreted in the urine, and to metabolically activate xenobiotics. About 25% of cardiac output continuously flows through the kidneys even though the relative weight of the kidneys is only 0.5% of the human body mass. Due to the key role of the kidney in the excretion of metabolic wastes, it inevitably becomes exposed to high concentrations of metabolic endproducts. The primary urine filtrated in the glomeruli is concentrated about 100-fold before its excretion. The amount of primary urine formed during a 24-hour period is about 100 L, and is being concentrated down to 1 L. Therefore the concentrations of several toxic compounds in the urine may become very high compared to their corresponding concentrations in blood. Furthermore excretory and reabsorptive

TABLE 5.16 Chemical compounds that induce acute liver damage.

Chemical compound	Necrosis	Steatosis
Natural compounds		
Aflatoxin	x	x
Ethanol	–	x
Pyrrolizidine alkaloids	x	x
Saffrole	?	?
Synthetic compounds		
Anabolic androgens	–	–
Dialkylnitroamines	x	x
Organochlorine pesticides	?	?
Polychlorinated hydrocarbons	?	?
Carbon tetrachloride	x	x
Chloroform	x	x
Vinyl chloride	?	?
Dimethylaminobenzene	x	x
Acetylaminofluorene	?	?
Thioacetamide	x	–
Urethane	x	–
Ethiomine	–	x
Dimethylbenzanthrazene	?	?
Galactosamine	x	x

?, Unknown; –, does not cause any effect.
Modified from Savolainen and Vähäkangas.[273]

functions may expose kidney cells to high concentrations of harmful compounds.[274,276,277]

Alterations in the ability of the kidneys to excrete or reabsorb compounds from the proximal and distal tubules are immediately reflected in the amount of extracellular fluid in the mammalian organism. For example, reduced excretion in the glomeruli leads to increased volume of extracellular fluid, and this may contribute to cardiac insufficiency, in which the working capacity of the heart is exceeded. The kidneys are active metabolic organs; this ability has toxicological significance when it leads to the formation of toxic reactive metabolites that damage kidney cells.[277–279]

Mechanisms of kidney toxicity. Direct mechanisms of toxicity include functional and morphological damage at the level of glomerular, tubular, interstitial and vascular cells. Alterations in the levels of free intracellular calcium in kidney cells are important in kidney toxicity caused by several chemical compounds since cell calcium participates in cell activation and the formation of ROS that contribute to hypoxic cell injury.[277] Kidney injury may also be due to an indirect mechanism: long-term hypotension may be a reason for kidney injuries, for example, due to reduced oxygen supply. Immunological mechanisms may also play a role in kidney injuries, and for example, metallothionein, a protein synthesized by the liver to complex heavy metals, may accumulate in the kidneys as a protein–metal complex and cause kidney injury. The primary goal of the protein is to protect the mammalian organism against metal toxicity, but excessive accumulation of the metal–metallothionein complex in the kidneys leads to cellular damage and impaired kidney function, for example, reduced formation of urine. Also accumulation of calcium oxalate, which occurs after exposure to ethylene glycol, the parent compound of the oxalate, may induce kidney injury. In addition, kidney injury may happen due to the necrosis, for example, tubular cell loss.[277–279]

Compounds that cause kidney damage. Several drugs and some anesthetic compounds such as methoxyflurane cause kidney damage when present at high doses. Kidney-toxic compounds found in occupational environments include mycotoxins, halogenated hydrocarbons, several metals, and solvents (see Table 5.17).

Many metals are potent kidney toxins. These metals cause similar signs and symptoms. By concentrating in tubular cells metals inhibit essential metabolic processes. At low doses, the symptoms include leakage of sugars and amino acids into the urine due to glomerular damage and polyuria due to lack of concentrating capability of the kidney. Large doses cause cellular necrosis, anuria, increased concentrations of blood–urea–nitrogen, and subsequently the total breakdown of kidney function and ultimately death. In addition to direct cell injury, some metals induce vasoconstriction in the kidney. Metals such as nickel and cadmium strongly induce the synthesis of metal binding proteins in the liver, notably metallothionein.[277,278]

Halogenated hydrocarbons may cause kidney damage in addition to liver damage. A nephrotoxic dose of carbon tetrachloride increases the relative weight of the kidneys, induces swelling of tubular epithelium in the kidneys, and causes lipid degeneration, tubular casts, and necrosis of the epithelium of the proximal tubulus. Several other halogenated hydrocarbons, for example, tri- and tetrachloroethylene, also induce this kind of kidney damage.[277,278]

Reproductive toxicity

The reproductive system is a very complex, hormonally controlled entity. The female endocrine system is more complex than that of the male, and toxic effects that are directed toward the female reproductive system are, therefore, more difficult to assess than

TABLE 5.17 Nephrotoxic compounds in occupational and general environments.

Mycotoxins
Aflatoxin B[227]
Fumonisin B1[278]
Cirinin[278]
Ochratoxin A[278]
Pyrrolizidine alkaloids[280]
Volatile hydrocarbons
Gasoline[227]
Herbicides and fungicides
Paraquat[278]
Diquat[227]
Succinimides[281]
2,4,5-Trichlorophenoxyacetic acid[282]
Metals
Cadmium[227]
Gold[227]
Lead[227]
Nickel[227]
Mercury[227]
Chromium[227]
Uranium[227]
Organic solvents
Ethylene glycol[278]
Diethylene glycol[278]
Toluene[227]
Halogenated aliphatic hydrocarbons
Bromobenzene[283]
Carbondichloride[284]
Chlorofluoroethene[285]
Dibromoethane[227]
HCBD[227]
TFE[285]
Trifluoroethane[286]
Bromodichloromethane[287]
Chloroform[227]
DBCP[288]
Dichloroethene (dichloroethylene)[227]
Pentachloroethane[227]
Trichloroethene[278]
Other compounds
Benzidine[227]
p-Aminophenol[227]
Maleate[289]

DBCP, Dibromochloropropane; *HCBD*, hexachlorobutadiene; *TFE*, tetrafluoroethene.

those targeted at the male system. In addition, the effects of toxic compounds on the reproductive system clearly differ between pregnant and nonpregnant females, because pregnancy changes female physiology and because the target of the toxic effects may also be the fetus. The effects of chemicals on the fetus will be discussed in the section on teratogenesis (see Section 5.3.4.5). The assessment of reproductive toxicity is further complicated by the fact that the timing of essential events, for example, the process of organogenesis, is different in different species, and therefore extrapolating results obtained in animal experiments to predict the toxic effects of chemicals on human reproduction is problematic.

It needs to be noted that a toxic effect on the reproductive system may be mediated through alterations in normal functions of the CNS, gonads (ovaries and testicles), the hypothalamus–pituitary–gonad axis, or on the pharmacokinetics of reproductive hormones.[290,291]

Compounds affecting reproduction. Compounds that can affect reproductive function include several drugs and occupationally important chemicals, such as solvents and pesticides as well as a number of environmentally relevant compounds. A group of chemical compounds that has received much attention is the chemically diverse group of endocrine disruptors. These are known to induce, for example, feminization in fish and other animal species.[291,292] There is intense debate about the significance of these compounds to human health. Tobacco smoke and ethyl alcohol also have major effects on human reproduction, the effects of alcohol being especially important. Table 5.18 lists compounds that may disturb the functions of female and male reproductive functions.

Toxicity to blood and blood-forming tissues

Blood-forming tissues consist of bone marrow, spleen, lymph nodes and the reticuloendothelial system. These produce the elements of blood and are important for the immunological defense systems.

There are undifferentiated stem cells of the blood elements in the bone marrow that differentiate and

TABLE 5.18 Examples of chemical compounds that affect the reproductive system.

Females	Males
Environmental chemicals	Environmental chemicals
Aniline	Carbon disulfide
Benzene	Chlordecone
Ethylene oxide	Dibromochloropropane
Glycol ethers	Ethylene dibromide
Formaldehyde	Ethylene oxide
Inorganic and organic lead	Glycol ethers
Carbon disulfide	Hexane
Methylmercury	Inorganic and organic lead
Pesticides (occupational exposure)	Pesticides (occupational exposure)
Phthalates	Vinyl chloride
Polychlorinated biphenyls	
Styrene	Drugs
Toluene	Steroids
Vinyl chloride	Cell cycle inhibitors
	Central nervous system drugs
Drugs	• Anesthetic gases
Anesthetic gases	• Levodopa
Steroids	• Opioids
Cell cycle inhibitors	• Tricyclic antidepressants
Opioids	Phenacetin
	Phenytoin
Social poisons	Thiazide diuretics
Tobacco smoke	
Alcohol	Social poisons
	Tobacco smoke
Narcotic drugs	Alcohol
Marijuana	Caffeine
Cocaine	Theobromine
Heroin	Narcotic drugs
	Marijuana
	Cocaine
	Heroin
	Physical factors
	Temperature
	Lightning
	Hypoxia
	Irradiation

Modified from Savolainen and Vähäkangas.[274]

mature into erythrocytes, (red blood cells), thrombocytes (platelets), and white blood cells (leukocytes and lymphocytes). The production of erythrocytes is regulated by erythropoietin (see the section on kidney toxicity) that is synthetized and excreted by the kidney. An increase in the number of premature erythrocytes is an indication of stimulation of erythropoiesis, that is, increased production of erythrocytes in anemia due to continuous bleeding.

Toxic effects on the blood-forming tissues. Reduced formation of erythrocytes and other elements of blood is an indication of damage to the bone marrow. Chemicals toxic to the bone marrow may cause pancytopenia, in which the levels of all elements of blood are reduced. Ionizing radiation, benzene, lindane, chlordane, arsenic, chloramphenicol, trinitrotoluene, gold salts, and phenylbutazone all induce pancytopenia. If the damage to the bone marrow is so severe that the production of blood elements is totally inhibited, the disease state is termed aplastic anemia. In the occupational environment, high concentrations of benzene can cause aplastic anemia.[293]

Platelets are essential as the first line of defense in clot formation to stop bleeding. Platelets gather quickly around the damaged vessel wall and clump together with fibrin filaments to form a clot that prevents bleeding. Platelets also become activated by exposure to adrenaline, thrombin, and collagen. Drugs and chemicals that disturb normal functioning of the bone marrow also decrease the number of circulating platelets, a state termed thrombocytopenia. Vinyl chloride is an example of a chemical which may cause this disturbance.

Specialized phagocytes (i.e., actively phagocytizing cells of the immune system) include granulocytes (neutrophils, eosinophils, and basophils), monocytes, and macrophages, which often originate from circulating monocytes. Many environmental factors may decrease the number of these cells. Ionizing radiation and several drugs may cause granulocytopenia. Lysis of erythrocytes leads to hemolytic anemia, which reduces the capacity of blood to carry oxygen and thereby prevents oxygenation of various tissues, especially the CNS and the heart, organs that are particularly sensitive due to their large oxygen need. Aniline and nitrobenzene cause hemolytic anemia, and several other nitrocompounds also induce this effect. Phenols and propylene glycol are also capable of inducing hemolytic anemia.[293]

Toxicity to the skin

The skin is the largest organ of the human being. In particular, the surface layer of the outer epidermis, the stratum corneum, usually provides quite good protection against chemical compounds. Nevertheless, the

skin is an important entry route for chemicals into the body.

Skin has several protective mechanisms in addition to its thick epidermis that prevent many chemicals from penetrating it. Eccrine (sweat) glands, phagocytic cells, skin metabolism and melanin pigmentation (which protects the skin from ultraviolet irradiation from the sun) belong to the battery of the dermal defense systems. However, skin is potentially exposed to many chemical cs. Skin diseases account for a considerable percentage of all occupational diseases (about 20% in Finland). Among exposure-induced skin diseases, inflammations due to both irritation and sensitization are common.

Assessment of skin exposure continues to be relatively difficult, because it is difficult to measure or estimate the dose actually absorbed by the skin.

Toxic reactions of the skin. Irritation is the most common reaction of the skin. Skin irritation is usually a local inflammatory reaction. The most common skin irritants are solvents, dehydrating, oxidizing or reducing compounds and cosmetic ingredients. Acids and alkalies are common irritants. Irritation reactions can be divided into acute irritation and corrosion. Necrosis of the surface of the skin is typical for corrosion. Acids and alkalies also cause chemical burns. Phenols, organotin compounds, hydrogen fluoride, and yellow phosphorus may cause serious burns.[294]

The common skin reaction allergic contact dermatitis is evoked subsequent to exposure to a chemical via a cell-mediated type IV allergic reaction (see below). Allergic contact dermatitis is also a common skin disease in the occupational environment. The reaction is compound-specific and reexposure to very small amounts of chemical compounds provokes a severe reaction. Skin allergens often have small molecular size and are frequently haptens that become bound to a protein and in that way induce an immunological reaction. Many chemical compounds can induce allergic contact dermatitis (see Table 5.19).[294] Especially important inducers of allergic contact dermatitis are metals (nickel) and metallic compounds (cobalt, chromium, and nickel salts as well as organic mercurial compounds). Also several cosmetic products, resins, a number of colors, rubber (latex) and leather additives, and pesticides (fungicides such as thiurams and dithiocarbamates) are skin allergens. Compounds that belong to the same group of chemical compounds may cross-sensitize sensitive individuals. Thiurams and dithiocarbamates are good examples of this: a person sensitive to one compound in this group is also allergic to all members of this group of chemicals.[296,297] Table 5.20 lists common cross-reacting chemicals.

Light and toxic reactions. In many individuals, exposure to ultraviolet radiation from the sun causes skin

TABLE 5.19 Common contact allergens.

Source	Common allergens	Source
Topical medications/ hygiene products	Antibiotics	Therapeutics
	Bacitracin	Benzocaine
	Neomycin	Fluorouracil
	Polymyxin	Idoxuridine
	Aminoglycosides	α-Tocopherol (vitamin E)
	Sulfonamides	Corticosteroids
	Preservatives	Others
	Benzalkonium chloride	Cinnamic aldehyde
	Formaldehyde	Ethylenediamine
	Formaldehyde releasers	Lanolin
	Quaternium 15	*p*-Phenylenediamine
	Imidazolidinyl urea	Propylene glycol
	Diazolidinyl urea	Benzophenones
	DMDM Hydantoin	Fragrances
	Methylchloroisothiazolone	Thioglycolates
Plants and trees	Abietic acid	Pentadecylcatechols
	Balsam of Peru	Sesquiterpene lactone
	Rosin (colophony)	Tuliposide A
Antiseptics	Chloramine	Glutaraldehyde
	Chlorohexidine	Hexachlorophene
	Chloroxylenol	Thimerosal
(Merthiolate)	Dichlorophene	Mercurials
	Dodecylaminoethyl glycine HCl	Triphenylmethane dyes
Rubber products	Diphenylguanidine	Resorcinol monobenzoate
	Hydroguinone	Benzothiazolesulfenamides
	Mercaptobenzothiazole	Dithiocarbamates
	p-Phenylenediamine	Thiurams
Leather	Formaldehyde	Potassium dichromate
	Glutaraldehyde	
Paper products	Abietic acid	Rosin (colophony)
	Formaldehyde	Triphenyl phosphate
	Nigrosine	Dyes
Glues and bonding agents	Bisphenol A	Epoxy resins
	Epichlorohydrin	*p*-(*t*-Butyl)formaldehyde resin
	Formaldehyde	Toluene sulfonamide resins
	Acrylic monomers	Urea formaldehyde resins
	Cyanoacrylates	
Metals	Chromium	Mercury
	Cobalt	Nickel

Modified from Rice RH, Cohen DE. Toxic responses of the skin. In: Klaassen CD, editor. Casarett and Doull's toxicology: the basic science of poisons. New York: McGraw-Hill; 1996. p. 529–46.[295]

TABLE 5.20 Common cross-reacting chemicals.

Chemical	Cross-reactor
Abietic acid	Pine resin (colophony)
Balsam of Peru	Pine resin, cinnamates, and benzoates
Bisphenol A	Diethylstilbestrol and hydroquinone monobenzyl ether
Canaga oil	Benzyl salicylate
Chlorocresol	Chloroxylenol
Diazolidinyl urea	Imidazolidinyl urea and formaldehyde
Ethylenediamine di-HCl	Aminophylline and piperazine
Formaldehyde	Arylsulfonamide resin and chloroallyl-hexaminium chloride
Hydroquinone	Resorcinol
Methyl hydroxybenzoate	Parabens and hydroquinone monobenzyl ether
p-Aminobenzoic acid	*p*-Aminosalicylic acid and sulfonamide
Phenylenediamine	Parabens and *p*-aminobenzoic acid
Propyl hydroxybenzoate	Hydroquinone monobenzyl ether
Phenol	Resorcinol, cresols, and hydroquinone
Tetramethylthiuram disulfide	Tetraethylthiuram mono- and disulfide

Modified from Rice RH, Cohen DE. Toxic responses of the skin. In: Klaassen CD, editor. Casarett and Doull's toxicology: the basic science of poisons. *New York: McGraw-Hill; 1996. p. 529–46.*[295]

reactions, such as erythema, thickening of the epidermis, and darkening of existing pigment. Exposure to ultraviolet light also increases the risk of different forms of skin cancers, especially malignant melanoma.[294]

Chemical acne. Many chemical compounds induce skin lesions that are similar to acne. Oils, tar, creosote, and several cosmetic products induce chemical acne. These compounds induce keratinization of the sebaceous glands of the skin, obstruction of the glands, and formation of acne. Chloracne is a specific skin lesion that is induced by dioxin like compounds. Chloracne is characterized by hyperplasia of the epithelial cells of the sebaceous glands associated with inflammatory skin changes typical of acne and is slow to heal and difficult to treat.[298]

TCDD is the most potent inducer of chloracne. This has been well known since the accident in Seveso, Italy, in 1976 in which a few kilograms of TCDD were distributed in the environment subsequent to an explosion in a factory producing a chlorophenoxy herbicide, 2,4,5-T. TCDD is an impurity produced during the production of 2,4,5-T, and the most common long-term effect of high dose TCDD exposure was chloracne.[215]

Allergies

Allergies are diseases in which immune responses to antigens, compounds which otherwise would be innocuous, cause inflammation. The immune response occurs in two stages. First, the person becomes sensitized to an antigen. He or she will remain asymptomatic until there is a new exposure, which will provoke an inflammatory response. Hypersensitivity is often used as a synonym for allergy. Allergic disease can be classified according to the immunologic mechanism provoking it. Traditionally, a classification into four types is used, as first presented by Gell and Coombs.[299]

Type I allergies are mediated by immunoglobulin E (IgE). Unlike the other immunoglobulins (G, M, A, and D), which are part of the essential defense mechanisms against foreign proteins, IgE is an antibody type that has virtually only adverse effects. Often allergy is defined to include only type I reactions. The symptoms due to IgE-mediated responses depend on the exposure route. In occupational environments, inhalation is usually the most important route and allergic rhinitis and asthma are common occupational diseases. Atopic dermatitis also belongs to this allergy type. The individual susceptibility for this kind of reaction varies considerably. Inherently sensitive persons are called atopics. The allergens are usually proteins or glycoproteins with molecular weights ranging from 10 to 40 kDa. Common allergen sources include pollen, mites, molds, and animal dander.[300]

Sensitization is a consequence of a complex chain of events which includes presentation of the allergen by antigen-presenting cells to naive (ThO) lymphocytes, which then differentiate into Th2 lymphocytes. These lymphocytes then release a barrage of cytokines (particularly IL-4) that cause B lymphocytes (B cells) to differentiate into specialized plasma cells, which secrete IgE antibodies (cytokines are chemical mediators, small soluble proteins that affect the specific receptors of other cells initiating and maintaining many biological processes). Circulating IgE binds to the receptors on the surfaces of mast cells (located mainly in the mucosal and epithelial tissues).[300]

When exposure is repeated, the allergen binds between two adjacent IgE molecules. This causes release of inflammatory mediators (histamine, leukotrienes, and chemotactic factors). These act locally and cause smooth muscle contraction, increased vascular permeability, mucous gland secretion, and infiltration of inflammatory cells (neutrophils and eosinophils). However, histamine can also be released by non-IgE-mediated mechanisms (e.g., due to exposure to certain fungi).[296,297]

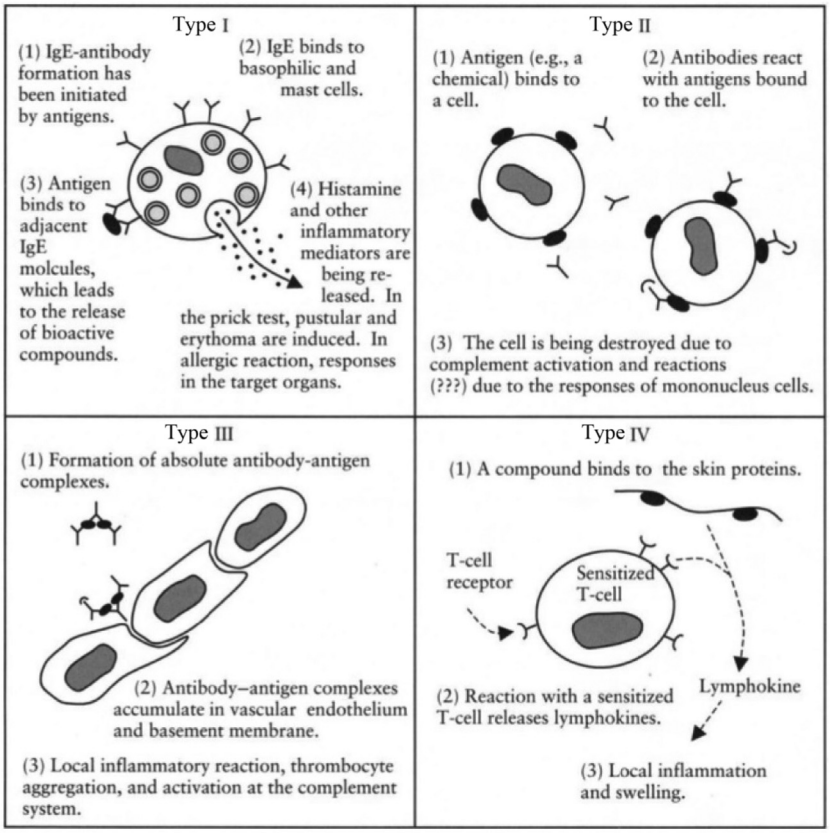


FIGURE 5.51 Four allergic (types I–IV) reaction types based on Coomb's classification. Type I reaction is an immediate allergic reaction. Type II reaction is an antibody-dependent cytotoxic reaction. In a type III reaction, injuries are due to soluble circulating antibody-antigen complexes. Type IV reactions are cell-mediated delayed allergic reactions. In the figure, the characteristics of different allergic reactions have been depicted in more detail.[300]

In addition to the proteins discussed above, a large number of reactive chemicals used in industry can cause asthma and rhinitis. Hypersensitivity pneumonias have also been described. Isocyanates and acid anhydrides are industrial chemicals that cause occupational asthma. Acid anhydrides, such as phthalic anhydride, seem to cause mainly type I reactions, whereas the IgE-mediated mechanism explains only a part of the sensitizations to isocyanates.

Type II reactions include cytotoxic reactions in which the antigen binds to the surface of certain cells (e.g., red blood cells) and B cells then produce IgG antibodies against these cells, which results in cytotoxic injury mediated by complement (a group of blood plasma proteins acting together) activation and an influx of inflammatory cells. For example, some drug allergies are caused by this mechanism. However, this is not an important mechanism in occupational allergies.[300]

In type III or immunocomplex-mediated allergy, IgG antibodies form complexes with antigen. At low exposures, the body is able to remove these complexes, but if there is a severe exposure, immunocomplexes release a variety of proinflammatory cytokines. The involvement of this mechanism is clearest in serum sickness. This mechanism is also considered to be most important in the development of extrinsic allergic alveolitis (hypersensitivity pneumonitis, especially the acute form) in persons having massive bioaerosol exposure. The symptoms include fever, cough, shortness of breath, and malaise. Prolonged exposure can result in lung fibrosis. The disease is common among farmers who handle moldy hay (the syndrome is also called farmers' lung disease). Trimellitic anhydride is an example of a reactive chemical causing a type III response (Fig. 5.51).[300]

Type IV reactions differ from the previous hypersensitivity reactions in that they are not immunoglobulin-mediated, but mediated by T cells. It is probable that this mechanism is also involved in the pathogenesis of extrinsic alveolar alveolitis, especially in the chronic form. Allergic contact dermatitis is the most common example of this allergy type. Allergic contact dermatitis is caused by substances with low molecular weights (below 500 Da). A small molecule (e.g., Ni and Co), called a haptene, cannot act as an allergen alone, but needs to bind to certain proteins (Ia-antigens) on the surface of Langerhans' cells. This combined hapten and Ia-antigen forms the allergen. Langerhans' cells then transfer the allergen to small lymphocytes. This is carried by the lymphatic vessel to the lymph node where it initiates the production of activated T cells (Th1 lymphocytes). When these encounter their antigens, cytokines are secreted (e.g., interferon γ).

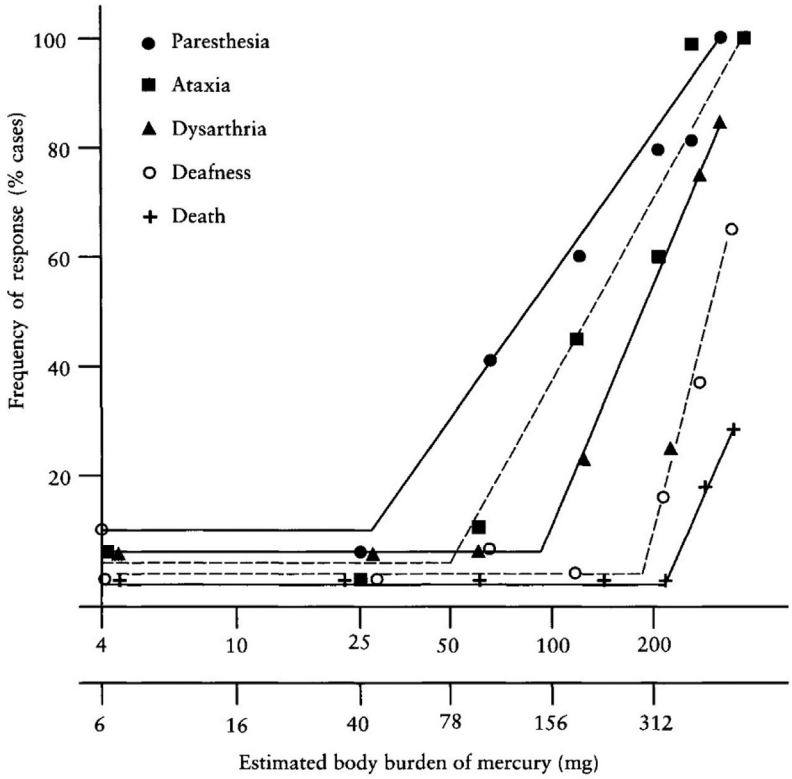


FIGURE 5.52 Dose–response relationships of estimated methylmercury body burden and the frequency of symptoms from the Iraq epidemic methylmercury poisoning in 1970s. Source: *Used with permission.*

These activate the inflammatory cells leading to visible eczema usually within 1–4 days. T memory cells remain viable for a long time (a year or even longer), after which the sensitivity disappears.[296,297] The mechanisms underlying type I–IV allergic reactions have been depicted in a simplified way in Fig. 5.52.

Over 3000 chemicals have been classified as contact allergens. Among them are some substances (so-called "superallergens") that are so potent that they sensitize most exposed persons possibly on the first contact (e.g., dinitrochlorobenzene). In practice, it would be useful to be able to classify contact allergens according to their potency. In the Nordic countries, a classification system for skin-contact allergens resembling the criteria of IARC for the classification of carcinogenic substances has been proposed, but it is not yet widely accepted. Many irritative chemicals may cause nonspecific hyperresponsitivity of the airways and skin. The number of irritating chemicals is very large, several thousands. The symptoms caused by exposure to irritants may resemble allergic symptoms. In addition, exposure to irritating substances (such as sulfur dioxide or solvent vapors) often triggers the symptoms in individuals with allergic asthma.

5.3.4.5 Developmental toxicity

Developmental toxicants are compounds that interfere with development and induce functional or structural alterations. A teratogen is a chemical that induces malformations or permanent damage in the fetus. Brain is especially sensitive to toxic effects of chemicals, because unlike most other organs, which undergo most of their organogenesis during the first trimester of pregnancy, the brain continues to develop throughout the entire pregnancy, and even after birth until early adulthood.[301]

About 2%–3% of children are born with major birth defects, about 14% with minor birth defects and 16%–17% with abnormal neurological function. In about 15%–25% of human birth defects a genetic reason can be found, and it has been estimated that 4% is due to maternal conditions, 3% due to maternal infections a less than 1% due to chemicals and other environmental influences. In about 65%, however, the reason for the developmental defects remains unknown. It should be stressed that it is extremely difficult to identify chemical compounds that cause functional damage, since regardless of the mechanism of the disturbance in development, the timing when it causes the damage is critical. Compounds with very different mechanisms can cause similar functional deficiency in a child.[301]

Mechanisms of chemical teratogenesis

Effects of a teratogen on a fetus depend on timing of exposure, that is, at which stage of organogenesis

the exposure takes place. Exposure to a teratogen before implantation usually leads to death and abortion of the fetus. However, experimental data provide evidence that exposure even at this stage may lead to birth of a malformed offspring. Organogenesis is the period between the pregnancy days 21 and 56 in humans, when most organs are undergoing rapid development. This time is the most sensitive period for a teratogen to exert its effects. For example, the closure of the palate in humans takes place between pregnancy days 56 and 58. This is the time when the risk of cleft palate is the greatest for the fetus if exposure to a teratogen occurs. After the end of organogenesis, morphological malformations are unlikely, but biochemical and functional alterations are still possible.[301]

Several chemical carcinogens are also chemical teratogens. In these cases, both carcinogens and teratogens may have an ultimate common mechanism, DNA damage. In this context, both chemical carcinogens and teratogens may require metabolic activation to be able to react with the nucleic acids in DNA. Like chemical carcinogenesis, chemical teratogenesis constitutes a cascade of complex events, and is rarely induced by a single factor. This is exemplified by the fact that, depending on the dose and timing of exposure, a chemical teratogen may cause death of the fetus, result in growth retardation or induce a malformation. If the dose is high, the fetus dies. If the dose is lower and exposure takes place during an early phase of a critical period, compensatory hyperplasia may replace the dead cells in the damaged organ resulting in growth retardation in a morphologically normal fetus. However, even a small dose of a teratogen may lead to specific malformations when the exposure takes place during a critical period of organogenesis of a given organ. In addition to chemical compounds, ionizing radiation may also cause DNA damage potentially leading to teratogenesis.[301]

In addition to direct effects of chemical compounds on the fetus, metabolic disturbances in the mother, such as diabetes or hyperthermia, or deficiencies of calories or specific nutrients such as vitamin A, zinc, or folic acid may lead to teratogenesis. Compounds that interfere with placental functions may also induce malformations. For example, hydroxyurea disrupts the placental circulation and induces malformations. In addition, it also induces DNA damage.[301]

Teratogens and developmental toxicants

More than 900 teratogens have been identified in experimental animals. However, only about 30 human teratogens have been identified. Human teratogens have been listed in Table 5.21. In this section, some of the best-known teratogenic compounds are briefly described.

TABLE 5.21 Human developmental toxicants.

- Radiation
  - Therapeutic
  - Radioiodine
  - Atomic fallout
- Infections
  - Rubella virus
  - Zika virus
  - Cytomegalovirus
  - Herpes simplex virus I and II
  - Toxoplasmosis
  - Varicella virus
  - Venezuelan equine encephalitis virus
  - Syphilis
- Parvovirus B-19
  - Maternal trauma and metabolic imbalances
  - Alcoholism
  - Amniocentesis, early
  - Chorionic villus sampling (before day 60)
  - Cretinism
  - Diabetes
  - Folic acid deficiency
  - Hyperthermia
  - Phenylketonuria
  - Rheumatic disease and congenital heart block
  - Sjögren syndrome
  - Virilizing tumors
- Drugs and chemicals
  - Androgenic chemicals
  - Angiotensin-converting enzyme inhibitors: captopril and enalapril
  - Angiotensin receptor antagonists: sartans
  - Antibiotics: aminoglycosides, tetracyclines
  - Anticancer drugs: aminopterin, methotrexate, chlorambusil, cyclophosphamide, and busulfan
  - Anticonvulsants: diphenylhydantoin, trimethadione, valproic acid, and carbamazepine
  - Antithyroid drugs: methimazole
  - Carbon monoxide
  - Cocaine
  - Coumarin anticoagulants
  - Cytarabine
  - Diethylstilbesterol
  - Danazol
  - Ergotamine

(*Continued*)

TABLE 5.21 (Continued)

- Ethanol
- Ethylene oxide
- Iodides
- Lithium
- Metals: lead and mercury (organic)
- Methylene blue
- Misoprostal
- Penicillamine
- Polychlorinated biphenyls
- Quinine (high dose)
- Retinoids: isotretinoin, etretinate, and acitretin
- Thalidomide
- Tobacco smoke
- Toluene
- Vitamin A (high dose)

Modified from Rogers JM. Developmental toxicology. In: Klaassen CD, editor. Casarett and Doull's toxicology: the basic science of poisons. *New York: McGraw-Hill; 2019.*

Thalidomide was introduced in 1956 as a sedative which also prevented nausea and vomiting. Since the compound was effective and did not induce addiction, and because its acute side-effects were minor, it became popular to prevent the nausea associated with early pregnancy. Within a few years after its introduction, there was an outbreak of an epidemic of very rare malformations of the extremities, hands, and legs. Typical malformations due to thalidomide were lack of extremities (anamely), a shortening of long bones of the extremities (phocomelia or seal-like limbs), and malformations of the heart, eyes, intestine, external ears, and kidney. Thalidomide was banned in 1961, and within a year no more children were born with its tragic trademark deformities.[302] Had the malformations induced by thalidomide been less spectacular and rare (e.g., if it had induced cleft palate), it would have taken much longer to identify the causal relationship between the use of thalidomide and the malformations. In the case of thalidomide, the causal relationship was clear; about 84% of mothers whose children had limb malformations had taken thalidomide. It is estimated that thalidomide damaged about 7000–10,000 children, mainly in Western European countries.

Another well-known chemical teratogen is methylmercury. Environmental health disasters in Japan, in Minamata Bay and Nigeta in the 1950s, and in Iraq in 1971, have provided detailed information of the effects of methylmercury on fetuses.[303] Exposure to methylmercury during pregnancy affected mainly the CNS of the children, and these changes were permanent. The most important sign was progressive retardation of psychomotor development of a child that seemed to be normal at birth. In addition, the exposure may also have caused blindness, deafness, and convulsions which appeared as the child grew (see Fig. 5.52). Methylmercury seriously disturbs the normal organization of various brain structures during organogenesis. It binds to SH groups of proteins and also disturbs DNA and RNA synthesis. Male fetuses seem to be more sensitive than female fetuses to the effects of this compound.[304]

Fetal alcohol syndrome (FAS) was only defined in 1973, even though harmful effects of ethyl alcohol on the fetus have been known for a long time. Today FAS is considered to represent the adverse end of the whole range of toxic effects of developmental ethanol exposure called fetal alcohol spectrum disorders, and the consumption of ethyl alcohol during pregnancy is not recommended at all. The incidence of FAS has been found in different epidemiological studies to be about 2–7 cases/1000 live births.[305]

FAS is normally characterized by growth retardation, anomalies of the head and face, and psychomotor and intellectual dysfunctions. Excessive consumption of ethyl alcohol may lead to malformations of the heart, extremities, and kidneys. Since consumption of ethyl alcohol has been socially acceptable and prevalent even in pregnant women, the risks associated with the use of ethyl alcohol are remarkable. However, it should be kept in mind that there are several chemicals in the occupational environment that may also cause developmental defects even at low doses. The occupationally important known human developmental toxicants include methylmercury, ethyl alcohol, PCB compounds, tobacco smoke, lead, CO, nitrogen dioxide, gasoline, and fluoride.[301]

5.3.4.6 *Carcinogens and mutagens*

A mutation is a change of DNA sequence in cells and mutagenic chemicals causes them. Mutations occurring in germ cells are inheritable and may lead to genetic diseases. If mutations take place in somatic cells, carcinogenesis may be initiated.

The International Agency for Research on Cancer (IARC) classifies carcinogens into four groups: carcinogenic to humans (group 1), probably carcinogenic to humans (group 2A), possibly carcinogenic to humans (group 2B), not classifiable as to human carcinogenecity (group 3). Before an agent can be classified as a human carcinogen, there must be sufficient epidemiological evidence for a causal association between exposure to this agent and cancer. Probable human carcinogens include agents for which the epidemiological evidence is more limited and/or animal test carcinogenicity evidence is available. A compound is classified as possible human carcinogen when there is

TABLE 5.22 Classification of carcinogenicity of chemicals according to the International Agency on Research on Cancer.

Class		Explanation
1.	Carcinogenic to humans	Enough epidemiological evidence on carcinogenicity in humans
2A.	Probably carcinogenic to humans	Limited evidence on carcinogenicity in humans and sufficient evidence on carcinogenicity in experimental animals and other relevant evidence
2B.	Possibly carcinogenic to humans	Limited evidence on carcinogenicity in humans and other relevant evidence missing; occasionally a compound with insufficient human evidence but limited evidence on carcinogenicity in experimental animals
3.	Not classifiable	Not enough scientifically relevant data available for classification

Modified from Vähäkangas K, Savolainen K. Toksikologian perusteet. Periaatteet ja yleistoksikologia. In: Pelkonen O, Ruskoaho H, editors. Lääketieteellinen Farmakologia ja Toksikologia. Helsinki, Finland: Duodecim; 1998.[308]

limited evidence of animal carcinogenicity. If the animal evidence is inadequate, the agent belongs to the not classifiable group. By the year 2019 IARC had classified the carcinogenicity of 1079 agents (chemical compounds, groups of chemical compounds, and mixtures of chemical compounds). Of these, 120 were placed in group 1. Group 2A includes 82 agents, and 311 compounds were placed in group 2B. The number of chemical compounds belonging to group 3 was 500. These figures reveal some of the difficulties associated with the assessment of the carcinogenicity of chemical compounds: (1) usually only a limited number of studies on the carcinogenicity of chemicals are available; (2) human carcinogenicity is difficult to demonstrate; and (3) experimental or epidemiological evidence of the lack of carcinogenicity is practically impossible to obtain. This is the reason for not having a "not carcinogenic in humans" group in the IARC classification.[306] Some of the difficulties in assessing the carcinogenicity of chemical compounds are discussed below.[307] Table 5.22 lists groups of human carcinogens according to IARC.

Since animals are biological systems which differ from humans both in toxicokinetics and dynamic effects, epidemiological evidence on the carcinogenicity of chemicals is naturally much stronger than that derived from experimental animal studies. However, it is often difficult to obtain conclusive evidence due to several problems which are characteristic of epidemiological studies (see Section 5.3 IPR, pages 3–5). Incidence of different types of cancer demonstrates this challenge well. Rare types of cancer are much easier to detect than those causing more common cancers. Angiosarcoma of the liver and adenocarcinoma of the nose are rare cancers (annual incidences about one per million in the general population); therefore, the human carcinogenicity of vinyl chloride (angiosarcoma) and wood dust (adenocarcinoma of nasal cavity) was identified on the basis of a few cases, whereas increased risk of lung cancer (annual incidence about 400–500 per million) is much more difficult to demonstrate. However, when the evidence derived from experimental animal studies on the carcinogenicity of a given chemical is utilized in assessing human risks of chemical carcinogenesis, several new difficulties are encountered.[307,309–311]

The biotransformation of a given chemical compound in experimental animals and in humans usually differs. Furthermore high doses of chemical compounds are used in testing their carcinogenecity with experimental animals, and this may cause alterations in biotransformation of the tested chemicals compared at the lower doses relevant to the human exposure situation. For example, a metabolic pathway dominating at low doses may become saturated, and a new metabolic pathway takes place. It may produce reactive intermediates of the compound. Since this intermediate would never be produced minimally or not at all at the exposure levels encountered in humans, the overall result of such a carcinogenicity test is indicating unlikely carcinogenic hazard and risk in human. It has also been argued that high doses of chemicals used in animal carcinogenicity bioassays induce mitogenesis (increased rate of cell division), and thus carcinogenesis, and are therefore not specific to the compound itself.[311,312]

Mechanisms of chemical carcinogenesis

Carcinogens can be divided into two broad classes based on their mechanism of chemical carcinogenicity: genotoxic and nongenotoxic (epigenetic) carcinogens. Genotoxic carcinogens initiate the process of chemical carcinogenesis by damaging DNA and acting as mutagens. Nongenotoxic carcinogens promote carcinogensis without binding, damaging of interacting with DNA. They act by causing cytotoxicity, binding to receptors such as estrogen, androgen, aryl hydrocarbon, perosisome or constitutive active receptors, suppressing immune system, increasing oxidative stress, or inhibiting DNA damage repair. They cannot initiate the

development of carcinogenesis. Chemical carcinogenesis is a very[313] complex cascade of events of initiation, promotion and progression of tumor development. Initiation takes place when some dividing somatic cell, for example, epithelial cell is mutated to and initiator (stem) cell of cancer. This cell divide (proliferate) and new cells accumulate cancer promoting mutations so that genotypically different types of cancer cell populations are formed in benign tumor. In bad scenario benign tumor will further accumulate mutations and will be transformed to malignant tumor during progression stage. Typical for this stage is the breaking and penetration of a basement membrane of a tissue. Mutation creates typical hallmarks of cancer. They are sustaining cell proliferation, resisting apoptosis, inducing angiogenesis, enabling immortality, activating invasion and metastasis, evading growth suppression and immune destruction, reprogramming of energy metabolism, genomic instability, and promoting inflammation. Carcinogenesis from initiation to tumor takes years and it is difficult if not impossible to find the point at which one step is over and the next one begins.[307,309]

Experimental animal studies have played a key role in the understanding of the mechanisms of chemical carcinogenesis. The duration of development of a cancer in humans may be several decades, and the development probably includes several steps. Furthermore individual susceptibility is also important for the disease, as there are many mechanisms opposing the carcinogenesis such as error free DNA repairs and immune defense. Development of tumor is silent and usually warning symptoms are observed in the late stage. Therefore it has been extremely difficult to make the required observations in exposed individuals.

Most genotoxic carcinogens require metabolic activation to electrophilic metabolites, which can react with nucleophilic macromolecules such as DNA. Electrophilic metabolites can be trapped efficiently and inactivated to GSH conjugates by GSH S-transferases. They are mostly formed in CYP enzymes catalyzed reactions from certain types of structures, for example, from aromatic ring and double bonds are oxidized to epoxides and nitrosamines to carbonyl radicals. CYP1A2 is involved in the activation of polycyclic aromatic hydrocarbons and of aromatic amines, and CYP3A in the activation of aflatoxins.[227] Activated metabolites are reactive, mutagenic and carcinogenic. They can bind and damage the genetic material, that is, form carcinogenecity promoting DNA adducts and mutations in the genes. Typically, activated compounds favor guanine as the base to which they bind. In addition to the balance between the activity of enzymes that activate chemicals and the activity of enzymes that inactivate reactive metabolites, several DNA damage repair systems are important factors in the initial stages of chemical carcinogenesis induced by genotoxic carcinogens.[314–316]

If enzymes responsible for DNA repair are unable to remove the DNA adduct, or if an error takes place in the repair, then the mutation in the genetic code remains when the cell divides. Mutation is irreversible toxic effect, if the cell survives. Thus cellular proliferation is necessary, in addition to a mutation, for a permanent effect of a chemical compound. Accumulation of mutations is the key factor in chemical carcinogenesis.[312,315]

The number of genes that are important in chemical carcinogenesis has been identified. List of somatic mutations of cancer are listed in the homepage of COSMIC. (https://cancer.sanger.ac.uk/cosmic/signatures) These include oncogenes (genes that promote carcinogenesis) and tumor suppressor genes (genes that prevent carcinogenesis). A mutation of a proto-oncogene may be required for the transformation of a proto-oncogene to an oncogene, which then increase cell proliferation and amplifies the carcinogenic process. On the other hand, mutations that inactivate tumor suppressor genes remove the brake against carcinogenesis, and also amplify carcinogenesis. Both types of mutations can be increased by exposure to chemical compounds.[317,318] For example, mutations of ras-, raf-, jun-, fur-, and myc-oncogenes are known to be crucial in the development of lung cancer.[319] Table 5.23 lists important oncogenes and tumor suppressor genes that may be involved in human carcinogenesis.

Tumor suppressor genes have also been identified. The most important of these are the p53 tumor suppressor gene and the retinoblastoma gene.[320] When functioning normally, the p53 tumor suppressor gene will stop cell division after DNA damage to give the cell time to repair the damage or in large DNA damage guide cell to the apoptotic cell death. Inactivating mutations in the p53 tumor suppressor gene may, therefore, contribute carcinogenesis by preventing the cell from repairing damage to its genetic material. In fact, mutations of p53 tumor suppressor gene are the most usual genetic changes in human cancers, and it seems that some chemical carcinogens induce typical and very specific mutations in the p53 tumor suppressor gene. One example is the aflatoxin-induced mutation in codon 249 in the p53 gene.[321] In contrast, benzo (a)pyrene, present in tobacco smoke, does not bind to this codon, but does bind to other areas of the gene, so-called hot spots. Exposure to UV light also seems to induce typical and specific mutations in the p53 gene. In addition, there are other typical mutations of the p53 gene that seem to be associated with cancer that are induced by environmental or occupational chemical carcinogens.[320]

TABLE 5.23 Important oncogenes and tumor suppressor genes in human cancers.

	Tissues associated with the cancer
Oncogene	
ras	Lung, colon, and pancreas
raf	Lung
jun	Lung
erb-B2(neu)	Breast and lung
fur	Lung
myb	Lung
myc	Bone marrow (acute leukemia)
	Lymphatic tissue (Burkitt lymphoma) and lung
	Nervous tissue (neuroblastoma)
abl	Bone marrow
Tumor suppressor gene	
Rb	Retina (retinoblastoma) and lung
p53	Lung, urinary bladder, intestine, and breast
WT-1	Kidney (Wilms' tumor)
APC	Colon
DCC	Colon
BRCA	Breast
HNPCC	Colon

Modified from Vähäkangas K, Savolainen K. Toksikologian perusteet. Periaatteet ja yleistoksikologia. In: Pelkonen O, Ruskoaho H, editors. Lääketieteellinen Farmakologia ja Toksikologia. *Helsinki, Finland: Duodecim; 1998.*[308]

Transplacental carcinogenesis

Transplacental carcinogenesis indicates that exposure of the mother during pregnancy may induce cancer in the child as it grows. In animals, more than 50 transplacental carcinogens have been found, but in humans only one such compound has been identified, diethylstilbesterol, a synthetic estrogen that was used to prevent spontaneous abortions. However, there is data to suggest that several chemical compounds that are important in the occupational environment may also mediate their effect transplacentally. Such compounds include polycyclic aromatic hydrocarbons, nitrosoamines, hydrazines, and isoniazide. Thus exposure to these compounds should be strictly controlled due to the potential hazard they pose to the developing fetus.[227]

5.3.5 Exposure assessment

Workers' exposure levels can be estimated either by occupational hygiene sampling or by biological monitoring. Since inhalation is usually the most important exposure route, occupational hygiene surveys generally include the measurements of airborne concentrations of many impurities in workroom air. However, dermal exposure is also important for many substances. It can be assessed by analyzing hand-wash and patch samples. In biological monitoring, the concentration of a substance or its metabolite is determined from biological samples. Urine, blood, and exhaled air are the most common biological samples. Furthermore molecular dosimetry, or target-dose monitoring, usually based on the analysis of DNA or protein adducts in lymphocytes or hemoglobin adducts in erythrocytes in exposed individuals, has become popular and holds great promise in the assessment of the association between exposure and the effects of carcinogens.

5.3.5.1 Determination of airborne concentrations

Major time variation is typical for occupational inhalation exposure. It is not unusual if a worker's daily average exposure levels vary by a factor of ten within a single week. The concentration distribution is usually close to lognormal (the logarithms of concentrations are distributed normally). In fact, the distribution may be slightly skewed so that its right side is less steep than its left. The concentration distributions can be characterized by their geometric mean (m_g) and geometric standard deviation (s_g). However, the geometric mean should never be used to describe exposure because the exposure dose depends on the arithmetic mean. The geometric standard deviation is typically 1.5–2.5. In industries with continuous processes, s_g may be lower (1.1–1.5), whereas s_g may exceed 2.5 in some manual occupations. The lognormal distribution becomes a straight line on logarithmic probability paper. The concentration corresponding to the probability of 50% is m_g (also the median) and s_g is obtained from the ratio $c_{50}/c_{15.9}$ or ($c_{84.1}/c_{50}$) as shown in Fig. 5.53.[322]

A European standard (EN 689/95) has been set for occupational exposure assessment. However, this is primarily intended to be used to guarantee that the concentrations of air impurities are in compliance with OELs. According to the standard, exposures exceeding 10% of OEL level should be followed with repeated measurements, the interval of which depends on the concentration observed. The interval decreases as the concentration approaches the OEL.[323] The standard also includes the concept that workers should be divided into homogeneous exposure groups (HEGs). These consist of workers who have similar jobs and are exposed to the same agents. This is practical because it would be unnecessarily laborious to investigate every worker. On the other hand, the prerequisite of the standard that the exposure levels of the

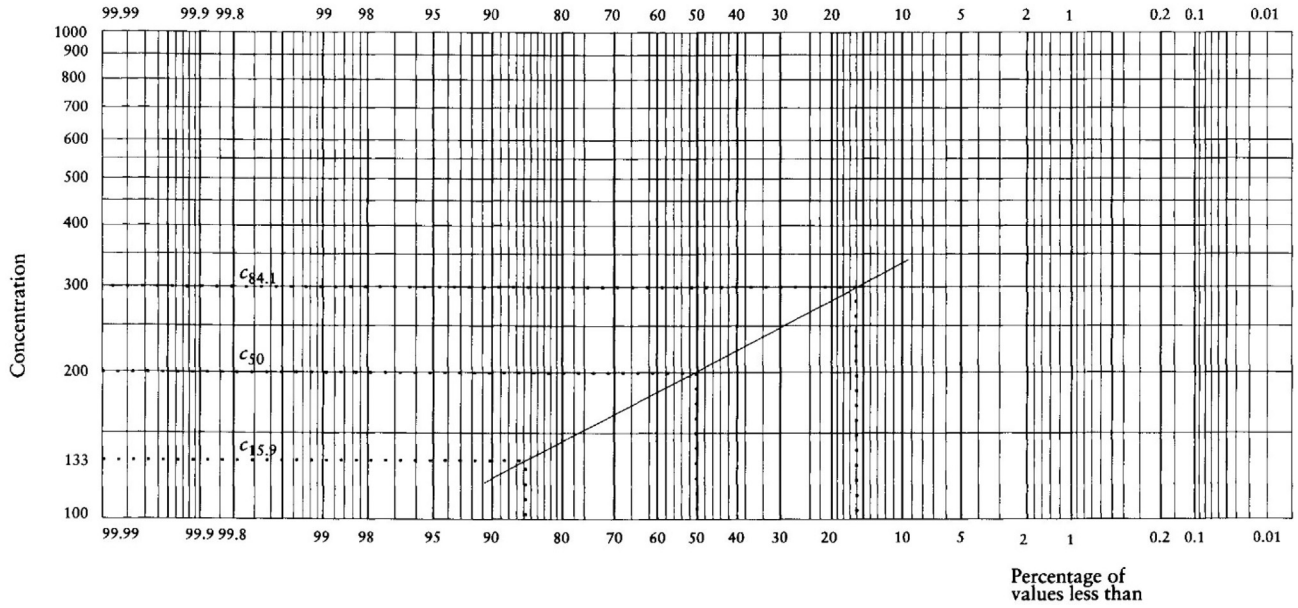


FIGURE 5.53 The determination of the geometric mean concentration and geometric standard deviation. In the sample: $m_g = c_{50} = 200$ ppm; $s_g = c_{50}/c_{15.9} = 200/133$ ppm $= 1.5 = c_{84.1}/c_{50} = 300/200$ ppm.[322]

members of a HEG remain within the range 0.5–2 times the mean exposure level is impractically tight. In addition, airborne concentrations usually fluctuate greatly with time. The within- and between-worker components of exposure variability can be calculated by using the random-effects analysis of variance.[324] However, this would require extensive sampling. Even though repeated random personal sampling is, in principle, the most accurate method for exposure assessment, it has the serious limitation that it does not provide information on the reasons for the exposure. Without this basic knowledge, it may be difficult to institute effective remedial measures.[216]

It is appropriate to consider the differences between manual tasks and process industries (see Section 5.3.2.1) while assessing the exposure, and to perform air sampling so that it also can support planning of engineering control. Because of steep concentration gradients, breathing zone sampling must be performed when investigating manual tasks. A worker often performs several tasks, and the exposure may be very different during different tasks. Therefore all major tasks done by the worker should be studied under various conditions. If the position of the local exhaust is not fixed, its influence should also be examined. The time-weighted average (TWA) concentration is obtained using the lengths of various tasks as the weights. It is common practice to determine the TWA of a working day (shift). Since the health effects usually depend on long-term average exposure level, this should also be estimated. Past exposures are often very difficult to assess because working conditions and methods may have been changed. However, the present (e.g., annual) average exposure level can be estimated by asking the worker how much time he/she spends on average (e.g., during the past year) for various tasks and use these as weights. For example, if we want to assess a construction painter's exposure to organic solvents, we must first list all tasks in which solvent-based paints are used. The exposure during painting depends mainly on the size of the surface painted (or on paint consumption rate), the room volume, and the ventilation. Since local exhausts cannot be used generally, the ventilation may depend on the possibility of keeping the doors and windows open. Breathing zone samples are collected during painting of doors, window frames, floors, walls, etc. in rooms of different size (e.g., small, medium, and large), both with doors and windows open and with them closed. The time use distribution can be obtained with a questionnaire.

In process industries, the areal distribution of airborne pollutant concentrations becomes important. Thus workers' exposure levels depend on their movement patterns during the working day. Ideally, the processes are closed, but, in practice, in-plant emissions occur from openings needed for material flows and sampling. Sometimes, in-plant emissions are intentionally allowed to be discharged into workroom air in areas where workers do not spend any time.

In addition, fugitive emissions commonly take place due to leaking seals in flanges, valves, pumps, and fans. For continuous processes, the time variation of airborne concentration often depends predominantly on relatively few process parameters, such as production rate, temperature, and pressure. These are also important for batch processes, but there are usually certain process phases during which the emissions are heaviest. Batch processes generally also include several manual tasks, such as emptying sacks and barrels.

Since the concentration gradients are not very steep at the actual working areas, it is more convenient to use stationary monitoring instead of personal sampling, and ask how much time, on average, each worker spends in various areas. Direct reading instruments provided with a multipoint sampling system are especially useful because they permit long-term concentration follow-up without excessive costs. Even though accurate information on time use cannot be obtained with questionnaires or interviews, and the coverage of stationary sampling points remains incomplete, the error due to these inadequacies is, nevertheless, usually much smaller than that caused by too brief a sampling time in personal monitoring. In addition, relationships between process parameters and airborne concentrations may be identified. This allows the assessment of long-term exposure because long-term statistics of the important process parameters are usually available. In industries using batch processes, the concentration variation during various process phases should also be taken into consideration. Fig. 5.54 shows the linear relationship between airborne toluene concentration and toluene concentration observed at stationary sampling sites in a printing plant. The annual average concentration is now obtained for each monitoring site simply from the point on the line corresponding to the average use of toluene during the year.[325]

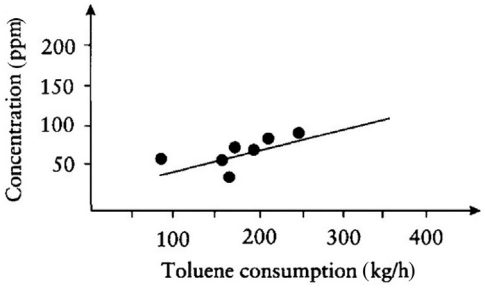


FIGURE 5.54 Relationship between the concentration of toluene in front of a gravure press and the consumption of toluene.[325]

5.3.5.2 *Biological monitoring*

While occupational hygiene measurements always measure only the concentrations of chemical compounds present in the occupational environment, that is, the potential dose, the analysis of biological specimens predominantly reflects the body burden. Furthermore biological monitoring is always limited to assessment of individual exposure. Personal occupational hygiene sampling takes into consideration only some of the individual factors, for example, working habits and height, which can affect exposure. In biological monitoring, factors such as physical activity, that is, cardiac output and minute volume of ventilation, metabolism, and the mass of depot tissues (e.g., adipose tissue) may also be considered.[230,231] Fig. 5.55 depicts the difference between occupational hygiene and biological monitoring.

Biological monitoring provides integrated information on exposure via all routes, including dermal and oral routes.[216] It also includes exposure that takes place outside the workplace. These are benefits in individual risk assessment; on the other hand, they can also be considered disadvantages in occupational health because its aim is to provide safe working conditions for everybody, irrespective of individual characteristics. Biological monitoring can also be used to ascertain effectiveness of personal protective equipment. It also has inherent benefits for substances with long half-lives. The accumulation of substances with very long biological half-lives, such as cadmium, is suitable for biological monitoring because a single sample can provide valuable information provided that a steady-state situation in the body has been reached. In addition, the variation of exposure with time will be attenuated for biological indicators with long half-lives. Therefore fewer biological monitoring samples are needed for long-term exposure assessment than with conventional occupational hygiene monitoring. However, even this advantage is occasionally negated by the large individual variability typical of biological indicators.

Biological monitoring has several other limitations, in addition to those presented above. Biological monitoring is not suitable for agents which do not need to enter blood, such as irritating gases and many dusts. Neither is it very useful for substances with high acute toxicity (in fact occupational hygiene surveys are not very practical in such cases, but the working area should be provided with some kind of continuous monitoring equipped with an automatic alarm system). Another limitation is the small number of compounds for which there are biological exposure limits or indices (BEI) compared to those for OELs (only c. 10%). However, it should be noted that biological monitoring of exposure to a certain agent is often useful even if no

BEI has been established for it. Biological monitoring is especially beneficial for substances with significant skin penetration. Urine sampling may well represent the most convenient means for exposure trend analysis.[326] Blood sampling may be slightly more difficult due to the analytical procedures and unpleasantness of blood sampling. The main limitation is, however, that biological monitoring as such does not provide any information on the causes of exposure. New technologies have become available in which cell samples can be collected, for example, from the oral cavity, and possible protein or DNA adducts (reaction products between a reactive compound and proteins or DNA) can be quantitated, for example, with high-pressure-liquid chromatography. Examples of such compounds are formaldehyde and isocyanates.

5.3.5.3 *Biomarkers*

Extensive research is currently underway to use biological markers (biomarkers) in exposure and risk assessment. Biomarkers include the reaction products of chemicals or their metabolic products with biological macromolecules, especially with DNA. They also involve indicators of effect, such as chromosomal damage, and indicators of individual genetic susceptibility.

Formation of DNA adducts has been demonstrated for many carcinogens. DNA bases are nucleophilic and react with electrophilic compounds. Guanine seems to be especially reactive. Several studies have described how adduct formation can increase with exposure. However, the individual variability is larger than with conventional biological monitoring. Very high interindividual variation has been observed with compounds that require metabolic activation (e.g., polycyclic aromatic compounds). Even though the formation of the adducts is an expression of an interaction of a carcinogen with DNA, the significance of these adducts in chemical carcinogenesis is not yet known. DNA repair and cell proliferation mechanisms remove damage caused by adducts. Peripheral white blood cells are often used in DNA adduct studies; T cells are especially popular because they are long-lived (half-life is about 3 years) and therefore they do not solely reflect current exposure. Peripheral white blood cells have also been frequently used for studies of chromosomal changes. Individuals who have high enzyme activity for formation of reactive metabolites and/or abnormally low metabolic activity of detoxifying enzymes are probably especially susceptible to toxicity.[327–329] The use of biomarkers in biomonitoring is likely to provide a valuable tool for this purpose in the future. This technology can also be used for molecular dosimetry, or target-dose monitoring, in exposed individuals. The goal is to assess the dose at a critical organ or site, such as DNA or a protein.[317,328,330] Fig. 5.55 depicts some essential features and prerequisites of biomonitoring.[227] Table 5.24 indicates the main purposes of biological monitoring of exposure to chemical compounds in the workplace.

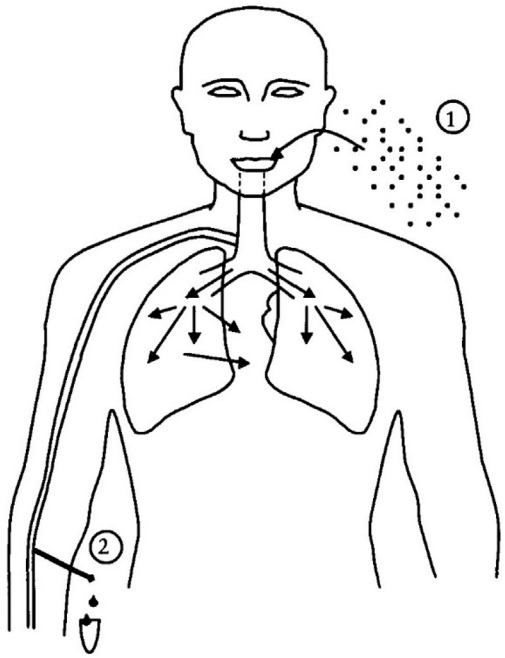


FIGURE 5.55 The idea of biomonitoring compared to the concept of occupational/environmental hygienic monitoring. Hygienic monitoring (1) means measurement of concentration of a compound or a factor (e.g., fungal spores) outside the organism, for example, air monitoring. Biomonitoring (2) means measurement of a compound or its metabolites within the organism, for example, in the blood, urine, or exhaled air; measurement of binding products in the blood or urine or assessment of an existing effect such as chromosomal or DNA damage in white blood cells.[227]

TABLE 5.24 Biomonitoring serves three different purposes of identifying and using.

1. Biomarkers for susceptibility of an individual within a population of one species to exposure to an intoxicant—genetically determined susceptibility.
2. Biomarkers for internal dose of the intoxicant—dose monitoring.
3. Biomarkers for early biological changes following exposure—effect monitoring.

Modified from Aldridge.[331]

5.3.6 Toxicity, risks, and risk assessment

Earlier in this chapter, a short introduction to risk assessment and the concept of risk was given (see Section 5.3.1.5). In this context, the same issues will not be repeated. However, the risk assessment concepts and methodologies will be discussed in more depth

after the reader has received more insight into the role of toxicology in risk assessment, and after many of the principles of risk assessment, such as dose–response relationship, have been clarified. It is still worth emphasizing that the concept of risk is utilized to indicate hazards in the traffic, sports, health care, and even in the monetary markets, not to mention in relation to energy production, for example, nuclear power and its utilization. Toxicology has taken advantage of the concept of risk because it so neatly crystallizes the key issues of toxicology, prevention of chemical and other health hazards, and guaranteeing safety to humans.[332]

The term risk implies the probability that a certain deleterious health effect will take place under defined circumstances, and with regard to chemicals risk is the product of hazardous property and exposure. Likewise, the term security implies the probability that no such deleterious incident will take place under defined circumstances. This kind of definition of risk or security has its foundation in an experimental settings. However, humans or wild animals do not live under defined conditions, but rather face a variety of challenges each day. Therefore reliable risk assessment is an extremely difficult and tedious undertaking. One of the most challenging issues of toxicology has been assessment of carcinogenic risks. In the first phase, we shall assess, on the basis of weight of evidence, whether a chemical is a carcinogen or not. This estimation is followed by another, even more demanding task with the goal of estimating the magnitude of the risk of humans exposed to a given chemical in an occupational setting or in their general environment. The outcome of such an assessment should be an estimate of the actual number of additional cases of cancer among exposed persons. This risk assessment utilizes data from experimental and epidemiological studies as well as all available information on human exposure under different occupational and other living conditions.[332]

5.3.6.1 *Phases of risk assessment*

Risk assessment is usually divided into four different and well defined phases to ensure that all important issues will be given a fair consideration. The phases of systematic risk assessment include:

1. hazard identification,
2. hazard characterization and delineation of dose–response relationships,
3. exposure assessment, and
4. Risk characterization.

In hazard identification, step one, the potential of the chemical to induce adverse health effects, such as acute toxicity, organ toxicity, skin irritation, sensitization, genotoxicity, and developmental toxicity is assessed. In the past hazardous properties of new chemicals were typically observed only after excessive occupational exposures or chemical accidents. Hazard identification is a preventive procedure based on safety evaluation studies conducted before a chemical compound or product reaches the market, and before humans are exposed to it.[332] In addition to animal experiments in vitro methods and even computer based in silico predictions are increasingly used for hazard identification.

Hazard characterization, step two, utilizes data from in vivo safety evaluation studies for establishing dose–response relationships of critical toxic effects that will be used for estimation of safe human exposure levels. In vitro methods are not currently suitable for human dose extrapolations, and there are no regulatory guidance values based on in vitro studies. Safety evaluation studies used for risk assessment of chemicals have to be carried out according to internationally accepted guidelines, such as the OECD Guidelines for Testing of Chemicals. Technical quality of the studies is regulated in detail by good laboratory practice guidelines. In Table 5.25, the safety evaluation studies utilizing experimental animals or in vitro test systems required for marketing authorization of industrial chemicals, drugs, pesticides, and food additives are listed.[332]

In exposure assessment, step three, the extent, duration and other characteristics of exposure are defined together with identification of possible high exposure populations and other special groups. This is a critical step of risk assessment as there is no toxic risk without exposure. Typical routes of exposure are inhalation, oral and dermal routes. In occupational environments widely used methods for exposure assessment include modeling based on exposure scenarios, chemical analyses for estimation of inhalation, dermal and oral exposures as well as biomonitoring of workers.

In risk characterization, step four, includes qualitative and quantitative risk assessment based on steps 1–3. The human exposure situation is compared to the toxicity data from animal studies, typically NOAEL (or another dose descriptor) for a selected sensitive and critical toxic effect, and often a margin of safety approach and the use of safety factors is utilized. Safety factors are based on a knowledge of interspecies (animal vs human) and intraspecies (among human individuals) variability in sensitivity. Usually one assumes that humans are more sensitive than experimental animals to the effects of chemicals due to their larger size and slower rate of metabolism. Therefore a "default" safety factor of 10 is usually applied for interspecies (e.g., rat vs human) sensitivity differences, and similarly, another "default" safety factor of 10 is applied for interindividual variability among humans. However, it should be noted that these default safety factors should be replaced with more accurate

TABLE 5.25 Toxicity studies for safety evaluation of drugs, pesticides, food additives, and other chemicals utilizing in vitro systems and animal experiments required by health authorities.

Name of study	Animal species	Duration of study	Outcome of study
Biotransformation	• Human microsomes • Rats/mice	1 day to weeks	Metabolism
Kinetic studies	Rats/mice	1 day to weeks	Absorption, distribution, and elimination
Acute toxicity	Rats/mice	2 weeks	Acute effects
Subacute toxicity	Rats/dogs/minipigs	2–4 weeks	Target organs and delayed effects
Subchronic study	Rats/dogs/minipigs	3–6 months	Target organs and delayed effects
Chronic studies	Rats/mice	12–24 months	Chronic effects of low exposures
Carcinogenicity	Rats/mice	18–24 months	Carcinogenic potential
Prenatal toxicity (teratogenicity)	Rats/rabbits	3–4 weeks	Development defects
Reproductive toxicity	Rats/mice/rabbits	Several months	Potential to affect reproduction
Skin and eye irritation	• Reconstituted human epidermis in vitro • Rabbits/rats	Few days	Irritation index
Sensitization	Mice/guinea pigs	Few weeks	Potential to sensitize
Mutagenicity and genotoxicity	Bacterial strains, yeasts, cells in vitro, mice, and rats	Few days to a week	Potential to cause mutations, chromosomal damage, and other genotoxic effects

chemical specific data whenever available. Both intraspecies and interspecies safety factors can be divided into a toxicokinetic and a toxicodynamic component. Typically the toxicokinetic component is larger for the interspecies differences in sensitivity, which reflects the significance of differences in xenobiotic metabolism (e.g., safety factor of 4 for toxicokinetics and 2.5 for toxicodynamics; $4 \times 2.5 = 10$). For example, if the lowest dose that does not cause any toxicity to rodents, rats, or mice, that is, the NOAEL is 100 mg/kg, this dose is divided by the safety factor of 100. The safe dose level for humans would be then 1 mg/kg. Occasionally, a NOAEL cannot be determined if there are adverse effects also at the lowest dose level, and one has to use the lowest observable-adverse-effect level in risk assessment. In this situation, often an additional uncertainty factor of 3 or 10 is added, and then the dose is divided by a factor of 300 or 1000. A similar approach is also used when one deals with an exceptionally serious toxicological end point, such as carcinogenesis or malformations, which has a threshold dose. This kind of approach is utilized when one deals with deterministic toxicological effects, for example, target organ toxicity that has a threshold for the effect, that is, that there is a safe dose below which no harmful effects occur, and the chemical has a typical sigmoidal dose–response curve in its toxic effects.[332]

5.3.6.2 The significance of health risks of chemical compounds

Assessing health risks induced by exposure to chemical compounds is different in different societies. Typically, in industrialized societies, traffic exhausts, exhausts of power plants, and indoor and outdoor emissions of the chemical industry are the greatest concerns. In the occupational environment, one deals with relatively high exposure levels, whereas among the general public, one deals with very low exposure concentrations but large exposed populations, which complicates the assessment of the additional risks caused by the exposure. Also the magnitude of risks may vary widely. The excess risks of the general population due to air pollution (nitric and sulfur dioxide, ozone, and small particles) in Europe and the United States are up to five percent in terms of excess mortality. In Europe this corresponds, however, to about 350,000 extra deaths. In occupational environments, the exposure levels may be up to several orders of magnitude higher than background exposure of general public, but usually the exposure levels are relatively low as compared with the situation some 30 years ago. Also in occupational environments, the exposed populations can be clearly defined, and appropriate measures undertaken to avoid excessive exposure. In industrialized countries, the exposures are nowadays much better regulated than before.

This does not imply that toxicity and risk assessment are less important for guaranteeing chemical safety for workers and the general population. It simply means that the nature of exposures and their consequences have changed; rather than causing acute poisonings, exposures cause long-term effects such as allergies, cancers, and other chronic diseases such as cardiovascular diseases and asthma. The risk assessment of long-term effects of chemical exposures is much more demanding than assessing the risks of deterministic effects due to high exposures.[332]

In developing countries, exposure to chemicals, for example, pesticides, is responsible for millions of acute poisoning cases and hundreds of thousands of fatalities every year. This is due to the low standard of living, poor education, failure to appreciate the significance of hygienic measures and the effects of the compounds, and general attitudes. Furthermore most developing countries are situated in subtropical or tropical areas where the use of a number of chemicals, such as pesticides, is a necessity. Thus inadequate safety measures, regulations, and education easily lead to careless use of highly toxic compounds. Much can be done to alleviate this situation, for example, experts can visit to highlight problems, but ultimately the situation can only be improved by developing the infrastructure, education and increased awareness of the toxicity of the compounds and appropriate safety measures.

A good example is the accident that took place in Bhopal, India, in 1984. An explosion in a pesticide plant producing carbaryl, a highly toxic insecticide, caused a release of the raw material of the pesticide, methyl isocyanate, into the environment. In all, 10–25 tons of this highly toxic and reactive compound and other toxic gases were released into the densely populated area surrounding the plant. It has been estimated that the exposure resulted in 10,000 immediate deaths, 25,000 died in the long run and 500,000 severe poisonings with respiratory problems, pulmonary edema, eye irritation and blindness.[333]

5.3.6.3 Perception of risks by experts and the general population

Communication of risks to the general public is extremely important for policy-making in consideration of toxic substances. Policy makers should be able to educate the public concerning chemical hazards, and that the magnitude of the exposure and thus the dose is essential for assessing the magnitude of the risk. The existence of a chemical hazard does not imply as such a risk to human populations. However, risk communication is usually a difficult task, because the concept of risk is so difficult to understand, and, especially in a crisis, failure in risk communication is not uncommon. Several investigators have extensively studied risk perception and risk communication between lay people and experts and found marked differences in risk perception between these groups. Especially among lay people the familiarity of risk (e.g., occupational exposure vs smoking), the magnitude of the outcome (release of a chemical in the workplace vs car accident), and the severity of the outcome of an event (release of minute amounts of radon within a nuclear plant vs fire) all have a major impact on the perception of risks (see Fig. 5.56).[332] Alcohol consumption is a good example. Consumption of ethyl alcohol is one of the most important health hazards in industrialized countries, whereas food additives or pesticide residues are not causes of concern. Nonetheless alcohol use is considered a minor risk compared to nonsignificant effect of food additives or pesticide residues. Similarly with occupational health risks, exposure to chemicals in industrialized countries is in most cases a minute hazard when compared to lifestyle factors. This does not mean, however, that one should not always strive to prevent industrial exposures and accidents at all times. Perception of risks is important because it ultimately determines how effectively the knowledge produced by toxicological research can be utilized in protection of human health in occupational settings and general environment.

5.3.6.4 Special considerations

After the use of a chemical becomes widespread, new deleterious effects on human health may be observed. In such situations, the occupational limit values will have to be modified. Usually the OELs tend to decrease when more information on the toxicity of a chemical is obtained.[216] Knowledge of the specific features of various chemicals is thus extremely important for planning ventilation of industrial premises. It is important to be especially aware of those chemicals and exposure conditions that may cause long-term effects without causing any acute effects. There are also compounds, such as isocyanates that are extremely irritating at concentrations as low as 0.5 ppm. However, some workers may become sensitized to isothiocyanates at a concentration of 10 ppb, and therefore this has to be taken into consideration when planning the industrial ventilation. Special attention has to be paid to such compounds that can cause serious health effects at concentrations at which their presence cannot be observed by the human senses, that is, irritation or odor.

Ultimately, the final stage of risk assessment, risk characterization, aims at achieving a synthesis from data gathered in steps 1–3. The goal of such a synthesis is, in addition to qualitative risk assessment, quantitative risk assessment. This implies that the outcome of

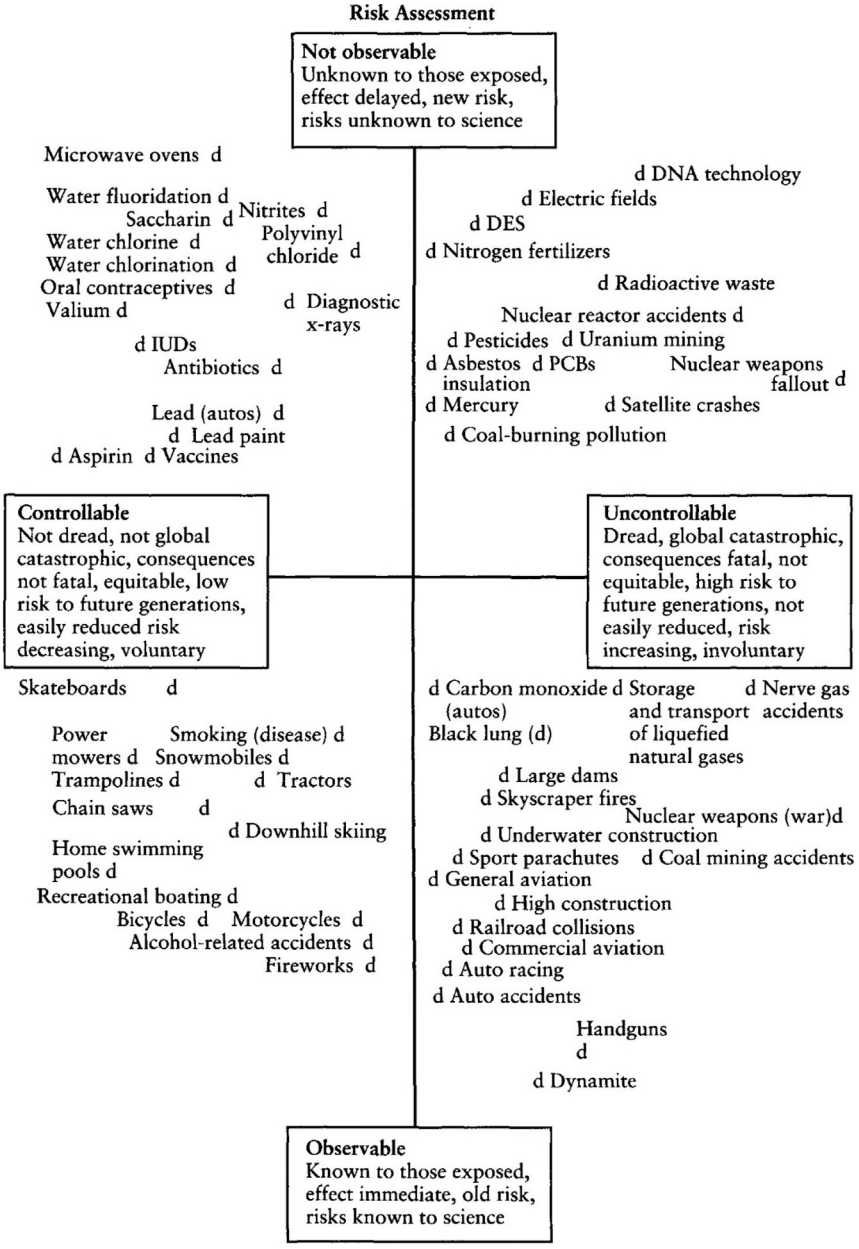


FIGURE 5.56 Risk space has axes that correspond roughly to a hazard's "dreadfulness" (d) and the degree to which it is understood.

the process should be numerical, for example, an estimate indicating how many extra cases of a deleterious health outcome are produced due to exposure to a given exposure level in a given population.[332] Decision makers demand that toxicologists be able to come up with a reliable estimate of the relative importance in terms of severity of the health outcome or the number of new cases of disease. This would then allow them to prioritize the health risks and carry out the expense/benefit analysis. It would then be easier to make decisions on which chemical problems to tackle first, and at which concentration the occupational limit value of a given chemical should be set. These measures are important for the preventive measures to be undertaken.

An important issue in the toxicity of chemicals and in assessing their risks is the *inherent toxicity* of a chemical. This implies the potency, that is, the dose at which the chemical can induce a toxic effect, whether cancer, liver damage, or nervous dysfunction. One example of a characteristic of a chemical is its reactivity, which may markedly affect its potential to cause allergic reactions or cancer or to induce irritation of the respiratory tract. Thus detailed information on the characteristics of a compound is of major significance in understanding the mechanisms of the effects that it can induce in

humans and in other living organisms and in understanding the effects themselves.[332]

5.3.6.5 Important chemical carcinogens

Polycyclic aromatic hydrocarbons have been classified as human carcinogens because they induce cancers in experimental animals and because smoking and exposure to mixtures of chemicals containing polycyclic aromatic hydrocarbons in the workplace increase the risk of lung cancer in exposed individuals. In experimental animals, benzo(a)pyrene induces cancer in different organs depending on the route of administration. Furthermore exposure to polycyclic aromatic hydrocarbons commonly occurs in occupations related to traffic (use of diesel engines in transportation and railways).

Tobacco smoke induces a myriad of deleterious health effects in exposed individuals. CO decreases oxygenation of tissues by erythrocytes, nicotine causes vasoconstriction and disturbs circulation especially in the periphery, for example, in the placenta, and tar contains a number of carcinogenic compounds. In addition, tobacco smoke irritates mucous membranes in respiratory airways and eyes, depresses cilia in bronchi, and also has immunosuppressive effects. These effects may also contribute to the increased risk of lung cancer. Furthermore all forms of smoking increase the risk of lung cancer. Association between smoking and lung cancer is no longer open to debate; there is a dose–response relationship between the number of cigarettes smoked per day and the magnitude of the risk, and an association between the duration of smoking and the lung cancer risk. Also an increased risk of bladder cancer and kidney-pelvis cancer is associated with smoking. These observations are not surprising because tobacco smoke contains many known carcinogens, such as benzo(a)pyrene, at relatively high concentrations.[334]

Asbestos fibers and arsenic compounds are also clear-cut human carcinogens.[335,336] Today, substitutes of asbestos or insulation materials, notably man-made vitreous fibers containing ceramic, glasswool, lockwool, and slog-wool fibers are suspected human carcinogens,[336] but further information is required before one can come to a final carcinogenic classification. Other potentially important human carcinogens include reactive agents such as formaldehyde and isocyanates.

IARC has also classified ethyl alcohol as a human carcinogen. The use of ethyl alcohol is associated with increased risk of cancers of the oral cavity, pharynx, larynx, esophagus, and liver. Ethanol is usually considered to be a cocarcinogen which amplifies the effects of other carcinogens. For example, the carcinogenic effects of tobacco smoke are amplified by ethyl alcohol. In addition, ethyl alcohol is also genotoxic, and causes chromosomal aberrations, sister chromatid exchanges, and point mutations in test systems where ethyl alcohol can be metabolized. Thus it seems likely that acetaldehyde, the primary metabolite of ethyl alcohol, is the compound responsible for mutagenicity of ethyl alcohol.[337]

5.3.6.6 Future perspectives

In the future, the preventive role of toxicology will be emphasized. It will be increasingly important to develop early indicators to monitor long-term subtle exposures that predict deleterious effects that are known to have a causal relationship with occupational exposures. In addition to collection of blood and urine samples, also collection of cells from points of entry into the body, for example, by nasal or bronchoalveolar lavage (BAL), will provide possibilities to explore functional chemical-induced changes at the cellular and molecular level. Routine measurements of alterations of gene expression in cells so collected may provide valuable information on causality between inhalation exposures and effects in target cells in the nasal cavity or lungs. In many instances, cells collected with nasal or BAL methods may be used to demonstrate a causal relationship between inhalation exposure and an effect in the airways. This would then allow protection of exposed workers by assessing the exposure through occupational hygienic measurements.

5.4 Ventilation noise—characteristics, effects, and suggested counter-measures

5.4.1 Occurrence

Ventilation is encountered today in practically all types of indoor environments, for example, in dwellings, leisure facilities, service facilities, schools, hospitals, factories, machine rooms, workshops, stores, offices, vehicles, meeting rooms, teaching areas, restrooms, and control rooms. The problem affects a very large number of people, both at work and during their leisure time. Complaints about ventilation noise have increased in recent years, at the same time as very limited efforts have been made to deal with the problem. The recommendations applicable to ventilation noise usually indicate a maximum acceptable level of 35–40 dB(A). The highest recommended levels are exceeded, however, in many environments.

A schematic view of a typical central station ventilation system, including a fan, ducts, and diffusers, is given in Fig. 5.57.

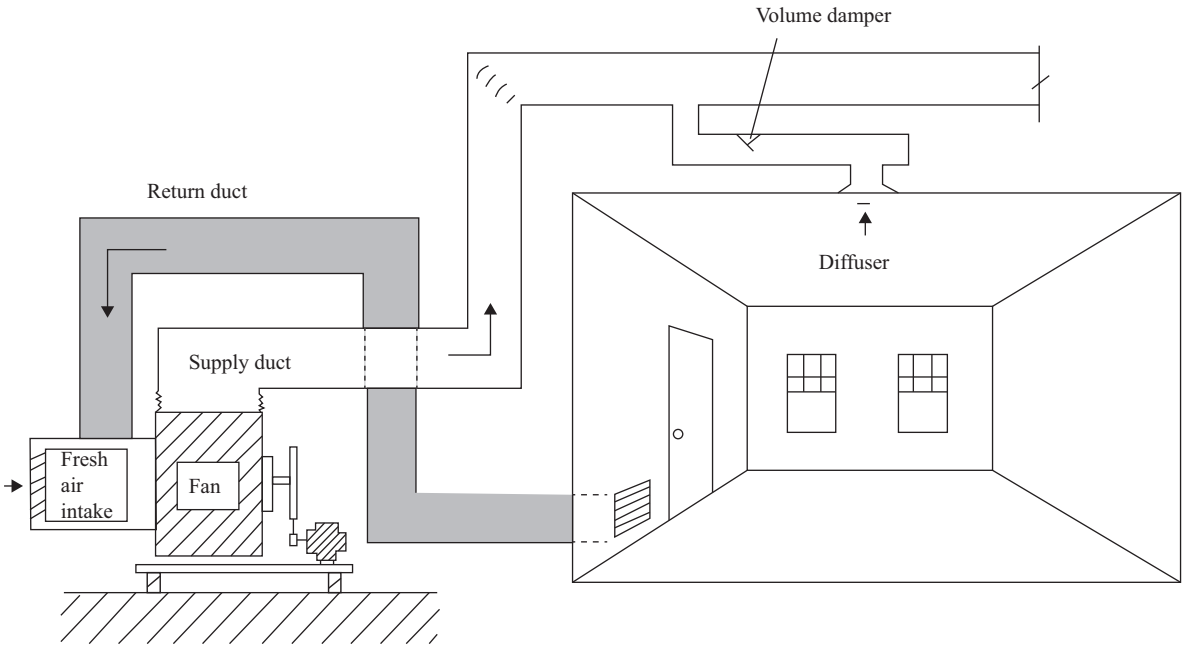


FIGURE 5.57 Capture for this is A schematic view of a typical central station ventilation system, including a fan, ducts, and diffusers.

5.4.2 Ventilation noise as an environmental problem

Noise generated from ventilation systems can constitute a big problem, particularly in environments where other ambient noise is low.[338] For this reason, ventilation noise has attracted particular attention in environments such as offices, schools, and public areas. The effects which occur there are due primarily to ventilation noise levels that are below the level at which there is a risk of hearing damage. The most common effects are a feeling of annoyance and disturbance of work due to fatigue or disturbed concentration. Ventilation noise also occurs, of course, in other environments with special demands on air quality and air change. In some environments, such as workshops, warehouses, machine rooms, and garages, the great need for air changing may lead to relatively high ventilation noise levels. The noise from large fans may in such cases sometimes reach levels around the threshold for hearing damage. A risk for hearing damage appears in cases of repeated daily exposure above 70 dB(A). Another problem which may arise at high levels of ventilation noise is masking of speech or of other sound signals. In most cases the sound from ventilation noise is dominated by low-frequency components, which means that the speech-masking effect is not always pronounced. The biggest speech-masking effect occurs if the background noise coincides with the speech frequency range, 500–4000 Hz.

Ventilation noise and the annoyance effects which may result have been a recurring question in recent years for researchers, occupational health services, and various authorities. In spite of this, there are still major shortcomings in our knowledge about the links between human effects and exposure to ventilation noise. Current regulations and recommendations are thus based on uncertain principles in certain respects.

Today there is a pronounced need to take more effective measures against this type of noise. The problem is complicated, however, by the fact that these measures in many instances are unilaterally targeted at achieving a lowering of the dB(A) level, which in many cases has resulted in only a marginal restriction of the inconvenience, or even none at all.

5.4.3 Physical characteristics

Ventilation noise originates primarily from fans and the air turbulence generated inside ducts and around supply air and exhaust air terminal devices. The appearance of the noise is, of course, affected by factors such as the speed of rotation and the power of the fan, and by how the fan is stabilized or in other ways acoustically insulated. The noise level and the frequency characteristics are also largely determined by the velocity of the air inside ducts and around terminal devices, where factors such as the dimensions and appearance of the ducts and terminal devices may play a decisive role in the appearance of the noise.

The description of the physical characteristics of ventilation noise is based on more reliable knowledge than the description of the human effect. Misconceptions about the levels and frequency characteristics of ventilation noise are still common. This in turn has sometimes led to wrong suggestions about the measures that should be taken in to eliminate the effects of a ventilation noise exposure.

The links between levels of exposure and inconvenience caused by ventilation noise are described in an investigation carried out on office workers.[339] Technical measurements and analyses of the ventilation noise at 155 typical office workplaces were in this study combined with assessments by the office workers of the level of disturbance that they experienced, the effect on working performance, fatigue, stress-related pain, and headaches. The average noise level was about 40 dB(A) at two of the workplaces, while it was about 35 dB(A) at two others. It emerged from the narrow-band analyses that the sound pressure levels (SPLs) of the infrasound were not in any event of an order that this type of sound frequencies (below 20 Hz) could contribute to any disturbance effects. Any steps taken to counter the sound frequencies of the ventilation noise under 50 Hz, that is, the point of intersection between the threshold curve of auditory perception and the spectral level distribution curve of the ventilation noise, would thus be ineffective in these cases. This conclusion is based on the fact that the SPLs of the ventilation noise frequency under 50 Hz were significantly below the threshold curve of auditory perception, and as such were not audible. This situation is considered to be representative of today's ventilation noise in offices. The same results were obtained in a study by Pääkkönen.[340] It must be pointed out, however, that levels above the perception threshold in the infrasound range can, of course, be generated from heavy-duty ventilation systems, for example, in factories, stores, and department stores.

It should also be pointed out that the levels from ventilation noise in, for example, workshop premises are often lower than those emanating from other sources, for example, machines of various kinds. It is not uncommon for the ventilation noise from industrial premises to cause more disturbance in adjacent offices than in the industrial premises themselves. The fact that ventilation noise propagates in this manner is due to its pronounced low-frequency character. The more low-frequency components in the noise, the greater the propagation. Sound radiation from industrial premises, via duct openings in facades and roofs, may in this manner also cause disturbances in nearby residential accommodations.

5.4.4 Noise generation

5.4.4.1 *Fan noise*

The fan is usually the main source of noise generation in a ventilation system. Rotating fans always constitute a source of noise generation. However, aerodynamically designed fan blades may reduce the noise generated. Another important source of noise is the bearings of various kinds inside the fan motor. Defective bearings add directly to the ventilation noise. Another common reason for high noise levels may be the imbalances which easily arise in a fan system or ductwork. Imbalances give rise to vibration and hence to noise generation. The noise generated by a fan may also be due to poor impact sound insulation in the fan mountings and duct connections. Noise from drive motors for the fan may cause strong radiation from a fan room, especially if it has poor sound and vibration insulation.

The noise from the fan is propagated in the duct to the openings inwards or outwards in the premises. The ventilation noise often propagates into the surrounding area from the supply air and exhaust air terminal units in the rooms. The ducts in themselves may also be important sources of noise, particularly if they are poorly insulated or otherwise designed in such a way that noise generation may occur.

5.4.4.2 *Flow noise*

Besides the fan noise, the duct flow may generate noise as it passes through different duct components, such as bends, branches, dampers, or terminal devices. Typically the noise is generated by turbulence when the flow detaches from a surface. At the sides of the detached flow, the shear velocity is high, and when vortex tubes develop in the shear layer, they are distorting and vibrating. In doing so, they behave like acoustic quadrupoles.[341] This kind of complicated noise source can be analyzed by simulation with a computational fluid dynamics (CFD) code. In a simulated flow, these vortical structures become clearly visible when plotting the vorticity $\nabla \times v$. Fig. 5.58 shows a contour plot of x-component of vorticity behind a whistling exit vent, drawn in the yz-section. Airflow is sucked between nested cones into the exit duct. The shear layer on both sides of the jet breaks up into large number of vibrating vortices, responsible of the noise. Noise can also be produced by fluctuating pressure at the surfaces of a flow obstacle facing the flow. Analytical calculation rules have been derived for this kind of flow noise.

5.4.4.3 *Noise simulation*

Since the acoustic wave equation governs mere propagation of sound waves, a more general equation

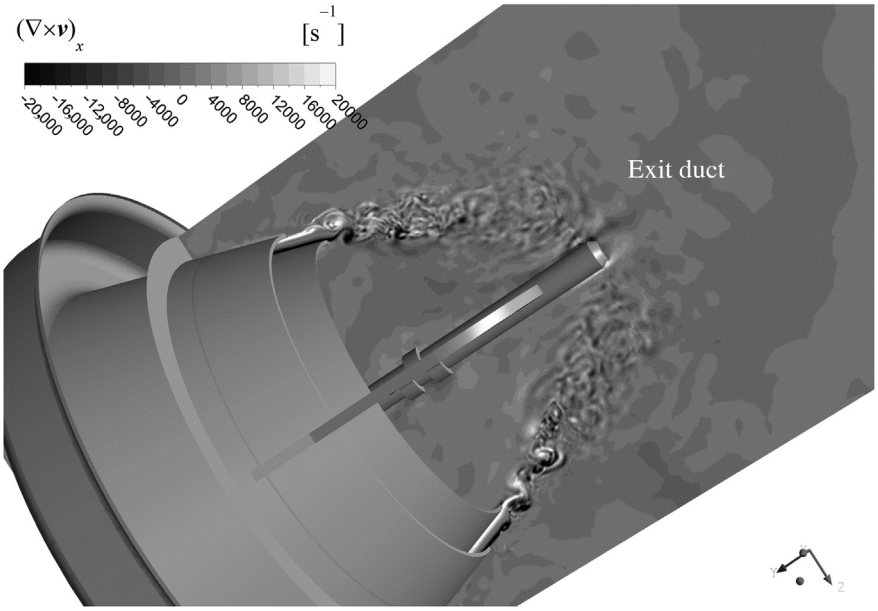


FIGURE 5.58 Vorticity component perpendicular to the plotting plane reveals the vibrating, noise-generating vortices emanating from an exit vent.

is needed to describe sound generation by turbulent flows. Such an equation is the Lighthill acoustic analogy, introduced by Lighthill in his pioneering work.[342] It can be derived from momentum equation and equation of continuity, and further simplified using adiabatic ideal gas law.

In a fluid, the rigid body equation of motion $\boldsymbol{F} = m d\boldsymbol{v}/dt$ is differentiated with respect to volume, giving $\boldsymbol{f} = \rho\, \mathrm{D}\boldsymbol{v}/\mathrm{D}t$, where $\boldsymbol{f}$ is the force density and ρ is density of the fluid. The convective derivative $\mathrm{D}\boldsymbol{v}/\mathrm{D}t$ takes into account both local, temporal velocity changes and velocity changes due to movement of fluid element in a locally changing velocity field. Thus

$$\frac{\mathrm{D}\boldsymbol{v}}{\mathrm{D}t} = \frac{\partial \boldsymbol{v}}{\partial t} + \boldsymbol{v}\cdot\nabla\boldsymbol{v}$$

The force density accelerating the flow comes from negative pressure gradient $-\nabla p$ and friction force density $\nabla\cdot\boldsymbol{T}$, where $\boldsymbol{T}$ is the stress tensor. In free flow with a high Reynolds number, the viscous forces are insignificant, and we can write the equation of motion, or momentum equation, as

$$\rho\frac{\mathrm{D}\boldsymbol{v}}{\mathrm{D}t} = -\nabla p \tag{5.54}$$

Mass flux is a physical quantity, telling the rate of mass flow through unit area. It is a vector quantity defined as $\rho\boldsymbol{v}$, where $\boldsymbol{v}$ is the flow velocity. If the density is changing in a volume element, it means that arriving and exiting fluxes are different, which in turn means that the flux is changing locally, or $\nabla\cdot\boldsymbol{v}\neq 0$. Combining, gives the continuation equation, or the equation of mass conservation,

$$\frac{\partial\rho}{\partial t} = -\nabla\cdot(\rho\boldsymbol{v}) \tag{5.55}$$

The third important equation is the ideal gas law $p = \rho TR/M$. In a sound wave, density oscillations occur so fast that air molecules have no time to exchange thermal energy. Therefore sound waves can be considered adiabatic. If, in addition, the background temperature is constant (no hot jets), temperature is a function of density alone, and can be eliminated from the equation. Then, the ideal gas law can be rewritten as ($\gamma = c_p/c_V = 1.400$ for air at 20°C)

$$p = p_0\left(\frac{\rho}{\rho_0}\right)^{\gamma} \tag{5.56}$$

Together with the approximation (c_0 is the speed of sound at equilibrium density)

$$c = \sqrt{\gamma\frac{p}{\rho}} \approx \sqrt{\gamma\frac{p_0}{\rho_0}} = c_0$$

the adiabatic ideal gas law simplifies the aeroacoustic wave equation considerably.

Combining Eqs. (5.54)–(5.56) gives the (inviscid) Lighthill equation

$$\frac{1}{c_0^2}\frac{\partial^2 p}{\partial t^2} - \nabla^2 p = \sum\nolimits_{i=x,\,y,\,z}\sum\nolimits_{j=x,\,y,\,z}\frac{\partial^2}{\partial i\,\partial j}\left(\rho v_i v_j\right) \tag{5.57}$$

While Lighthill used density as the aerodynamic variable, it has here been replaced by pressure, which is the primary quantity in acoustic measurements. It is important to remember that the source terms on the right-hand side of the equation are functions of time.

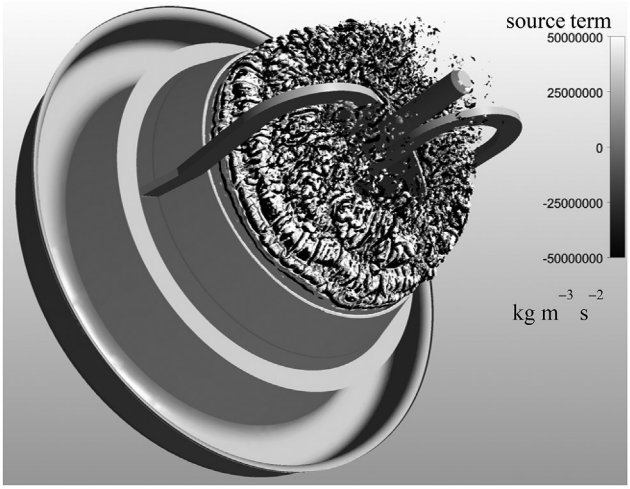


FIGURE 5.59 One frame of a video displaying the time-dependent source field of a whistling exit vent. The isosurfaces $\pm 5 \times 107$ kg/m^{3}s^2 of the source strength are drawn.

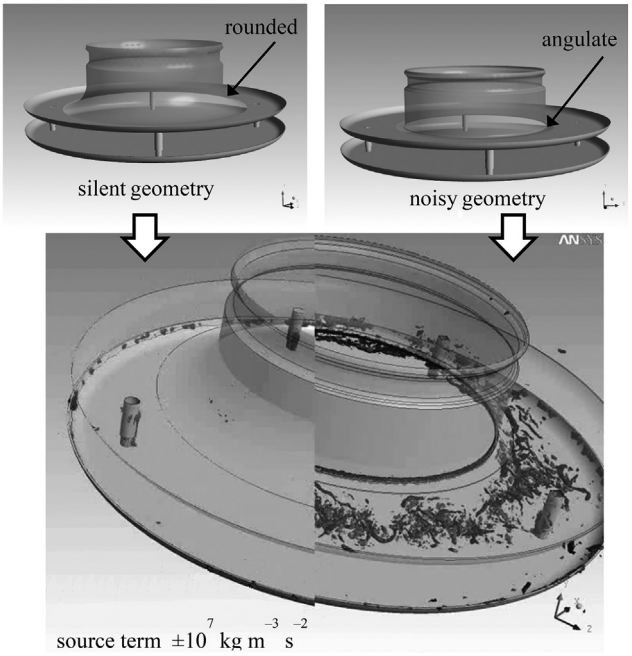


FIGURE 5.60 Acoustic source field of an HVAC inlet device. The difference between a silent and a noisy geometry is demonstrated.

Despite the fact that sound waves are density waves, in low Mach number flows the double sum on the right-hand side of Eq. (5.57) can be computed accurately enough using the incompressibility approximation ρ is the constant. Then, it can be shown[343] that it is directly proportional to the second invariant Q of the velocity gradient tensor:

$$\rho \sum_{i=x,\,y,\,z} \sum_{j=x,\,y,\,z} \frac{\partial^2}{\partial i\,\partial j}\left(v_i v_j\right) = -2\rho Q \qquad (5.58)$$

This simplifies the simulation of flow noise sources considerably, since Q is automatically computed by modern CFD codes. Fig. 5.59 displays the source field of the same noisy exit vent that was seen in Fig. 5.58. Two isosurfaces of $-2\rho Q$ are shown. The figure is actually one frame of a video demonstrating the behavior of the time-dependent source field. Another example is given in Fig. 5.60, showing two alternative geometries of a radial diffuser.[344] In the silent geometry, the joint between the inlet duct and the terminal device is rounded, while in the noisy geometry there is an angle. The split image on the bottom of the figure shows the source strength $-2\rho Q$ for both geometries. It is easy to see, which one is noisier. It is also evident that the noise is generated in the free turbulent shear flow after separation from the duct wall.

When calculating the time-dependent source field, a simulated time series of the velocity field is needed. Since the flow noise is generated by deforming vortices, the vortical structure of the flow must be resolved. This means, in practice, that the CFD has to be done using a large Eddy simulation (LES) solver. Then, the vortices are resolved down to the resolution of the computational grid. Since higher frequencies are generated by smaller vortices, the grid density in the source volume sets a limit to how high noise frequencies can be revealed. As a rule of thumb, at least four or five mesh cells should be available to span the smallest vortices resolved.[345] Consequently, a simple approximation rule[346] exists for the cut-off frequency f_M. Simplifying, it approximates the perimeter of the smallest resolvable vortex by $2d_i$, or twice the local diameter of the CFD computational cells (control volumes) in a given direction $i = x, y, z$, and frequency f_M by the corresponding frequency of rotation, or

$$f_M = v_i'/2d_i \qquad (5.59)$$

Here, velocity v_i' is the RMS mean of the velocity deviations due to turbulent fluctuation of the velocity component v_i. If it is not directly given by the CFD software, it can be approximated using the turbulence kinetic energy $k = (1/2)(v')^2 \approx (3/2)(v_i')^2$. If the diameter d_i is unavailable, it can be approximated from the local volume of computational cells as $V \approx d_i^3$. The time step used must also adapt to the size of the smallest control volumes (cell volumes), since a vortex moving through more than one cell in one time step will not be optimally resolved.[345] Since LES modeling

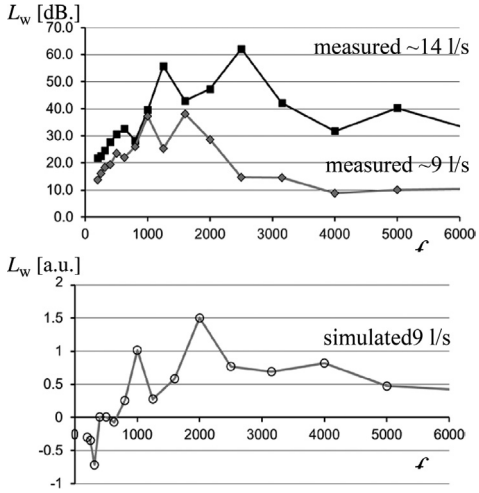


FIGURE 5.61 Measured 1/3 octave sound power levels of the exit vent of Fig. 5.59 at two different airflow rates (top). Fourier power spectrum of simulated pressure in the turbulent source volume as integrated over 1/3 octave band, in arbitrary units (bottom).

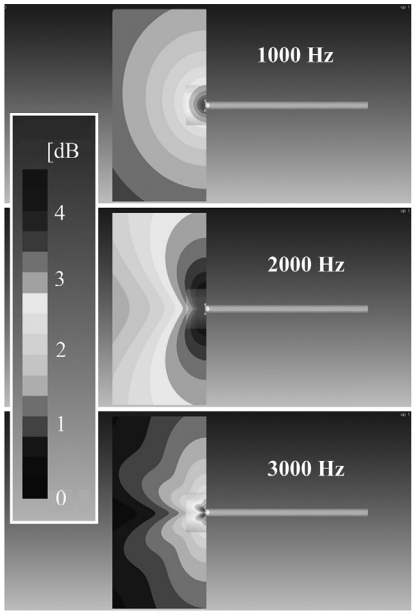


FIGURE 5.62 Contour maps of 1/3-octave band sound pressure level emanated from the whistling exit vent of Figs. 5.60 and 5.61, as given by an aeroacoustic solver. The maps demonstrate how directivity increases with increasing frequency.

approaches direct numerical simulation asymptotically when the spatial and temporal resolutions are improved, in the end, the quality of the simulation is limited merely by the hardware resources. A precursor LES must first be run until a fully developed flow field is achieved. Inlet boundary condition should not be too close to the source volume, since realistic turbulence must have given space to build up.

If the flow noise is tonal, the same frequencies are likely to be present in the most powerful source locations. In Fig. 5.61 bottom, Fourier power spectrum of simulated pressure in the vortex street of Fig. 5.58 is integrated over 1/3-octave bands, and drawn in a logarithmic scale. A comparison with measured sound power level on the top shows that it indeed resembles the measured spectrum. Since the spectrum is dependent on the air flow rate, and accurate measurement of small air flows is challenging, the measurement has been performed at two different air flow rates. The validation tests suggest that the simulated noise sources can be used to predict whether there are distinct tones present in the noise. They are also useful when comparing noise production of alternate geometries and for locating the flow noise sources.

Because pressure is a field quantity and is affected by interference and reflections from the surfaces, the far-field SPL is much more difficult to predict than the source field or the radiated sound power level. Solving SPL requires precise modeling of the pressure wave propagation from the primary sources to the far field. This task can be performed by an acoustic solver able to handle Lighthill sources, or an aeroacoustic solver. In addition, use of an aeroacoustic solver software adds the possibility to model the effect of absorbing materials and perform purely acoustic simulations by using simple artificial acoustic sources. Moreover, acoustic solvers often have add-ons for performing vibroacoustic simulations and some solvers are able to model sound propagation in a duct analytically. While flow solvers operate in the time domain, an aeroacoustic solver operates in the Fourier, that is, in the frequency domain, which makes separation of different frequencies easy. Fig. 5.62 shows an example, how an aeroacoustic solver can be used to separate frequency bands and track the propagation of different frequencies separately. Note that to calculate the acoustic sources, an aeroacoustic solver requires a time-accurate velocity field in the source volume, calculated by a CFD code.

5.4.4.4 *Noise calculation rules for duct components*

For flow noise produced at low Mach numbers by simple duct components, such as bends and constrictions, specific analytic calculation rules exist. The most advanced tools are pressure-based methods that use the drop Δp of static pressure, caused by the flow obstacle, to calculate the noise power spectrum radiated into the duct. These methods are modifications of the method by Nelson and Morfey,[347] which is valid for in-duct strip spoilers in rectangular low-speed ducts. They give the entire broad-band power spectrum of the flow noise generated, excluding possible

TABLE 5.26 Case-specific definitions for different obstacle geometries.

Obstacle type	Open area ratio, σ	Strouhal number, St	Cut-on frequency, f_0
Nelson–Morfey	a_{free}/a	$\frac{f_c(a-a_{\text{free}})}{U_c}$	$\frac{c_0}{2max\{a,b\}}$
Oldham–Ukpoho	• $1-h^2/r^2$ (disk) • h^2/r^2 (orifice)	$\frac{f_c\pi r(1-\sigma)}{2U_c}$	$\frac{100\text{ m}}{r}\times 1\text{Hz}$
Oldham–Waddington	$\frac{C_L^{1/2}-1}{C_L-1}$	$\frac{f_c}{U_c}(1-\sigma)b$	$\frac{c_0}{2max\{a,b\}}$

A separate function K(St) is also needed for each obstacle type. Average flow velocity at the obstacle is $U_c = U/\sigma$.

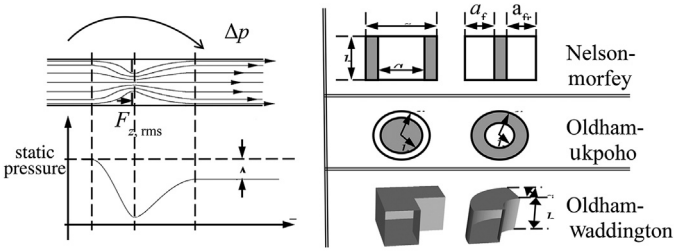


FIGURE 5.63 Pressure-based flow noise calculation rules assume that the interaction force $F_{z,\text{ rms}}$ between the obstacle and airflow is directly proportional to the pressure drop Δp (left). The original work of Nelson and Morfey was generalized to round obstacles by Oldham and Ukpoho and to mitred bends by Oldham and Waddington (right).

narrow-band tonal components. The starting point in the Nelson–Morfey method is to replace the pressure gradient force in the momentum equation by a fluctuating interaction force $\boldsymbol{F}$ between airflow and the obstacle. This force affects only in the immediate vicinity of the noisy obstacle. Thus in Eq. (5.54), the pressure gradient force density $-\nabla p$ is replaced by force density $d\boldsymbol{F}/dV = \boldsymbol{f}$. (Here $\boldsymbol{f}$ is force per unit volume, while Nelson and Morfey used force per unit mass.) The Lighthill Eq. (5.57) then reads as

$$\frac{1}{c_0^2}\frac{\partial^2 p}{\partial t^2} - \nabla^2 p = -\nabla\cdot\boldsymbol{f}$$

Mathematically, this means that the (lateral) quadrupole sources distributed in the free flow have been replaced by dipole sources located on the obstacle surface (one space derivative instead of two). Nelson and Morfey proceeded by Fourier transforming the above equation and solving it in the frequency domain.

General calculation rules are obtained by using dimensionless frequency, the Strouhal number St. Its exact definition varies depending on the type of the flow obstacle, see Table 5.26. The pressure-based methods are based on combining the following two assumptions (see Fig. 5.63):

1. For every frequency band $[f_c/\alpha, \alpha f_c]$ the time-averaged, perpendicular net interaction force $F_{z,\text{ rms}}$ between flow and obstacle, and the pressure force $A\Delta p$, where A is the duct cross-sectional area, are related by $F_{z,\text{ rms}} = (1/2)KA\Delta p$, where K is a function of the Strouhal number St alone.
2. The sound power generated into the frequency band is directly proportional to the square of $F_{z,\text{ rms}}$.

Combining these results, Nelson and Morfey obtained the following result for the sound power level, radiated from the strip spoiler to one frequency band and to both directions:

$$L_w(f_c)[dB] = \begin{cases} 120 + 20\log K(\text{St}) + 10\log\left[\dfrac{A(\Delta p)^2}{4\rho_0 c_0}\right], & f_c \le f_0 \\ 120 + 20\log K(\text{St}) + 10\log\left[\dfrac{\pi A^2 f_c^2(\Delta p)^2}{6\rho_0 c_0^3}\right] & \\ \quad + 10\log\left[1+\dfrac{3c_0\phi}{16Af_c}\right], & f_c > f_0 \end{cases} \tag{5.60}$$

Here, frequency f_0 is the cut-on frequency for the first transverse duct mode. Furthermore $\phi = 2(a+b)$ is the perimeter of the duct cross section. Function K(St), or $120 + 20\log K$(St), is a kind of default spectrum. If the Strouhal number is defined wisely, the same K(St) fits to all situations where the flow velocity or the size of the channel or the obstacle is changed. (Nelson and Morfey called this effect the collapse of data.) Function K(St) is nevertheless not given by the theory. Therefore to determine it, a measurement is required in at least one situation. After that, Eq. (5.60) can be used to predict the sound power spectrum for different flow rates and different sizes of the duct and the spoiler. Note that function K(St) also depends on the width of the frequency bands used. Taking into account that $\Delta p \propto U^2$ and $f_c \propto \text{St}U$, where U is the free flow velocity in the duct, shows that if we eliminate f_c and state L_w purely as a function of dimensionless frequency St (as was done by Nelson and Morfey), it is proportional to U^4 for plane wave sound ($f_c \le f_0$) propagation and U^6 for multimodal ($f_c > f_0$) propagation.

According to Oldham and Ukpoho,[348] Eq. (5.60) is valid also for round orifice or disk obstacles in round ducts. Then, of course, $\phi = 2\pi r$ and $A = \pi r^2$. Likewise, definitions of St and f_0 change as shown in Table 5.26.

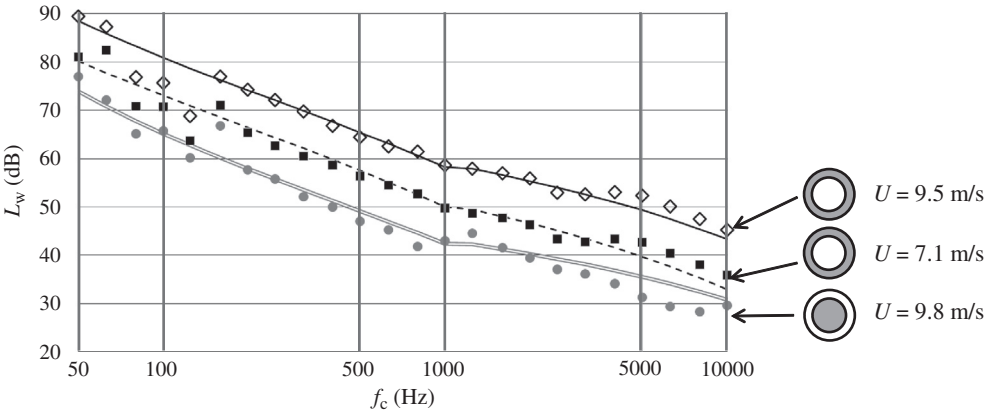


FIGURE 5.64 Eq. (5.60), with a common function K(St), fits the measured data (markers) in the three different situations shown. Displayed on the vertical axis is 1/3-octave band noise power level radiated into the duct (to both directions from the flow obstacle).

Eq. (5.60) can be applied as such even to 90 degrees mitred bends in rectangular ducts, as shown by Oldham and Waddington.[349] Then, however, calculation of the Strouhal number requires knowledge about the pressure loss coefficient C_L. The coefficient is useful also when the pressure drop Δp is not known, because

$$\Delta p = \frac{1}{2} C_L \rho_0 U^2 \tag{5.61}$$

This equation is also used, for example, when measuring airflow rate by an orifice constriction.

The pressure loss coefficient can be calculated from the same measurement used to obtain the function K(St). However, when the geometry is changed, one should remember that C_L is dependent on the open area ratio σ. This dependence can be described as

$$C_L = \frac{k_1}{\sigma^2} - \frac{k_2}{\sigma} + k_3 \tag{5.62}$$

In literature, different values for the dimensionless coefficients k_1, k_2, and k_3 can be found for different types of flow obstacles. Saarinen[350] suggests using $k_1 = 3.01$, $k_2 = 7.10$, and $k_3 = 5.39$ for Oldham–Ukpoho type obstacles. Fig. 5.64 shows a comparison of measured (markers) and predicted (continuous lines) 1/3-octave band noise power spectra for three different situations. Predicted spectra have been calculated using Eq. (5.60), with Δp approximated from Eqs. (5.62) and (5.61). The same function K(St) has been used for each spectrum, with the Strouhal number defined as

$$\mathrm{St} = \frac{f_c \pi r (1-\sigma)}{2U/\sigma^{1/2}}$$

which gave better collapse of experimental data than the original definition shown in Table 5.26. As the figure shows, the same function (5.60) fits the experimental data despite a change of U and even when orifice obstacle has been replaced by a disk obstacle.

In design guidebooks by ASHRAE and SMACNA, simplified calculation rules can be found for radius bends, mitred bends with and without vanes, T- and X-branches, and straight ducts. These obstacles have no constrictions, and thereby no variable open area ratio σ. Thus C_L is constant for each obstacle type, and Δp depends on U alone. Therefore U is substituted for Δp in the calculation rules. These rules are simplified in that frequencies below and above the set-on frequency f_0 have not been treated separately. A summary of these rules can be found in the paper by Marks.[351]

5.4.5 Effects on humans

5.4.5.1 Influence on disturbance and working performance

The office workers involved in the study mentioned above[339] rated the ventilation noise as "somewhat disturbing" to "quite disturbing" at the two workplaces where the level of exposure was about 40 dB(A). At the two workplaces where the mean level was 5 dB lower, the mean rating lay between "not at all disturbing" and "somewhat disturbing." The difference in level resulted in a clearly perceivable lowering of the average disturbance and inconvenience levels. The fact that a 5-dB reduction in the ventilation noise level can result in such a pronounced reduction of the perceived inconvenience can be explained by the circumstance that a change in level in the low-frequency range has a significantly greater effect on the loudness than would be the case in a high-frequency sound range. Measures to achieve a reduction in ventilation noise of the order

of 5 dB can thus result in measurable gains in the form of lower inconvenience reactions.

The range of answers to the question, "How does ventilation noise affect your ability to perform your tasks?" reveals that about one in every five office workers on average felt that ventilation noise made their work more difficult. A significantly greater number assessed the higher level at 40 dB(A) as an aggravating factor in the performance of their tasks at the office. About 20% considered that the higher level made their work "somewhat" or "much" more difficult. About 10% made a similar assessment at the noise level of 35 dB(A).

5.4.5.2 *Influence due to spectral distribution*

Systematic studies have been carried out for the purpose of studying how low-frequency tones, broad-band components, and/or time fluctuations in ventilation noise interfere with disturbance reactions.[352] In one of these studies, the respondents were exposed to ventilation noise that is representative of the noise encountered in office premises. The respondents were asked to use a rotating potentiometer to "set" the "most acceptable noise level" and the "least acceptable noise level" for each noise, taking account of comfort, disturbance, and performance, while performing their work at the same time. The noise level was maintained at a constant level of 40 dB(A).

When setting the most acceptable ventilation noise level consisting of a single tone, all the respondents selected a lower tone frequency for both settings than when they set the least acceptable level. The average frequencies set for the most acceptable and the least acceptable noise levels were 58 and 380 Hz, respectively. The disturbance experienced and the discomfort experienced were significantly higher, and the performance was significantly lower, during exposure to the least acceptable noise. A higher level of exertion was also experienced when exposed to the most acceptable noise.

The results clearly indicated that the ventilation noise was perceived as most acceptable when the tone was situated in the lower part of the frequency range. The experience of disturbance and the associated effects occur at exposure levels above the auditory perception threshold. Above this level, the risk of these effects increases as the perceived loudness increases, provided that the other conditions remain constant. Since the loudness can be predicted relatively accurately by means of technical measurements, any differences in the degree of disturbance can also be predicted by reference to these measurements, provided that they are dependent on differences in the loudness.

At the same dB(A) level, the noise with the stronger low-frequency feature was thus experienced as being significantly less disturbing than the more high-frequency noise. This result suggests that the A-weighting overestimates the contribution made by low-frequency tones to the disturbance experienced. This could be taken to mean that the general applicability of the dB(A) level is extremely limited at times when the goal is to carry out evaluations of the anticipated disturbance effects of ventilation noise containing tones.

An investigation designed as a tone experiment was also carried out on a broad-band ventilation noise. The average mid-range frequency for the broad band indicated by each respondent when the most acceptable and least acceptable noise levels were set reflects the situation applicable to tone exposures. The average set mid-range frequencies for the broad-band components were 129 and 456 Hz, respectively. The most acceptable noise level had a lower frequency than the least acceptable noise level for all the respondents. The estimate of the mean values in all the inconvenience variables was significantly higher for the least acceptable noise level than for the most acceptable noise level in all variables.

The results revealed by these investigations on the whole indicated that the measures taken to counter ventilation noise to reduce the effects on disturbance, performance, and exertion should be directed at higher frequency components within the low-frequency range. A greater general lowering of the dB(A) level based on measures to counter the low-frequency parts of the ventilation noise may involve a smaller limitation of the inconvenience effects than a smaller comprehensive lowering of the dB(A) level based on a measure to counter the higher frequencies of the ventilation noise.

5.4.5.3 *Influence due to exposure period*

The influence of the period of exposure on the disturbance experienced due to ventilation noise has been studied both in authentic exposure situations in offices and in laboratory experiments. The link between the estimated disturbance experienced due to ventilation noise and the period of exposure, that is, the time during which the office personnel stated that they could hear the ventilation, was tested on quite a large group of respondents.[339] The whole test group was divided into two groups, based on estimated values above or below 50 mm on a 100 mm estimation scale. The group with a lower average disturbance experience exhibited significantly lower experience periods (231 minutes) than the group with a higher average disturbance experience (390 minutes). The link between the estimated disturbance experience and the time for which

the ventilation noise could be heard points to a positive linear correlation, according to which the disturbance experience increases in line with the increase in the period for which it is experienced. This can be interpreted as indicating that exposed persons become habituated to or adapt to the low-frequency ventilation noise only to a very small extent, or not at all. It is felt that similar conclusions can be drawn from a recent, more systematic field study into the significance of the period of exposure prior to the experience of disturbance.[353] The phenomenon of habituation and adaptation is believed to be consistently stronger for high-frequency noises than for low-frequency noises.

Laboratory experiments strengthen the picture and the conclusions formed from the field studies. Both the estimated disturbance experience and the degree of exertion exhibit gradually higher values over time during a studied exposure period of 60 minutes.[354] The investigation included exposure to ventilation noises with different characteristics at levels ranging from 35 to 40 dB(A). The requirement to correlate the disturbance experience to the period for which a ventilation noise was experienced gave rise to the idea of possibly masking the experience of an unfavorable ventilation noise. Pure-tone (100 Hz), broad-band, and masked ventilation noise were compared in a laboratory experiment with regard to the effects on performance, alertness, and experience of disturbance.[355] When a masking "pink noise" was added to the pure-tone ventilation noise, there was a tendency for performance to improve and for alertness to increase, although at the same time people were more disturbed by the noise. All the effects were weak, however, and in most cases they were not statistically confirmed. The opportunities for improving the acoustic climate in an environment with ventilation noise via masking effects are thus regarded as limited. Efforts should rather be targeted at a more general reduction of the level of those parts of the ventilation noise that contribute to the overall experience of loudness.

In an office environment, the most distracting source of noise is speech.[356] It disturbs particularly creative work tasks relying on working memory or verbal processes. The degree of disturbance is highest in open offices, as was found out in a questionnaire consisting of 689 respondents.[356] Rather than intensity, noise complaints seem to be related to intelligibility of the speech, which can be measured by the speech transmission index (STI).[357] Then, the distraction distance may be decreased by using masking sound that diminishes the STI values in the office. Constant ventilation sound may then be a better choice than no background noise at all.[358]

5.4.5.4 *Influence due to time fluctuations*

Laboratory studies have demonstrated clearly that the experience of disturbance, the degree of exertion, and the performance are consistently affected more negatively when exposed to intermittent noises than when exposed to continuous noises at the same equivalent level.[359] Studies into ventilation noise in authentic environments are still very limited, however. Systematic evaluations of the links between inconvenience symptoms and fluctuations have, as expected, indicated an increased risk of symptoms with an increased breadth of fluctuation in the level.[359] The effect of an increased breadth of fluctuation in the level was also higher at higher dB(A) levels. Fluctuations in level in the vicinity of the threshold of auditory perception were correlated to lower disturbance reactions. More rapid fluctuations (2 Hz) were also experienced as more disturbing than slower fluctuations (0.5 and 1 Hz).[359] Comparisons also indicate that fluctuating tones and fluctuating higher noise frequencies are experienced as more disturbing than corresponding broad-band noise and lower noise frequencies.[359] The situation relating to the effect of fluctuations on inconvenience reactions thus reflects, as anticipated, an increased risk of influence with greater psychophysical potential for experiencing a ventilation noise. There are strong indications that the particularly disturbing effects of ventilation noise can be explained in many cases by the pronounced fluctuations which often characterize experiences of this type of noise. Fluctuations in ventilation noise can be the reason for marked increases in the experiences of inconvenience. The importance of countering the fluctuating characteristics of the ventilation noise in various ways should be emphasized.

5.4.5.5 *Effects on hearing*

The most usual effect of exposure to ventilation noise, as previously mentioned, consists of annoyance and disturbance of various kinds. Such effects may occur as a result of the relatively low levels of exposure occurring in offices, schools, etc. In industrial environments, workshops, warehouses, etc., however, the levels from a fan system may sometimes even reach the level of risk of hearing damage or of speech-masking. The risk of hearing damage and the speech-masking effect arise at levels around 70 dB(A). Pronounced or well-defined health effects expressed as a function of long time or repeated exposure to ventilation noise have not been demonstrated. However, the possibility that repeated exposure to ventilation noise may cause increased stress and in this way may have an effect on health cannot be ruled out. An increased risk of stress-related complaints may occur, not least

because human ability to acclimatize to low-frequency noise seems very limited.[353]

5.4.6 Measures

The significant differences in disturbance, when evaluated with regard to the average noise levels, show that the noise level is a decisive factor with regard to disturbance. Measures to limit the disturbance reaction due to ventilation noise should, therefore, naturally be directed in the first instance at lowering the noise level.

The extent to which a ventilation noise is perceived as disturbing depends not only on its dB(A) level, but also on the spectral distribution and the presence of tones or intermittent components in the noise. From an experiment carried out on respondents exposed to ventilation noises with different characteristics in a simulated office room, it emerged that the highest acceptable level was about 7 dB higher for ventilation noise with a superimposed tone at 30 Hz than for other types of noise.[359] In another experiment, it was found that the tolerance level was much higher for a tone than for a noise at 100 Hz, whereas the opposite tendency applied at 1000 Hz.[359]

Earlier experiments indicate clearly that a lowered SPL can be an effective measure to reduce the inconvenience reactions due to a ventilation noise, provided that it is targeted at the most critical frequency range from the point of view of influence or that the measure results in a general lowering over the entire spectral range of the ventilation noise.

5.4.7 Elimination of different ventilation noise sources

Efforts to reduce the noise from a ventilation system may be concentrated on measures concerning the fan, the fan room, the fan ducts, and the supply and exhaust air terminals.[338,360]

5.4.7.1 The fan

Fans with poorly designed or excessively simple straight blades should be replaced with quality fans with lower noise generation. As the accumulation of dirt on impellers often causes imbalance, leading to vibration and unnecessary noise, these should be cleaned regularly. Imbalance, whether due to dirt or to other causes, should be corrected by adjustment. Defective bearings should be adjusted or replaced. Struts and sharp edges in front of an impeller should be avoided. Impact noise insulation should be introduced between fan room and floor structure and between fan room and connecting ducts. The fan unit should be enclosed, with satisfactory airborne sound insulation as the objective.

5.4.7.2 The fan room

The roof and walls of the fan room should be lined internally with absorbent materials to reduce the sound level in the fan room. The wall insulation should be sufficient to reduce the transmission of sound to adjoining silent premises.

5.4.7.3 The fan ducts

Straight, internally smooth ducts should be avoided as these give very little noise reduction. Fan noise can pass virtually unobstructed. Silencers should be installed inside the ducts by covering the walls with absorbent material. In this way noise in the higher frequency range may be reduced. The low-frequency components of the fan noise are more difficult to attenuate. Very thick absorbent linings are needed to reduce such noise. Altering the area of the duct produces a damping of the noise because parts of the sound are reflected back into the duct. This kind of damping, which may also be achieved at the openings of a ventilation duct, is most effective for low-frequency noise. By inserting internally smooth bends in the duct, damping of the noise may be obtained. The larger the duct widths, the better the damping for low frequencies. Narrow ducts dampen very little. Sound-absorbent bends may produce very sharp reductions in noise level. High frequencies are dampened most easily with duct bends. To obtain damping in the low-frequency range with bends, wide ducts are required. Larger spaces with absorbent walls, "absorption chambers," built into the ductwork, also give effective damping in the lower frequency ranges. The ventilation duct often consists of large noise-generating surfaces which may need to be insulated or enclosed. Counter noise may be an alternative method of reducing the noise level in a ventilation duct. However, this method is relatively costly compared with other technical solutions.

5.4.7.4 The supply and exhaust air terminals

There is usually a certain damping of the fan noise at the opening of the duct. The damping is greatest if the opening consists of a pipe projecting clear of the wall. The noise radiation is also lowest if it is on the level of the roof or the wall in a corner of the room. Excessive air velocities in the opening may also cause noise in the terminal device, as may inappropriately shaped devices with sharp edges, etc. Supply and exhaust air terminal devices may be fitted with silencers or absorbents.

5.4.8 Exposure limits

The link between the experience of disturbance and the dB(A) level, which has been analyzed in various studies, indicates that 40 dB(A) would be equivalent to a degree of disturbance immediately below "somewhat disturbing." This result indicates that the ventilation noise in offices should lie beneath this level. When assessing noise, however, it is necessary to take into account the fact that the reaction to disturbance is influenced by many noise characteristics other than the noise level, and also by working environment factors other than the noise environment. The tolerance to noise is also reduced when working on particularly challenging tasks.[359] Under conditions of exposure and working situations of this kind, the noise level should not exceed 35 dB(A). Concerning the dB(A) weighting of the ventilation noise, it should be pointed out that this alternative can exhibit a very poor correlation to the experience of disturbance by the noise in question.[359] Previous analyses indicate, however, that the correlation of the A-weighting procedure is no poorer than that of other weightings, for example, dB(B), dB (C), or dB(D).[359] The most reliable conclusions with respect to the negative influence of ventilation noise on exposure are obtained from an analysis of the actual symptoms of inconvenience.

5.5 Glossary

Airstream Volume of air traversing a portion of or the entire respiratory tract.

Airway defense mechanisms Group of physical, physiological, and immunological mechanisms that protect the respiratory tract against disease or injury.

Airway generation Theoretical representation of bronchi position within the airway based on the number of successive bifurcations leading to a given level. Generally assumes a symmetric series of bronchial bifurcations. Asymmetric models typically use the concept of order, based on branching angle of daughter tubes, to describe relative position within the airway.

Airway lumen Opening in conducting airway through which air moves during inhalation and exhalation.

Airway surface liquid (ASL) A mixture of periciliary fluid and submucosal gland secretions.

Alveolar duct Airway distal to respiratory bronchiole leading to individual alveoli and alveolar sacs.

Alveolar gas transport Exchange of oxygen and carbon dioxide between alveolar gases and the adjacent capillary bloodstream.

Alveolar sac Group of alveoli originating from an expansion of the alveolar duct surface.

Alveolar ventilation Volume of air passing through the alveoli and alveolar ducts in 1 minute.

Anastomoses Lattice-like network of direct connections between arterioles and venules. They can allow for flow regulation and pressure equalization.

Apical epithelial surface In the airway, surface interfacing with lumen.

Atelectasis Collapse of the expanded lung.

Axial diffusion Mass transfer by diffusion along streamlines that occurs at very low velocities. In the respiratory tract, axial diffusion likely occurs in the pulmonary airways.

Basal epithelial surface In the airway, surface interfacing with basement membrane.

Basal cells Stem cells for other airway cell types that do not interface with the airway lumen.

Basement membrane Layer of dense amorphous material on which cells associated with connective tissue rest (e.g., epithelia). Appear to structurally support cells and may play a role in regulating ion and molecular transport across tissues.

Bifurcation In the airway, a relatively large bronchi divides into two smaller, more distal branches.

Bloodstream Volume of blood circulating through the heart, arteries, capillaries, and veins or within a certain anatomical region.

Body core temperature Hypothetical "average" internal organ temperature, typically referenced to either right atrial or brain temperature. A reference value of 37°C is generally used under normal environmental conditions.

Breathing frequency Number of breaths per minute.

Bronchioles Noncartilaginous, smaller, more distal subdivision of tracheobronchial tree. Walls consist of smooth muscle and elastic fibers.

Buccal Pertaining to the lateral inner surface of the oral cavity (cheek).

Bulk transport Transport of relatively large quantities of material by forced convection.

Capacitance vessels Larger venules and veins forming a large-volume, low-pressure system of blood vessels.

Carbon dioxide production Rate at which the pulmonary bloodstream transports carbon dioxide, produced by metabolic processes, to the pulmonary airstream.

Cartilaginous Consisting of cartilage.

Catabolism Destructive metabolism; breakdown of complex chemical compounds into simpler ones.

Chemical neutralization Chemical reaction that converts acids or bases to nonreactive salts.

Cilia Hair-like motile extensions of a cell wall. Airway cells use cilia to propel mucus gel toward the epiglottis.

Ciliary beat frequency Rate at which cilia travel through both the power and recovery phases of the ciliary beat cycle.

Ciliary beat power phase Interval during which forward ciliary movement propels mucus gel toward the epiglottis.

Ciliary beat recovery phase Interval during which cilia bends and returns to initial position before power stroke. Minimal mucus gel retrograde movement is thought to occur.

Clearance Removal of a substance from the airway.

Concentration gradient Difference in concentration measured between two points.

Concha One of three bony projections in the nasal turbinate region.

Conducting airways Portion of respiratory tract through which air is transported but in which oxygen and carbon dioxide are not exchanged with the bloodstream.

Dead space The portion of each breath that does not participate in gas exchange. Anatomical dead space is the volume of the conducting airways; physiological dead space also includes the contribution of pulmonary airways that are well-ventilated but poorly perfused.

Dental plaque Mass of microorganisms attached to a tooth surface.

Deoxygenated blood Blood containing hemoglobin with oxygen levels below fully saturated.

Diaphragm Large abdominal muscle that varies pleural pressure resulting in movement of air through the respiratory tract.

Diffusion-limited A chemical or physical process that depends upon the supply of material via diffusion.

Distal In the airways, positioned relatively further from the nares.

Donnan equilibrium Both concentration gradients and charge gradients contribute to the distribution of ions on either side of a membrane. Consequently, if there is a concentration gradient of an impermeable charged solute (e.g., protein) across a semipermeable membrane, then concentrations of permeable ions on either side of the membrane will not be equal.

Dose-dependent Response to applied stimuli directly proportional to concentration of stimuli.

Edema Excessive accumulation of fluid in cells, interstitial spaces, or tissues.

Eddy currents Vortices that characterize turbulent flow.

End-expiratory Airstream conditions measured when expiration ceases and just prior to initiating inspiration.

Endogenous ammonia By-product of metabolism and bacterial catabolism that diffuses into the airway lumen. Highest concentrations are found in the oral cavity.

Endothelium Layer of flat cells lining blood vessels.

Entrance flow Flow within the inlet region of a conduit that has not developed a parabolic velocity profile. Airflows within the respiratory tract are not fully developed (parabolic) because of the relatively short tube lengths and irregular geometry.

Epiglottis Leaf-shaped cartilage which closes larynx during swallowing.

Epiphase Airway surface liquid gel layer composed of mucins in the form of droplets, sheets, or blankets.

Epithelium Cellular layer interfacing with external environment which contains no blood vessels. In the airway, the epithelium lines the airway lumen.

Expiratory reserve volume (ERV) Maximum additional volume one can expire from end-tidal expiration.

Extrapulmonary airways All airways not involved in gas exchange. These include the extrathoracic airways and the tracheobronchial tree down to the terminal bronchioles.

Extrathoracic airways The portion of the human conducting airways proximal to and including the larynx. Also called the upper airways.

Fibroelastic Fibrous material possessing elastic properties. In the airway, fibroelastic tissue throughout the lung contributes to its overall elasticity, generating a positive recoil force at the functional residual capacity, or resting state of the lungs.

Flow distortion Nonuniform airstream velocity profile due to asymmetric shear, as in inspiratory bronchial airflow distal to a bifurcation.

Flow separation Formation of turbulent eddies away from boundary as flow streamlines diverge.

Forced expired volume (FEV_t) Gas volume forcibly expired within the time interval t (typically $t = 1.0$ seconds).

Forced vital capacity (FVC) Maximum forced expired volume following a maximum inspiratory effort.

Fully conditioned airstream Inspired airstream which has been warmed and humidified to approximate alveolar conditions (theoretically 37°C, 100% relative humidity).

Functional residual capacity (FRC) Gas volume remaining in the airway at end-tidal exhalation.

Gingival crevicular fluid Liquid found in gingival crevices located around the base of teeth.

Hemoglobin saturation level The extent to which the oxygen-bearing capacity of hemoglobin in red blood cells is utilized.

Homeostasis Tendency for an organism to maintain internal physiological stability.

Hydrostatic pressure Force generated by a fluid at rest, directed perpendicular to a surface.

Hygroscopic Material that readily adsorbs or absorbs moisture from the atmosphere.

Hyperbaria Pressures greater than standard atmospheric pressure (760 mmHg).

Inspiratory reserve volume (IRV) Maximum additional volume one can inspire from end-tidal inspiration.

Intercostal muscles Muscles connecting the ribs that aid the diaphragm in propelling air through the respiratory tract.

Interstitial Space found between cells.

Jet Rapidly expanding flow exiting from a very small orifice.

Lower airways The portion of the human conducting airways distal to the larynx.

Macrophage A large ameboid phagocytic cell.

Mean mass aerodynamic diameter (MMAD) Mean diameter of theoretical particles with a 1 g/cm density having the same settling velocity as an actual group of measured particles calculated on the basis of particle mass.

MeatusMetachronal wave Synchronized ciliary movement over a relatively large airway region that is responsible for the transport of objects and materials along the mucociliary escalator.

Microvilli Minute projections of cell membrane that greatly increase apical surface area.

Minute ventilation Volume of air expired or inspired during 1 minute of breathing.

Mucociliary escalator Mechanism that removes extracellularly derived materials from the conducting airways by entrapping these materials in mucus that is continuously moved toward the epiglottis by synchronized ciliary movement.

Mucus Viscous glycoprotein, proteoglycan secretion of goblet cells and mucus glands.

Nares Orifices leading into the nasal cavity; nostrils.

Nasal cavity Airway passages between the nares and posterior termination of the nasal septum.

Nasal turbinates Region within the nasal cavity denoted by convoluted bony projections (conchae).

Nasopharynx Airway passage between the posterior termination of the nasal septum and lower border of the soft palate.

Nonhygroscopic Material that resists adsorbing or absorbing atmospheric water vapor.

Noxious Injurious.

Olfaction The physiological function of sensing odors.

Oral cavity Airway passage between the lips and lower border of the soft palate.

Oronasal breathing Breathing simultaneously through both the nasal and oral cavities.

Oropharynx Airway passage between the lower border of the soft palate and epiglottis.

Oxygen uptake Rate of oxygen transfer from air resident in the pulmonary airways to the pulmonary bloodstream. This is driven by the oxygen concentration gradient and depends on metabolic demand.

Parenchyma The essential or specialized part of an organ; gas exchange portions of the respiratory tract (alveoli, respiratory bronchioles).

Particle growth Increase in particle size due to hydration.

Partition coefficient Quantitative expression of the partition equilibrium of a material between two immiscible liquid phases; usually expressed as the ratio of concentrations between the two phases.

Patency Extent of a conduit (airway, blood vessel) being open or not obstructed.

Pathogen Disease-producing organism or substance.

Peak expired flow (PEF) Gas volume forcibly expired within the time interval t (typically $t = 1.0$ seconds).

Perfusion Passage of blood through a blood vessel.

Peribronchial surface Surface surrounding a bronchus.

Periciliary fluid Transepithelial secretion along the conducting airways consisting primarily of water.

Phagocytosis Process describing the engulfment and destruction of extracellularly derived materials by phagocytic cells, such as macrophages and neutrophils.
Pleura Folded membrane surrounding the lungs. Space between the visceral and parietal layers (pleural space) is fluid filled and determines transpulmonary pressure.
Poiseuille flow Parabolic laminar flow in a straight tube.
Portal The point at which something enters the body; in the airway, the nares or lips.
Proximal In the airways, positioned relatively closer to the nares.
Pulmonary Pertaining to or affecting the lungs.
Pulmonary airways Portion of respiratory tract (alveoli, respiratory bronchioles) where gas exchange occurs.
Pulmonary perfusion rate Volumetric flow rate within the pulmonary veins.
Reentrainment Return of material to an airstream after deposition onto a surface.
Residence time Time interval during which an identifiable portion of a fluid flow remains within a given volume.
Residual volume (RV) Minimum noncollapsible volume within the airway.
Resistance vessels Microcirculatory blood vessels (arterioles, precapillary sphincters) used to regulate blood flow in a specific tissue.
Respiration Physiological process of taking in oxygen and expelling oxidative waste products (carbon dioxide, water).
Respiratory air conditioning Heat and water vapor exchange occurring in proximal airways that warms and humidifies inspired air to approximate alveolar conditions.

Respiratory bronchioles

Respiratory exchange ration Ratio of carbon dioxide production to oxygen uptake, a measure of aerobic metabolism.
Reverse flow Portion of a flow moving in a direction opposite that of the bulk of the flow.
Secretory cells Cells producing substances (e.g., mucus) with physiochemical properties differing from cellular components.
Smooth muscle Involuntary muscle tissue found in viscera and blood vessel walls.
Soft palate Movable fold along the posterior superior portion of the oral cavity dividing the nasopharynx and oropharynx.
Squamous epithelium Flattened, interlocking, toughened epithelial cells.
Submicrometric particle Airborne particle with a diameter less than one micrometer.
Submucosa Layer of tissue beneath the airway epithelium.
Surfactant Monomolecular layer of material secreted by Type II alveolar cells that lowers alveolar surface tension and stabilizes alveolar volume.
Temperature gradient Difference in temperature measured between two points.
Thermoregulation Physiological process attempting to maintain body core temperature at approximately 37°C.
Tidal volume (V_T) Volume of air inspired or expired with each breath.
Total lung capacity (TLC) Total volume of air that can be contained within the respiratory tract during maximal inspiration.
Toxin Poisonous material.
Tracheobronchial tree Series of bifurcating tubes originating at the trachea that conduct air to and from the respiratory airways.
Transepithelial Passing across a layer of epithelial cells.
Upper airway The portion of the human conducting airways proximal to and including the larynx.
Ureolysis Physiological process that breaks down urea and releases ammonia.
Vasculature Blood supply consisting of arteries, capillaries, and veins.
Vital capacity (VC) Greatest possible inspired volume.
Wetted perimeter Perimeter of conduit in contact with moving fluid.
Work of breathing Metabolic cost of breathing.

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CHAPTER

6

Target levels

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6.1 Overview of target levels

6.1.1 Introduction

In process technology and in the manufacture of equipment and systems, the starting point of design includes very precise targets.

Target levels (TLs) refer to the expected values of the system that are determined at the early stages of the design process. Fulfillment of the TLs should be validated or measured by the individuals or organizations responsible at the end of the construction process. TLs are needed as a standard against which system solutions are compared. During the comparison, it may be found that target values cannot be met by any solution or they can lead to very expensive solutions. In such cases the TLs have to be reconsidered. The TL assessment is a feedback process.

6.1.2 Factors affecting the target levels

The goal of industrial ventilation technology is to control indoor environmental conditions and pollutant emissions, so special attention must be paid to indoor conditions and contaminant levels. The target levels for indoor conditions and outdoor air emissions are derived from requirements of human health, production processes, and equipment, as well as housing and building types. The TLs should be determined for the ventilation system and for many other factors in the construction process, and the TLs should always be treated simultaneously with the design methodology. The expected value of the system as the starting point of the design has many factors that have an important impact on it. The various factors are summarized as follows.

6.1.2.1 Laws and regulations

Laws and regulations regulate the behavior standards of all walks of life, and industrial ventilation is no exception. Relevant laws and regulations on industrial processes and equipment have provisions on the ventilation temperature, heat stress, occupational exposure limits, environmental humidity, and air speed of industrial process. When determining the TL, the relevant provisions of relevant laws and regulations must be taken into account and relevant requirements must be clearly defined.

6.1.2.2 Trade standards

In addition to laws and regulations, each industry also has specific trade standards. Trade standards have specific production specifications and emission requirements for specific industrial processes. Therefore the relevant standards must be fully understood and learned before determining the expected value of the industrial ventilation system.

6.1.2.3 Nonbinding standards

In addition to relevant laws and regulations as well as trade standards, there are also requirements for industrial operating environments and production environments, such as standards related to human comfort, codes of conduct, and custom requirements. These requirements are scientific provisions related to industrial production, such as the safety of production personnel and the safe operation of production equipment. Otherwise, it will affect the physical health of relevant operators or cause equipment operation failure. Therefore when determining the TL, the relevant personnel health standards and equipment operation specifications must be fully demonstrated and considered.

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6.1.2.4 Architectural type

The determination of TL needs to be carried out according to different architecture types. Because it determines the selection process and the layout form of the system.

6.1.3 Setting principles of target level

TL setting is common. In setting the TL, the following principles should be followed.

6.1.3.1 Principle of comprehensiveness

Many factors need to be considered in setting the TL, such as laws and regulations, Party A's needs, health and comfort of personnel, production process and equipment, and type of building structure. Among them, laws and regulations and Party A's needs are the conditions that must be met, while others, such as the health and comfort of personnel, are the conditions that need to be met. The previous two conditions need to be taken into account in determining the TL.

6.1.3.2 Principle of readjustment

In the process of comparing the level reached by the solution with the TL, it may be found that not all the solutions can reach the TL, or even if the target level can be reached, the economic cost of the solution is very high. At this time the TL needs to be readjusted.

6.1.3.3 Principle of integrity

In the absence of a clear TL, the TL is usually determined by the relevant laws and regulations. In most cases the target levels set according to the laws and regulations will lead to the performance of device be emphasized and the performance of the system be ignored. Therefore the setting of TL should pay attention to the overall performance of the system.

6.1.4 Use of target levels

The use of clearly defined TLs has become more and more important in industrial ventilation. The targets must be realistic and verifiable by measurements. Different kinds of TLs can be set—for example, for indoor air quality, temperature, energy utilization, and various efficiencies.

The use of TLs is spreading to other branches of industrial ventilation, and one big problem associated with the verification of system performance has occurred. In the absence of clearly defined target values, administrative regulations have been used as targets. However, administrative regulations, such as occupational exposure limits, are seldom rigorous. In most cases the fulfillment of these data guarantees only satisfactory performance of the system. On the other hand, the use of administrative regulations as targets has obviously led to the prominence of equipment-based thinking because of the lack of other exact figures. Typically, in equipment-based thinking the focus is placed on such parameters as fan power, performance of filters, and efficiency of heat exchangers instead of the target value for the entire system.

In most cases the main steps in defining TLs relating to industrial ventilation are as follows:

6.1.4.1 Step 1: Musts

Ascertain the requirements of laws, regulations, and standards related to legislation, processes, and equipment, and compare them with customer needs. Of course, before this step, needs of the end user—for example, economical boundary conditions—are identified. At this stage the temporary TLs have also been selected.

6.1.4.2 Step 2: Needs

Ascertain nonbinding standards, human comfort standards, guidelines, codes of practice, and custom needs.

6.1.4.3 Step 3: Target levels

Define the TLs based on musts and needs.

6.1.4.4 Step 4: Design conditions

Suggest and confirm with customer the outdoor or process conditions that must reach the target temperature.

6.1.4.5 Step 5: Reliability

Find out customer requirements for process reliability. Define and obtain the customer's approval of ventilation system reliability requirements (e.g., the allowed down time).

6.1.5 Combination of target levels and design methodology

The combination of TLs and design methodology is of vital importance. Together, these two concepts form the basis for industrial ventilation systems.

The TL concept was introduced and developed by the Association of Finnish Manufacturers of Air Handling Equipment (AFMAHE) in 1985.[1,2] Indoor climate TLs have been utilized in the ventilation of public buildings, apartments, and offices for years.[3–5]

In the design methodology, the whole life cycle of the process must be considered. The life cycle of the process can be divided into four parts: design,

construction, operation, and end of the process. Each consists of different tasks. The design methodology process can be described as follows.

6.1.5.1 Given data

- Collect and identify data that do not change during the design process, such as outdoor conditions.

6.1.5.2 Process description

- Understand the industrial process and identify subprocesses.
- Identify possible emission sources, occupational areas, effects of environmental parameters on production, needs for enclosure, and ventilation equipment.
- Divide the process in parts such that their inputs and outputs to the environment can be defined.

6.1.5.3 Building layout and structures

- Collect data on building layout, openings, and their properties as basic values for load calculations.
- Complete zoning of the building based on division of the process and building layout.
- Make space reservations and add structures needed for ventilation equipment.

6.1.5.4 Target level assessment

- Define TLs for indoor zones and outdoor conditions.
- Specify design conditions for which the TLs are to be met.
- Define TLs for ventilation system, such as reliability, energy consumption, investment, and life-cycle costs.

6.1.5.5 Source description

- Determine the characteristics of the sources and methods for the calculation of local loads.

6.1.5.6 Calculation of local loads

- Calculate loads from individual sources to the environment.

6.1.5.7 Calculation of total building loads

- Calculate total loads (heat, humidity, and contaminants) from different subprocesses and the environment to ventilated enclosures.
- Take into account the fact that loads are usually time-dependent.

6.1.5.8 Selection of system

- Select an applicable system on the basis of the TLs.
- Compare acceptable systems to choose the most desirable one.

6.1.5.9 Detailed design

- Provide detailed layout and dimensioning.
- Design adjustment and control system.
- Consider special issues, such as thermal insulation, condensation risk, fire protection, and sound and vibration damping.

These are the main steps of design methodology. In some cases, all the steps are not needed, but in most cases, it is important to take all of them into consideration. In addition to the construction and the use of the system, attention should also be paid to its demolition. It is worth noting that the feedback is always a typical feature of the design methodology.

Using the procedure described previously, the TLs can be determined. Relevant calculation methods and expertise are needed in all the phases. Although the calculation of TLs takes place in different ways for particular cases, the basic procedure remains the same. The TL calculation also varies for different outside temperatures and different process parameters.

6.2 Occupational exposure limit

6.2.1 Introduction

When workers are engaged in the process of occupational production activities, they are often exposed to various external harmful substances. Depending on the amount of exposure of harmful substances, they may have acute or chronic health effects on the human body, leading to disease or even death. In order to effectively prevent and control occupational hazards, the concept of occupational exposure limits (OELs) has been proposed to quantify and improve working conditions in the workplace. Physical exposures such as radiation, noise, and vibration can also cause harm to human health. For industrial ventilation, only chemically harmful substances exposed to the air, such as toxic gases and dust, are considered.

OELs are acceptable levels of exposure that do not cause harmful effects in the health of the vast majority of contacts in occupational production activities, usually expressed as the concentration of hazardous substances, such as ppm or mg/m^3. It should be noted that OEL does not adequately protect all workers, and some people may experience discomfort or other adverse health effects even when exposed to OEL concentrations. There are numerous possible reasons for increased susceptibility to a chemical substance, including age, gender, ethnicity, genetic factors (predisposition), lifestyle choices (e.g., diet, smoking, and abuse of alcohol and other drugs), medications, and preexisting medical conditions.[6] The OEL is the

minimum air-quality standard for maintaining human health, but this standard limit is constantly changing as the technology advances. Therefore for the entire life cycle of industrial ventilation control systems, the target level of environmental control needs to be determined based on OELs, combined with production processes and economic benefits.

6.2.2 Types of occupational exposure limits

OELs are the concentration of pollutants and are closely related to contact time. Therefore three types of OELs can be defined:

- Time-weighted average (TWA) exposure limit
- Short-term exposure limit (STEL)
- Instantaneous exposure limit

TWA is the average concentration of pollutants which is the main indicator for evaluating workplace sanitation and worker exposure. The evaluation of occupational disease hazard control effects, such as project acceptance, periodic hazard assessment, system contact assessment, and the need to reevaluate the impact of work environment due to changes in production processes, raw materials, and equipment, should focus on TWA testing and evaluation. Fixed-point monitoring is a method of measuring TWA, which is calculated by collecting the average concentration of a working place within one working day (set as 8 hours per day), as shown in the following formula:

$$\mathrm{OEL_{TWA}} = \frac{C_1T_1 + C_2T_2 + \ldots + C_nT_n}{8} \qquad (6.1)$$

where $\mathrm{OEL_{TWA}}$ is the average concentration of chemical substances, $\mathrm{mg/m^3}$; T_n is the contact time at concentration of C_n, hour; C_n is the corresponding concentration in time T_n, $\mathrm{mg/m^3}$.

STEL is short-time (usually 15–30 minutes) exposure limit associated with TWA and can be considered as a supplement to TWA. It is only used for short-term exposure to chemicals that can cause acute effects, such as irritation, asphyxia, central nervous system depression, and chronic irreversible tissue damage.[7]

For many chemical substances with an OEL-TWA but without OEL-STEL, American Conference of Governmental Industrial Hygienists proposed that transient increases in workers' exposure levels may exceed three times the value of the OEL-TWA level for no more than 15 minutes at a time, or no more than four occasions spaced 1 hour apart during a workday, and under no circumstances should they exceed five times the value of the OEL-TWA level. In addition, the 8-hour TWA is not to be exceeded for an 8-hour work period.[6]

Instantaneous exposure limit represents concentrations that should not be exceeded during any part of the working exposure. These OELs are also called ceiling values. If the final average of the contaminant concentration is lower than the OEL-TWA, a limited concentration upward excursion is allowed but must be below the ceiling values.

These three types of OELs are not universal. For most substances a TWA alone or with a STEL is relevant. For some substances (e.g., irritants), only the instantaneous exposure limit is applicable.[6]

For certain chemically harmful substances, it can be absorbed through the skin. Even if the OELs are within the standard, you may be overexposed to a chemical by skin absorption. Therefore additional risk-management measures are needed to prevent skin contact when handling such chemicals.

In Council Directive 98/24/EC on the protection of the health and safety of workers related to chemical agents at work, the EN defined two types of OELs[8]:

- indicative OEL values (IOELVs)
- binding OEL values (BOELVs)

IOELVs are health-based, nonbinding values based on the latest scientific data, taking into account the availability of measurement techniques. They are designed to help employers identify and assess risks, but IOELVs are built without consideration of socio-economic and technical feasibility factors.

BOELVs consider socioeconomic and technical feasibility factors, and each member country shall determine the corresponding national binding OELs according to the European Community limits, but not exceeding the same.

The EU member states establish OELs for countries based on IOELV, such as WELs (workplace exposure limits) in the United Kingdom and MAX (maximum workplace concentration) in Germany.

Immediately Dangerous to Life and Health (IDLH) is established by the NIOSH. The purpose for establishing an IDLH value was to determine the airborne concentration from which a worker could escape without injury or irreversible health effects from an IDLH exposure in the event of the failure of respiratory protection equipment.[9]

Table 6.1 shows the OELs for some common contaminants in industrial environments.[9,10] More detailed data can be found in *EH40/2005 Workplace Exposure Limits* and *NIOSH Pocket Guide to Chemical Hazards*.

6.2.3 Setting occupational exposure limits

The setting of OEL usually requires analysis in many fields, including chemistry, toxicology, epidemiology,

TABLE 6.1 Occupational exposure limits (OELs) for some common air contaminants in industrial environments.

		Workplace exposure limit				
		OEL-TWA		STEL-TWA		IDLH
Substance	CAS number	ppm	mg/m^3	ppm	mg/m^3	ppm
Formaldehyde	50-00-0	2	2.5	2	2.5	20
Nitrogen dioxide	10102-44-0	0.5	0.96	1	1.91	20
Ozone	10028-15-6	–	–	0.2	0.4	5
Acetone	67-64-1	500	1210	1500	3620	2500
Toluene	108-88-3	50	191	100	384	500

IDLH, Immediately Dangerous to Life and Health; *STEL*, short-term exposure limit; *TWA*, time-weighted average.

occupational medicine, and occupational health. In addition to the characteristics of chemical substances, it should be considered as far as possible the actual production process and contact mode. In developing the OEL, all relevant organizations such as regulatory agencies, industry, e-employers and employees associations, consumers, and scientific communities can make recommendations for new OELs values and eventually reach a balance among all relevant organizations and be considered by governments as adopted exposure limits. Due to differences in evaluation criteria and setting procedures, there may be differences in OELs levels of some substances in different countries. In addition, countries also have different emphasis on the selection of OELs for the selection of hazardous substances. For example, the OELs value set by China in heavy metals is not available in the relevant US standards.

In general, the OEL is based on the no observed or lowest observed adverse effect level (NOAEL or LOAEL)[11] for the most critical effect seen in one or more repeated dose animal studies. EC Scientific Committee for Occupational Exposure Limits (SCOEL) recommends that good human data be used instead of animal data. However, human data are often unavailable or scientifically inadequate. In such cases the OEL is derived from well-conducted animal studies and the use of assessment factors. In a risk assessment for humans, the NOAEL from an animal study is the typical starting point, and assessment factors are then applied to account for both uncertainty and variability in the subsequent extrapolation elements. If an appropriate NOAEL is available, then no extrapolation and hence, no assessment factor is necessary. There are cases where the critical effect of NOAEL cannot be determined and also where the LOAEL is considered a more appropriate starting point.[11] Where, only the LOAEL is available an additional assessment factor is used typically. So far these assessment factors have been applied only to single substances.

According to SCOEL, a health-based OEL for specific substances is derived generally by means of the following steps[11]:

- Collecting information on all hazards of the substance, that is, all physical, chemical, toxicological, and epidemiological data.
- Deciding if data are sufficient to derive a health-based OEL.
- Evaluating all adverse effects. Establishing which adverse effect occurs at the lowest exposure. That is the critical effect for setting an OEL.
- Selecting relevant human and animal studies of sufficient quality, in which the critical effect has been shown.
- Establishing the mode of action and mechanism, threshold or nonthreshold; evaluating the dose–response relationship for all relevant adverse effects and establishing the NOAEL and the LOAEL.
- Recommending a numerical value for an OEL expressed as a TWA of 8 hours for a substance below the NOAEL while applying appropriate uncertainty factors.
- Deciding if a STEL is needed in addition to an OEL expressed as a TWA of 8 hours and recommending a numerical value for a STEL, if necessary.
- Documenting the full process of deriving the OEL for the substance.
- Determining the appropriate method for air monitoring in human and animal studies.

For hydrocarbon solvents the reciprocal calculation procedure methodology that takes into account the properties of the individual constituents is recommended by HSE. The OEL of a mixture can be calculated as the following formula[10]:

$$\frac{1}{\mathrm{OEL_{sol}}} = \frac{\mathrm{FR}_a}{\mathrm{OEL}_a} + \frac{\mathrm{FR}_b}{\mathrm{OEL}_b} + \ldots + \frac{\mathrm{FR}_n}{\mathrm{OEL}_n} \tag{6.2}$$

where OEL_{sol} is the OEL of the hydrocarbon solvent mixture, mg/m^3; OEL_a is the OEL of the component a, mg/m^3; and FR_a is the fraction (w/w) of component a in the solvent mixture.

When there are two or more chemical substances in the workplace, if the toxicological data of the combined action is lacking, the concentration of each chemical substance should be determined separately and evaluated according to the OEL of each substance.

However, when two or more toxic substances act together on the same organ, system, or have similar toxic effects (such as stimulating effects), or if these substances are known to produce additive effects, the following formula should be satisfied[7]:

$$\frac{C_1}{OEL_1} + \frac{C_2}{OEL_2} + \ldots + \frac{C_n}{OEL_n} \leq 1 \qquad (6.3)$$

where C_n is the observed concentration of chemical substances, mg/m^3 and OEL_n is the corresponding OEL, mg/m^3.

For workers working under unusual work schedules, the intake of harmful factors may increase, and it is necessary to adjust the corresponding OELs. In Patty's industrial hygiene, a comprehensive introduction to adjusting OELs to accommodate unusual work schedules is provided.

6.2.4 Occupational exposure assessment

Occupational exposure assessment is a process of identifying occupational hazards present in the workplace and evaluating the likelihood of personal injury from these hazards. In order to assess chemical exposures and ensure that exposures do not exceed OELs, the actual exposure per worker per day needs to be measured, but such measurements are often unrealistic.

In order to reduce the number of exposure measurements and reduce the cost of evaluation, workers within similar exposure groups (SEGs) are usually evaluated during the actual exposure assessment. SEGs refer to a population of workers who have the same general exposure to the chemical being studied due to the similarity of the manner in which the tasks are performed.

By measuring the exposure level of a few workers in SEG and comparing with OELs, if the requirements are met, all workers in SEG are considered to meet the relevant requirements.

In the initial exposure assessment, if some SEGs did not meet the OELs, then control measures should be taken and the exposure assessment repeated. After the first contact assessment, a reassessment should be carried out on a regular basis. EN 689: 2018 gives the exposure assessment strategy shown in Fig. 6.1.[12]

6.3 Target level of thermal environment

A target of HVAC is to achieve comfort thermal environment for people. In ASHRAE Standard 55, thermal comfort is defined as "that condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation." The definition is a bit cryptic about what "satisfaction" means. But it reveals the involving factors influencing human comfort, such as physical, physiological, and psychological level judgment.

People tend to evaluate thermal comfort and discomfort through temperature and moisture sensations from the skin, deep body temperatures, and the efforts necessary to regulate body temperatures. Generally, comfort occurs when body temperatures are held within narrow ranges, skin moisture is low, and the physiological effort of regulation is minimized. Some behaviors may change the thermal and moisture sensations to reduce thermal discomfort. Some examples are altering clothing, altering activity, changing location, or changing the thermostat setting.

6.3.1 Introduction

Different people have different sensations for thermal comfort. Therefore, certain methods are needed to judge the thermal conditions in a space, which are able to evaluate the acceptance percentage of occupants in that space.

When conditions for thermal comfort are defined, six primary factors must be considered. And some secondary factors will also affect comfort in some situations.

The six primary factors are listed next:

1. metabolic rate
2. clothing insulation
3. air temperature
4. radiant temperature
5. air speed
6. humidity

6.3.1.1 Metabolic rate

When choosing the best conditions for comfort and health, it is necessary to know the ratio of work done in daily sports activities, because metabolic rate is directly proportional to exercise intensity. The metabolic rate varies widely, depending on the activity, the person and the conditions under which the activity takes place.Table 6.2 lists typical metabolic rates for an average adult ($A_D = 1.8$ m^2) for activities performed continuously. In the table, met is the unit to express the metabolic rate per unit DuBois area, which is

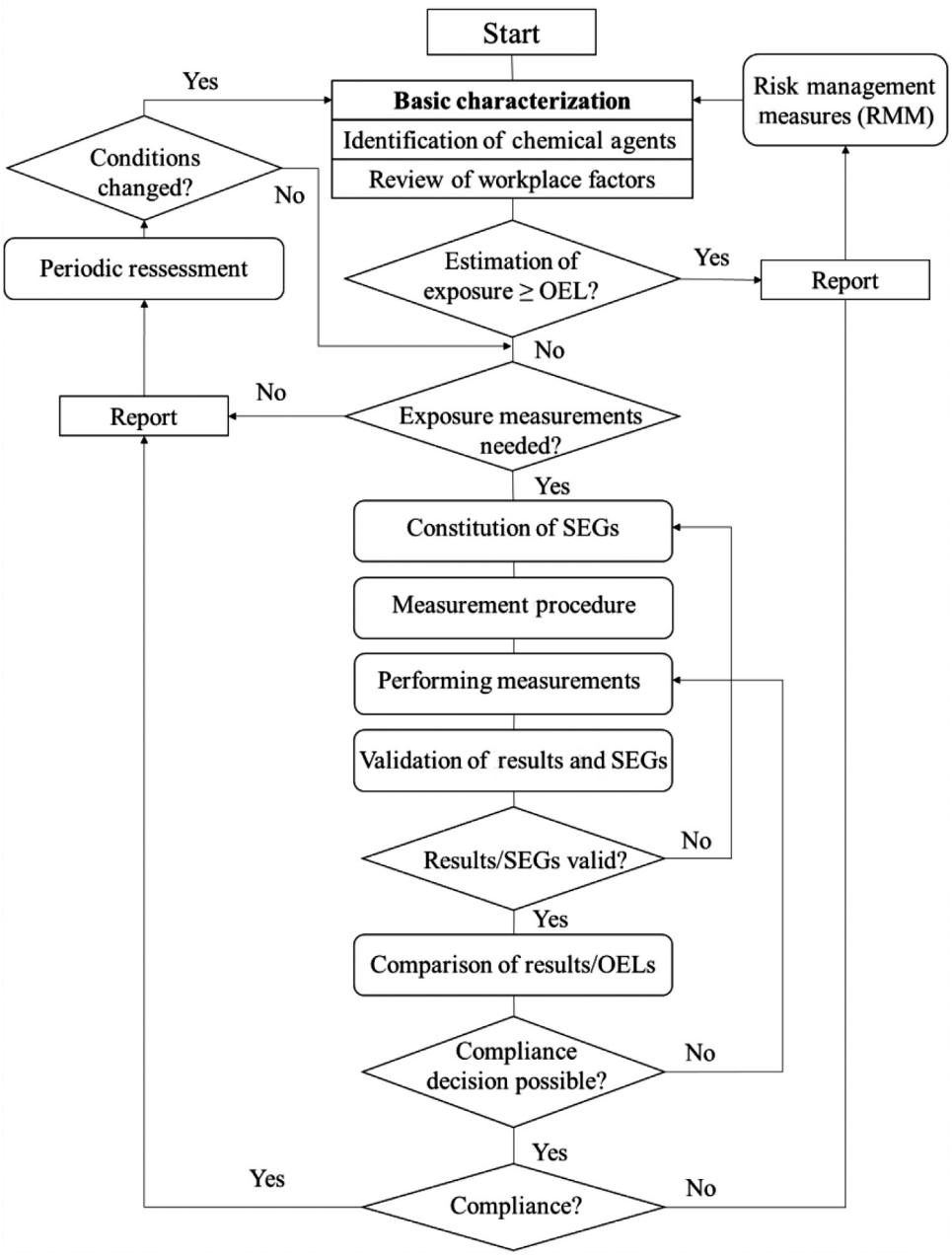


FIGURE 6.1 Exposure assessment strategy.

difined as the metabolic rate of a sedentary person (seated, quiet): 1 met = 58.1 W/m^2 = 50 kcal/(h · m^2).

6.3.1.2 *Clothing thermal insulation*

There are two accurate ways to determine clothing insulation, one is measurements on heated manikins and the other is measurements on active subjects. For most routine engineering work, estimates are sufficient using tables and equations in ASHRAE Standard 55-2017. The sensible heat loss from skin through convection and radiation in a given environment can be measured using thermal manikins.

Clothing insulation value may be expressed in clo units. To avoid confusion the symbol I is used with the clo unit instead of the symbol R. The relationship between the two is $R = 0.155I$, or 1.0 clo is equivalent to 0.155 (m^2 K)/W.

6.3.1.3 *Mean radiant temperature*

The mean radiation temperature is a key variable in the calculation of human body heat. It is a hypothetical

TABLE 6.2 Typical metabolic heat generation for various activities.

Activities	Metabolic heat generation (met)	Activities	Metabolic heat generation (met)
Sleeping	0.7	Car driving	1.0–2.0
Seated, quiet	1.0	Cooking	3.2
Walking (0.9 m/s)	2.0	Light machine work	2.0–2.4
Walking (1.8 m/s)	3.8	Heavy machine work	4
Office writing	1.0	Basketball sport	5.0–7.6

uniform temperature of the enclosed space, in which the radiant heat transfer from the human body is equal to the radiant heat transfer of the actual non-uniform enclosed space. Globe temperature, air temperature and air velocity measurements can be combined to estimate the mean radiant temperature.

The mean radiant temperature can also be calculated based on the measured temperatures of surrounding walls and surfaces and their positions relative to people. Most building materials have a high emissivity, so all surfaces in the room can be assumed to be black. The following equation is then used:

$$T_r^4 = T_1^4 F_{p-1} + T_2^4 F_{p-2} + T_N^4 F_{p-N}$$

where T_r is the mean radiant temperature, K; T_N is the surface temperature of surface N, K; and F_{p-N} is the angle factor between a person and surface N.

Since the sum of the angle factors is uniform, the fourth power of the mean radiation temperature is equal to the mean value of the surrounding surface temperatures to the fourth power, and is weighted by the respective angle factors.In 2017 ASHRAE Handbook-Fundamentals, there are figures used to estimate the angle factors for rectangular surfaces.

6.3.2 Thermal environment assessment

As individuals can react very differently, both regarding acceptance of a given environment and regarding the strain that a given environment imposes, it may under certain circumstances be beneficial to incorporate individual physiological and subjective measurements. Also, there may be a need for evaluation of an individual's capability for performing a certain job under severe conditions in the field or in an ergonomics laboratory investigation.

6.3.2.1 *Cold stress*

Definition

Cold stress threshold limit value (TLV) is designed to protect workers from the most severe effects of cold stress (hypothermia) and cold damage, and to describe exposure to cold working conditions where almost all workers can be exposed repeatedly without adverse health effects. The purpose of TLV is to prevent deep body temperature from falling below 36°C (96.8°F) and to protect the limb from cold damage (deep body temperature is the core temperature of the body determined by conventional rectal temperature measurement methods). For a single environment that is occasionally exposed to a cold environment, the core temperature should be allowed to drop to no less than 35°C (95°F). In addition to the provisions for total body protection, the objective of TLV is to protect all parts of the body, with emphasis on hands, feet, and head from cold injury.[13]

The clinical presentations of victims of hypothermia are shown in Table 6.3.[13]

Pain in extremities may be the first early warning of the danger to cold stress. In a cold environment, when the body temperature drops to 35°C, maximum severe shivering will occur. This must be regarded as a sign of danger to the workers. When the workers tremble obviously, they should stop contacting the cold immediately. When severe shivering occur, useful physical or mental work is limited.

If operating in air below 4°C (40°F), workers must be provided with sufficient insulated dry clothes to keep the core temperature above 36°C (96.8°F. Wind chill cooling rate and cooling power are the key factors. (Wind chill cooling rate is defined as heat loss from a body expressed in watts per meter squared which is a function of the air temperature and wind velocity upon the exposed body.) The higher the air speed, the lower the temperature in the working area, the greater the insulation value of the protective clothing required. Table 6.4 gives the equivalent chill temperature chart related to the actual dry bulb temperature and air speed. When estimating the combined cooling effect of wind and low air temperatures on exposed skin, or when determining the insulation requirements of clothing to maintain the core temperature of the deep body, an equivalent chill temperature should be used.

6.3.2.2 *Heat stress and heat strain*

The aim of TLV is to keep the core temperature of the human body within the range of 37°C ± 1°C. In certain cases, the core temperature of the human body will exceed this range by selecting people, conducting environmental and physiological monitoring and other control methods.

TABLE 6.3 Progressive clinical presentations of hypothermia.[a]

Core temperature		
°C	°F	Clinical signs
37.6	99.6	"Normal" rectal temperature
37	98.6	"Normal" oral temperature
36	96.8	Metabolic rate increases in an attempt to compensate for heat loss
35	95	Maximum shivering
34	93.2	Victim conscious and responsive, with normal blood pressure
33	91.4	Severe hypothermia below this temperature
32	89.6	Consciousness clouded; blood pressure becomes difficult to obtain; pupils dilated but react to light; shivering ceases
31	87.8	
30	86	Progressive loss of consciousness; muscular rigidity increases; pulse and blood pressure difficult to obtain; respiratory rate decreases
29	84.2	
28	82.4	Ventricular fibrillation possible with myocardial irritability
27	80.6	Voluntary motion ceases; pupils nonreactive to light; deep tendon and superficial reflexes absent
26	78.8	Victim seldom conscious
25	77	Ventricular fibrillation may occur spontaneously
24	75.2	Pulmonary edema
22	71.6	Maximum risk of ventricular fibrillation
21	69.8	
20	68	Cardiac standstill
18	64.4	Lowest accidental hypothermia victim to recover
17	62.6	Isoelectric electroencephalogram
9	48.2	Lowest artificially cooled hypothermia patient to recover

[a]*Presentation approximately related to core temperature.*
Reprinted from the January 1982 issue of American Family Physician. Published by the American Academy of Family Physicians.

TABLE 6.4 Cooling power of wind on exposed flesh expressed as equivalent temperature[a] (under calm conditions).

Estimated wind speed (m/s)	Actual temperature reading (°C)										
	10.0	**4.4**	**−1.1**	**−6.7**	**−12.2**	**−17.8**	**−23.3**	**−28.9**	**−34.4**	**−40.0**	**−45.6**	**−51.1**
	Equivalent chill temperature (°C)											
Calm	10.0	4.4	−1.1	−6.7	−12.2	−17.8	−23.3	−28.9	−34.4	−40.0	−45.6	−51.1
2.24	8.9	2.8	−2.8	−8.9	−14.4	−20.6	−26.1	−32.2	−37.8	−43.9	−49.4	−55.6
4.47	4.4	−2.2	−8.9	−15.6	−22.8	−31.1	−36.1	−43.3	−50.0	−56.7	−63.9	−70.6
6.71	2.2	−5.6	−12.8	−20.6	−27.8	−35.6	−42.8	−50.0	−57.8	−65.0	−72.8	−80.0
8.94	0.0	−7.8	−15.6	−23.3	−31.7	−39.4	−47.2	−55.0	−63.3	−71.1	−78.9	−85.0
11.18	−1.1	−8.9	−17.8	−26.1	−33.9	−42.2	−50.6	−58.9	−66.7	−75.6	−83.3	−91.7
13.41	−2.2	−10.6	−18.9	−27.8	−36.1	−44.4	−52.8	−61.7	−70.0	−78.3	−87.2	−95.6
15.65	−2.8	−11.7	−20.0	−28.9	−37.2	−46.1	−55.0	−63.3	−72.2	−80.6	−89.4	−98.3
17.88	−3.3	−12.2	−21.1	−29.4	−38.3	−47.2	−56.1	−65.0	−73.3	−82.2	−91.1	−100.0
Wind speeds greater than 17.88 m/s have little additional effect	Little danger In < h with dry skin. Maximum danger of false sense of security.				Increasing danger Danger from freezing of exposed flesh within one minute.			Great danger Flesh may freeze within 30 s.				
	Trench foot and immersion foot may occur at any point on this chart											

[a]*Developed by US Army Research, Institute of Environmental Medicine, Natick, MA.*
Equivalent chill temperature requiring dry clothing to maintain core body temperature above 36°C per cold stress TLV.
TLV, *Threshold limit value.*

The potential health hazards of working in a high-temperature environment significantly, more than any other physical factors, depend on physiological factors that can cause a range of susceptibilities, depending on the level of acclimatization. Therefore, professional judgment is particularly important to assess the level of heat stress and physiological heat strain, so as to fully consider personal factors and work types, and provide sufficient guidance for the protection of almost all healthy workers. The assessment of heat stress and heat strain can be used to assess the safety and health risks of workers.[13]

Definition

Heat stress refers to the net heat load that workers sustain under the combined effect of metabolic heat production, environmental factors (ie. air temperature, humidity, air flow and heat radiation) and clothing requirements. Slight or moderate heat stress may lead to discomfort and may detrimentally affect work efficiendy and occupational safety, but is not harmful to health. With the heat stress approaching human tolerance limits, the risk of getting heat-related diseases increases.[13]

Heat strain is the overall physiological response caused by heat stress. The physiological response is aimed at eliminating excess heat in the body.

Thermal acclimatization is a gradual physiological adaptation that can improve an individual's ability to withstand heat stress. It requires physical activity under heat stress conditions similar to worker's expected environment. If the recent history of heat stress exposure has been at least two consecutive hours, as far as TLV is concerned, the worker can be considered acclimatized. When the activity is no longer continuous under those heat stress conditions, the thermal adaption begins to degenerate, and distinctly regress after four days and may disappear completely within three to four weeks. As thermal acclimatization is related to the level of heat stress exposure, people will not be acclimated to sudden and intense heat stress, such as a heat wave.

Assessment

Wet-Bulb Globe Temperature (WBGT) is a practical first-order indicator used to assess the effect of the environment on heat stress. It is affected by air temperature, air humidity, heat radiation and air movement. As an approximate indicator, it cannot completely consider all interactions between individuals and the environment, such as some special conditions like heating from radio frequency or microwave.

WBGT is calculated by the following equations:

With direct exposure to sunlight:

$$\mathrm{WBGT_{out}} = 0.7T_{\mathrm{nwb}} + 0.2T_{\mathrm{g}} + 0.1T_{\mathrm{a}}$$

Without direct exposure to the Sun:

$$\mathrm{WBGT_{out}} = 0.7T_{\mathrm{nwb}} + 0.3T_{\mathrm{g}}$$

where T_{nwb} is the natural wet-bulb temperature, T_{g} is the globe temperature, and T_{a} is the dry-bulb (air) temperature.

Since WBGT is only an indicator for the environment, the screening criteria is amended for the work requirements and the impact of clothing.Table 6.5 lists the clothing adjustment factors added to the environment WBGT.

In order to determine the degree of heat stress exposure, the working mode and requirements must be considered. If the working area, as well as rest area, is distributed in multiple locations, the time-weighted average WBGT should be used.

The values in the table cannot be used for completely enclosed clothing; nor can the values of multiple layers of clothing be superimposed. These coveralls assume that only moderate underwear is worn underneath, rather than a second layer of clothing.

If there is enough information about the heat stress effect of the required clothing, the first item of detailed analysis is task analysis, which includes time-weighted average effective WBGT (environmental WBGT plus clothing adjustment factor) and metabolic rate. The TLV and action limit against the metabolic rate are shown in Fig. 6.2.[13]

Even under the same heat stress conditions, the risk and severity of excessive heat strain will vary greatly among the population.

The normal physiological responses to heat stress make it possible to monitor worker's heat strain, to use this information to assess the level of heat strain present

TABLE 6.5 Clothing-adjustment factors for some clothing ensembles.[a]

Clothing type	Addition to WBGT (°C)
Work clothes (long sleeve shirt and pants)	0
Cloth (woven material) coveralls	0
Double-layer woven clothing	3
Spunbond Meltblown Spunbond (SMS) polypropylene coveralls	0.5
Polyolefin coveralls	1
Limited-use vapor-barrier coveralls	11

[a]*These values must not be used for completely encapsulating suits, often called Level A. Clothing-adjustment factors cannot be added for multiple layers. The coveralls assume that only modesty clothing is worn underneath, not a second layer of clothing. SMS*, Spunbond Meltblown Spunbond; *WBGT*, Wet-bulb globe temperature.

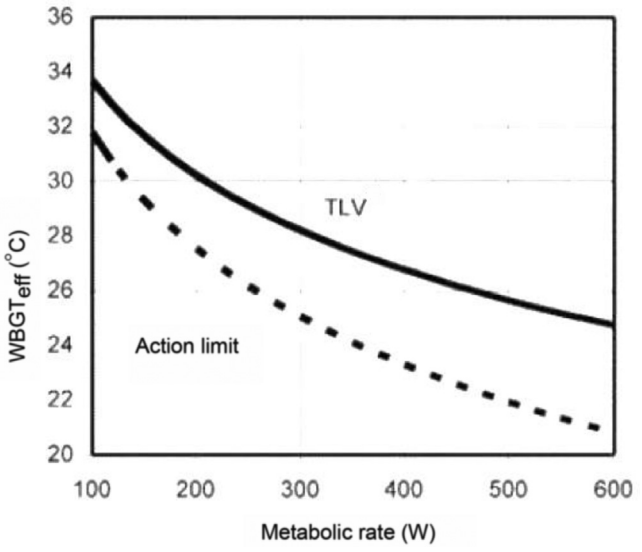


FIGURE 6.2 TLV and Action Limit for heat stress

in the workforce, to control exposures, and to assess the effectiveness of implemented control measures.

6.3.2.3 *Thermal comfort*

Some other factors may also have effect on thermal response and comfort except for the previously discussed independent environmental and personal variables. The secondary factors include environmental heterogeneity, visual stimuli, age and outdoor climate, etc. A study of 1600 college students by Rohles in1973 and Rohles and Nevins in 1971 showed that there is a correlation between comfort, temperature, humidity, gender, and exposure time. The thermal sensation scale developed from these studies is called ASHRAE thermal sensation scale and is shown as follows.[14]

+3 hot
+2 warm
+1 slightly warm
0 neutral
−1 slightly cool
−2 cool
−3 cold

Environmental factors affecting comfort

Operative temperature Under the given air humidity, air movement, metabolic rate, and clothing thermal resistance, the comfort zone can be determined. The comfort zone is defined according to the operating temperature range that provides acceptable thermal environmental conditions, or according to a combination of air temperature and average radiant temperature that is considered thermally acceptable for people.

People wear different clothes according to different situations and seasonal weather. In ASHRAE Standard 55-2013, the comfort zones of 0.5 and 1.0 clo dressing are given, as shown in Fig. 6.3[15]. The thermal resistance of winter suits is about 1clo, and the thermal resistance of short-sleeved shirts and pants is about 0.5clo. Because the hot and cold boundaries of the comfort zone are affected by humidity, in the middle of the comfort zone, a typical person will wear neutral clothes to produce neutral or near-neutral thermal sensation. But near the hotter boundary of the comfort zone, a person will feel an additional thermal sensation of +0.5 on the basis of ASHRAE's thermal sensation scale; near the boundary of the colder area, the person may feel an additional heat sensation of -0.5.

Humidity threshold The upper limit of humidity in the comfort zone is 0.012 $kg_w/kg_{dry\ air}$, and the lower limit of humidity is not clearly specified.

Usually when the dew point temperature is below 0°C, low humidity may dry the skin and mucus surface and cause discomfort in the nose, throat, eyes and skin.

At high humidity, excessive skin moisture can increase discomfort, especially skin moisture from physiological sources.

Air speed The comfort zone of Fig. 6.3 is suitable for situations where the air speed does not exceed 0.2 m / s. However, increasing the air speed can improve comfort in areas outside the maximum temperature limit in the figure. Fig. 6.4 shows the air velocity required to compensate for the temperature rise above the warm boundary. The combination of air velocity and temperature defined by the curve in this figure corresponds to the same skin heat loss.[14]

The increase amount in air speed on the improvement of comfort is affected by the mean radiation temperature $tr_{_ave}$. The curves in Fig. 6.4 are for different levels of $tr_{_ave}-t_a$. When the mean radiation temperature is low and the air temperature is high, the increase in air speed has little effect on increasing heat dissipation. For a given temperature rise,a higher air speed is required to increase heat dissipation. On the contrary, when the average radiant temperature is high and the air temperature is low, the effect of increasing the air velocity is obvious, and the air speed to be increased is less. Fig. 6.4 is suitable for people wearing light-colored clothing with the thermal insulation of clothing between 0.5 and 0.7 clo for nearly fixed physical activity.

Asymmetric thermal radiation Causes of asymmetric or non-uniform thermal radiation in the space include cold windows, uninsulated walls, cold surfaces, cold or hot mechanical equipment, or inappropriately sized heating boards installed on walls or ceilings. In residential buildings, offices and other

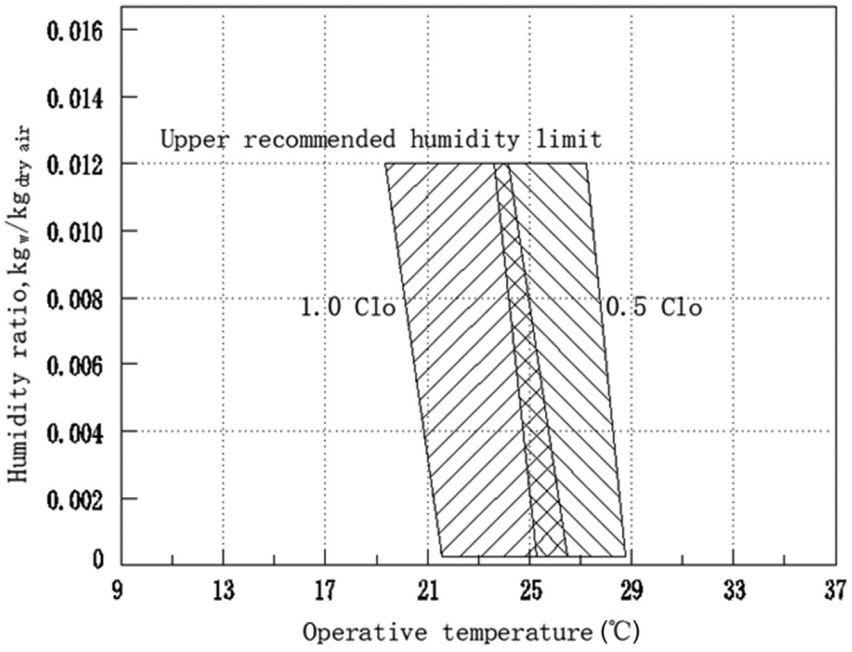


FIGURE 6.3 ASHRAE Summer and Winter Comfort Zones.

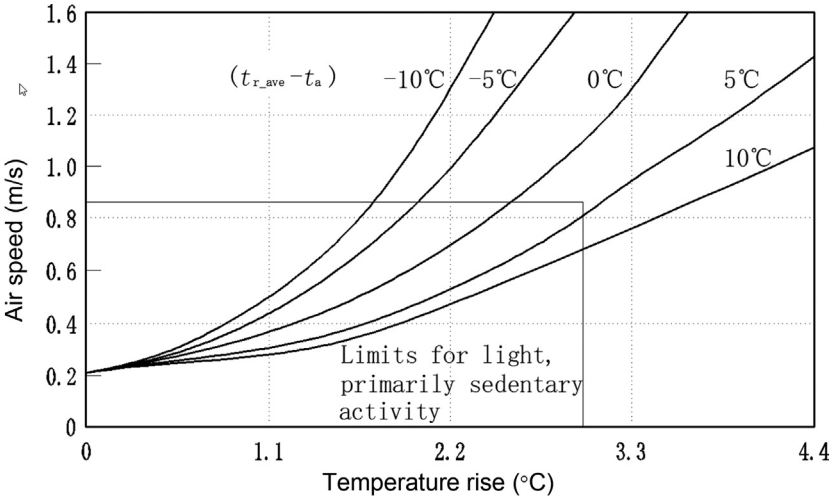


FIGURE 6.4 Air Speed to Offset Temperatures Above Warm-Temperature Boundaries

places, the most common reasons are cold windows, inappropriately sized heating boards, or heating boards installed on the ceiling. In industrial plants, cold or hot objects, cold or hot equipment, etc. can cause asymmetric or uneven heat radiation.[14]

Thermal radiation asymmetry refers to the difference in ambient radiation temperature between the two sides of the human body. More precisely, thermal radiation asymmetry refers to the difference in radiation temperature observed from a facet in the opposite direction.

As shown in Fig. 6.5, people are more sensitive to asymmetric radiation caused by hot surfaces above the head than vertical cold walls. The cold surface above head or vertical warm wall have much less impact on dissatisfaction. These data are especially important when using radiant panels to improve comfort in large space with cold surfaces or cold windows.

Draft sensation The draft sensation is an unpleasant local cold sensation caused by the air flow. Not only in many ventilated buildings, but also in cars,

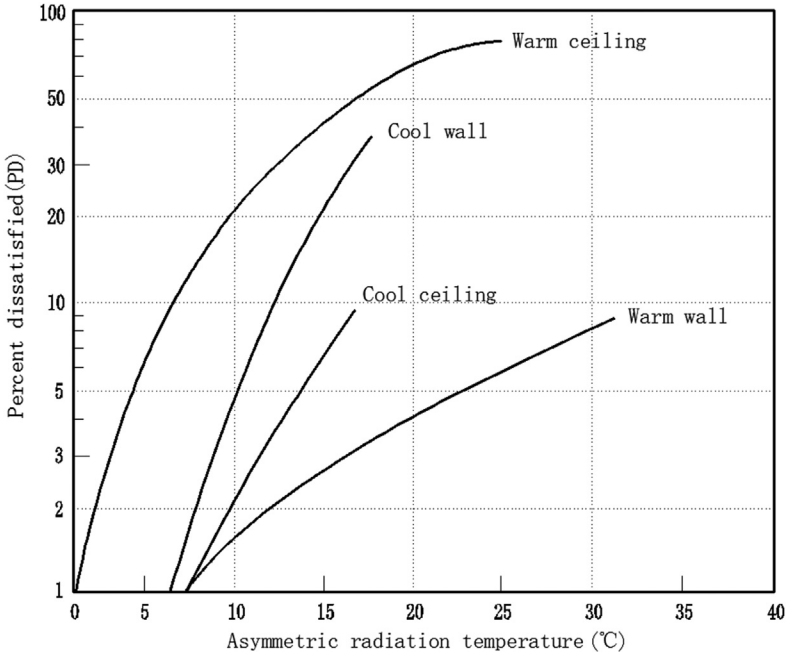


FIGURE 6.5 Percentage Dissatisfied Caused by Asymmetric Radiation.

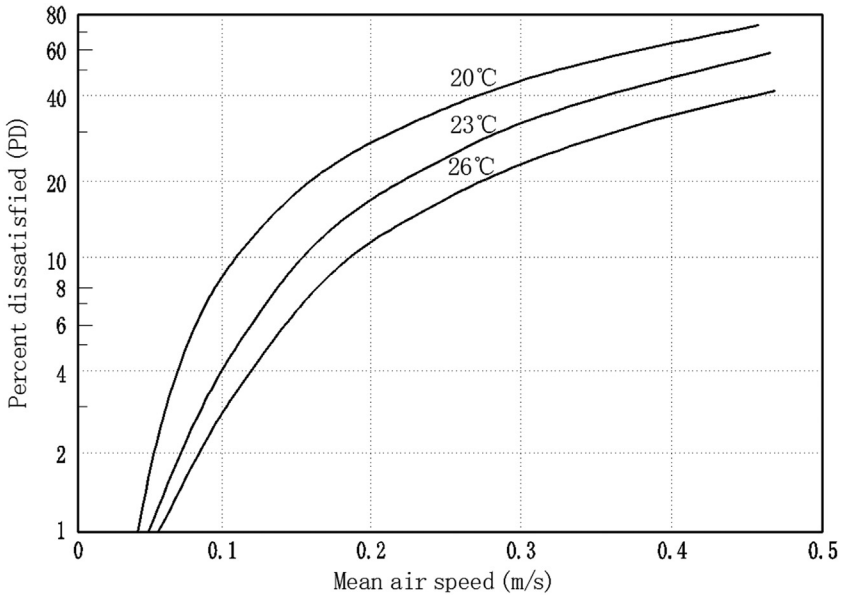


FIGURE 6.6 Relationship bewteen Percentage of People Dissatisfied and Mean Air Velocity.

trains and airplanes, this is a problem that cannot be ignored. When people feel the sensation of draft, they usually ask to increase the temperature of the air in the room or turn off the ventilation system.

Fig. 6.6 shows the relationship between the percentage of dissatisfied and the mean air speed at the neck who feels the draft around the head region. The head region includes the head, neck, shoulders and back. Air temperature significantly affects the dissatisfaction.[14] The data in Fig. 6.6 is only suitable for people who wear normal indoor clothes and do light physical work, mainly sedentary work. People with high activity intensity are not so sensitive to the draft.

Vertical air temperature difference In most indoor spaces, the air temperature usually rises with the height above the floor. If the temperature gradient is large enough, even if the entire body is thermally neutral, there may still be thermal discomfort at the head or cold discomfort at feet. Fig. 6.7 shows the relationship between dissatisfaction and the vertical air temperature difference between head and ankles.[14]

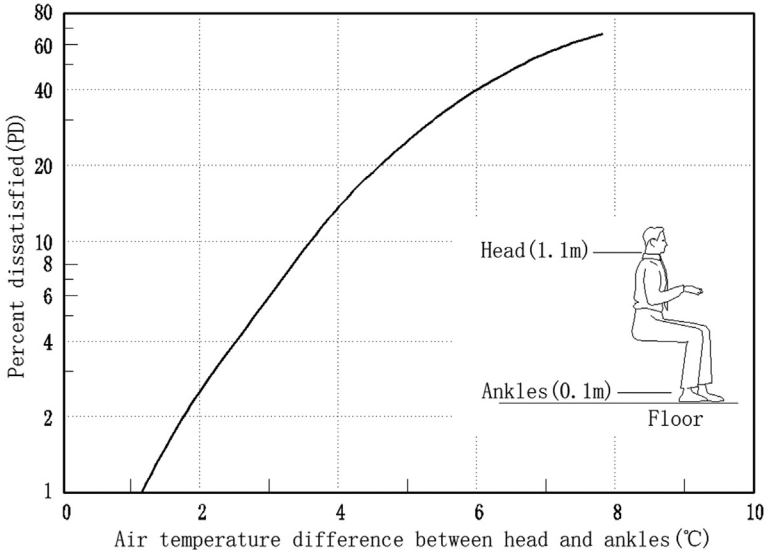


FIGURE 6.7 Relationship Between Percentage Dissatisfied and Air Temperature Difference Between Head and Ankles.

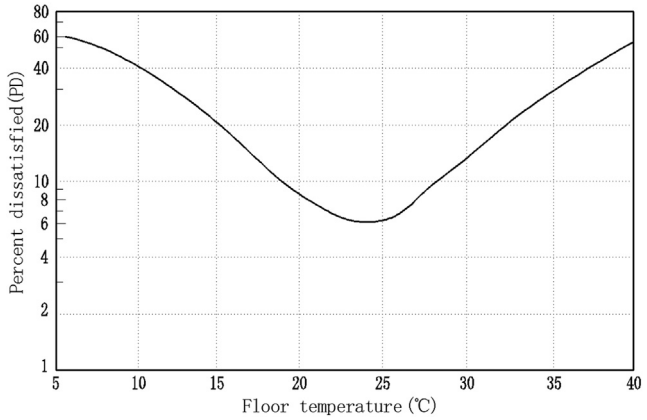


FIGURE 6.8 Relationship Between Percent Dissatisfied and Floor Temperature.

The air temperature at the head zone should be lower than that at feet is not as critical for people. If the air temperature in the head area is lower, one can tolerate a greater vertical temperature difference.

Warm or cold floors Usually the feet are in direct contact with the floor. The local discomfort of the feet is usually caused by the floor temperature being too high or too low. At the same time, the floor temperature will significantly affect the mean radiation temperature of the room. If the floor is too cold and the feet feel cold and uncomfortable, the usual reaction is to increase the temperature of the room; during the heating season, this will increase heating energy consumption. Floor radiant heating system can prevent cold floor from causing discomfort.

Fig. 6.8 shows the relationship between floor temperature and dissatisfaction, combining experimental data with seated and standing individuals. In all experiments, the people were in a thermally neutral state. Therefore, dissatisfaction is only related to discomfort caused by cold or warm feet. There is no significant difference in preference for floor temperature between women and men.[14]

Personal factors affecting comfort

The air temperature, humidity, velocity and its variation range, as well as individual parameters such as metabolism and clothing insulation are the main factors that directly affect the heat balance and thermal comfort. However, many secondary personal factors may affect comfort in more subtle ways.

Age Human metabolism will decline slightly with age, and some people believe that the comfortable conditions derived from experiments on young healthy subjects cannot be used for other age groups. Related research shows that the thermal environment preferred by the old people is not different from the thermal environment preferred by the young. Old people's lower metabolism can be compensated by lower evaporative heat loss.

Sex According to related research, men and women like almost the same thermal environment. Women's skin temperature and evaporative heat loss are slightly lower than that of men, which is balanced with women's slightly lower metabolism levels. Women usually prefer higher ambient temperatures than men, partly because women often wear lighter clothes.

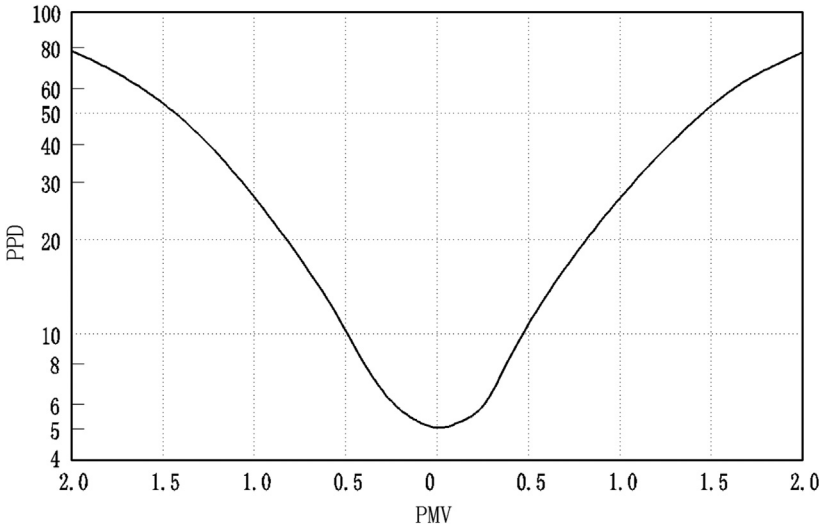


FIGURE 6.9 Relationship Between PPD and PMV.

Acclimation The thermal acclimation has little effect on the ambient temperature of personal preference. However, in uncomfortable warm or cold environments, thermal acclimation usually has a certain influence. Compared with people in cold climates, people who work and live in warm climates can more easily accept warmer environments and maintain relatively higher work efficiency.

Prediction of thermal comfort

There are several ways to predict thermal comfort and thermal sensation. One method is to use Fig. 6.3 and adjust the insulation of clothing and activity level. A more accurate numerical prediction is to use the PMV-PPD model.

Predicted mean vote

The PMV (Predicted Mean Vote) index predicts the mean response of a large group of people based on ASHRAE's thermal sensation scale. Fanger (1970) correlated the imbalance between PMV and the actual heat flow of the human body in a given environment and the heat flow required for optimal comfort under specified activities, as shown in the following formula:[14,16]

$$\mathrm{PMV} = [0.303\exp(-0.036M) + 0.028]L$$

Where L is the heat load of the human body, which is defined as the difference between heat generation and heat loss in the human body in the actual environment. It is assumed that the human body with skin temperature and sweat evaporation heat loss values is among comfort zone at actual activity level. M is the body's metabolic heat production rate.

TABLE 6.6 Acceptable thermal environment for general comfort.

PPD	PMV range
<10	$-0.5 < \mathrm{PMV} < +0.5$

PMV, Predicted mean vote; *PPD*, predicted percent dissatisfied.

Predicted percent dissatisfied

After predicting PMV, the predicted perpcent dissatisfied (PPD) under certain conditions can also be predicted. Fanger (1982) associates PPD with PMV as follows:[17]

$$\mathrm{PPD} = 100 - 95\exp[-(0.03353\mathrm{PMV}^4 + 0.2179\mathrm{PMV}^2)]$$

The definition of dissatisfaction is anyone who does not vote -1, +1 or 0. This relationship is shown in Fig. 6.9. When PPD is 10%, the corresponding PMV range is ± 0.5. Even if PMV = 0, about 5% of people are not satisfied.[14]

The PMV-PPD model is widely used in the design and on-site evaluation of comfort conditions.

The operating temperature range shown inFig. 6.3 is for a situation acceptable to 80% of the personnel. This is based on 10% dissatisfaction with general thermal comfort based on the PMV-PPD index, plus an average 10% dissatisfaction caused by local thermal discomfort. And Table 6.6 defines the recommended PPD and PMV range for typical applications.

6.4 Target levels for industrial air quality

6.4.1 Introduction

The need for exact target values relating to processes and products is self-evident in the design phase

of process technology, equipment manufacture, and many other areas of engineering. Industrial ventilation is defined as "airflow technologies" to "control the indoor environment and emissions of the workplace." It is therefore logical that the goals of industrial ventilation are unambiguously quantified. In the past the design goals of industrial ventilation have been expressed in many terms, such as airflow rate, filter classes, control velocity of a local exhaust hood, and surface temperature of a radiator. Although these are indispensable quantities in the design and realization processes, they account only indirectly for the environment within the premises. Therefore the goal of industrial air quality should be defined using target values of the relevant contaminants occurring in the room.

The need for the implementation of target levels (TLs) for air quality in industrial work rooms stems from different concerns. In addition to technological factors, the systematic design methodology, life-cycle assessment, advances in air-distribution methods, and increased integration with the process and building automation have to be considered. The recent changes in the standards of working conditions that favor the TL process also must not be forgotten.

Occupational exposure limits (OELs) have been used for decades as exposure criteria for air contaminants. OELs are quantitative health standards expressed as a maximum mean concentration of dangerous air contaminants over a given reference period. Although OELs are required for the establishment of health-based standards, the limits entail a great deal of uncertainty. They indicate the minimum level of air quality corresponding to the present understanding of what is an acceptable risk, but they do not serve as a criterion for planning a comfortable environment and control technologies for the whole life cycle of the system, say over a period of 20 years.

Although high exposures to air contaminants still occur in a number of industries, the general trend is that the current levels of most commonly used chemicals are decreasing and are clearly lower than the corresponding OELs. There is the opinion that the OELs have less impact on the improvement of occupational environment than in previous decades, and that they focus on compliance testing instead of control. It is also worth noting that compliance with the OELs does not guarantee 100% protection for all individuals.

Nowadays, many companies have adopted a policy of continuous improvement of working conditions. Therefore it is desirable to create TLs for those who want to pursue more efficient control by applying the best available control technologies. There are also endeavors to create optimal working conditions in order to improve the performance and the innovativeness of a staff and hence enhance productivity. A series of laboratory and case studies show that employee productivity is higher when the work environment is appropriate for the tasks being done.[18] Such efforts are typical in the advanced sector of industry. One can say that there is a transition from "blue-collar to white-collar work."

Today occupants, building owners, and other end users of ventilation systems are more interested in the level of air quality and thermal climate than in the techniques by which that level is achieved. This is supported by the fact that industrial ventilation systems in modern premises are more complicated and tightly integrated with the process and building automation. It is therefore difficult for end users or nonprofessionals to evaluate whether a ventilation system is functioning correctly.

The aim of this section is to consider the scientific and technological grounds for assessing TLs of contaminants that frequently occur in the occupational environment as well as use of the TLs. The TL of a contaminant is defined as the predetermined concentration of a dominant contaminant to be achieved by air technology or other control methods. A TL can be considered for an entire room volume or a zone, such as an occupied zone or a limited part of the occupied zone. Unlike OELs the proposed TLs for air contaminants are voluntary guidelines.

6.4.2 Grounds for assessing target levels for industrial air quality

In order to assess the target concentrations of air contaminants both human risk–based and technology-based approaches can be used (Fig. 6.10). Various approaches are dealt with in more detail elsewhere.[19,20]

The procedure of assessing health-based OELs for chemical substances includes determination of no observed adverse effect level for the critical toxic effect and application of an appropriate safety factor based on expert judgment (see Section 5.3). In principle the same procedure could be used for assessing the TLs. However, the quantitative risk assessment procedure entails notable uncertainties at low-dose regions. In addition, exposure limits are revised at certain intervals in the light of new research information and actual policy objectives. In most cases the limits have been reduced over the years. In theory, one possibility for assessing a TL for desired air quality could be the determination of an exposure that cannot be distinguished from the biological monitoring values of the nonoccupational population. However, adequate data for this purpose exist only for a few substances in advanced industrialized countries, and for that reason a technology-based approach for TL assessment is considered in this chapter. Similar control strategies, based on performance standards and risk assessment, have been proposed for some industries—for example, the

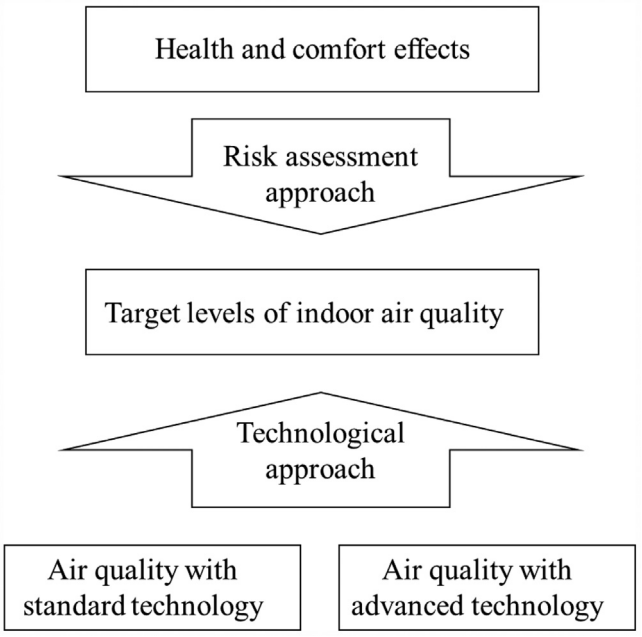


FIGURE 6.10 Approaches for the assessment of target level air quality.

pharmaceutical industry and technology transition in the defense sector.[21,22]

In the technological approach, qualitative and quantitative information on emissions released by various production and work processes, as well as data on control technology performance, are required in order to specify the air quality TLs that are technically and economically feasible. The approach is based on information on current concentration levels that are achieved by different control technologies, ranging from standard practices to the most advanced technology options (Fig. 6.10).

Existing contaminant exposure data banks can be utilized to survey the standard practices.[23,24]

In recent years, benchmarking has proved to be a very successful tool in total quality management.[25] Basically, benchmarking is a target-setting and comparison process in which the current standard performance is compared with the best possible performance. A typical feature of the benchmarking process is periodic upgrading of the targets. Applying the benchmark philosophy to air quality control means that the air quality level produced by the best available technology must be defined. The benchmark air quality is obtained by determining the contaminant concentrations in plants with advanced production and control technology. For selection of the benchmark plants, the following criteria were set:

- Effective elimination of emission sources through the selection of the best process technology or effective control of emissions from sources that cannot be avoided.
- Balanced mechanical supply and exhaust ventilation equipped with an advanced air-distribution strategy to accurately control the flow patterns in a work space.
- Air-handling units equipped with heat recovery and sophisticated control of the key parameters of HVAC systems, such as temperature, airflow rate, and pressure difference.
- New or renovated premises.

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CHAPTER

7

Principles of air and contaminant movement inside and around buildings

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7.1 Introduction

Proper selection and sizing of ventilation systems require knowledge of emissions from internal contaminant and heat sources and an understanding of the mechanisms and characteristics of air and contaminant movement. Traditionally indoor air environment is built with the theory of "dilution" and all the parameters are treated to be uniform; however, the actual indoor environment is nonuniform for various airflow patterns, including the common mixing ventilation. Under the nonuniform indoor environment, the distribution of air parameters is affected not only by the emission rates of contaminant and heat source as well as the ventilation rate but also by the number and locations of the sources, as well as the specific airflow pattern formed by multiple influencing factors. To achieve timely and effective ventilation and create a desired nonuniform indoor environment, it is crucial to clarify the capacity of each air supply jet to deliver fresh air to different local areas and the quantitative effect of each contaminant source on the contaminant concentration at different positions in a specific airflow field. In the condition that the transient parameters are concerned, the impact of the initial condition of contaminant should not be neglected, especially in the initial period of contaminant dispersion.

In this chapter, the major factors affecting the air movement and contaminant distribution inside ventilated space are summarized and classified as:

- contaminant sources (Section 7.2);
- transport mechanism of contaminant in ventilated space (Section 7.3);
- forced convection or supply air jets introduced into the room by mechanical or natural ventilation systems or their combination (Section 7.4);
- free convection flows along heated and cooled vertical surfaces and above heat sources, covered in Section 7.5;
- airflow created in the vicinity of local and general exhausts (Section 7.6);
- aerodynamic means of the large opening protection, described in Section 7.7; and
- airflow through intended and unintended openings and cracks in the building envelope (Section 7.8).

In some cases, the ventilation process in the room can be simplified and mechanisms of air and contaminant movement under the influence of each of the above factors can be described using simplified

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theoretical principles of fluid mechanics, empirical data, and observations from numerous research studies. In general, the ventilation process in a room is complex and different factors have a joint effect on airflow patterns and characteristics, in continued spaces, and in industrial buildings particularly.

Although there are no ready recipes available, this chapter provides some guidance on

- How to select predominant factors affecting air and contaminant movement in ventilated spaces?
- How to account for their joint effects?
- How to predict the airflow characteristics with an accuracy acceptable in ventilation design?

7.2 Contaminant sources

7.2.1 Classification

Knowledge of the process or operation and contaminant sources is essential before ventilation systems can be selected and designed. Contaminant sources affecting the working environment may be external, associated with the elements of HVAC systems, or internal.

7.2.1.1 External sources

Outdoor air is generally less polluted than the system return air. However, problems with reentry of previously exhausted air occur as a result of improperly located exhaust and intake vents or periodic changes in wind conditions. Other outdoor contamination problems include contaminants from other industrial sources, power plants, motor vehicle exhaust, dust, asphalt vapors, and solvents from construction or renovation. Also heat gains and losses through the building envelope due to heat conduction through exterior walls, floor, and roof and due to solar radiation and infiltration can be attributed to effects from external sources.

7.2.1.2 HVAC system

The HVAC system also acts as a pollutant source when it is not maintained properly. Microorganisms breed in various environments present within components (e.g., cooling coils, ducts, humidifier, filter, etc.) of the system and may be distributed throughout the building. Improper maintenance of filters leads to loss of efficiency and reemission of contaminants.

7.2.1.3 Internal sources

This section discusses primarily internal sources of contaminants and other occupational hazards related to the process or the building envelope.

Among the major potential hazards affecting working environment are chemical (airborne contaminants), biological, and physical hazards. Air contaminants are commonly classified as either particulate contaminants or gas and vapor contaminants.[1] Common particulate contaminants include dusts, fumes, mists, aerosols, and fibers.

Dusts are solid particles generated by such processes as handling, crushing, and grinding.
Fumes are formed when material from a volatilized solid condenses in cool air (e.g., welding fumes).
Fibers are solid particles whose length is several times their diameter, such as asbestos.
Gases are formless fluids that expand to occupy the space enclosure in which they are confined. They are atomic, diatomic, or molecular in nature, as opposed to droplets or particles, which are made up of millions of atoms or molecules.
Vapors are the volatile form of substances that are normally in a solid or liquid state at room temperature and pressure. Through evaporation, liquids change into vapors and mix with the surrounding atmosphere.
Mist is a liquid suspended in air. Mists are generated by liquids condensing from a vapor back to a liquid or by a liquid being dispersed by splashing or atomizing.
Aerosols are also a form of a mist characterized by highly respirable, minute liquid particles. They can be formed by atomizing, spraying, or mixing or by violent chemical reactions, evolution of gas from a liquid, or escape of a dissolved gas when pressure is released.
Fogs are fine airborne droplets usually formed by condensation of vapor. Many droplets in fogs are microscopic and submicroscopic and serve as a transition stage between mists and vapors.
Smog commonly refers to air pollution; it implies an air mixture of smoke particles, mists, and fog droplets of such concentration and composition as to impair visibility, in addition to being irritating or harmful. Smog is often associated with temperature inversions in the atmosphere that prevent normal dispersion of contaminants.

Biological hazards include bacteria, viruses, fungi, and other living organisms that can cause acute and chronic infections by entering the body either directly or through breaks in the skin. Physical hazards include thermal parameters (temperature, relative humidity, and velocity) beyond the comfort range, excessive levels of ionizing and nonionizing electromagnetic radiation, noise, vibration, and illumination.

Airborne contaminant movement in the building depends on the type of heat and contaminant sources, which can be classified as (1) buoyant (e.g., heat) sources, (2) nonbuoyant (diffusion) sources, and (3) dynamic sources.[2] With the first type of sources, contaminants move in the space primarily due to the heat energy as buoyant plumes over the heated surfaces. The second type of sources is characterized by contaminant diffusion in the room in all directions due to the

concentration gradient in all directions (e.g., in the case of emission from painted surfaces). The emission rate in this case is significantly affected by the intensity of the ambient air turbulence and air velocity. The third type of sources is characterized by contaminant movement in the space with an air jet (e.g., linear jet over the tank with a push–pull ventilation) or particle flow (e.g., from a grinding wheel). In some cases, the above factors influencing contaminant distribution in the room are combined.

The effect of buoyancy in gases released into the air can be related either to the difference in the molecular weights or to the difference in temperature. To characterize the buoyancy for gases with a molecular density significantly different from the density of air, Elterman proposed a parameter P, with units of $g/(m^3\,K)$[3]:

$$P = C\frac{1 - 29/M_g}{\Delta\theta}, \tag{7.1}$$

where C is the gas concentration in the air, g/m^3; M_g is the relative molecular density of the gas; $\Delta\theta = \theta - \theta_o$ is the air temperature difference between the reference point and the air supply, °C.

According to Elterman, when $P < 5 \times 10^{-3}\,g/(m^3\,K)$, the distribution of gas concentrations along the room height is similar to the temperature distribution, and thus the contaminant removal efficiency and heat removal coefficients will have the same values. When $5 \times 10^{-3} < P < 0.1$, gas concentration in the air of the upper zone is higher than that in the occupied zone, and the gas removal efficiency is higher than the heat removal efficiency. When $P \approx 0.4$, the gas concentration distribution is uniform along the room height. Only when $P > 0.4\,g/(m^3\,K)$ the concentration of the gas, which is heavier than air, is higher in the occupied zone than in the upper zone.

7.2.2 Nonbuoyant contaminant sources

Gases, vapors, and small dust particulates are distributed in the space by airflows produced by supply jets, convective flows, or air currents entering the building through the building apertures and cracks. Also gases and vapors are distributed due to turbulent and molecular diffusion. Distribution of contaminants with airflows is significantly faster (hundreds of times) than distribution due to molecular diffusion.

Theories of hood performance with nonbuoyant pollution sources are based on the equation of turbulent diffusion. The following equation allows the engineer to determine the contaminant concentration decay in the uniform airflow upstream from the contaminant source:

$$C_x = C_o e^{-\frac{V}{D}X}, \tag{7.2}$$

where, X is the distance from the source, m, C_o is the contaminant concentration at the source, mg/m^3, C_x is the contaminant concentration at the distance X from the source, mg/m^3, V is the air velocity in the flow, m/s, and D is the coefficient of the turbulent diffusion, m^2/s.

The value of the coefficient of turbulent diffusion, D, depends on the air change rate in the ventilated space and the method of air supply. Studies by Posokhin[2] show that approximate D values for locations outside supply air jets is equal to $0.025\,m^2/s$. Air disturbance caused by operator or robot movement results in an increase in the D value of at least two times. Studies by Zhivov et al.[4] showed that the D value is affected by the velocity and direction of cross-drafts against the hood face, and the presence of an operator; for example, for a cross-draft directed along the hood face with velocity $v = 0.5$ m/s with $D = 0.15\,m^2/s$ (with the presence of an operator), an increase to $v = 1.0$ m/s results in $D = 0.3\,m^2/s$.

7.2.2.1 *Contaminant emission by a process*

The quantity of contaminant (fume, oil mist, VOC, gas, or particulates), G, kg/h, generated in the space can be calculated using one of the following equations:

$$G = R_1 \times T_{process}, \tag{7.3}$$

where R_1 is a fume, oil mist, VOC, gas, or particulate generation rate, kg/min, and $T_{process}$ is an average contaminant release time per hour (e.g., arc time for the welding process), min/h; or

$$G = R_2 \times U, \tag{7.4}$$

where R_2 is a contaminant generation rate per production unit, kg/(production unit), and U is an average production unit output (e.g., units/h) for the given process.

For a welding process, for example, total welding fume generation rate R_1 is a fume (gas, particulate) generation rate, kg/min; T_{arc} is an average arc time for the welding process used, min/h; R_2 is a fume generation rate, kg, per kg of electrodes used; and U is average electrode usage, kg/h, in the given welding process. R_1 and R_2, percentages of the critical components in the fume for the typical welding processes, and an average arc time for the typical welding processes are listed in AWS.[5]

7.2.2.2 *Gas and vapor emission through looseness in process equipment and pipelines*

When the pressure inside the equipment/pipeline is greater than the room pressure, this emission can be calculated using the equation suggested by Repin[6]:

$$G = kCV\sqrt{\frac{m}{T}}, \tag{7.5}$$

TABLE 7.1 Coefficient C for Eq. (7.5).

Gas pressure, atmospheres	<2	2	7	17	41	161	401	1001
C	0.121	0.166	0.182	0.189	0.152	0.289	0.297	0.370

where k is a reserve coefficient that varies from 1 to 2 depending on the state of the equipment, C is a coefficient that depends on the gas pressure inside the equipment (see Table 7.1), V is the internal volume of the equipment/pipeline with an excessive gas pressure (m^3), m is the molecular weight of gas/vapor, and T is the gas/vapor temperature inside the equipment (°C).

7.2.2.3 Gas and vapor emission processes from an open liquid face

Emission from an open liquid face (e.g., open tanks and liquid spills on the floor surface) can be evaluated using equations based on criteria relations and empirical data. Assuming that the heat and mass transfer processes can be described using similar differential equations, the criteria equation describing the evaporation process will be similar to one describing the heat transfer[3]:

$$\mathrm{Nu} = C(\mathrm{Gr}\cdot\mathrm{PR})^n, \tag{7.6}$$

where Nu, Gr, and Pr are the Nusselt, Grashof, and Prandtl numbers, respectively, for evaporation processes:

$$\mathrm{Nu} = \frac{g_0 d}{D}, \quad Pr = \frac{v}{D}, \quad \mathrm{Gr} = \frac{g d^3(\rho_0 - \rho_1)}{v^2 \rho_1},$$

where, g_0 is the mass rate of liquid evaporation (liquid mass evaporated from unit of surface area in a unit of time and related to the unit of vapor concentration at the surface and in the ambient air), m/s, d is the characteristic dimension, m, D is the molecular diffusion coefficient, m^2/s, v is the kinematic viscosity coefficient, m^2/s, ρ_0 is the ambient air density, kg/m^3, and ρ_1 is the air density near the liquid surface at the surface temperature, kg/m^3

Film regime: in the film regime, there is a thick film of undisturbed air formed adjacent to the liquid surface (e.g., evaporation from the surface of small mercury droplets). In Eq. (7.6), $\mathrm{Gr}\times\mathrm{Pr} < 1$, $n = 0$, and Nu is constant.

The mass flow rate, G, g/s, from the surface can be evaluated using

$$G = 2\,Dd(C_1 - C_0), \tag{7.7}$$

where C_1 and C_0 are vapor concentration, g/m^3, in the air adjacent to the liquid surface and in the ambient air.

Laminar regime: in the laminar regime, $2\times10^2 < \mathrm{Gr}\times\mathrm{Pr} < 2.3\times10^8$, and $n = 1/4$. The mass flow rate, G, g/s, from the surface can be evaluated using

$$G = FAd^{-1/4}D^{1/2}(C_1 - C_0)^{5/4}\left|\left(M_{\mathrm{air}}/M_l - 1\right)\right|^{1/4}, \tag{7.8}$$

where M_{air} is the relative molecular weight of air, M_l is the relative molecular weight of vapor evaporated from the liquid surface, and A is the surface area, m^2.

For a horizontal surface, $F = 0.334$ when $M_{\mathrm{air}} > M_l$, and $F = 0.184$ when $M_{\mathrm{air}} < M_l$. For a wet vertical surface, $F = 0.224$.

Turbulent regime: in the turbulent regime, $\mathrm{Gr}\times\mathrm{Pr} > 2.3\times10^8$, and $n = 1/3$. This regime may occur only when the area of evaporating liquid is very large (tens of square meters).

The mass flow rate, G, g/s, from the surface can be evaluated using

$$G = FAD^{1/3}(C_1 - C_0)^{4/3}\left|\left(M_{\mathrm{air}}/M_l - 1\right)\right|^{1/3}. \tag{7.9}$$

For a horizontal surface, $F = 150$ when $M_{\mathrm{air}} > M_l$, and $F = 75$ when $M_{\mathrm{air}} < M_l$. For a wet vertical surface, $F = 113$.

7.2.3 Emission from heat sources

Heating and cooling load calculation for HVAC system design is based on the heat balance principle. For the given building, room, or independent building zone, heat balance components should be established and analyzed. The major heat sources and sinks in industrial buildings are

- heat losses and gains by heat conduction through the building envelope,
- heat gains by solar radiation,
- heat losses and gains with infiltration and exfiltration through cracks in the building envelope,
- heat gains from people activity,
- heat gains from lighting,
- heat gains from process equipment powered by electric motors,
- heat gains from processes with conversion of mechanical energy into heat,
- heat gains and losses for heating or cooling raw materials and parts brought into or taken out of the building, melted metal solidification, vapor condensation, or liquid evaporation, and
- heat losses with vehicles entering the building, etc.

The total heat gains and losses, ΔW, which should be compensated by the HVAC system, can be determined by

$$\Delta W = \sum_1^n W_{\mathrm{gains}_i} - \sum_1^m W_{\mathrm{losses}_i}. \tag{7.10}$$

Heat gains and losses can be only sensible or sensible and latent. Sensible heat gains result from conduction, convection, and/or radiation. Latent heat gains

TABLE 7.2 Coefficient ψ for Eq. (7.5).

	Surface temperature, θ_{surf} (°C)										
€	40	50	60	100	150	200	300	500	800	1000	1200
0.8	0.42	0.44	0.45	0.48	0.45	0.4	0.32	0.2	0.1	0.1	0
0.5	0.52	0.55	0.58	0.59	0.56	0.51	0.42	0.29	0.14	0.1	0.1
0.2	0.73	0.76	0.77	0.78	0.76	0.73	0.65	0.59	0.3	0.2	0.14

TABLE 7.3 Coefficient $\varphi_{horizontal}$.

	B/H			
Source location in the room	1	2	3	4
Along the room axis	0.3	0.12	0.04	0
Between the axis and the wall	0.38	0.17	0.11	0.07
Close to the wall	0.51	0.3	0.23	0.16

TABLE 7.4 Coefficient $\varphi_{vertical}$.

	B/H			
Source location in the room	1	2	3	4
Source along the room axis	0.8	0.7	0.65	0.6
Between the axis and the wall	0.8	0.72	0.67	0.63
Close to the wall	0.85	0.75	0.7	0.68

occur when moisture is added to the space (e.g., from evaporation).

7.2.3.1 Sensible heat sources

The total heat load is introduced by each source by convection and radiation[7]:

$$W_0 = W_{conv} + W_{rad} = \psi W_0 + (1-\psi)W_0. \quad (7.11)$$

The total radiant component of the heat load introduced by each source into the space can be divided between the upper and the lower (occupied) zones of this space[7]:

$$\begin{aligned} W_{rad} &= W_{rad\ low} + W_{rad\ up} \\ &= \phi(1-\psi)W_0 + (1-\phi)(1-\psi)W_0. \end{aligned} \quad (7.12)$$

The total convective component of the heat load introduced by each source is

$$W_{conv} = W_{conv\ low} + W_{conv\ up} = \beta\psi W_0 + (1-\beta)\psi W_0, \quad (7.13)$$

where ψ, φ, and β are nondimensional coefficients reflecting the portion of the convective component of the total heat load released into the space for each heat source, the portion of the radiant component of the total radiant heat load in the low zone, and the portion of the convective component of the total convective heat load in the low zone, respectively.

Coefficients ψ, φ, and β vary within a range from 0 to 1. The coefficient ψ value depends on the heated surface temperature and emittance and can be estimated from Table 7.2.

The coefficient φ value depends on the source location in the ventilated room (e.g., in the center, close to the wall, etc.) and the source dimensions relative to the room size. Coefficient φ values for small sources (<1/10 of the room size) can be estimated using Tables 7.3 and 7.4, where B and H are the width and height of the room, respectively.

Coefficients ψ and φ for some typical heat sources are as follows:

for a sitting or standing person $\psi = 0.57$, $\varphi = 0.63$ and for machining equipment $\psi = 0.5$, $\varphi = 0.6$.

The β coefficient value depends on the supply air method (e.g., $\beta = 0$ with displacement and natural ventilation, $\beta = 1$ with convective plume dissipating within the occupied zone due to interaction with supply jets, airflows created by moving objects, etc.).

7.2.3.2 Heat gain from process equipment

Heat load from hot process equipment with a relatively simple configuration (e.g., tanks with hot water, solution, or oil) can be calculated using the following equation:

$$W_{eq} = \alpha_{conv}A_{conv}(\theta_{surf} - \theta_0) + \alpha_{rad}A_{rad}(\theta_{surf} - \theta_0), \quad (7.14)$$

where θ_{surf} is the surface temperature, °C; θ_0 is the room air temperature, °C; A_{conv} is the convective heat exchange surface area, m^2; and A_{rad} is the radiant heat exchange surface area, m^2. In general, $A_{rad} \le A_{conv}$ (i.e., the ratio K_A is the $A_{rad}/A_{conv} \le 1$). The convective and radiant heat flux, α_{conv} and α_{rad}, $W/(m^2\,°C)$, are

$$\begin{aligned} \alpha_{conv} &= K\sqrt[3]{\theta_{surf} - \theta_0} \\ \alpha_{rad} &= \epsilon_0\, C_0 b, \end{aligned} \quad (7.15)$$

$$W_{eq} = \epsilon_0 C_0 b + K\sqrt[3]{\theta_{surf} - \theta_0}(\theta_{surf} - \theta_0)A, \quad (7.16)$$

where $C_0\epsilon_0$ is the surface emittance, $W/m^2\,°C$, and K and b are coefficients depending on the surface temperature (see Table 7.5).

For horizontal surfaces facing upward, coefficient K should be increased by 30%, and for horizontal surfaces facing downward, it should be decreased by 30%, compared with the data from Table 7.5.[8]

The total unit heat gain from the process equipment, W_{eq}, can be evaluated using the graph in Fig. 7.1.

TABLE 7.5 Values of K and b coefficients.[8]

θ_{surf} (°C)	b	K
20	1.01	1.67
80	1.36	1.6
180	2.3	1.53
280	3.3	1.47
380	4.87	1.41
480	6.92	1.36
580	9.43	1.33
980	25.5	1.19

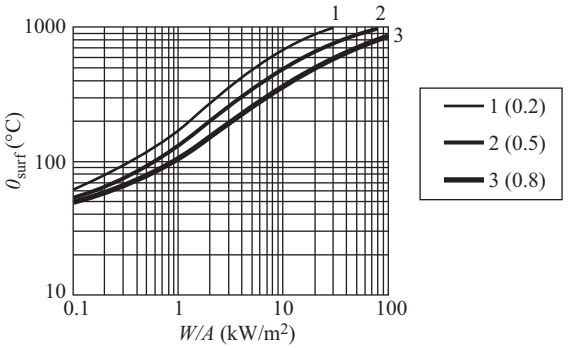


FIGURE 7.1 Relationship between the heat source surface temperature, θ_{surf}, heat flux, W/A_{surf}, and the heat source emittance ϵ/K_A: 1—$K_A = 0.2$; 2—$K_A = 0.5$; and 3—$K_A = 0.8$.

7.2.3.3 Heat gain from lighting

This can be calculated from

$$W_{light} = K_u K_{sa}\, W_0, \tag{7.17}$$

where K_u is the lighting use factor (applied to lighting when use is known to be intermittent), K_{sa} is the special allowance factor, and W_0 is the total light wattage. The special allowance factor is for fluorescent fixtures and fixtures that are either ventilated or installed so that only part of their heat contributes to a space heat load.[1]

7.2.3.4 Heat gain from equipment operated by electric motors

Heat gain related to electric motors is calculated as[1]

$$W_{eq} = \left(\frac{P}{E_M}\right) F_{UM} F_{LM}, \tag{7.18}$$

where P is the motor power rating, W; E_M is the motor efficiency, expressed as a decimal fraction <1.0; F_{UM} is the motor use factor, 1.0 or a decimal fraction <1.0; and F_{LM} is the motor load factor, 1.0 or a decimal fraction <1.0.

7.2.3.5 Heat loss/gain for heating or cooling materials and parts brought into or taken out of the space

This heat loss/gain is calculated as

$$W_{mat} = c(\theta_1 - \theta_0)GB, \tag{7.19}$$

where c is the specific heat, W/(m^3 °C); t_1 is the material initial temperature, °C; G is the material mass, kg; B is the share of heat load lost/gained during the time period ΔZ from the time the material was brought in (see Fig. 7.2), $B = f(\text{Fo})$; and $\text{Fo} = \Delta Z/cGR$, where

$$R = \frac{G}{\rho\lambda A^2} + \frac{1}{\alpha_0 A}, \tag{7.20}$$

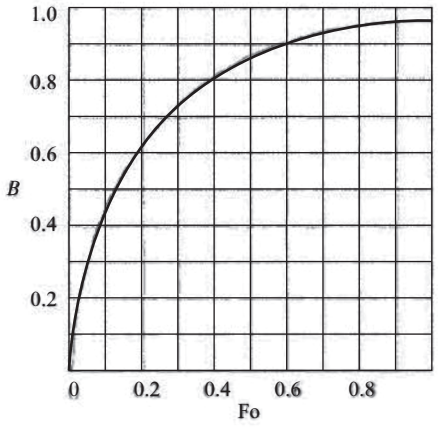


FIGURE 7.2 Influence of Fo criteria on coefficient B.[8]

where λ is the heat conductivity coefficient, W/(m °C) (increased by 25% for loose materials), and α_0 is the total heat flux coefficient, W/(m^2 °C).

7.2.3.6 Heat load from molten metal cooling

This can be calculated as

$$W_{molten\ mat} = [c_l(\theta_1 - \theta_s) + c_s(\theta_s - \theta_0)]G, \tag{7.21}$$

where c_l is the specific heat of the material in the liquid phase, W/(m^3 °C); c_s is the specific heat of the material in the solid phase, W/(m^3 °C); and θ_S is the temperature of solidification, °C.

7.2.4 Sources of dust

Many industries, such as chemical, printing, metallurgy, fine ceramics, and welding operations, have processes generating dust. Crushing, blasting, material transporting and processing, finishing, and other operations result in dust release into the building

space. Other particle sources may include construction and renovation, deteriorated insulation, cleaning, etc. The dust resulting from each of these processes differs in its chemical and physical characteristics. Particle size, shape, density, and release method influence the distribution of dusts inside the space, and the methods of particle control and air cleaning technology used.

Dusts in the work environment vary in particle size from 0.1 to 25 μm (Fig. 7.3). Respirable particle sizes vary from less than 1 to 10 μm. Large particles with a size of tens of microns deposit on surfaces, while smaller particles remain suspended in the air and migrate in the space with airflows.

Grinding, polishing, and other finishing operations produce particles imparted with some momentum and thus are considered to be dynamic sources. A particle ejected into still air with an initial velocity will travel some distance (stopping distance) before decelerating to rest due to drag forces. The data from Hinds show the difficulty in throwing even fairly large particles an appreciable distance in still air.[9] When the receiving hood is designed, any particles that are in the inhalable range should be considered immovable.[10]

Granular and other bulk materials processing and conveying, bag emptying and disposal, and similar operations air into the process enclosure. Induced air picks up dust. If the system component, such as a bin, is tight, the induced air will reverse its path and carry the entrained dust back through the upstream opening, as shown in Fig. 7.4.[10]

Resistance and arc welding operations and plasma and laser cutting produce fumes by expulsion or evaporation of the base material, coating, and electrode wear. Larger particles deposit on the surrounding surfaces, while smaller particles move upward with convective flows. Specific contaminants associated with different welding and cutting operations are listed in AWS.[5]

To prevent the transport of dusts in the working environment, proper exhaust ventilation can be used, such as the local exhaust ventilation. The calculation of the minimum volume flow rate for supply and exhaust air can be found in ASHRAE Handbook[11]. In addition, the chosed ventilation systems must prevent dust pocketing in the equipment.

7.2.5 Sources of moisture emission

Moisture control in industrial buildings is necessary to avoid problems related to

- Human comfort, health, and productivity,
- process requirements (e.g., in pharmaceutical and semiconductor manufacturing, painting processes, plastic injection molding, and breweries),
- building durability, by preventing decay of wood-based materials, corrosion of metals, and spalling of

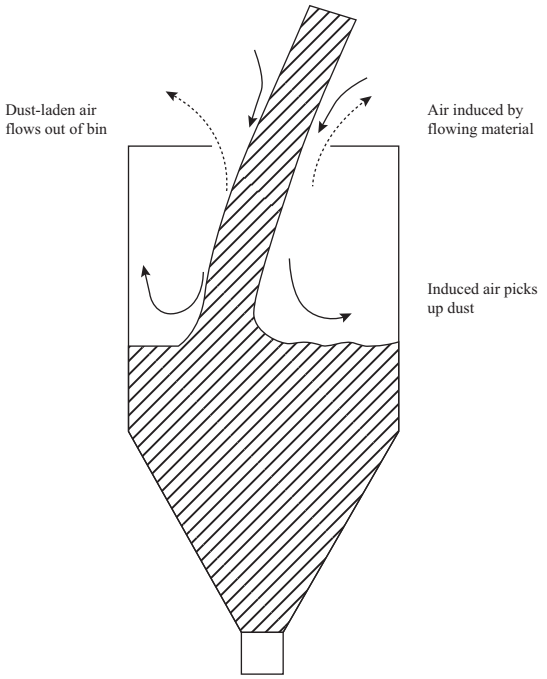


FIGURE 7.4 Air induction into a bin from flowing granular materials.

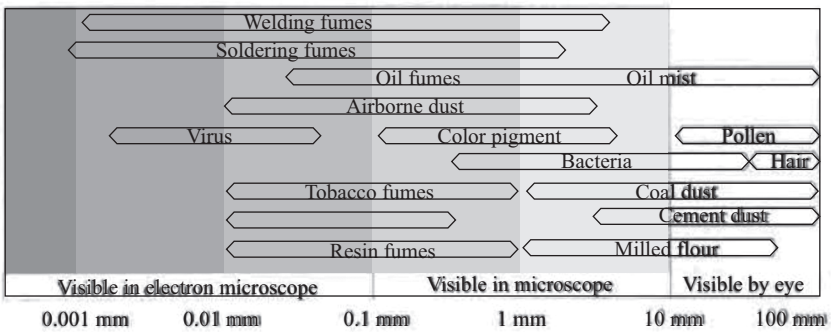


FIGURE 7.3 Range of particle sizes for typical dusts.

masonry and concrete caused by freeze–thaw cycles, and
- degradation of the thermal resistance of building materials.

Principal sources of moisture in the building include

- evaporation from wet surfaces and open tanks,
- steam leakage from process equipment or pipelines,
- evaporation from people breathing and from perspiration,
- permeation through floors, walls, and ceiling,
- desorption from moist products,
- generation from combustion, that is, open flame in the space,
- air infiltration through leaks, holes, and door openings, and
- untreated outside air supplied by the mechanical or natural ventilation system.

7.2.5.1 Moisture diffusion through the building envelope

The permeation moisture load through building materials can be calculated using

$$M = pA\,\Delta\mathrm{VP}, \tag{7.22}$$

where p is the permeance factor, g/(h m^2 kPa), A is the surface area, m^2; ΔVP is the difference in vapor pressure across the material, kPa.

7.2.5.2 Evaporation from wet surfaces and open tanks

The amount of water evaporated from wet surfaces (i.e., process equipment, floors) or water tanks, M, kg/h, is proportional to the difference in vapor pressure between the surface and air, the surface temperature, and the air velocity across the water surface[12]:

$$M = 7.4(a + 0.017V)(P_2 - P_1)101.3\frac{A}{P_b}, \tag{7.23}$$

where M is the evaporation load, kg/h; V is the air velocity across the surface, m/s; P_2 is the water vapor pressure in the air above the surface, kPa; P_1 is the vapor pressure of air saturated at the water temperature, kPa; A is the total surface area wetted or of the water face, m^2; P_b is the barometric pressure, kPa; and a is the coefficient reflecting the influence of air movement. For air temperature between 15°C and 30°C, a can be evaluated as shown in Table 7.6.

If the water temperature is held constant and the water is still, Table 7.7 can be used to evaluate the temperature of the water surface (at room air temperature 20°C and RH = 70%). When the water is stirred, the surface temperature can be assumed to be equal to the mean water temperature.

TABLE 7.6 Determination of a for various water temperatures.

Water temperature								
°C	30	40	50	60	70	80	90	100
a	0.022	0.03	0.03	0.04	0.041	0.05	0.051	0.06

TABLE 7.7 Water surface temperature evaluation.

Water temperature (°C)	20	30	40	50	60	70	80	90	100
Water surface temperature (°C)	18	28	37	45	51	58	69	82	97

Evaporation load from wet surfaces or floors can be evaluated using the following equation[12]:

$$M \approx (6 \sim 6.5)(\theta_0 - \theta_w)A, \tag{7.24}$$

where θ_0 is the dry-bulb room air temperature, °C and θ_w is the wet-bulb room air temperature, °C.

7.2.5.3 Moisture from air leaks through cracks and apertures

Moisture load from infiltrating air can be evaluated as

$$M = G_{inf}(m_{\mathrm{out}} - m_0), \tag{7.25}$$

where G_{inf} is the infiltrating air flow rate, kg/h; m_{out} and m_0 are the moisture content in outside and inside air, g/kg.

7.2.5.4 Moisture from personnel

The moisture release rate from people's respiration and perspiration can be calculated as follows[13]:

$$M = (P_A \times F_A) + (P_B \times F_B) + (P_C \times F_C) + (P_D \times F_D), \tag{7.26}$$

where (P_A, F_A), (P_B, F_B), (P_C, F_C), and (P_D, F_D) are the evaporation rate and number of people seated (A), standing (B), with light activity (C), and with moderate activity (D) (see Fig. 7.5).

7.2.5.5 Moisture from combustion

The amount of water vapor resulting from combustion varies with composition of the burned gas. When the value is unknown, one can estimate that each cubic meter of gas burned produces 42 g (650 grains) of water vapor.

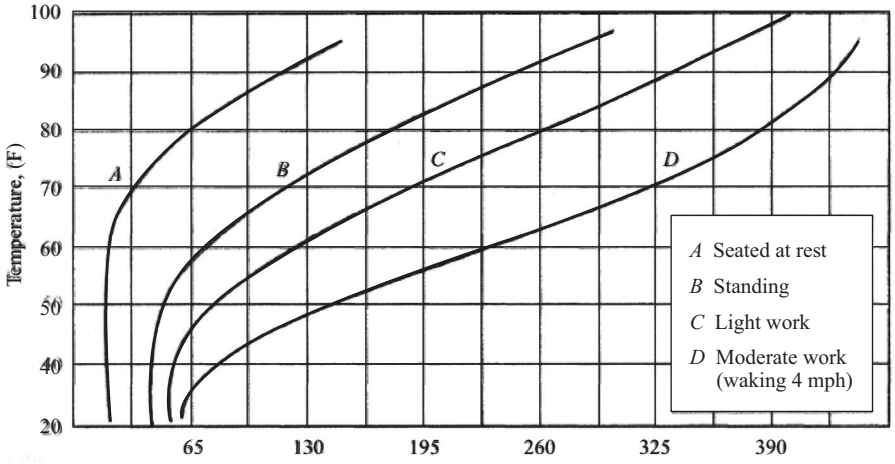


FIGURE 7.5 Moisture evaporation, g/h, per average man.

7.2.6 Source of mist emission

7.2.6.1 Mechanism of mist generation

Mists consisting of droplets are generated by vapor condensation, bubble burst, or splashing dispersion. Different droplet generation types cause the differences of physical properties of droplets, such as initial diameter and velocity, which may lead to the control strategy to vary. The properties of droplets generated by splashing or atomizing are highly relevant to the generation devices and have huge differences. Thus this section focuses on the droplet generation due to vapor condensation and bubble burst.

7.2.6.2 Vapor condensation

The formation of droplet due to condensation contains two processes: the nucleation of droplets and droplet growth. Currently engineering projects still use the classical Becker–Döring theory[14] as the droplet nucleation model. The classical nucleation rate is[15]

$$J_{CL} = q_c \frac{\rho_g^2}{\rho_l}\left(\frac{2\sigma}{\pi m^3}\right)^{1/2} \exp\left(-\frac{4\pi r^{*2}\sigma}{3kT_g}\right) \tag{7.27}$$

where q_c is the condensation coefficient; ρ_g is the gas-phase density, kg/m^3; ρ_l is the liquid-phase density, kg/m^3; σ is the surface tension of the molecular aggregation, N/m; m is the mass of a single molecule, kg; r^* is the critical droplet size, m; k is Boltzmann constant, and T_g is the gas-phase temperature, K.

The growth rate of a droplet depends on Kn (Kn is the ratio of the size of the droplets and the mean free path of the gas molecules) and could be mainly divided into three periods: the radius of the droplets is much smaller than ($\mathrm{Kn} \gg 1$), approximately equal to ($\mathrm{Kn} \approx 1$), or much larger than($\mathrm{Kn} \ll 1$) the mean free path of the gas molecules.[16] The growth rate model, based on mass and heat transfer coupling, proposed by Han et al. is[17] ($\zeta$is calculated based on Kn):

$$\frac{dm}{dt} = 4\pi r^2 \xi\left(\frac{p}{RT_g} - \frac{p_{s(T_r,r)}}{RT_r}\right) \tag{7.28}$$

$$h_{fg}\frac{dm}{dt} = 4\pi r^2 h\left(T_r - T_g\right) \tag{7.29}$$

where m is the droplet mass, kg; r is the droplet diameter, m; ζ is the mass transfer coefficient; p is the gas-phase pressure, Pa; R is the gas constant; T_g is the gas temperature, K; $p_s(T_r, r)$ is the saturation pressure of water surface, Pa; T_r is the droplet temperature, K; h_{fg} is the condensation latent heat; and h is the convective heat transfer coefficient between water droplets and surrounding steam.

This model is suitable for determining the growth rate of a droplet with a diameter within the above three ranges ($\mathrm{Kn} \gg 1$, $\mathrm{Kn} \approx 1$, and $\mathrm{Kn} \ll 1$). The calculation of ζ is different in the three ranges of Kn.

7.2.6.3 Bubble burst

Some chemical reactions produce bubbles in the liquid. When a bubble bursts at the free surface of a liquid, it usually produces two types of drops: film drops (Fig. 7.6A) which are formed during the large bubble burst and jet drops which are formed after the collapse of the small bubbles (Fig. 7.6B).[18]

7.2.6.4 Source characteristics

The diameter of droplets directly formed by condensation is usually less than 1 μm. The bubble burst may produce larger droplets. According to the experimental data, the diameter (D_d in μm) and speed (S_d in m/s) of film drops can be achieved as follows[19]:

$$D_d = 2 \times 3.57\left\{2\gamma R_0(1-\cos(31.3))/\left(S_f^2\rho\pi\sin(31.3)\right)\right\}^{1/2} \tag{7.30}$$

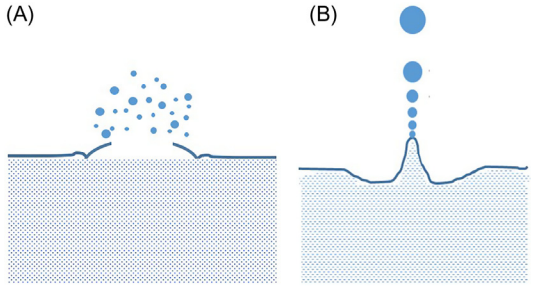


FIGURE 7.6 Examples of (A) film drops and (B) jet drops. Source: *Characteristics*.

$$S_d = 21.6\exp(-D_b/7.862) \tag{7.31}$$

where γ is surface tension; R_0 is the cap's radius of curvature; S_f is the film roll-up speed; ρ is the water's density; and D_b is the bubble diameter.

For the jet droplets, the following empirical correlations were derived, from the experimental work[20] on sea water bubble at 27°C–29°C, to predict the radius of the first three drops (R_{drop} in μm) after the collapse of a microbubble with a radius of R_{bubble}.

$$\begin{aligned} R_{drop1} &= 0.0345R_{bubble}^{1.206} \\ R_{drop2} &= 0.0172R_{bubble}^{1.316} \\ R_{drop3} &= 0.0006R_{bubble}^{1.796} \end{aligned} \tag{7.32}$$

The following empirical equations can be used to approximate the speed of ejection of a jet drop (s_e in m/s) based on the size of the bubble.[20]

$$\begin{aligned} s_{edrop1} &= 12.12\,exp(-1.390\times10^{-3}R_{bubble}) \\ s_{edrop2} &= 9.64\,exp(-2.806\times10^{-3}R_{bubble}) \\ s_{edrop3} &= 6.53\,exp(-2.916\times10^{-3}R_{bubble}) \end{aligned} \tag{7.33}$$

7.2.6.5 *Droplet evaporation and movement*

Droplet movement in the building rooms is accompanied by evaporation. The water vapor mass is diffused due to the difference between the partial pressure of water vapor on the surface of a droplet and the water vapor of the surrounding air. The evaporation of water, the evaporation rate, energy equation, and momentum equation could be expressed as[21]:

$$\frac{dm_p}{dt} = -Sh\frac{2\pi pD_PD_\infty M_vC}{RT_\infty}ln\left(\frac{p-p_{vap.d}}{p-p_{vap.\infty}}\right) \tag{7.34}$$

$$m_pc_p\frac{dT_p}{dt} = 4\pi r_p^2K_g\frac{T_\infty - T_p}{r_p}Nu - L_v\frac{dm_d}{dt} \tag{7.35}$$

$$\frac{\overrightarrow{dV_p}}{dt} = \vec{g} - \frac{3C_d\rho_g|\overrightarrow{V_p}-\overrightarrow{V_g}|(\overrightarrow{V_p}-\overrightarrow{V_g})}{8\rho_pr_p} \tag{7.36}$$

where c_p is the droplet heat capacity, J/(kg K); m_p is the mass of the droplet, kg; D_p is the diameter of droplet, m; M_v is the molecular weight of water vapor, kg/mol; D_∞ is the binary diffusion coefficient far from the droplet, m^2/s; R is the universal gas constant, J/(mol · K); p is the total pressure, Pa; C is a correction factor because of the temperature dependence of the diffusion coefficient; g is gravity acceleration, m/s^2; ρ_p is density of droplet, kg/m^3; and C_d is the drag coefficient. The Sherwood and Nusselt numbers for moving droplets are used to consider the effect of relative velocity of airflow and droplet on mass transfer and heat transfer.

For a solution drop, the presence of the solute causes the surface water vapor partial pressure of the droplet to be lower than that of the pure water droplet under the same conditions. The partial pressure of water vapor on the surface of the droplet is one of the most important factors affecting droplet evaporation. The partial pressure of water vapor on the surface of droplet can be modified according to the theory of equal chemical potential of gas and liquid[22]:

$$P_{vap,d} = a_wP_{sat,w} \tag{7.37}$$

where $P_{sat,w}$ is the saturation vapor pressure of water droplet, Pa and a_w is the water activity.

7.2.7 Explosive gases, vapors, and dust mixtures

Some gases, vapors, and dust mixtures with air or oxygen may produce explosions. For explosion to occur, a flammable gas or combustible material mixture with air or oxygen must be in proportion within the explosive ("flammable") limits, and an ignition source must be present and have sufficient energy to ignite the explosive mixture. Ignition can be caused by electrical arc, spark, or a hot surface. The surface temperature that can cause ignition varies for different gas mixtures. Ignition of carbon disulfide and ethyl nitrite, for example, requires a surface temperature of only 85°C.

Flammable gases and vapors include acetylene, hydrogen, butadiene, ethylene oxide, propylene oxide, acrolein, ethyl ether, ethylene, acetone, ammonia, benzene, butane, cyclopropane, ethanol, gasoline, hexane, methanol, methane, natural gas, naphtha, and propane.

Combustible dusts include metal dust (e.g., aluminum, magnesium, and their commercial alloys), carbonaceous dust (e.g., carbon black, charcoal, and coal), flour, grain, wood, plastics, and chemicals.

Flammable gases and vapors or combustible dust may be present in quantities sufficient to produce explosive or ignitable mixtures due to[23]

- Leakage from maintenance operations, breakdown of equipment, or faulty operation of equipment.
- Escape in the event of an accidental rupture or breakdown of equipment, or in abnormal equipment operation.

Among common areas where explosion can occur are coal mines, petrochemical plants, chemical plants, paint shops, grain handling industry, etc. Explosive limits for gases and vapors are expressed as percentages and may be defined as minimum and maximum concentrations of a flammable gas or vapor between which ignition occurs.[24] Concentrations below the lower explosive limit (LEL) are too lean to burn, while those above the upper explosive limit are too rich. Table 7.8 lists explosive limits for some common gases.

Upper and LELs EL_{mix} for mixtures of several gases can be calculated using the Le Chatelier equation[25]

$$EL_{mix} = \frac{100}{(P_1/EL_1) + (P_2/EL_2) + (P_3/EL_3) + \cdots}, \quad (7.38)$$

where P_i is the proportion of gas i in the gas mixture and EL_i is the explosive limit for gas i.

Dust particles have a LEL expressed in mg/m^3 and almost no upper limit. Examples of LEL for dusts are polystyrene, 0.02 mg/m^3; corn starch, 0.04 mg/m^3; and coal, 0.055 mg/m^3.

A liquid which is not considered as flammable may still have an explosive potential. An example is dichloromethane or methylene chloride, often used in paint strippers, which evaporates very quickly. It is not flammable, but its vapors may be explosive (explosive limits 12%–22%).

TABLE 7.8 Explosive limits of some gases in air and oxygen mixtures.[24]

Gas	Explosive limits in air	Explosive limits in oxygen
Ammonia	15.50–27.00	13.50–79.00
Carbon monoxide	12.50–4.20	15.50–93.90
Hydrogen	4.00–74.20	4.65–93.90
Methane	5.00–15.00	5.40–59.20
Diethyl ether	1.85–36.50	2.10–82.00
Propylene	2.00–11.10	2.10–52.80

7.2.8 Identification of contaminant sources

The earlier sections introduce the contaminant sources that affect the working environment. In this section, their identification is discussed. Identification of the contaminant sources is critical for the source control. Only when the contaminant sources are identified, can the emergency actions be appropriately taken. Current measurements such as sensors can only tell the concentration locally, where, when, and how the contaminants have been released are still required to be known, inference of the contaminant sources based on the limited available concentrations constitutes inverse modeling.

Recently some pioneering studies started to inversely identify the sources. The adopted strategies can generally be divided into two categories, namely, directly reversing transport equations to track a source and solving forward transport equations to match a source. For the first strategy, quasireversibility method[26,27] and pseudoreversibility method[28] were proposed to inversely identify location of a pollutant source. A linear scaling method[28] was utilized to determine the total amount of an instantaneously released source, and a matrix inversion method[29] was proposed to determine the temporally varied release rates of a gaseous source, with the source location as the known input.

The first strategy does not require extensive a priori source information but may suffer from numerical instability. Hence some researchers turned to the second strategy. Different methods have been used by the second strategy to locate a source, such as solving the adjoint equation of pollutant source location probability,[30,31] applying neural network to match the pollutant source concentration,[32,33] or using two nondimensional indices, response coefficient to supply air and response coefficient to contaminant source, to help judge the position of a source.[34] The second strategy is numerically stable, but may produce significant computing expense.

Compared with a single source, identification for multiple sources that emit the same pollutant is more challenging. The monitored concentration at a sensor is the superposition of concentrations from each source, so the inverse modeling must decouple the combined concentration response into that from each source to determine the pollutant sources. A reversed time-marching method[35,36] was utilized to identify the locations of the multiple sources at fixed constant rates. Girault et al.[37] applied a reduced model that linked the monitored concentrations with source rates to estimate the varied pollutant emission rates, with the source locations to be known. Cai et al.[38] determined multiple constant release source locations and constant emission rates by matching concentrations. Wei et al.[39] proposed an inverse model based on the matrix inversion and

Bayesian model to determine the locations and temporal varied release rate profiles of multiple sources. Next, the inverse model based on the matrix inversion and Bayesian model is introduced briefly to illustrate the identification of the multiple sources.

The governing equation for each specific pollutant source in the total n sources all has the following form:

$$\frac{\partial(\rho c_i)}{\partial t} + \mathrm{div}(\rho \boldsymbol{u} c_i) = \mathrm{div}\left[\Gamma \mathrm{grad}(c_i)\right] + q_i \tag{7.39}$$

where ρ is the air density, c_i is the concentration response due to the ith pollutant source, t is time, $\boldsymbol{u}$ is air velocity vector, Γ is effective mass diffusion coefficient, q_i is the temporal release rate of the ith pollutant source, and i is from 1 to n, where n is the total number of the pollutant sources.

If the flow field is fixed, the concentration response of an arbitrary release rate q_i can be expressed as the convolution integral between the source release rate and the concentration response of a unit impulse release, shown as[29,40]:

$$c_i(t) = \int_{-\infty}^{+\infty} q_i(\tau) F[\delta(t-\tau)] d\tau \tag{7.40}$$

where $F[\delta(t-\tau)]$ is the concentration response of a unit impulse release at time $t=\tau$, which is also called the response factor.[40] By discretizing Eq. (7.40), the concentration response for an arbitrary release rate q_i can be written as[40]:

$$c_{i,t_l} = \sum\nolimits_{k=0}^{l} q_{i,t_{l-k}} F_{t_k} = q_{i,t_l} F_{t_0} + \cdots + q_{i,t_{l-k}} F_{t_k} + \cdots + q_{i,t_0} F_{t_l} \tag{7.41}$$

where t_l is the time at the lth time step. Eq. (7.41) can be rewritten into a matrix:

$$\boldsymbol{c}_i = \boldsymbol{A}_i \boldsymbol{q}_i \tag{7.42}$$

where $\mathbf{c}_i$ is a concentration vector (composed of discrete concentrations at different time steps) due to the presence of the release rate vector $\mathbf{q}_i$ (composed of discrete rates at different time steps), $\mathbf{A}_i$ is the linear matrix that describes the cause–effect relationship between the release rate of the ith pollutant source and the exhibited pollutant concentration. The matrix is subject to the flow pattern, pollutant source location, and the location at which the concentration is concerned. However, the matrix is independent of the pollutant release rate profile.

When there are multiple sources, the total concentration at a point in space is the sum of the component concentration of each source, so the total concentration at a specific point j can be written as:

$$\boldsymbol{c}^j = \sum\nolimits_{i=1}^{n} \boldsymbol{c}_i^j = \sum\nolimits_{i=1}^{n} \boldsymbol{A}_i^j \boldsymbol{q}_i, \tag{7.43}$$

where $\boldsymbol{c}^j$ is the total concentrations at point j from n sources, $\boldsymbol{A}_i^j$ is the matrix for the relationship between the release rate vector $\boldsymbol{q}_i$ from the ith pollutant source and the pollutant concentration $\boldsymbol{c}^j$ at point j in space. To prevent confusion, the pollutant source index i is expressed using a subscript, while the concentration concerning point index j is noted with a superscript.

Then the total concentrations at n different concerning points in space due to n different sources can be formulated into:

$$\begin{bmatrix} \boldsymbol{c}^1 \\ \vdots \\ \boldsymbol{c}^j \\ \vdots \\ \boldsymbol{c}^n \end{bmatrix} = \begin{bmatrix} \boldsymbol{A}_1^1 & \cdots & \boldsymbol{A}_i^1 & \cdots & \boldsymbol{A}_n^1 \\ \vdots & \ddots & \ddots & \ddots & \vdots \\ \boldsymbol{A}_1^j & \ddots & \boldsymbol{A}_i^j & \ddots & \boldsymbol{A}_n^j \\ \vdots & \ddots & \ddots & \ddots & \vdots \\ \boldsymbol{A}_1^n & \cdots & \boldsymbol{A}_i^n & \cdots & \boldsymbol{A}_n^n \end{bmatrix} \begin{bmatrix} \boldsymbol{q}_1 \\ \vdots \\ \boldsymbol{q}_i \\ \vdots \\ \boldsymbol{q}_n \end{bmatrix} \tag{7.44}$$

or

$$\boldsymbol{C} = \mathrm{A}\boldsymbol{Q} \tag{7.45}$$

where matrix $\boldsymbol{C} = \left[\boldsymbol{c}^1, \ldots, \boldsymbol{c}^j, \ldots, \boldsymbol{c}^n\right]^{\mathrm{T}}$, matrix $\boldsymbol{A}$ is the whole square matrix in the right-hand side of Eq. (7.44), and $\boldsymbol{Q} = \left[\boldsymbol{q}_1, \ldots, \boldsymbol{q}_i, \ldots, \boldsymbol{q}_n\right]^{\mathrm{T}}$. Eqs. (7.44) and (7.45) outline the concentration responses of multiple pollutant sources. The matrix $\boldsymbol{A}$ is also subject to the flow field, pollutant source locations and locations of the concentration concerning points. Since $\boldsymbol{A}$ does not rely on the pollutant source release rates, the matrix can be presolved by computational fluid dynamics (CFD) simulation prior to inverse modeling. More details on constructing the matrix $\boldsymbol{A}$ can be found in Refs. [29,40].

7.2.8.1 *Inverse identification of multiple temporal release rates*

The release rate matrix $\boldsymbol{Q}$ should be solved first with the input of the concentration matrix $\boldsymbol{C}$. An intuitive strategy is inverting matrix $\boldsymbol{A}$. However, matrix $\boldsymbol{A}$ is not reversible due to the ill-posedness of the inverted cause–effect relationship, so Tikhonov regularization[41] is applied in the model to add an additional stabilized term when implementing inversion as:

$$\boldsymbol{Q} = \left(\boldsymbol{A}^T\boldsymbol{A} + \lambda^2 \boldsymbol{L}^T\boldsymbol{L}\right)^{-1}\left(\boldsymbol{A}^T\boldsymbol{C}\right) \tag{7.46}$$

where the matrices $\boldsymbol{A}$, $\boldsymbol{Q}$, and $\boldsymbol{C}$ are a combined matrix for multiple sources and concentrations. λ is the regularized parameter that adjusts strength of the regularization, and $\boldsymbol{L}$ is the regularized matrix, which usually has as a second-order derivative format.[42] With an appropriate λ, the matrix $\left(\boldsymbol{A}^T\boldsymbol{A} + \lambda^2\boldsymbol{L}^T\boldsymbol{L}\right)$ can be inverted. In Eq. (7.46), the dimensions of $\boldsymbol{C}$ should be equal to the dimensions of $\boldsymbol{Q}$ to make the equation solvable. This implies that the number of concentration monitoring points should be equal to the number of pollutant sources to be identified.

7.2.8.2 Inverse identification of multiple pollutant source locations

When constructing a matrix based on Eq. (7.41), the pollutant source locations are presumed to be known. This means that the potential pollutant sources (PPSs) have to be sampled with known locations toward establishing a pool. However, the exact sources in the pool of potential sources that are responsible for the detected concentrations are unknown. Suppose there are n possible isolated source locations in the pool and m sources have released the same gaseous pollutant and that $2 \le m \le n$. The number of candidate groups of sources in the pool to be identified is $C_n^m = n!/[(n-m)! \times m!]$, where $n! = n \times (n-1)\ldots 3 \times \ 2 \times 1$. If each candidate group has an equal chance to be the actual sources, then the prior probability of each group is $1/C_n^m$. By providing more concentration information, the probability of each candidate group as the actual sources can be further quantified. This can be done by adopting the Bayesian inference with the formula:

$$p(G_m|O) = \frac{p(G_m)L(O|G_m)}{\sum_{s=1}^{M} L(O|G_s)p(G_s)} \tag{7.47}$$

where $p(G_m|O)$ is the probability of the candidate group G_m being the actual sources based on the observation O, $p(G_m)$ is the prior probability of group G_m being the sources, which does not rely on the observation, $L(O|G_s)$ is the likelihood of acquiring the measured observation O for the candidate group G_s, and $M = C_n^m$. As mentioned, if all C_n^m possible groups have identical probabilities as the sources, then $p(G_s) = 1/C_n^m$. The likelihood function can be assigned an appropriate value by matching the predicted concentrations from a candidate group of sources with the measured concentrations at a sensor.

If the bias between the predicted and the observed concentrations indicates the likelihood of the candidate group of sources as the actual sources, the likelihood function can be constructed based on the Gaussian normal distribution. Then the likelihood of acquiring the observation O for a candidate group G_m is[43]

$$L(O|G_m) = \frac{1}{\sqrt{2\pi}\sigma} exp\left[-\frac{1}{2}\left(\frac{c_O - c_{G_m}}{\sigma}\right)^2\right] \tag{7.48}$$

where c_O is the measured concentration by a sensor that is different from those sensors providing concentrations to constitute the matrix $\boldsymbol{C}$ in Eq. (7.48), c_{G_m} is the predicted concentration at the sensor's location for a candidate group G_m, and σ^2 is the differentiating error variance between the observed and the predicted concentrations. The value of σ indicates the tolerance to judge identity of both concentrations. A closer match between the measured and predicted concentrations results in a higher probability of the candidate group matching the actual sources.

7.2.8.3 Solution procedure

Fig. 7.7 illustrates the solution procedure for the identification of multiple pollutant sources. Suppose that n PPSs and m sources are supposed to release the pollutant ($2 \le m \le n$). If $m < n$, m is the number of isolated sources with nonzero release rates. Consequently C_n^m candidate groups of sources can be constituted by selecting m sources from n PPSs ($C_n^m \ge 1$). The thermoflow fields and the unit-impulse tracer gas concentrations can be resolved by CFD. The governing matrix $\boldsymbol{A}$ is established once the sensor locations are known. With the retrieved m concentrations monitored by m sensors, the matrix $\boldsymbol{C}$ can be constructed. The release rate profiles for each candidate group of sources are solved by Eq. (7.46). If $m < n$, there is more than one

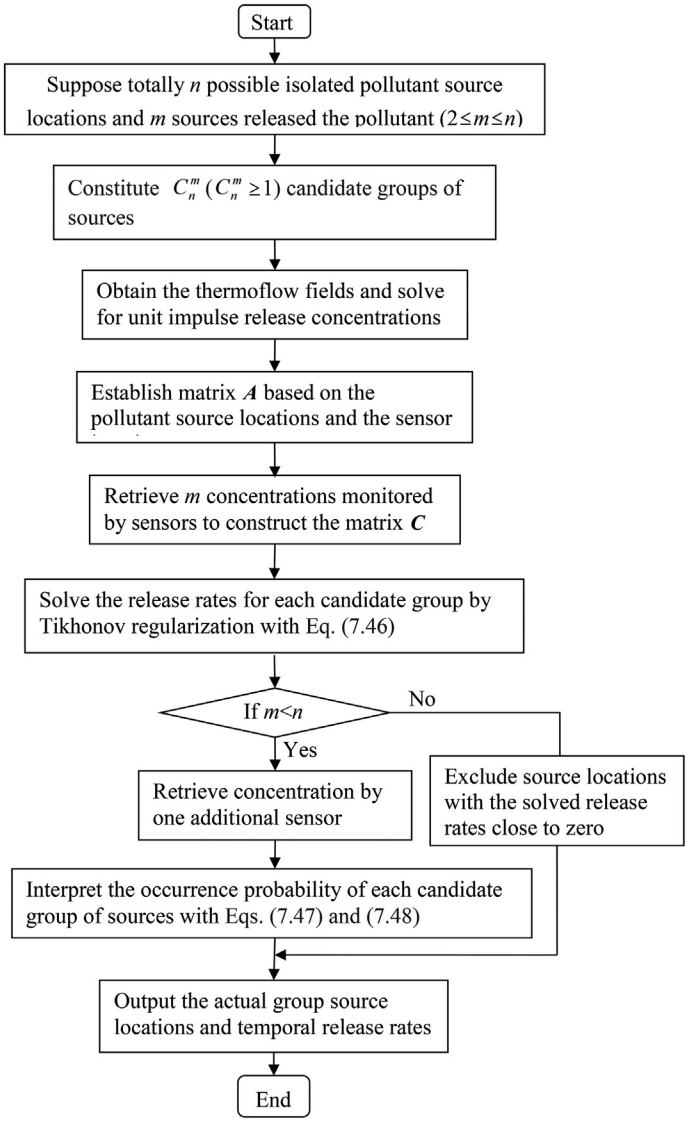


FIGURE 7.7 Solution procedure for the identification of multiple pollutant sources.

candidate group of sources. The concentration monitored by an additional sensor is required to further interpret the occurrence probability of each candidate group by solving Eqs. (7.47) and (7.48). In such situations, $m+1$ sensors are required to fulfill the inverse identification. However, if $m=n$, no interpretation of the occurrence probability is needed. The actual sources must be determined by excluding those with the solved release rates close to zero. Because no Bayesian inference is operated, m sensors are required. Finally the identified group of source locations and their temporal release rate profiles are outputted.

7.3 Transport mechanism of contaminant in ventilated space

7.3.1 Factors influencing room airflow

The airflow field in the ventilated space is affected by different external and internal forces, such as

- supply air jets forced into the room by mechanical systems;
- free convection currents generated by air heating or cooling by surfaces (process equipment, external walls);
- airflow in the vicinity of local exhausts (hoods) or general exhaust (due to negative pressure in the duct produced by mechanical systems);
- airflow forced through intended and unintended openings in the building envelope, which depends on the pressure difference across the opening resulting from wind pressure on the building envelope, temperature difference between the indoor and outdoor air, and an imbalance in the mechanical exhaust ventilation system performance versus the mechanical air supply (positive or negative pressure building); and
- air currents produced by process equipment or moving people (e.g., high-speed rotating machines such as pulverizers, high-speed belt material transfer systems, falling granular materials, and escaping compressed air from pneumatic tools).

The airflow pattern and the scale of air currents in the room depend on the types of sources and the energy introduced by each source, as well as the configuration and dimensions of the room. The energy of the predominant turbulent flow created by each source transfers into transverse turbulent pulsations, which convert large eddies into smaller eddies. This energy is finally converted into heat. Kinetic energy of air leaving the room through exhaust openings can be neglected. Typically exhaust openings are protected by a grill, which does not let through large or medium-size energy-containing eddies.

The energy of large- and medium-size eddies can be characterized by the turbulent diffusion coefficient, A, m^2/s. This parameter is similar to the parameter used by Richardson to describe turbulent diffusion of clouds in the atmosphere.[44] Turbulent diffusion affects heat and mass transfer between different zones in the room and thus affects temperature and contaminant distribution in the room (e.g., temperature and contaminant stratification along the room height—see Chapter 8: Room Air Conditioning). Also the turbulent diffusion coefficient is used in local exhaust design (Section 7.6).

Studies by Elterman show that turbulent diffusion coefficients in ventilated rooms outside jets and plumes can be described using the relationship[3]

$$A = C\epsilon^{1/3}l^{4/3}, \tag{7.49}$$

where C can be evaluated from the equation

$$C = 0.25 \pm \Delta, \tag{7.50}$$

where Δ is the confidence interval, which depends on the required confidence probability, as shown in Table 7.9. In most cases, the average value $C=0.25$ can be used in Eq. (7.49).

Characteristic length, l in Eq. (7.49), depends on the application; for example, for local exhaust design l equals the characteristic hood dimension and for room air distribution design with a temperature or contaminant stratification, l equals the room height.

Another important parameter used in Eq. (7.49) is ϵ, which is the kinetic energy, E_{room}, kg m^2/s, dissipated in the mass of air, M, kg, in time τ, s:

$$\epsilon = \frac{E_{room}}{M\tau} \tag{7.51}$$

The total kinetic energy introduced by different sources can be calculated by summation of all sources:

$$E_{room} = \sum E_{jet} + \sum E_{conv} + \sum E_{m.o.} \tag{7.52}$$

Contributing energies can be calculated using the following equations[3]:

- Kinetic energy introduced by supply air jet:

$$E_{jet} = \frac{1}{2}\rho_0 Q_0 V_0^2 \tag{7.53}$$

TABLE 7.9 Coefficient C for Eq. (7.49).

Confidence probability (%)	80	85	90	95	97
Confidence interval (Δ)	0.051	0.063	0.078	0.10	0.114
Maximum (°C)	0.301	0.313	0.328	0.35	0.364
Minimum	0.199	0.187	0.172	0.15	0.136
Average	0.25	0.25	0.25	0.25	0.25

- Kinetic energy generated by convective heat source (W_{conv}):

$$E_{conv} = \frac{gW_{conv}H_r}{1.8C_pT_0} \quad (7.54)$$

where W_{conv} is a convective component of the heat source, H_r is the room height above the heat source, and C_p and T_0 are the specific heat and absolute temperature of the room air, respectively.

- Kinetic energy from the moving objects, calculated from the body's drag coefficient k, area A, velocity V, percent movement t, and the room air density ρ_0:

$$E_{m.o.} = \frac{1}{2}kAV^2\rho_0 t \quad (7.55)$$

7.3.2 Typical airflow patterns

7.3.2.1 Airflow dominated by supply jets

In rooms where energy is introduced primarily by supply air jets, air distribution methods are referred to as mixing type. With a perfect mixing-type air distribution, airflow pattern and air velocity at any point in the room are governed by supply jet momentum. In this case, if the air supply and air exhaust openings are located close to each other, a large proportion of supply air is extracted from the room without passing the occupied zone. Such a situation, called short-circuiting, results from poor design and leads to undesirable airflow patterns.

Current mixing-type air distribution methods typically consider ventilation of the occupied zone with jets intercepting its upper boundary (e.g., Fig. 7.8A and C). Also the occupied zone can be ventilated by the reverse flow produced as the supply jet degrades above the occupied zone level (Fig. 7.8B). Mixing-type air distribution methods include air supply with jets projected vertically downward, inclined jets, jets directed vertically upward, and horizontal jets along room surfaces.

7.3.2.2 Airflow dominated by thermal plumes

In rooms where air and contaminant movement is dominated by thermal energy of heat sources (e.g., in rooms with natural or displacement ventilation), temperature and contaminant stratification along the room height is created. Air supply and exhaust in such rooms are designed not to disturb the natural pattern of air movement created by heat sources: cooled air enters the room in the lower zone close to the floor level and is exhausted from the upper zone. Under the influence of buoyancy, cold air spreads along the floor and floods the lower zone of the room. The air close to the heat source is heated and rises upward as a convective airstream (Fig. 7.9). In the upper zone this stream spreads along the ceiling. The lower part of the convective stream induces the colder air of the lower zone of

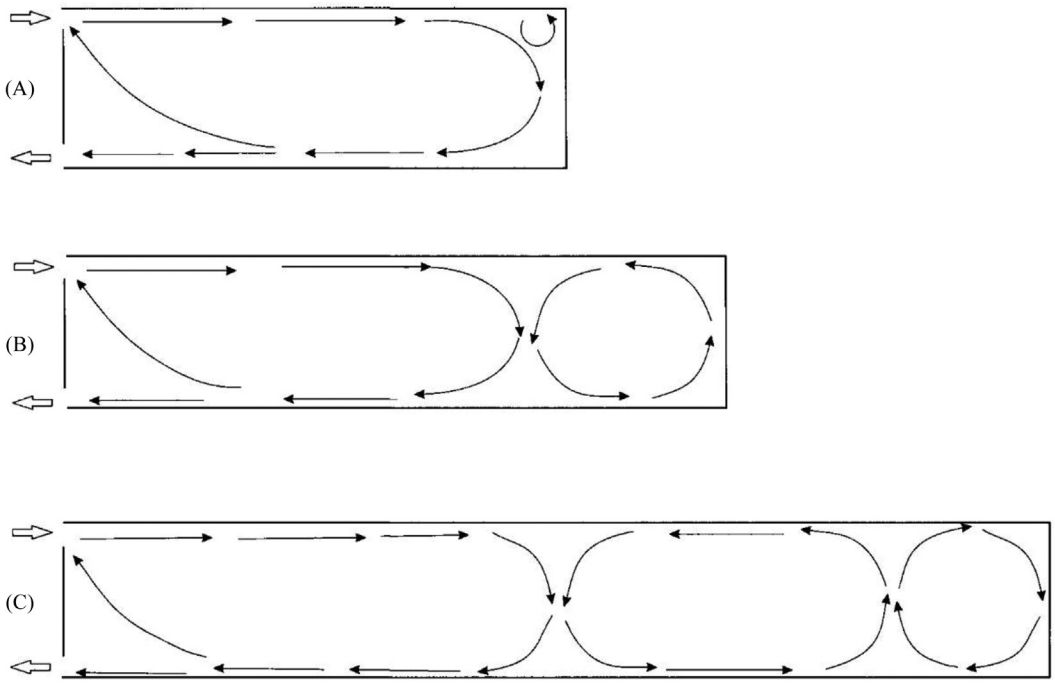


FIGURE 7.8 Flow patterns in rooms with horizontal air supply along the ceiling surface: (A) primary airflow in a short room; (B) primary airflow; and (C) secondary and tertiary eddies in long rooms.[45]

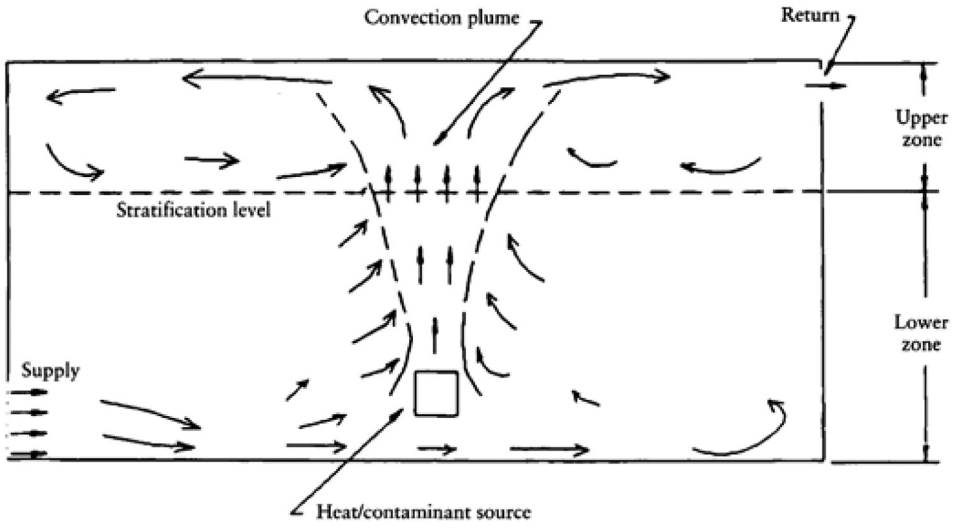


FIGURE 7.9 Schematic of airflow in rooms with a displacement ventilation.[46]

the room, and the upper part of the convective airstream induces the heated air of the upper zone of the room. The height of the lower zone depends on the air volume discharged into the occupied zone and on the amounts of convective heat discharged by the sources.

7.3.2.3 *Unidirectional flow*

To create unidirectional low-turbulence flow, air is supplied with a low velocity; supply diffusers and exhaust openings have large surfaces (e.g., filter mats). Airflow can be either vertical (air supplied from the ceiling and exhausted through the floor or vice versa, Fig. 7.10A), or horizontal (air supplied through one wall and exhausted through returns located on the opposite wall, Fig. 7.10B). The outlets are uniformly distributed over the ceiling, floor, or wall to provide a low-turbulence plug-type flow across the entire room. This type of system is mainly used for ventilating clean rooms, in which the main objective is to remove contaminant particles within the room, or in halls with high heat and/or contaminant loads and a high air change rate.

One of the unidirectional flow system modifications is air supply through diffusers located above the occupied zone. The supply air temperature is lower than the desired room air temperature in the occupied zone, and air velocity is lower compared with a mixing-type air supply but higher than for a thermal displacement ventilation. Polluted air of the occupied zone is suppressed by an overlying air cushion that displaces the contaminated air toward floor-level exhausts (Fig. 7.11).

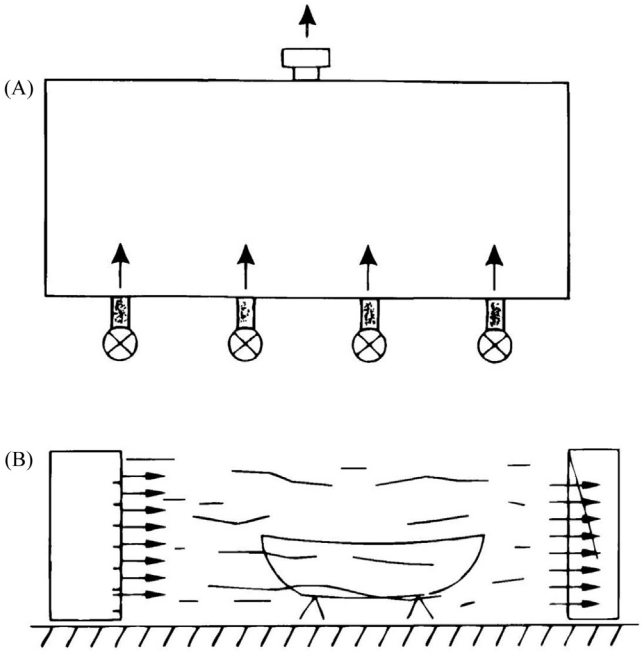


FIGURE 7.10 Unidirectional flow.[47]

7.3.2.4 *Spiral vortex flow*

Spiral vortex air distribution can be used to localize air contaminants in certain room areas and to evacuate the polluted air from those areas. A spiral vortex in a space can be formed by supplying air through the vertical supply ducts located along a closed contour (preferably along the walls), thus generating a vertical vortex. An exhaust outlet can be located in the ceiling near the center of the rotational flow. Such a combination of air supply and exhaust-systems allows a means for concentrating contaminants in the vortex core and transporting them to the exhaust outlet along the core axis (Fig. 7.12). Low pressure in the vortex core allows for the collection of contaminants and for preventing their diffusion to the clean space.[49,50]

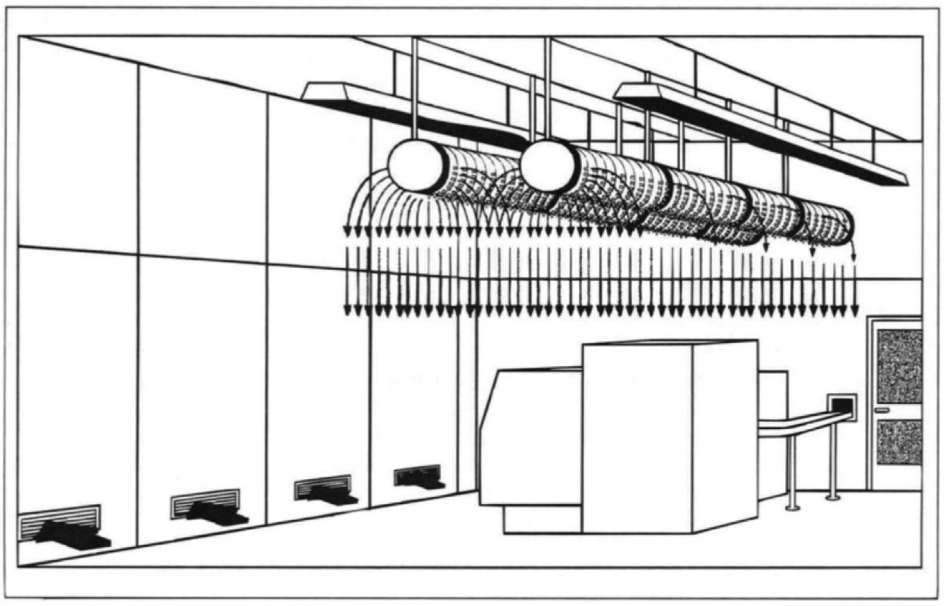


FIGURE 7.11 Unidirectional flow system with air supply through diffusers located above the occupied zone.[48]

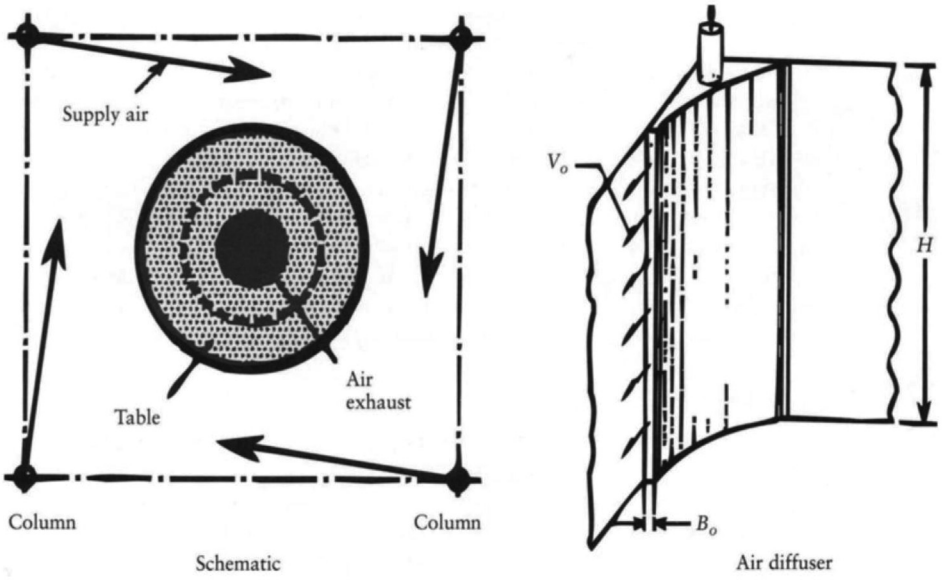


FIGURE 7.12 Spiral vortex flow.[49,50]

7.3.2.5 *Airflow created by exhausts*

The effect of exhaust performance on room air movement is limited compared with the effect produced by air jets. The distance from the opening to the point where air velocity drops to 10% of the initial velocity value (Fig. 7.13) is approximately equal to one characteristic size of the exhaust opening (D for the round duct) and 60 characteristic sizes for the supply outlet ($60D$ for the round nozzle).

Local exhausts are designed to capture air pollutants and heat at the source, and thus their location and the exhausted airflow rate should ensure sufficient capture velocity.

General exhausts typically do not prevent contaminants and heat from mixing with the room air, and thus, theoretically, it does not matter where the exhaust openings are located (Fig. 7.14).

In practice, the air seldom mixes as completely as in theory. The pollutants spread with the room air, but there may be areas where the concentration is higher than the average for the room. This has to be taken into account in determining the location of the exhaust opening.

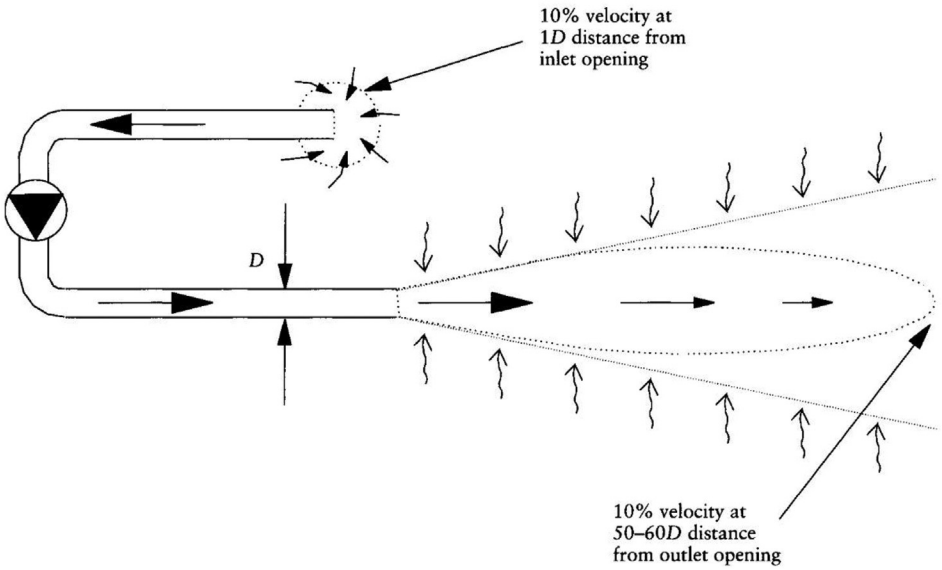


FIGURE 7.13 A jet reaches a long way in one direction, but the capture length of a sink is very limited.

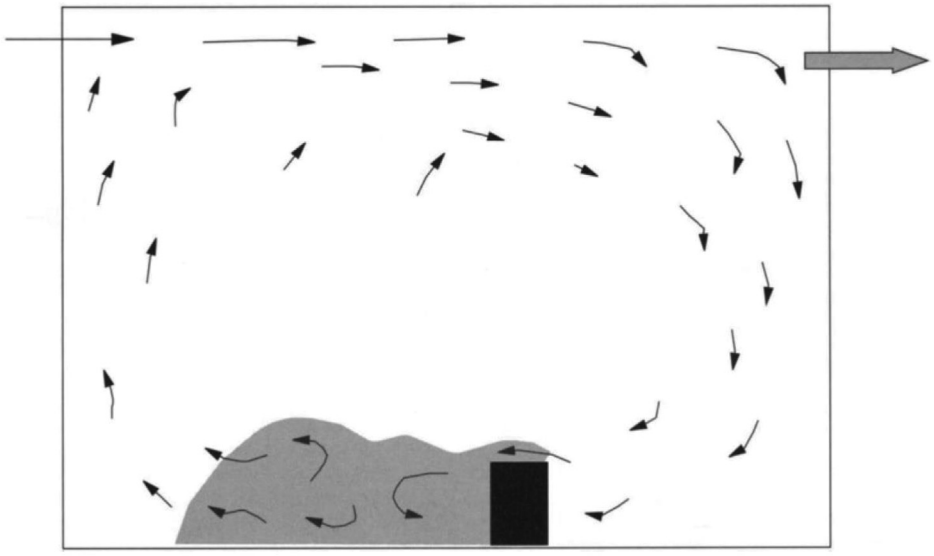


FIGURE 7.14 Exhaust from a room with perfect mixing. The location of exhaust openings is not important.

7.3.3 Quantitative effects of various factors on contaminant distribution

The ventilation and pollution removal performance in a ventilated space could be quantified by a number of indices, where "age of air" is a classical concept to evaluate the effectiveness of an air distribution system in delivering outdoor air to different positions. The age of air is defined as the elapsed time since the air entered the room.[51] At the air supply point the age of air is usually treated as zero. The fresher air at a point has a younger age of air. The local mean age of air is obtained by using tracer gas measurement or by solving the age equation in a CFD code. However, most of the traditional indices including local mean age of air are applicable to the evaluation of steady state ventilation performance, not to the evaluation of transient performance.

The distributions of the scalar parameters of indoor air (i.e., concentrations of various contaminants, temperature, humidity, etc.) are determined by specific airflow patterns and various boundary conditions, and can be regarded as the summation of three parts:

- contribution from the supply air,
- contribution from the released sources, and
- contribution from the initial distribution.

The room airflow is normally a complex turbulent flow, which exhibits a strong nonlinear characteristic. It is difficult to make a simple description or prediction of the scalar distribution under the complex and

transient airflow field. However, in condition that the airflow pattern does not change notably during a period of time, the indoor airflow field could be assumed as steady. In many circumstances, various types of contaminants can be treated to be passively transported, and therefore their dispersion process under the steady flow field becomes linear and the superposition theorem for the effects of different influencing factors is practicable. If the boundary conditions of the supply air and contaminant sources are constant, the so-called index of "transient accessibility" can be defined to quantify the transient effect of each independent boundary factor on the establishment of the contaminant concentration field, based on which the transient concentration field can be predicted by using the superposition theorem.[52]

7.3.3.1 Transient accessibility of supply air

For a steady flow field, assume that the initial contaminant concentration is 0, the emission rates from all of the contaminant sources are 0, and the contaminant concentration in the supply air from the n_S^{th} inlet is $C_S^{n_S}$ whereas those of the other inlets are 0. Then, the transient accessibility of supply air (TASA), $a_{S,p}^{n_S}(\tau)$, at an arbitrary point p from the n_S^{th} inlet is defined as[52]:

$$a_{S,p}^{n_S}(\tau) = \frac{C_p(\tau)}{C_S^{n_S}} \tag{7.56}$$

where $C_p(\tau)$ is the contaminant concentration of point p at moment τ, kg/m^3, $C_S^{n_S}$ is the contaminant concentration in the supply air from the n_S^{th} inlet, kg/m^3, and τ is the predicted moment, s.

TASA is a function only related to the room airflow characteristic and has nothing to do with the location and emission rate of contaminant source. TASA quantifies how easily the contaminant from the supply air of each inlet is transiently delivered to an arbitrary location. It is a good measure to the ability of each air supply inlet to influence different local zones.

7.3.3.2 Transient accessibility of contaminant source

For a steady flow field, assume that the initial contaminant concentration is 0, the contaminant concentrations in the supply air from all of the inlets are 0, and the emission rate from the n_C^{th} contaminant source is J^{nC}, whereas the rates from the other sources are 0. Then, the transient accessibility of contaminant source (TACS), $a_{C,p}^{n_C}(\tau)$, at an arbitrary point p from the n_C^{th} contaminant source is defined as[52]:

$$a_{C,p}^{n_C}(\tau) = \frac{C_p(\tau)}{\overline{C}_E^{n_C}} \tag{7.57}$$

where $\overline{C}_E^{n_C}$ is the average concentration from all of the outlets caused by the n_C^{th} contaminant source, kg/m^3, $\overline{C}_E^{n_C} = \frac{J^{n_C}}{Q}$, J^{n_C} is the emission rate from the n_C^{th} contaminant source, kg/s, and Q is the ventilation rate of the room, m^3/s.

TACS is a function both related to the flow characteristic and location of contaminant source, having nothing to do with emission rate of contaminant source. TACS quantifies how easily the contaminant source transiently transports contaminant to an arbitrary location.

7.3.3.3 Transient accessibility of initial condition

If the ventilated space is divided into N_I zones, the entire initial condition of contaminant $C_p(0)$ could be divided into N_I subinitial conditions, correspondingly. For the n_I^{th} subinitial condition $C_p^{n_I}(0)$, the contaminant distribution in the n_I^{th} zone is the same as the entire initial condition, whereas the contaminant concentrations in the other zones are zero. For a steady flow field, assume that the contaminant concentrations in the supply air from all of the inlets are 0 and the emission rates from all of the contaminant sources are 0, and the initial distribution of contaminant of the n_I^{th} subinitial condition is $C_p^{n_I}(0)$, whereas the initial distribution of contaminant of the other subinitial conditions are 0. Then, the transient accessibility of initial condition (TAIC), $a_{I,p}^{n_I}(\tau)$, at an arbitrary point p from the n_I^{th} subinitial condition is defined as[52,53]:

$$a_{I,p}^{n_I}(\tau) = \frac{C_p(\tau)}{\overline{C}_0^{n_I}} \tag{7.58}$$

where $\overline{C}_0^{n_I}$ is the average concentration of the n_I^{th} subinitial condition, kg/m^3, $\overline{C}_0^{n_I} = \frac{\oint C_p^{n_I}(0)dV}{V}$, V is the volume of the room, m^3, and $C_p^{n_I}(0)$ is the contaminant concentration of point p for the n_I^{th} subinitial condition, kg/m^3.

TAIC is a function only related to the flow characteristic. It quantifies the transient effect of the initial contaminant distribution in each zone on an arbitrary location.

The indices TASA, TACS, and TAIC are dimensionless and they reflect the quantitative effect of contaminant in supply air, contaminant source and initial condition of concentration on the concentration in an arbitrary location at transient moment. The indices can be obtained from both numerical simulation and experiment. By releasing the contaminant at a target boundary while keeping the other boundaries free of contaminant, the transient concentration of contaminant can be measured or simulated, and the accessibility index can be further calculated following the corresponding definition. The demonstrations of the above indices can be found in Sections 7.4.8, 7.5.6, and 7.7.7.

7.3.4 Analytical expression for transient transport of passive contaminant

7.3.4.1 Contribution from supply air

For a room with N_S supply air inlets, the contribution from all of the supply air inlets to the formation of the transient concentration at an arbitrary point p is expressed as:

$$C_{p,sa}(\tau) = \sum_{n_S=1}^{N_S} \left[C_S^{n_S} a_{S,p}^{n_S}(\tau) \right] \tag{7.59}$$

where $C_{p,sa}(\tau)$ is the contaminant concentration of point p at moment τ resulted from concentrations in the supply air from all of the inlets, kg/m^3 and N_S is the number of the supply air inlets.

7.3.4.2 Contribution from contaminant source

For a room with N_C contaminant sources, the contribution from all of the sources to the formation of the transient concentration at an arbitrary point p is expressed as:

$$C_{p,cs}(\tau) = \sum_{n_C=1}^{N_C} \left[\frac{J^{n_C}}{Q} a_{C,p}^{n_C}(\tau) \right] \tag{7.60}$$

where $C_{p,cs}(\tau)$ is the contaminant concentration of point p at moment τ resulted from all of the contaminant sources, kg/m^3, and N_C is the number of the contaminant sources.

7.3.4.3 Contribution from initial condition

For a room with an initial condition of contaminant consisting of N_I subinitial conditions, the contribution from the initial condition to the formation of the transient concentration at an arbitrary point p can be expressed as:

$$C_{p,ic}(\tau) = \sum_{n_I=1}^{N_I} \left[\overline{C}_0^{n_I} a_{I,p}^{n_I}(\tau) \right] \tag{7.61}$$

where $C_{p,ic}(\tau)$ = contaminant concentration of point p at moment τ resulted from the entire initial contaminant distribution, kg/m^3.

7.3.4.4 Expression for transient concentration of contaminant

In a steady flow field, the transient contaminant concentration at an arbitrary point p can be predicted by the summation of three contributions: supply air, contaminant source and initial condition. The transient concentration is expressed as[52,53]:

$$\begin{aligned} C_p(\tau) &= C_{p,sa}(\tau) + C_{p,cs}(\tau) + C_{p,ic}(\tau) \\ &= \sum_{n_S=1}^{N_S} \left[C_S^{n_S} a_{S,p}^{n_S}(\tau) \right] + \sum_{n_C=1}^{N_C} \left[\frac{J^{n_C}}{Q} a_{C,p}^{n_C}(\tau) \right] \\ &\quad + \sum_{n_I=1}^{N_I} \left[\overline{C}_0^{n_I} a_{I,p}^{n_I}(\tau) \right] \end{aligned} \tag{7.62}$$

Eq. (7.62) is a simple algebraic form, and therefore has two main advantages: (1) from this expression, contribution of each boundary condition to the concentration at an arbitrary location can be easily recognized, which results in increasing knowledge on the transporting principle of real flow field and (2) when any one or more boundary conditions of concentration are changed, the resulted concentration value at an arbitrary location p can be quickly obtained only through the algebraic calculation. The advantage in computing speed appears to be more obvious when a large amount of cases are calculated, since repeated simulation iterations can be avoided by using the algebraic expression.

Eq. (7.62) is used to describe the transport mechanism of contaminants for constant boundary conditions. If the boundary conditions vary with time, the total effect of each independent boundary factor (concentration in supply air from each supply opening, emission rate from each contaminant source) at a specified time should be further decomposed into the summation of the effect of the boundary parameter at each time step along the past period of time. The effect of each boundary condition at one time step is quantified by the so-called index of "response coefficient"[34] instead of the "transient accessibility." With the response coefficient, the contribution from each changing boundary condition can be mathematically expressed and the expression of the transient concentration of contaminant at an arbitrary point can be established by using the superposition theorem.[34]

The above mentioned equations are suitable for passive contaminants. For the dense gas with the density significantly larger than the air density, the transport of the contaminant will not completely follow the airflow, and therefore a certain prediction discrepancy may exist when using the abovementioned equations for the dense gas distribution. In addition, room airflow field is sensitive to the change in the heat source intensity due to the buoyancy effect, and a prediction discrepancy may exist as well when adopting a linear superposition expression for the transient temperature distribution at scenarios with changing heat source. The research results show that the prediction deviation decreases as the decrease of intensity difference between the heat source used to build the fixed flow field and the real intensity of heat source.[54] The accuracy is acceptable in predicting the average temperature of the room and temperature at positions outside the heat source area.

7.4 Air jets

7.4.1 Introduction

Air supplied into the room through the various types of outlets (grills, ceiling-mounted air diffusers,

perforated panels, etc.) is distributed by turbulent air jets. In mixing-type air distribution systems, these air jets are the primary factor affecting room air motion. Numerous theoretical and experimental studies that developed a solid base for turbulent air jets theory were conducted from the 1930s. Theory of air jets and air distribution design principles are discussed in this section.

7.4.2 Classification

If there is no influence of the walls, ceiling, or obstructions on the air jet, it can be considered a free jet. If the air jet is attached to a surface, it is called an *attached air jet*.

Characteristics of the air jet in the room might be influenced by reverse flows, created by the jet entraining the ambient air. This air jet is called a *confined jet*. If the temperature of the supplied air is equal to the temperature of the ambient room air, the jet is an *isothermal jet*. A jet with an initial temperature different from the temperature of the ambient air is called a *nonisothermal jet*. The air temperature differential between supplied and ambient room air generates buoyancy forces in the jet, affecting the trajectory of the jet, the location at which the jet attaches and separates from the ceiling/floor, and the throw of the jet. The significance of these effects depends on the relative strength of the thermal buoyancy and inertial forces (characterized by the Archimedes number).

Air jets can be classified according to the diffuser type as follows (Fig. 7.15):

1. *Compact* air jets[55] are formed by cylindrical tubes, nozzles, and square or rectangular openings with a small aspect ratio that are unshaded or shaded by perforated plates, grills, etc. Compact air jets are three dimensional and axisymmetric at least at some distance from the diffuser opening. The maximum velocity in the cross-section of the compact jet is on the axis.
2. *Linear* air jets are formed by slots or rectangular openings with a large aspect ratio. The jet flows are approximately two dimensional. Air velocities are symmetric in the plane at which air velocities in the cross-section are maximum. At some distance from the diffuser, linear air jets tend to transform into compact jets.
3. *Radial* air jets are formed by ceiling cylindrical air diffusers with flat disks or multidiffusers that direct the air horizontally in all directions.
4. *Conical* air jets are formed by cone type or regulated multidiffusers in ceiling air distribution devices, and have an axis of symmetry. The vectors of air velocities are parallel to the conical surface (with an angle at the top of the cone equal to 120 degrees). Maximum velocities in cross-sections perpendicular to the axis occur at the conical surface.
5. *Incomplete radial jets* are supplied through outlets with grills having diverging vanes, and have a coerced angle of expansion. At some distance this kind of jet tends to transform into a compact one.
6. Swirling jets are supplied to the room through air diffusers with vortex-forming devices creating rotation motion, and have tangential as well as radial velocity vectors. Depending on the type of air diffuser, swirling jets can be compact, conical, or radial.

In these years, fabric-based air duct, which usually provide compact air jets by cylindrical shape, has been widely used in many fields, such as the catering and food processing industries.[56] Relevant researches show that, comparing with the conventional rigid ducting, fabric ducting has shown great promise toward less costing, easier maintenance, as well as providing much more uniform velocities and comfortable environment.[57] However, it can only be used to supply air. For outlets, rigid ducting should be installed.

7.4.3 Isothermal free jet

7.4.3.1 Zones in a jet

For different types of free jets and air diffusers there are similarities in the resulting flows. Four major zones are recognized along a free jet. These zones, as described by Tuve,[58] may be roughly defined in terms of the maximum or center core velocity that exists at the jet cross-section being considered (see Fig. 7.16A).

Zone I is a short zone, extending about two to six diffuser diameters (for compact and radial jets) or slot widths (for linear jets) from the diffuser face, depends on the type of diffusers and the turbulence of the air supply. In this zone, the centerline velocity of the jet remains nearly equal to the original supply velocity throughout its length.

Zone II is a transition zone, and its length depends on the diffuser type. For a compact jet the transition zone typically extends to eight or ten diameters from the outlet. Within this zone, the centerline velocity begins to gradually decrease as[59]:

$$U_m/U_o \propto 1/x^n \tag{7.63}$$

where x is the distance from the supply, and n is an index between 0.33 and 1.00. Some researchers[60–62] suggest use of a simplified scheme of the jet (Fig. 7.16B) with a transition cross-section for practical purposes.

Zone III is the zone of fully established turbulent flow. It has major engineering importance, since it is usually in this zone that the jet enters the occupied

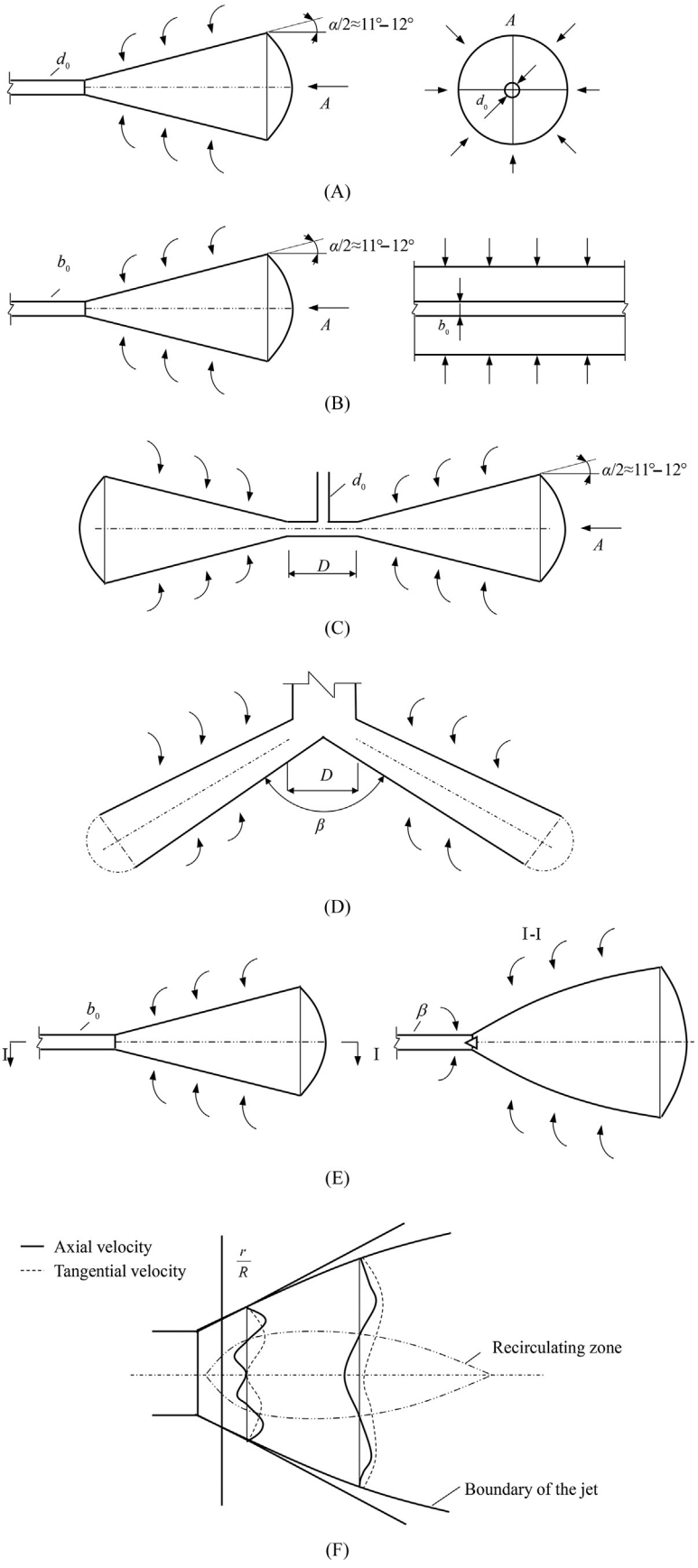


FIGURE 7.15 Types of diffuser jets: (A) compact, (B) linear, (C) radial, (D) incomplete radial, (E) conical, and (F) swirlin.[55]

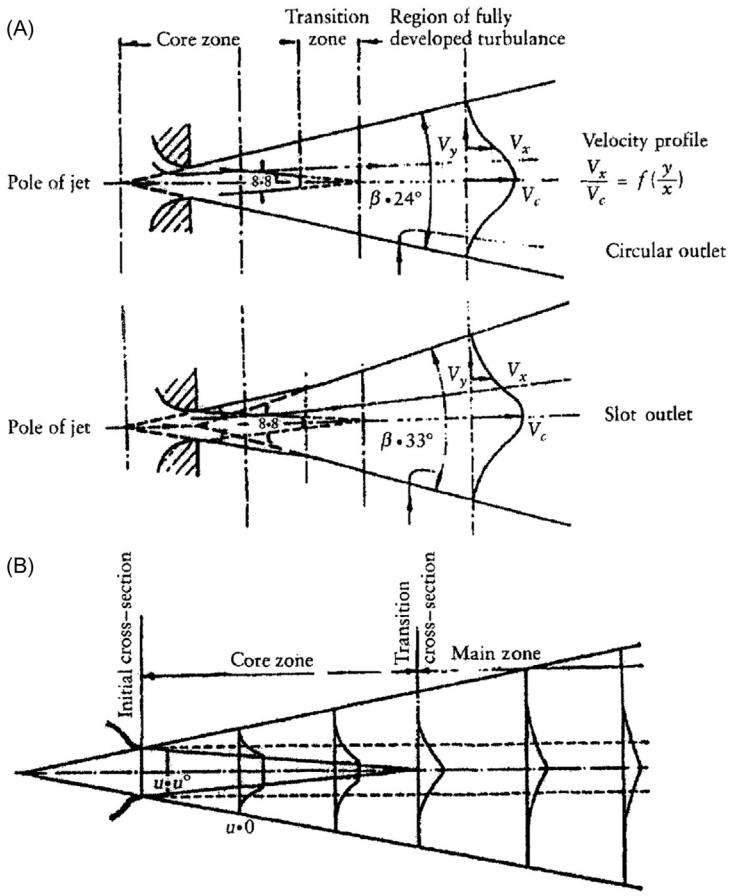


FIGURE 7.16 Turbulent jet: (A) schematic with four zones and (B) simplified jet schematic.

region. The length of this zone depends on the air jet shape, the type and size of supply air diffuser, the initial velocity of the air jet, and the turbulence characteristics of the ambient air. The centerline velocity here decreases inversely with the distance from the opening, that is[59]:

$$U_m/U_o \propto 1/x \tag{7.64}$$

Zone IV is a terminal zone in which the residual velocity decays quickly into large-scale turbulence. Within a few diameters, the maximum velocity subsides to below 0.25 m/s. This zone has been studied by several researchers,[63,64] and Awbi pointed that the centerline velocity in this region decays with the square of the distance[59]:

$$U_m/U_o \propto 1/x^2 \tag{7.65}$$

In some practical applications of air supply (e.g., multiple-jet ceiling diffusers, an annular jet collapsing into a compact jet, jets from rectangular outlets becoming round or elliptical, or multiple streams merging when air is supplied through perforated panels), measurements and accurate jet description in Zones I and II may be difficult.

7.4.3.2 *Velocity distribution in jet cross-section within Zone III*

Velocity distribution profiles in Zone III of the jet were found to be similar.[65–67] They can be computed by applying momentum-transfer theory (Prandtl–Tollmein) and vorticity-transfer theory (Taylor–Goldstein). Modification of these theories with different assumptions has resulted in several equations for jet velocity profiles. These equations are presented in Table 7.10 and can be divided into two groups:

- Profiles with finite boundaries[68,70] with a velocity of zero at the specified distance from the jet axis.
- Profiles with indefinite boundaries[69,71] with air velocity decreasing with distance from the axis asymptotically approaching zero.

In the equations listed in Table 7.10, $r(y)$ is the distance from the point of interest to the jet axis, and δ is the distance to the jet boundary, which can be obtained from equations summarized in Table 7.11.

It has been demonstrated by Ruden,[73] Albertson et al.,[67] Taylor et al.,[74] Keagy and Weller,[75] Pai,[76] Becher,[77] Förthmann,[66] Nottage et al.,[78] Shepelev,[61] Grimitlyn,[55] and other researchers that the Gauss

TABLE 7.10 Equations for velocity profiles in a free jet.

	v_i/v_x	
Authors	**Round jet**	**Linear jet**
Schlichting[68]	$(1-(r/\delta)^{3/2})^2$	$(1-(y/\delta)^{3/2})^2$
Reichardt[69]	$e^{-(r/\delta)2}$	$e^{-(r/\delta)2}$

TABLE 7.11 Equations for jet boundaries.

	δ		
Authors	**Round**	**Linear**	**Radial**
Tollmein[70]	$0.151x$	$0.272x$	–
Reichardt[69]	$0.085x$	–	–
Heskestad[72]	–	–	$0.288x$
Tuve[58]	–	–	$0.185x$

TABLE 7.12 Practical modifications of the Gauss error-function velocity profile equations.

	vv_x		
Authors	**Round jet**	**Linear jet**	**Radial jet**
Tuve[58]	$e^{-0.7(y/y_{0.5v})^2}$	$e^{-0.7(y/y_{0.5v})^2}$	$e^{-0.7(y/y_{0.5v})^2}$
Grimitlyn[79]	$y_{0.5v}=x\tan\alpha_{0.5v}$	$y_{0.5v}=x\tan\alpha_{0.5v}$	$y_{0.5v}=x\tan\alpha_{0.5v}$
Shepelev[61]	$e^{-(1/2)(y/0.082x)^2}$	$e^{-(1/2)(y/0.1x)^2}$	$e^{-(1/2)(y/0.1x)^2}$

error-function profile by Reichardt is in agreement with data from studies of both nozzle jets and manufactured air diffusers supplying similar jets. This profile is utilized mostly by researchers using the analytical (semiempirical) approach in air jet studies. Table 7.12 lists some modifications of the Gauss error-function velocity profile equations used for practical applications.

The Schlichting finite-boundaries profile is another one that is frequently used.[80] Utilization of this profile is specifically fruitful for describing velocity distribution in complex flows, for example, a jet in a cross-flow[81] and jet interaction under the right angle.[82] In such cases, distance from the jet axis, r_i, to the point with an air velocity V_i is replaced by the parameter $r_i=(S_i/\pi)^{1/2}$, where S_i is the area within a contour with a constant velocity V_i.

7.4.3.3 *Centerline velocity in Zone III*

Compact jet

Centerline velocity in Zone III of the supply jet can be calculated from the equations based on the principle of momentum conservation along the jet[83,84]:

$$M_0=2\pi\rho\int_0^{y^*}V^2y\,dy \tag{7.66}$$

where $M_0=\rho\nu_0^2A_0$ is the initial jet momentum and y^* is the distance from the axis to the jet boundary.

Application of the Gauss error-function equation for velocity profile in the form proposed by Shepelev (Table 7.12) in Eq. (7.66) results in the following formula for the centerline velocity in Zone III of the compact jet:

$$v_x=\frac{1}{\sqrt{\pi}c}\sqrt{\frac{M_0}{\rho_\infty}}\frac{1}{x}. \tag{7.67}$$

Eq. (7.67) can be also presented as

$$v_x=\frac{\theta\phi}{\sqrt{\pi}}c\frac{V_0\sqrt{A_0}}{x}, \tag{7.68}$$

where

$$\theta=\sqrt{\rho_0/\rho_\infty}=\sqrt{T_\infty/T_0} \tag{7.69}$$

and

$$\frac{A}{A_0}\phi=\left[\int_0^1\left(\frac{V}{V_0}\right)^2d\left(\frac{A}{A_0}\phi\right)\right]^{1/2} \tag{7.70}$$

are the coefficients of the velocity distribution at the diffuser outlet.

When outlet velocity distribution is uniform, $\varphi=1$, and v_0 is the average air velocity at the diffuser outlet ($v_0=q_0/A_0$). The complex of coefficients $\Theta\phi/\sqrt{\pi}c$ reflecting the conditions of the air supply has a constant value for a given situation and is called the dynamic characteristic of the diffuser jet.[61] Dynamic characteristic describes the intensity of velocity decay along the air jet axis:

$$K_1=\frac{\overline{O}\phi}{\sqrt{\pi}c}. \tag{7.71}$$

The abovementioned approach for the centerline velocity computation was utilized by different researchers using other velocity profiles and resulted in

$$v_x=K_1v_0\frac{\sqrt{A_0}}{x}. \tag{7.72}$$

Theoretical values of characteristic K_1 depend on the type of velocity profile equation and supply conditions assumed. According to Shepelev, $K_1=6.88$; another estimate is $K_1=6.7$.[85] The Schlichting profile results in $K_1=7.4$, and with the Tollmein profile $K_1=7.76$.[86] According to experimental studies reported

by Tuve,[58] the range of K_1 characteristic for compact jets discharged from round outlets varies between 5.7 and 7 depending on supply air velocity and type of outlet. Analysis of experimental data from different researchers by Rodi[87] indicates that K_1 is close to 7.

Some researchers (e.g., Abramovich,[83] Baturin,[88] Rajaratnam,[89] and Nielsen and Moller[90]) consider x to be the distance from a point located at some distance x_0 upstream from the diffuser face. Equations for the jet boundaries and velocity profile used in the centerline velocity derivation assume that the jet is supplied from the point source. Addition of the distance x_0 to the distance from the outlet corrects for the influence of the outlet size on the jet geometry. For practical reasons some researchers neglect x_0.

Linear jet

The equations for centerline velocities in a linear diffuser jet can be derived using the same principles as in the case of a compact jet. For a linear jet:

$$v_x = K_1 v_0 \sqrt{\frac{H_0}{x}}, \tag{7.73}$$

where H_0 is the height of the slot.

Similar to the case with compact air jet supply, theoretical values of characteristic K_1 depend on the type of velocity profile equation and supply conditions. According to Shepelev, $K_1 = 2.62$. The Görtler profile results in $K_1 = 2.43$ and the Tollmein profile in $K_1 = 2.51$.[86] Becher[91] reported the K_1 characteristic for a linear jet to be equal to 2.55. Experimental results by Heskestad,[92] Miller and Comings,[93] van der Hegge Zijnen,[94,95] Gutmark and Wygnanski,[96] and Kotsovinos and List[97] appear to satisfy $K_1 = 2.43$.

Radial jet

The application of the principle of momentum conservation to the radial jet by Koestel[98] resulted in the following equation for the centerline velocity (Fig. 7.17):

$$\frac{v_R}{v_0} = \frac{\sqrt{K(H_0/R_0)cos\theta[K(H_0/R_0)cos\theta + 1]}}{\sqrt{R(R-R_0)}/R_0}. \tag{7.74}$$

The value of the numerator of the right-hand side of Eq. (7.74) depends on the geometric configuration of the outlet (R_0, H_0, cos θ). The denominator represents the dimensionless distance from the outlet. For a given diffuser or a plaque Eq. (7.74) becomes

$$\frac{v_R}{v_0} = \frac{CR_0}{\sqrt{R(R-R_0)}}. \tag{7.75}$$

Experimental C values for a radial slot and a radial nozzle tested by Koestel are 1.13 and 1.19, respectively. Eq. (7.75) is similar to Eq. (7.72) if the distance from the outlet is large enough so that $(R - R_0)$ is approximately equal to R. The theoretical value of the characteristic K_1 in Eq. (7.72) applied to a radial jet is equal to 1.05, according to Shepelev.[99]

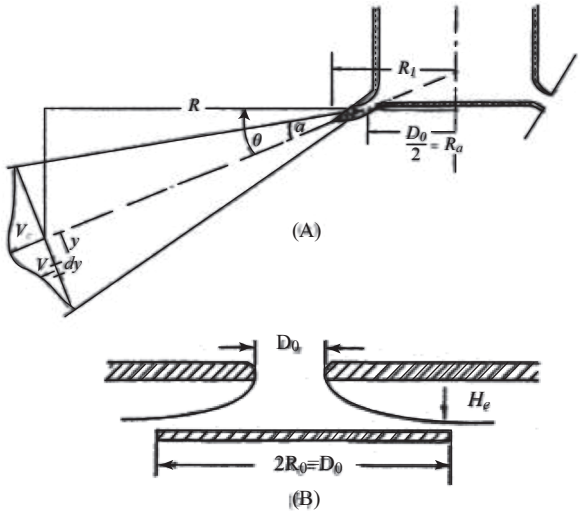


FIGURE 7.17 Schematic of a radial jet: (A) general case $\theta \geq 0$ and (B) air supplied through a diffuser with a plaque: $\theta = 0$.[98]

Similar derivations by Regenscheit[100] resulted in the following equation for the radial jet centerline velocity:

$$\frac{v_R}{v_0} = \frac{R_0}{R}\frac{A_F}{\sqrt{1 + R_0/R}}, \tag{7.76}$$

where

$$\alpha_F = \frac{h_0}{R_0 m} \tag{7.77}$$

and

$$A_F = \alpha_F \sqrt{1 + 1/\alpha_F}. \tag{7.78}$$

Referring to the data from Baturin,[101] Regenscheit evaluated α_F to be 0.377. At a significant distance from the supply outlet ($R_0/R \to 0$), Eq. (7.76) can be transformed into Eq. (7.72).

Eqs. (7.72) and (7.73) show that air velocity along compact, linear, and radial jets is proportional to the value of K_1. This parameter depends on

1. jet type,
2. diffuser type, and
3. initial air velocity[46] or Reynolds number.[102–104]

7.4.3.4 Universal equations for velocity computation along jets supplied from outlets with finite dimensions

A fruitful approach for velocity computation in the first three zones of jets supplied from outlets with finite size was developed based on the hypothesis that

momentum diffuses with distance from the source in the same manner as heat energy.[105,106] This approach, developed by Elrod,[107] Shepelev and Gelman,[108] and Regenscheit,[109] utilizes the method of superposition of jet momentum from the multiple-jet system. These jets originate from the points with supply air velocity equal to the average air velocity at the outlet. This approach utilizes the following principles:

1. Momentum conservation along the jet,
2. air velocity in each jet cross-section, described using the Reichardt Gauss error-function profile, and
3. constant angle of divergence along the jet.

This method was applied by Shepelev and Gelman[108] to air supply through a rectangular outlet with dimensions $2L \times 2B$ (Fig. 7.18), resulting in the following equation for air velocities in the jet cross-section located at the distance x from the outlet:

$$v_{x,y,z} = \frac{v_0}{2}\sqrt{\left(\operatorname{erf}\frac{L-y}{cx} + \operatorname{erf}\frac{L+y}{cx}\right)\left(\operatorname{erf}\frac{B-z}{cx} + \operatorname{erf}\frac{B+z}{cx}\right)}. \tag{7.79}$$

From Eq. (7.79) the centerline velocity can be calculated by substituting $y = 0$ and $z = 0$:

$$v_x = v_0\sqrt{\operatorname{erf}\frac{L}{cx}\operatorname{erf}\frac{B}{cx}}. \tag{7.80}$$

Eqs. (7.78) and (7.79) describe air velocity in cross-sections of the jet located in Zones I–III. The shape of the outlet can be from square ($2B \times 2B$) to infinite slot with a width $2B$ ($L = \infty$). In the case of a linear jet supplied through the slot, Eqs. (7.79) and (7.80) become

$$v_{x,y,z} = \frac{v_0}{\sqrt{2}}\sqrt{\operatorname{erf}\frac{B-z}{cx} + \operatorname{erf}\frac{B+z}{cx}} \tag{7.81}$$

and

$$v_x = v_0\sqrt{\operatorname{erf}\frac{B}{cx}}. \tag{7.82}$$

The solution for a circular jet as presented by Elrod[107] and Regenscheit[109] cannot be evaluated in closed form. The exact solution can be determined only for the centerline velocity:

$$\frac{v_x}{v_0} = 1 - + \left(\frac{-D^2}{2c^2x^2}\right). \tag{7.83}$$

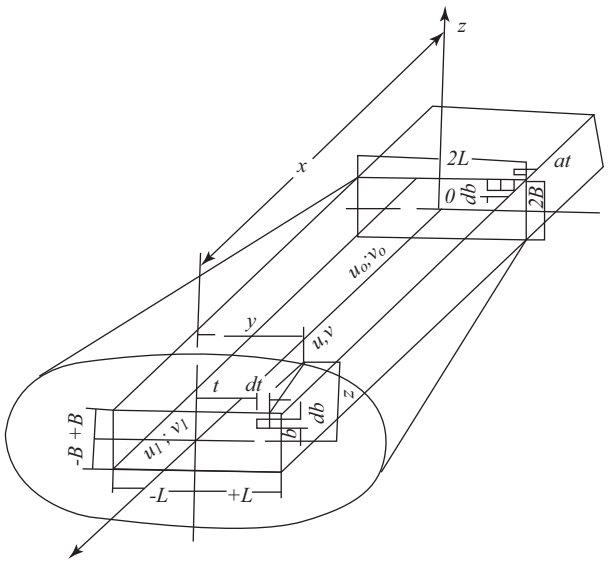


FIGURE 7.18 Schematic of a jet supplied through a rectangular outlet.[108]

7.4.3.5 Jet throw

Diffuser jet throw, L, is a parameter commonly used in air diffuser sizing defined as the distance from the diffuser face to the jet cross-section where the centerline velocity equals a terminal velocity v_x (v_x is often assumed to be 0.25 m/s). Therefore the throw (L) can be determined by velocity decay equations with v_x equal to the terminal velocity:

$$L = K_1\sqrt{A_0}\frac{v_0}{v_x}. \tag{7.84}$$

7.4.3.6 Entrainment ratio

Entrainment ratio is another jet characteristic commonly used in air distribution design practice. Specifically it is used in analytical multizone models (see Chapter 8: Room Air Conditioning) when one needs to evaluate the total airflow rate transported by the jet to some distance from a diffuser face. Airflow rate in the jet, Q_x, can be derived by integrating the air velocity profile within the jet boundaries:

$$Q_x = 2\pi\int_0^{y^*} Vydy. \tag{7.85}$$

Equations for airflow rate computation in compact, linear, and radial jets are presented in Table 7.13.

For a given area of diffuser opening A_0, the entrainment ratio is proportional to the distance x (for compact, radial, and conical diffuser jets) or proportional to the square root of the distance x (for linear jets). For the same type of jet, the entrainment ratio is less with a large K_1 than with a small K_1. Radial and conical diffuser jets have a smaller entrainment ratio than compact or incomplete radial jets with the same K_1 value. Linear diffuser jets have a smaller entrainment ratio than radial and conical jets.

TABLE 7.13 Airflow rate through a jet cross-sectional area.

	Q_x/Q_0		
Authors	Compact jet	Linear jet (per 1 m slot length)	Radial jet
Baturin[110]	$0.275\frac{x-x_0}{\sqrt{A_0}}$	$0.530\sqrt{\frac{x-x_0}{H_0}}$	$0.530\sqrt{(1-R/R_0)\frac{R-R_0}{H_0}}$
Regenscheit[100]	$1.77m\frac{x}{\sqrt{A_0}}$	$\sqrt{m}\sqrt{\frac{x}{H_0}}$	$1.96\frac{R\sqrt{1-R_0/R}}{R_0}$
Shepelev[61]	$0.29\frac{x}{\sqrt{A_0}}$	$0.43\sqrt{\frac{x}{H_0}}$	$0.069\frac{R}{\sqrt{R_0H_0}}$
Grimitlyn[55]	$\frac{2}{K_1}\frac{x}{\sqrt{A_0}}$	$\frac{\sqrt{2}}{K_1}\sqrt{\frac{x}{H_0}}$	$\frac{\sqrt{2}}{K_1}\frac{x}{\sqrt{A_0}}$

7.4.4 Nonisothermal free jets

7.4.4.1 *Criteria for nonisothermal jets*

Buoyant flows can be classified as[111]

- Buoyant jets when the buoyant force acts in the direction of the jet supply velocity at the origin, that is, upward-projected heated air jet or downward-projected cooled air jet.
- Negative buoyant jets when the buoyant force acts in the opposite direction, that is, downward-projected heated air jet or upward-projected cooled air jet.
- Nonbuoyant jets when the effect of buoyancy is negligible.
- Plume when the buoyant force completely dominates the flow, as for flow generated with a heat source.

In the general case, a buoyant jet has an initial momentum. In the region close to discharge, momentum forces dominate the flow, so it behaves like a nonbuoyant jet. There is an intermediate region where the influence of the initial momentum forces becomes smaller and smaller. In the final region, the buoyancy forces completely dominate the flow and it behaves like a plume. When the jet is supplied at an angle to the vertical direction, it is turned upward by the buoyancy forces and behaves virtually like a vertical buoyant jet in a far field. A negative buoyant jet continuously loses momentum due the opposite direction of buoyancy forces to the supply air momentum and eventually turns downward.

Between the nonbuoyant jet region and the plume region lies an intermediate region in which the flow changes from the former to the latter.[111] The axial location of the beginning of this intermediate region depends primarily on the exit Froude number:

$$F = \frac{V_0^2}{gD_0}\frac{T_\infty}{(T_0 - T_\infty)}. \tag{7.86}$$

The beginning of the transition zone in the linear jet is at approximately

$$\frac{x_{\text{beg}}}{H_0} = 0.5F^{2/3}\left(\frac{T_\infty}{T_0}\right)^{-1/3}. \tag{7.87}$$

Using the relation between the Froude number and the Archimedes number, $\text{Ar}_0 = 1/F$, the length of the linear jet zone, x, where the buoyancy forces are negligibly small can be calculated as follows:

$$\frac{x}{H_0} = \frac{0.5}{\text{Ar}_0^{2/3}}\left(\frac{T_\infty}{T_0}\right)^{-1/3}. \tag{7.88}$$

To characterize the relationship between the buoyancy forces and momentum flux in different cross-sections of a nonisothermal jet at some distance x, Grimitlyn[112] proposed a local Archimedes number:

$$\text{Ar}_x = \frac{gx\Delta\theta_x}{V_x^2 T_\infty} \tag{7.89}$$

where g is acceleration due to buoyancy. Equations for the local Archimedes number can be derived by substituting the expressions for axial velocity V_x and temperature differential $\Delta\theta_x$ into Eq. (7.89).

- For compact and radial jets

$$\text{Ar}_x = \frac{K_2}{K_1^2}(\text{Ar}_0)\left(\frac{x}{\sqrt{A_0}}\right)^2 \tag{7.90}$$

- For linear jets

$$\text{Ar}_x = \frac{K_2}{K_1^2}(\text{Ar}_0)\left(\frac{x}{H_0}\right)^{3/2}, \tag{7.91}$$

where the Archimedes number at the outlet is

$$\text{Ar}_0 = \frac{g\sqrt{A_0}}{V_0^2}\frac{\Delta\theta_0}{T_r}, \tag{7.92}$$

characterizing the ratio of buoyancy and inertial forces at the jet discharge from the outlet.

When calculating Ar_0 for a linear jet, $\sqrt{A_0}$ is replaced by the width of the slot H_0.

Introduction of the local Archimedes criterion helped to clarify nonisothermal jet design procedure. Grimitlyn suggested critical local Archimedes number values, $\text{Ar}_x^{\text{crit}}$, below which a jet can be considered unaffected by buoyancy forces (moderate nonisothermal jet): $\text{Ar}_x \le 0.1$ for a compact jet, $\text{Ar}_x \le 0.15$ for a linear jet.

A similar limitation for a linear jet from Eq. (7.91) at $K_1 = 2.5$, $K_2 = 2.0$, $T_0 = 293\text{K}$, and $T_\infty = 313\text{K}$ is $\text{Ar}_x < 0.14$.

7.4.4.2 Temperature profile distribution in a jet

Along with a constant velocity zone (Zone I), there is a constant temperature zone in a jet. Heat diffusion in a jet is more intense than momentum diffusion; therefore the core of constant temperatures fades away faster than that of constant velocities and the temperature profile is flatter than the velocity profile. Thus the length of the zone with constant temperature (Fig. 7.19) is shorter than the length of the constant velocity zone (Zone I).[62,113,114]

From Tolmin's theory and experimental data (e.g., Reichardt[115]) the relationship between velocity profile and temperature profile in the jet cross-section can be expressed using an overall turbulent Prandtl number $\mathrm{Pr} = v_t/\alpha_t$, where v_t is a turbulent momentum exchange coefficient and α_t is a turbulent heat exchange coefficient:

$$\frac{v}{v_x} = \left(\frac{\theta-\theta_\infty}{\theta_x-\theta_\infty}\right)^{1/Pr} \tag{7.93}$$

where θ is the air temperature at the point of consideration, and θ_x is the maximum air temperature in the cross-section of the jet, θ_∞ is the air temperature when the jets are well mixed with the surrounding air.

A Prandtl number of 0.7 has been suggested for nonisothermal jets by Nottage,[116] Forstall and Shapiro,[117] Corrsin and Uberoi,[118] and Grimitlyn.[119] Abramovich[60] suggested a Prandtl number for a compact jet of 0.75, and for a linear jet, 0.5. O'Callaghan et al.[120] also proved that the Prandtl number is 0.5 for linear jets by experiment.

According to Abramovich,[113] Regenscheit,[121] and Shepelev,[61] the relation between the velocity distribution and the temperature distribution in the cross-section of nonisothermal compact, linear, or radial jets within Zone III can be expressed as

$$\frac{\Delta_\theta}{\Delta_{\theta x}} = \frac{(\theta-\theta_\infty)}{(\theta_x-\theta_\infty)} = \sqrt{\frac{v}{v_x}}, \tag{7.94}$$

and thus the Prandtl number is equal to 0.5. Table 7.14 lists some temperature profile equations as they are used for practical applications.

7.4.4.3 Centerline temperature differential in a horizontally supplied jet

Compact jet

The centerline temperature differential within the zone of fully established turbulent flow (Zone III) of a nonisothermal jet can be derived using equations of momentum (Eq. 7.66) and excessive heat conservation

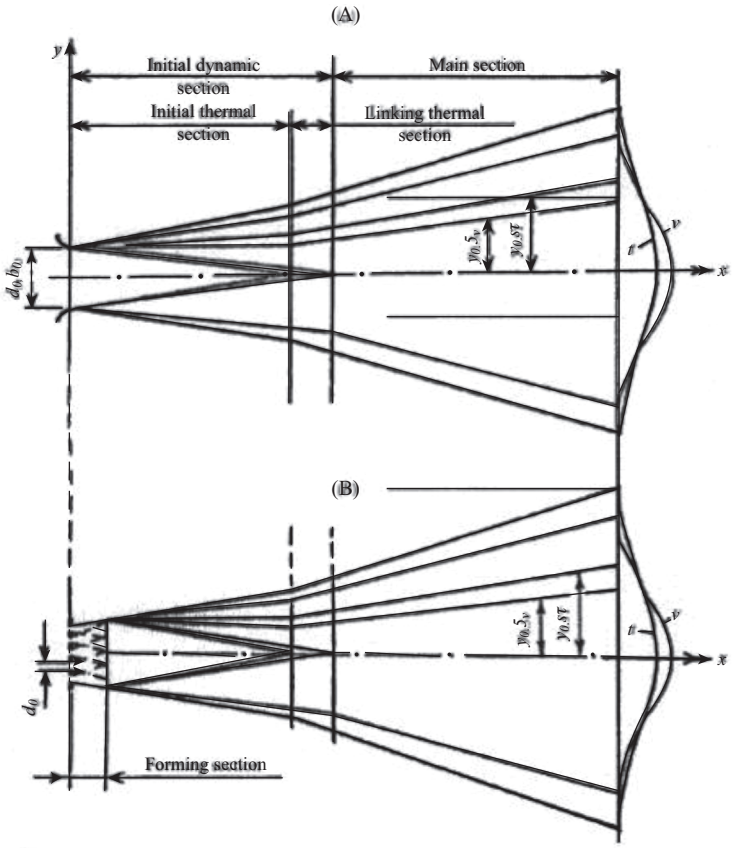


FIGURE 7.19 Schematic of a jet: (A) from an open outlet and (B) from a louvered (perforated) outlet.[112]

TABLE 7.14 Temperature profile equations.

Authors	$\frac{\theta-\theta_\infty}{\theta_x-\theta_\infty}$ Round jet	Linear jet	Radial jet
Shepelev[61]	$e^{-(1/4)(y/0.082)^2}$	$e^{-(1/4)(y/0.1x)^2}$	$e^{-(1/4)(y/0.1x)^2}$
Grimitlyn[55]	$e^{-0.7(y/y_{0.5t})^2}$	$e^{-0.7(y/y_{0.5t})^2}$	$e^{-0.7(y/y_{0.5t})^2}$
	$y_{0.5t}=x\tan\alpha_{0.5t}=x\frac{tg\alpha_{0.5v}}{\sqrt{Pr}}$	$y_{0.5t}=x\tan\alpha_{0.5t}=x\frac{tg\alpha_{0.5v}}{\sqrt{Pr}}$	$y_{0.5t}=x\tan\alpha_{0.5t}=x\frac{tg\alpha_{0.5v}}{\sqrt{Pr}}$
Abramovich[80]	$(1-r/\delta^{3/2})^{1.4}$	$(1-y/\delta^{3/2})$	–

along the jet:[60,61,114]

$$W_0=2\pi C_p\rho_\infty\int_0^{y^*}V(\theta-\theta_\infty)ydy, \tag{7.95}$$

where $W_0=c_p\rho_\infty\nu_0(\theta_0-\theta_\infty)A_0$is the excessive heat in supplied air, and C_p is the specific heat of air.

The equation for the centerline temperature differential in Zone III of the compact jet derived[59] from Eq. (7.88) using the Gauss error-function temperature profile (Table 7.14) is

$$\Delta\theta_x=\frac{1+\sigma}{2\sqrt{\pi c}}\frac{W_0}{C_p\rho_\infty}\frac{1}{\sqrt{M_0/\rho_\infty}}\frac{1}{x}. \tag{7.96}$$

Eq. (7.96) also can be presented as follows:

$$\Delta\theta_x=\frac{(1+\sigma)\theta}{2\sqrt{\pi c\phi}}\frac{\Delta\theta_0\sqrt{A_0}}{x}, \tag{7.97}$$

where

$$\Delta\theta_x=T_0-T_\infty=\frac{W_0}{C_p\rho_0Q_0}. \tag{7.98}$$

The complex of coefficients having constant value $(1+\sigma)\theta/2\sqrt{\pi c\varphi}$ is called[59] the thermal characteristic of the diffuser jet, K_2, and characterizes the temperature decay along the air jet. Assuming perfect mixing in the room (i.e., $\theta_{oz}\approx\theta_\infty$), θ_∞ can be substituted for θ_{oz}, and Eq. (7.97) can be presented as follows:

$$\frac{\theta_x-\theta_{oz}}{\theta_0-\theta_{oz}}=K_2\frac{\sqrt{A_0}}{x}. \tag{7.99}$$

As in the case of equations for the velocity decay computation, in the equations for the temperature decay computation, some researchers consider x to be a distance starting from some virtual source located at some distance x_0 from the diffuser face; others for practical reasons neglect x_0.

As in the case of the K_1 characteristic, theoretical values of characteristic K_2 depend on supply conditions. According to Shepelev,[61] in the case of air supply through a nozzle with a uniform outlet velocity profile, $K_2\times K_1=(1+\mathrm{Pr})/(2\pi0.082^2)$. Thus when $K_1=6.88$ and $\mathrm{Pr}=0.7$, $K_2=5.85$. Grimitlyn[55] suggests the following relation between K_2 and K_1 coefficients:

$$K_2=\sqrt{\frac{1+Pr}{2}}K_1. \tag{7.100}$$

According to Helander's unpublished data (Progress Report, Downward Projection of Heated Air, January 6, 1951, referenced by Koestel[114]), $K_2=6$.

Linear jet

Derivation of the equation for the centerline temperature differential in a linear jet is based on the same principles that are used in the case of a compact jet. For the linear diffuser jet, centerline temperature differential can be computed from the following equation:

$$\frac{\theta_x-\theta_{oz}}{\theta_0-\theta_{oz}}=K_2\frac{\sqrt{H_0}}{x}. \tag{7.101}$$

The centerline temperature differential in Zone III of the diffuser jet is proportional to the value of the K_2 coefficient, which, along with the K_1 coefficient, depends on jet and diffuser types and supply conditions. The theoretical value of the K_2 coefficient, according to Shepelev,[61] is 2.49. Experimental data reported by Grimitlyn[55] show K_2 to be 2.0.

Radial jet

Equations for the centerline temperature differential in radial and in conical jets (Fig. 7.17) are derived in the same way as for compact and linear jets[61] and is similar to Eq. (7.97):

$$\frac{\theta_x-\theta_\infty}{\theta_0-\theta_\infty}=\sqrt{\frac{1+\mathrm{Pr}}{4\sqrt{\pi c}\sin\alpha}}\frac{\theta}{\varphi}\frac{1}{\sqrt{\beta}}\sqrt{\frac{A_0}{x}} \tag{7.102}$$

The complex of parameters

$$\sqrt{\frac{1+\mathrm{Pr}}{4\sqrt{\pi c}\sin\alpha}}\frac{\theta}{\varphi}\frac{1}{\sqrt{\beta}} \tag{7.103}$$

is a thermal characteristic, K_2. In the case of a radial jet $\beta=2\pi$ and $\alpha=90$ degrees. Assuming $\theta=1$, $\varphi=1$, $c=0.082$, and $\mathrm{Pr}=0.5$, $K_2=1$. For conical jet with $\alpha=60$ degrees ($\beta=2\pi$, $\theta=1$, $\varphi=1$, $c=0.082$, and

TABLE 7.15 Centerline temperature differential in horizontal jets.

Authors	$\frac{\theta-\theta_\infty}{\theta_x-\theta_\infty}$ Round jet	Linear jet	Radial jet
Shepelev[61]	$K_2\frac{\sqrt{A_0}}{x}$	$K_2\frac{\sqrt{H_0}}{x}$	$K_2\frac{1}{\sqrt{\beta\sin\alpha}}\frac{\sqrt{A_0}}{x}$
Grimitlyn[79]			$K_2\frac{\sqrt{A_0}}{x}$
Abramovich[113]	$\frac{0.7}{\frac{ax}{R_0}+0.29}$	$\frac{1.04}{\sqrt{\frac{ax}{B_0}+0.41}}$	–
	$a=0.66-0.76$	$a=0.09-0.12$	
Koestel[114]	$5.35\frac{D}{x}$	–	–
Baturin[110]	$\frac{9.24R_0}{x-x_0}\sqrt{\frac{\theta_x}{\theta_0}}$	$3.27\sqrt{\frac{B_0}{x}}\sqrt{\frac{\theta_x}{\theta_0}}$	$3.27\sqrt{\frac{R_0}{(x-x_0)\left(1+\frac{x}{x_c}\right)}}\sqrt{\frac{\theta_x}{\theta_0}}$

$\mathrm{Pr}=0.5$), $K_2=1.07$. Table 7.15 lists some equations used for centerline temperature differential computation in horizontal jets.

According to Shepelev,[99] the theoretical values of the K_2/K_1 ratio for air supply through nozzles with uniform velocity distribution at the outlet cross-section is 0.9 for compact jets and 0.95 for radial, conical, and linear jets. Practically for different types of air diffusers, this ratio can vary from 0.7 to 3.0.

7.4.4.4 *Universal equations for temperature difference computation along jets supplied from outlets with finite dimensions*

Shepelev and Gelman[108] and Regenscheit[109] computed air temperature along the first three zones of jets supplied from outlets with finite size using the method of superposition of the multiple-jet system. These jets originate from the points with supply air velocity equal to the average air velocity at the outlet.

Along with principles described in Section 7.4.3, this approach utilizes the following equations describing temperature distribution in a compact jet and a heat flux at a given point (x, y, z):

$$\frac{\theta-\theta_\infty}{\theta_0-\theta_\infty}=\frac{1+\mathrm{Pr}}{2\times0.082\sqrt{\pi}}\frac{\sqrt{A_0}}{x}e^{-\frac{\mathrm{Pr}}{2}\left(\frac{r}{0.082x}\right)^2} \tag{7.104}$$

$$\frac{v(\theta-\theta_\infty)}{v_0(\theta_0-\theta_\infty)}=\frac{1+\mathrm{Pr}}{2\pi0.082^2}\frac{A_0}{x^2}e^{-\frac{1+\mathrm{Pr}}{2}\left(\frac{r}{0.082x}\right)^2}. \tag{7.105}$$

The heat flux through the finite element dA of the jet cross-section at the distance x from the outlet can be calculated as

$$\frac{d[v(\theta-\theta_\infty)]}{v_0(\theta_0-\theta_\infty)}=\frac{1+Pr}{2\pi0.082^2}\frac{dA_0}{x^2}e^{-\frac{1+Pr}{2}\left(\frac{r}{0.082x}\right)^2}. \tag{7.106}$$

The double integral of Eq. (7.106) across the outlet area $2A\times2B$ results in the following equation for a heat flux through a given point of a jet supplied through a rectangular outlet:

$$\frac{v_{xyz}(\theta_{xyz}-\theta_\infty)}{v_0(\theta_0-\theta_\infty)}=\frac{1}{4}\left(\mathrm{erf}\sqrt{\frac{1+\mathrm{Pr}}{2}}\frac{y+A}{0.082x}-\mathrm{erf}\sqrt{\frac{1+\mathrm{Pr}}{2}}\frac{y-A}{0.082x}\right)\times\left(\mathrm{erf}\sqrt{\frac{1+\mathrm{Pr}}{2}}\frac{z+B}{0.082x}-\mathrm{erf}\sqrt{\frac{1+\mathrm{Pr}}{2}}\frac{z-B}{0.082x}\right). \tag{7.107}$$

Joint solution of Eqs. (7.95) and (7.107) results in the following equation for the temperature differential along the jet axis:

$$\frac{\theta_{xyz}-\theta_\infty}{\theta_0-\theta_\infty}=\frac{\mathrm{erf}(\sqrt{(1+\mathrm{Pr})/2})(A/0.082x)\mathrm{erf}(\sqrt{(1+\mathrm{Pr})/2})(B/0.082x)}{\sqrt{\mathrm{erf}(A/0.082x)\mathrm{erf}(B/0.082x)}}. \tag{7.108}$$

In the case when air is supplied through a slot with a width $2B$ ($A=\infty$), Eq. (7.108) can be converted into

$$\frac{\theta_{xyz}-\theta_\infty}{\theta_0-\theta_\infty}=\frac{\mathrm{erf}(\sqrt{(1+\mathrm{Pr})/2})(B/0.082x)}{\sqrt{\mathrm{erf}(B/0.082x)}}. \tag{7.109}$$

Velocities and temperatures in vertical nonisothermal jets

Studies by Helander et al.,[122–125] Knaak,[126] Koestel,[114] Shepelev,[61] Regenscheit,[127] and Grimitlyn[62] resulted in equations for downward and upward projected diffuser jets.

For the circular jet, Regenscheit[127] obtained empirical equations for the maximum velocity in the downward and upward vertical jets of heated and cooled air. For a compact (round) jet,

$$\frac{v_x}{v_0}=\frac{m\sqrt{A_0}}{x}\pm\sqrt{\frac{\mathrm{Ar}_0}{m}\left[1+\ln\frac{2x}{m\sqrt{A_0}}\right]}, \tag{7.110}$$

where m is the parameter characterizing the diffuser jet (m from 0.1 to 0.3). For a linear jet,

$$\frac{v_x}{v_0}=\frac{\sqrt{H_0}}{x}\pm\sqrt{\frac{\mathrm{Ar}_0}{0.2}\left(2.83\sqrt{\frac{x}{H_0}}-1\right)}. \tag{7.111}$$

Based on theoretical analyses, Koestel,[114] Shepelev,[61] and Grimitlyn[62] developed equations for velocities and temperatures in vertical heated and chilled air jets. The assumptions used by these authors are similar, and the method used is described in Koestel.[114] The assumptions used in the analysis can be summarized as follows:

1. The jet of warmed or cooled air is projected into an unbounded atmosphere of still air of uniform temperature.

2. The only force opposing the downward flow of the heated air or upward flow of the cooled air is a buoyancy force. In their analysis, Helander and Jakowatz[128] also suggested accounting for inertial forces due to the entrainment of room air. However, this suggestion is not in an agreement with a principle of momentum conservation used in most of the existing models for isothermal jets.
3. The air entrained by the jet has room air temperature.
4. A velocity profile and a temperature difference profile have shapes that can be approximated by an error-function type curve.

For practical use the influence of buoyancy forces on temperature and velocity decay in vertical nonisothermal jets, as proposed by Grimitlyn,[112] can be accounted for by the coefficient K_n of nonisothermality. For compact jets,

$$\frac{v_x}{v_0} = K_1 \frac{\sqrt{A_0}}{x} K_n \tag{7.112}$$

$$\frac{\theta_x - \theta_{oz}}{\theta_0 - \theta_{oz}} = K_2 \frac{\sqrt{A_0}}{x} \frac{1}{K_n}, \tag{7.113}$$

where K_n for a compact jet can be computed as

$$K_n = \sqrt[3]{1 \pm 2.5 \frac{K_2}{K_1^2} \mathrm{Ar}_0 \left(\frac{x}{\sqrt{A_0}} \right)^2}. \tag{7.114}$$

For linear jets,

$$\frac{v_x}{v_0} = K_1 \frac{\sqrt{H_0}}{x} K_n, \tag{7.115}$$

$$\frac{\theta_x - \theta_{oz}}{\theta_0 - \theta_{oz}} = K_2 \frac{\sqrt{H_0}}{x} \frac{1}{K_n}, \tag{7.116}$$

where

$$K_n = \sqrt[3]{1 \pm 1.8 \frac{K_2}{K_1^2} \mathrm{Ar}_0 \left(\frac{x}{H_0} \right)^{1.5}}. \tag{7.117}$$

By applying the Ar_x criterion, Eqs. (7.114) and (7.117) can be transformed into

$$K_n = \sqrt[3]{1 \pm a \mathrm{Ar}_x}, \tag{7.118}$$

where $a = 2.5$ for axially symmetric and incomplete radial jets and $a = 1.8$ for linear jets. The plus sign in Eq. (7.118) corresponds to the situation when the directions of buoyancy and inertia forces coincide, whereas the minus sign corresponds to their counteraction. This equation can be used for vertical nonisothermal jets at $\mathrm{Ar}_x \le 0.25$.

Nonisothermal jet throw

The throw of downward-projected heated jets or upward-projected chilled jets can be derived from Eqs. (7.112) and (7.115) for K_n equal to some value, for example, 0.1. Helander and Jakowatz,[122] in their work on heated jets projected downward, have called attention to some of the differences between the actual conditions and those assumed for analysis. One of these is the radial escape of warm air in the terminal zone of a hot stream projected downward. This escaping warm air then rises and causes a change in ambient conditions for the upper part of the jet. The terminal zone and the edges of the jet are zones of marked instability, with definite surges and fluctuations, so that the jet envelope is very difficult to define or to determine experimentally. In the closure to the paper presented by Knaak,[126] Dr. Helander suggested that from the point of view of practical application, the distance to the beginning of the unstable, terminal zone of the jet is about 80% of the jet throw.

The data from Baturin and Shepelev,[129] Helander et al.,[123] Regenscheit,[121] Turner,[130] Shepelev,[61] Seban et al.,[131] Sato et al.,[132] Grimitlyn,[112] Mizuchina,[133] and Weidemann and Hanel[134] show that maximum downward/upward travel of heated/cooled compact jets can be evaluated using the equation

$$\frac{z_{\max}}{\sqrt{A_0}} = \frac{a}{\sqrt{\mathrm{Ar}_0}}, \tag{7.119}$$

where the coefficient a varies between 1.59 and 2.57 with a mean value of $a = 1.8 \pm 0.3$.

Considering that most of the data were obtained using round nozzles with jet characteristics close to ideal discharge conditions ($K_1 = 6.88$ and $K_2 = 5.85$), Eq. (7.119) can be presented as[112]

$$\frac{z_{\max}}{D} = \frac{0.63 K_1}{\sqrt{K_2 \mathrm{Ar}_0}}. \tag{7.120}$$

The throw of downward-projected heated linear jets or upward-projected chilled air jets can be calculated as[112]

$$\frac{z_{\max}}{D} = \frac{0.67 K_1^{4/3}}{(K_2 \mathrm{Ar}_0)^{2/3}}. \tag{7.121}$$

Trajectory of horizontal and inclined jets

Buoyancy forces influence the trajectory of horizontally projected air jets or air jets supplied at some angle to the horizontal plane (Fig. 7.20). Most nonisothermal air jet studies were devoted to horizontally projected compact air jets. Based on the analytical studies,[60,99,135–138] the trajectory axis of inclined jets can be described by a polynomial function

$$\frac{z}{\sqrt{A_0}} = \Psi \frac{K_2}{K_1^2} \mathrm{Ar}_0 \left(\frac{x}{\sqrt{A_0}} \right)^3. \tag{7.122}$$

Similar equations were suggested by experimental studies.[128,139,140] Some authors determine the trajectory

axis from equations of another kind—parabola, for example.[141,142]

In some studies on inclined air jets, the equation for the trajectory differs from Eq. (7.122) by the additional term as follows:

$$\frac{z}{\sqrt{A_0}} = \frac{x}{\sqrt{A_0}} tg\alpha_0 \pm \Psi \frac{K_2}{K_1^2} \mathrm{Ar}_0 \left(\frac{x}{\sqrt{A_0}}\right)^3. \tag{7.123}$$

Taliev[143] and Schneider[144] derived the equations for the trajectories by numerical methods. Experimental data for the trajectory of the inclined jet ($\alpha_0 \neq 0$) were obtained only by Fleishbacker and Schneider.[138]

As was shown by Zhivov,[145] the main difference in most of the equations for the jet trajectory is the value of the coefficient ψ. The differences in experimental data obtained by different authors are mainly due to the difficulties in the measurements of nonisothermal air jets supplied with low initial velocities (2−10 m/s). There is also a different understanding of the term "air jet trajectory." Some authors mean the points with maximum velocity values, while others mean the centers of gravity of the cross-sections of the jet.

The analytical method of jet trajectory study developed by Shepelev[61] allows the derivation of several other useful features and is worth describing. On the schematic of a nonisothermal jet supplied at some angle α_0 to the horizon (Fig. 7.21), S is the jet's axis, X is the horizontal axis, and Z is the vertical axis. The ordinate of the trajectory of this jet can be described as $z = xtg\alpha + \Delta z$, where Δz is the jet's rise due to buoyancy forces. To evaluate Δz, the elementary volume dW with a mass equal to $dm = \rho_s\, dV$ on the jet's trajectory was considered. The buoyancy force influencing this volume can be described as $dP = g(\rho_\infty - \rho_s)$. Vertical acceleration of the volume under the consideration is $j = dP/dm = g(\rho_\infty - \rho_s)/\rho_s \approx g(T_s - T_\infty)/T_s$. Vertical acceleration can be presented with the help of the vertical velocity component $j = dV_z/d\tau$, where the time interval $d\tau$ can be described as dS/V_s. Based on these equations, the vertical component of air velocity can be presented as

$$v_z = \frac{g}{T_\infty} \int_0^s \frac{T_s - T_\infty}{v_s} dS. \tag{7.124}$$

The ratio of local temperature difference and velocity on the jet's axis in Eq. (7.124) can be substituted by

$$\frac{T_s - T_\infty}{v_s} = \frac{K_2}{K_1} \frac{g}{T_\infty} \frac{T_0 - T_\infty}{v_0}, \tag{7.125}$$

resulting in

$$v_z = \frac{K_2}{K_1} \frac{g}{T_\infty} \frac{T_0 - T_\infty}{v_0} S. \tag{7.126}$$

Considering that $v_z = dz/d\tau$, $v_z/v_s = dz/dS$ and $v_s = K_1 v_0 \sqrt{A_0}/S$, the equation for calculating Δz can be rewritten as

$$\Delta z = \frac{K_2}{K_1} \frac{g}{T_\infty} \frac{T_0 - T_\infty}{v_0} \int_0^s \frac{S dS}{v_s} \tag{7.127}$$

or

$$\Delta z = \frac{K_2 g (T_0 - T_\infty)}{3 K_1^2 T_\infty v_0^2 \sqrt{A_0}} S^3. \tag{7.128}$$

Substituting $x \cos \alpha_0$ for S, and Ar_0 for the complex of parameters, the resulting equation for the trajectory is

$$z = xtg\alpha_0 \pm \frac{1}{3} \frac{K_2}{K_1^2} \mathrm{Ar}_0 \left(\frac{x}{cos\alpha_0}\right)^3. \tag{7.129}$$

When a chilled air jet is supplied at the angle α_0 upward, it will cross the level of the supply outlet at the distance x_0. This distance can be calculated by

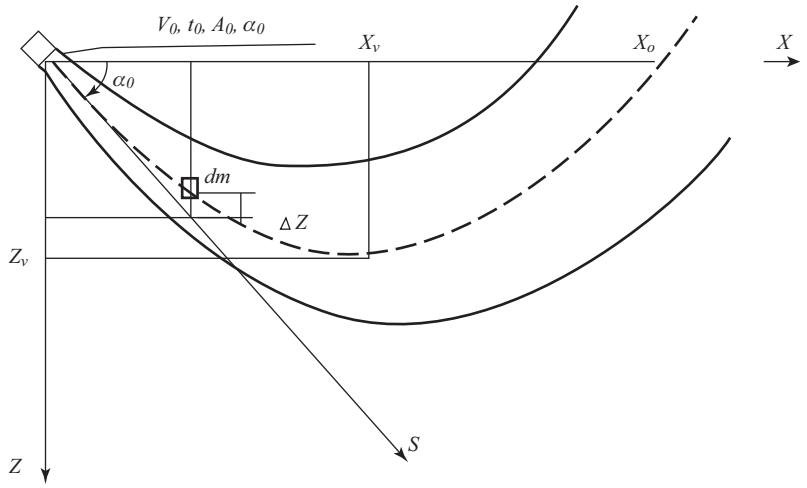


FIGURE 7.20 Schematic of an inclined jet.

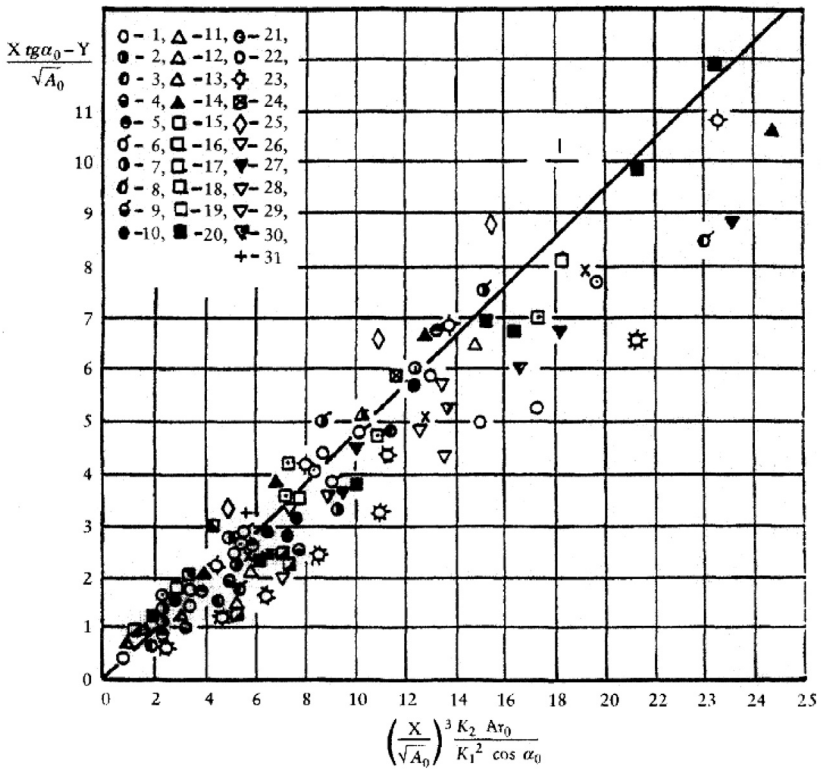


FIGURE 7.21 Experimental evaluation of the coefficient ψ. Source: *Reproduced from Zhivov AM. Theory and practice of air distribution with inclined jets. ASHRAE Trans 1993;**99**:1.*

substituting $z = 0$ in Eq. (7.129):

$$x_0 = \frac{\sqrt{3}K_1 cos\alpha_0 \sqrt{sin\alpha_0}}{\sqrt{K_2 Ar_0}}. \tag{7.130}$$

The abscissa and ordinate of the jet vortex in the case of upward inclined cold air jet supply or downward inclined warm air supply were derived from Eq. (7.130):

$$x_v = \frac{K_1 \cos\alpha_0 \sqrt{\sin\alpha_0}}{\sqrt{K_2 Ar_0}}. \tag{7.131}$$

$$z_v = \frac{2}{3}\frac{K_1 (\sin\alpha_0)^{3/2}}{\sqrt{K_2 Ar_0}}. \tag{7.132}$$

The ratio x_v to z_v depends only on $tg\alpha_0$: $z_v/x_v = 2/3\ tg\alpha_0$, and the ratio of x_v/x_0 has a constant value equal to 0.578. To clarify the trajectory equation of inclined jets for the cases of air supply through different types of nozzles and grills, a series of experiments were conducted.[145] The trajectory coordinates were defined as the points where the mean values of the temperatures and velocities reached their maximum in the vertical cross-sections of the jet. It is important to mention that, in such experiments, one meets with a number of problems, such as deformation of temperature and velocity profiles and fluctuation of the air jet trajectory, which reduce the accuracy in the results. The mean value of the coefficient ψ obtained from experimental data (Fig. 7.21) is 0.47 ± 0.06. Thus the trajectory of the nonisothermal jet supplied through different types of outlets can be calculated from

$$\frac{z}{\sqrt{A_0}} = \frac{x}{\sqrt{A_0}} tg\alpha_0 \pm 0.47 \frac{K_2}{K_1^2} Ar_0 \left(\frac{x}{\sqrt{A_0}}\right)^3. \tag{7.133}$$

The accuracy of this value is sufficient (Fig. 7.22) to be used in designing the trajectory of inclined ventilation jets at an angle $\alpha_0 \leq \pm 45$ degrees. Using the experimental value of the coefficient, the equation for the vortex abscissa x_v can be presented as follows:

$$x_v = \frac{\cos\alpha_0 \sqrt{|\sin\alpha_0|\sqrt{A_0}}}{\sqrt{3 \times 0.47 (K_1/K_2^2) Ar_0}}. \tag{7.134}$$

However, this clarification does not affect the ordinate-to-abscissa ratio, which remains equal to 0.578.

For the nonisothermal linear air jet, the trajectory equation is derived by Shepelev[61] is

$$\frac{z}{H_0} = \frac{x}{H_0} tg\alpha_0 \pm 0.4 \frac{K_2}{K_1^2} Ar_0 \left(\frac{x}{H_0}\right)^{5/2}. \tag{7.135}$$

7.4.4.5 *Jet attachment*

Jets discharging close to the plane of the ceiling or wall are common in ventilation practice. The presence of an adjacent surface restricts air entrainment from the side of this surface. This results in a pressure

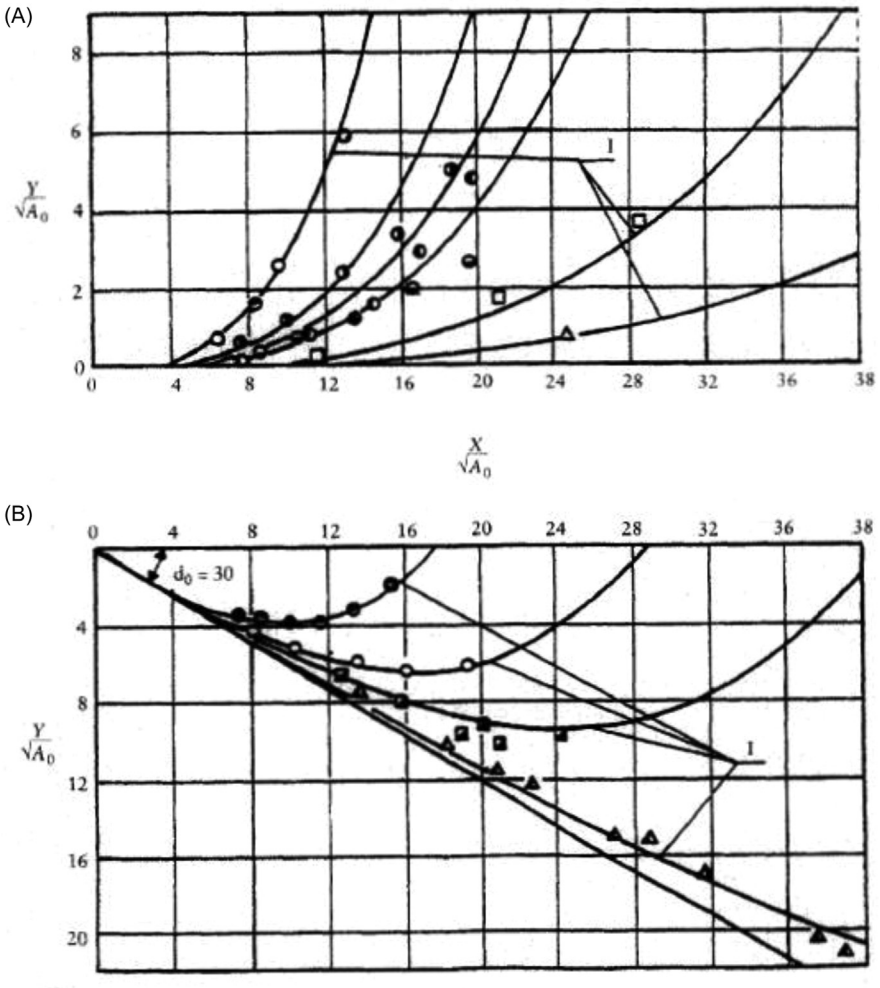


FIGURE 7.22 Trajectory of a nonisothermal jet supplied at an angle: (A) $\alpha_0 = 30$ degrees and (B) Eq. (7.133). Source: *Reproduced from Zhivov AM. Theory and practice of air distribution with inclined jets.* ASHRAE Trans *1993;**99**:1.*

difference across the jet, which therefore curves toward the surface. The curvature of the jet increases until it attaches to the surface. This phenomenon is usually referred to as a Coanda effect. The attached jet or, as it is commonly called, wall jet, can result from air supply through an outlet with one edge coincident with the plane of the wall or ceiling (Fig. 7.23). Jets supplied at some distance from the surface or at some angle to the surface can also become attached (Fig. 7.24)

The results of studies by Kerka of jets supplied through rectangular outlets[58] with and without an adjacent surface indicate an increase of 1.27−1.45 times in the velocity decay coefficient for wall jets compared with free jets from the same outlets. The angle of divergence of the wall jet in the direction perpendicular to the wall was slightly less than one-half of a free jet, while the angle of spread of the jet along the wall was greater than the divergence of a free jet.

The results of experimental studies of compact wall jets by Mitkalinny,[148] Abdushev, Baharev, and Fedorov; on linear wall jets by Kerka and Sakipov; and on radial wall jets by Gelman are summarized by Grimitlyn[55] and presented in Fig. 7.25. These data indicate that the parameter $K^{wall} = K_1^{wall}/K_1$ reflecting the influence of the wall on the velocity decay along the jet increases from 1 to 1.4 with distance from the outlet. For a compact jet $K^{wall} = 1$ when $x < 5d_0$; for a linear jet K^{wall} reaches its maximum value of 1.4 only at $x < 20b_0$, where b_0 is an outlet width.

Studies of wall jets[110,111] show that they have two layers: a turbulent boundary layer close to the wall and an outer shear layer. The thickness of the boundary wall layer can be neglected for practical purpose. Accordingly to compute the maximum velocity in the wall jet, researchers[51,99,121] apply the method of images by treating the wall jet as one-half of a free jet. Application of this method gives a relationship between the characteristics of a wall jet and a free jet, which results in a correction factor equal to $\sqrt{2}$. This approach has some inaccuracy even with linear and radial jets. For a three-dimensional wall jet, the procedure is even more approximate. Discussion by Etheridge and Sandberg[146] of some previous studies of attached jets indicates some loss of momentum in an

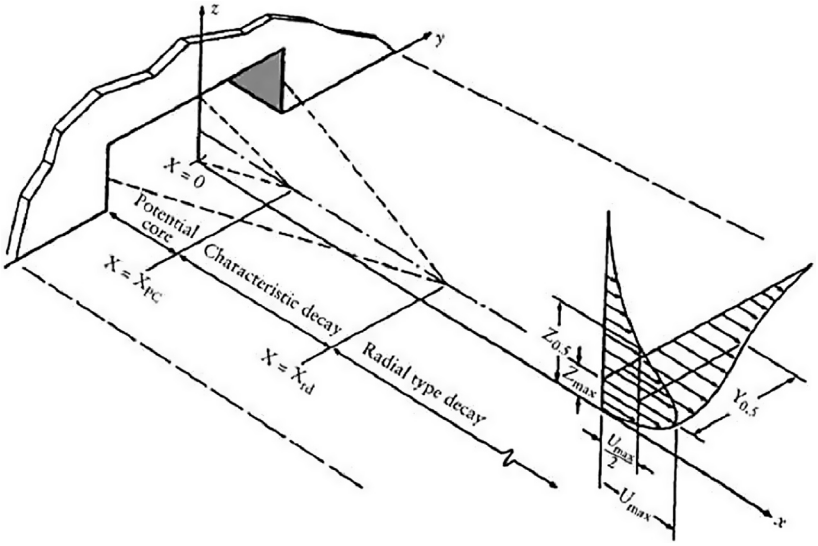


FIGURE 7.23 Three-dimensional wall jet. Source: *Reproduced from Etheridge D, Sandberg M. Building ventilation: theory and measurements. Chichester; New York: John Wiley & Sons; 1996.*[146]

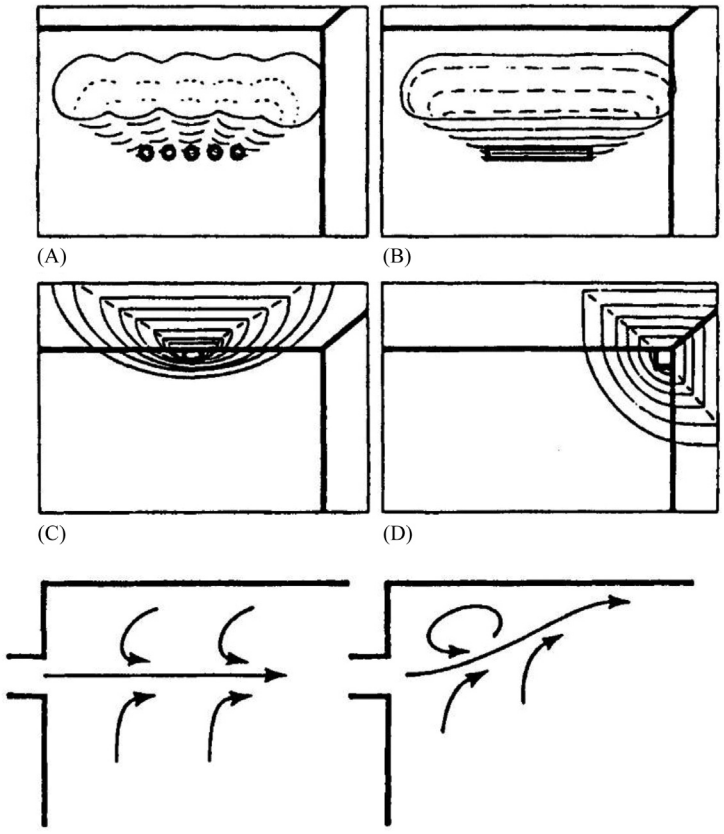


FIGURE 7.24 Jet attachment with air supply through outlets located at some distance from the surface: (A) multiple jets, (B) long slot jet, (C) rectangular jet, (D) rectangular corner jet, and (E) general schematic. Source: *Reproduced from Nielsen PV.* Luftstrømning i Ventilerede Arbejdslokaler [Ventilation of working areas]. *SBI-Rapport 128. Statens Byggefoskningsinstitut; 1981.*[147]

attached jet due to the friction against the surface. The authors compiled information from previous studies, which is summarized in Table 7.16. According to the equations presented in Table 7.16, the maximum velocity in a wall jet is inversely proportional to the distance from the outlet in a different power compared with a free jet.

It is not uncommon to supply air into the room with jets attached both to the ceiling and to the wall surfaces.[55,147] Air jets can be parallel to both surfaces or be directed at some angle to one or both surfaces (Fig. 7.24). Studies of compact wall jets supplied parallel to both surfaces reported by Grimitlyn[55] show that the correction factor value is in the range from 1.6 to

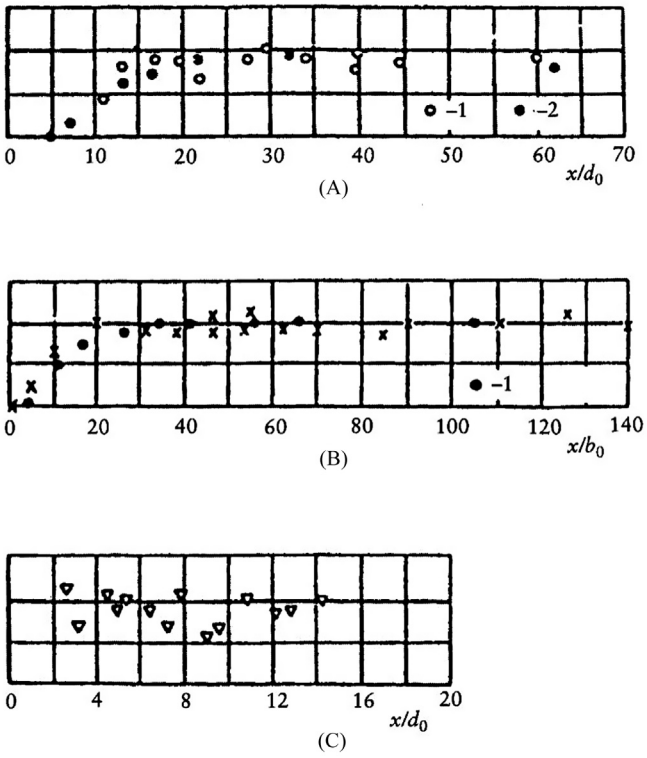


FIGURE 7.25 Correction parameter K_{wall} reflecting the influence of the wall: (A) compact jets (experimental data from V. Mitkalinny, A. Abdushev, V. Baharev, and L. Fedorov), (B) linear jets (experimental data from W. Kerka and Z. Sakipov), and (C) radial jets (experimental data from N. Gelman). Source: *Reproduced from Grimitlyn MI.* Air distribution in rooms. *Moscow, Russia: Stroiizdat; 1982.*

TABLE 7.16 Maximum velocity decay in wall jets.

Type of jet	Decay of the maximum velocity	References
Linear wall jet	$\sim x^{-0.555}$	Schwarz and Cosart[149]
	$\sim x^{-0.375}$	Regenscheit[121]
Radial wall jet	$\sim x^{-1.12}$	Bakke[150]
	$\sim x^{-1.15}$	Waschke[151]
Rectangular wall jet		
$H_0/L = 0.025$	$\sim x^{-1.15}$	Sforza and Herbst[152]
$H_0/L = 0.05$	$\sim x^{-1.09}$	
$H_0/L = 0.01$	$\sim x^{-1.15}$	
$H_0/L = 1.0$	$\sim x^{-1.14}$	
Compact wall jet	$\sim x^{-1}$	Nielsen[153]

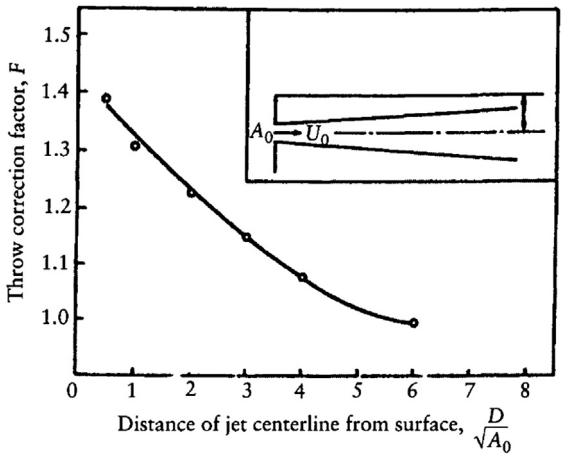


FIGURE 7.26 Correction factor F for surface proximity. Source: *Reproduced from Farquharson IMC. The ventilating air jet.* JIHVE *1952;**19**:449–69.*

1.7, which means that restriction of entrainment from two sides reduces velocity decay by 20%–30% compared with the case of a wall jet.

When a jet is supplied at some distance from the surface, the attachment occurs when the distance between the outlet and the surface is below a critical distance; otherwise the jet will propagate as a free jet.[154] If the jet attaches to the surface, the flow downstream of the attached point is similar to that of a wall jet. For a compact isothermal jet, the critical distance for jet attachment to the surface is $L_{crit} = 6A_0^{1/2}$.[155] For $L_{crit} < 6A_0^{1/2}$ the velocity decay coefficient K_1 becomes greater than it would be in the case of free jet, and should be corrected using correction factor F (see Fig. 7.26) to compensate for surface proximity.

The length of the recirculation zone, x_a, for a linear jet (the distance to the point of jet attachment to the surface) was studied by Sawyer,[156] Miller and Comings,[93] and Bourque and Newman.[157] The results of these studies, summarized in Awbi,[154] show that the length of the recirculation zone (Fig. 7.27) is proportional to the distance from the outlet to the surface and can be described as

$$x_a/H_0 = 0.73(D/H_0) - 2.3, \qquad (7.136)$$

where H_0 is the width of the outlet.

Sandberg et al.[158] conducted similar tests with a heated linear jet so that the buoyancy forces opposed the forces due to the lower pressure in the circulation zone (bubble). Based on the results of these tests, it was concluded that heating the jet does not change the location of the attachment point. For $5 < D/H_0 < 13$ the length of the circulation zone followed the relation

$$x_a/H_0 = 1.175(D/H_0) + 6.25. \qquad (7.137)$$

Calculation of x_a/H_0 using Eqs. (7.136) and (7.137) results in significantly different results. For $D/H_0 = 8$, $x_a/H_0 = 3.54$ according to Eq. (7.136), and $x_a/H_0 = 15.65$ according to Eq. (7.137).

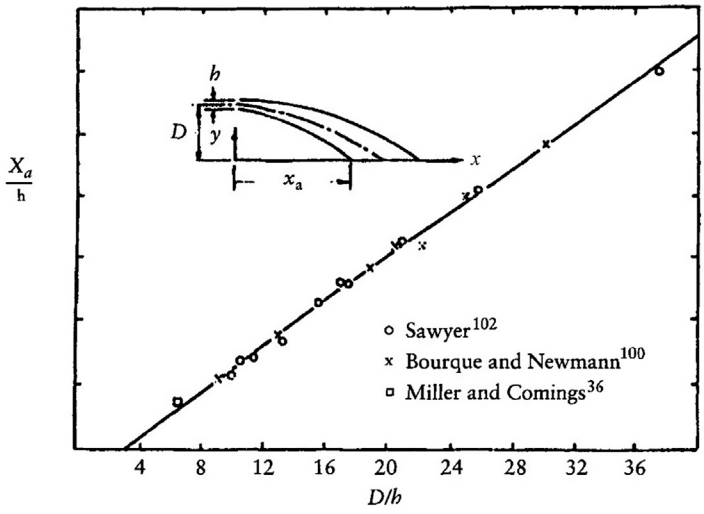


FIGURE 7.27 Effect of supply distance from surface on the attachment distance for a linear jet. Source: *Reproduced from Awbi HB.* Ventilation of buildings. *London: Chapman and Hall; 1991.*

The length of the circulation zone (bubble), L_b, created when the linear jet is supplied at an angle α to the surface was studied experimentally by Bourque and Newman[157] and theoretically by Sawyer.[156] The effect of the angle between the jet axis at the outlet and the surface on the length of the circulation bubble is shown in Fig. 7.28, reproduced from Awbi.[154] The data presented in Fig. 7.28 show that at sufficiently high Reynolds number the length of the circulation zone is independent of the Reynolds number.

Linear jet attachment to a plane not parallel to the supply direction was studied by Katz.[159] The critical angle, θ_c, of the plane to the jet supply direction, as indicated in Fig. 7.29, was found to be dependent on the supply velocity (Reynolds number). It also depends on the distance of the plane edge from the supply outlet (see Fig. 7.30).

Baturin[110] studied air jets supplied from rectangular nozzles at some angle to the plane with an edge of the nozzle coincident with the plane. The results of his studies indicate that the critical value of the angle of the jet supply direction to the plane is 45 degrees. It was also shown that the jet supplied through a rectangular outlet with a nozzle located at some distance from the plane does not attach to the surface.

7.4.4.6 Jet separation

When the temperature of an air jet attached to the ceiling is lower than the temperature of the ambient air, the jet will remain attached to the ceiling until the downward buoyant force becomes greater than the upward static pressure (Coanda force). At this point, the jet separates from the ceiling and begins a downward-curving trajectory.[160] Studies of nonisothermal jets conducted by Grimitlyn,[62] Schwenke,[161] Nielsen and Moller,[162] Miller,[160] Anderson et al.,[163]

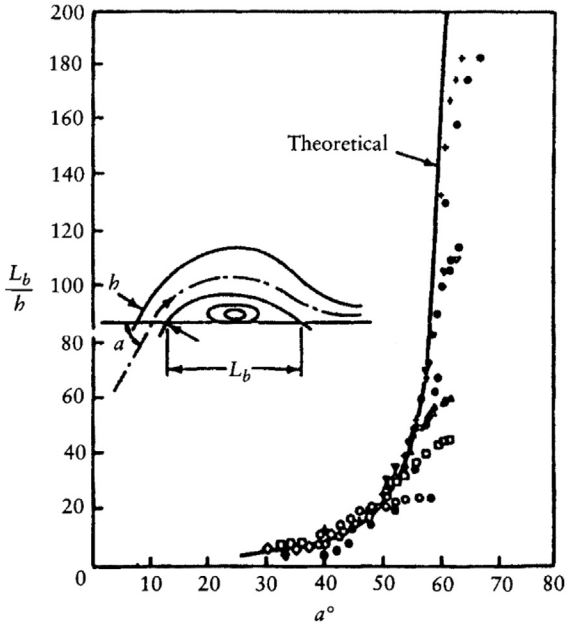


FIGURE 7.28 Effect of supply jet angle on recirculation bubble length. Experimental data and theoretical curve from Sawyer.[99] Source: *Reproduced from Awbi HB.* Ventilation of buildings. *London: Chapman and Hall; 1991.*

and Kirkpatrick et al.[164] showed that the distance to the point of the jet's separation can be computed using the following equation:

$$x_{sep} = \frac{a}{(\mathrm{Ar}_0)^b}. \tag{7.138}$$

For linear diffuser jets[165] $a = 2.5H_0$ and $b = 2/3$. For compact diffuser jets $a = 1.6A_0^{0.5}$ and $b = 0.5$.[163,164] According to theoretical analysis and experimental data collected by Grimitlyn,[60] the separation distance of jets could be expressed by equations similar to

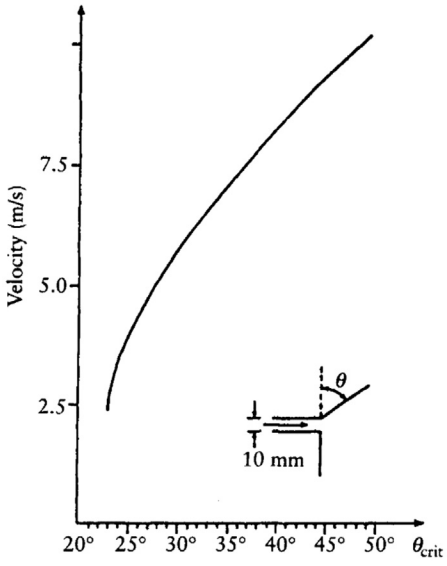


FIGURE 7.29 Critical angle. Wall is located close to supply outlet. Source: *Reproduced from Katz P. Der Coanda-effect. Gesundheits-Ingenieur 1973;6:169–92.*

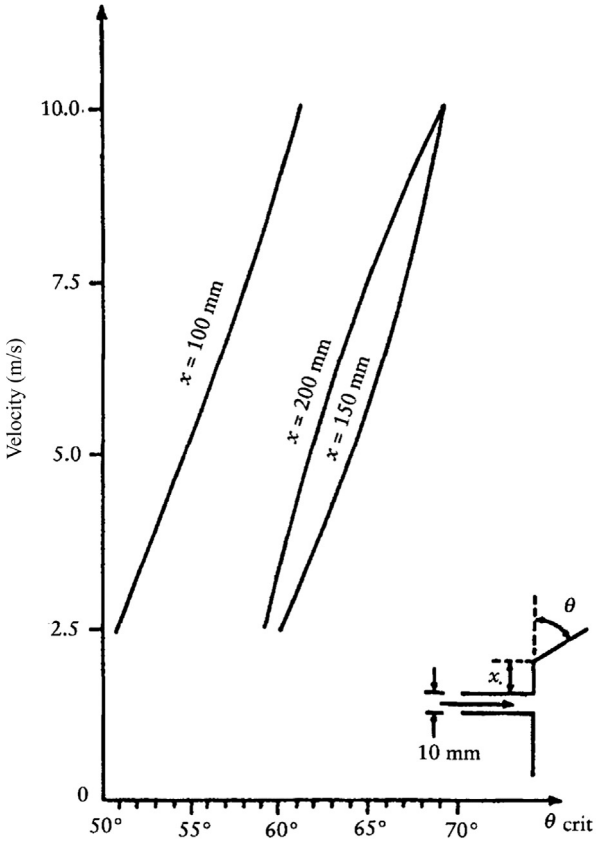


FIGURE 7.30 Critical angle. Wall is located at different distances from the air supply outlet. Source: *Reproduced from Katz P. Der Coanda-effect.* Gesundheits-Ingenieur *1973;6:169–92.*

Eq. (7.138) considering diffuser characteristics K_1 and K_2. For compact and incomplete radial jets,

$$x_s = \frac{0.55K_1\sqrt{A_0}}{\sqrt{K_2 \mathrm{Ar}_0}}. \tag{7.139}$$

For linear jets,

$$x_s = \frac{0.4K_1^{3/4}H_0}{(K_2 \mathrm{Ar}_0)^{2/3}}. \tag{7.140}$$

For radial jets,

$$x_s = \frac{0.45K_1\sqrt{A_0}}{\sqrt{K_2 \mathrm{Ar}_0}}. \tag{7.141}$$

7.4.5 Jets in confined spaces

7.4.5.1 General description of confined flow

Current mixing-type air distribution methods typically consider occupied zone ventilation with jets intercepting its upper boundary. These methods include air supply with vertical jets through ceiling-mounted air diffusers and air supply with inclined jets. They also include air supply with vertical upward-directed jets or horizontal jets along room surfaces. In the latter case, the jet reaches the opposite wall/ceiling and follows room surfaces until it reaches the occupied zone (Fig. 7.31). If the combination of room sizes (height, length, and width) allows such an airflow pattern, this room is considered to be "short."[51] The room in which air jets dissolve before reaching the opposite wall is considered to be "long." In such rooms, the occupied zone is ventilated by "reverse" flow, and secondary and tertiary vortexes. Initially studies of jets in confined spaces were carried out for mining, chemical, and mechanical engineering applications.[60,89,141] In this chapter four methods of air supply in confined spaces are discussed:

- horizontal jet supply,
- inclined jet supply,
- horizontal jet supply with directing jets, and
- vertical jets.

7.4.5.2 Experimental studies of isothermal horizontal jets in confined spaces: airflow pattern, throw, and velocities

The first experimental data on confined air jets used for ventilation date back to 1939, when Baturin and Hanzhonkov studied air supply method with the occupied zone ventilation by "reverse" flow. Later, this method was called concentrated air jet supply. Baturin and Hanzhonkov concluded that the airflow pattern in the ventilated space depends on the location of air supply outlets and practically does not depend on location of air exhausts.[166]

Studies by Nelson and Stewart,[167] Bromley,[168] and Gunes[169] provide experimental data on air velocities and temperature distribution for this method of air supply at different room configurations, locations of air supply outlets, and velocities and temperatures of air supply.

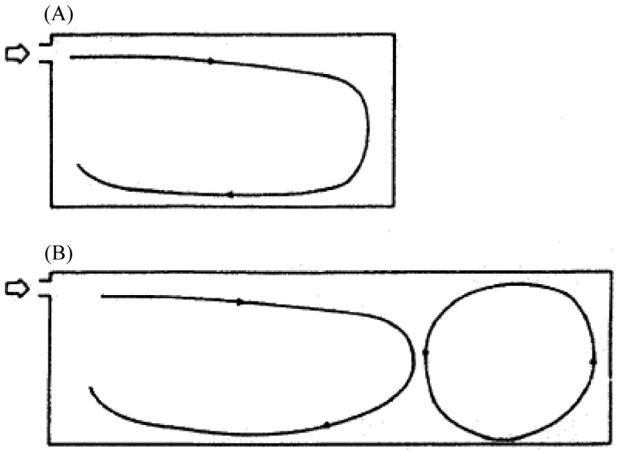


FIGURE 7.31 Jet flow in a room: (A) "short" room and (B) "long" room. Source: *Reproduced from Etheridge D, Sandberg M. Building ventilation: theory and measurements. Chichester; New York: John Wiley & Sons; 1996.*[147]

The effect of the room length and position and shape of the air supply outlets was studied by Linke.[170,171] These studies show that there is a maximum room length that can be effectively ventilated by the supply air jet (Fig. 7.32A). For the linear (2D) air jet attached to the ceiling supplied at the Reynolds number in the range from 1825 to 12,000, the maximum room length does not exceed three times the room width. The rest of the room downstream is poorly ventilated. When the air supply slot is symmetrical (located at $1/2H$), the effectively ventilated room length increases to four room widths. Air supply through a round nozzle with a nonattached jet allows the effectively ventilated room length to increase up to five transversal cross-sectional sizes, $(B \times H)^{1/2}$, of the room.

The airflow pattern in rooms ventilated by linear attached jets with L/H ratio greater than that for effectively ventilated rooms was studied by Schwenke[161] and Müller.[45] The results of their air velocity measurements and visualization studies indicate that there are secondary vortexes formed downstream in the room and in the room corners. The number of downstream vortexes and their size depend on the room length (Fig. 7.32B). Mass transfer between the primary vortex and the secondary vortex depends on the difference in characteristic air velocities in the corresponding flows

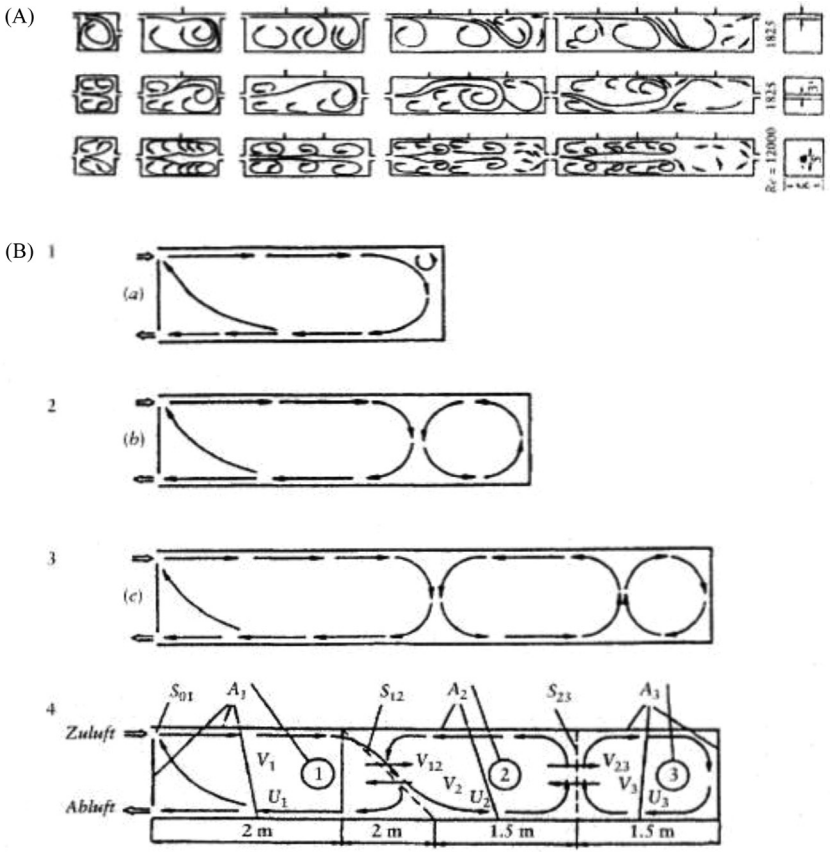


FIGURE 7.32 Flow patterns in rooms of different lengths with various types of air supply and exhaust. 1—$L_r/H_r = 3$; 2—$L_r/H_r = 4$; 3—$L_r/H_r = 6$; and 4—schematic of primary, secondary, and tertiary vortexes in the room with $L_r/H_r = 6$. Source: *Reproduced from (A) Linke W. Eigenschaften der strahlüftung (Aspects of jet ventilation). Kältetechnik Klimatech 1966;**18**:3 (BSRAE Translation 103); (B) Müller HJ. Einfluß der geometrischen Verhältnisse auf die raumunströmung bei der Strahllüftung. Luft-und Kältetechnik 1977;**6**.*

U_1 and U_2 and can be described using the Stanton number, St[45]:

$$\mathrm{St} = \frac{U_1 + U_2}{U_1 - U_2}\frac{1}{4\sigma\sqrt{\pi}}, \tag{7.142}$$

where

$$\sigma = \frac{1}{2\sqrt{\chi c \frac{U_1 - U_2}{U_1 + U_2}}}, \tag{7.143}$$

where χ and c are empirical coefficients. For the jet spreading along the wall ($U_2 = 0$), the Stanton number is equal to 0.01. This approach was used to predict mass transfer between the primary and secondary vortexes and the characteristic air velocities in the secondary vortexes. These predictions were compared with experimental data. Though experimental data deviate from predicted air velocities, the proposed model provides some understanding of the mechanism of mass transfer between different room zones. Average rotation velocity and mass transfer decreases from the primary vortexes to the secondary and the subsequent vortexes.

The influence of room transverse cross-sectional configuration on airflow patterns created by air jets supplied through round nozzles in proximity to the ceiling was studied by Baharev and Troyanovsky[172] and Nielsen[146] (see Fig. 7.33). Based on experimental data, they concluded that when the room width B is less than $3.5H$, the jet attaches to the ceiling and spreads, filling the whole width of the room in the manner of a linear jet. The reverse flow develops under the jet. When $B > 4H$, the reverse flow also develops along the jet sides. Baharev and Troyanovsky[172] indicated that air temperature and velocity distribution in the occupied zone is more uniform when the jet develops in the upper zone and the occupied zone is ventilated by the reverse flow. Thus they proposed limiting room width to $3-3.5H_r$.

Detailed experimental data were obtained by Sadovskaya[173,174] on a physical model in isothermal conditions. She has found that the confined air jet has two critical cross-sections (Fig. 7.34). In the first cross-section, where the ratio of jet cross-sectional area to the area of ventilated space equals 0.24, the jet develops as a free jet. Between the first and second critical cross-sections, where the jet occupies 40% of the room cross-sectional area, is the zone of confined jet. Beyond the second critical cross-section is the zone of jet degradation. Sadovskaya has found that the lengths of all three zones depend on the coefficient of turbulent structure a of the jet at the air supply and determined empirical equations for the length of each zone and air velocities in the air jet and in the reverse flow:

$$x_1 = \frac{0.1\sqrt{BH}}{a} \tag{7.144}$$

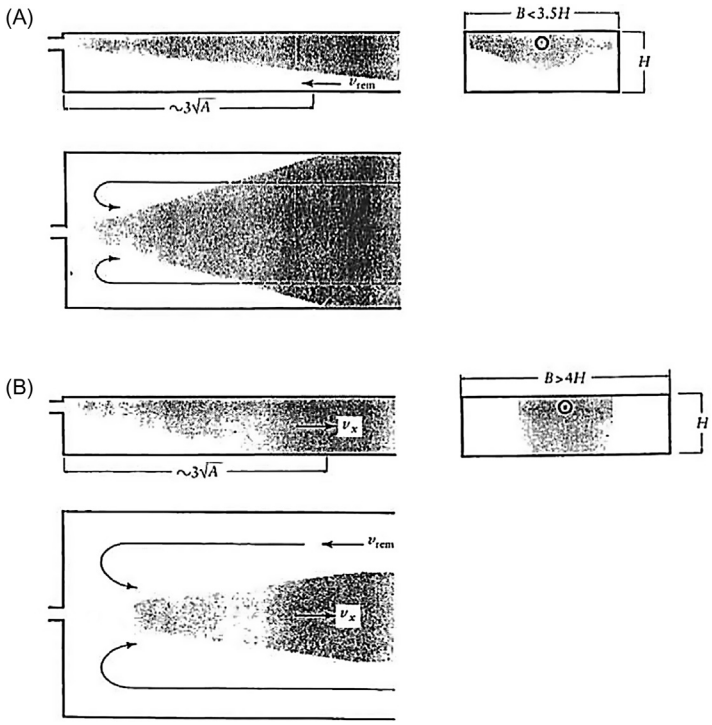


FIGURE 7.33 Influence of room configuration on airflow pattern: (A) $B/H < 3.5$ and (B) $B/H > 4$. Source: *Reproduced from Nielsen PV. Luftstrømning i Ventilerede Arbejdslokaler [Ventilation of working areas]. SBI-Rapport 128. Statens Byggeforskningsinstitut; 1981.*

$$x_2 = [0.21(BH^{1/2})]/a. \tag{7.145}$$

In these studies, nozzles with $a = 0.07$ were used. For the values of parameter $(BH/A_0)^{1/2}$ used in the studies, from 2.44 to 71.5, maximum jet throw is in the range

$$x_{\max} = (\text{from } 4.07 \text{ to } 5.1)\sqrt{BH}.$$

Based on experimental data from Sadovskaya[173,174] and Rozenberg[175] as well their own experimental results, Baharev and Troyanovsky[172] derived empirical equations to design air distribution with horizontally supplied confined jets.

Studies conducted by Grimitlyn[112] led to generalized Eqs. (7.144) and (7.145) for air diffusers with different velocity decay characteristics K_1:

$$x_1 = 0.22K_1\sqrt{BH}$$
$$x_2 = 0.31K_1\sqrt{BH}$$
$$x_{\max} = 0.62K_1\sqrt{BH}$$

for compact jets, and

$$x_1 = 0.1K_1^2H_r$$
$$x_2 = 0.15K_1^2H_r$$
$$x_{\max} = 0.3K_1^2\sqrt{BH}$$

for linear jets.

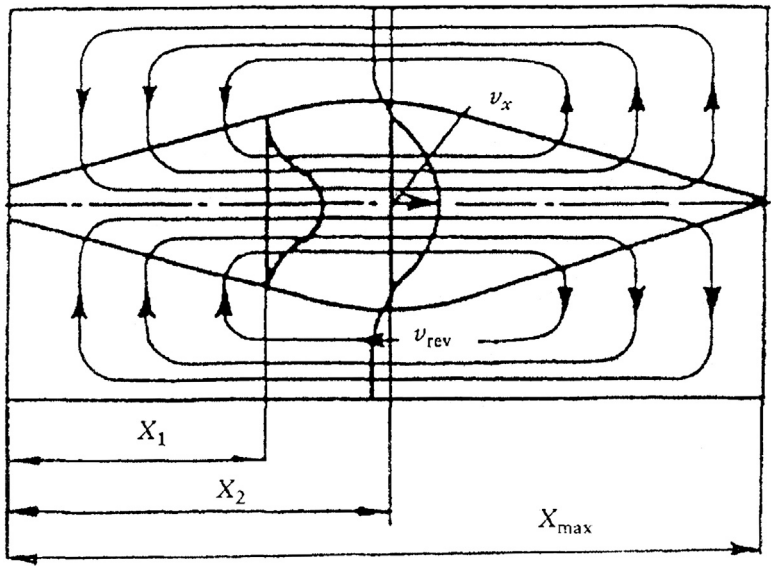


FIGURE 7.34 Schematic of air jet in confined space proposed by N.N. Sadovskaya. Source: *Reproduced from Grimitlyn M. Zurluftverteilung in raumen. Luft und Kältetechnik 1970;5:246–56.*

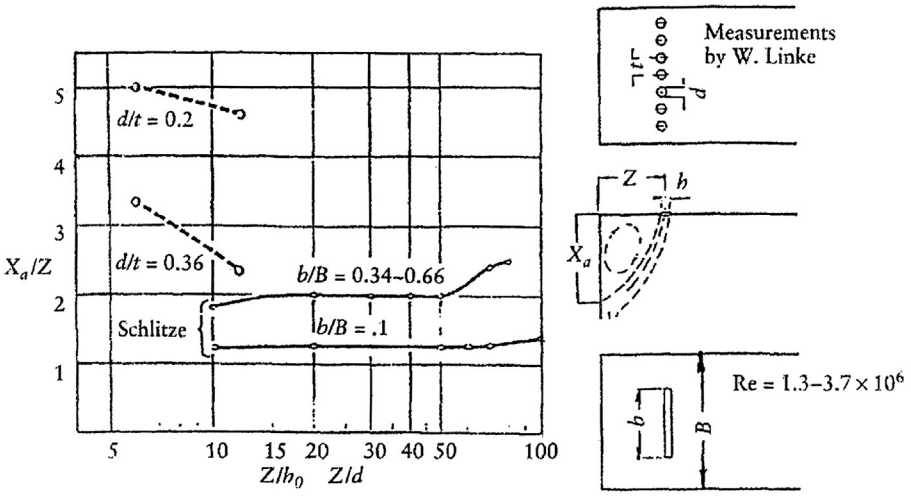


FIGURE 7.35 Reattachment length x_a versus the distance z from the ceiling surface to the supply outlet. Source: *Reproduced from Regenscheit B.* Das Wierderlengen eines ebenen Strahles an eine Wand. *Hausbericht H. Krantz Lufttechnik. E. Bericht 2376; 1962.*

To avoid high velocities in the occupied zone due to direct effect from the supply air jet, and to increase the length of the effectively ventilated zone for a single jet in rooms with height H_r from 4 to 10 m, Baharev and Troyanovsky[172] proposed supplying air from the height $h_0 = 0.6–0.7H_r$.

Effect of jet proximity to the ceiling

Studies by Sawyer,[176] Bourque and Newman,[157] and Regenscheit[177] showed (Fig. 7.35) that a two-dimensional jet supplied from a slot with a width h_0 at the distance z from the ceiling surface will become attached to this surface at the distance x_a given by

$$\frac{x_a}{h_0} = 0.2 + 2.7\left(\frac{z}{h_0}\right)^{0.8}. \qquad (7.146)$$

Research reported by Jackman[178] showed that the effect of proximity of air supply to the ceiling is also important when air is supplied with compact jets. The nonuniform entrainment from either side of the jet resulted in a force deflecting the jet toward the ceiling. The attraction of the jet toward the surface is greater the closer the air supply is to the ceiling and the higher its aspect ratio (grill width over grill height).

Effect of ceiling beams or obstructions in the jet zone

Ceiling beams will not affect jet attachment to the ceiling if they are located further than $1.6x_a$ from the air supply outlet.[179] If a beam is located closer than $1.6x_a$, the impingement of the jet on this beam will change the jet direction.

The effect of ceiling beams and light fittings on ventilation jets was also studied by Holmes and Sachariewicz.[180] Their studies were limited to the two-dimensional case: air supply through linear slot and a two-dimensional barrier (Fig. 7.36). The results of these

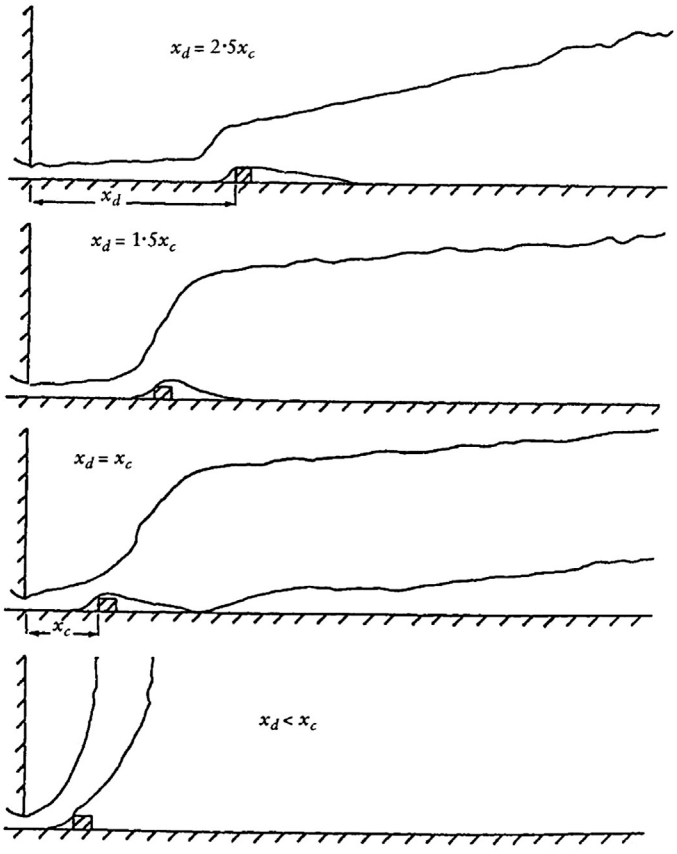


FIGURE 7.36 Beam influence on the airflow pattern along the ceiling. Source: *Reproduced from Holmes MJ, Sachariewicz E. The effect of ceiling beams and light fittings on ventilating jets. Heating and Ventilating Research Association Laboratory report no. 79. Bracknell, UK; 1973.*

studies show that the ceiling jet can take one of the three courses when it encounters an obstruction:

1. separate from the surface and take up a flow angle approximately equal to the angle between the upstream face of the obstruction and the surface;
2. separate from the surface and reattach some distance downstream from the barrier; or
3. almost ignore the existence of the barrier.

The jet will separate from the surface if the axial distance between the slot and the obstruction, x_d, is less than a specified critical distance x_c (Fig. 7.37). The values of x_c given in Fig. 7.37 are only for an obstruction of transverse dimension w equal to or less than the slot span s.

In the second course, the flow downstream of the barrier can be adequately represented by a determination of both the maximum separation of the line of maximum velocity from the surface (Fig. 7.38) and the velocity decay after the barrier

$$\frac{v_{\max}}{v_0} = 2.2\left(1-0.785\frac{w}{s}\right)^{1/2}\left(\frac{b_0}{x-x_d}\right)^{1/2}, \qquad (7.147)$$

where x_d is the distance from obstruction to the slot, $0.5 < w/s < 1$.

Obstruction does not affect the jet if the obstruction is further than about eight critical distances (x_c) from the slot. In this case the velocity decay of the jet may be obtained from

$$\frac{v_m}{v_0} = 2.2\left(\frac{b_0}{x-x_d}\right)^{1/2}, \qquad (7.148)$$

where b_0 is an effective slot width and x_0 is the virtual origin of the jet.

Although the tests were conducted only for two-dimensional cases, the authors suggest that their results can be extended to three-dimensional cases as follows:

- If the obstruction span is less than half the slot span, the effect of obstructions can be ignored provided $x_d > x_c$.
- The value of x_c for a short barrier will be less than for a long one and it will be safe to use the values obtained in the studies.
- The velocity decay downstream of a short barrier may be represented by

$$\frac{v_m}{v_0} = 2.2\left(1-0.785\frac{w}{s}\right)^{1/2}\left(\frac{b_0}{x-x_d}\right)^{1/2}, \qquad (7.149)$$

  where $0.5 < s/w < 1$.
- If the barrier is longer than the slot, the flow will be deflected at right angles to the normal jet trajectory, causing a possible thinning of the jet at the barrier and increasing the critical barrier distance. It is not possible to estimate accurately the extent of such an increase from the studies of two-dimensional situations. The authors consider that such an increase will not exceed 100%.

Graphical interpretation of the factors influencing the critical distance x_c from air supply to the linear obstacle with a height d_c for air supply through a slot diffuser with height h_0 and for air supply through a round nozzle with outlet diameter d_0[181] are presented in Fig. 7.39. Nonisothermal flow has an influence on the critical distance, and Archimedes number, Ar, is an important parameter together with the geometrical relations.[181] Ventilation with cooled air increases the effect of obstacles, and warm air supply decreases this effect.[182,183]

Air can be supplied in rooms by one or several jets. Air supply openings can be located along one wall—parallel air jet supply (Fig. 7.40A)—and/or on opposite walls—contrary-directed jets supply (Fig. 7.40B). In special cases air can be supplied in a fan-type manner (Fig. 7.40C).

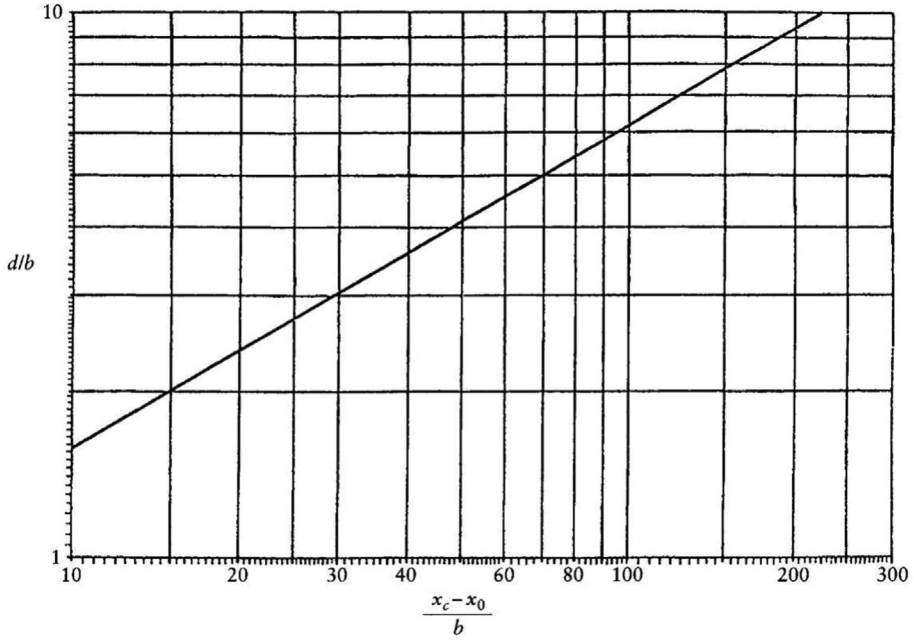
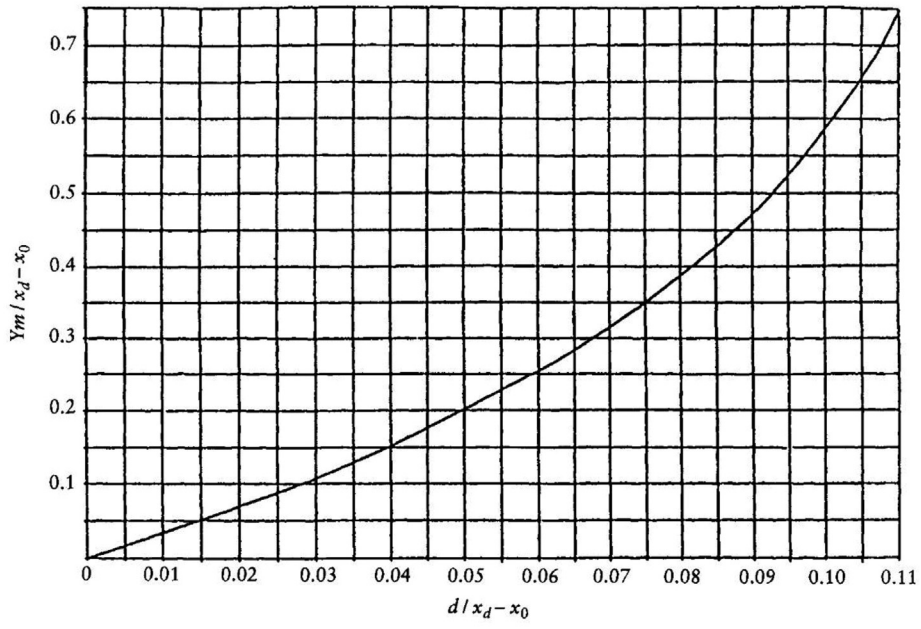


FIGURE 7.37 Critical distance x_c from the slot to the beam: b is the width of two-dimensional slot, d is the beam height, and x_0 is the jet core zone length. Source: *Reproduced from Holmes MJ, Sachariewicz E.* The effect of ceiling beams and light fittings on ventilating jets. *Heating and Ventilating Research Association Laboratory report no. 79. Bracknell, UK; 1973.*



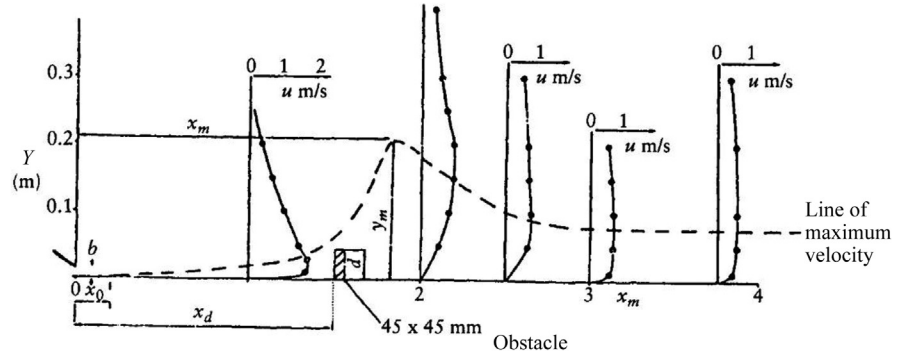


FIGURE 7.38 Maximum jet separation γ_m from ceiling: b is the width of two-dimensional slot, b is the beam height, and x_0 is the jet core zone length. Source: *Reproduced from Holmes MJ, Sachariewicz E.* The effect of ceiling beams and light fittings on ventilating jets. *Heating and Ventilating Research Association Laboratory report no. 79. Bracknell, UK; 1973.*

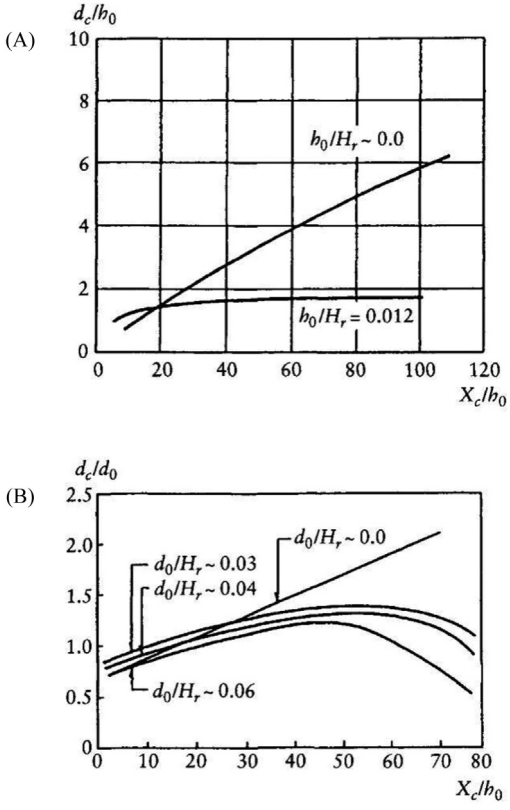


FIGURE 7.39 Critical height of an obstacle d_c versus distance from supply slot with a height h_0 (A) and supply nozzle with diameter d_0 (B). Source: *Reproduced from Nielsen PV. Lecture notes on mixing ventilation. Denmark: Department of Building Technology, University of Aalborg; 1995.*

Air circulation with a parallel jet supply is illustrated in Fig. 7.41. Jets are located at distance t from each other, and each jet forms return flow similar to that induced by a single jet in the room with a width $B = t$. Thus in the case of N parallel jets, the room should be considered divided into several zones with a width $B = B_r/N$, separated from each other by airtight walls.

7.4.5.3 *Analytical studies*

Abramovich[60] was the first to study axisymmetric confined jets analytically. He suggested the method based on utilizing the equations of continuity and momentum conservation. He also assumed that the width of the layer of a jet mixing with a counterflow equals the width of a free jet with a velocity distribution according to Schlichting's formula:

$$\frac{v - v_{\text{rev}}}{v_x - v_{\text{rev}}} = \left[1 - \left(\frac{y}{b}\right)^{3/2}\right]^2 \tag{7.150}$$

where $b = 0.22x$ is the half of the free jet width.

Analytical methods suggested by Shepelev and Tarnopolsky,[184] Grimitlyn and Pozin,[185] and Sychev and Volov[186] differ from the one described earlier only in the way the authors described velocity distribution in the mixing layer:

- Shepelev and Tarnopolsky:

$$\frac{v - v_{\text{rev}}}{v_x} = e^{-(1/2)(y/cx)^2} \tag{7.151}$$

- Grimitlyn and Pozin:

$$\frac{v - v_{\text{rev}}}{v_x} = e^{-0.7((k_1 y)(0.66x))^2} \tag{7.152}$$

- Sychev and Volov:

$$\frac{v - v_{\text{rev}}}{v_x - v_{\text{rev}}} = 1 - 6\left(\frac{y}{b}\right)^2 + 8\left(\frac{y}{b}\right)^3 - 3\left(\frac{y}{b}\right)^4 \tag{7.153}$$

It is assumed in the abovementioned methods that the influence of confined space on the supplied jet can be described by the reduction of the axial component and the value v_{rev}, as for jet development in the counterflow. The value of v_{rev}, is assumed to be the same throughout each cross-section but variable along the jet length. The value of v_{rev} can be found from the continuity equation, which in the case of jet distribution in a space of cylindrical shape can be presented as

$$2\int_0^{r*} vrdr + v_{\text{rev}}(R^2 - r_*^2) = 0 \tag{7.154}$$

for air supply and air exhaust located in the same wall and for air supply and air exhaust located in opposite walls.

According to Shepelev and Tarnopolsky[184] air velocity on the axis of the jet at the distance x from the outlet for air supply and exhaust located on the same wall can be calculated from

$$v_{xc} = v_x\left[1 - \frac{r_0}{R}\frac{1 - \exp[-(1/2)(R/cx)^2]}{(1/2)(R/cx)^2 K_1\sqrt{\pi}(r_0/R)}\right], \tag{7.155}$$

and the maximum (in the cross-section) velocity in the reverse flow can be calculated from

$$v_{\text{rev}} = v_0 \left(K_1\frac{\pi r_0^2}{x}\exp\left[-\frac{1}{2}\left(\frac{R}{cx}\right)^2\right]\left(-\frac{r_0}{R}\frac{1 - \exp[-(1/2)(R/cx)^2]}{(1/2)(R/cx)}\right)\right). \tag{7.156}$$

Velocity in the reverse flow reaches its maximum value at $x^* = 4.88R$ equal to

$$v_{\text{rev}}^{max} = 0.656\sqrt{\frac{M_0}{\rho\pi R^2}}, \tag{7.157}$$

The equations presented earlier can be applied to spaces of rectangular shape by replacing πR^2 by BH.

A similar approach was used by Zhivov[187,188] in his studies of systems of coaxial jets in confined space.

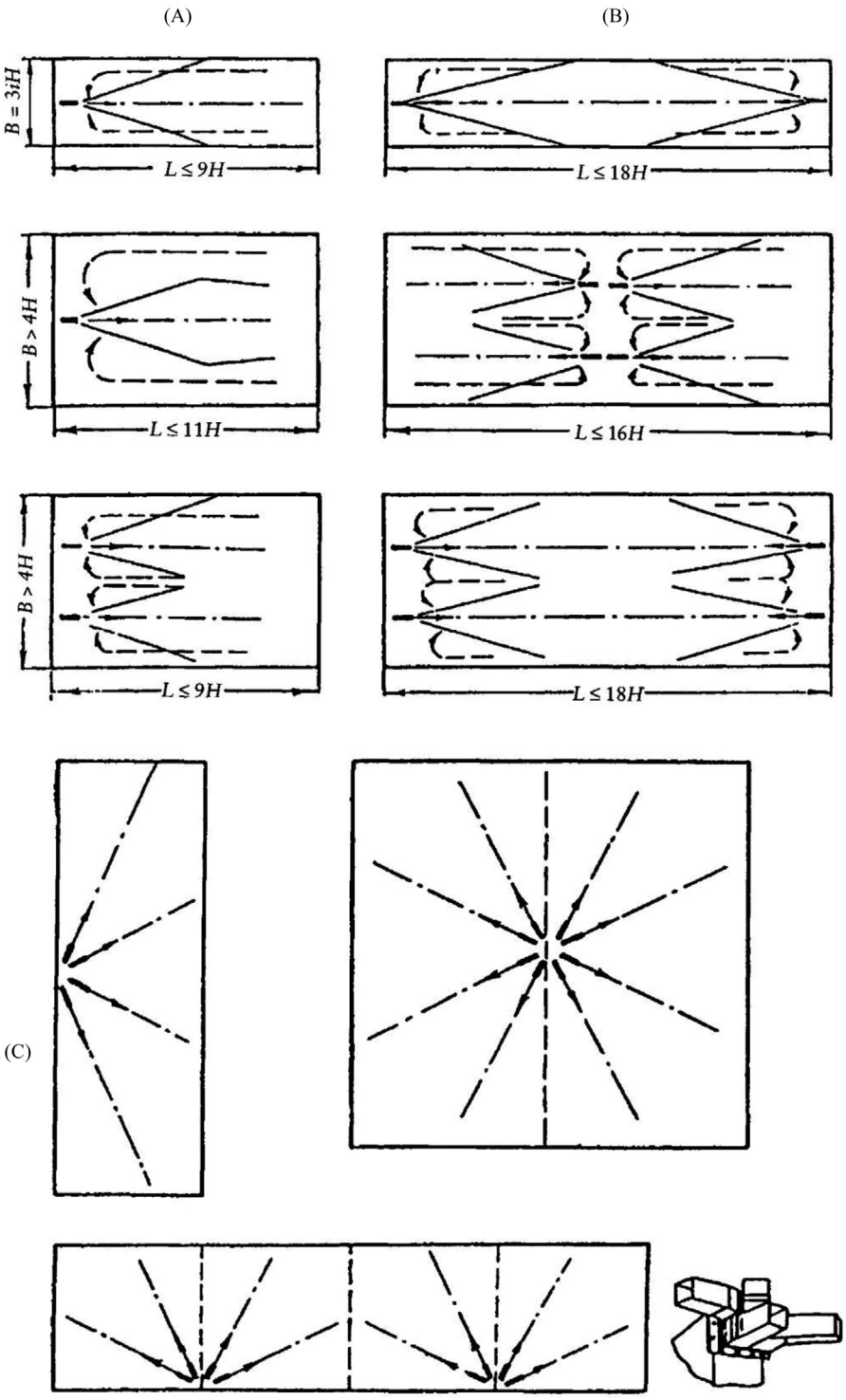


FIGURE 7.40 Schemes of room ventilation with parallel jet supplied from the same wall (A), from opposite walls (B), and in a fan-type manner (C). Source: *Reproduced from Baharev VA, Troyanovsky VN. The basis of heating and ventilating systems with the concentrated air supply design and calculation. Moscow, Russia: Profizdat; 1958.*

The distance x from the air diffuser to the cross-section with a maximum velocity in the reverse flow for the case without coaxial jets was found to be $1.9(BH)^{0.5}$ at $K_1 = 6.2$, and $1.4(BH)^{0.5}$ at $K_1 = 4.5$. For a nonisothermal jet, it was also found[188] that the reverse flow and confining surfaces increase the upper limit of the cold or heated supply air temperature Δt_0, which ensures a horizontal jet projection.

The results of different analytical and experimental studies of the confined horizontal jet described earlier are presented in Table 7.17. The main reason for the differences in the analytical results is different approximations of reverse flow velocity profiles.

The influence of the reverse flow on the centerline velocities v_{xc} was proposed[55] to be expressed by the coefficient K_c,

$$v_{xc} = v_x K_c,$$

which, as was shown earlier, can be derived analytically. The value of K_c depends on the ratio of the cross-sectional area of the free jet and the corresponding cross-section of the room. The graphs for evaluating K_c for compact, radial, and linear air jets are presented in Fig. 7.42.

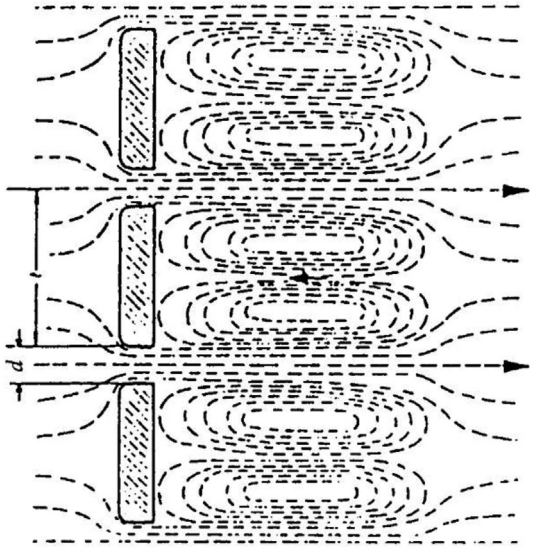


FIGURE 7.41 Room ventilation by parallel jets. Source: *Reproduced from Regenscheit B. Strahlgesetze und raumströmung. Klima-Kälte-technik 1975;6:147–50.*

7.4.5.4 Experimental studies of horizontal heated and cooled air supply in confined spaces

Gobza[191,192] studied air supply with concentrated jets on physical models and in the field and concluded that temperature stratification along the room height may occur if improper supply air temperature difference and air exchange rate are selected. Among the field cases reported by Gobza are industrial halls as long as 150 m (at width equal to 50 m and height 12–15 m).

The effect of supply air temperature on jet behavior in confined spaces was studied by Müllejans.[104] Studies of cooled air jets were conducted in rooms with a size from $1.0\ \text{m} \times 1.0\ \text{m} \times 1.6\ \text{m}$ to $2.27\ \text{m} \times 3.33\ \text{m} \times 5.31\ \text{m}$ with an air supply through the slot ($b = B_r$) or rectangular opening ($b \ll B_r$). Numerous smoke photographs were taken reflecting supply situations with different Re and Ar numbers. Archimedes number was defined by Müllejans as

$$\text{Ar} = \frac{gD_h(\theta_w - \theta_0)}{v^2(T_w + T_0)/2}, \tag{7.158}$$

where $v = Q_0/(B_r H_r)$ (m/s), $\theta_w(T_w)$ and $\theta_0(T_0)$ are the wall and supply air temperatures (°C, K), g is the acceleration due to gravity, and D_h is the hydraulic diameter,

$$D_h = \frac{4B_r H_r}{2(B_r + H_r)}. \tag{7.159}$$

Müllejans has reported that with air supply through rectangular openings the jet behaves more or less as an isothermal flow when $\text{Ar} < 10^4$.

TABLE 7.17 Results of experimental and analytical studies of the compact confined jet.

References	Throw	Throw definition	Zone I + Zone II	Maximum velocity in the reverse flow
Baharev and Troyanovsky[172]	$(4.7 - 5.4)\sqrt{BH}$	$v_{\text{rev}} = 0.05 - 0.01 v_0 \sqrt{A_0/BH}$	$2.0\sqrt{BH}$	$0.78 v_0 \sqrt{(A_0/BH)}$
Abramovich[60]	$3.5\sqrt{BH}$	$v_x = 0.05–0.01\ v_0$	$2.4\sqrt{BH}$	$0.88 v_0 \sqrt{A_0/BH}$
Shepelev and Tarnopolsky[184]	$5\sqrt{BH}$	$v_{\text{rew}} = 0.07 v_0 \sqrt{A_0/BH}$	$2.75\sqrt{BH}$	$0.66 v_0 \sqrt{(A_0/BH)}$
Grimitlyn and Pozin[185]	$0.7K_1\sqrt{BH}$	–	$0.31K_1\sqrt{BH}$	$0.78 v_0 \sqrt{(A_0/BH)}$
Gnatyuk et al.[189]	$4.3\sqrt{BH} - 3.3\sqrt{A_0/BH}$	$v_x = (0.07–0.01)\ v_0$	–	–
Sychev and Volov[186]	$5.5\sqrt{BH}$	$v_x = 0$	$3.2\sqrt{BH}$	$0.7 v_0 \sqrt{(A_0/BH)}$
Schwenke[161]	$5.0\sqrt{BH}$ – compact jet	$v_{x\ \text{ave}} = 0.2$ m/s or	–	
	$3H$ – linear nonattached jet	$v_{x\ \max} = 0.5$ m/s		
Nielsen[147,153]	$5.0\sqrt{BH}$ – compact jet	–	$3\sqrt{BH}$	$0.95 v_0 \sqrt{(A_0/BH)}$
	$4H$ – linear attached jet			Round attached jet
Zhivov[187,188]	$3.9\sqrt{BH}, K_1 = 6.2$	Cross-sectional velocities are uniform with an unstable direction	$1.9\sqrt{BH}, K_1 = 6.2$	$0.73 v_0 \sqrt{(A_0/BH)}$
	$3.4\sqrt{BH}, K_1 = 4.5$		$1.4\sqrt{BH}, K_1 = 4.5$	

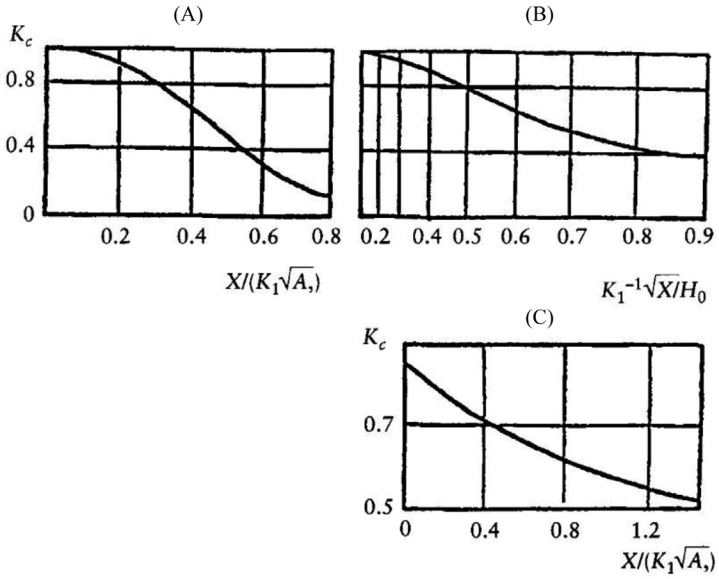


FIGURE 7.42 Coefficient of confinement: (A) compact jet, $A_r = B \times H$; (B) linear jet; and (C) radial jet, $A_r = B \times L$. Source: *Reproduced from Grimitlyn MI, Pozin GM. Fundamentals of optimizing air distribution in ventilated spaces. ASHRAE Trans 1993;99:1.*[190]

To establish a criterion for any size room and outlet, the Ar number was adjusted using a geometrical factor. The modified Ar* was defined as

$$\mathrm{Ar}* = \mathrm{Ar}\frac{b_0 h_0}{D_h^2}. \tag{7.160}$$

With a common value of $b_0h_0/D_h{}^2 = 1/250$, the airflow pattern will be similar to isothermal with a modified Ar* number limited to 40.

In the case of a room ventilated by a linear jet, this jet deflects toward the ceiling immediately after entering the room. Maximum Ar values depend on the L/H ratio and are shown in Table 7.18.

Experimental studies conducted by Grimitlyn[112] on heated and chilled confined jets showed that the airflow pattern remains the same as for isothermal air supply when $\mathrm{Ar}_x < 0.2$ at $x = 0.22K_1(BH)^{0.5}$, in rooms with H/B ratio from 0.3 to 1.0, where

$$\mathrm{Ar}_x = \frac{K_2}{K_1^2}\frac{g\sqrt{A_0}\Delta\theta_0}{v_0^2 T_{oz}}\left(\frac{x}{\sqrt{A_0}}\right)^2. \tag{7.161}$$

The abovementioned limitation on the local Archimedes number results in the following equation for maximum temperature difference of supplied air:

$$\Delta\theta_0 = 122\frac{v_0^2\sqrt{A_0}}{K_2 BH}. \tag{7.162}$$

Similar studies were conducted by Troyanovsky,[193] who concluded that to maintain the airflow pattern in rooms with heated or cooled air supply as in isothermal conditions, it is necessary that the rise of horizontally supplied jet does not exceed $\Delta y = 0.1BH$ at the distance from the outlet $x = 0.15K_1(BH)^{0.5}$. From this assumption the following equation for the maximum air temperature difference was derived:

$$\Delta\theta_0 = 1300\frac{v_0^2\sqrt{A_0}}{K_1 K_2 BH}. \tag{7.163}$$

TABLE 7.18 Ar values.

L/H	4.7	3.0	2.0	1.0
Ar_{max}	2000	3000	10,000	11,000

Comparing Eq. (7.162) with Eq. (7.163), one can see that the value of the maximum temperature difference computed using Eq. (7.163) is higher than that determined using Eq. (7.162). Results of experimental studies on physical models[55] indicate that when $H/B > 1$, the limitation on supply air temperature difference should be even more restrictive.

7.4.5.5 Computational fluid dynamics simulation of heated and cooled air supply in confined spaces

CFD simulation has become more and more popular in predicting the airflows produced by heated and cooled air supply in confined spaces. A review[194] shows that in 2007, more than 60% of the papers have used CFD to predict ventilation performance in buildings. The used methods include Reynolds-averaged Navier–Stokes (RANS) equation modeling, large-eddy simulation (LES), detached-eddy simulation (DES), etc. Performance of the models was not always consistent for different flows.[195]. Details of CFD modeling are discussed later in this book.

Besides the models, accuracy of the air terminal devices described by CFD also influenced the simulation results. The air jets in ventilated confined spaces

are usually supplied by air terminal devices, such as diffusers, nozzles, and grilles. Compared with nozzles and grilles, CFD simulations for diffusers on displacement ventilation systems are more complicated since the diffusers cannot be treated as a simple opening for their small effective area ratios. Zhang et al.[196] proposed a simplified approach by assigning the actual velocities on a certain ratio of CFD cells to describe the complex diffusers. Results show that the method is capable of describing the quarter-circular-perforated, grille, floor-perforated, and swirl diffusers under good agreement in computed profiles with the measured data. In addition, the proposed method is better for diffusers with large supply areas such as the quarter-circular-perforated and the floor-perforated diffusers since their discharge flows are more uniform and stable, which are easier to model.

When a room is ventilated by air terminal devices such as nozzles, the jet will deflect toward the ceiling immediately after entering the room. Uncomfortable feeling will be produced by the inside human body if improper air supply is applied. However, it is challenging to determine the exact air supply that can create the target performance. Current CFD simulation has to enumerate every possible air supply to find the optimal solution, which is not straightforward and would require a lengthy iterative procedure. Inverse methods are able to provide a faster approach for fulfilling the task by setting the design targets first and then inversely solving for the required causal boundary parameters. Recently several inverse methods have been applied to promptly search for the required air supply parameters based on the expected target, such as CFD-based genetic algorithm,[197–200] CFD-based proper orthogonal decomposition (POD),[201,202] and CFD-based adjoint method.[202–205] Also a method which integrates the abovementioned three methods was proposed to further expedite the inverse design process.[206] For these methods, the target environmental performance is specified first, and then the corresponding air-supply parameters can be inversely solved with the use of a particular method.

7.4.5.6 The effect of confinement on inclined air jets

The investigations of horizontal and inclined air jet trajectory, velocity, and temperature decay under buoyancy discussed in the previous section were conducted with free (nonconfined) jets. Only limited research data is available describing the behavior of inclined jets in confined spaces. Studies by Regenscheit[127] of horizontal cooled air supply from linear and rectangular openings can be related to this topic. Graphs in Fig. 7.43 show how the relative distance x_0/L from the supply opening to the point of jet impingement with the floor surface is influenced by the modified Archimedes number Ar:

$$\mathrm{Ar} = \frac{g\Delta T_0 4(BH)^3}{T_{0z}Q_0^2 2(B+H)}. \tag{7.164}$$

Other experimental and analytical studies of nonisothermal inclined jets in confined spaces were carried out by Zhivov.[145] Experimental studies were conducted on the physical models. The ratio of the model dimensions $L \times B \times H$ was changed so that the value H/B was from 0.3 to 3.0 and $L/(B \times H) = 2.4 - 4.9$.

Visualization of airflow in the room with smoke and silk threads was used to describe airflow patterns in rooms with inclined jet supply. Airflow created by inclined jets impinging on the floor surface can be divided conditionally into three zones (Fig. 7.44): (1) free or confined jet, (2) impingement zone, and (3) flow along the floor. The width of the jet depends on the supply characteristics, which can be primarily described by the velocity decay characteristic K_1. Some air diffusers (e.g., ventilation grills) can create jets with coerced angle of deflection only in one direction.

The first zone of the jet can be described using equations for velocity and temperature decay as well as jet trajectory with a coefficient K_c accounting for jet confinement. The impingement zone can be characterized by a significant change in the static pressure and great curvature of the air current lines. After the impingement, the radial flow is formed as if it is supplied from the side surface of the truncated cylinder with a uniform initial velocity of U_m^*. In the basement of the cylinder, there is a particular line that crosses the quasisource of the radial flow. Equations provided in the paper can be used to evaluate velocities along the branches with maximum airflow, minimum airflow, and along the particular line.

When the width of the jet (calculated for free conditions) is less than the width of the room, airflow after jet impingement on a floor is similar to that in nonconfined conditions. When the horizontally directed flow (along the particular line) reaches the wall, it is divided into two branches: one following the direction of the branch with a maximum airflow and another flowing in the opposite direction.

When the air directed backward reaches the back wall of the room, it flows upward to be induced by the jet within its first zone (Fig. 7.45).

The circulation zone is created above the branch with maximum airflow spreading along the floor. The reverse flow is also induced by the inclined jet within its first zone.

If the width of the jet (calculated for free conditions) at the point of its intercept with the occupied zone

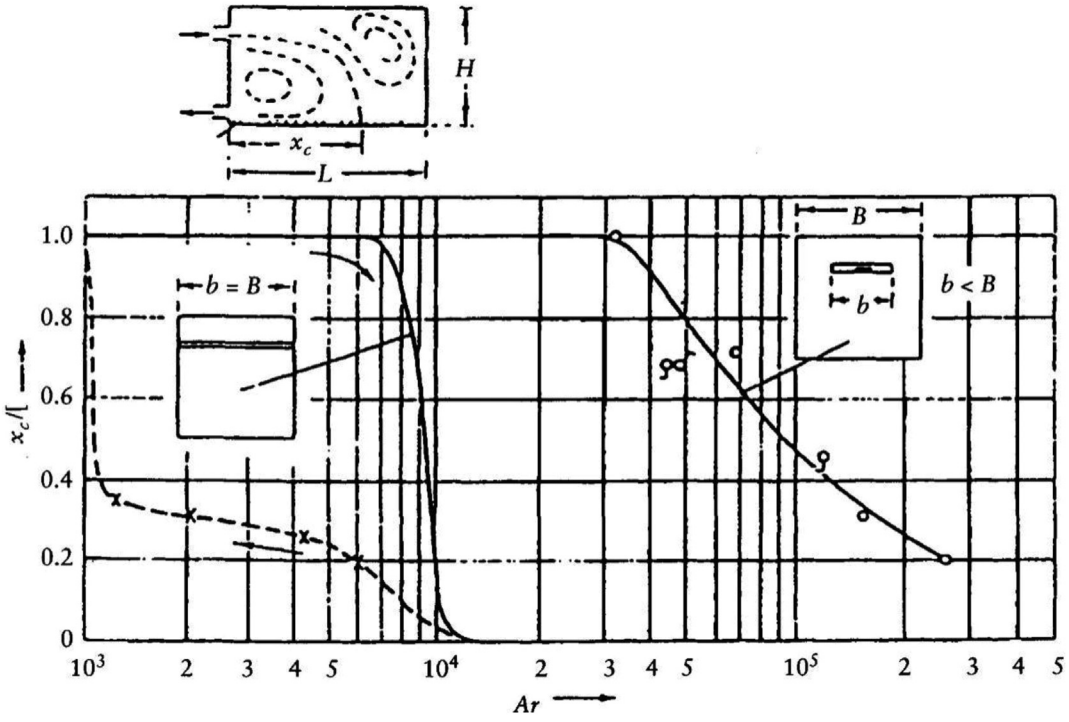


FIGURE 7.43 Nonisothermal jet trajectory in a room. Source: *Reproduced from Regenscheit B. Die Archmedes-zhal-kenzah zur beurteilung von raumstromungen. Gesundh Ing 1970;**91**:6.*

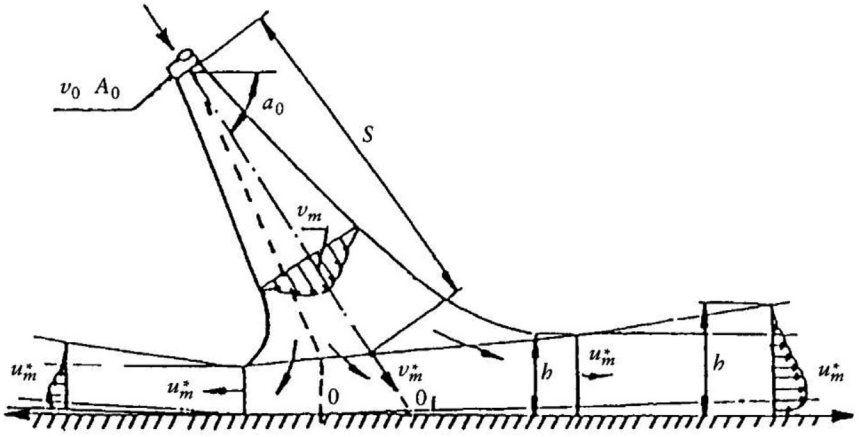


FIGURE 7.44 Schematic of inclined jet impingement with a floor surface. Source: *Reproduced from Zhivov AM. Theory and practice of air distribution with inclined jets.* ASHRAE Trans *1993;**99**:1.*

exceeds the room width, side walls transform the jet as if it was formed by the linear jet impingement on a floor.

In design of air distribution with inclined cooled air jets, the following parameters should be considered: air velocity and temperature at the point of jet intercept with the occupied zone—for practical purposes this cross-section can be considered the border between the first and the second zone of the impinging jet; and velocities along jet branches—maximum airflow branch, minimum airflow branch, and the branch along the particular line.

The latter information is important in evaluating the size of the occupied zone that can be effectively ventilated by inclined jets. It was proposed[145] that the occupied zone of rooms is well ventilated by inclined jets (particularly in industrial rooms with contaminant release) if air velocity in the occupied zone exceeds 0.1 m/s.

The influence of confinement on air velocity u_m in the flow along the floor can be accounted for with the coefficient K_c:

$$u_m^c = K_c u_m. \tag{7.165}$$

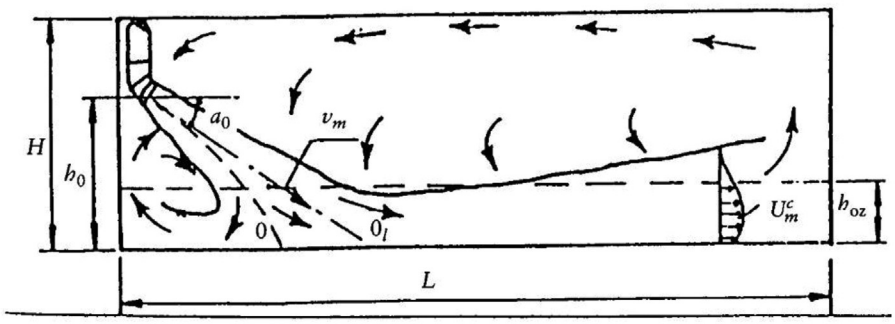


FIGURE 7.45 Airflow in a room with an inclined jet supply. Source: *Reproduced from Zhivov AM. Theory and practice of air distribution with inclined jets.* ASHRAE Trans *1993;**99**:1.*

The value of this coefficient depends on the relative height of the flow along the floor h_f/H_r (calculated in the free conditions), where

$$K_c = 1 \text{ when } h_f \le 1.1, K_c = 1.79 - 0.72h_f/H_r \text{ when } h_f > 1.1. \quad (7.166)$$

7.4.5.7 Air supply with vertical jets

Air supplied in confined space by downward vertical jets creates a similar flow pattern as in the case of air supply by horizontal nonattached jets. With vertical air supply, the occupied zone is ventilated directly by air jets. Grimitlyn[190] suggests that the area of occupied zone ventilated by one jet be sized based on the jet's cross-sectional area at the point it enters the occupied zone. The jet cross-sectional area and configuration depend on the height of the air supply, the type of air jet, and diffuser characteristics (K_1 and K_2).

7.4.5.8 Airflow patterns and airflow in occupied zone

Different air distribution systems may produce different room air stratification. The flow patterns are primarily determined by the characteristics of the room supply airflow and heat load configuration. For a given heat load and room air supply rate, air velocity in the occupied zone increases when outlet discharge velocity increases[201,207] and extremely large air velocity and turbulence may cause thermal discomfort of the inside occupants. So air distribution systems should be designed properly to prevent occupants from subjecting to the region where large room air velocity occurs. In addition, Buoyant forces, for example, when supply air is heated, can significantly affect the airflow pattern created by supply jets. Applying proper design principles prevents warm air from rising to the upper zone of the room without heating the occupied zone.

7.4.6 Jet interaction

Diffuser jet interaction is a common case when air is supplied into ventilated rooms through multiple air diffusers (e.g., sidewall-mounted grills or ceiling-mounted air diffusers) or using speciality air distribution systems. Interacting jets can be supplied

- parallel to each other,
- in the opposite direction toward each other (e.g., from the outlets located on the opposite walls),
- coaxially, and
- at an angle to each other.

Studies of some common cases of jet interaction are discussed in this section.

7.4.6.1 Interaction of parallel jets

Interaction of parallel air jets is the most common case. It was thoroughly studied by Baturin,[208] Koestel and Austin,[209] Grimitlyn,[112,210] Bashus and Kocheva,[211] Posokhin,[212] Nosovitsky,[213] Kuzmina,[214] Vasilyeva,[215] and Shepelev.[99] Researchers developed equations describing velocities and temperatures in two or more interacting jets assuming that momentum and heat content of the flow through the elementary area in the cross-section of the resulting jet is equal to the sum of momentums and heat contents through the same area of the separate interacting jets. It was also assumed that separate jets do not influence each other. Derivation of velocities based on these assumptions described by Shepelev[99] is presented later.

Air velocity v_Σ at any point (X, Y, and Z) of the flow created by interaction of two parallel jets supplied from outlets located at a distance 2 a from each other (Fig. 7.46) can be described by

$$v_\Sigma^2 = v_1^2 + v_2^2, \quad (7.167)$$

where

$$v_1 = \frac{K_1 v_0 \sqrt{A_0}}{X} e^{-(1/2)((((Y-a)^2+Z^2)/cX^2))} \quad (7.168)$$

$$v_2 = \frac{K_1 v_0 \sqrt{A_0}}{X} e^{-(1/2)((((Y+a)^2+Z^2)/cX^2))} \quad (7.169)$$

Air velocity in the resulting flow in the plane of the interacting jets' axis ($Z = 0$) derived from

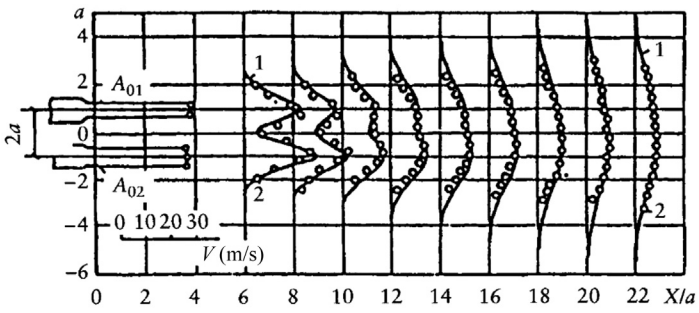


FIGURE 7.46 Interaction of two parallel compact jets.

Eqs. (7.167)–(7.169) is

$$v_\Sigma = \frac{K_1 v_0 \sqrt{A_0}}{X}\left[e^{-\left((Y-a)/cX\right)^2} + e^{-\left((Y+a)/cX\right)^2(1/2)}\right]. \tag{7.170}$$

Air velocity on the axis of one of interacting jets ($Y = a$ or $Y = -a$) is

$$v_{x1} = \frac{K_1 v_0 \sqrt{A_0}}{X}\left[1 + e^{-(2a/cX)^2}\right]^{\frac{1}{2}} \tag{7.171}$$

If $\lambda = (v_{x1} - v_x)/v_x$ is the relative increase of axial air velocity in one of the interacting jets due to the influence of the other, then the length of the jet X^* within which the influence of the interacting jet would be less than, for example, 10% ($\lambda = 0.1$) can be obtained from

$$X^* = \frac{2a}{c}\frac{1}{\sqrt{-ln\lambda(2-\lambda)}} \approx \frac{24.4a}{\sqrt{-\lambda(2-\lambda)}} = 26.4a, \tag{7.172}$$

which means that the axial velocity of the interacting jet is influenced by another jet only beginning with X^* exceeding 13.2 times the distance between the outlets.

Velocity along the axis of symmetry between the interacting jets can be calculated assuming $Y = 0$ in Eq. (7.170):

$$v_{\Sigma x} = \frac{K_1 v_0 \sqrt{2A_0}}{X} e^{-(a/cX)^2}. \tag{7.173}$$

Air velocity along the jet supplied from the outlet with opening area equal to $2A_0$ is

$$v_x = \frac{K_1 v_0 \sqrt{2A_0}}{X}. \tag{7.174}$$

If $\lambda = (v_x - v_{x1})/v_x$ is the relative difference between the axial air velocity in the jet with double opening area and the air velocity along the axis of symmetry of the interacting jets, then the length of the jet X^{**} within which this ratio would be less than, for example, 10% ($\lambda = 0.1$) can be obtained from

$$X^{**} = \frac{a}{\sqrt{2c}}\frac{1}{\sqrt{-\ln(1-\lambda)}} \approx \frac{8.62}{\sqrt{-\ln(1-\lambda)}} = 26.5a. \tag{7.175}$$

This approach can be used for interacting jets supplied from outlets with different area of discharge with different initial air velocities. In this case the equation for air velocity in the flow of interacting jets will be

$$v_\Sigma = \frac{K_1}{X}\left[v_{01}^2 A_{01} e^{-((Y-a)/cX)^2} + v_{02} e^{-((Y+a)/cX)^2}\right]. \tag{7.176}$$

Comparison of the calculations using Eq. (7.170) with experimental data collected by Vasilyeva[215] at $v_{01} = 38.1$ m/s, $v_{02} = 36.6$ m/s, $D_{01} = 0.03$ m, $D_{02} = 0.04$ m, and $a = 0.05$ m is presented in Fig. 7.46 from Shepelev.[99]

Jet interaction should not be taken into account when the jets are closely adjacent to each other, are propagated in confined conditions, and entrainment of the ambient air is restricted. This may be the case for concentrated air supply when air diffusers are uniformly positioned across the wall and the jets are replenished by the reverse flow, which decreases the jet velocity. This effect should be taken into consideration using the confinement coefficient K_c discussed in Section 7.4.5. For the same reason, jet interaction should not be taken into consideration when air is supplied through the ceiling-mounted air diffusers and they are uniformly distributed across the ceiling.[112]

For the most common practical situation, when air is supplied by parallel jets from several diffusers placed in one plane and having the same outlet area A_0 and discharge velocity v_0, the resulting velocity on the axis of the coalesced flow v_Σ can be found[112]:

- For compact and incomplete radial jets from

$$\frac{v_\Sigma}{v_0} = K_1 \frac{\sqrt{A_0}}{X} K_{\text{int}} \tag{7.177}$$

- For linear jets from

$$\frac{v_\Sigma}{v_0} = K_1 \sqrt{\frac{H_0}{X}} K_{\text{int}}. \tag{7.178}$$

The abovementioned relations can also be used as a first approximation to find the temperature drop in interacting jets by substituting $\Delta\theta_\Sigma$, $\Delta\theta_0$, and ΔK_2 for v_Σ, v_0, and K_2, respectively. The values of interaction coefficient K_{int} for even and odd numbers of outlets are given in Fig. 7.47, reproduced from Grimitlyn.[112]

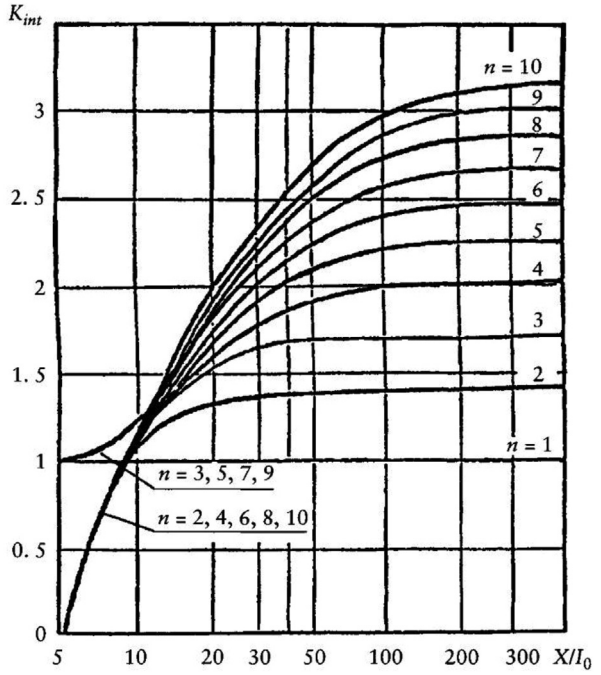


FIGURE 7.47 Coefficient K_{int} of interaction for the jets discharging from the opening located in a single row. Source: *Reproduced from Grimitlyn MI. Air distribution in rooms. Moscow, Russia: Stroiizdat; 1982.*

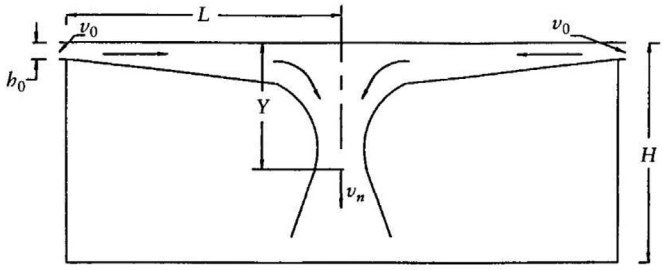


FIGURE 7.48 Interaction of two jets supplied from the opposite walls.

7.4.6.2 Interaction of jets supplied from opposite directions

For practical applications related to space ventilation and air-conditioning, interaction of similar jets with equal size and initial momentum supplied from the opposite walls (Fig. 7.48) were studied by Roeder,[216] Conrad,[217] Urbach,[218] Regenscheit,[219] and Smirnova.[55]

Conrad[217] compared impingement of two opposite linear jets attached to the ceiling with a linear jet changing its direction after impingement with a wall. For the attached jet, maximum air velocity along the jet can be described by

$$\frac{v_m}{v_0} = \left(\frac{h_0}{mX}\right)^{0.375}, \tag{7.179}$$

where m is a supply outlet characteristic that can range from 0.1 to 0.4,[114] and X is the distance from the slot to the point of interest. After changing the jet direction, the velocity in the vertical jet can be obtained from

$$\frac{v_m}{v_0} = K\left(\frac{h_0}{mL}\right)^{0.375}\left(\frac{L}{L-Y}\right)^{q}, \tag{7.180}$$

where Y = vertical distance from the ceiling to the point of interest, L = length of jet travel along the ceiling; $K = 1$, $q = 0.2$ when the jet travels vertically along the wall; $K = 0.65$, $q = 1$ in the case of two-jet interaction. In discussion of the data obtained by Conrad and Roeder, Regenscheit[219] suggested that the values of K and q also depend on the relative distances L/h_0 and Y/h_0 and the characteristic m. Based on the data by Urbach,[218] Regenscheit concluded that K and q parameters also depend on ratios L/H and h_0/H. Research data also show that air velocities in the combined vertical jet are lower than in the jet after its interaction with a wall.

The abovementioned data as well as studies of compact and radial jet interaction conducted by Smirnova, Avdeeva, and Gunes were summarized by Grimitlyn.[55] Grimitlyn suggested that the air velocity in the jet resulting from impingement of two similar opposite jets is

$$v_\Sigma = K_{\text{int}}^{\text{op}} v_x \tag{7.181}$$

where K_{int}^{op} can be evaluated from the graph in Fig. 7.49. The graph shows that smaller relative distance between jet supply outlets and the point of jet interaction, a/b_0 (for linear jets) or $a/\sqrt{A_0}$ (for radial and compact jets), results in smaller air velocities in the combined jet.

7.4.6.3 Interaction of coaxial jets

During the past two decades, a new generation of HVAC systems with concentrated air supply assisted by directing jets was introduced in several European countries.[220,221] In one common modification, the main streams of ventilating air (heated or chilled) are supplied through a small number of air openings (grills) at low initial velocities and distributed within the space by horizontal (coaxial with main streams) and vertical (supplied perpendicular to the main streams), or only horizontal, directing jets (Fig. 7.50). These jets are discharged at high velocities from nozzles having small outlet diameters. The air is delivered to these nozzles from a separate air handling unit. The same principle is utilized by the "air piston system,"[221] in which horizontal directing jets are created by axial or radial fans located along the main streams. Analytical

and experimental studies on the interaction of main streams of supplied air and horizontal directing jets in laboratories and in the field conducted by Zhivov[82,188,222] laid the ground for the design method of such systems.

Interaction of the free isothermal main stream and horizontal directing jets

The characteristic feature of the main stream and horizontal directing jet interaction is that the directing jets are supplied through nozzles located at some distance from each other and from the outlet supplying the main stream (Fig. 7.51).

Experimental studies with propane, as a tracer gas, introduced into the main stream showed that directing jets make the main stream narrower. Velocity profiles $v_{i\Sigma}$ in cross-sections of the resulting stream (created by the main stream and directing jets) can be described by the formula derived, assuming the resulting stream momentum is equal to the sum of interacting jet initial momentums. For maximum velocity in the resulting airflow in the cross-section of the $(N-1)$ nozzle, this equation is

$$v_{m\Sigma} = v_{02}\frac{d_{02}}{l_i}\sqrt{\frac{I_{01}}{I_{02}}\left(\frac{K_{11}}{K_{12}}\right)^2 + \sum_{k=1}^{N-1}\frac{1}{k^2}}, \qquad (7.182)$$

where K_{11} and K_{12} are coefficients of velocity decay in the main stream and directing jets, respectively, v_{02} is the horizontal directing jet supply air velocity, I_{01} and I_{02} are main stream and horizontal directing jet momentum, respectively, and d_{02} is the horizontal directing jet nozzle diameter.

To derive the equation for the jet boundary resulting from the interaction of coaxial main flow and a directing jet supplied at the distance l_0 from the main outlet, this interaction was presented[222] as the interaction of the main jet with a sink distributed along its axis (Fig. 7.51). Considering the influence of the directing jet on the main flow boundary as ΔY, the half width of the resulting flow can be presented as

$$Y_b = \Lambda X = \Delta Y, \qquad (7.183)$$

where

$$\Delta Y = \int_x^0 \frac{d(\Delta Y)}{dX}dX. \qquad (7.184)$$

Λ is the coefficient characterizing the angle γ of the main flow divergence (Fig. 7.51) without the directing jets' influence. The following relationship was derived for the resulting flow boundary:

$$Y_b = \Lambda X - \frac{\theta}{K_{12}^2}\frac{I_{02}}{I_{01}}l_0\Phi(\overline{X}) \qquad (7.185)$$

$$\Phi(\overline{X}) = \int_0^x \left(1 + \frac{1}{\sqrt{1+K_{11}^2(X/(X-1))^2}}\right)^2 d\overline{X}, \qquad (7.186)$$

where θ is an experimental coefficient and $\overline{X} = X/l_0$.

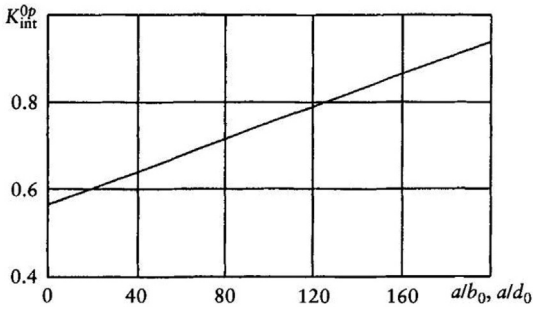


FIGURE 7.49 Coefficient K_{int} of opposite jets interaction. Source: *Reproduced from Grimitlyn MI.* Air distribution in rooms. *St. Petersburg; 1994.*

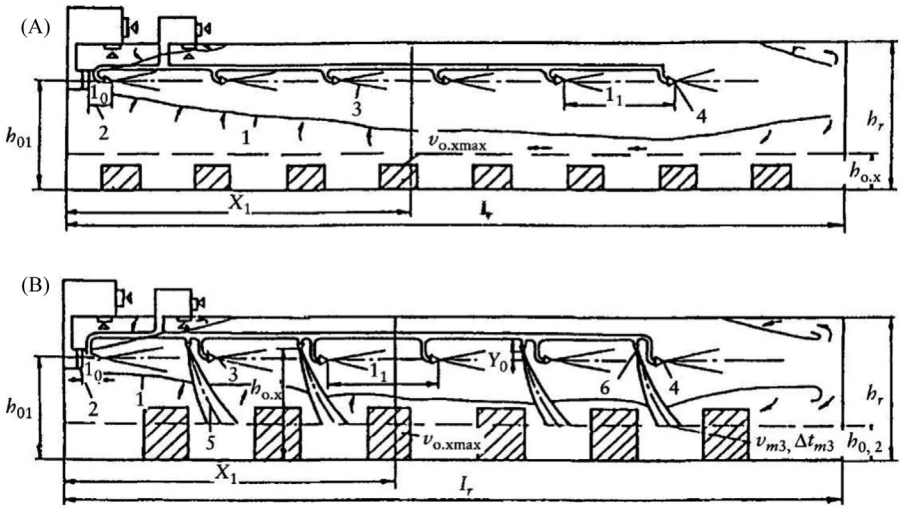


FIGURE 7.50 Concentrated air supply with directing jets: (A) with horizontal directing jets, (B) with vertical directing jets, 1—main stream, 2—main stream air diffuser, 3—horizontal directing jet, 4—horizontal directing jet nozzle, 5—vertical directing jet, and 6—vertical directing jet nozzle. Source: *Reproduced from Zhivov AM.* Concentrated air distribution with directing jets *[Ph.D. thesis]. All-Union Research Institute for Labor Protection, St. Petersburg, Russia; 1983.*

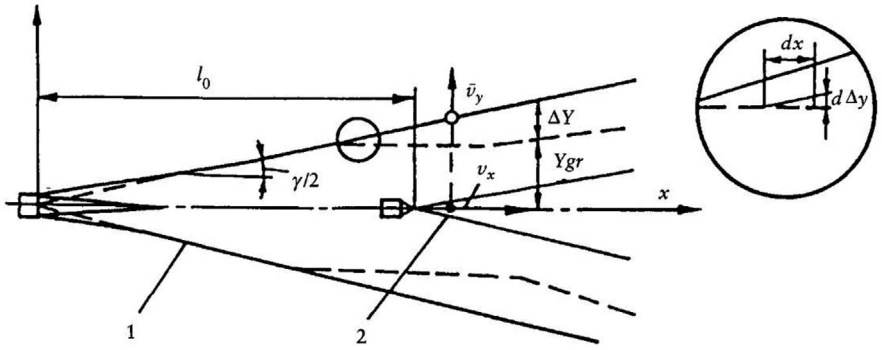


FIGURE 7.51 Schematic of free isothermal main stream and horizontal directing jet interaction: I—main stream (D_{01}, V_{01}, I_{01}, d_{01}, and K_{11}) and 2—directing jet (D_{02}, V_{02}, I_{02}, d_{02}, and K_{12}). Source: *Reproduced from Zhivov AM.* Concentrated air distribution with directing jets *[Ph.D. thesis]. All-Union Research Institute for Labor Protection, St. Petersburg, Russia; 1983.*

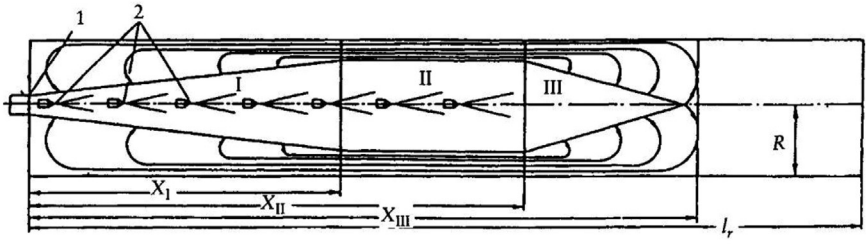


FIGURE 7.52 Schematic of confined main and horizontal directing jets interaction: 1—main flow and 2—directing jet. Source: *Reproduced from Zhivov AM. Air supply with directing jets. In:* Ventilation '94: proceedings of the fourth international symposium on ventilation for contaminant control. *Stockholm, Sweden; 1994.*

In the case of several directing jets interacting with a main stream, the abovementioned approach was used assuming that each following directing jet interacts with the resulting flow created by the main flow and the previous directing jets. The equation for the resulting flow boundary differs from Eq. (7.185) only by the expression for the $\Phi(X)$ function.

Interaction of the confined isothermal main stream with horizontal directing jets

The experimental studies[222] show that the resulting jet length in the confined space can be divided into three zones (Fig. 7.52). In the first zone, there is an expansion of the resulting jet boundaries. The length of this zone (X_I) depends on the relative momentum (I_{02}/I_{01}) value and the relative distance ($l_i/\sqrt{A_r}$) between the directing nozzles, where A_r is the room vertical cross-sectional area ($b_r \times h_r$). The distance (X_I) increases when the relative momentum (I_{02}/I_{01}) increases and the relative distance ($l_i/\sqrt{A_r}$) decreases. In the second zone, the resulting jet width stays relatively constant. It expands up to the last nozzle, and its length is equal to ($X_{II} - X_I$) and depends on the number of directing nozzles and the distance between them.

In the third zone, there is a significant decrease in resulting jet width. The cross-section in which the jet flow degrades is considered the end of the third zone. The length of the third zone ($X_{III} - X_{II}$) is practically equal to the length of the jet's degradation zone in the confined space without directing jets, which is $2\sqrt{A_r}$. Within the studied range of parameters, the resulting jet throw (X_{III}) reached $10\sqrt{A_r}$.

Beyond the third jet zone, there is a stagnant zone in which the velocity values are relatively uniform and have an unstable direction. There is reverse flow in Zones I–III which is located between the jet boundaries and the cylinder walls. The maximum value of the velocity in the reverse flow is in the cross-section at the end of zone I at the distance X_I. The following equation was derived to calculate the length of the first zone X_I:

$$X_1 = \frac{0.755}{\Lambda} Y_b + \frac{\theta}{\Lambda K_{11}^2} \frac{I_{02}}{I_{01}} l_i \Phi(\overline{X}). \tag{7.187}$$

The average experimental value of the coefficient θ is 1.7 with ass standard deviation (σ_θ) of 0.05. Eq. (7.187) allows one to calculate the momentum ratio (I_{02}/I_{01}) required to extend the length of zone I to the value equal to X_I, given that the distance between the directing nozzles is equal to l_i. The graph presented in Fig. 7.52 is plotted according to Eq. (7.187) for K_{11} and K_{12} equal to 6.2. The maximum value of reverse flow velocity (v_{rev}) was found to be in the cross-section at X equal to X_I:

$$v_r^{max} = 0.73 v_{01} \sqrt{\frac{A_{01}}{A_r}\left(1 + \frac{SI_{02}}{I_{01}}\right)}, \tag{7.188}$$

where A_0 is the main stream supply air diffuser area, and S is the number of horizontal directing jets in Zone I.

Interaction of a nonisothermal main stream with horizontal directing jets

Studies of nonisothermal main stream and horizontal directing jet interaction were conducted to evaluate the maximum heat load that can be effectively

supplied by such HVAC systems. To summarize experimental data both in free and confined conditions, it was suggested that the abovementioned limiting condition is achieved when the current Archimedes number Ar_x (ratio of the buoyancy forces over inertia forces along the resulting jet axis) does not exceed some value μ:

$$Ar_x = \frac{gX}{v_{x\Sigma}^2}\frac{2\theta_m - \theta_r}{273 + \theta_r} \le \mu. \quad (7.189)$$

For the interaction of the main stream with N directing nozzles, the resulting expression for Ar_x can be presented as:

$$Ar_x = \frac{K_{21}}{K_{11}^2} Ar_0 \left(\frac{X}{d_{01}}\right)^2 \frac{1}{f}, \quad (7.190)$$

where

$$f = \frac{1}{N^2} + \frac{I_{02}}{I_{01}}\left(\frac{K_{12}}{K_{11}}\right)\left(\frac{X}{l_i}\right)^2 \sum_{i=1}^{N} \frac{1}{i^2}. \quad (7.191)$$

The current Archimedes number for the resulting jet grows along the jet as it does in any nonisothermal jet. However, the consequent momentum additions by directing jets increases the inertial forces in the resulting jet and thus at a certain cross-section the current Archimedes number falls. The number of directing jets after which Ar_x reaches the peak can be calculated using

$$N^* = \frac{1}{\sqrt{\left(I_{02}/I_{01}\right)\left(K_{12}/K_{11}\right)^2\left(1.202\sum_{i=1}^{N^*} i^{1/3}\right)}}. \quad (7.192)$$

From the experimental data for the free resulting jet, the μ value in Eq. (7.189) is equal to 0.075. Tests in field showed that the influence of the reverse flow and confining surfaces increases the value of μ to 0.2. Based on Eq. (7.189) and experimental μ values, the maximum initial air temperature supplied by the main stream is limited by

$$\theta_0 - \theta_r \le a\frac{K_{11}^2}{K_{21}}\frac{v_{01}^2 d_{01}}{l_i^2} f, \quad (7.193)$$

where a is a coefficient equal to 2.65 for a free jet and 7.07 for a confined jet.

7.4.6.4 *Interaction of jets supplied at an angle to each other*

There are only a few studies of air jets supplied at some angle α ($0 < \alpha < 90$ degrees) to each other. To predict characteristics (trajectory, velocity decay, etc.) of the flow resulting from interaction of two jets supplied at some angle toward each other, Hudenko[223] proposed to sum momentums of interacting jets as in the case with parallel jets. He has estimated that the error of prediction will be smaller at a smaller

- interaction angle,
- distance between the supply nozzles,
- difference in the nozzle sizes, and
- supply air velocity values.

Meshalin[224] conducted experimental studies of two equal jets supplied at angles of 15, 30, and 45 degrees to each other. Based on the results of his studies, the author concluded that

- The turbulent mass transfer in the flow resulting from the interaction of the two jets is more extensive than in a single jet under the same supply conditions.
- The intensity of mass transfer in the flow in the plane of the interacting jets' axis is lower than in the plane of symmetry.
- The intensity of mass transfer in the resulting flow increases with the angle of interaction.

Numerous studies of jets supplied into a uniform or nonuniform cross-flow were conducted in application to such areas as air pollution control, burning processes, etc. Detailed discussion of these studies is beyond the current review. However, some results of these studies will be mentioned as needed in the following section.

Interaction of a free isothermal main stream with directing jets supplied at a right angle to the main stream

As in the case of the interaction of coaxial directing jets, the interaction of main streams with directing jets supplied at a right angle was studied[82,187] to develop a design method for air distribution with horizontal and vertical directing jets.

The discussion of the interaction of air jets supplied at some angle to each other shows that application of the method of superposition of the interacting jets' momentums and surplus heat to predict velocity and temperatures in the combined flow results in inaccuracy when two unequal jets are supplied at a right angle. A different approach was undertaken in the studies of interaction of the main stream with vertical directing jets.[82,187]

Visualization studies of the resulting flow showed that the directing jet changes its initial direction due to its interaction with a main stream. The interaction results in two separate flows; the first is a continuation of the main stream and the second is a continuation of the directing jet. The specific feature of this interaction is that vertical directing nozzles are located within a main stream (Fig. 7.53). The median diameter of the directing jet is significantly (several times) smaller than the main stream diameter within a zone of their

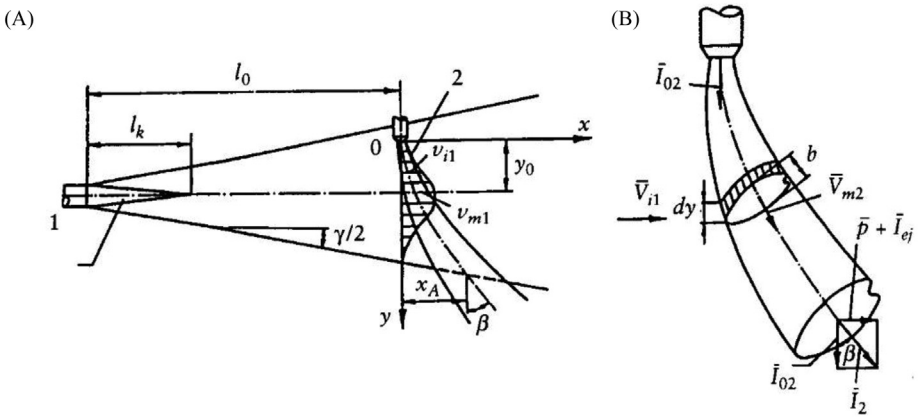


FIGURE 7.53 Schematic of free isothermal main stream and vertical directing jet interaction: I—main stream (D_{01}, V_{01}, I_{01}, d_{01}, and K_{11}) and 2—directing jet (D_{03}, V_{03}, I_{03}, d_{03}, and K_{13}). Source: *Reproduced from Zhivov AM. Concentrated air distribution with directing jets [Ph.D. thesis]. All-Union Research Institute for Labor Protection, St. Petersburg, Russia; 1983.*

interaction. Thus the interaction of the main stream and the vertical directing jet can be seen as an interaction of the axisymmetric (directing) jet with an infinite cross-draft with a nonuniform velocity profile.

An analytical solution of the interaction in the case of isothermal main and directing jets, assume that the main stream (Fig. 7.53), supplied with initial velocity (v_{01}) through a nozzle that has internal diameter (d_{01}), is developing within a zone ($-l_0$, 0) as a free jet. The momentum (I_1) of the jet within the zone ($-l_0 + l_c$, 0) remains equal to the initial momentum (I_{01}), and the velocity distribution in the cross-section of interaction in the plane XY remains the same within the zone (0, X_A). The axisymmetric main stream within the zone (0, X_A) is substituted by the linear flow with velocity profile that can be described by the formula

$$v_{i1} = v_{m1} e^{-(1/2)((Y-Y_0)/cl_0)^2}, X \in [0, X_A], c = 0.082 \quad (7.194)$$

The directing jet is supplied at a right angle to the main stream axis with an initial velocity of v_{03} from the nozzle with an inner diameter (d_{03}) located at the distance (l_0) from the plane of main stream supply and at the distance Y_0 from its geometrical axis. The momentum vector component along the Y-axis remains constant and equal to the initial momentum (Fig. 7.53):

$$I_3 \cos\beta = I_{03}, \quad (7.195)$$

where β is the angle between the Y-axis and the tangent to the directing jet trajectory

$$I_{03} = \rho_{03} v_{03}^2 \frac{\pi d_{03}^2}{4}. \quad (7.196)$$

The aerodynamic force (P) of the main stream and the momentum of the injected air (I_{inj}) change the X component of the directing jet. The X component of the directing jet can be calculated from

$$I_3 \sin\beta = P + I_{\text{inj}}. \quad (7.197)$$

Experimental studies have shown that velocity distribution in the cross-section of the directing jet can be described by the same equation as those in the axisymmetric jet in a cross-draft,

$$\frac{V_{i3}}{V_{03}} \cos(\bar{v}_{i3}, \bar{n}) = \left[1 - \left(\frac{r_i}{R_b}\right)^{1.5}\right]^2, \quad (7.198)$$

where $r_i = \sqrt{S_i/\pi}, R_b = b/2$, and S_i is a cross-sectional area limited by the constant velocity line. The joint solution of Eqs. (7.195), (7.197), and (7.198) results in the following expression for the maximum velocity along the directing jet:

$$\frac{V_{m3}}{V_{03}} = \chi_v \frac{d_{03}}{Y}, \quad (7.199)$$

where

$$\chi_v = \frac{m_2}{1 + (a/q^k)} \frac{1}{\sqrt{\cos\beta}}. \quad (7.200)$$

When $\beta < 25$ degrees, $\sqrt{\cos\beta} \approx 1$.

The differential equation for the directing jet trajectory was obtained by the joint solution of Eqs. (7.195) and (7.197) assuming $\sqrt{\cos\beta} = 1$:

$$tg\beta = \frac{P + I_{inj}}{I_{03}} = \frac{c^2 k_0}{3} \frac{m_1}{\chi_v} I_0 \frac{1}{I_{03}} A, \quad (7.201)$$

where

$$A = e^{-\left(\frac{Y_0}{cl_0}\right)^2} - e^{-\left(\frac{Y-Y_0}{cl_0}\right)^2} + \frac{\sqrt{\pi} Y_0}{cl_0}\left[\text{erf}\left(\frac{Y - Y_0}{cl_0}\right) + \text{erf}\left(\frac{Y}{cl_0}\right)\right] \quad (7.202)$$

The equation for the directing jet trajectory was obtained by the integration of Eq. (7.201) at $X = 0, Y = 0$:

$$X = \frac{ck_0}{3} \frac{m_1^2}{\chi_v} \frac{I_{01}}{I_{03}} B, \quad (7.203)$$

where

$$B=(Y-Y_0)\left[e^{-(Y_0/cl_0)^2}+\frac{\sqrt{\pi}Y_0}{cl_0}\left(\frac{Y_0}{cl_0}\right)\right]-\frac{cl_0\sqrt{\pi}}{2}\left[\mathrm{erf}\left(\frac{Y-Y_0}{cl_0}\right)+\mathrm{erf}\left(\frac{Y_0}{cl_0}\right)\right]+Y_0\left[e^{-((Y-Y_0)/cl_0)^2}+\frac{\sqrt{\pi}(Y-Y_0)}{cl_0}\mathrm{erf}\left(\frac{Y-Y_0}{cl_0}\right)\right]. \quad (7.204)$$

Based on Eqs. (7.202′) and (7.204) one can conclude that beyond the boundary of the main stream the directing jet has a straight trajectory.

Visualization studies of the directing jet showed that after interacting with a main stream the directing jet has a straight trajectory when β is less than 50 degrees. At a greater value of β ($tg\beta > 1.2$) the directing jet trajectory is significantly curved.

In the case of a nonisothermal directing jet, the abovementioned assumptions are true, except that the momentum vector component along the Y-axis changes due to the buoyancy force:

$$dI_{y3} = \pm dG \quad (7.205)$$

The amount of heat W_3 along the directing jets remains constant.

Experimental studies have shown[187] that the temperature distribution in the cross-section of the directing jets can be described as follows:

$$\frac{\theta_{i3}-\theta_r}{\theta_{o3}-\theta_r}=1-\left(\frac{r_i}{R_b}\right)^{1.5} \quad (7.206)$$

Based on the abovementioned assumptions, the following equations were derived to calculate the velocity and temperature decay along the nonisothermal directing jet:

$$\frac{V_{m3}}{V_{03}}=\chi_r\frac{d_{03}}{Y}\sqrt{1\pm 1.8\mathrm{Ar}_y\sqrt{\cos\beta}\sqrt{\frac{\rho 03}{\rho_r}}} \quad (7.207)$$

$$\frac{\theta_{m3}-\theta_r}{\theta_{o3}-\theta_r}=\chi_t\frac{\sqrt{\cos\beta}\left(\sqrt{\rho_{03}/\rho_r}\right)}{\sqrt{1\pm 1.8\mathrm{Ar}_y\sqrt{\cos\beta}\left(\sqrt{\rho_{03}/\rho_r}\right)}}\frac{d_{03}}{Y} \quad (7.208)$$

where

$$\mathrm{Ar}_r=\frac{1}{\chi_v}\frac{gd_{03}}{v_{03}^2}\frac{\theta_{03}-\theta_r}{273+\theta_r}\left(\frac{Y}{d_{03}}\right)^2 \quad (7.209)$$

and

$$\chi_t=\frac{n_3}{1+(a/q^k)}. \quad (7.210)$$

In the case when there is no main stream ($v_{01}=0$, $q=\infty$, and $\beta=0$), Eqs. (7.207)–(7.210) reduce to those for free jets.

Joint solution of Eqs. (7.197) and (7.205) allows one to calculate the maximum amount of heat supplied by a directing jet with the assumption that the jet reaches the occupied zone ($v_{m3}>0.1$ m/s) and $(tg\beta_n - tg\beta)/tg\beta$ is less than 0.2 at the point where it enters the occupied zone. The maximum initial temperature difference of the air supplied by vertical directing jet is

$$(\theta_{03}-\theta_\tau)^{\max}=32\frac{v_{03}^2 d_{03}}{\left(h_{03}-h_{o,z}\right)^2}\frac{1}{1+23.7\sqrt{\frac{d_{01}}{l_0}}} \quad (7.211)$$

Based on the experimental results, the following values of coefficients were found: $a=9.5$, $k=0.25$, and $k_0=3.3$ at $0<\beta<25$ degrees and $k_0=2.4$ at $25<\beta<50$ degrees.

7.4.7 Applications of air jets

In industrial ventilation, both the experiment and CFD simulation have been used for the study of air jets. Conventional experiments used single-point measurements, include fast-response aerodynamic probe,[225,226] thermal anemometer,[227] laser Doppler velocimetry (or laser Doppler anemonetry),[228,229] etc. With advances in laser imaging techniques, whole-field measurements have been developed recently, include particle-image velocimetry,[230] laser-induced fluorescence (LIF).[231,232] For the whole-field measurements, high-speed digital cameras were often used for recording the flows, to obtain the high resolution of the images.

Besides the experiment, CFD simulation is also a competitive means to characterize the free jet flows for its convenient and rapidity. RANS turbulence models,[233,234] LES,[235,236] and direct numerical simulation (DNS)[237,238] were often used to solve the turbulent free jets. For RANS models, only time-averaged properties of the flow are required to be known, which may discard the details of the flow in the instantaneous fluctuations. While in DNS, no turbulence models are required, and the whole spectrum of turbulent scales are directly resolved, which results in huge computing burden. To balance the accuracy and computing load between the RANS and DNS models, LES was developed, in which large eddies are resolved directly and small eddies are modeled. Even though, CFD simulations for jets still need a large amount of computing burden to capture the characteristic of the flow.

Recently some pioneering studies started to apply POD to reconstruct the air flow jets. Lam[231] used POD to extract the crucial modes for jet reconstruction, and

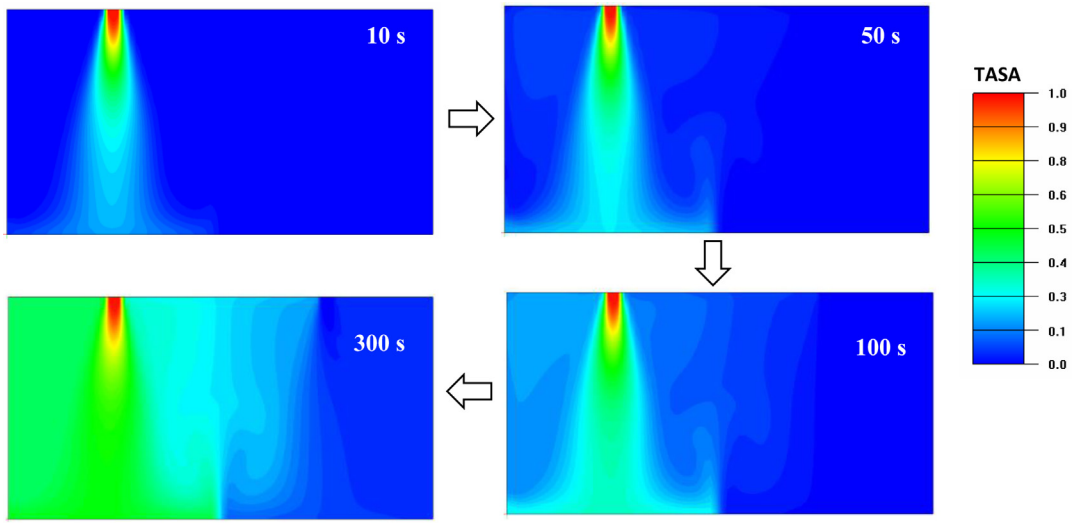


FIGURE 7.54 Impact of air jet on areas at different moments.

compared the velocity fields from POD with that from the LIF experiment. Results show that with sufficient POD modes, the reconstructed field can be very similar with the experiments.

7.4.8 Effectiveness of air jet to different areas

The characteristics of the jet flow are introduced in detail in the earlier sections. While the roles of the various air jets in transporting different types of contaminants, as well as influencing the air parameter distributions need to be revealed by adopting the appropriate evaluation indices. TASA is an alternative index to evaluate the effectiveness of an air jet in transporting the contaminant to different areas (Section 7.3.3). As a demonstration, Fig. 7.54 shows the evaluation result of the left air jet in a typical room with dimensions of 6 m (length) × 2.6 m (height) × 3 m (width).

As illustrated in Fig. 7.54, the left air jet has a larger impact (with a larger TASA value) on the jet path area only at 10 seconds. As time goes by, more areas are notably influenced by the supply air. At 300 seconds, most of the left space has been dominated by the left air jet.

A comparison of the impacts between two different air jets at 100 seconds is shown in Fig. 7.55, where the flow rate of the left air jet in the right figure is twice of that in the left figure.

After the flow rate of the left air jet is doubled, the jet momentum is enhanced. Therefore the impact of the left jet on different areas becomes stronger. As shown in Fig. 7.55, the TASA values become larger, indicating a stronger effect. However, the influence from the left air jet on the right space is still much lower than that on the left space.

7.5 Plumes

7.5.1 Natural convection flows

When an object is warmer than the surrounding air, the air is heated, and the warm air moves upward due to buoyancy. The air current created in this way is called a natural convection flow or plume. Also if the object is colder than the surroundings, the descending cool air current is called a natural convection flow or plume. Generally heat transfer involving motion of air or some other fluid caused by a difference in density is called natural or free convection. As a result of free convection, a flow of air or other fluid is produced in the form of a boundary layer moving along a surface or as a thermal plume above a surface. In a building, natural convection flows can be formed along the cold or warm vertical surfaces of the external walls and windows, along vertical hot surfaces of process equipment, etc., as shown in Fig. 7.56. Convection flows or thermal plumes are created above people, lights, hot horizontal surfaces of process equipment, and other objects with a surface temperature greater than the room air temperature (Fig. 7.57).

As shown in Figs. 7.56 and 7.57, the amount of air in the convection flows increases with height, due to entrainment of the surrounding air. The amount of air transported in a natural convection flow depends on the temperature and the geometry of the surface or source and the temperature of the surrounding air. Because the driving force in convection flows is the buoyancy force caused by the density difference (i.e., the temperature

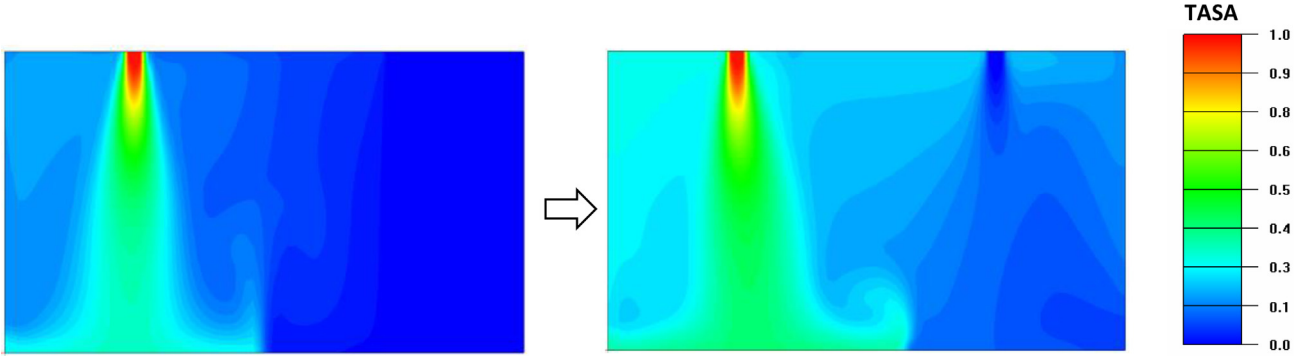


FIGURE 7.55 Change in the impact level of air jet when increasing the flow rate of the left air jet at 100 seconds.

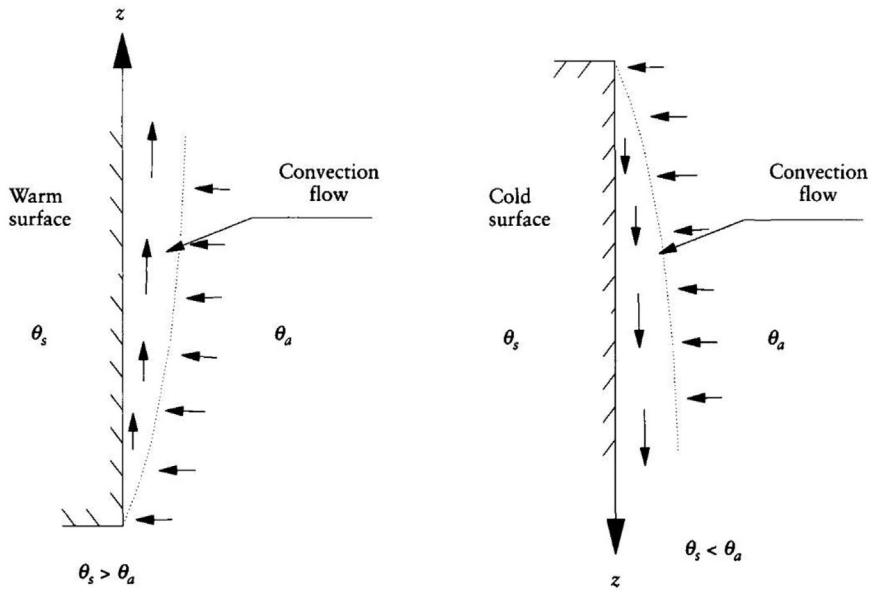


FIGURE 7.56 Convection flows along vertical surfaces.

difference), a temperature gradient in the room influences the plume rise height. Information on thermal plume characteristics is essential for designing ventilation systems with displacement air supply and for dimensioning overhead hoods above heat sources. Empirical, analytical, and CFD are the commonly used approaches to evaluate air temperatures, velocities, and airflow rates in thermal plumes above different heat sources and in convection flows at vertical surfaces.

This section treats

- natural convection flows in nonconfined and nonstratified environments (Section 7.5.2),
- plume interaction (Section 7.5.3),
- the influence of a confined space on convection flows (Section 7.5.4), and
- the influence of a temperature stratification (Section 7.5.5).

7.5.2 Nonconfined and nonstratified environments

7.5.2.1 *Plumes from point and line sources*

Thermal plumes above point (Fig. 7.58) and line (Fig. 7.59) sources have been studied for many years. Among the earliest publications are those from Zeldovich[239] and Schmidt.[240] Analytical equations to calculate velocities, temperatures, and airflow rates in thermal plumes over point and line heat sources with given heat loads were derived based on the momentum and energy conservation equations, assuming Gaussian velocity and excessive temperature distribution in thermal plume cross-sections.[241] These equations correspond to those determined experimentally by other researchers[242,243] and are listed in Table 7.19. The equations in Table 7.19 were derived with the

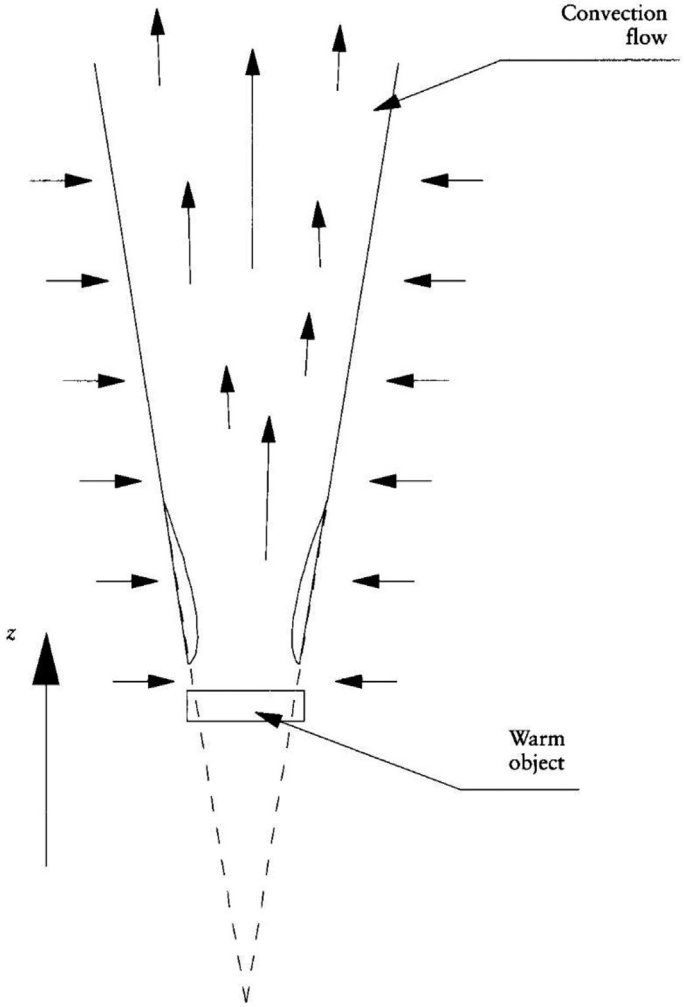


FIGURE 7.57 Thermal plume above a horizontal surface.

assumption that the heat source size was very small and did not account for the actual source dimensions.

The coefficients in the equations differ slightly in different references, depending on the entrainment coefficients used. The convective heat flux Φ, in W or W/m from the heat source, can be estimated from the energy consumption of the heat source Φ_{tot} by

$$\Phi = k\Phi_{tot} \tag{7.212}$$

The value of the coefficient k is 0.7–0.9 for pipes and ducts, 0.4–0.6 for smaller components, and 0.3–0.5 for larger machines and components.[244]

Example 7.5.1

A point source has a convective heat output of 100 W (see Fig. 7.60). Determine the airflow rate 1 m above the source.

Solution Table 7.19 gives the equation to be used:

$$q_{v,z} = 0.005\Phi^{1/3}z^{5/3}. \tag{7.213}$$

This gives

$$q_{v,z} = 0.005 \times 1^{(5/3)} = 0.23\text{m}^3/\text{s}.$$

Moreover, the mixing and dispersion of passive plumes emitted from two line sources or point sources are also investigated in recent years.[245,246] The interference arising from the dispersion of passive scalar plumes released from a pair of point sources in fully developed wall-bounded shear flow is investigated by numerical simulation.[247]

7.5.2.2 *Convection flow along vertical surfaces*

Convection flow along vertical surfaces (Fig. 7.61) is also of major interest in industrial ventilation, where large production units with a vertical extension are often present.

When the vertical extension of the surface is small, the convection flow is mainly laminar, but at larger extensions the flow is turbulent. The basic equations for a surface with a constant temperature are given in Table 7.20.[51,248] The temperature difference between the surface and the surrounding air is $\Delta\theta$, and z is the height from the bottom of the surface. The flow changes from laminar to turbulent at

$$\text{Gr} \times \text{Pr} = 7 \times 10^8 \tag{7.214}$$

which for air at moderate temperature differences means around $z = 1$ m and for air at higher temperatures around $z = 0.5$ m.

Example 7.5.2

Calculate the airflow rate along an external wall with a surface temperature 3°C above room temperature, at a height of 4 m above the lower edge of the surface.

Table 7.20 gives the equation to be used:

$$q_{v,z} = 0.00275\Delta\theta^{0.4}z^{1.2} \tag{7.215}$$

which gives

$$q_{v,z} = 0.00275 \times 3^{0,4} \times 4^{1.2} = 0.0225\text{m}^3/\text{s}$$

7.5.2.3 *Convection flow from horizontal surfaces*

Convection flows from horizontal surfaces are very difficult to determine in the same basic way as for point, line, or vertical sources. The reason is that the flows behave in a very unstable way and leave the flat surface from different positions at different times, partly depending on the total air movement in the room. These surfaces are mostly treated as plumes from extended surfaces; see later.

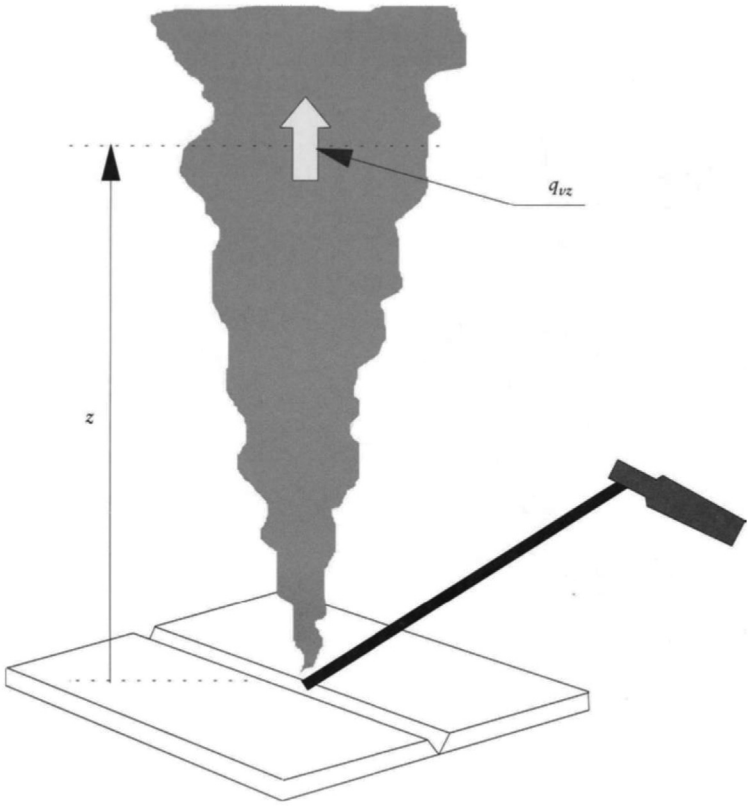


FIGURE 7.58 Plume from a point source.

7.5.2.4 *Plumes from extended sources*

In reality, heat sources are seldom a point, a line, or a plane vertical surface. The most common approach to account for the real source dimensions is to use a virtual source from which the airflow rates are calculated,[3,249–251] see Fig. 7.62. The virtual origin is located along the plume axis at a distance z_0 on the other side of the real source surface. The adjustment of the point source model to the realistic sources using the virtual source method gives a reasonable estimate of the airflow rate in thermal plumes. The weakness of this method is in estimating the location of the virtual point source.

The method of determining a maximum case and a minimum case provides a tool for such estimation; see Fig. 7.63.[251] In the maximum case, the real source is replaced by a virtual point source such that the border of the plume above the point source passes through the top edge of the real source (e.g., cylinder).

The minimum case is when the diameter of vena contracta of the plume is about 80% of the upper surface diameter and is located approximately one-third of a diameter above the source. The spreading angle of the plume is set to 25 degrees. For low-temperature sources, Skistad[252] recommends the maximum case, whereas the minimum case best fits the measurements for larger, high-temperature sources.

For a flat heat source Morton et al.[252] suggested that the position of the virtual source be located at $z_0 = 1.7 - 2.1 \times D$ below the real source (Fig. 7.64). This corresponds well with the maximum and minimum method described above, which gives $z_0 = 1.47 - 2.25 \times D$ below a flat plate.

Mundt[241] calculates the thickness of the boundary layer (see Table 7.20) at the top of a vertical extended heat source and adds this to the source radii, and then calculates the position of the virtual source as $z_0 = 2.1$ $(D + 2\delta)$ before using the point source equation.

According to Bach et al.[253] the volume flow from the vertical surfaces should be added to the volume flow calculated by the equations for point or line sources.

Example 7.5.3

Calculate the convection flow rate, q_v at a height $z_{\text{floor}} = 2$ m above the floor in the plume above a hot cylinder with a diameter of $D = 0.66$ m and a height of $H = 0.66$ m. The convective heat flux is $\Phi = 5$ kW.

1. *The maximum case* (Fig. 7.65) In the maximum case we get

$$z_0 = D/(2 \tan 12.5^\circ) = 2.25D = 1.49 \text{ m},$$

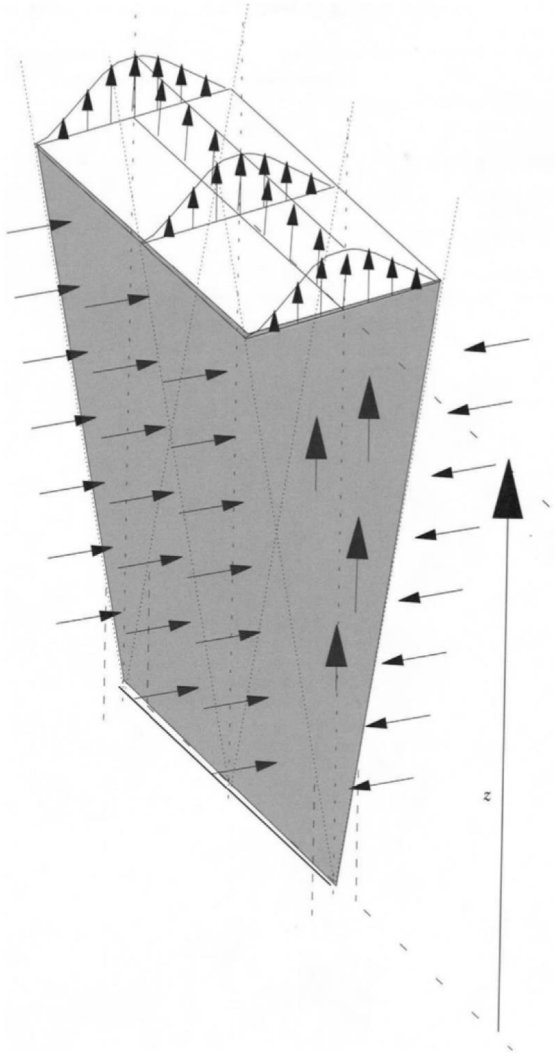


FIGURE 7.59 Plume from a line source.

TABLE 7.19 Characteristics of thermal plumes above point and line sources.

Parameter	Point source	Line source
Centerline velocity (m/s)	$v_z = 0.128\Phi^{1/3}z^{-1/3}$	$v_z = 0.067\Phi^{1/3}$
Centerline excess temperature (°C)	$\Delta\theta = 0.329\Phi^{2/3}z^{-5/3}$	$\Delta\theta = 0.094\Phi^{2/3}z^{-1}$
Airflow rate (m^3/s)	$q_{v,z} = 0.005\Phi^{1/3}z^{5/3}$	$q_{v,z} = 0.013\Phi^{1/3}z$
Φ = convective heat output (W/m)		
z = height above source (m)		

which means that the virtual source is located below the floor:

$$z_p = H - z_0 = 0.66\text{ m} - 1.49\text{ m} = -0.83\text{ m}.$$

The vertical distance to be used in the plume formula is

$$z = z_{\text{floor}} - z_p = 2.0\text{ m} - (-0.83\text{ m}) = 2.83\text{ m},$$

and from Table 7.19, we find the formula for the volume flow above a point source:

$$q_v = 0.005\Phi^{1/3}z^{5/3}, \tag{7.216}$$

which gives

$$q_v = 0.005 \cdot 5000^{1/3} \cdot 2.83^{5/3} = 0.48 m^3/s$$

2. *The minimum case* (Fig. 7.66) In the minimum case we get

$$z_0 = 0.8D/(2\tan 12.5^\circ) = 1.804D = 1.19\text{ m}.$$

The virtual source is located slightly below the floor:

$$z_p = (D/3) + H - z_0 = 0.22\text{ m} + 0.66\text{ m} - 1.19\text{ m} = -0.31\text{ m}.$$

The vertical distance to be used in the plume formula is

$$z = z_{\text{floor}} - z_p = 2.00\text{ m} - (-0.31\text{ m}) = 2.31\text{ m},$$

which gives the volume flow rate from Eq. (7.216):

$$q_v = 0.005 \times 5000^{1/3} \times 2.31^{5/3} = 0.35\text{ m}^3/s$$

Conclusion: According to the maximum and minimum method, the convection flow rate through a level 2 m above the floor is between 0.35 and 0.48 m^3/s.

Moreover, the plumes around human body are also typical thermal plume from extended sources. The radiative and convective heat transfer coefficients of human body in natural convection are also investigated experimentally.[254,255] The plumes in different sitting postures around human body are investigated by performing CFD simulations.[256–259]

7.5.3 Plume interaction

When a heat source is located close to a wall, the plume may attach to the wall; see Fig. 7.67. In this case the entrainment will be reduced compared with the entrainment in a free plume, and the attached plume can be regarded as half the plume from the source, with its mirror image on the other side of the wall; see Fig. 7.68.

The airflow rate from a heat source can then be calculated as half of the flow from a source with a heat emission of 2Φ[244]:

$$q_v = 1/2 \cdot 0.0052\Phi^{1/3}z^{5/3} = 0.0032\Phi^{1/3}z^{5/3}. \tag{7.217}$$

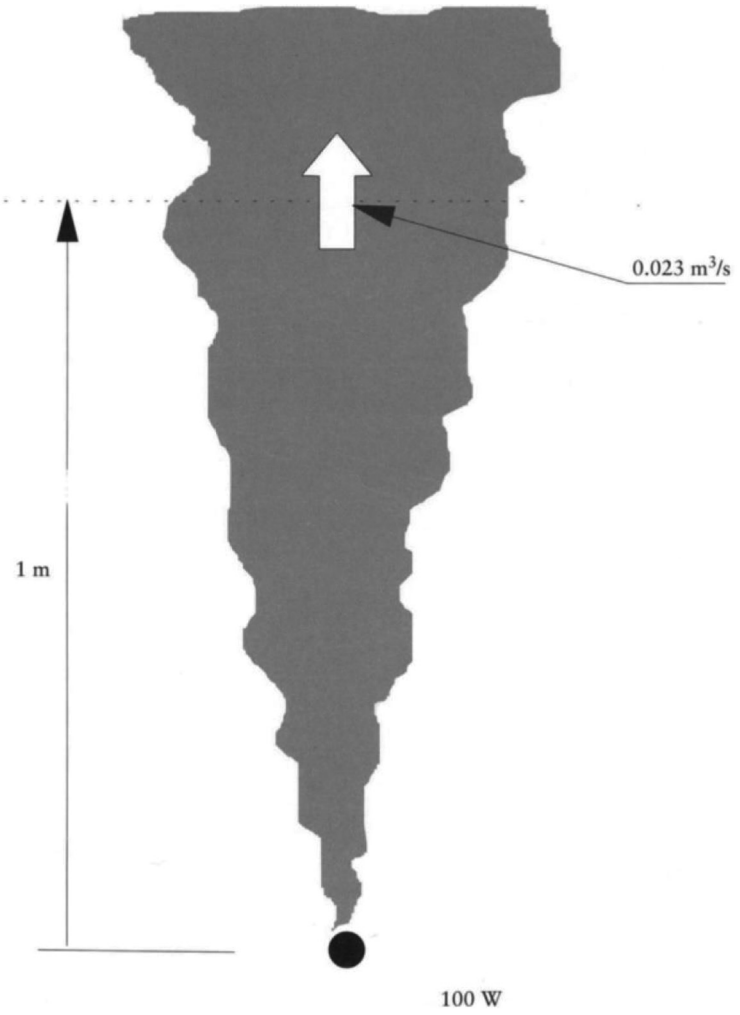


FIGURE 7.60 The plume of Example 7.5.1.

If the heat source is located in a corner, the airflow rate is equal to 25% of the airflow from a heat source with a heat emission of 4Φ[260]:

$$q_v = 1/4 \cdot 0.0025(4\Phi)^{1/3} z^{5/3} = 0.002\Phi^{1/3} z^{5/3}. \quad (7.218)$$

This follows from the same reasoning as above. See Fig. 7.69.

When several heat sources are located close to each other, the plumes may merge into a single plume; see Fig. 7.70. In this case, the source should be regarded as one single source, with the heat emission equal to the sum of the heat emission from each of the sources:

$$q_v = 0.005(\Sigma\Phi)^{1/3} z^{5/3} \quad (7.219)$$

where $\Sigma\Phi$ is the sum of the individual heat emissions (W), and z is the height above the virtual source (m). The total flow from N identical sources is then given by[261]

$$q_{v,N} = q_{v,1} N^{1/3} \quad (7.220)$$

where $q_{v,1}$ is the volume flow in the plume from one of the sources.

Plume interaction can be observed by water towing-tank experiment.[262] In addition, similar phenomena also exist in other fields besides industrial ventilation. Aerodynamic plume-on-plume interactions and plume-on-wall interactions can significantly change the heat transfer to the propellant grain in solid rocket motor.[263] Interaction of side-by-side cooling tower plumes is dominated by the interaction of the rotating vortex pairs within the plumes.[264]

7.5.4 Plumes in confined spaces

Fig. 7.71 shows a plume in an open environment. The hot air from the source entrains ambient air into the convection current (the plume), thus making the air volume flow increase with height.

Imagine that we enclose the plume, as shown in Fig. 7.72. The plume still entrains air from the

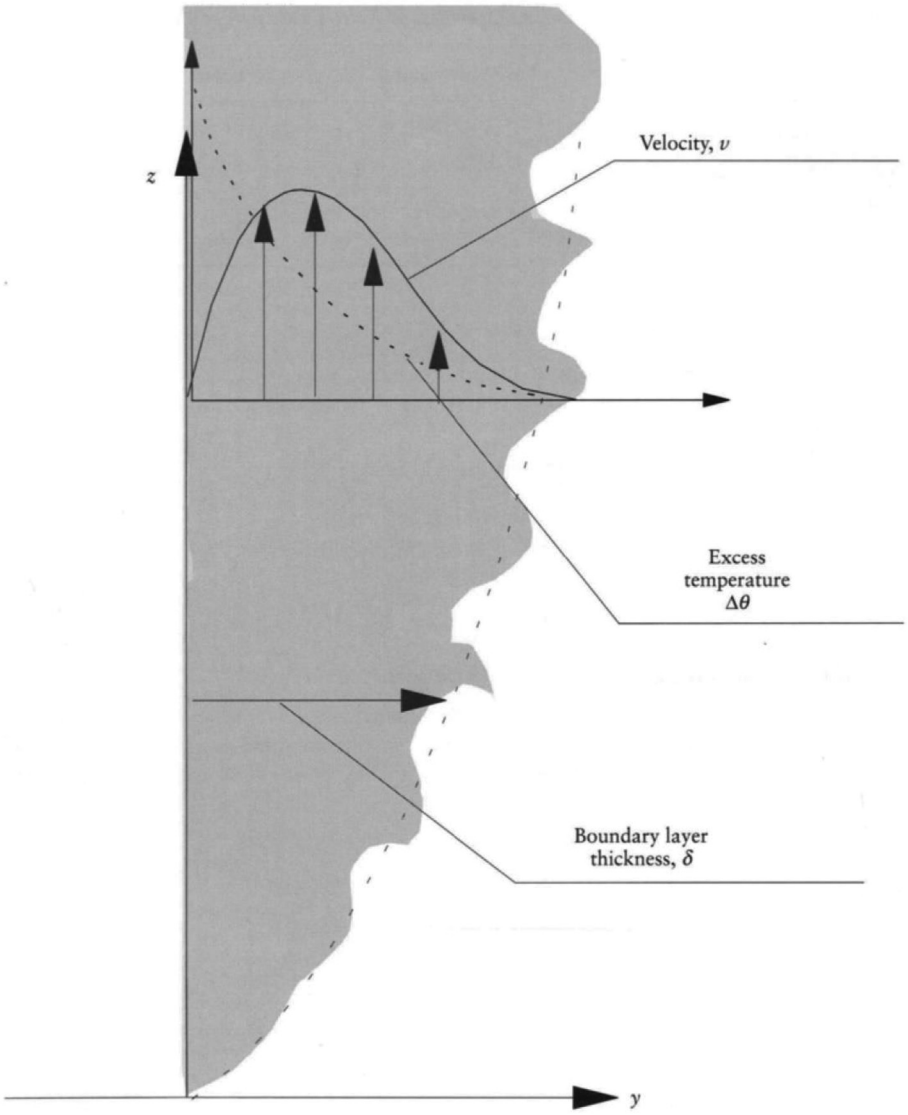


FIGURE 7.61 Convection flow along a vertical surface.

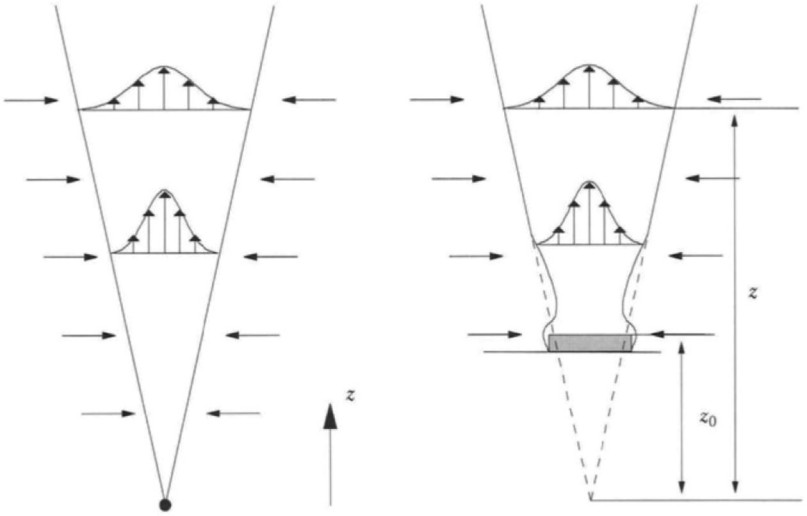
TABLE 7.20 Characteristics of convection flows along vertical surfaces.

Parameter	Laminar region	Turbulent region
Maximum velocity (m/s)	$v_z = 0.1(\Delta\theta z)^{1/2}$	
Thickness of boundary layer (m)	$\delta_z = 0.05(z/\Delta\theta)^{1/4}$	$\delta_z = 0.11 z^{0.7} \Delta\theta^{-0.1}$
Airflow rate (m^3/s)	$q_{v,z} = 0.00287 \Delta\theta^{1/4} z^{3/4}$	$q_{v,z} = 0.00275 \Delta\theta^{0.4} z^{1.2}$

surroundings, but the available fresh air is limited. This means that fresh air will surround the plume only up to a certain level. Above this level, the entrained air has to be recirculated from the plume itself. This leads to a two-zone flow model, with a layer of fresh air at the bottom, and warmer air from the plume at the top. The interface between the two layers is located at the height at which the entrained air in the plume equals the supplied air. This can be found from the volume-flow formula of Section 7.5.2. In addition, these characteristics can be observed by some experimental studies[265–267] or numerical studies.[268]. Similar phenomena also exist in other fields besides industrial ventilation, for example, shockwave dynamics.[269,270]

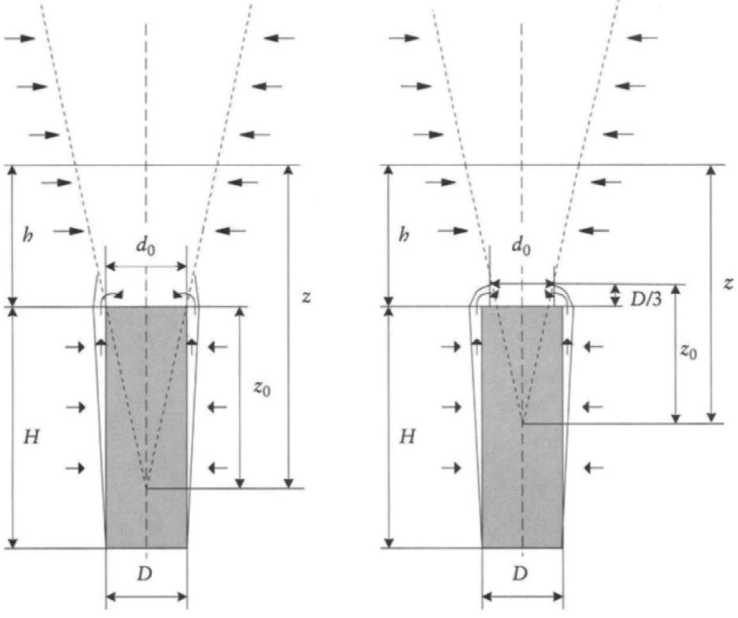
Example 7.5.4
Confined plume

We now put the hot cylinder of Example 7.5.3 inside a room. The convective heat output is still $\Phi = 5$ kW. The air is supplied at the floor, as shown in Fig. 7.73, at a rate of $q_v = 1\ m^3/s$ at 10°C, corresponding to an air mass flow of 1.25 kg/s. The same air mass flow is allowed to escape through the opening in the ceiling. This corresponds to a temperature



(A) Point source (B) Extended source

FIGURE 7.62 Illustration of the position of the virtual source.



(A) Maximum case (B) Minimum case

FIGURE 7.63 Convection flow above a vertical cylinder.

increase of $\Delta\theta = 4°C$. At what height will the interface between the fresh bottom layer and the upper recirculation layer stabilize?

From Table 7.19 we find the air volume flow (q_v) in the plume as a function of the height above the floor (z):

$$q_v = 0.005\Phi^{1/3}z^{5/3} \tag{7.221}$$

We can rearrange the equation so that we find the height for a given air volume flow in the plume:

$$z = \left(\frac{q_v}{0.005}\right)^{3/5}\frac{1}{\Phi^{1/5}},$$

which yields the height of the interface:

$$z = (1/0.005)^{3/5} \times (5000)^{-1/5} = 4.37 \text{ m}.$$

See Fig. 7.73.

7.5.5 Plumes in rooms with temperature stratification

The idealized case of Fig. 7.73 assumes that the room air temperature is constant in the lower, fresh air layer. In reality, the temperature increases with height

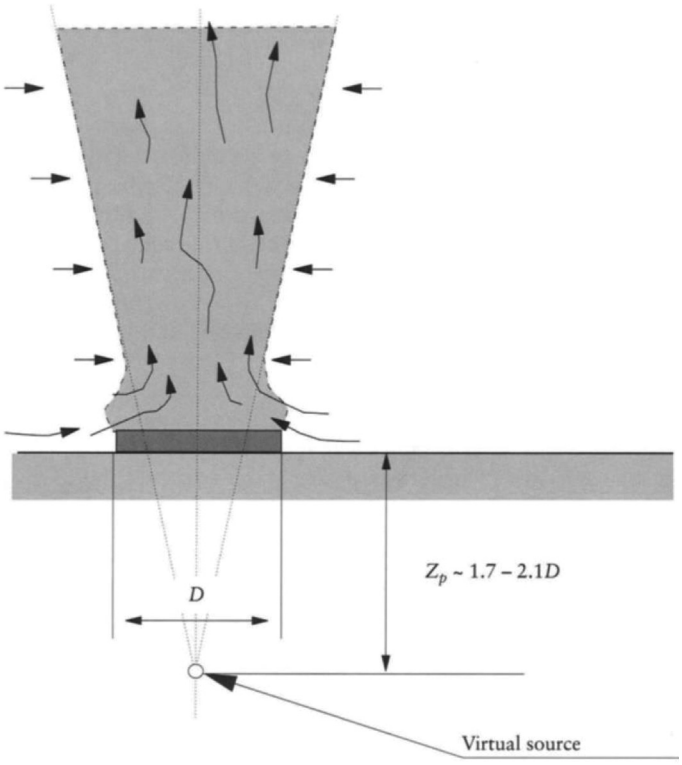


FIGURE 7.64 Virtual source according to Morton.

in both the lower and upper layers. Fig. 7.74 shows a typical vertical temperature distribution in a room enclosing a convection current. The air enters the room at low temperature, and is mixed slightly with the room air and heated by convection from the floor, thus making the air temperature at floor level higher than the supply temperature. The air temperature increases from the floor level up to the ceiling, more or less linearly, because the hottest air (in the core of the plume) rises to the ceiling due to its buoyancy, while the outer, cooler parts of the plume layer according to their temperature.

Plumes are influenced by the temperature stratification. The driving force of the plume is the temperature difference between the plume and the surroundings. When this difference diminishes, the plumes will disintegrate and spread horizontally in the room; see Fig. 7.75. The multiple sources produce a multiple layered stratification with each plume terminating in a given layer.

Batchelor[271] noticed the influence of a temperature gradient in the surroundings, and Morton et al.[252] gave a method for calculating the maximum plume rise from a point source in surroundings with a temperature gradient. The volume flow rate in plumes in a room with a temperature stratification is slightly

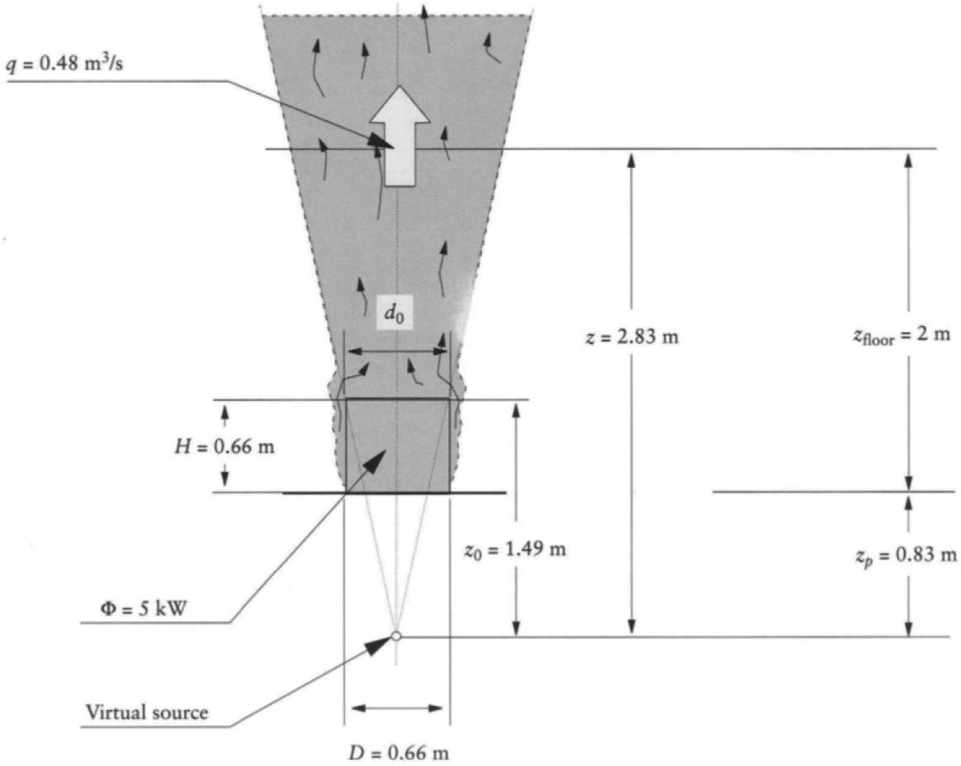


FIGURE 7.65 Plume above a cylinder, maximum case.

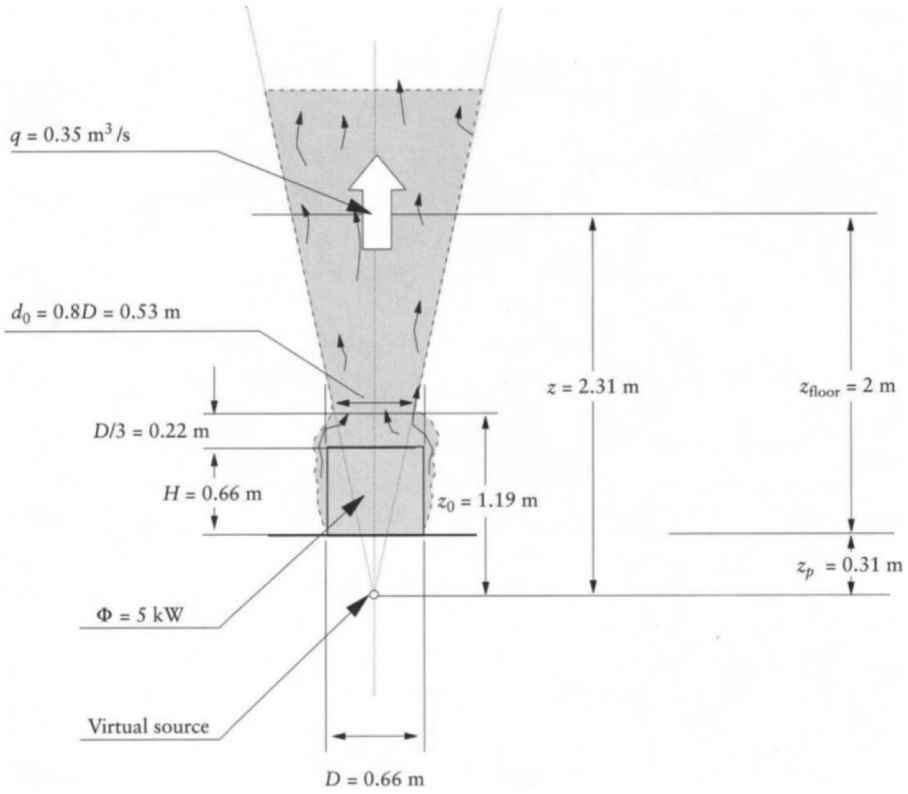


FIGURE 7.66 Plume above a cylinder, minimum case.

decreased compared with the volume flow rates calculated with the equations presented for nonstratified media.[251] Jin[272] studied the maximum plume rise height for plumes above welding arcs.

In the presence of the temperature gradient, the convective plume reaches its terminal height (z_t), where the temperature difference between the plume and the ambient air disappears. Also there is another point in the plume at which the air velocity equals zero. This is referred to as the maximum height of the plume (z_{max}). The plume spreads horizontally between these two heights.

7.5.5.1 Point source

For a point source, Mundt[241] gives the following plume rise formulae:

Maximum height:

$$z_{max} = 0.98\Phi^{1/4}\left(\frac{d\theta}{dz}\right)^{-3/8} \tag{7.222}$$

Equilibrium height:

$$z_t = 0.74\Phi^{1/4}\left(\frac{d\theta}{dz}\right)^{-3/8} \tag{7.223}$$

where $d\theta/dz$ is the vertical temperature gradient in the room air (°C/m), Φ is the convective heat from the source (W), z_{max} is the maximum plume rise height (m), and z_t is the equilibrium height of the plume (m).

Volume flow rate: the volume flow rate through a given height above the virtual point source in the plume can be found by the following calculation procedure, according to Mundt[241]:

1. Calculate the location of the virtual source and the corresponding z.
2. For the height z above the virtual source, calculate z_1 according to the formula

$$z_1 = 2.86z\left(\frac{d\theta}{dz}\right)^{3/8}\Phi^{-1/4} \tag{7.224}$$

   If $2.125 < z_1 < 2.8$, the density difference disappears and the calculations become uncertain; if $z_1 \geq 2.8$, the plume has reached its maximum height below the actual level.
3. Calculate

$$m_1 = 0.004 + 0.039z_1 + 0.380z_1^2 - 0.062\cdot z_1^3 \tag{7.225}$$

4. The volume flow rate in the plume through the height z can be found by

$$q_v = 0.00238\Phi^{3/4}\left(\frac{d\theta}{dz}\right)^{-5/8}m_1, \tag{7.226}$$

where q_v is the volume flow rate in m^3/s.

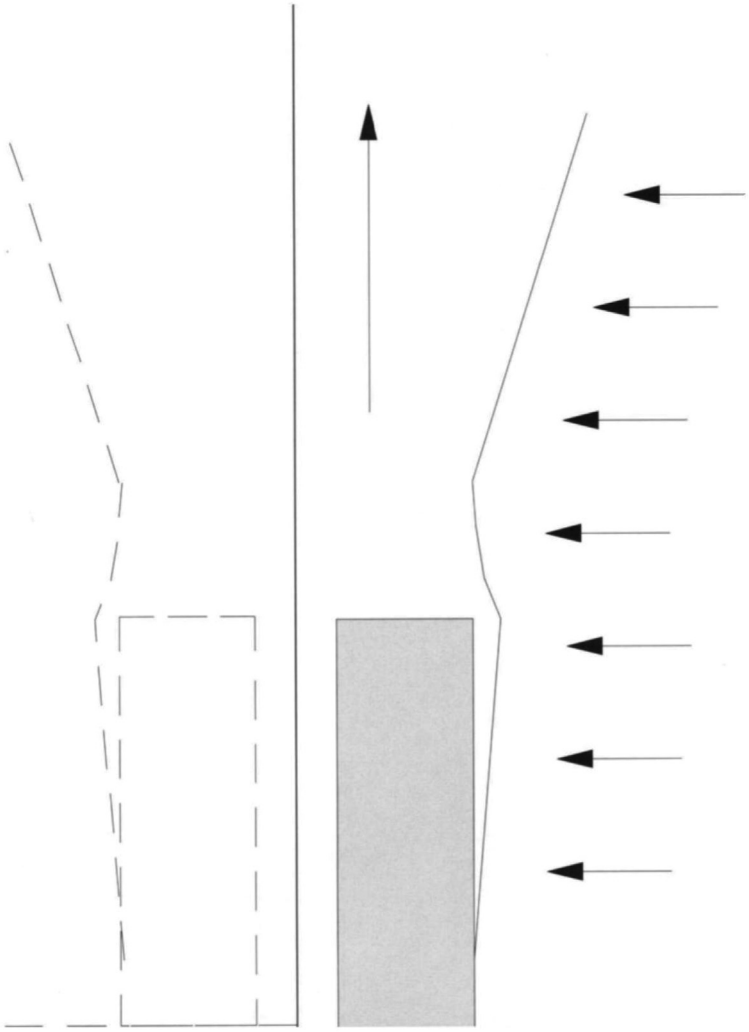


FIGURE 7.67 Thermal plume attached to a wall.

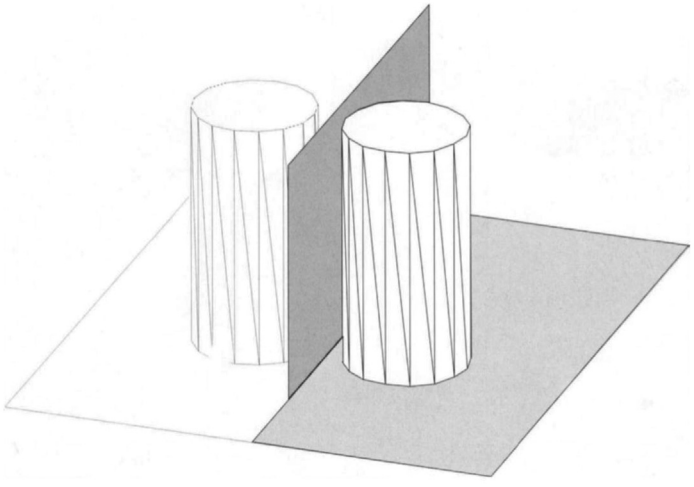


FIGURE 7.68 Convection source close to a wall.

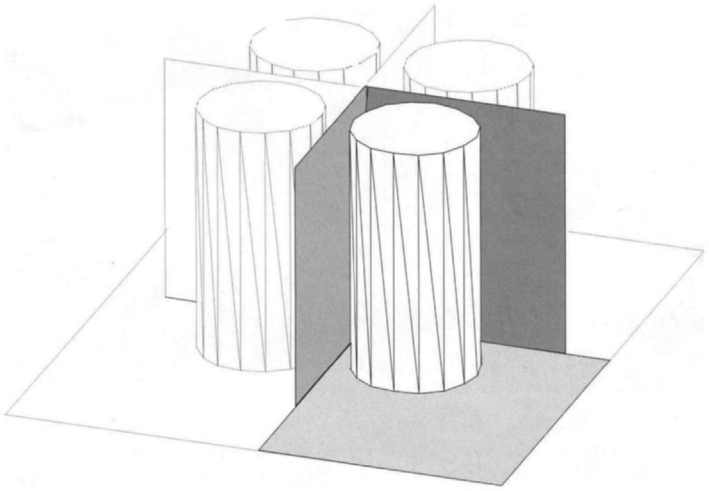


FIGURE 7.69 Convection source in a corner.

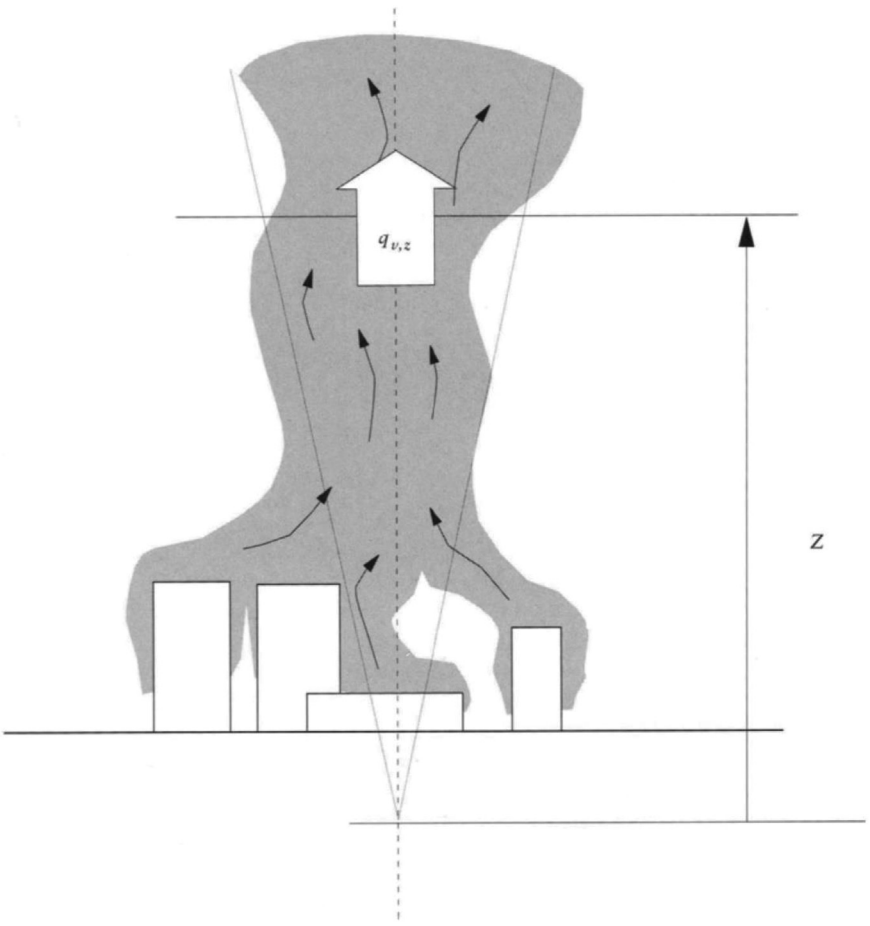


FIGURE 7.70 Interaction between several plumes.

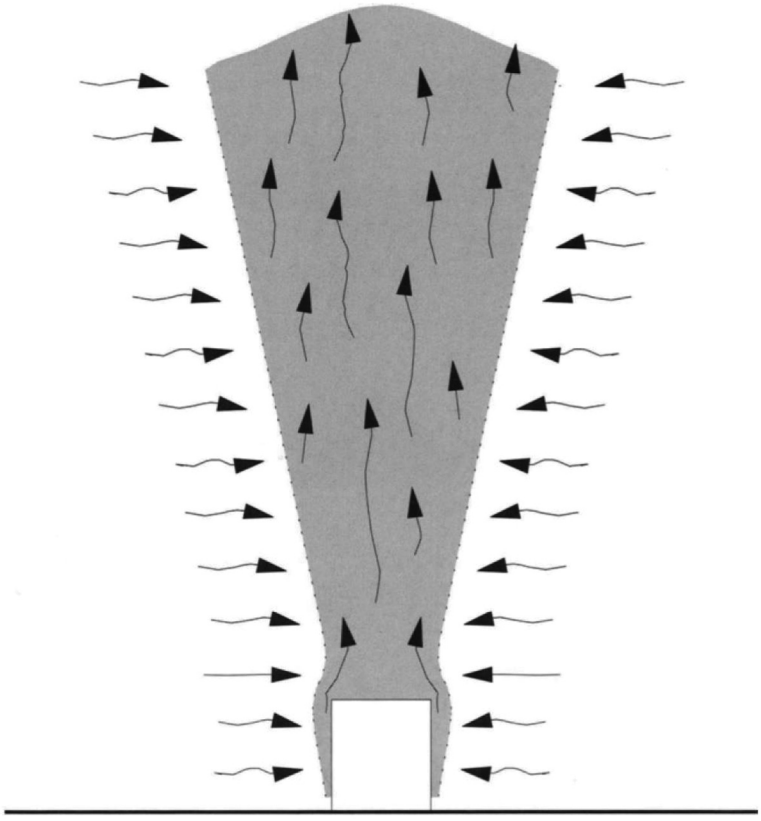


FIGURE 7.71 Plume in an open environment.

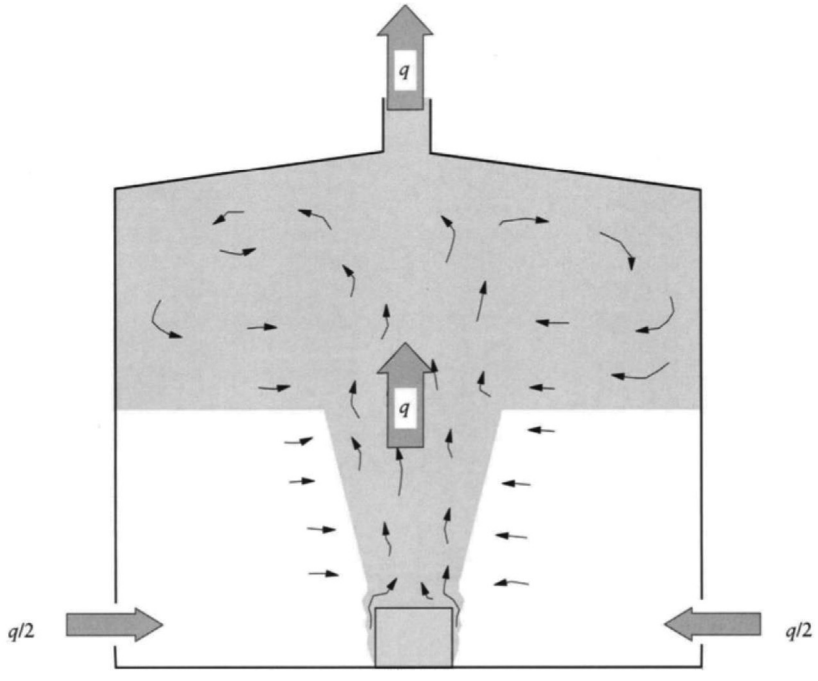


FIGURE 7.72 Plume in an enclosure.

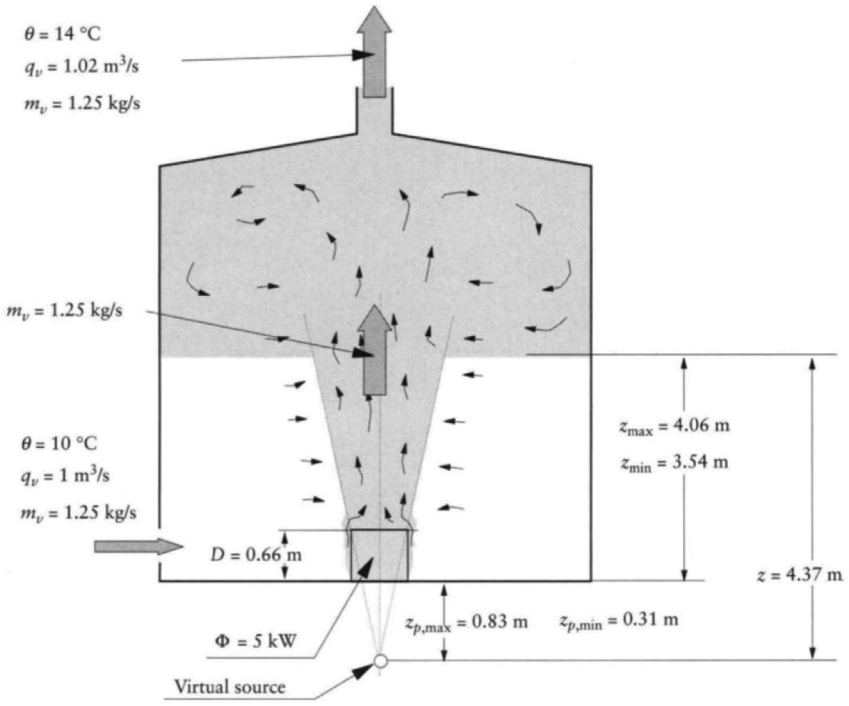


FIGURE 7.73 Calculated results of Example 7.5.4.

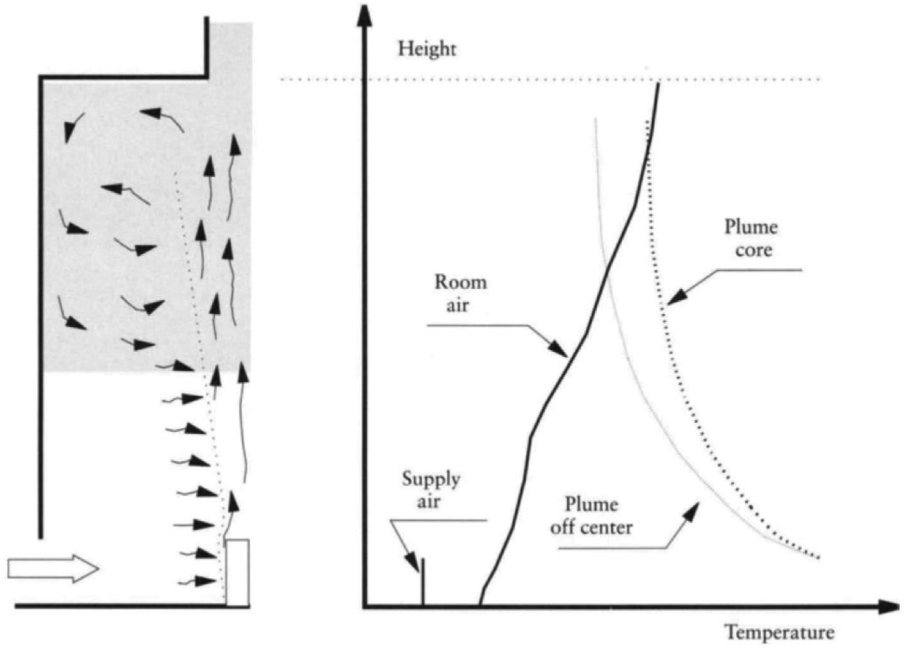


FIGURE 7.74 Vertical temperature distribution in the room of Fig. 7.72.

7.5.5.2 *Line source*

For a line source, Mundt[241] gives the following plume rise formulae:

Maximum height:

$$z_{\max} = 0.51\Phi^{1/3}\left(\frac{d\theta}{dz}\right)^{-1/2} \tag{7.227}$$

Equilibrium height:

$$z_t = 0.35\Phi^{1/3}\left(\frac{d\theta}{dz}\right)^{-1/2} \tag{7.228}$$

Volume flow rate:

- Calculate the location of the virtual source and the height z above the virtual source.

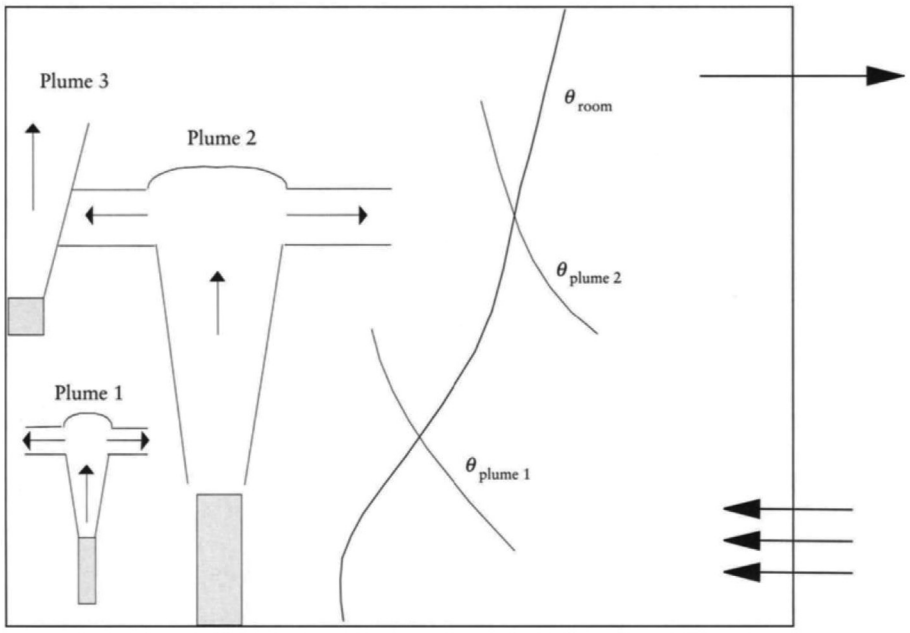


FIGURE 7.75 Schematic illustration of the airflow pattern in a room ventilated by displacement.

- For the height z above the virtual source, calculate z_1:

$$z_1 = 5.78z\left(d\theta/dz\right)^{1/2}\Phi^{-1/3} \tag{7.229}$$

If $2.0 < z_1 < 2.95$, the density difference disappears and the calculations become uncertain; if $z_1 \geq 2.95$, the plume has reached its maximum height below the actual level.

- Calculate

$$\sqrt{a} = 0.004 + 0.477z_1 + 0.029z_1^2 - 0.018z_1^3 \tag{7.230}$$

- The volume flow rate is given by

$$q_{v,1} = 0.00482\Phi^{2/3}\left(d\theta/dz\right)^{-1/2}\sqrt{a} \tag{7.231}$$

where

$q_{v,1}$ is the volume flow rate in m^3/s m.

Example 7.5.5

Point source in a room with thermal stratification

We now put the cylinder of Example 7.5.4 inside a room with a vertical temperature gradient of 1.5°C/m (see Fig. 7.76). In this case we assume that there are other heat sources in the room. We want to investigate how this temperature stratification influences the volume flow in the plume above the cylinder, and at what height the plume stops.

Following the formulae and calculation procedure of Section 7.5.5.1, we get

1. *Location of the virtual source:* we use the location calculated for the maximum case in Example 7.5.4, that is,

$$z_p = -0.83 \text{ m}.$$

2. *Maximum plume rise:* the maximum height above the virtual source is found from Eq. (7.222):

$$z_{max} = 0.98 \times 5000^{1/4} \times 1.5^{-3/8} = 7.08 \text{ m},$$

that is, the maximum height is 7.08 m above the virtual source, which is

$$7.08 \text{ m} - 0.83 \text{ m} = 6.25 \text{ m above the floor}$$

Equilibrium height (Eq. 7.223):

$$z_t = 0.74 \times 5000^{1/4} \times 1.5^{-3/8} = 5.34 \text{ m},$$

that is, the equilibrium or terminal height is 5.34 m above the virtual source, which is

$$5.34 - 0.83 = 4.51 \text{ m above the floor.}$$

3. *Volume flow rate in the plume:* to find the air volume flow rate through a level $z_{floor} = 3.5$ m, we must first calculate z and then z_1 at this level with Eq. (7.224):

$$z_1 = 2.86 \times (3.5 + 0.83) \times 1.5^{3/8}\ 5000^{-1/4} = 1.715$$

With z_1 we can calculate m_1 according to Eq. (7.225):

$$m_1 = 0.004 + 0.039z_1 + 0.380z_1^2 - 0.062z_1^3 = 0.875,$$

which gives the convection airflow rate from Eq. (7.226):

$$q_v = 0.00238 \times \left(5000^{3/4}\right) \times 1.5^{-5/8} \times 0.875 = 0.96 \text{ m}^3/s.$$

The volume flow calculated 3.5 m above the floor is slightly lower than that calculated for an isothermal atmosphere. The deviation between the volume flows calculated for isothermal room air and stratified room air can be seen is shown in Fig. 7.77.

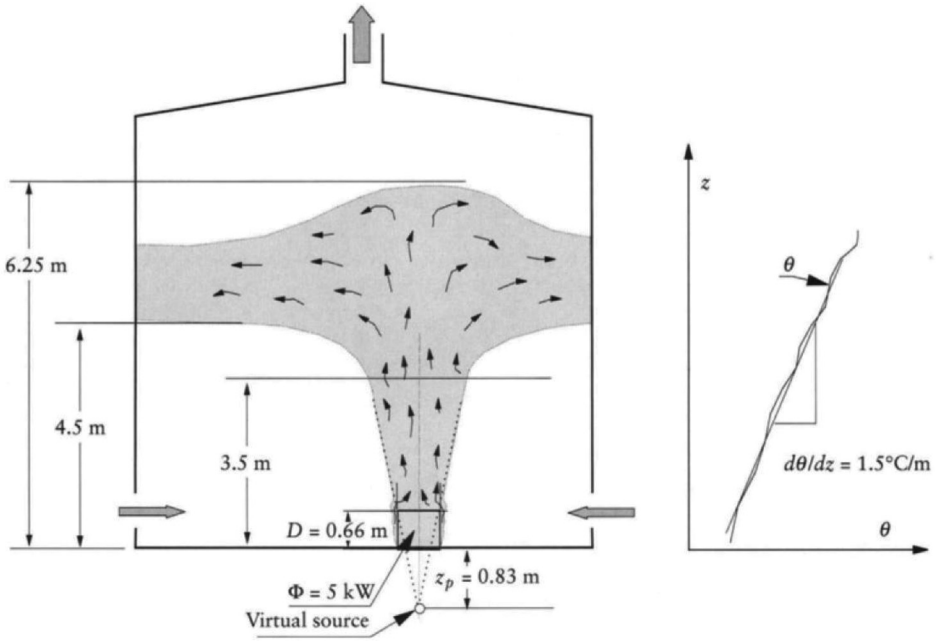


FIGURE 7.76 The cylinder of Example 7.5.4 in a room with thermal stratification.

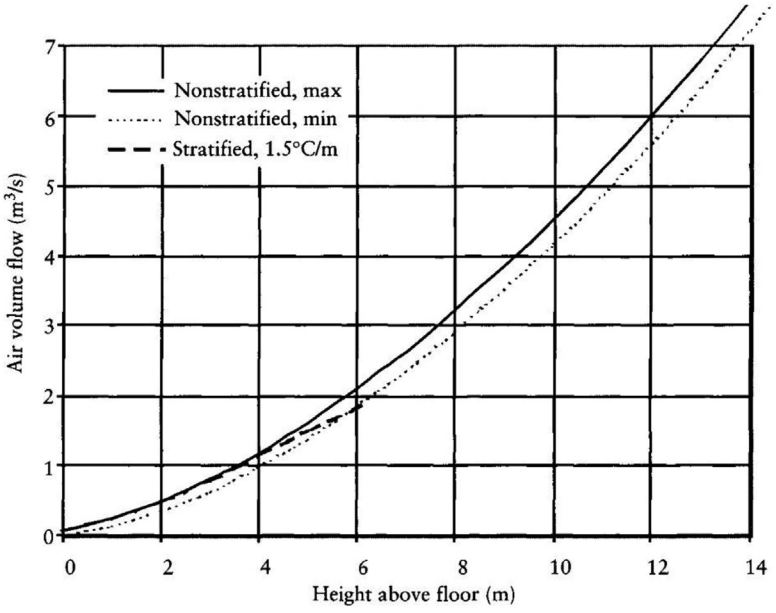


FIGURE 7.77 Airflow rate in the plume above the cylinder of Example 7.5.5.

Example 7.5.6

Comparison with numerical modeling

Calculate maximum air velocity, airflow rate, and excessive temperature (relative to the ambient air temperature equal to 20°C) in thermal plume above the heated cube (0.66 m × 0.66 m × 0.66 m) with convective heat production $W_{conv} = 225$ W, at heights of 2.0 and 4.0 m above the floor level. Neglect temperature gradient along the room height. Compare the results with predictions made for the same case using CFD code[273] (Fig. 7.78).

To calculate the thermal plume, the cube can be presented as a cylinder with a diameter equivalent to the hydraulic diameter of the top of the cube:

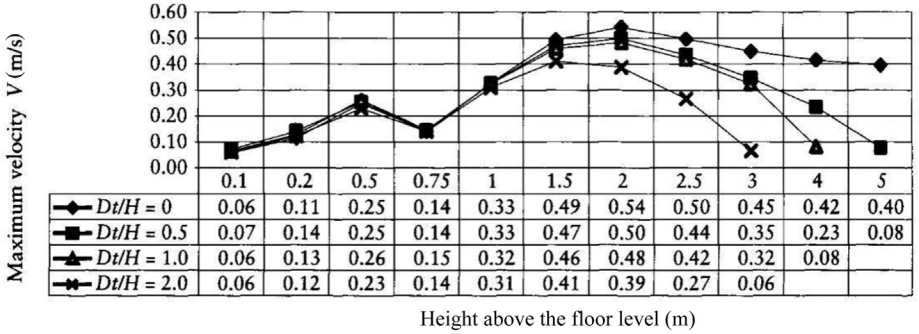
$$D = \frac{4 \times 0.66 \times 0.66}{4 \times 0.66} = 0.66\text{m}.$$

In the minimum case,

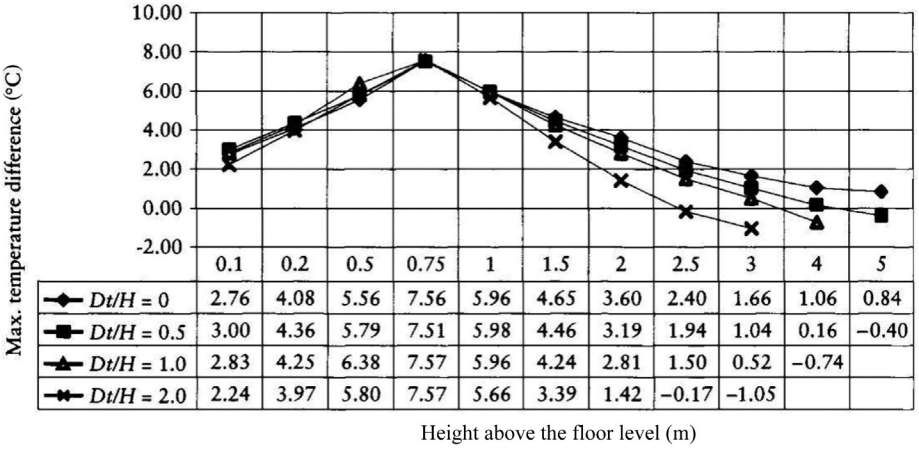
$$z_o = 0.8\text{D}/(2 \tan 12.5^\circ) = 1.19 \text{ m}.$$

The virtual source is located below the floor level:

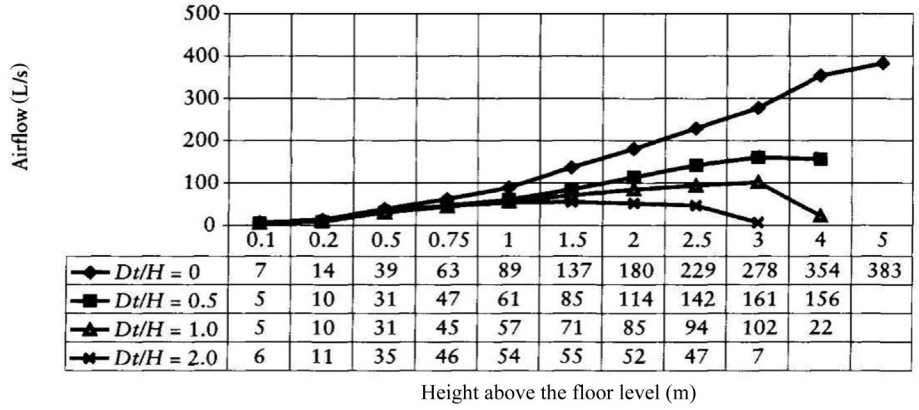
$$z_p = D/3 + H - z_o = 0.66/3 + 0.66 - 1.19 = -0.31 \text{ m}.$$



	0.1	0.2	0.5	0.75	1	1.5	2	2.5	3	4	5
Dt/H = 0	0.06	0.11	0.25	0.14	0.33	0.49	0.54	0.50	0.45	0.42	0.40
Dt/H = 0.5	0.07	0.14	0.25	0.14	0.33	0.47	0.50	0.44	0.35	0.23	0.08
Dt/H = 1.0	0.06	0.13	0.26	0.15	0.32	0.46	0.48	0.42	0.32	0.08	
Dt/H = 2.0	0.06	0.12	0.23	0.14	0.31	0.41	0.39	0.27	0.06		



	0.1	0.2	0.5	0.75	1	1.5	2	2.5	3	4	5
Dt/H = 0	2.76	4.08	5.56	7.56	5.96	4.65	3.60	2.40	1.66	1.06	0.84
Dt/H = 0.5	3.00	4.36	5.79	7.51	5.98	4.46	3.19	1.94	1.04	0.16	−0.40
Dt/H = 1.0	2.83	4.25	6.38	7.57	5.96	4.24	2.81	1.50	0.52	−0.74	
Dt/H = 2.0	2.24	3.97	5.80	7.57	5.66	3.39	1.42	−0.17	−1.05		



	0.1	0.2	0.5	0.75	1	1.5	2	2.5	3	4	5
Dt/H = 0	7	14	39	63	89	137	180	229	278	354	383
Dt/H = 0.5	5	10	31	47	61	85	114	142	161	156	
Dt/H = 1.0	5	10	31	45	57	71	85	94	102	22	
Dt/H = 2.0	6	11	35	46	54	55	52	47	7		

FIGURE 7.78 CDF-predicted values of maximum velocity V, temperature differential, θ_{max} (°C), and airflow, q (L/s), in the horizontal cross-section of the buoyant plume above the heated cube (0.66 m × 0.66 m × 0.66 m, 225 W).[273]

Thus the vertical distances to be used in the plume characteristics calculation are correspondingly equal to 2.31 and 4.31 m.

The maximum velocity in the thermal plume, from Table 7.19, is

$$v_{z,2m} = 0.128 \cdot 225^{1/3} \cdot 2.31^{-1/3} = 0.59\ m/s$$

and

$$v_{z,4m} = 0.128 \cdot 225^{1/3} \cdot 4.31^{-1/3} = 0.48\ m/s.$$

The corresponding values of maximum velocities in the plume at heights 2.0 and 4.0 m above the floor level, from the table in Fig. 7.78, are 0.54 and 0.42 m/s.

The maximum excessive temperature in the thermal plume, from Table 7.19, is

$$\Delta\theta = 0.329 \times 225^{2/3} \times 2.31^{-5/3} = 3.0°C$$

and

$$\Delta\theta = 0.329 \times 225^{2/3} \times 4.31^{-5/3} = 1.06°C.$$

The corresponding values of maximum excessive temperatures in the plume at heights 2.0 and 4.0 m above the floor level, from the table in Fig. 7.78, are 3.6°C and 1.06°C.

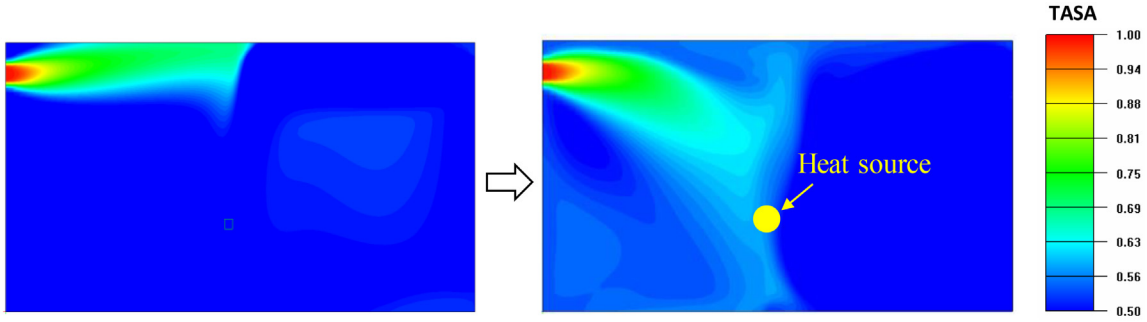


FIGURE 7.79 Impact of the plume on jet effectiveness at the steady state.

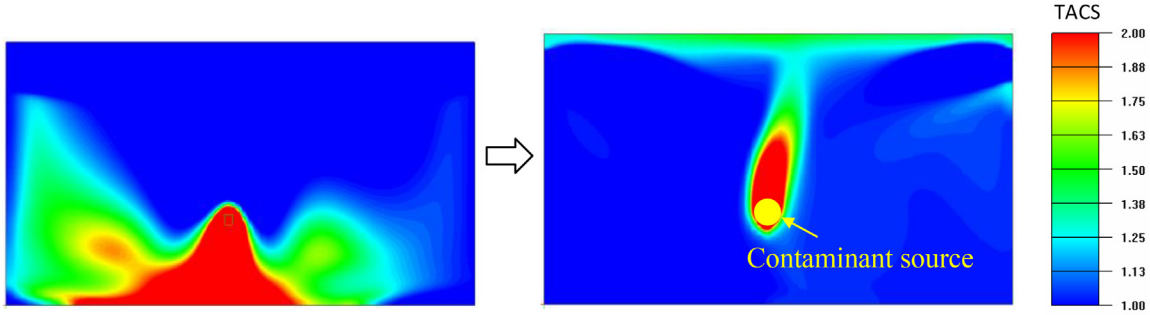


FIGURE 7.80 Impact of the plume on the contaminant transport at the steady state.

The maximum airflow in the thermal plume, from Table 7.19, is

$q_{v,z} = 0.005 \times 225^{1/3} \times 2.31^{5/3} = 0.123\ \text{m}^3\text{s}$ and

$$q_{v,Z} = 0.005 \times 225^{1/3} \times 4.31^{5/3} = 0.347\ \text{m}^3\text{s}.$$

The corresponding values of airflow rates in the plume at heights 2.0 and 4.0 m above the floor level, from the table in Fig. 7.78, are 0.18 and 0.35 m^3/s.

There is a very good correspondence between the analytical and numerical results for temperature and velocity. The airflow rates differ, however, with a factor of 1.46 at a height of 2 m, whereas the correspondence at 4 m is very good.

7.5.6 Effect of plumes on transport of contaminant

The characteristics of the plume are introduced in detail in the earlier sections. The existence of the thermal plume may notably influence the contaminant transport surrounding the plume source, and the impact range of the air jet. The roles of the various plumes in transporting different types of contaminants, as well as influencing the air parameter distributions could be revealed by using indices of TASA and TACS (Section 7.3.3). As a demonstration, Fig. 7.79 shows the impact of the plume on TASA distribution of the air jet in a typical room with dimensions of 6 m (length) × 2.6 m (height) × 3 m (width), where there is no heat source in the left figure, while there is a heat source with an intensity of 1000 W in the right figure.

As affected by the plume, the left air jet inclines downward, and therefore has a greater effect on the lower areas with respect to the nonplume condition. Fig. 7.80 shows the impact of plume on TACS distribution of the contaminant source, where the contaminant source is at the same position as heat source.

With the effect of the plume, the contaminant is transported upward, and therefore the areas directly above the plume are seriously polluted by the contaminant source (a larger TACS). However, if there is no plume, the contaminant released is transported downward (the left figure in Fig. 7.80) and the areas below the contaminant source are greatly polluted, indicating a significant difference under the impact of the contaminant source.

7.6 Airflow near exhausts

7.6.1 Introduction

The pollutant-capturing efficiency of local ventilation systems depends on hood design, the hood's positioning near the source of contamination, and the exhaust air flow. The type and the size of the hood depend on the type and geometry of the pollution

source and its characteristics. Contaminant movement in the source vicinity is specific to the source type. Pollution sources can be classified (see Section 7.2) as

- a nonbuoyant (diffusion) source,
- a buoyant (heat) source, and
- a dynamic source.

The first type of source is characterized by contaminant diffusion in the room in all directions due to the concentration gradient in all directions (e.g., emission from a painted surface). The emission rate in this case is significantly affected by the intensity of the ambient air turbulence and air velocity. With the second type of source, contaminants move in the space primarily due to heat energy as buoyant plumes over the heated surfaces. The third type of source is characterized by contaminant movement in a space with an air jet (e.g., a linear jet over the tank with push–pull ventilation) or particle flow from a grinding wheel. In some cases these factors influencing contaminant distribution are combined.

The geometry of the contaminant source can be *compact* or *linear*. The source geometry affects the hood geometry: round, rectangular, or slot.

Hoods are either *enclosing* or *nonenclosing*. Enclosing hoods provide better and more economical contaminant control because their exhaust rates and the effects of room air currents are minimal compared with nonenclosing hoods.

For nonenclosing hoods, the airflow rate that allows contaminant capture is called a *target airflow*.[274] The target airflow rate q_0^* is proportional to some characteristic flow rate q that depends on the type of contaminant source:

$$q_0^* = Kq \tag{7.232}$$

where K is a dimensionless coefficient depending on the hood design. q is the characteristic airflow rate depending on the contaminant source.

For a nonenclosing hood with a nonbuoyant contaminant source, the characteristic airflow can be calculated using the following equation:

$$q = V_0 \cdot A_0, \tag{7.233}$$

where V_0 is average air velocity in the hood opening that ensures capture velocity at the point of contaminant release, m/s and A_0 is hood opening area, m^2.

For a buoyant source q can be equal to the airflow in the convective plume at the hood suction cross-section. For a dynamic source q can be equal to the airflow rate in the jet.

An exhaust airflow rate lower than q_0^* results in reduced contaminant capturing effectiveness. An exhaust airflow rate greater than q_0^* results in excessive capturing effectiveness (Fig. 7.81).

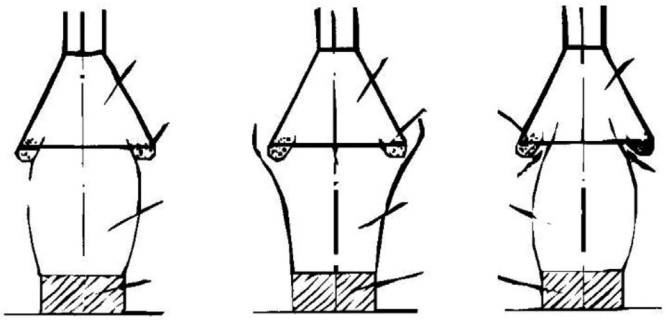


FIGURE 7.81 Hood performance for different exhaust airflow rates. (A) Target airflow rate $q = q^*$. (B) Target airflow rate $q < q^*$. (C) Target airflow rate $q > q^*$.

7.6.2 Air movement near sinks

7.6.2.1 *Theoretical considerations*

Airflow near the hood can be described using the incompressible, irrotational flow (i.e., potential flow) model. The potential flow theory is based on several assumptions.[275] For instance, the fluid is assumed frictionless. Another assumption is that the flow is steady. That means there are no changes in velocity at a given point with respect to time.

The total pressure p_{tot} in the area upstream of the hood remains constant and can be described with the following equation:

$$p_{tot} = p_{st} + p_d = \text{constant} \tag{7.234}$$

where p_{st} is the static pressure, Pa, at any point of the flow; p_d is $\rho v^2/2$ is the dynamic pressure, Pa, at any point of the flow; ρ is air density, kg/m^3; v is air velocity, m/s; and g is gravitational acceleration, 9.8 m/s^2.

At some distance from the hood, the total pressure in the airflow p_{tot} is equal to the ambient air pressure, for example, $p_{tot} = 0$. Thus

$$p_d = \frac{\rho v^2}{2} = -P_{st} \tag{7.235}$$

The abovementioned discussion does not apply to the wakes, with a vortex air movement (Fig. 7.82).

Numerical simulation of hood performance is complex, and results depend on hood design, flow restriction by surrounding surfaces, source strength, and other boundary conditions. Thus most currently used methods of hood design are based on experimental studies and analytical models. According to these models, the exhaust airflow rate is calculated based on the desired capture velocity at a particular location in front of the hood. The capture velocity is the air velocity at the point of contaminant generation upstream of a hood. The concept of capture velocity is primarily used by designers to select a volumetric flow rate for

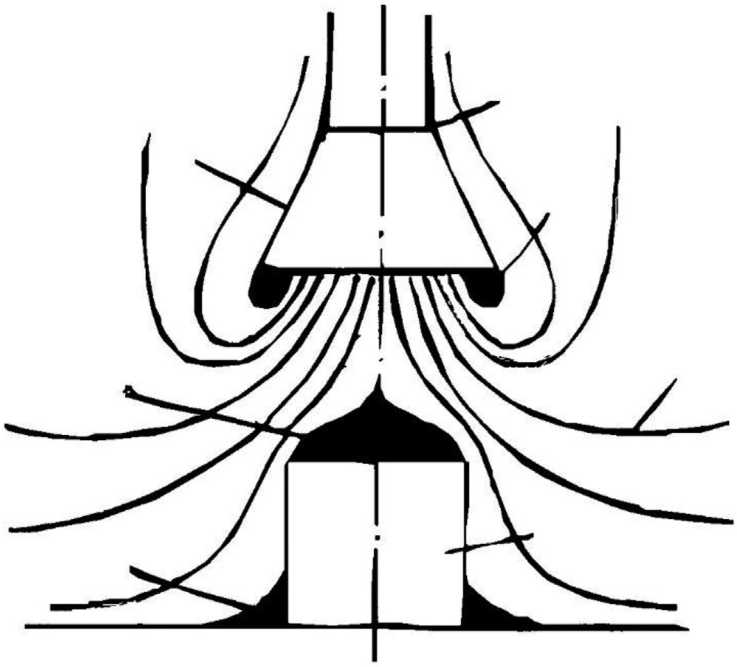


FIGURE 7.82 Airflow in the hood vicinity.

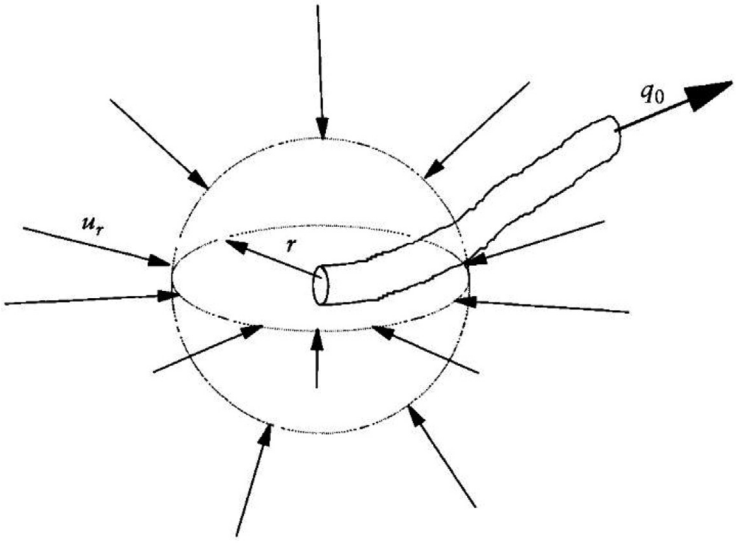


FIGURE 7.83 Airflows toward the point sink.

withdrawing air through a hood in the case of a non-buoyant contaminant source. It is easier to understand the design process for the sink with vanishingly small dimensions—a point or a linear source of suction.

7.6.2.2 *Air movement near a point sink*

The point sink can approximate airflow near a hood with round or square/rectangular shape. The point sink will draw air equally from all directions (Fig. 7.83). The radial velocity v_r (m/s) at a distance r (m) from the sink can be calculated as a volume rate of exhaust airflow q (m^3/s) divided by the surface area of an imaginary sphere of radius r:

$$v_r = \frac{q}{4\pi r^2}. \tag{7.236}$$

Restriction of the airflow by surfaces decreases the area through which the air flows toward the sink, which results in increased radial velocity. For cases with a restricted airflow created by the sink, Eq. (7.236) can be modified to

$$v_r = \frac{q}{\alpha r^2} \tag{7.237}$$

where α is in radians. Values for some typical airflow restrictions are listed in Table 7.21.

TABLE 7.21 Values of α (rad) for some typical point sink locations.

Type of airflow restriction	α (rad)
Unrestricted airflow	4π
Sink within the infinite surface	2π
Sink in the vertex of the dihedral angle with the right angle (90 degrees) of deflection	p
Sink in the vertex of the trihedral angle with the right angle (90 degrees) of deflection in all directions	$\pi/2$
Sink in the vertex of the dihedral angle with the angle Φ, rad, of deflection	2φ
Sink in the vertex of the cone with an angle of deflection Φ (rad)	$2\pi(1-\cos\Phi/2)$

Equations for the inflow velocity (v_r) and the corresponding capture distance (r_c) are listed in Table 7.22 for the most common point sink locations.

7.6.2.3 Air movement near a linear sink

A linear sink will create a two-dimensional airflow. The radial velocity v_r (m/s) at a distance r (m) from the sink is calculated as a volume rate of q(m^3/s) per meter of linear sink length divided by the surface area of an imaginary cylinder of radius r:

$$v_r = \frac{q}{2\pi r} \tag{7.238}$$

The effect of restricting surfaces on the flow created by the linear sink will be similar to that described for the point source. The equations for the inflow velocity (v_r) and the corresponding capture distance (r_c) for some typical situations are listed in Table 7.23.

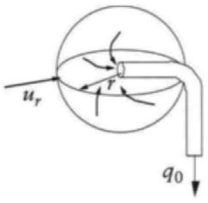
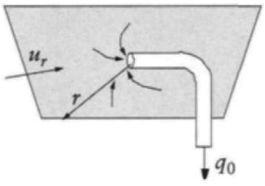
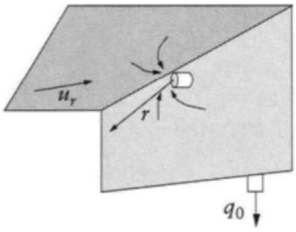
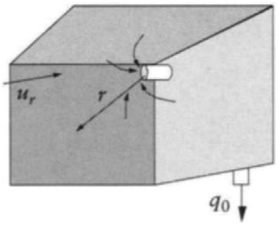
7.6.2.4 Air movement near sinks with finite dimensions

Realistic exhausts used to capture contaminants are complex, varying in their geometry and size. In many cases, the airflow rate ensuring a desired capture velocity at a particular location can be obtained only from empirical studies. Air velocities in front of the hood suction opening depend on the exhaust airflow rate, the geometry of the hood, and the surfaces comprising the suction zone. Studies have established the principle of similarity of velocity contours (expressed as a percentage of the hood face velocity) for zones with similar geometry.[276]

To describe an airflow in the vicinity of some realistic, finite-dimensional hoods, theoretical considerations that are valid for hypothetical point or linear sinks can be applied.

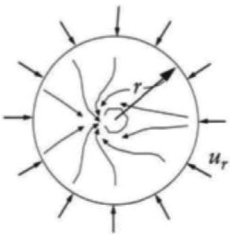
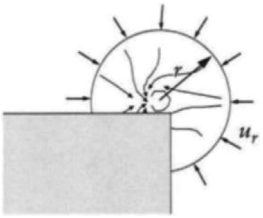
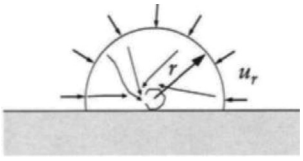
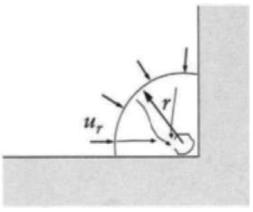
Typically the velocity distribution in the hood face area is not uniform. Wakes formed close to the hood sides, or vena contracta, reduce the "effective suction area" of the hood. Fig. 7.82 shows the wakes in the suction area of an unrestricted rectangular duct. The suction occurs only in the central part of the duct, which constitutes approximately 75% of the duct width.[277] Within the "effective suction width" airflow velocities are relatively uniform, with a maximum air velocity v_m in the center equal to $1.29v_0$, where v_0 is the average velocity calculated based on the total duct width. The velocity distribution in the suction area of

TABLE 7.22 Inflow velocity and capture distance for some common locations of a point sink.

Schematic of airflow restriction	Inflow velocity, v_r	Capture distance, r_c
 Unrestricted airflow	$v_r = \frac{q}{4\pi r^2}$ (1)	$r_c = \sqrt{\frac{q}{4\pi v_c}}$ (2)
 Sink within the infinite surface	$v_r = \frac{q}{2\pi r^2}$ (3)	$r_c = \sqrt{\frac{q}{2\pi v_c}}$ (4)
 Sink in the vertex of the dihedral angle with the right angle (90 degrees) of deflection	$v_r = \frac{q}{\pi r^2}$ (5)	$r_c = \sqrt{\frac{q}{\pi v_c}}$ (6)
 Sink in the vertex of the trihedral angle with the right angle (90 degrees) of deflection in all directions	$v_r = \frac{2q}{\pi r^2}$ (7)	$r_c = \sqrt{\frac{2q}{\pi v_c}}$ (8)

Ill. courtesy H. Skistad.

TABLE 7.23 Inflow velocity and capture distance for some common locations of a linear sink.

Schematic of airflow restriction	Inflow velocity, v_r	Capture distance, r_c
 Unrestricted flow	$v_r = \frac{q}{2\pi rL}$ (1)	$r_c = \frac{q}{2\pi v_c L}$ (2)
 Sink at the vertex of the dihedral angle with the right angle of deflection (270 degrees)	$v_r = \frac{2q}{3\pi rL}$ (3)	$r_c = \frac{2q}{3\pi v_c L}$ (4)
 Sink within the infinite surface	$v_r = \frac{q}{\pi rL}$ (5)	$r_c = \frac{q}{\pi v_c L}$ (6)
 Sink at the vertex of the dihedral angle with the right angle of deflection (90 degrees)	$v_r = \frac{2q}{\pi rL}$ (7)	$r_c = \frac{2q}{\pi v_c L}$ (8)

Ill. courtesy H. Skistad.

an unrestricted round duct is similar, with an "effective suction diameter" $D_e = 0.81D$ and a maximum velocity $V_m \approx 1.1v_0$.

The size of these wakes and the velocity uniformity level depend on the hood design and the airflow pattern in close proximity to the hood face.

Fig. 7.84 shows the approximate relation between the wake size and the angle of cone deflection for the typical hood.[277] Wake size increases ("effective suction area" decreases) with an increase in the angle of the hood deflection. Thus it can be recommended that the value for hood deflection angle not exceed $\pi/4$.

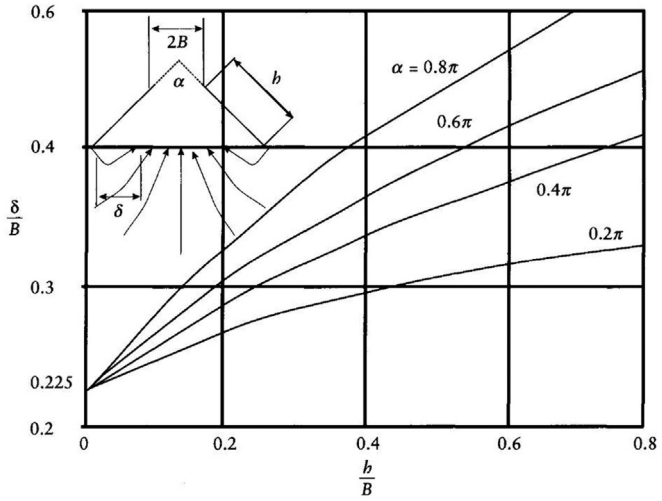


FIGURE 7.84 Influence of hood configuration on hood entrance wake size, δ.

Extensive review of equations for centerline velocities in flows in the vicinity of realistic hoods resulting from experimental and theoretical studies was performed by Braconnier.[278] This review shows certain inconsistencies in equations available from the technical literature due to effects of parameters related to opening (shape, length-to-width ratio, presence of a flange) and the opening location (in an open space or limited by surfaces). The summary of equations from this review complemented by information from Posokhin[277] is presented in Tables 7.24 and 7.25.

Comparison of the relative velocity change in the airflow created by a hood with a finite face area and by a point source is graphically illustrated in Fig. 7.85. At a distance greater than $X/R = 1$, the velocities induced by a realistic hood and by a point source are practically equal. This means that in some cases airflow in front of realistic hoods can be described using the simplified point source equations.

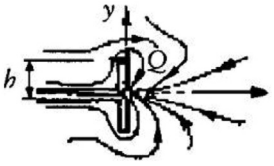
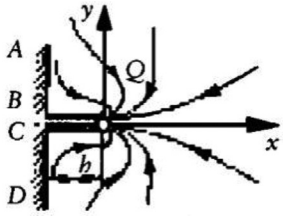
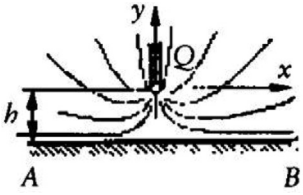
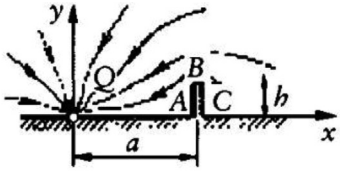
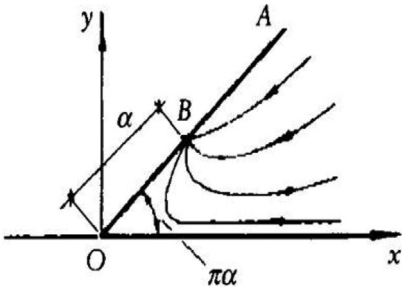
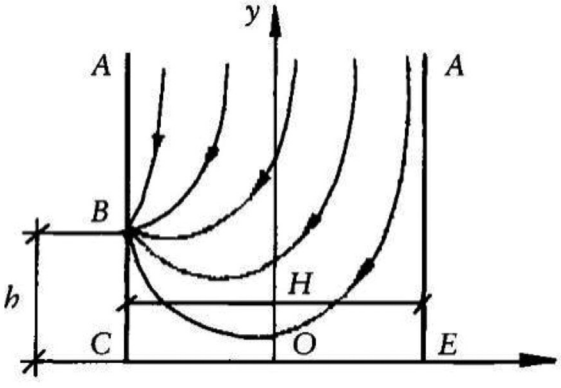
7.7 Air curtains

7.7.1 Introduction

Air curtains are local ventilation devices that are used in industrial buildings to reduce leakage of airflow through apertures in building enclosures and process equipment. Their operation is based on the damping effect of air jets that are supplied into the area of the open aperture. The main advantages of air curtains include

- improvement in working conditions near open apertures,
- reduction of heat (cold) consumption and electrical energy for heating (cooling) of buildings and process equipment,

TABLE 7.24 Inflow velocity for some other common locations of a linear sink.

Schematic of airflow restriction	Inflow velocity, v
 Sink with flanges	$V_{0X} = \frac{Q}{2\pi} \frac{X}{X^2 + h^2 - h\sqrt{X^2 + h^2}}$
 Sink set forth from the infinite surface	$v_{AB} = -v_{CD} = -\frac{Q}{\pi} \frac{Y}{Y^2 + h^2}$ $v_{0X} = \frac{Q}{\pi} \frac{X - h}{X^2 + 2Xh}$
 Sink facing the infinite surface	$v_{AB} = -\frac{Q}{\pi} \frac{X}{X^2 + h^2}$ $v_{0C} = -\frac{Q}{\pi} \frac{(h + Y)}{(2h + Y)}, Y < 0$
 Sink within the infinite surface with an airflow obstruction from one side	$v_{OA} = \frac{Q}{\pi} \frac{X - z}{\sqrt{(a^2 + h^2)\left[(X-a)^2 + h^2\right] - (X-a)^2 - h^2}}$ $v_{CD} = \frac{Q}{\pi} \frac{a - X}{\sqrt{(a^2 + h^2)(X-a)^2 - (z - X^2) + h^2}}$
 Sink on one of the surfaces of the dihedral angle $\alpha\pi$ (rad)	$v_{0X} = -\frac{Q}{\alpha\pi} \frac{x\left(\frac{1-\alpha}{\alpha}\right)}{x^{1/\alpha} + a^{1/\alpha}}$ $v_{OA} = -\frac{Q}{\alpha\pi} \frac{\left(\frac{x}{\cos\frac{\pi}{\alpha}}\right)^{\left(\frac{1-\alpha}{\alpha}\right)}}{\left(\frac{x}{\cos\pi\alpha}\right)^{1/\alpha}} - a^{1/\alpha}$
 Linear exhaust from confined space (a)	$v_{AC} = \frac{Q}{H} \frac{\sin h(\pi\overline{Y})}{\cos h(\pi h) - \cos h(\pi\overline{Y})}, \overline{h} = \frac{h}{H}$

(*Continued*)

TABLE 7.24 (Continued)

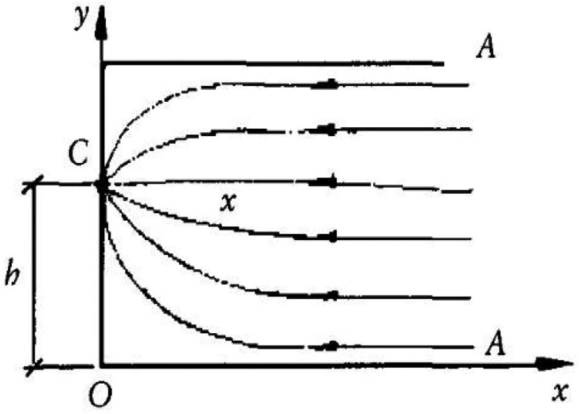
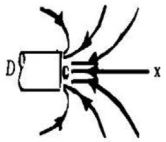
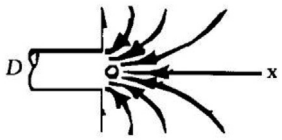
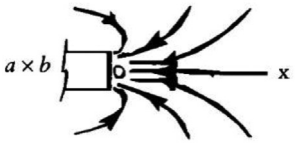
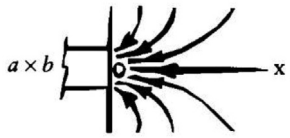
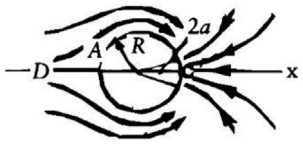
Schematic of airflow restriction		Inflow velocity, v
	Linear exhaust from confined space (b)	$v_{OA} = -\frac{Q}{2H}\frac{\sin h(\pi\overline{X})}{\cos h^2\left(\frac{\pi}{2}\overline{X}\right) - \cos^2\left(\frac{\pi}{2}\overline{h}\right)}$ $v_{BA} = \frac{Q}{2H}\frac{\sin h(\pi\overline{X})}{\cos h^2\left(\frac{\pi}{2}\overline{X}\right) + \cos^2\left(\frac{\pi}{2}\overline{h}\right)}$ $v_{OB} = -\frac{Q}{2H}\frac{\sin(\pi\overline{Y})}{\cos^2\left(\frac{\pi}{2}\overline{Y}\right) - \cos^2\left(\frac{\pi}{2}\overline{h}\right)}$

TABLE 7.25 Centerline air velocities induced by sinks with finite dimensions.

Hood type	Schematic	Equation	Applicable range	References
Round free-standing hood, unflanged		$\frac{v_X}{v_0} = \left(1 + \frac{10x^2}{A}\right)^{-1}$	$x \le 1.7\sqrt{A}$ $\alpha \le 30$ degrees	Dalla Valle[276]
Round free-standing hood, flanged		$\frac{v_X}{v_0} = 1.1(0.07)^{-\frac{x}{D}}$ $\frac{v_X}{v_0} = 1(x/D)^{-1.5}$	$0 \le \frac{x}{D} \le 0.5, C \ge D$ $0.5 \le \frac{x}{D} \le 1.5, C \ge D$	Garrison[279]
Rectangular free-standing hood, unflanged		$\frac{v_X}{v_0} = \left(0.93 + 8.58\alpha_F^2\right)^{-1}$ $\alpha_F = \left(x/\sqrt{A}\right)(\alpha/b)^{\beta F}$ $\beta_F = 0.2\left(x/\sqrt{A}\right)^{-1.3}$	$1 \le \frac{a}{b} \le 16$ $0.05 \le \frac{x}{\sqrt{A}} \le 3$ $a \le 30$ degrees	Fletcher[280]
Rectangular free-standing hood, flanged		$\left[\frac{v_X}{v_0} = 1 - \frac{2}{\pi}atan\left(\frac{2x\sqrt{x^2 + a^2 + b^2}}{ab}\right)\right]$	$1 \le \frac{a}{b} \le 16$ $0.05 \le \frac{x}{\sqrt{A}} \le 3$ $\frac{C}{\sqrt{A}} \ge 1$	Tyaglo and Shepelev[281]
Slot in the pipe wall		$\frac{v_{ez}}{v_p} = \frac{v_{AD}}{v_0} = \frac{2R}{R\pi}atan\left[\frac{x+R}{x-R}\tan\left(\frac{\alpha}{2}\right)\right]$	$\lvert x\rvert \ge R$	Posokhin[277]

- reduction of heat loss in the building by using overheated air from an upper zone of the room, and
- reduced loss of usable working area near gates due to the ingress of outdoor air into the building

Traditional air curtains, which utilize only indoor air heated in the curtain heaters, are not always economical (due to considerable thermal energy consumption). Reduction of heat consumption is achieved by curtains that utilize unheated indoor or outdoor air, and also combined air curtains, which heat only part

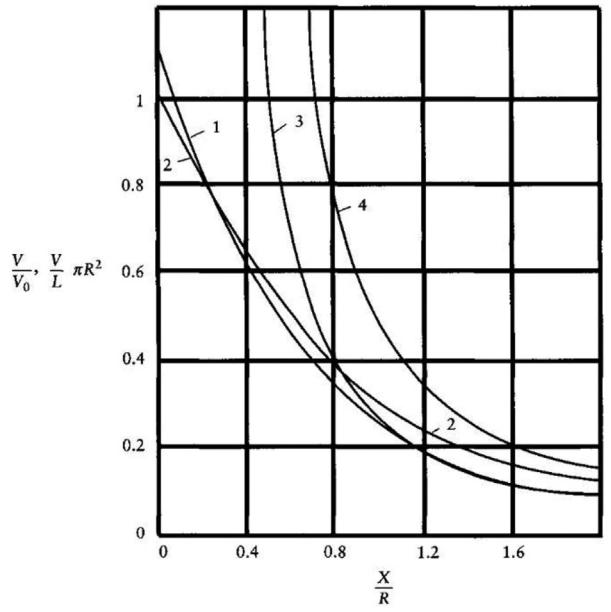


FIGURE 7.85 Centerline velocity, V, and decay in the flow created by exhausts with finite dimensions and point sinks. 1, round free-standing pipe; 2, round opening in an infinite surface; 3, unrestricted point sink, $V = q/\pi^2$; and 4, point sink in an infinite surface, $V = 2q/\pi r^2$.

of the supplied air. Air curtains that use unheated air conserve 30%–70% of thermal energy.

7.7.2 Types of air curtains

According to aerodynamic pattern, the following types of air curtains have been designed:

- air curtains with heated indoor air,
- air curtains with unheated indoor air,
- air curtains with unheated outdoor air, and
- combined air curtains with indoor air.

The abovementioned air curtain types are installed in outer apertures of gates in heated rooms and also in unheated rooms where a standard temperature should be maintained in the working zone. They are designed to prevent the ingress of outdoor air during the cold season of the year.

7.7.2.1 Air curtains with heated indoor air

The traditional design pattern (to be described in Section 7.7.5) is recommended for air curtains with heated indoor air. The air curtain may be double sided with horizontal supply (Fig. 7.86A) or it may be single sided (Fig. 7.86B) with horizontal or vertical supply. In all cases, the air curtain is a flat jet discharged at an angle toward the pressure side of the opening.

Air curtains with heated indoor air are used for relatively small gates (up to 3.6×3.6) in genial climate and in buildings without skylights, and also in case constructions prevent the installation of combined air curtains (e.g., not enough space at the gate). Air curtains with heated indoor air guarantee the necessary temperature of the air mixture entering through the gate, but they also consume a relatively large amount of thermal energy.

7.7.2.2 Air curtains with unheated indoor air

Air curtains with unheated indoor air have a similar air supply arrangement. Only the design of the air intake duct might differ; it may be extended to take in warm air, for example, from the upper overheated zone of the room. Air curtains with unheated indoor air are recommended in case the standard temperature of the air coming in through the aperture of the open gate can be maintained without heating the air in the curtain, as in the following cases:

- Rooms with an overheated upper zone of over $2 \times C$ (if it is possible to use the air from the upper overheated zone of the room),
- rooms with excess heat (if it is technically possible and reasonable to utilize it), and
- rooms with a low standard air temperature in the working zone near the gates (8°C and below).

Air curtains with unheated indoor air are more restricted in application than air curtains with heated indoor air, but they do not require the installation of air heaters in the curtain devices.

7.7.2.3 Air curtains with unheated outdoor air

Air curtains with unheated outdoor air are recommended with a one-sided lateral supply of outdoor air in the form of a flat jet at an angle to the plane surface of the gate aperture toward the outdoor air (Fig. 7.86B). In this case the outer surface of the enclosure should not have any obstructions on the opposite side that might prevent the jet from flowing along the surface of the gate. This free length should be no less than the approximate width of the gate.

Air curtains with unheated outdoor air find application in unheated rooms and also in case there are no strict hygiene requirements for the microclimate in the gate zone (no working places near to the gate, e.g.). Air curtains with unheated outdoor air should not be used in double-wing gates, in humid rooms, and in cases of transport with an open driver's cab through the gate.

Air curtains with unheated outdoor air do not provide for the necessary microclimate in the immediate vicinity of the gate, but since no thermal energy is used, they reduce heat losses from the room.

7.7.2.4 Combined air curtains with indoor air

Combined air curtains with indoor air are recommended with a double-sided supply of heated and

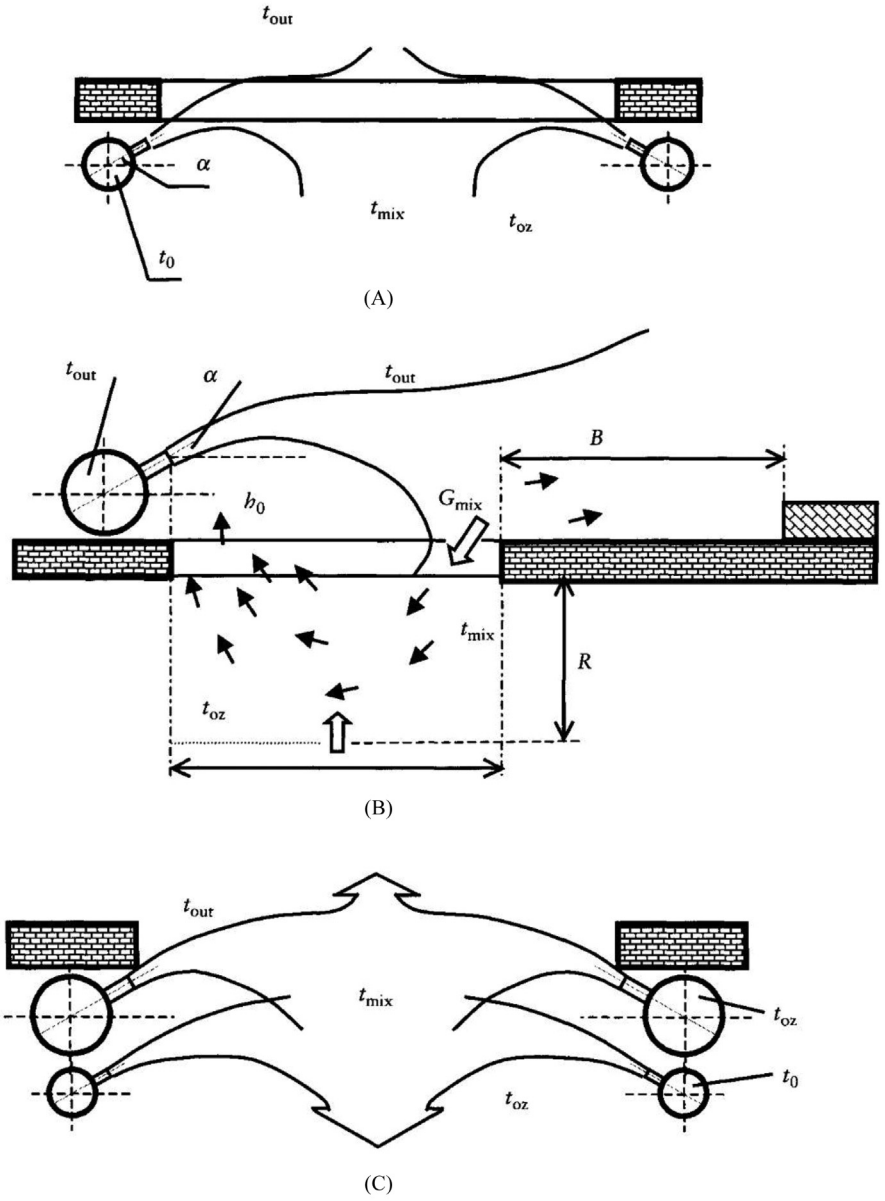


FIGURE 7.86 Schematic of shutter-type air curtains. (A) Heated or unheated air supply. (B) Outdoor air supply. (C) Combined air curtain.

unheated air fed to the room in the form of plane jets at an angle of 15 degrees to one another (Fig. 7.86C).

Delivery ducts for unheated (outdoor) air are located in the room right against the gate. Delivery ducts for air heated in the air curtain device (indoor) are installed inside the room behind the outdoor delivery ducts, with a clearance for exhaust air between the outdoor and indoor delivery ducts of the air curtain.

Recommended applications of combined air curtains with indoor air include

- Severe climate areas,
- gates of 3.6×3.6 m or larger, and
- locations with several gates (three or more).

Combined air curtains with indoor air cut down expenses due to rational utilization of energy by the heated jets. Thermal energy savings are 25%–60% depending on the dimensions of the gate and climatic region, and the reduction in expenditures is 30%–70%.

7.7.3 Applications of air curtains

Through years of development, air curtain has been widely used in many applications. In addition to classic applications, some new applications have recently been proposed.

7.7.3.1 Air curtains for cooled rooms

Air curtains for cooled rooms are recommended with a one-sided supply of outdoor air from above in

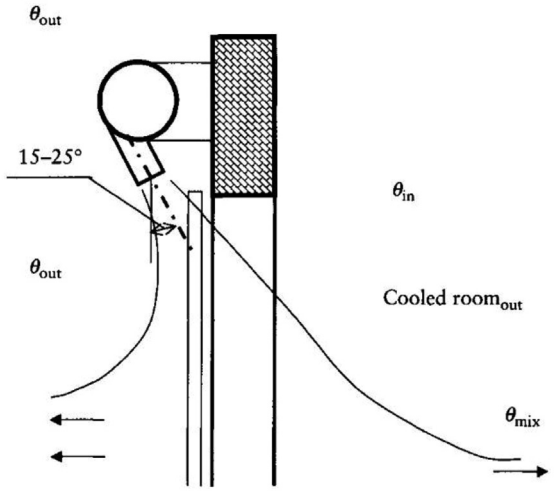


FIGURE 7.87 Schematic of air curtains for cooled rooms.

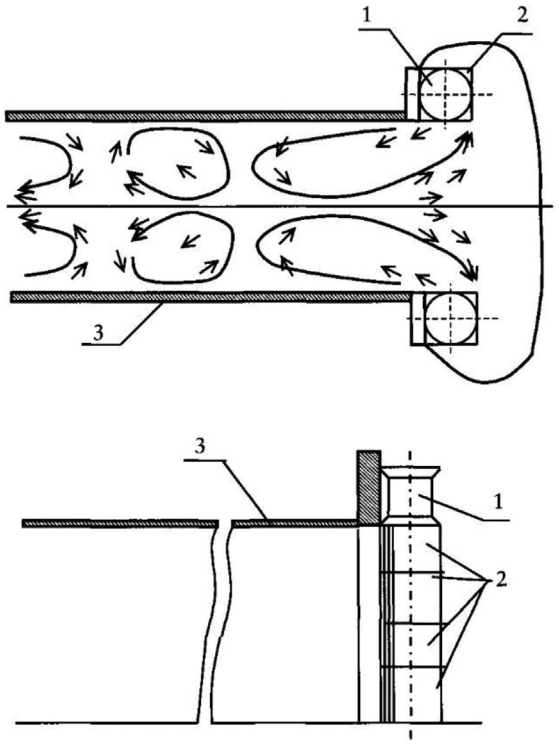


FIGURE 7.88 Air curtain for a medium-size gate with a lobby: 1, fan; 2, distribution duct; and 3, lobby.

the form of a flat jet at an angle to the plane surface of the gate toward the cold indoor air (Fig. 7.87).

Air curtains for cooled rooms are used in all types of rooms with artificial cooling of air: vegetable stores, cold rooms, freezers, air-conditioned plants and storehouses, etc. Installation of air curtains for cooled rooms considerably reduces cold losses through the open gate and also reduces undesirable variations in temperature in the gate zone inside and outside the cooled room.

7.7.3.2 Air curtains for gates with long passages

Air curtains for gates with long passages use a pattern of air supply "curtains in a channel" (Fig. 7.88). The operating principle is based on the complete conversion of the jet impulse to counterpressure that prevents outdoor air from bursting into the room.

The air is fed against the bursting airflow or at a minor angle. The jet of the curtain developing in the channel is dampened, and from the end of the channel it turns to the opposite direction. This creates a closed circulation proof against outside effects.

The length of the passage should be chosen so that it prevents the air from being released outside. To reduce the length of the passage, the air is normally supplied in the form of incomplete spray jets with nozzles of special design. Another alternative is to feed outdoor air into the passage (Fig. 7.89).

7.7.3.3 Air curtains for process equipment

Air curtains for process equipment are designed to prevent the ingress of toxic constituents (gases, aerosols, and heat flows) into the room through open apertures of the process equipment. They also support necessary parameters of technological processes in the plant. Processes running in the technological equipment are classified as isothermal (e.g., spray-painting chambers) and nonisothermal (e.g., heat dryers). In isothermal processes one uses damper-type air curtains for process equipment in combination with an exhaust system (Fig. 7.90). In nonisothermal processes one uses a circulation system based on the "curtains in a channel" operating principle (Fig. 7.91).

7.7.3.4 Air curtains for tunnels

Taking appropriate methods to confine the spread of the smoke and toxic gases in case of a fire is a serious concern for smoke management in tunnels. Air curtain is an alternative approach to restrict the fire-induced smoke and carbon monoxide to spread along channels. Both one air curtain[282] and two air curtains[283] can be designed. The use of a single air curtain can prevent smoke from propagating in one direction only. For two air curtains, the jet patterns of each air curtain can be single-jet without recirculation, single-jet with full recirculation, double-jet without recirculation, and double-jet with partial recirculation.[283]

7.7.3.5 Air curtains for relics preservation

As an important carrier of human culture, the relics possess high scientific and historical values. However, the relics are far from well preserved in the archaeology museum. The air pollution and fluctuation of environmental parameters are the main causes of desiccation cracking, efflorescence of relics. To preserve the immobile relics and earthen sites in

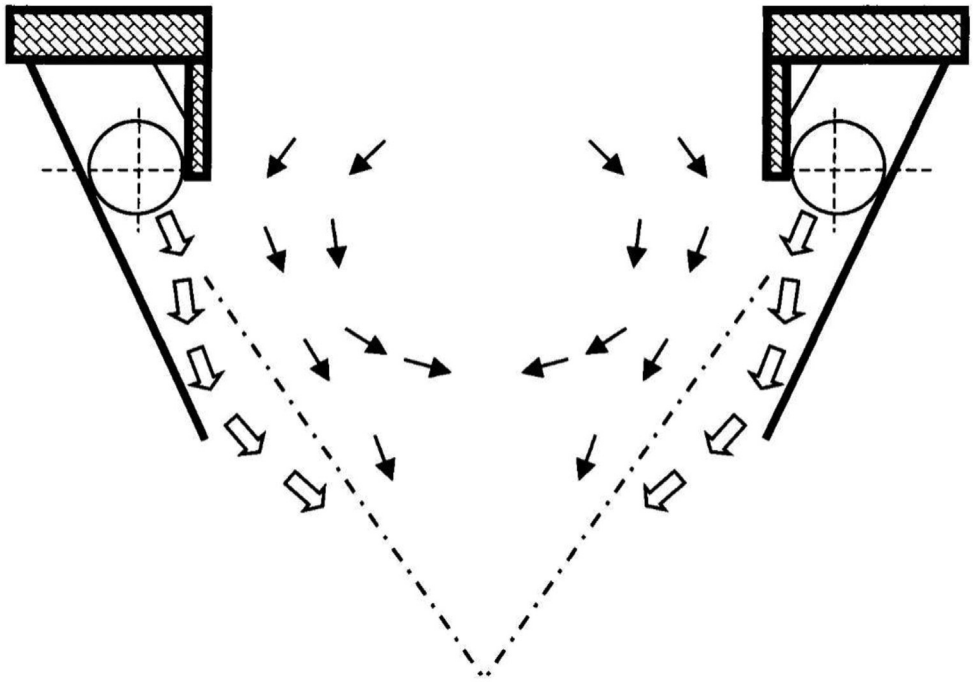


FIGURE 7.89 Air curtain with a lobby and outside air supply.

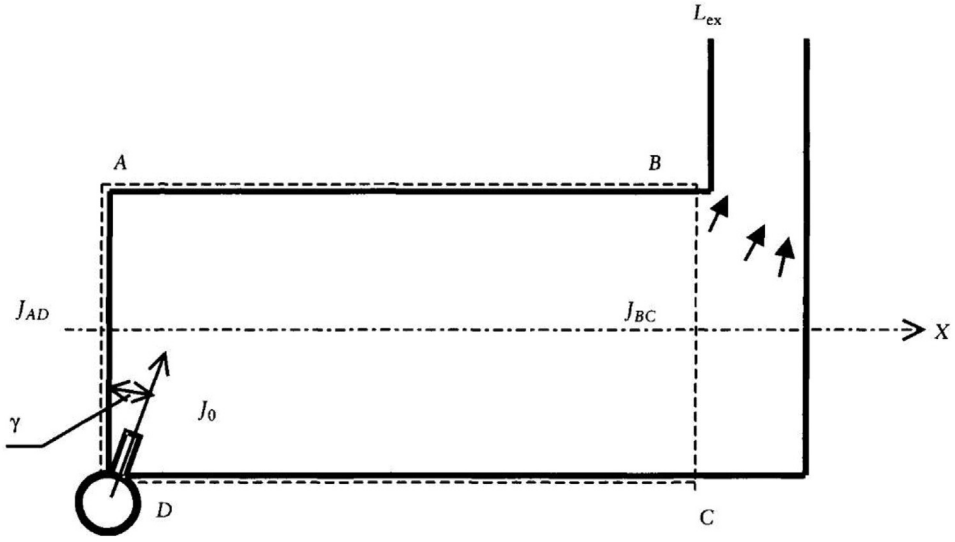


FIGURE 7.90 Schematic of air curtains for process equipment: isothermal processes.

archaeology museum, an air curtain system with jet blown horizontally could provide an air barrier at the top of the funerary pit to prevent the pollutants from entering the funerary pit.[284] The air curtain separates the local environment in funerary pit from the large-space exhibition hall. The environment in the funerary pit can be greatly preserved by the air curtain, and the temperature difference between the top and bottom of funerary pit could be reduced significantly.

7.7.3.6 Air curtains for aerodynamic noise reduction

During a landing process for aircraft, when engines are operating at low thrust, the noise from the landing gear and the wheel bay cavity contributes substantially and can often dominate the overall noise signature of modern aircraft. An air curtain, which is really a planar jet in crossflow,[285] is a noise reduction technology for aircraft landing gear. Both the single planar jet and dual

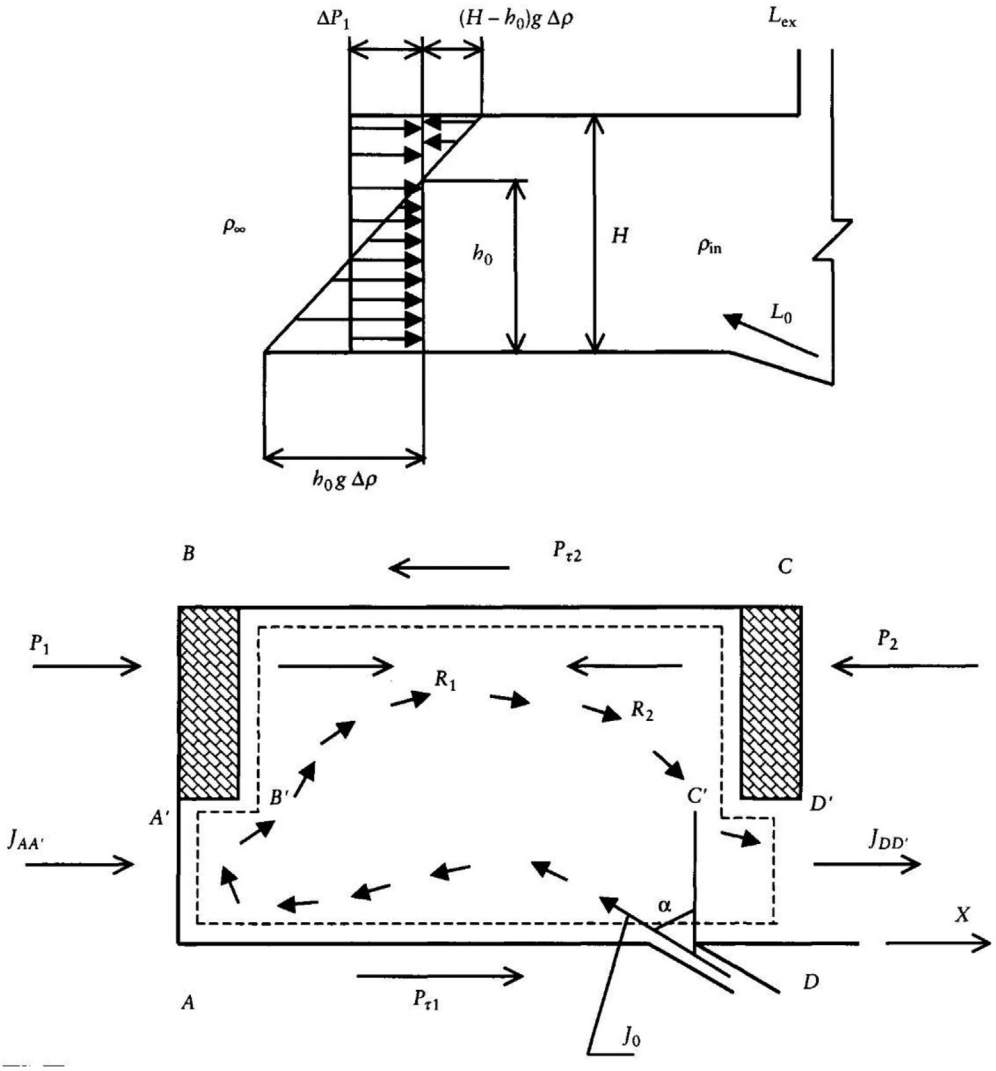


FIGURE 7.91 Schematic of air curtains for process equipment: nonisothermal processes.

planar jet can be designed.[286] Single-jet air curtains have shown significant promise as a low noise technology but can introduce additional noise sources such as lip and mixing noise. Dual-jet planar air curtains are demonstrated to be able to successfully remove aerodynamic noise radiated from tandem rods in a crossflow. Compared with single jet air curtains, the secondary upstream air curtain allows the primary air curtain to provide the same shielding height but at a lower velocity. Dual-jet air curtains achieve the same shielding height with a significantly lower overall system noise.[286]

7.7.3.7 *Air curtains for cleanrooms*

Contamination control of cleanrooms is a vital issue to assure the product reliability and personnel safety associated with the fabrication process in a variety of high-precision industries, for instance thin film transistor-liquid crystal display, semiconductor, microelectronics, and biotechnology. Pollutant organic gases, unintentionally emitted from certain solids or liquids during operations, may also cause serious safety problems in cleanrooms. Main efforts have been made to realize pollution management in cleanrooms by delivering the cleaned air to remove hazardous VOC. In case of emergency response, arranging air curtains in a cleanroom is beneficial for preventing the contaminated species from dispersion for personnel safety concern.[287] The air curtain can be installed right above the doorway. The installation height of the air curtain is a key factor, which significantly influences the sealing efficiency. In addition, air curtain can be used to protect the process equipment in cleanroom. For instance, an air curtain can be used to prevent moisture entering a purged front-opening unified-pod (a container for the storage of wafers) when its door is opened in a minienvironment.[288]

7.7.3.8 Air curtains for antiinsect barrier

Vector-borne diseases are of high concern for human, animal and plant health. In humans, such diseases are often transmitted by flying insects. In pastry-making, meat-processing and other food industry structures, insects such as flies, honey bees, wasps, and hornets are often attracted by food products from which they draw nutrients and on which they can lay eggs or deposit feces. By contact with meat and food products, insects can also transmit pathogens and parasites that are responsible for food-borne diseases. The best way to combat insect infestation is to prevent insects from entering buildings rather than killing them with electronic insect-killing devices, pesticides, or traps. Air curtains may represent an efficient, alternative approach to reduce or avoid the presence of flying insects in buildings, thus not only increasing food safety and hygiene but also decreasing the incidence of vector-borne diseases. An air curtain placed above the doorway of a building may prevent a strong flyer with high kinetic energy, such as the honey bee, from entering a building.[289]

7.7.4 Principle of calculation

There are currently two approaches to the design of damper-type air heat curtains: cinematic and dynamic. In the cinematic method the airflow in the aperture is understood to be the result of interaction of the air curtain jet and the incident flow. Some of the cinematic methods that were developed[290–295] did not apply the laws of conservation of the impulse and mechanical energy. These methods did not correspond satisfactorily to test results and were not developed further. In these cases the determination of the jet trajectory does not take into account the effect of the enclosures and the interaction of the jets, and the division of airflows between the room and the outer atmosphere is performed with an arbitrary geometrical construction. The above mentioned facts lead to divergence of design results and existing test results as to both the release speed and the initial temperature of the air curtain.[296,297]

The dynamic method sees airflow as a result of the effect of differential pressure on the jet in the gate aperture. Dynamic methods do consider the law of conservation of the impulse in the isolated circuit. According to the type of isolation of the circuit, dynamic methods are divided into methods that determine the trajectory of the jet[298–303] and methods that determine the integral flow rate of air through the aperture.[296,297,304,305] This method, due to consideration of the aperture and surrounding enclosures in the design circuit and application of the law of conservation of mechanical energy, achieved for the design of the air release speed a dependence that corresponds well to test results.[296,297] Figs. 7.90–7.92 illustrate examples of isolated circuit design.

In the following the dynamic method of air curtain design is applied (see Fig. 7.92). The basic dependency is illustrated for a one-sided air curtain that is supplied at angle a and developed on the plane surface XOY. Since the jet of the air curtain is bent by the effect of differential pressure from outside (P_{out}) and inside (P_{in}) the building, the jet of the air curtain flows to the opposite side of the aperture and splits into two parts. After the division, one part of the jet flows along the outer surface of the enclosure and the other one enters the room at an angle β to the plane surface of the aperture. We isolate the circuit $ABVGDKMN$. Surfaces AB,

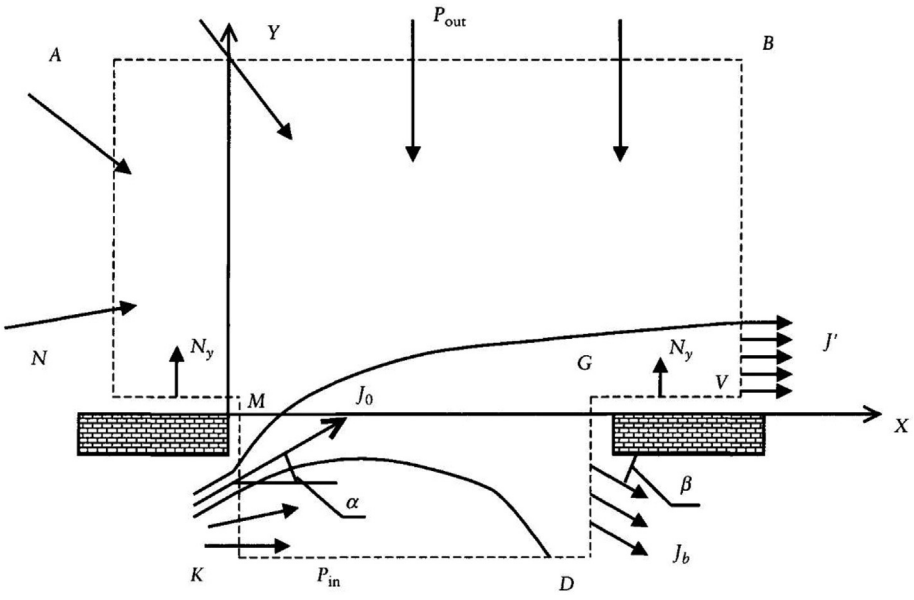


FIGURE 7.92 Theoretical model of air curtains.

AN, and *BV* are led at a distance from the gate where the speed of air flowing to the jet is near zero and the quantity of air impulse coming in through the surface *NABV* may be disregarded.

The equation of momentum of the isolated circuit in projection at the *Y*-axis is

$$J_0 \sin\alpha + J_b \sin\beta = \Delta P A_0 + \left(P_{\text{in}} - N_y\right)(A_{AB} - A_0), \quad (7.239)$$

$$\Delta P = P_{\text{out}} - P_{\text{in}}(\text{Pa}) \quad (7.240)$$

where J_0 and J_b are impulses of airflows supplied by the air curtain that flow into and out of the room (N). A_{AB} and A_0 are the areas of the surface *AB* and the gate aperture (m^2). N_y is the average value of reactive pressure in the scope of plane surfaces *VG* and *MN* (Pa).

The concept of the dynamic efficiency of the air curtain, *E* is derived, which is equivalent to the ratio between aerostatic pressure forces affecting the gate aperture and the doubled initial impulse of the air curtain jets:

$$E = \Delta P A_0 / 2 J_0. \quad (7.241)$$

The factor *E* shows the efficiency of utilization of the initial impulse of the curtain jets. Using this form to represent factor *E* allows us to estimate the efficiency of the curtain in fractions of the unit.

The dependence for factor *E* results from the joint solution of the Eqs. (7.239) and (7.241) as follows:

$$E = 0.5\left(\sin\alpha + R_y\right). \quad (7.242)$$

The quantity R_y is a function of geometrical parameters and is determined experimentally. Similar dependencies for the factor *E* have been obtained for all patterns of the air curtains introduced earlier.

The initial speed of the air supplied by the air curtain is determined according to the following universal dependence, which results from the joint solution of Eqs. (7.239)–(7.242):

$$v_0 = \left(\Delta P A_0 / 2\beta_0 \rho A_S E\right)^{1/2}(\text{m/s}), \quad (7.243)$$

where β_0 is the Boussinesq factor (1.05–1.1). ρ is the density of air supplied by the air curtain (kg/m^3). A_S is the total area of the outlet apertures of the air curtain (m^2).

The value of the ratio $f = A_0/A_S$ is recommended to be taken based on the following technical and economical considerations:

- For air curtains with heated indoor air, unheated indoor air, or combined air curtains with indoor air: $f = 10–20$.
- For air curtains with unheated outdoor air, air curtains for cooled rooms, air curtains with long passages, or air curtains for process equipment: $f = 20–40$.

The average values of factor *E* in the recommended range of *f* are shown in Table 7.26.

TABLE 7.26 Factor of dynamic efficiency, *E*.

	Type of air curtain		
sin α	With heated indoor air, with unheated indoor air, or combined curtains with indoor air	With unheated outdoor air or for cooled rooms	For gates with a long passage
0.1	0.1	0.15	–
0.2	0.15	0.2	–
0.3	0.2	0.25	–
0.4	0.25	0.3	–
0.5	0.3	0.4	–
0.6	0.35	–	–
0.9–1.0	–	–	−1.0

The mass flow of air supplied by the air curtain is

$$G_0 = \rho_0 v_0 A_s(\text{kg/s}). \quad (7.244)$$

The purpose of the thermal design of the air curtain is to find the dependence between the average initial temperature of the jets supplied by the air curtain and the average temperature of the part of damping airflow coming in through the gate aperture. The temperature distribution of the air curtain jet significantly differs from the temperature distribution of the free jet as a result of the different temperatures of the air masses joined to the air curtain jet (Figs. 7.93 and 7.94).

The distribution of excess temperature of the curtain air (in relation to the outdoor air temperature, t_{out}) is found from the conservation of the heat content of the jet as follows:

$$\Phi = \Delta\theta_{o1} N \cosh^{-1} Y/CX + 0.5\Delta\theta_B\left(1 - (\tanh Y)/CX\right)\left(1 - N\cosh^{-1} Y/CX\right), \quad (7.245)$$

where

$$\text{N} = (4/\pi\sqrt{3}Cx)\sqrt{T_\infty}/\sqrt{T_0}\cdot(1/\phi)$$
$$\Delta\theta_{o1} = \theta_0 - \theta_{\text{out}}$$
$$\Delta\theta_B = \theta_{\text{in}} - \theta_{\text{out}}$$

C is an experimental constant ($C \approx 0.1$), *x* is the X/b_o, b_o is the width of the air outlet slot, $\sqrt{T_\infty}/\sqrt{T_0}$ is the correction factor for the nonisothermal jet, T_∞ is the average temperature of the environment (K), $\varphi = 4\sqrt{\zeta}$ is the correction factor for the jet impulse, and ζ is the factor of the local resistance of the air outlet nozzle.

The formula for the determination of the necessary initial temperature of the air curtain jet results from the integration of the equation for the heat content of the jet in the section X_A corresponding to the inlet of the jet into the room in the scope ($-\infty$ $-Y_A$). The

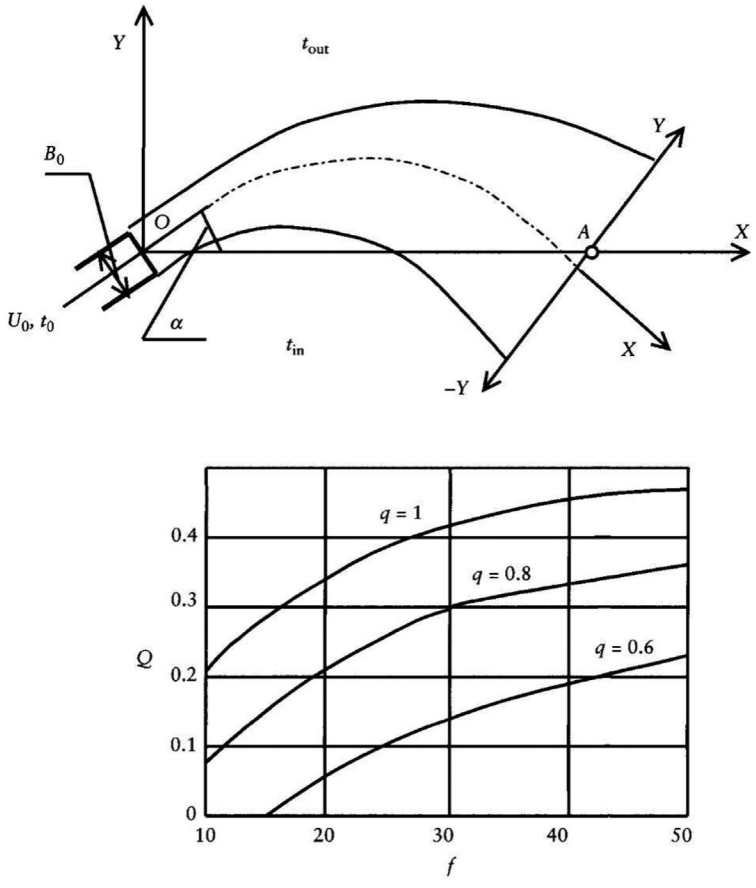


FIGURE 7.93 Heat losses of air curtains.

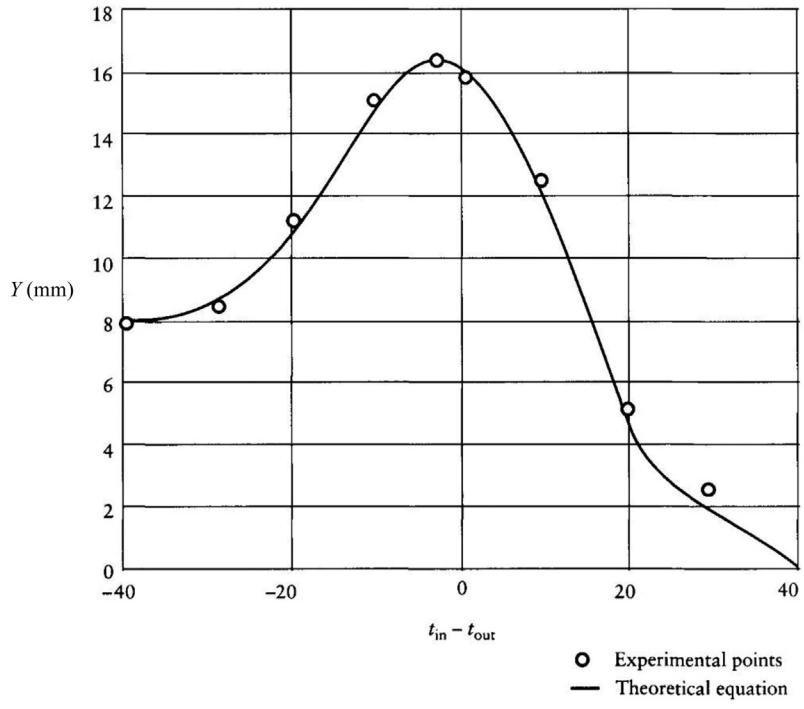


FIGURE 7.94 Air temperature distribution in air curtain jet.

TABLE 7.27 Values of the factors m_1 and m_2.

f	m_1	m_2
10	2	−0.6
15	2.5	−1
20	2.9	−1.3
30	3.5	−1.8
40	4.1	−2.2

ordinate of point A is obtained from the law of conservation of the mass in the section X_A:

$$Y_A = CX \text{ arc tan h} D,$$

where

$$D = [2 - (p_{mix}q/p_0)](1/\phi)(1/\sqrt{T_\infty}/\sqrt{T_0})/(p_{mix}q/p_0)(\sqrt{3}Cx) \quad (7.246)$$

where $q = G_0/G_g$ is the relative mass flow of air through the curtain and G_g is the mass flow of air through the aperture.

Simplified formulas are applied in engineering design. For example, in the case of an air curtain with heated indoor air the necessary temperature of the supplied air may be calculated according to the following formula:

$$\theta_0 = \theta_{out} + m_1(\theta_{mix} - \theta_{out}) + m_2(\theta_{in} - \theta_{out}), \quad (7.247)$$

where θ_{out} and θ_{in} are design temperatures of the outdoor air and room air, respectively (°C), θ_{mix} is the temperature of the air mixture coming through the aperture of the gate, and m_1 and m_2 are factors with average values as shown in Table 7.27 for total damping of the aperture ($q = 1$).

The temperature of air supplied by the air curtain should not exceed 70°C in case the technological process does not require any other temperature. The thermal capacity of the air curtain is determined according to the equation

$$P = c_p G_0(\theta_0 - \theta_H)(\text{W}), \quad (7.248)$$

where c_p is the heat capacity of air, 1004 J/kg °C and θ_H is the discharge temperature of the air curtain.

Parameters for combined air curtains are determined by optimized calculations based on minimal expenses. Usually the share of heated air in a combined air curtain is 20%–40% of the total flow and its temperature is 60°C–70°C.

Besides, a procedure for the technical dimensioning of a vertically upward blowing air curtain was proposed,[306,307] which can be taken as a guide for the users. The flow chart of the technical dimensioning is shown in Fig. 7.95. The detailed information can be found in Refs.[306,307]

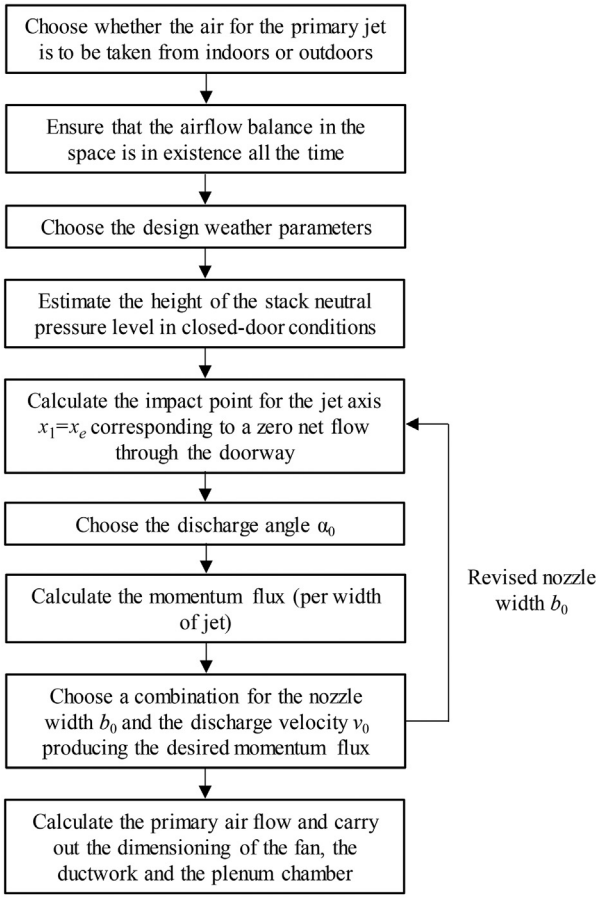


FIGURE 7.95 Technical dimensioning procedure of the vertically upward blowing air curtain.[306,307]

7.7.5 Operation of the air curtain

The automation of an air curtain should provide for the following:

- Starting of the fan when the serviced aperture is opened and when the air temperature near the closed aperture deviates from a set value.
- Stopping of the fan after the serviced aperture is closed and restoration of the air temperature near the closed aperture to a set value.
- Controlling supply air flow in proportion to the square root of the change in the aerostatic pressure difference across the aperture: $q_{m,0} \sim \Delta P^{0.5}$.
- Controlling the supply air temperature (in the case of air curtains with heated air) in proportion to the change in temperature of the outdoor air.

7.7.6 Design of an air curtain device

7.7.6.1 The task

An air curtain is to be used in the doorway of an industrial building to prevent the penetration of cold

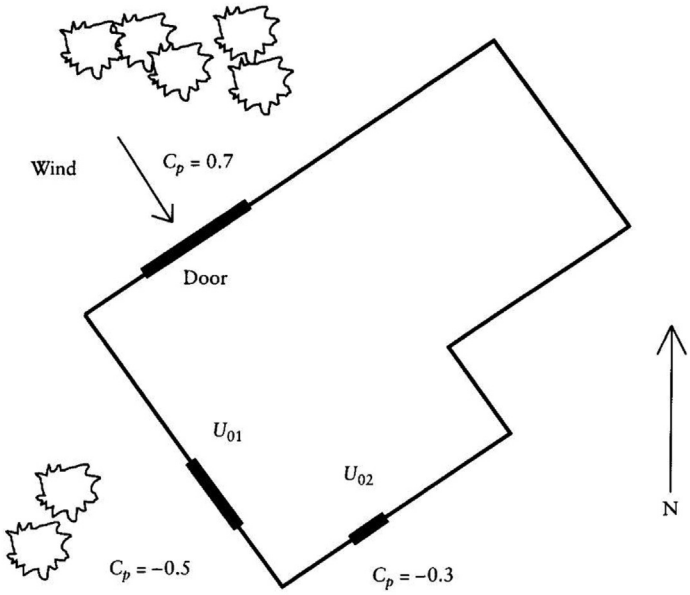


FIGURE 7.96 The object building.

air into the building. The task is to dimension this air curtain device for the following climatic conditions:

Case 1:
- Outdoor air temperature $\theta_{out} = -12°C$.
- Wind velocity $v_w = 0$ m/s.

Case 2:
- Outdoor air temperature $\theta_{out} = -5°C$.
- Wind velocity $v_w = 5$ m/s.

The wind velocities are local values at the height of the doorway, so no additional wind sheltering coefficients or height corrections are necessary. A solution is sought for which both situations above can be handled with the same air curtain device by just changing the fan speed and the heating power of the air curtain.

7.7.6.2 Data

The building (see Fig. 7.96) is 12 m high and has three large openings. The doorway for which the air curtain is designed is on the ground level, has dimensions 4 m (height) × 6 m (width), and is intended primarily for vehicular traffic.

There are two large openings in the upper part of the building: U_{01}, which is 4 m wide × 2 m high, and U_{02}, which is 2 m wide × 1 m high. The lowerlevel of 5 m. Otherwise the building envelope is assumed to be airtight. This information is summarized in Table 7.28.

The indoor air temperature and humidity are assumed to be uniform:

- Temperature: $\theta_{in} = 20°C$.
- Humidity: $\Phi = 60\%$.

The general ventilation airflows are in balance.

TABLE 7.28 Characteristics of the building apertures in the example.

	Gate	Opening 1	Opening 2
Width, L (m)	6	4	2
Height, H (m)	4	2	1
Area, A (m^2)	24	8	2
Center height above ground, h_g (m)	2	9	5.5
Pressure coefficient, c_p (−)	0.7	−0.5	−0.3
Discharge coefficient, μ (−)	0.25	0.8	0.8

The unbalanced local exhausts take their makeup air through the building openings:

$$q_{v,m} = 4\ \mathrm{m}^3/s,\quad q_{m,m} = 4.8\ \mathrm{kg}/s.$$

7.7.6.3 Pressure distribution in the building

Calculation is made in accordance with Section 7.8.2, *Infiltration and Exfiltration*.

Mass balance of the airflows through the building envelope:

$$q_{m,g} + q_{m,1} + q_{m,2} + q_{m,m} = 0, \tag{7.249}$$

where

$$q_{m,i} = \mu_i A_i \left[2\Delta p_i\right]^{1/2}(\mathrm{kg/s}) \tag{7.250}$$

$$\Delta p_i = p_{i,\mathrm{out}} - p_{i,\mathrm{in}} \tag{7.251}$$

$$p_{i,\mathrm{in}} = p_x = \mathrm{const} \tag{7.252}$$

Air densities:

- Outdoor air density: $\rho_{out,1} = 1.35\ \mathrm{kg/m^3}$.
- Indoor air density: $\rho_{in} = 1.2\ \mathrm{kg/m^3}$.

Reference level 0 is located in the center of the gate opening, that is, 2 m above ground level.

Case 1: temperature −12°C, calm conditions

Design pressures

$$\begin{aligned}
\Delta p_{1,\mathrm{out}} &= 0\\
\Delta p_{2,\mathrm{out}} &= 0 + (h_1 - h_2)g\left(\rho_{\mathrm{out},1} - \rho_{\mathrm{in},1}\right)\\
&= (9 - 5.5\mathrm{m}) \times 9.81\mathrm{m/s^2}\left(1.35 - 1.2\mathrm{kg/m^3}\right)\\
&= 5.15\mathrm{Pa}\\
\Delta p_{\mathrm{g.out}} &= 0 + (h_1 - h_2)g\left(\rho_{\mathrm{out},1} - \rho_{\mathrm{in},1}\right)\\
&= (9 - 2\mathrm{m}) \times 9.81\mathrm{m/s^2}\left(1.35 - 1.2\mathrm{kg/m^3}\right)\\
&= 10.3\mathrm{Pa}
\end{aligned}$$

The design airflows are found from Eq. (7.250):

$$\begin{aligned}
q_{m,g} &= 0.25 \times 24\mathrm{m^2}\left[2\left(10.3\mathrm{Pa} - p_x\right) \times 1.35\mathrm{kg/m^3}\right]^{1/2}\\
&= 9.86\left(10.3\mathrm{Pa} - p_x\right)^{1/2}(\mathrm{kg/s})\\
q_{m,1} &= 0.8 \times 8\left[2\left(p_x - 0.675\right) \times 1.2\mathrm{kg/m^3}\right]^{1/2} = 9.91\left(p_x - 5.15\right)^{1/2}(\mathrm{kg/s})\\
q_{m,2} &= 0.8 \times 2\left[2\left(p_x - 0.0\right) \times 1.2\mathrm{kg/m^3}\right]^{1/2} = 2.48\left(p_x\right)^{1/2}\\
q_{m,m} &= 4.8\mathrm{kg/s}
\end{aligned}$$

The mass balance Eq. (7.249) now gives

$$9.68(10.3-p_x)^{1/2} = 9.91(p_x-5.15)^{1/2} + 2.48(p_x)^{1/2} + 4.8p_x \approx 6.05\text{Pa}.$$

The design differential pressure is

$$\Delta p_{g,1} = 10.3 - 6.05 = 4.25\text{Pa}.$$

Case 2: temperature $-5°C$, Wind 5 m/s
Design pressures

$$\Delta p_{1,\text{out}} = 0$$

$$\Delta p_{2,\text{out}} = 0 + (h_1 - h_2)g(\rho_{\text{out},1} - \rho_{\text{in},1}) + (c_{p,2} - c_{p,1})p_w$$
$$= (9-5.5)\cdot 9.81\cdot(1.32-1.2) + [-0.3-(-0.5)]\cdot 16.5$$
$$= 7.42\text{Pa}$$

$$\Delta p_{g,\text{out}} = 0 + (h_1 - h_g)g(\rho_{\text{out},1} - \rho_{in,1}) + (c_{p,g} - c_{p,1})p_w$$
$$= (9-2)\cdot 9.81\cdot(1.32-1.2) + [0.7-(-0.5)]\cdot 16.5$$
$$= 28.04\text{Pa}$$

The mass balance Eq. (7.249) is

$$9.86(28.04-p_x)^{1/2} = 9.91(p_x-7.42)^{1/2} + 2.48(p_x)^{1/2} + 4.8$$
$$p_x \approx 13.2\text{Pa}.$$

The design differential pressure is

$$\Delta p_g = 28.04 - 13.2 = 14.84\text{Pa}.$$

7.7.6.4 Calculation of the parameters of the air curtain

We choose the ratio between the gate area and the discharge opening area,

$$f = \frac{A_0}{A_S} = 20,$$

which implies an air curtain discharge aperture of

$$A_s = 1.2\text{m}^2.$$

The aperture height equals the height of the gate, 4 meters, which yields the width of the air outlet slot:

$$b_s = \frac{A_s}{4} = 0.2\text{m}.$$

The discharge angle of the air curtain jet is

$$\alpha = 30^\circ (\text{i.e.}, \sin\alpha = 0.5).$$

The efficiency factor E, according to Table 7.26 is $E = 0.3$.

Based on the given value f the values of the factors m_1 and m_2 are determined from Table 7.27:

$$m_1 = 2.9$$
$$m_2 = -1.3.$$

To determine the supply temperature of the curtain, θ_0, and the thermal capacity P, we use

$$\theta_{\text{mix}} = \theta_{\text{in}} = \theta_H.$$

The parameters of the air curtain can now be determined.

The initial discharge velocity is calculated according to formula (7.243):

Version a
$$v_0 = [(4.25\times 20)/(2\times 1.05\times 1.2\times 0.3)]0.5 = 10.6\ m/s$$
Version b
$$v_0 = [(14.84\times 20)/(2\times 1.05\times 1.2\times 0.3)]0.5 = 19.8\ m/s$$

The mass flow of the air supplied by the air curtain is determined according to formula (7.244):

Version a
$$q_{m,0} = 1.2\times 10.6\times 1.2 = 15.26\ \text{kg}/s$$
Version b
$$q_{m,0} = 1.2\times 19.8\times 1.2 = 28.51\ \text{kg}/s$$

The discharge temperature of the air curtain is determined from formula (7.247):

Version a
$$\theta_0 = -12 + 2.9(20+12) - 1.3(20+12) = 39.2°C$$
Version b
$$\theta = -5 + 2.9(20+5) - 1.3(20+5) = 35°\text{C}$$

The heat capacity of the air curtain is determined according to formula (7.248):

Version a
$$P = 1.01\times 10^3 \times 15.26(39.2-20) = 295.9\times 10^3\ \text{W}$$
Version b
$$P = 1.01\times 10^3 \times 28.51(35-20) = 431.9\times 10^3\ \text{W}$$

Thus version b is more unfavorable, as the differential pressures ΔP are the greatest. The selection of air curtain equipment is made for version b.

The fan of the air curtain is to be provided with a device regulating the supply airflow, for example, by frequency transformer, multispeed motor, etc. The supply airflow is to change according to the dependence $q_{v,0} \sim \Delta p^{0.5}$, which enables, in particular, provision of the necessary efficiency for version a.

As the temperature of the supplied air increases in case of decreasing outdoor temperature, the heater of the air curtain selected for version b is to be tested according to the calculation designed for the lowest outdoor temperature (version a). The heater is to be provided with devices for automatic regulation of the temperature of the supplied air according to the dependence $\theta_0 \sim \theta_{\text{out}}$.

7.7.7 Effect of air curtain on transport of contaminant

The installation of the air curtain may notably change the transport path of the contaminant released from the contaminant source, and influence the impact range of the air supply jet. The effect of the air curtain could be evaluated by TASA and TACS as well

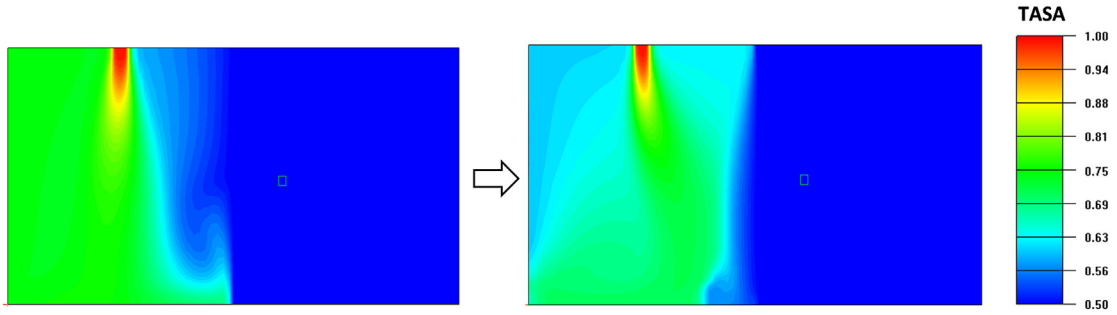


FIGURE 7.97 Impact of the air curtain on jet effectiveness at the steady state.

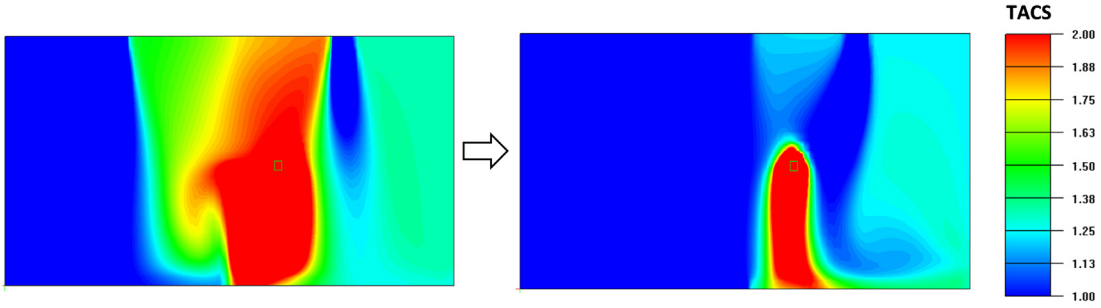


FIGURE 7.98 Impact of the air curtain on the contaminant transport at the steady state.

(Section 7.3.3). As a demonstration, Fig. 7.97 shows the impact of the air curtain on TASA distribution of the air jet in a typical room with dimensions of 6 m (length) × 2.6 m (height) × 3 m (width), where there is no air curtain in the left figure, while there is an air curtain installed in the middle of the room in the right figure.

When there is no air curtain, the left air jet has a large effect primarily on the left areas of the jet flow path. After installing the air curtain, the dominant impact area of the left air jet becomes larger, covering almost the entire left half of the space. Fig. 7.98 shows the impact of the air curtain on TACS distribution of the contaminant source, where the contaminant source is located on the right side of air curtain.

Before installing the air curtain, both the left half and the right half of the space are greatly polluted by the contaminant source (the left figure in Fig. 7.98). However, after installing the air curtain, the dominant impact range is limited to a small area below the source and the entire left half of the space is effectively protected with reduced TACS values (the right figure in Fig. 7.98).

7.8 Air movement around buildings and through a building envelope

7.8.1 Airflow around buildings

Airflow around buildings consists of natural winds that travel around and possibly through buildings. Airflow around buildings has two influences on industrial ventilation:

1. Wind pressures exerted on the exterior building surfaces, which can influence air movement indoors.
2. The outdoor movement of air contaminants, which can degrade indoor air quality if brought indoors with insufficient dilution.

This section will describe general features of airflow patterns and then present information on the dimensions and locations of recirculating (stagnant) zones around the building envelope, which determine wind pressures and contaminant dilution. This knowledge allows one to select the locations of stacks and air intakes and to calculate infiltration and natural ventilation rates.

7.8.1.1 General features of airflow around buildings

Buildings are immersed in an atmospheric boundary layer in which the wind is influenced by friction with the earth's surface. In this layer, wind speed tends to gradually increase with height and turbulence levels decrease with height, as described in many texts.[308] Surrounding buildings, terrain, and vegetation strongly influence wind and turbulence at a building site. The wind and turbulence levels are also influenced by thermal stratification of the atmosphere, such

as ground-level inversion layers. The major parameters of the wind at a building site depend on the Reynolds (Re), Karman (Ka), and Richardson (Ri) dimensionless characteristics:

$$\text{Re} = VI/v \quad \text{Ka} = \sqrt{(V')^2}/V$$
$$\text{Ri} = g1/\rho\,(\partial\rho/\partial Z)/(\partial V/\partial Z)^2, \tag{7.253}$$

where l is the building characteristic dimension (height or width), v is the kinematic viscosity, V' is the velocity fluctuation, V is the mean velocity, ρ is the air density, $\partial p/\partial Z$ is the vertical density gradient, and $\partial V/\partial Z$ is the vertical velocity gradient.

Winds traveling past a building will be greatly modified compared with winds in the absence of the building. Hosker reviews the information on flow around both isolated structures and building clusters.[309] Snyder and Lawson present detailed trajectories around isolated buildings obtained with detailed flow measurements.[310]

Fig. 7.99 illustrates typical flow patterns for wind directly approaching a building face. Airflow in the undisturbed zone has a speed profile dependent on the terrain roughness and the level of atmospheric stratification. Obviously most wind will be deflected around and over the building. Wind traveling toward the upwind face of the building will tend to stagnate, creating relatively high static pressures at the upwind face.

As the wind travels past a building corner or over a roof, it will be unable to negotiate the sharp turn and will separate from the building side and roof surfaces. The flow separation creates strong speed variations (shear), reversed flow directions (upwind vortex), negative (suction) pressures on building surfaces, and added turbulence. Another separation zone, the building or cavity wake, is created immediately downstream of the building as the wind encounters the downwind building corners and roof edge. Negative pressures are also seen on the downwind side of the building within the cavity wake. As the wind travels farther downstream, the shear and turbulence gradually diminish and settle to the condition that would apply without the building in place.

The flow pattern depends on the building's relative dimensions: long buildings (along wind length, $L > 2.5H$) versus short buildings ($L < 2.5H$).

The flow pattern shown in Fig. 7.99 illustrates the case of a short building ($L < 2.5H$) with a W/H ratio greater than 10. Near the upwind wall, a stagnation (recirculating) zone is formed. The direction of airflow in the area close to the surface is opposite the main wind direction. The flow separates at the sharp edge of the building to generate a recirculating flow zone. The boundaries of this zone can be described by the equation

$$\int_0^{Z_b} V dz = 0, \tag{7.254}$$

where V is the velocity at the point (x, z); and Z_b is the height of the recirculating zone boundary at the distance X.

Wilson indicates that for a flat-roofed building, the recirculation region maximum height H_c at location X_c, and the reattachment length L_r shown in Fig. 7.99 can be evaluated using the following equations[311]:

$$H_c = 0.22R \tag{7.255}$$
$$X_c = 0.5R \tag{7.256}$$
$$L_r = 1.0R, \tag{7.257}$$

where

$$R = B_s^{0.67} B_L^{0.33}, \tag{7.258}$$

where $B_s = \min[H, W]$, $B_L = \max[H, W]$, and H and W are the upwind building face height and width, respectively.

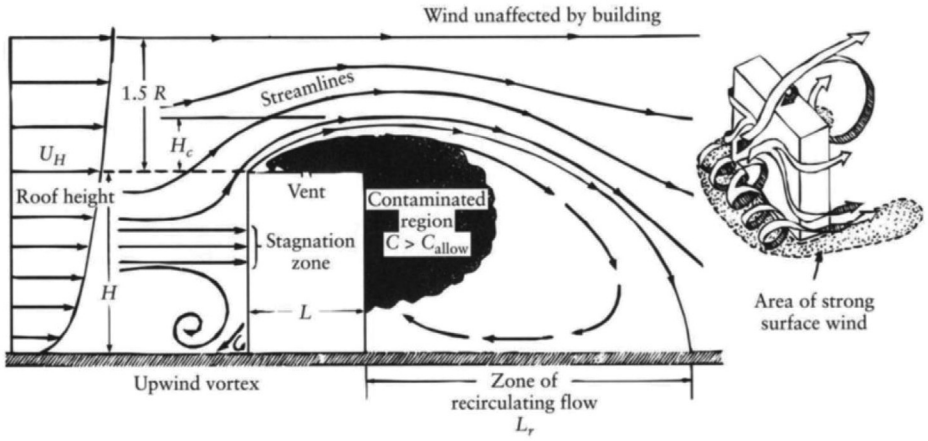


FIGURE 7.99 Flow pattern around a rectangular building.

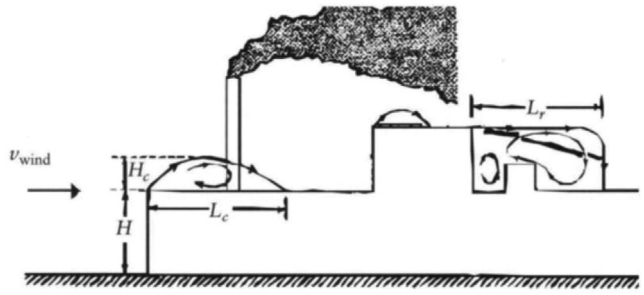


FIGURE 7.100 Flow pattern around a long rectangular building ($L > 2.5H$).

TABLE 7.29 Values of coefficient C_r

W/H	1	2	3	4	5	6	7	8	9	10
C_r	0.32	0.45	0.55	0.64	0.71	0.78	0.84	0.90	0.95	1.0

If the building has significant length L in the windward direction (Fig. 7.100), the flow will reattach to the building and may generate two distinct recirculating zones—on the building and in its wake. In the case of a long building ($L > 2.5H$), the recirculation zone created by separation of the flow at the front edge of the roof extends to some distance, L_c, smaller than the length of the building ($L_c \sim 0.9R$). Beyond this zone, at a distance of approximately $10H$ to $12H$, the flow streamlines along the roof surface become similar to those absent the effect of the building.

For $W/H < 10$, the length of this recirculating zone is reduced. The values of L_r and H_c can be calculated using the reduction coefficient C_r from Table 7.29.

Winds approaching a rectangular building at an angle will have different flow patterns than winds directly approaching a building face. Fig. 7.101 illustrates the case for approach from a 45 degrees angle. On the roof a pair of horizontal, counterrotating vortices emanate from the upwind roof corner. The negative (suction) surface pressures near the upwind corner can be intense, several times the magnitude of the dynamic pressure of the approaching wind.

The airflow in the center of the roof tends to be downward. The downward motions in the downstream cavity wake are also more intense for winds approaching from an angle.[310]

The abovementioned discussion is applicable only to the isolated building. Hosker reviews the effect of nearby buildings on the flow around the building of interest.[309,312]

7.8.1.2 *Building surface wind pressures*

One of the effects of airflow or wind around buildings is the exertion of wind pressure forces on the surface of the building, which contributes to natural ventilation of the building and infiltration of outside air into the building. As discussed earlier, pressures tend to be positive (into the building) on upwind surfaces and negative (suction) on lateral, downwind, and roof surfaces.

Pressure at a given location on the building surface is usually expressed as a pressure coefficient times a reference wind pressure at the building height without the building in place:

$$P = \frac{C_P \rho v_m^2}{2}, \tag{7.259}$$

where C_p is the pressure coefficient, ρ is the air density, and v_m is the general wind speed at the building height.

Estimation of building pressure comprises two steps:

1. Determination of the reference wind speed for the building site, v_m.
2. Determination of the pressure coefficient, C_p, for the particular location on the building.

Wind speed

Wind has a highly turbulent and gusting character. In addition, a time-mean speed varies with the height from the ground and the roughness of the terrain over which the wind passes. The time-mean wind speed profile can be determined using the following expression:

$$\frac{v_m}{v_{\text{met}}} = cH^a, \tag{7.260}$$

where v_m is the mean wind speed at height H above the ground, m/s, v_{met} is the mean wind speed measured at a weather station, normally at a height of 10 m above the ground, m/s, c is the parameter relating wind speed to nature of the terrain (see Table 7.30), and a is the exponent relating wind speed to height above ground.

To evaluate the wind speed at height H it is necessary to know the value of v_r for the required location. This may be obtained either from a local weather station or from wind contour maps of the country. Normally v_r represents the hourly mean wind speed that is exceeded 50% of the time at a particular site.

Other equations describing the wind profile are available from ASHRAE[313] and from Sherman and Grimsrud.[314]

Surface pressure coefficient

The second part of computing building pressures involves the pressure coefficient for a particular spot on the building. The surface pressure coefficient, C_p,

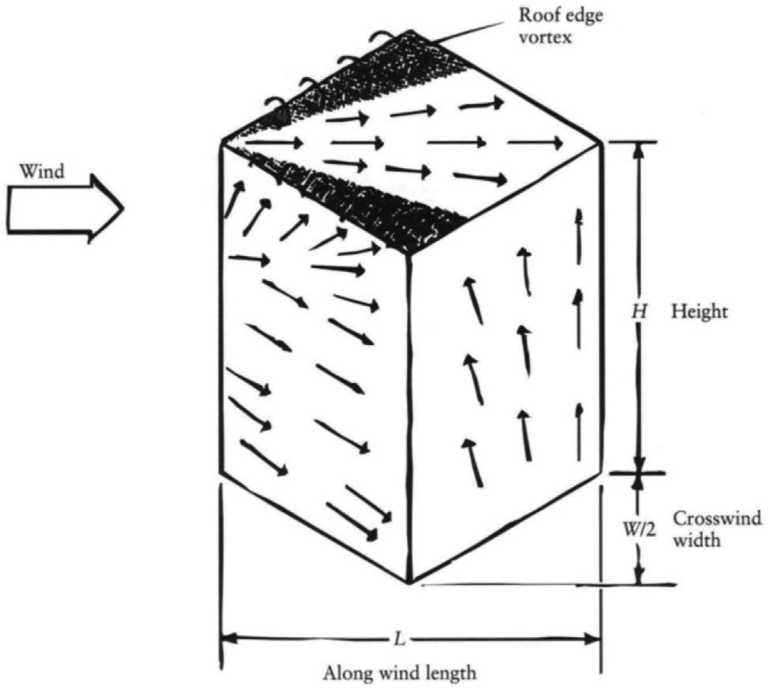


FIGURE 7.101 Flow pattern around a rectangular building with the wind approaching the building at 45 degrees.

TABLE 7.30 Terrain factors for Eq. (7.260).

Terrain	c	a
Open flat country	0.68	0.17
Country with scattered wind breaks	0.52	0.20
Urban	0.35	0.25
City	0.21	0.33

indicates the share of the wind kinetic energy that is transferred to the static pressure:

$$C_P = \frac{2P}{\rho v^{m2}}. \tag{7.261}$$

The value of coefficient C_p at the point on the building surface changes within a range of $-2 < C_p \leq 1$ and is determined by

1. the building geometry,
2. the wind velocity (i.e., speed and direction) relative to the building,
3. the location of the building relative to other buildings and the topography and roughness of the terrain in the wind direction, and
4. the location of a point on the building envelope.

The surface pressure coefficient is normally derived from pressure measurements in wind tunnels using reduced-scale models of buildings or building components, or from pressure measurements in the actual buildings. In general, the C_p values depend on the Re, Ka, and Ri numbers. However, experimental tests are typically conducted at high values of Re (Re ≥ 1000), in isothermal conditions (Ri = 0), and considering self-similarity against the Ka number (Ka = idem). Most test conditions allow only geometrical scaling.

For a building with sharp corners, C_p is almost independent of the wind speed (i.e., Reynolds number) because the flow separation points normally occur at the sharp edges. This may not be the case for round buildings, where the position of the separation point can be affected by the wind speed. For the most common case of the building with a rectangular shape, C_p values are normally between 0.6 and 0.8 for the upwind wall, and for the leeward wall 0.6 $C_p < -0.4$. Fig. 7.102 and Table 7.31 show an example of the distribution of surface pressure coefficient values on the typical industrial building envelope.

Values of C_p for simple building geometries may be obtained from the British Standards Institution[315,316] or from Liddament.[317] The following relationship between wind incident angle α, building side ratio, and average surface pressure coefficient is based on the database developed by Swami and Chandra[318]:

$$NC_p = \ln[1.248 - 0.703\sin(\alpha/2) - 1.175\sin^2(\alpha) + 0.131\sin^3(2\alpha G) + 0.769\cos(\alpha/2) + 0.07G^2\sin^2(\alpha/2) + 0.717\cos^2(\alpha/2)], \tag{7.262}$$

where NC_p is the normalized C_p, α is the angle in degrees between wind direction and outward normal of wall under consideration, and G is the natural log of

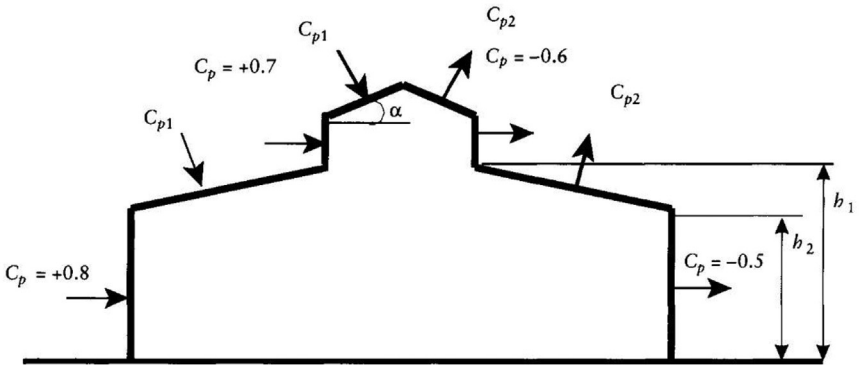


FIGURE 7.102 Example of surface pressure coefficient values for a typical industrial building envelope.

TABLE 7.31 Approximate surface pressure coefficient values for a building with a rooftop vent.

		C_{p1} and C_{p2}			
		h_1/L			
Coefficient	α	**0**	**0.5**	**1.0**	**≥2**
C_{p1}	0	0	−0.6	−0.7	−0.8
	20	+0.2	−0.4	−0.7	−0.8
	40	+0.4	+0.3	−0.2	−0.4
	60	+0.8	+0.8	+0.8	+0.8
C_{p2}	≤60	−0.4	−0.4	−0.5	−0.8
		C_{p3}			
		h_1/L			
Coefficient	*W/L*		**≤0.5**	**1.0**	**≥2**
C_{p3}	≤1		−0.4	−0.5	−0.6
	≥2		−0.5	−0.6	−0.6

ratio of width of wall under consideration to width of adjacent wall.

A detailed method of determining pressure coefficients is to perform experiments with a wind tunnel facility. Cochran and Cermak compared wind tunnel pressure coefficient measurements with field measures on a test building and found excellent results, with the exception of small areas beneath the vortices near the upwind roof corner for winds approaching at 45 degrees.[319] For infiltration and natural ventilation designs, wind tunnel results should be sufficiently accurate. With the progress of current techniques, the prediction of pressure coefficients as affected by unsteady turbulence wind flow has been possible. Chu et al. successfully predicted mean and unsteady pressure coefficients of single-sided ventilation with two openings under different wind directions using wind tunnel experiments. The external and internal pressures were measured by a multichannel high-speed pressure scanner (ZOC33/64PX, ScanivalveInc.). The measuring range of the pressure sensor was ±2758 Pa, with a resolution of ±2.2 Pa. The sampling frequency was 250 Hz and the sampling duration 65.5 seconds. The pressure scanner was connected to the pressure taps (diameter 1.5 mm, flush to the wall) with plastic tubing (length 0.3 m). The external pressures were measured 30 mm above the edge of the external openings and the internal pressure tap was at the center of the ceiling.[320] Daish et al. also used wind tunnel experiments to predict the pressure coefficients at the openings of a single-sided ventilated building with two apertures. Time-resolved pressure measurements (sampled at 1000 Hz) were obtained by means of flush surface pressure mounts located on building walls and around building openings.[321]

Another detailed method of determining pressures is CFD, which uses a numerical solution of simplified equations of motion over a dense grid of points around the building. Murakami et al.[322] and Zhoy and Stathopoulos[323] found less agreement with CFD methods using the $k-\epsilon$ turbulence model typically used in current commercial codes. More advanced turbulence models such as LES were more successful but much more costly.[322] In the industrial field, the geometrical models are always very complicated so that the LES will have high demand for the hardware of computers and thus is inefficient in terms of both time and economic cost. Thus the LES can not be widely applied despite its relatively higher accuracy. In recent two decades, the standard $k-\epsilon$ turbulence model has been modified into many other RANS turbulence models, such as the Realizable $k-\epsilon$ model, the RNG $k-\epsilon$ model, the SST $k-\omega$ model, and so on. These RANS models have been widely used in both industrial and academic field due to their much lower computing demands and sufficient predicting accuracy. The RANS method can be mainly classified into steady and unsteady RANS methods, according to the characteristics of studied flow fields. Montaxeri et al. used steady RANS method with Realizable $k-\epsilon$ model to mean

wind pressure distributions on windward and leeward surfaces of a medium-rise building with and without balconies. The turbulence model was proved to be able to accurately reproduce the mean wind pressure distribution across the windward facade of the building based on validation with wind tunnel experiments.[324] Zhong et al. used 3D unsteady RANS method with SST $k-\omega$ model to predict both the time-averaged and fluctuating pressure coefficients at the openings on the downstream building under the effect of an upstream building. This model was proclaimed to have sufficient accuracy in predicting the pressure coefficients at openings and building surfaces.[325]

For simplified buildings or in cases where detailed modeling is not practical, simplified tables of coefficients are presented by Liddament for low-rise buildings with two building shapes, for open and sheltered buildings, and for various walls and approach wind angles.[326] ASHRAE also summarizes results from other studies.[313]

7.8.1.3 Contaminant transport around buildings

Transport of outdoor contaminants is controlled by both the mean motion of winds and dispersal by turbulence. Since airflow around buildings has distorted wind trajectories and enhanced turbulence compared with the airflow without buildings, contaminant transport requires special consideration in the presence of buildings.

Contaminants can either be generated by sources within the building or be emitted from exhaust stacks or nearby locations. The goal is to maintain indoor air quality by minimizing the entry of outdoor contaminants into the interior of the building through careful design and location of exhausts, intakes, and building openings. Since most industrial exhausts and air-handling systems are located on the roof, the airflow on the roof is of great interest. As discussed earlier, when wind directly approaches a building side, a separation zone is created on the roof near the upwind edge. This zone will have high turbulence intensities and greatly distorted streamlines. In fact, flow reversal occurs, in which the airflow is in the opposite direction to the approaching wind. Any building exhaust emitted into this zone is quickly dispersed and spread over the roof surface, creating large rooftop contaminant concentrations. It is especially important to avoid placing low-momentum exhausts and air intakes within the same recirculating zone on the roof or side of the building.

Furthermore winds at an angle to the building side will have downward air motions above the roof and downwind of the building, which also will increase rooftop concentrations.

Therefore to avoid separation and high-turbulence zones on the roof, contaminant exhaust systems should be designed to extend as high above the roof as practically possible. Two ways to achieve high exhaust trajectories are with the use of taller stacks and greater exhaust vertical momentum, which increases the throw or rise of the exhaust. Tall stacks are an obvious solution but have esthetic problems, even in an industrial setting. Achieving greater vertical momentum can be gained in several ways:

- avoiding raincaps over exhausts, which eliminate vertical momentum,
- increasing exit velocity by narrowing the exhaust opening,
- manifolding separate exhausts into fewer stacks, and
- placing exhausts *very* close together so that the exhaust plumes can merge (which can help exhausts that cannot be manifolded).

In the design of an industrial facility, a detailed analysis of contaminant transport can be beneficial in specifying exhaust designs and intake locations. As with building pressures discussed earlier, analysis techniques range from simplified models to experimental methods and sophisticated computational methods. The greater the building geometry complexity, the less useful are the simplified models.

One simplified method for determining stack height is a geometric method described in ASHRAE.[313] The geometric method assumes an exhaust plume shape with a lower boundary having a 1:5 slope relative to the horizontal. The stack and plume are raised until the lower plume boundary is above rooftop penthouses, separation zones, and zones of high turbulence. ASHRAE provides equations for the sizes and locations of the separation and turbulence zones.[313] A stack height reduction credit is provided to account for the vertical exhaust momentum.

Another method is a series of exhaust dilution equations based on Wilson and Lamb[327] and a series of earlier papers summarized in ASHRAE.[313] This method is based on wind tunnel tests on simplified buildings and is intended to provide conservative (low dilution) results. Wilson and Lamb compared the model to actual field data collected at a university campus and found that the model did indeed predict dilutions similar to measured worst-case dilutions suitable for a screening model. However, many cases resulted in conservative underpredictions of dilutions.[327]

An alternative simple model for contaminant dilution of rooftop exhaust stacks is presented in Halitsky.[328] This model combines a jet region specification for the upward exhaust movement with a more traditional Gaussian plume region controlled by atmospheric and building-generated turbulent dilution.

For computing dilutions in the downstream cavity wake (intakes or building openings on the side of a building) due to contaminants from a rooftop stack, another model is presented by Schulman and Scire[329] and is incorporated in a US EPA screening model.[330] More sophisticated analysis may be achieved with wind tunnel models. Saathoff et al. compared wind tunnel dilution predictions with measurements on an actual building and found agreement within a factor of 2, a reasonable limit for dilutions, which can vary over many orders of magnitude.[331] Higson et al. found greater differences between wind tunnel measurements and a miniature building in field conditions, corresponding to a factor of 4 or more.[332]

CFD is becoming more and more popular, as discussed earlier for predicting building surface pressures. Although certain early paper found difficulties in the practical use of commercial codes due to the wide range of user inputs and decisions.[333] Efforts were once made to replace standard $k-\epsilon$ model typically used in commercial codes.[334,335] As various modified RANS turbulence model have been explored to increase the accuracy and efficiency in CFD simulation, the RANS methods have become the most popular and efficient method in the field of industry.

7.8.2 Infiltration and exfiltration

Most building envelopes have purposely provided openings (i.e., doors, windows, vents, flues, chimneys, and other ducts) and unintentional openings (i.e., cracks, mortar joints, and gaps around closed windows and doors). Air leakage through unintentional openings in the building envelope result in exchange between outside and indoor air. Uncontrolled outside airflow through cracks and other unintentional openings is called infiltration; the uncontrolled indoor airflow through unintentional openings is called exfiltration. Air leakage through the building envelope is a measure of the air tightness of the building envelope. Air leakage through the building envelope has a positive effect by allowing for natural (free) building ventilation. On the other hand, infiltration increases heat losses (in winter) and gains (in summer) through the building envelope, and also may result in reduced control over contaminant movement within the building.

In general, the air leakage rate through a building envelope is dependent on

- the sizes and distribution of leakage paths,
- the flow characteristics of the leakage paths, and
- the pressure difference across the leakage paths.

The flows through the openings in a building are not independent but are based on the mass balance across the whole building envelope. The flow rate through an opening depends on the pressure difference across it. Normally the pressure difference occurs due to the wind effect and a temperature difference between the indoor and outdoor air. Also an imbalance in the mechanical exhaust ventilation system performance over the mechanical air supply (positive or negative pressure building) might be a factor influencing infiltration and exfiltration.

A typical envelope opening has a complicated shape and is often subject to unsteady flow conditions at its inlet and outlet.[51] There are no simple analytical solutions for the flow through such openings. The most-used equation representing flow characteristics is the so-called power law:

$$Q = C_d A \left(\frac{\Delta P}{\rho}\right)^{\beta}, \tag{7.263}$$

where Q is the airflow through the opening, C_dA is the effective leakage area, and β is a coefficient.

Experimental evidence regarding the power law is somewhat contradictory. A constant value of $\beta = 0.5$ is considered to give a good fit to experimental data by many authors.[51] According to Awbi, β depends on the flow regime and has a value of 0.5 for fully turbulent flow and 1.0 for laminar flow.[154] In practice the value of β tends to be between 0.6 and 0.7.

Limited information is available on values of C_d for industrial buildings. The effective leakage area, C_dA, can be determined by means of a building pressurization or depressurization test.[336,337] A range of values of C_d for cracks formed around closed windows are given in Table 7.32. These should be used with a value of $\beta = 0.67$.

For practical situations Etheridge et al. suggest representing the typical unintentional opening by a long, narrow straight pipe or duct and describing the flow characteristics by a quadratic relation between Q and ΔP[51]:

$$\Delta P = aQ^2 + bQ, \tag{7.264}$$

where a and b are constants. The first term on the right-hand side represents turbulent flow and the second represents laminar flow. The values of a and b can

TABLE 7.32 Values of C_d for windows.[315]

	Value of $C_d \times 10^{-5}$	
Window type	**Average**	**Range**
Sliding	8	2–30
Pivoted	21	6–80
Pivoted (weather-stripped)	8	0.5–20

be obtained from experimental tests on the openings, and some values can be found in Baker et al.[338]

For extremely narrow openings (cracks) with deep flow paths (such as mortar joints and tight-fitting components) the flow is laminar and the flow rate, Q (m^3/s), can be described by the Couette flow equation[154]:

$$Q = \left[\frac{bh^3}{12\mu L}\right]\Delta P, \tag{7.265}$$

where b is the length of crack, m, h is the height of crack, m, L is the depth of crack in flow direction, m, and μ is the absolute viscosity of air, Pa s.

7.8.2.1 Pressure difference due to stack effect

The pressure difference across the crack can result from the difference in temperature (air density) between the air inside and outside the building. Static pressure in the vertical column of air varies with height and can be described by the following equation:

$$dp = g\rho\, dh, \tag{7.266}$$

where g is the gravitational acceleration, m/s^2 and ρ is the density of air, kg/m^3.

Thus static pressure, p_h, at height h can be calculated as

$$p_h = p_o - g\int_0^h \rho\ dh, \tag{7.267}$$

where p_0 is the static pressure at the reference height in undisturbed flow outdoors, Pa. Assuming that the air density does not change along the building height, Eq. (7.267) can be simplified to

$$p_h = p_o - g\rho h. \tag{7.268}$$

When the air temperature inside the building is greater than the outside air temperature, air infiltrates through the lower openings in the envelope (the pressure difference between the outside air and inside air is positive) and exfiltrates through the upper opening (pressure difference is negative). The height of the neutral plane where the pressure difference across the crack due to stack effect equals 0 depends on the crack's size, location, and characteristics.

The stack pressure can be expressed relative to the lowest opening height,[154,339] relative to the static pressure at the floor level,[88] or relative to the point with the minimum static pressure.[340]

7.8.2.2 Wind pressure

Static pressure over the building surfaces produced by the action of wind is generally positive on the windward side and negative on the leeward side. Pressures on the other sides of a building are negative or positive, depending on wind angle and building shape. The pressure difference across the crack produced by the action of wind can be calculated using Bernoulli's equation[313]:

$$\Delta P = P_o + C_p\rho V^2/2 - P_i, \tag{7.269}$$

where ΔP is the $P - P_r$ is the pressure difference across the crack (between outdoors and indoors) at the height of the crack, Pa, P_o is the static pressure at the reference height in undisturbed flow outdoors, Pa, P_i is the interior pressure, Pa, C_p is the surface pressure coefficient, V is the wind speed at the datum level (usually the height of the building), m/s, and ρ is the air density, kg/m^3.

Principles of wind speed and surface pressure coefficient evaluation were covered earlier.

7.8.2.3 Effect of ventilation system performance

General and local supply and exhaust ventilation systems can create negative, positive, or neutral pressure in the building. Static pressure created by a mechanical ventilation system inside the building does not change with height. The pressure difference across the crack due to the mechanical system's performance does not change with height and depends on the fan's performance curves and the crack's characteristics. The pressure across cracks due to unbalanced ventilation system performance depends on the difference in supply and exhaust airflow rates, ΔQ_{mech}, and the effective leakage area, C_dA.

7.8.2.4 Combined effect of gravity forces, wind, and mechanical ventilation

The airflow rate infiltrating and exfiltrating through each air leakage pass, Q_i, due to the combined effect of wind, stack, and mechanical ventilation system performance can be calculated from the mass balance equation

$$\sum Q_i + \Delta Q_{mech} = 0. \tag{7.270}$$

The airflow rate Q_i for each air leakage path is expressed with Eqs. (7.263), (7.268), and (7.269) using the information on effective leakage area, C_dA, and a pressure difference across the path. The total pressure acting on an opening from the outside is the sum of the pressure due to wind, gravity forces, and mechanical ventilation performance, and the static pressure inside the building results from Eq. (7.270).

The graphs in Fig. 7.103A show the pressure distribution along the building height due to gravity forces using the method suggested by Titov.[340] The reference point with a pressure equal to 0 is located in the upper point outside the building. The graph shape is not affected by the number of openings and their locations.

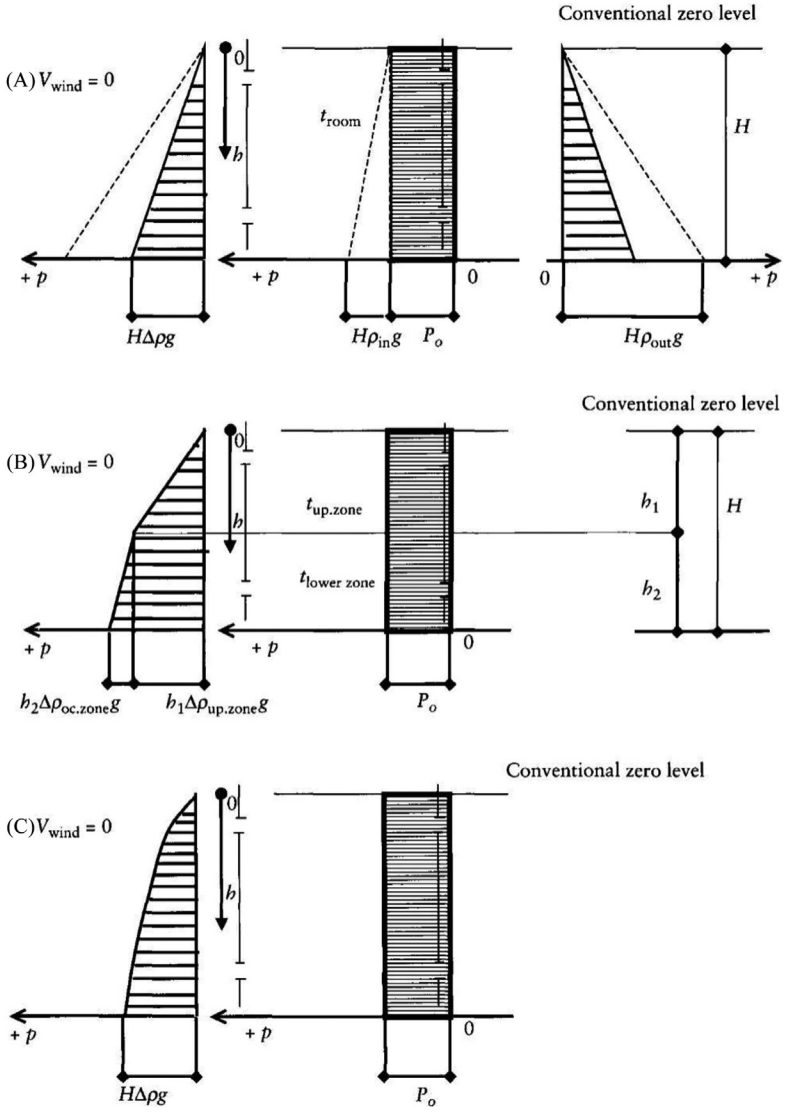


FIGURE 7.103 Pressure change along the building height due to gravity forces: (A) even temperature in the room volume; (B) room height divided into two zones, upper and lower (two-zone model); and (C) temperature gradient along the room height.

The same approach can be applied to the case with temperature stratification along the room height. The graph in Fig. 7.103B illustrates the case for the two-zone model, and the graph in Fig. 7.103C illustrates a temperature gradient.

The graphs in Fig. 7.104A show the pressure distribution along the building height due to the wind effect (wind velocity does not change along the height). The reference point with a pressure equal to 0 is located at the point with minimum surface pressure coefficient, C_p. The graph reflecting the influence of the nonuniform velocity distribution along the height is presented in Fig. 7.104B.

The combined effect of gravity forces, wind, and mechanical ventilation on the pressure across the opening is illustrated in Fig. 7.105.

When the opening in the building envelope is horizontal or is vertical with a relatively small length, the change in pressure difference along this opening can be neglected. For long vertical openings the change in pressure along the height should be considered, because the lower part of the opening can provide infiltration and the upper part exfiltration

7.8.2.5 *Calculation methods*

The principles discussed in this section enable the calculation of air infiltration/exfiltration, provided that the following quantities are known or can be evaluated[154]:

- wind speed and direction,
- internal and external air temperature,
- position and flow characteristics of all openings,
- pressure coefficients over the building envelope for the wind directions under consideration, and
- supply and exhaust ventilation airflow rates.

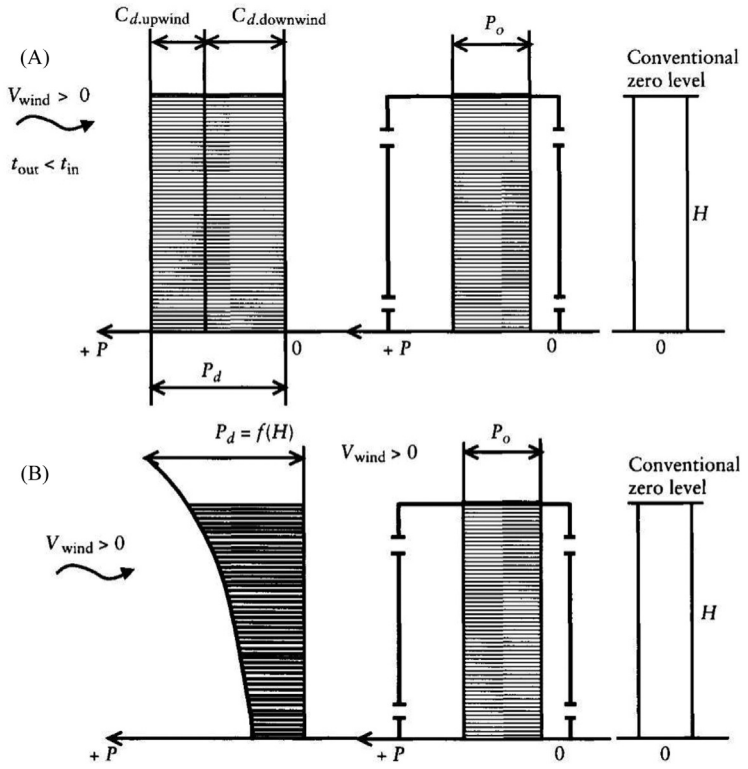


FIGURE 7.104 Pressure distribution due to wind effect: (A) uniform wind velocity profile and (B) nonuniform wind velocity profile.

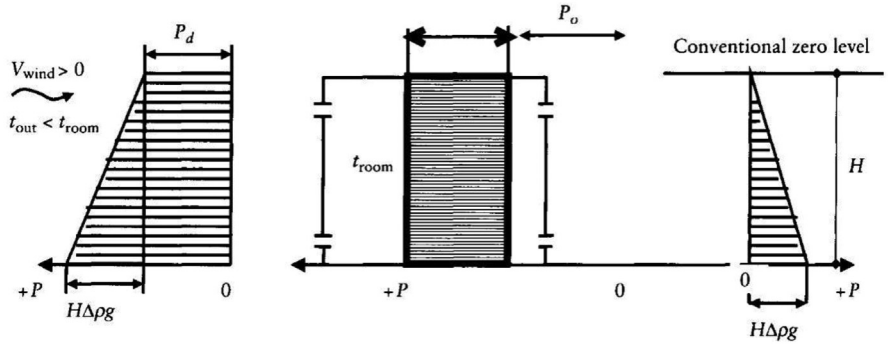


FIGURE 7.105 Pressure change along the building height due to the combined effect of wind and gravity forces, for the case of even temperature in the room volume.

In practice it is difficult, if not impossible, to determine all these quantities accurately, and the following simplified calculation methods based principally on equations discussed in this section are used:

- empirical methods,[313,315]
- simplified theoretical methods,[88,314,339,341] and
- network models, primarily for multizone buildings.[317,342]

7.8.3 Airflow through large openings and gates

When the building envelope has high openings (e.g., gates or large glazing), the pressure difference across such openings changes with height. In some situations, airflow through a part of the opening enters the building and the airflow through the rest of this opening exits the building. In certain cases this can be neglected, and the pressure difference across the whole opening is assumed to be equal to the one at the center of this opening. However, for openings such as high gates, such simplification results in significant error.

The pressure change outside the building envelope with such an opening is illustrated in Fig. 7.106. The outside air static pressure, P_{out}, is greater than the inside air static pressure, P_o, at the lower part of the opening, $h > h_o$, and is lower at its upper part, $h < h_o$. Thus air is infiltrating through the lower part of the opening and exfiltrating through its upper part.

The airflow rate through an opening area with a height of dh can be calculated as

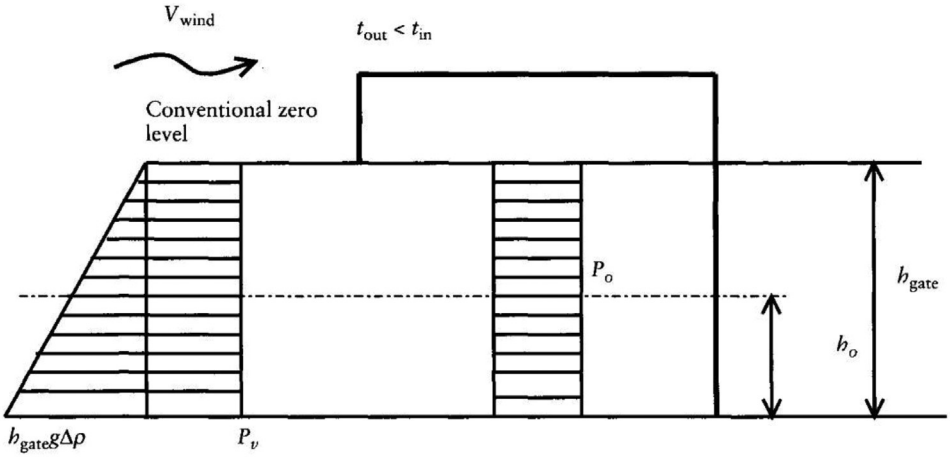


FIGURE 7.106 Pressure distribution inside and outside a building with a single large opening.

$$dG = C_d(2\rho_h)^{1/2}\,(P_{out} - P_o)^{1/2}\,dh. \tag{7.271}$$

The infiltrating and exfiltrating airflow rates can be calculated using the following equations:

$$G_{inf} = \int_H^h dG \tag{7.272}$$

$$G_{exf} = \int_h^0 dG. \tag{7.273}$$

Let us consider airflow through the gate under the influence of buoyancy forces and wind (Fig. 7.106). The gate has width b and height h_g and is located on the upwind face of the building. The gate pressure loss characteristic does not change with height. Also assume there are no apertures or cracks in the building envelope, other than the gate.

The mass balance equation for the airflow through the gate can be described as

$$G_{exf} + G_{inf} = 0. \tag{7.274}$$

At the height h_o of the boundary between the infiltrating and exfiltrating flows, air pressure across the gate, $P_{out} - P_o$, equals 0, and thus

$$P_o = \Delta\rho g h_o + C_{gate}\frac{\rho\,\nu^2}{2}. \tag{7.275}$$

Assuming $(\rho_{out}/\rho_o)^{1/3} \approx 1$, the equation allowing calculation of the pressure inside the building can be presented as

$$P_o = \frac{h_o}{2}\Delta\rho g + C_{gate}\frac{\rho\nu^2}{2}. \tag{7.276}$$

A properly designed air curtain protects the building from outside airflow through the gate. Only air entrained by the air curtain enters the building through the gate.

Total gate protection occurs when the air flow rate exiting the building through the gate, G_{ex} equals the air flow rate, G_o, supplied through the air curtain slot:

$$G_{ex} = G_o \tag{7.277}$$

When the gate is totally protected, there are no other openings in the building envelope and there is a balance between supply and exhaust ventilation systems, $h_o = h_{gate}$ (Fig. 7.107). In this case the pressure difference across the gate is influenced only by the buoyancy forces, and the airflow through the gate can be evaluated using the following equation:

$$G_{ex} = C_d A_o[h_{gate}g(\rho_{out} - \rho_o)\rho_{out}]^{1/2}, \tag{7.278}$$

where C_d is the pressure loss coefficient for the gate and C_d typically has a value between 0.2 and 0.3 (it can be assumed equal to 0.25). For more information about gate protection with air curtains, see Section 7.7.

7.8.4 Principles of natural ventilation and "pumping mechanism"

7.8.4.1 Design principles of natural ventilation

Natural ventilation is used in spaces with a significant heat release, when process and hygienic requirements for indoor air quality allow outdoor air supply without filtration and treatment. Natural ventilation cannot be used when incoming outdoor air causes mist or condensation. Natural ventilation allows significant air change rates (20–50 ach) for heat removal with minimal operation costs.

Though airflow through the building with natural ventilation is caused by both wind effect and buoyancy forces, design principles typically do not include a wind pressure component. Wind speed and direction can change over wide ranges and thus wind does not

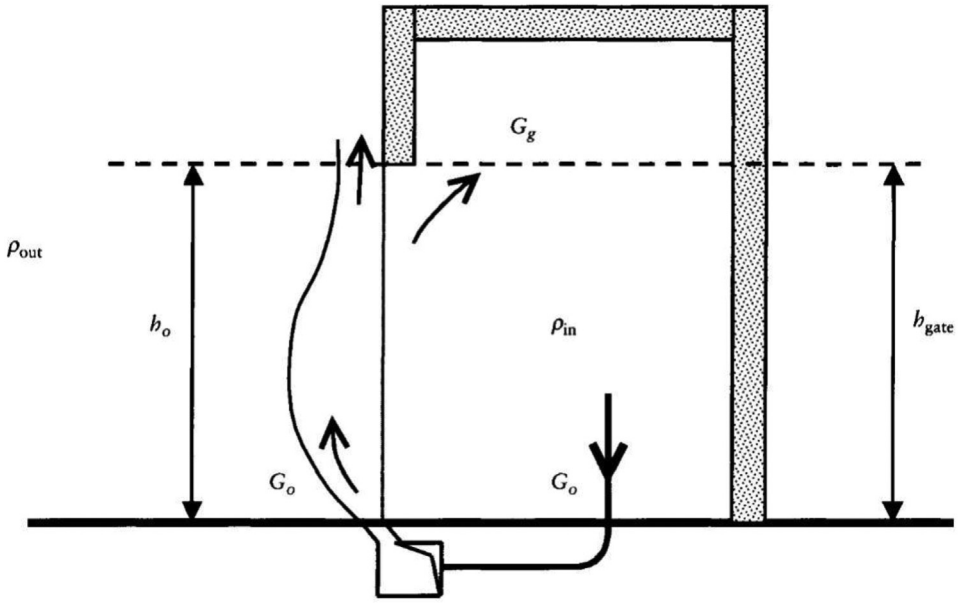


FIGURE 7.107 Schematics of airflow through the gate with an air curtain, for a building having no other openings.

provide a stable force to move air through the building. Thus the wind effect is frequently considered as a reserve. In general, maximum air change in the building is required in the summer, with a maximum heat load and a minimal temperature difference between outdoor and indoor air. With a change in the heat load, natural ventilation allows for self-adjusting airflow through the building.

Airflow through an opening used for natural building ventilation is approximately turbulent. In this case the flow rate, Q (m^3/s), can be calculated from an equation similar to Eq. (7.263):

$$Q = C_d A \left(\frac{\Delta p}{\rho}\right)^{0.5}, \qquad (7.279)$$

where C_d is the pressure loss coefficient of opening, A is the area of the opening, m^2, Δp is the pressure difference across the opening, Pa, and ρ is the air density, kg/m^3.

Air outlets are designed such that their pressure characteristics are negative, which improves their performance in the presence of wind. Different types of ridge vents and roof ventilators have been designed; their sizes and pressure loss coefficients are available from the manufacturers. Examples of some air inlets and their pressure loss coefficients are listed in Table 7.33. All inlets and outlets for natural ventilation must be supplied with controls for easy opening and closing. Ventilated interior halls can obtain outside air through ridge vents in adjacent "cold" halls.

In summer, air inlets for natural ventilation are located in exterior walls; the lower level of the openings is 0.3–1.8 m above the floor. Air inlets can be arranged in one, two, or more rows in the longitudinal exterior walls. Windows, doorways, and other types of openings in the exterior walls, or apertures in floors over basements (with air transportation along special channels) also can be used as air inlets. During periods of the year other than summer, air inlets must located higher than 4 m. These inlets must be supplied with baffles to direct air at an upward angle. Air is evacuated from naturally ventilated spaces through wind-protected continuous ridge vents, skylights, or round roof ventilators.

Natural ventilation design allows one to size the inlets, A_{inl}, and outlets, A_{out}, based on their pressure loss characteristics, C_p, and on the airflow rate, G_o, required to maintain the occupied zone within desired limits. The reverse design procedure is commonly used to evaluate the airflow rate through the building given the sizes, characteristics, and locations of inlets and outlets and the heat load and characteristics of heat sources.

The use of a natural ventilation system assumes temperature stratification throughout the room height. Air close to heat sources is heated and rises as a thermal plume (Fig. 7.108). Part of this heated air is evacuated through air outlets in the upper zone, and part of it remains in the upper zone, in the so-called heat cushion. The separation level between the upper and lower zones is defined in terms of the equality of G_{conv} and G_o, which are the airflow rate in thermal plumes above heat sources and the airflow supplied to the occupied zone, respectively. It is assumed that the air temperature in the lower zone is equal to that in the occupied zone, t_{oz}, and that the air temperature in the upper zone is equal to that of the evacuated air, t_{exh}.

TABLE 7.33 Pressure loss coefficients for inlets.[12]

		Pressure loss coefficient				
		Baffle angle, α				
Baffle type	***h/b***	**15 °**	**30 °**	**45 °**	**60 °**	**90 °**
Single, top-hinged	0	30.8	9.2	5.2	3.5	2.6
	0.5	20.6	6.9	4.0	3.2	2.6
	1	16.0	5.7	3.7	3.1	2.6
	0	59.0	13.6	6.6	3.2	2.7
Single, center-hinged	1.0	45.3	11.1	5.2	3.2	2.4
	0.5	30.8	9.8	5.2	3.5	2.4
Double, top-hinged	1.0	14.8	4.9	3.8	3.0	2.4

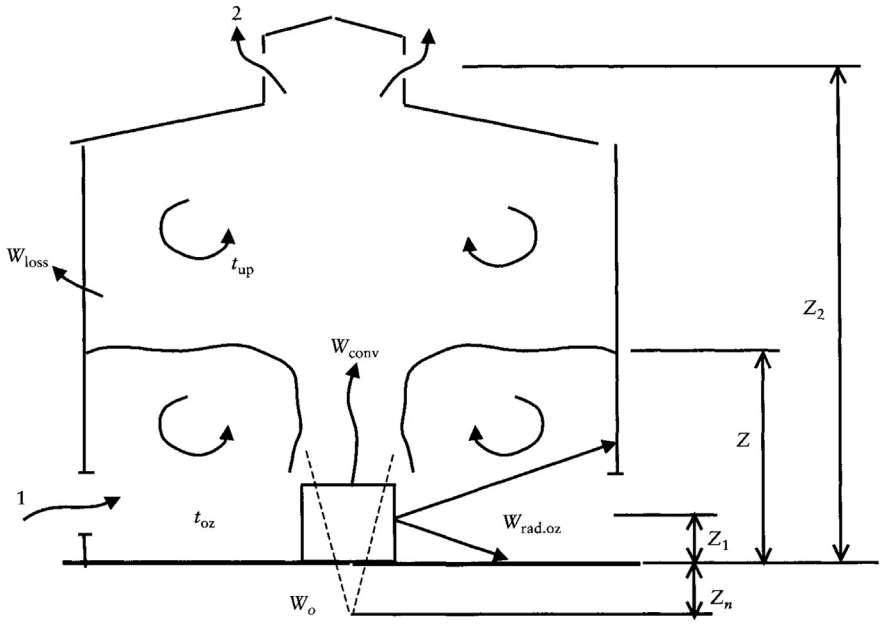


FIGURE 7.108 Schematics of naturally ventilated space: 1, air inlet and 2, exhaust vents.

The air exchange rate, G_o, required for temperature control in the occupied zone can be calculated from the room heat balance equation:

$$G_o = W[C_p K_\theta(\theta_{uz} - \theta_{oz})], \tag{7.280}$$

where G_o is the air exchange rate, kg/s, W is the total surplus heat released in the space, kW, C_p is the specific heat of air, kJ/(kg K), and K_θ is the coefficient of heat removal efficiency, calculated from the air temperatures of the occupied zone, θ_{oz}, the air removed from the upper zone, θ_{exh}, and the outdoor air, θ_o:

$$K_\theta = \frac{(\theta_{exh} - \theta_o)}{(\theta_{oz} - \theta_o)},$$

Coefficient K_θ can be evaluated through measurements in field or on a scaled model. Also it is possible to predict the K_θ value using the method of zone-by-zone heat balances.[343] According to this method,

$$K_\theta = 1/\left\{[\phi(1-\psi) - \lambda]/2\left\{[\phi(1-\psi) - \lambda]^2/4 + \lambda\right\}^{1/2}\right\}, \tag{7.281}$$

where

$$\psi = \sum (W_o \times \psi)_i / \sum W_o \tag{7.282}$$

$$\phi = \sum [(W_o \times (1-\psi)\phi)_i] / \sum [W_o(1-\psi)] \tag{7.283}$$

$$\lambda = \alpha_{rad} A_{oz}(\theta_{oz} - \theta_{out}) / \sum W_o \tag{7.284}$$

where θ_{oz} is the occupied zone air temperature, °C; θ_{out} is the outside air temperature, °C; and A_{oz} is the occupied zone area, m^2. A graphical interpretation of Eq. (7.281) is presented in Fig. 7.109.

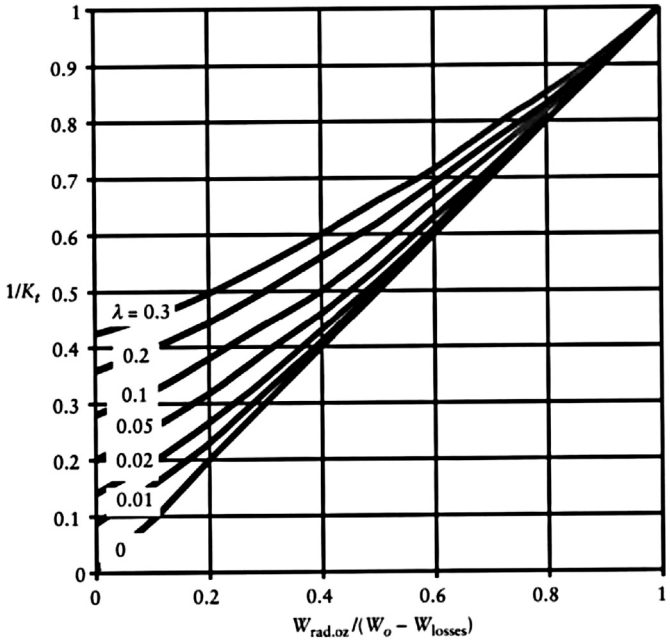


FIGURE 7.109 Supporting graph for the K_t calculation (Eq. 7.281). $\lambda = \alpha_{rad}\, A_{oz}\, \Delta t_{oz}/(W_o - W_{losses})$; $0 \le \lambda \le 1$.

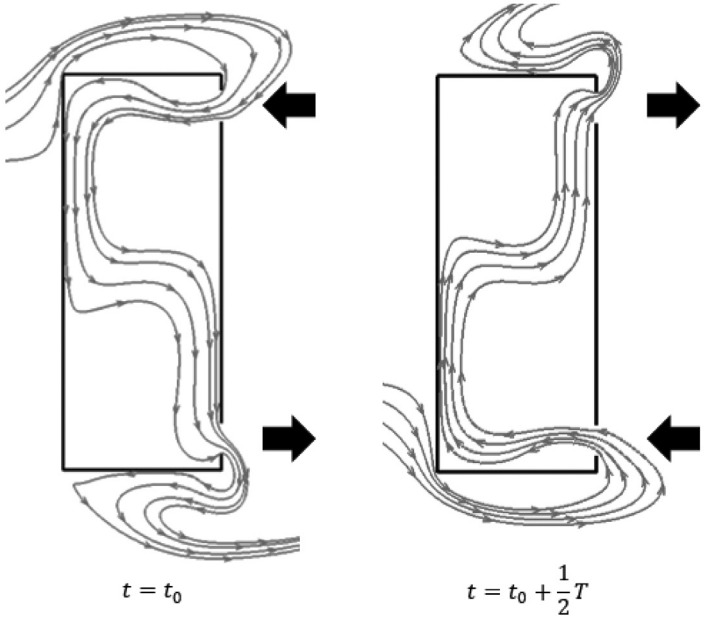


FIGURE 7.110 The two alternating phases of pumping flow (driven by vortex shedding).

Typically inlet and outlet locations and heights can be obtained prior to ventilation system design from construction drawings. The static pressure difference across inlets and outlets can be calculated based on the height of the location (Fig. 7.109) and the air density at the respective height:

$$\Delta p = g(Z - Z_1)(p_o - p_{oz}) - g(Z_2 - Z)(p_o - p_{exh}), \quad (7.285)$$

where Z_1 and Z_2 are the heights of inlet and outlet centers, m; and Z is the separation zone height (typically between $0.4\,H$ and $0.7\,H$), with H is the ventilated space height, m. For a more detailed calculation of K_t and Z, see Chapter 8, Room Air Conditioning and Stroiizdat.[12]

The temperature of exhausted air can be derived from Eq. (7.272):

$$\theta_{exh} = \theta_o + W/(C_p G_o). \quad (7.286)$$

Based on the static pressure across the inlets, $\Delta p_{inl} = \beta\, \Delta p$, one can calculate the required inlet area, A_{inl}:

$$A_{inl} = G_o/\left(2\rho_o \Delta p_{inl}/C_{pinl}\right)^{1/2}. \quad (7.287)$$

Typically the share of the static pressure across the inlets, β, is selected to be between 0.1 and 0.4. This allows one to keep a low velocity of airflow through inlets so as not to disturb thermal plumes above heat sources.

The residual static pressure available for outlets, Δp_{exh}, and the required outlet area, A_{out}, can be calculated as

$$\Delta p_{exh} = \Delta p - \Delta p_{inl} \quad (7.288)$$

$$A_{exh} = G_{exh}/\left(2p_{exh}\Delta p_{exh}/C_{pexh}\right)^{1/2}. \quad (7.289)$$

In the case where the area of inlets and outlets is given prior to ventilation design, the static pressure across them can be calculated using the following equations:

$$\Delta p_{exh} = (C_{pexh}/2\rho_{exh})\left(G_o/A_{exh}\right)^2 \quad (7.290)$$

$$\Delta p_{inl} = \Delta p - \Delta p_{exh}. \quad (7.291)$$

7.8.4.2 *Pumping flow mechanism—a special wind driven ventilation*

Recently completed wind tunnel tests showed that vortex shedding could drive airflow through openings in the leeward side of isolated building.[321] Authors called this NV flow type "pumping mechanism". The single-sided ventilation is driven by a combination of steady and unsteady pressure differences. The unsteady pressure component can contribute to the ventilation flow especially when the mean pressure difference is approximately zero. An extreme manifestation of these unsteady contributions occurs when the openings are on the leeward side of the building. In this case the pressure difference at the two openings changes sign with time, usually as a result of vortex shedding from the corners of the building.

Fig. 7.110 presents two key moments of a pumping ventilation flow. This oscillating "pumping" flow mechanism is unique in that it displays a clear periodic behavior, a known characteristic of vortex shedding in the back of bluff bodies.

In addition to the experimental study on the "pumping" flow, CFD simulations were also conducted to investigate the impact of incoming wind speeds and the side ratios of the building on the ventilation rate and the oscillating frequency.[344]

Figs. 7.111 and 7.112 show the 2D building sketch and the computational domain, respectively. The hexahedron building is perpendicular in the height direction and long enough so that two-dimensional assumption can be applied. For any incoming wind speed, only two openings are operated in open states, namely, the combinations of S1 and S8, S2 and S7, S3 and S6, as well as S4 and S5 could be arranged, with the corresponding aperture separations $s' = 0.84$, 0.60, 0.36 and 0.12, respectively. Unsteady RANS method with SST $k-\omega$ turbulent model was used.

As shown in Fig. 7.110, the CFD simulation successfully reproduced the oscillating phenomenon of "pumping" flow. The airflow firstly enters through the upper opening and exits through the bottom one; immediately after half a period time, the stream flow inlet and outlet alternate each other.

Fig. 7.113A illustrates the nondimensional flow rate Q' across the building enclosure as a function of incoming wind velocity U for $s' = 0.84$ (apertures S1 and S8) and Fig. 7.113B the corresponding mean pressure coefficient $|\triangle C_p|$. Detailed analysis of these three plots demonstrate that: (1) nondimensional flow rate Q' increases when the wind flow velocity is less than 5 m/s, but it would not continue to rise when the wind speed is further promoted. This correlation suggests that the wind flow velocity could improve the ventilation rate when the wind speed is not very high. However, it would make little contribution to improve the ventilation rate of backward ventilation in such like building structures when the wind speed is sufficiently high. (2) For the wind-induced natural ventilation, pressure difference could be the major contributor to the airflow rate. Thus $|\triangle C_p|$ shows a similar variation with that of incoming wind flow velocity as the Q' does. This result demonstrates again that a promotion of the incoming wind flow velocity alone could not contribute to the increase of the nondimensional natural ventilation flow rate unless the wind speed is not very high.

The correlation between the wind flow frequency f and incoming wind velocity U is shown in Fig. 7.114. The wind flow frequency increases slowly with the promotion of wind flow velocity at relatively low Reynolds numbers, that is, $U \leq 4$ m/s, and later it is improved dramatically when the wind flow velocity further increases. Anyway, the airflow frequencies have a strong correlation with the wind speed and can be promoted as the wind speed increases.

Observing from Fig. 7.115, the aperture separation has a deep impact on the natural ventilation flow rate, whereas it has almost negligible influence on the wind flow frequency. Carefully securitizing this plot, nondimensional ventilation flow rate increases with aperture

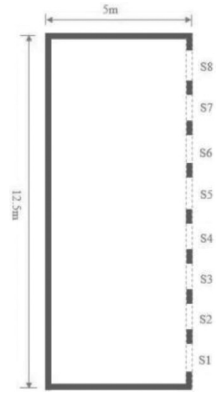


FIGURE 7.111 General presentation of the building.

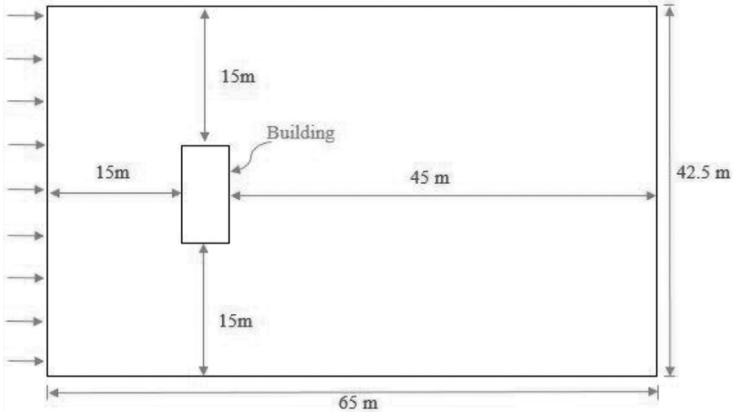


FIGURE 7.112 Computational domain.

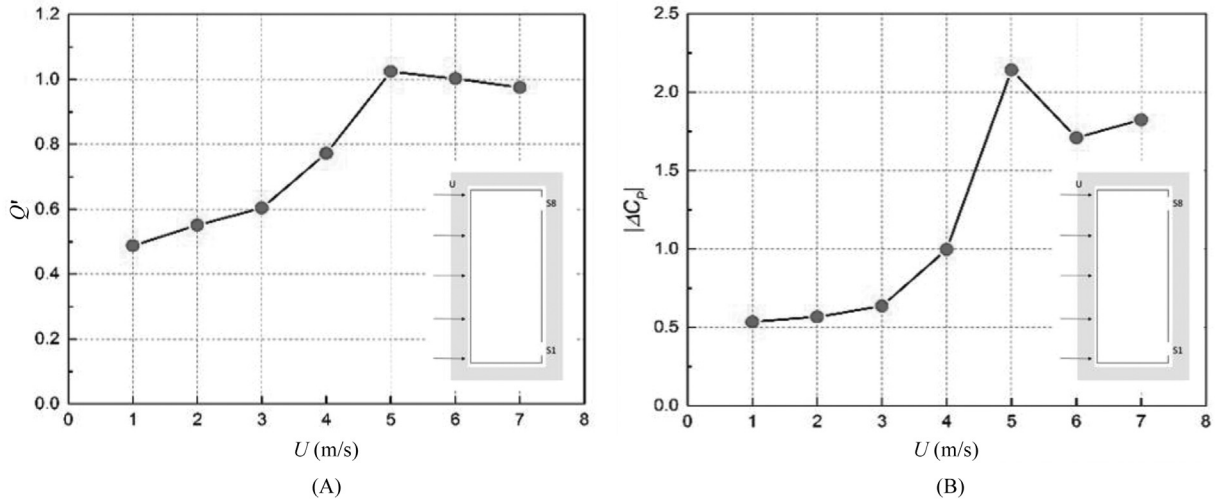


FIGURE 7.113 Variation with incoming wind velocity: (A) nondimensional ventilation rate and (B) modulus of mean pressure coefficient.

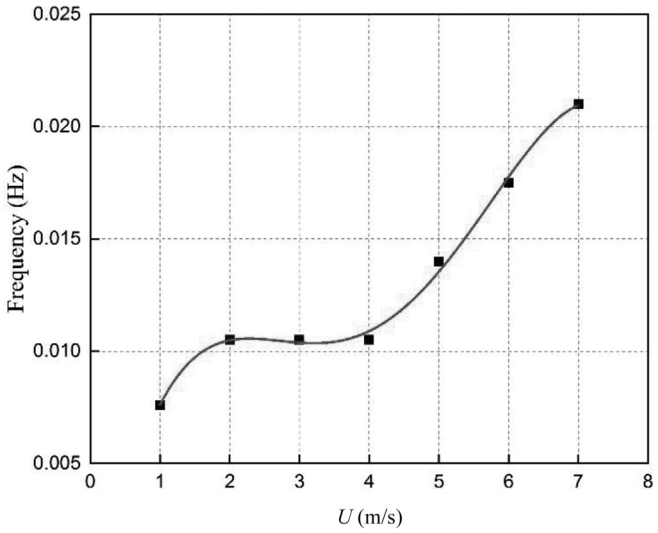


FIGURE 7.114 Variation of the flow frequency with incoming wind velocity.

separation and it reaches maximum when the aperture separation achieves the largest, for all the three different wind velocities. Furthermore the wind flow frequency is primarily determined by the incoming wind flow velocity and it does not change with the aperture separation.

The effects of side ratios on the nondimensional flow rate and oscillating frequency are displayed in Fig. 7.116. Both the nondimensional ventilation rate and the airflow frequency are improved when L/D is increased from 0.50 to 1.25, and the values of them drop a lot when L/D is further promoted to 2.50.

To conclude, the incoming wind speed has a significant effect on both nondimensional ventilation rate and oscillating frequency. The increase of aperture separation only promotes the ventilation rate but has no effect on the oscillating frequency. The influence of side ratio is much more sophisticated, but the effects on both ventilation rate and frequency are synchronous.

The characteristics of the "pumping mechanism" need to be further investigated. The building geometry and opening configurations also need to be well arranged so that the "pumping" flow can better improve the indoor air quality and ventilation efficiency.

7.8.5 Air and contaminant movement between building zones

To evaluate air and contaminant movement within the building, the following classification of building zones is used:

- Building areas separated by physical walls (e.g., halls, rooms, booths) located on the same level. The wall has either intentional apertures or leaks (Fig. 7.117A).
- Building areas separated by physical walls located on different levels (floors).

  Air movement between these zones may occur (e.g., through stairways or ducts) (Fig. 7.117B).
- Building areas separated by air jets located on the same level (e.g., jet-assisted hoods and air curtains) (Fig. 7.117C).
- Building areas within the same room (with no physical partitions) having different requirements for air cleanliness ("clean" and "dirty" areas) located on the same level (Fig. 7.117D).

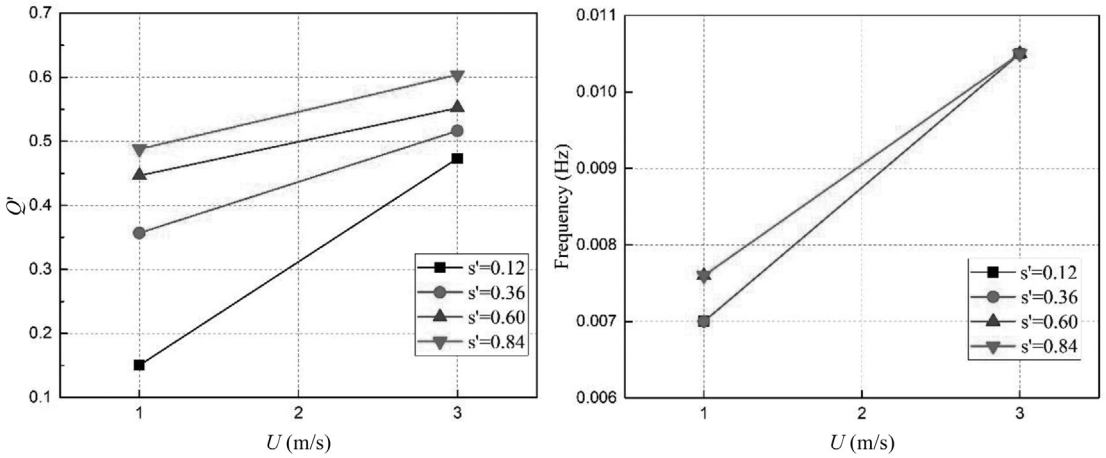


FIGURE 7.115 Influence of aperture separations on nondimensional ventilation flow rate and wind flow frequency, concerning on different wind flow velocities.

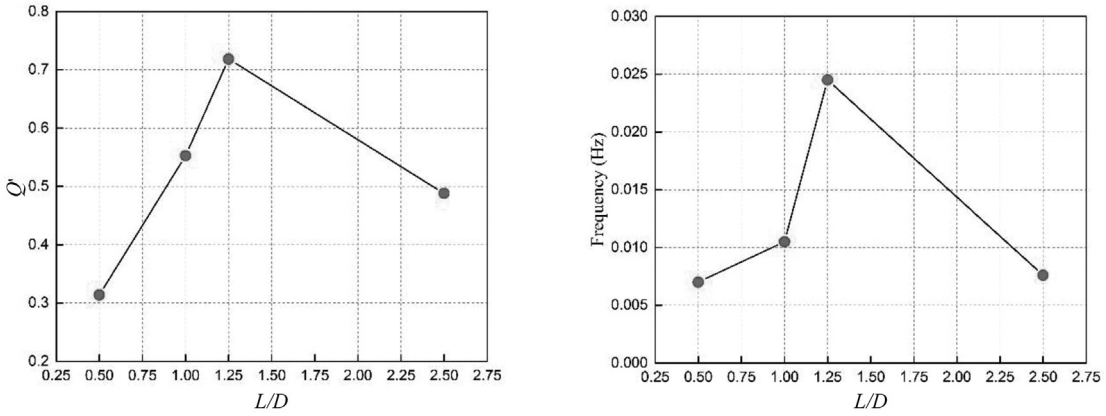


FIGURE 7.116 Effect of side ratio on the nondimensional flow rate and ventilation frequency.

- Zones located within the same room on different levels. These zones have different air temperatures and/or contaminant concentrations (Fig. 7.117E).

Air and contaminant movement between different zones may be caused by one or several of the following mechanisms:

- Static pressure difference between two zones resulting from the unbalanced air supply and exhaust in each zone. Air and contaminants move from the zone with higher static pressure to the zone with lower static pressure (Fig. 7.118A).
- Static pressure difference between two zones resulting from the wind effect on the building envelope (Fig. 7.118B).
- Buoyancy forces creating vertical air movement along the passage between two rooms located on different levels, or thermal plumes creating temperature and contaminant differences between two zones located on different levels of the same room (Fig. 7.118C).
- Turbulent exchange between air in different zones due to energy introduced by supply air jets, convective currents, or moving objects. In this case the resulting mass transferred between the zones equals 0 (Fig. 7.118D).

To control air and contaminant movement between zones, different construction, process-related, and ventilation techniques are used. "Clean" and "dirty" areas can be separated using solid walls, curtains, or partitions (Fig. 7.119A). Ventilation techniques used to separate zones include

- pressure management in different zones (Fig. 7.119B),
- air oasis with specially organized local air supply and exhaust (Fig. 7.119C),
- natural or displacement ventilation systems creating temperature stratification (Fig. 7.119D), and
- air curtains and jet-assisted hoods separating "dirty" zones from "clean" zones (Fig. 7.119E).

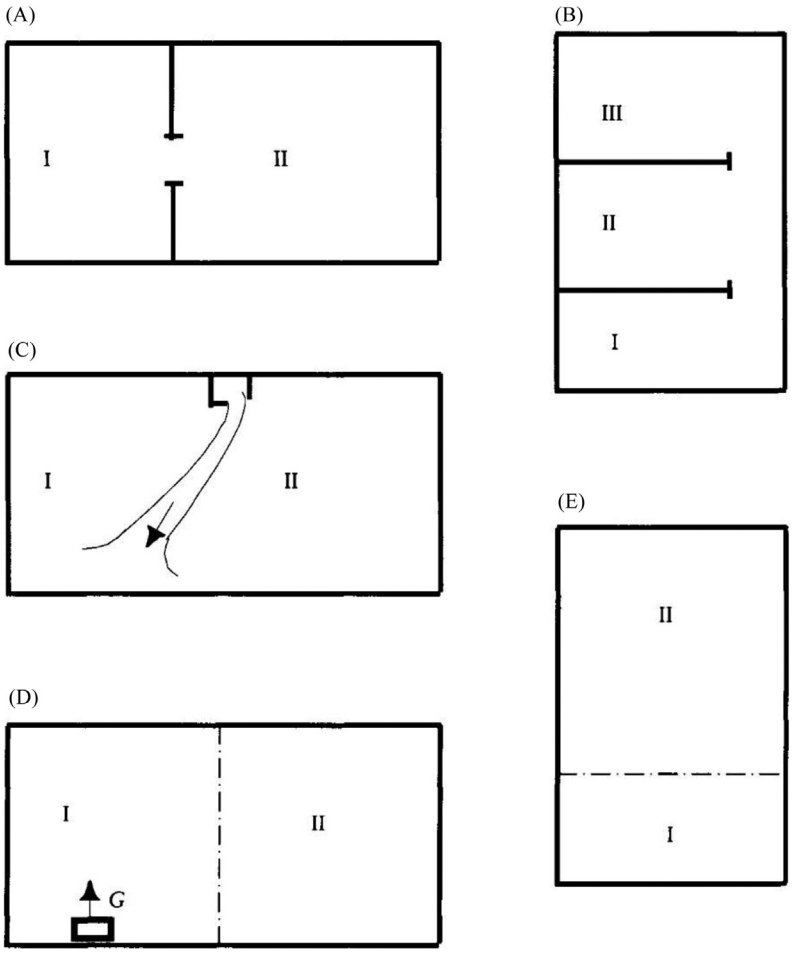


FIGURE 7.117 Zones within a building: (A) located on the same level and separated by physical walls with apertures; (B) located on different levels and separated by physical walls with apertures; (C) located on the same level and separated by air jets; (D) located on the same level with no physical separation; and (E) located in the same room on different levels without physical separation.

Combined construction, process-related, and ventilation measures include "air locks" between two zones (Fig. 7.119F) and process equipment enclosures with air exhaust from the enclosures (Fig. 7.119G).

For pressure management between two zones, refer to the information provided in Section 7.8.2. It is important to mention, that the airflow created by pressure difference between the "clean" and "dirty" zones does not completely prevent contaminant movement from the dirty to the clean zone. Let us consider an example of a two-bay building with an air supply into the high "clean" bay and an air exhaust from the low "dirty" bay (Fig. 7.120A). The high bay has a higher air temperature than the low bay. The stack effect created between the two bays creates contaminant movement from the dirty zone to the clean zone through leaks and other openings in the walls separating these zones. To prevent contaminant movement between the clean and dirty zones, these zones can be separated by an air lock (Fig. 7.120B).

Another factor influencing contaminant and heat transfer from dirty to clean zones against the stable airflow is a turbulent exchange between these zones. This process should be considered in the design of displacement or natural ventilation systems and evaluation of the emission rate of contaminants from the encapsulated process equipment (Fig. 7.121A).

The effect of turbulent exchange between the contaminated air in the process equipment enclosure and the room air (Fig. 7.121B) is described by Elterman.[3] According to Elterman, the air velocity V_{exh} in the process equipment enclosure opening assuring contaminant concentration C_l at the distance l from the opening can be calculated from

$$V_{\text{exh}} = \frac{A}{l} \ln \frac{C_o - C_\infty}{C_i - C_\infty}, \tag{7.292}$$

where C_o is the contaminant concentration under the enclosure, C_∞ is the contaminant concentration in the room at the distant point, C_l is the contaminant concentration at the point located distance L from the enclosure opening, and A is the turbulent exchange coefficient.

Among the most important factors affecting the coefficient A value are the characteristic enclosure opening size and the air turbulence level under the

enclosure. To decrease turbulent exchange with the room air, one can decrease the characteristic opening size (e.g., by inserting a flow equalizer) or install a spigot into the enclosure opening.

7.8.6 Air and contaminant movement in neighborhood scale and urban scale

7.8.6.1 Ventilation in street canyon

A street canyon is said to be symmetrical if the adjacent building heights are equal. The dimensions of a street canyon are expressed by its aspect ratio, H/W (building height to street width) and L/W (building length to street width). Based on the approximate values of aspect ratios, street canyons can be classified in Table 7.34. On the other hand, a street canyon is said to be asymmetrical if the relative height of buildings on the opposite sides of the street canyon are unequal and can be distinguished into two categories depending on the height of the upwind (H_A) or downwind (H_B) buildings (Table 7.34) with respect to the approaching wind flow.

The wind flow pattern inside street canyons depends on their geometry, in particular, the building-height-to-street-width (aspect) ratio ($H = W$, where H represents the building height and W is the street width). Fig. 7.122 shows that the street canyon ventilation structure is different by different aspect ratio (H/W): isolated roughness flow ($H/W = 0.1667$), wake interference flow ($H/W = 0.25$), and skimming flow ($H/W = 1$).

Apart from different aspect ratios, a numerical study by Oke[345] reported the influence of street length on the pollutant dispersion. In short street length, the pollutant concentration along the bottom street is almost uniform. However, for longer street canyons, the pollutants were mostly concentrated at the center of the street canyon. They described the production of a jet effect along the street canyon which helped flush away the concentration from the street canyon. The symmetric street canyon was the earliest street canyon layout to be investigated, based on the abovementioned literature. Some important features of flow structures and pollutant dispersion characteristics identified in the references are summarized in Table 7.35, respectively. In all, the formation of vortex strongly depends on the street canyon ratio (H/W and L/W).

With regard to their geometry, buildings comprise a large variety of configurations, as classified in Table 7.34. One of the most commonly seen in a typical urban city is an asymmetric street canyon. The asymmetric street canyon still holds the unique features of symmetric street canyon, which is the production of recirculation vortex but differs in terms of flow structure (number of vortex, vortex size, and strength) and pollutant dispersion (pollutant concentration and accumulation) depending on the buildings height of

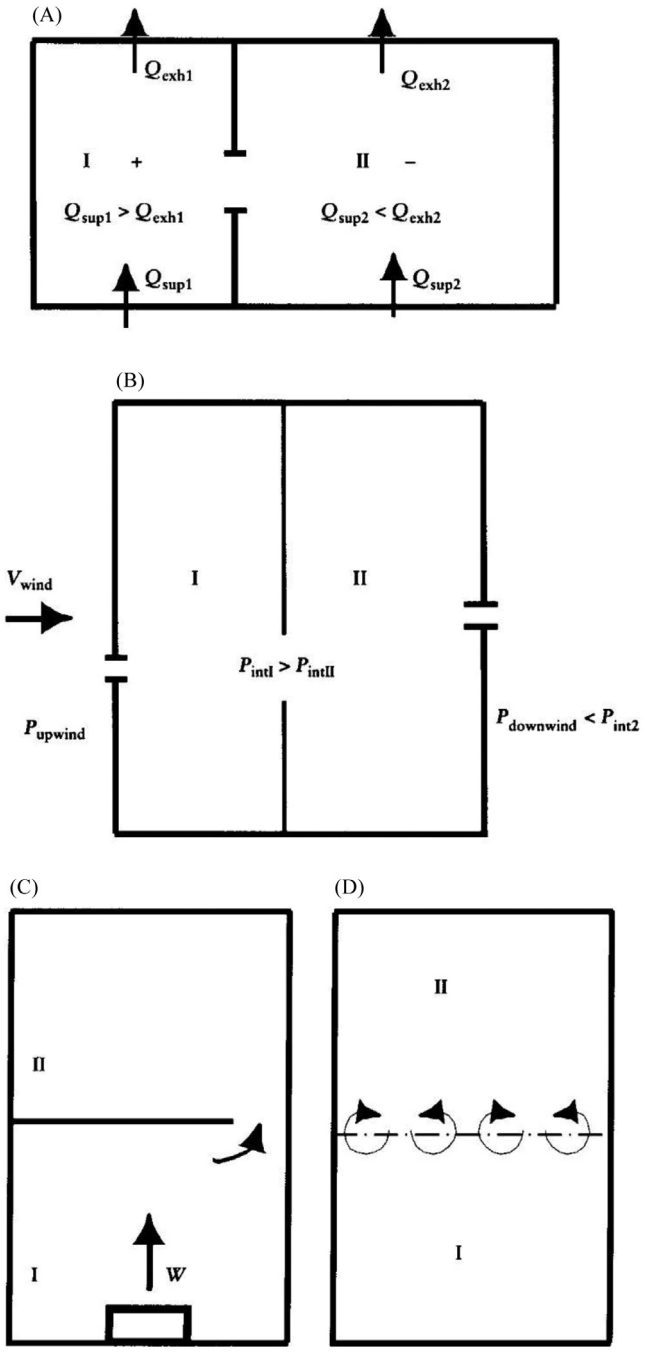


FIGURE 7.118 Mechanisms of contaminant movement between zones: (A) difference in static pressure resulting from unbalanced air supply and return; (B) difference in static pressure resulting from wind effect; (C) buoyancy forces create vertical air movement along the passage between zones located on different levels; and (D) turbulent exchange between air in different zones due to energy introduced by supply air jets, moving objects, etc.

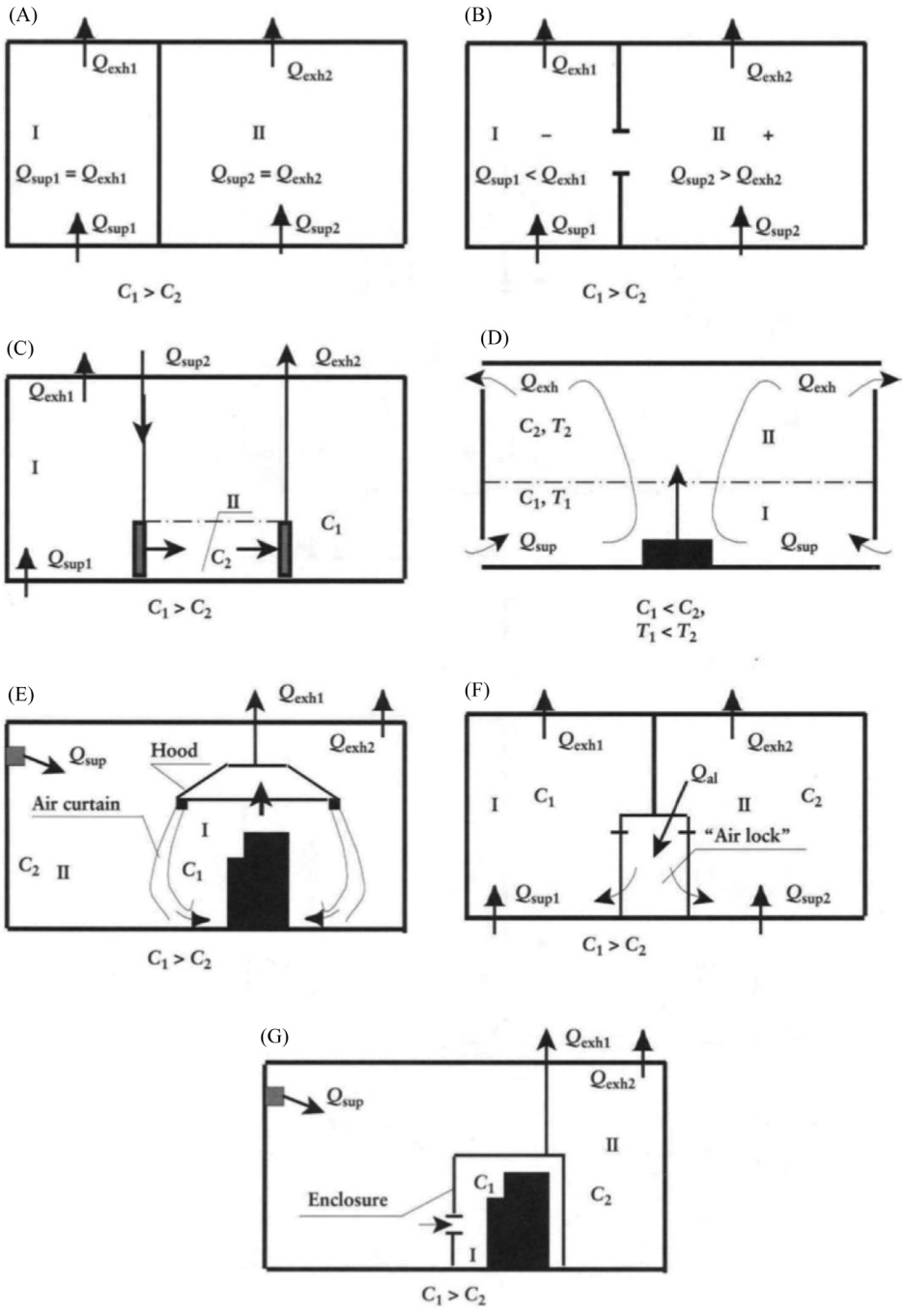


FIGURE 7.119 Construction, process-related, and ventilation techniques used to separate building zones: (A) solid wall; (B) unbalanced supply and return airflow rates; (C) specially arranged local supply and return, creating an "air oasis" with cleaner and cooler air in the desired zone of the building; (D) temperature and contaminant stratification along the room height using a natural or displacement ventilation system; (E) air curtain supplied around the perimeter of the canopy hood, which separates the contaminated process zone; (F) an "air lock" located between two zones; and (G) enclosing process equipment and extracting air from the enclosure.

opposite street canyon and adjacent building height along the street canyon.

Field measurement by Xie et al.[348] in a street canyon of $H/W = 1.1$ showed a distinctive O_3 concentration distribution with chemically produced pollutant as a result of the dispersion by the primary vortex typically found in urban street canyon. They observed that CO, NO, NO_2, and NO_x concentrations decreased on the vertical section of a leeward wall driven by the primary vortex. In contrast, the O_3 distribution on the leeward wall was quickly consumed by the already accumulated vehicle pollutants which contribute to the low concentration. The first CFD method applied to further analyze this phenomenon was reported by Baker et al.,[349] where they simulated reactive gases using LES. The results showed significant impact of reactive pollutant on the distribution of individual levels of NO, NO_2, and O_3 in a street canyon. Moreover, their preliminary tests were consistent with previous results obtained by Xie et al.[348] They

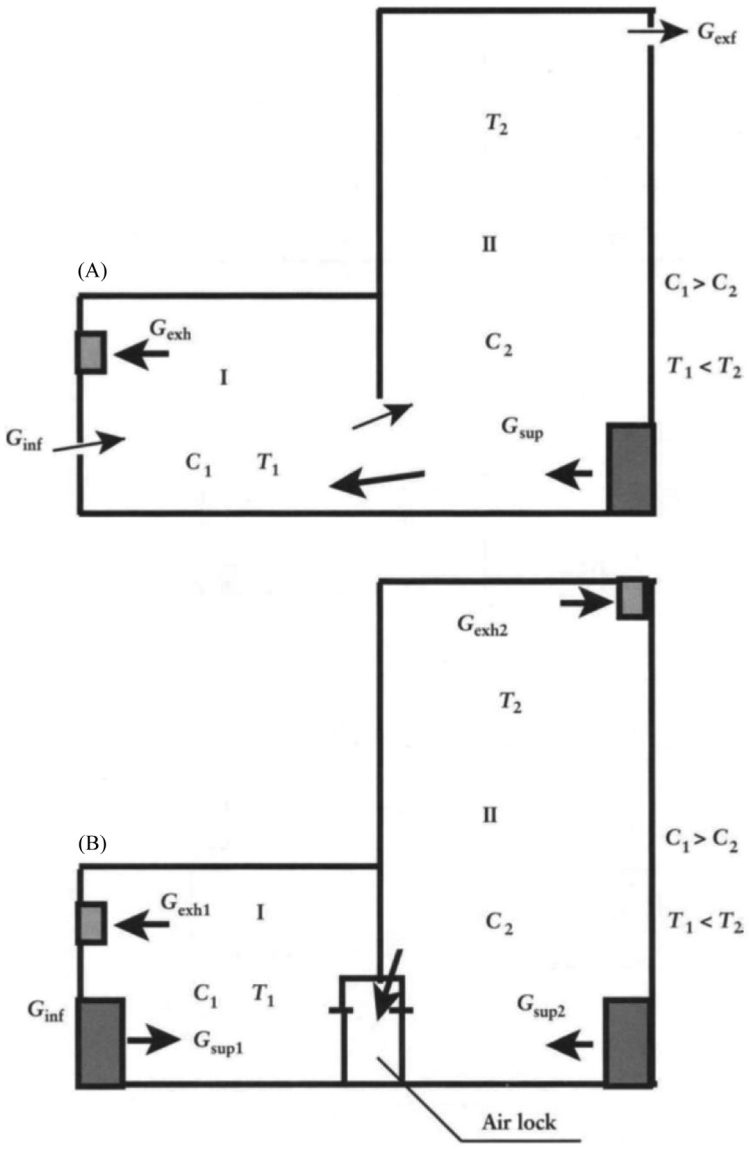


FIGURE 7.120 (A) Contaminant movement between two building zones with different ceiling heights due to the combined effect of unbalanced mechanical systems and the "stack" effect. (B) Prevention of contaminant movement between two zones using an "air lock."

demonstrated that the mixing of chemical species was stable at the center core and the ground level wind-ward corner of the street canyons, while along the wind-ward wall and near emission source, the chemical species were unstable. Using the same method, the concentration of NO_2 is fairly low near the pollution source; because of this, the production by chemical reactions in the canyon could be clearly observed. Fig. 7.123 shows the spatial contours of the O_3, O_2, and NO_2 concentration (in ppb) in the case of NO_x source in the center of the canyons. Grawe et al.[350] extended the investigation of Baker et al.[346] by including the effect of shaded windward/leeward walls in a street canyon, the reaction is varied with the presence of other species. Future prospects of this aspect will be on the inclusion of more chemicals and with different building layouts and solar radiation.

7.8.6.2 Airflow within building arrays

The prediction of effects of wind flow around buildings is of primary importance to natural ventilation of building array. The wind flows in the atmospheric boundary layer over buildings are inherently complex and exhibits a wide range of physical phenomena including large low-speed areas, strong pressure gradients, unsteady flow regions, three-dimensional effects, and confluence of boundary layers and wakes.[351] Predicting the turbulent flow is prerequisite to the dispersion of contaminant in the urban complex.[352]

These flow patterns around buildings within urban canopy layer are influenced by a large number of parameters (e.g., the thickness of the boundary layer, the layout of the buildings, and characteristics of the approach flow) that are identified and investigated in

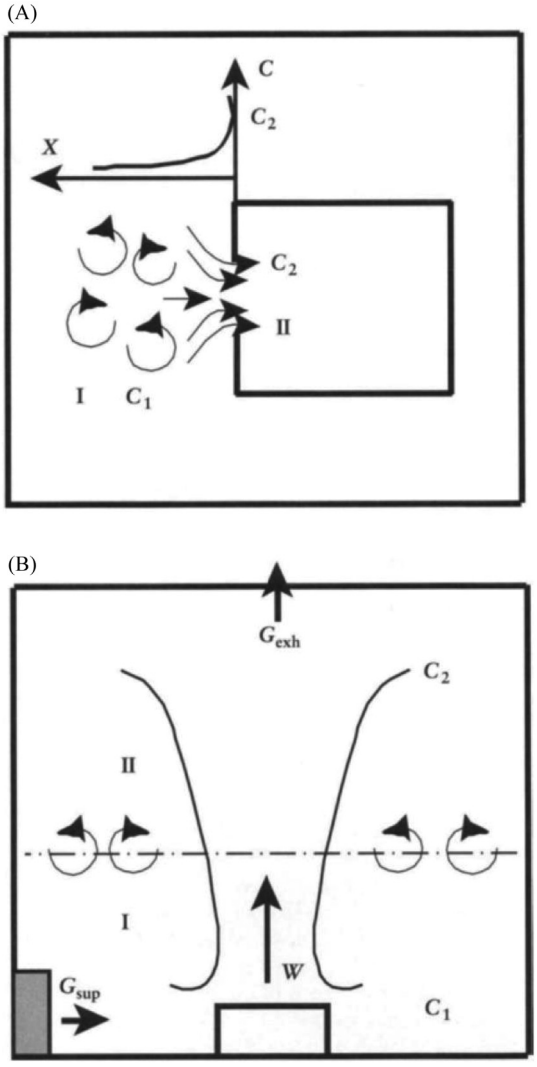


FIGURE 7.121 Contaminant and heat transfer due to turbulent exchange between building zones: (A) contaminant movement against the airflow near the vicinity of local exhaust and (B) heat and contaminant transfer between the lower and upper zones of the building with displacement ventilation.

TABLE 7.34 Classification of street canyons.

Aspect ratio	Classification
$H/W \leq 0.5$	Avenue canyon
$0.5 < H/W < 2$	Regular canyon
$H/W \geq 2$	Deep canyon
$L/W \leq 1$	Short canyon
$1 < L/W < 5$	Medium canyon
$L/W \geq 5$	Long canyon

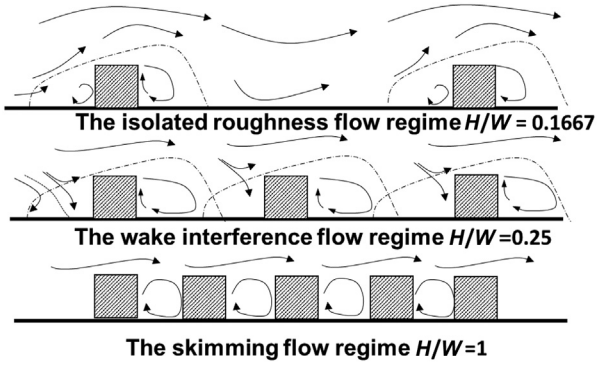


FIGURE 7.122 Street canyon ventilation structure with different H/W.

TABLE 7.35 Flow and vortex structure within symmetric street canyons.

References	Research method	Street canyon shapes	Flow structure
Oke[345]	Review and analysis of literature	Regular canyon	Found three regimes correlated to the geometrical
Sini et al.[346]	CFD (RANS)		The double-eddy circulation was not found
Baik and Kim[347]	CFD (RANS)		Its core being vertically aligned with the other two vortices

CFD, Computational fluid dynamics; *RANS*, Reynolds-averaged Navier–Stokes.

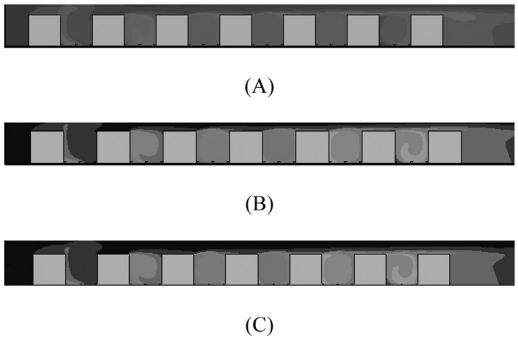


FIGURE 7.123 Spatial contours of O_3, O_2, and NO_2 concentration (in ppb) in the case A_1 of $\mathbf{NO}_x$ source position in middle center ($NO = 11.2\ \mu g/m\ s$): (A) concentration of O_3; (B) concentration of O_2; and (C) concentration of NO_2.

details.[353] Fig. 7.124 shows the airflow pattern within urban canopy with different packaging density. In these cases, the flow patterns are characterized by complex flow phenomena due to the interactions

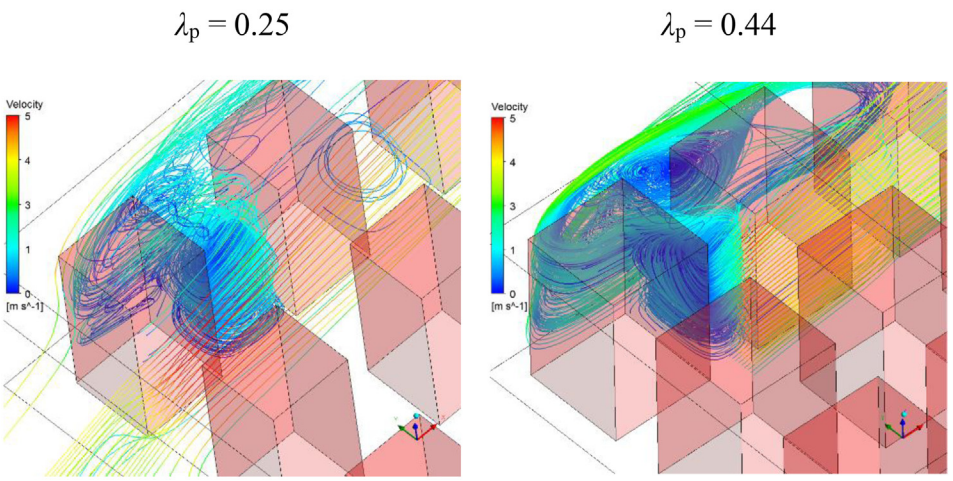


FIGURE 7.124 Streamlines within the urban canopy layer in the $\lambda_p = 0.25$ and 0.44 array.[354]

produced between the various buildings already existing within the site, however some of the results cannot be generalized since they probably include local effects such as secondary structures.[355,356]

7.8.6.3 *Ventilation evaluation of building arrays*

The air exchange and pollutant concentration are good indices for the ventilation performance and air quality at building arrays. There are many ventilation indices that can be used.

Ventilation flow rate and air change rate

Turbulent flows through street openings and open street roofs of the urban canopy are similar with those through large openings or windows in houses. The ventilation flow rates and air change rates due to mean flows and turbulent fluctuations across street roofs and street openings are calculated.

$$Q = \int_A \vec{V} \cdot \vec{n}\, dA \tag{7.293}$$

$$Q_{tur\pm} = \pm \int_{A_{\text{roof}}} 0.5\sigma_w dA. \tag{7.294}$$

where $\vec{V}$ is the velocity vector, $\vec{n}$ is the normal direction of a surface (street openings or roof), A is the area of roof or openings, $\sigma_w = \sqrt{\overline{w'w'}} = \sqrt{2k/3}$ is the fluctuation velocity on street roof based on the approximation of isotropic turbulence ($u' = v' = w'$) that velocity fluctuations in the stream-wise u', span-wise v', vertical directions w' are the same ($k = 0.5(\overline{u'u'} + \overline{v'v'} + \overline{w'w'})$), and A_{roof} is the area of street roofs.

Due to the flow balance, the total flow rate leaving (Q_{out}) the street through openings and roof should equals to that entering the street (Q_{in}). The total advective flow rate is defined as Q_T ($Q_T = Q_{in} = Q_{out}$). Then the air change rate (air exchange per hour for a control volume-ACH) due to the total mean flow rates and the effective flow rate due to turbulent exchange across street roofs were defined as

$$ACH_{turb} = 3600 Q_{turb}/vol \tag{7.295}$$

$$ACH = 3600 Q_T/vol \tag{7.296}$$

where vol is the control volume of the long street, 3600 means that 1 hour is 3600 seconds.

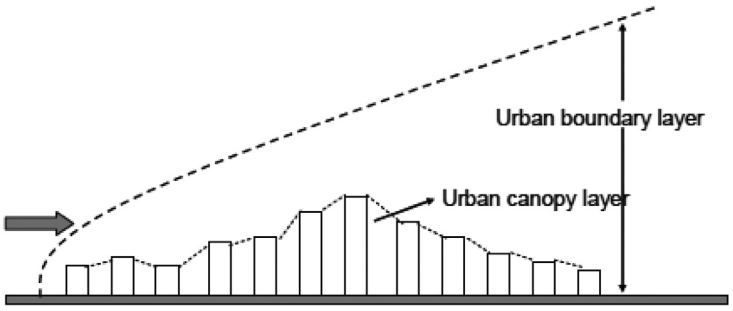
Pollutant removal rate

As pointed out by Fenger,[357] vehicles emissions are significant airborne pollutant sources in street networks of cities. Wind can redistribute and remove these pollutants from urban area. If the flow field is solved, the pollutant transport equation can be solved:

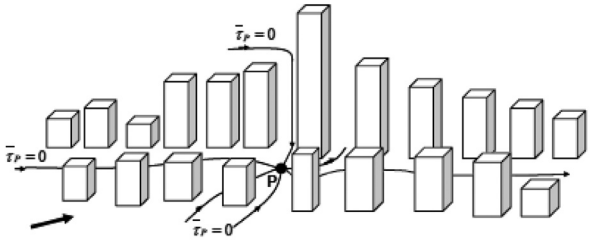
$$\overline{u_j}\frac{\partial \overline{c}}{\partial x_j} = \frac{\partial}{\partial x_j}\left(K_c \frac{\partial \overline{c}}{\partial x_j}\right) + S_c \tag{7.297}$$

where $\overline{c}$ is pollutant concentration (kg/m^3), $\overline{u_j}$ and x_j are velocity components and coordinate components. The diffusivity coefficient of the tra$_c$er gas K_c, which is known as a function of the turbulent Schmidt number ($K_c = \nu_t/S_{ct}$) and varies with different concentration gradient. Obviously the larger of K_c value, the stronger for the pollutant diffusion process. A widely used assumption in previous studies[348,358–360] was adopted that the ratio between the diffusivity coefficient of the tracer gas K_c and the eddy diffusivity of the momentum ν_t is constant, that is, the turbulent Schimidt number $S_{ct} = 0.7$ here as recommended in Sabatino et al.[359] and Riddle et al.[358]

Distribution of concentrations can display the urban air quality. To qualify the process of pollutant dispersion, the normalized pollutant removal rate across street openings and the street roof by mean flows (FA_m^*) could be defined as, Eq. (7.298), that due to the



(A)



(B)

FIGURE 7.125 Definition of urban canopy layer and illustration of the concept of age of air in the urban canopy layer: (A) urban canopy layer and (B) age of air.

turbulent flow across the street roof (FA_t^*), Eq. (7.299):

$$FA_m^* = \int \vec{V} \cdot \vec{n}\, \bar{c} dA/M. \tag{7.298}$$

$$FA_t^* = \int \overline{c'w'} dA/M = \int -K_c \frac{\partial \bar{c}}{\partial z} dA/M. \tag{7.299}$$

where $\vec{V}$ is velocity vector, $\vec{n}$ is the normal direction of street openings or the street roof, c' and w' are concentration fluctuation and vertical velocity fluctuation, M (kg/s) is the total mass release rate of pollutant source near the ground of street network. According to the mass balance of pollutants, the sum of all the normalized pollutant removal rates at all the street openings and roofs equals to 1.0, that is, the total pollutant removal rate equals to the total source release rate. The pollutant removal rates can be used to show the mechanism of pollutant removal at each street openings and street roofs quantitatively.

Local mean age of air and air exchange efficiency

For considerations of ventilation effectiveness, the local mean age of air $\overline{\tau_p}$ at an arbitrarily position P is most pertinent. The air in rural area outside of the urban canopy layer could be assumed fresh (i.e., the age of air is zero), the local mean age of air at Point P in urban air environment represents how long the external clean air can be transported to this point after it enters the urban canopy (see Fig. 7.125). There are several ways of determining the age of air by labeling the air with a tracer.[51] The equation for the mean age of air can be determined from transients (e.g., decay of a tracer) or from stationary concentrations. The transport equation for the mean age of air could be formulated.[361]

$$\overline{u_j} \frac{\partial \overline{\tau_p}}{\partial x_j} - \frac{\partial}{\partial x_k}\left(K_c \frac{\partial \overline{\tau_p}}{\partial x_k}\right) = 1 \tag{7.300}$$

where K_c is the diffusivity coefficient of the tracer gas. Here $K_c = \nu_t / S_{ct}$ and $S_{ct} = 0.7$. Eq. (7.300) can be derived from the pollutant transport equation with uniform pollutant source mathematically.[51]

As long as the flow field is known and the boundary condition for the age of air is defined, the age of air may be solved numerically. But if defining $\overline{\tau_p} = 0$ at upstream domain inlet, the age of air in far downstream above the urban canopy will be very "large" in value however the air is actually fresh. It is obviously unreasonable. Numerical studies of the age of air were performed in mechanically ventilated buildings,[362] in which the boundary condition for the age equation is relatively easy to be defined. The ventilation performance could also be evaluated by the homogeneous emission method, which was originally developed for the tracer gas techniques[51,361] in indoor experiments.

In the homogeneous emission method of tracer gas technique,[51] the local mean age of air can also be determined from stationary concentrations generated by a homogenous emission source. This means that the release rate R_s (kg/s) at each location P (theoretically arbitrarily small) within the flow domain must be the

same. Instruments have finite dimensions and, likewise, CFD approximations must be based on grid dimensions that are finite. Therefore in practical applications, a homogeneous emission implies that the release rate (R_s) at location P must be proportional to the local control volume surrounding this location. Given the effective uniformly distributed emission rate R_s at each point of the entire control volume and the corresponding tracer concentration field, the local mean age of air $\overline{\tau_p}$ can be calculated from the local concentration c (kg/kg) as:

$$\overline{\tau_p} = \bar{c}/R_s \tag{7.301}$$

The lower the local mean age of air is, the better the local ventilation will be. The practical applicability of relation Eq. (7.301) is that it can be inverted to predict the concentration from a homogeneous emission source when the local mean age of air is known.

The difficulty in the definition of open boundary condition for the age of air disappears. One can predict or measure the spatial distribution of the local mean age of air to characterize the spatial variation of ventilation effectiveness in an urban air environment. The homogenous emission method can be used to an urban air environment and the difficulty in the definition of open boundary condition for the age of air disappears. Indeed, it represents the worst situation in which the entire volume of urban canopy layer is covered by vehicles with their engines producing, effectively, a homogeneous emission at each point. This approach works well within a control volume when the air exits the control volume boundary without returning. A tight room ventilated by a mechanical ventilation system is an example of a control volume where the air never returns. A complication in urban environments is that there are "open roofs" at the top. Strong vertical turbulence can lead air and tracer gas to reentry the given control volume after exiting it. Turbulence introduces vertical fluxes on open roof both upwardly and downwardly. A nice example[363] using a LES prediction of 2D street canyon models, reported this downward flux (reentry) of pollutant is smaller than the upward flux (removal) of pollutant by turbulence on roof due to the fact that the concentration above roof is relatively low. The downward flux implies that some "air parcels" reentry the below volume, that is, visiting this volume at least twice. The number of times an air parcel visits the domain is defined as the visitation frequency (VF). Bady et al.[360] studied the VF using numerical simulations of urban air environments.

In case without returning air through roof and openings, the effective mean age of air is equal to the true mean age of air. The actual flow conditions within a given urban area will determine whether or not the effective mean age of air is equal to the true age of air. In the idealized city models, there is little air reentries urban domain through street openings but a small value of downward influx through the street roof exists in some cases. If the polluted air above roof returns, the local concentration will necessarily increase, thus the effective mean age of air will increase and the ventilation performance will decrease. That air parcels return through the roof does not limit the practical applicability the concept of effective mean age because the effective mean age can be used to predict the concentration due to a homogeneous emission.

In addition, the following concepts for building ventilation[51] are also introduced here for city ventilation. The turn-over time of the air (τ_n) is defined as the shortest possible time it takes to replace all the air in the urban air environment:

$$\tau_n = Vol/Q_T \tag{7.302}$$

The air change rate (ACH) per hour has a relation with the turn-over time as:

$$ACH = 3600Q_T/Vol = 3600/\tau_n \tag{7.303}$$

If the residence time could be defined as the age when the air leaves the city, then the mean residence time of the air in a volume is twice the spatial mean age:

$$\overline{\tau_r} = 2 < \overline{\tau_p} > \tag{7.304}$$

In general, the turn-over time does not equal to the mean residence time and the air change rate does not mean the air in the entire volume can be exchanged ACH times. So, the air exchange efficiency is defined as the efficiency of airflow flushing a volume with external air. It can be calculated as the ratio between the turn-over time (τ_n) and the mean residence time (τ_r):

$$\varepsilon_a = \tau_n/\tau_r = \tau_n/(2 < \overline{\tau_p} >) \times 100\% \tag{7.305}$$

Obviously there is a relationship between these parameters.

$$2 < \overline{\tau_p} > /\tau_n = \tau_r/\tau_n \geq 1 \text{ or } \varepsilon_a \leq 100\% \tag{7.306}$$

In an ideal piston flow, the whole volume is "flushed" by the total flow rate (not just a fraction), the time it takes to exchange all the air within the space is exactly equal to the turn-over time, so the spatial mean age is half of the turn-over time and the air exchange efficiency is 100%. In a system with perfect mixing, where the conditions are always uniform at any given moment (for example, the concentration is the same at each point), the spatial mean age equals to the turn-over time and the air exchange efficiency is 50%. In a short-circuiting system, a large fraction of the airflow resides a short time (much less than the turn-over

time) without passing the whole occupied volume, the air exchange efficiency may be less than 50% when only little fraction of flow rate flushes the total volume. In general, turbulent flows in urban area consists of piston flows, perfect mixing flow and short-circuiting flows, considering that the approaching flows may be weakened and some air leaves the urban canopy due to the blockage of buildings at the same time recirculation flows commonly exist in urban area.[364,365]

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CHAPTER

8

Room air conditioning

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8.1 Introduction

Air conditioning design for an industrial space must be focused on providing a safe and comfortable environment with a low health risk for workers. The main goal for indoor climate design for an industrial hall is that the selected solution be effective for workers with respect to air quality and thermal comfort. Thus prescribed design criteria must be met. Second, the climatization design should be as energy efficient as possible, which normally involves the minimization of outdoor airflow rate use through application of source control, local ventilation, and efficient space ventilation.

In Chapter 2, Terminology, the terminology of zones, systems, and basic strategies for room air conditioning are explained.

The scope of this chapter is to show how to select a strategy and to make calculations to provide efficient room air conditioning for an industrial enclosure. Design examples for simplified scenarios include different ways to condition industrial halls.

Upon starting design for a large enclosure, expected use should be described by informing and questioning the building owner and future users in order to locate zones of occupancy, the related activity level and clothing, and how that is expected to vary over a year. This is also the time to discuss and decide on adequate criteria for acceptable indoor air quality and thermal comfort with national health and building codes and recommendations and international standards.

Assumed heat and contaminant emissions inside the room should be estimated. Also, contaminants being admitted into the room by airflow and heat admitted by conduction and radiation through the room envelope should be assessed.

In the industrial ventilation design the possibilities of reducing or eliminating emission sources and using local extracts to avoid spreading to the rest of the room should be addressed.

The prevention of unnecessary heat loss or load, causing poor thermal comfort due to large glazing areas and infiltration, should also be considered.

The use of local climatization should be considered one way of reducing the requirement for heating, cooling, or ventilating the entire enclosure volume.

8.2 Basis for air conditioning design

Upon starting design for an industrial enclosure, expected use should be described by informing and questioning the building owner and future users.

A scenario of the planned industrial enclosure should be sketched. The scenario should describe:

- the different steps of the flow and batch processes of the industrial production;
- space demands for machinery, tools, working, and stocking areas for raw materials and products;
- demand for worker involvement in each part of the industrial process;
- emission of heat and contaminants from each part of the industrial process; and
- environmental requirements for each part of the industrial process.

The factory scenario must be updated with new information as the design process evolves.

8.2.1 Industrial process description

8.2.1.1 Stages of the industrial production process

The different steps of the flow and batch processes of the industrial production must be roughly described

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early in the design process. A more detailed description may be needed in regions where heat and contaminants are released. Production design engineers are likely to provide the information needed.

8.2.1.2 Space demands for the production process

Space demands for machinery, tools, work areas, and for stocking raw materials and products must be outlined. The logistics for the production line are often the basis for this consideration.

8.2.2 Requirements for indoor environment

8.2.2.1 Air conditioning demands for human occupancy

Requirements for indoor environment quality must be discussed and decided before the air conditioning design is performed. Criteria for acceptable indoor air quality and thermal comfort must be set.

Health hazard due to exposure to contaminants by inhalation is the most important issue. International or national health and building regulations or codes and recommendations are used as basis for the discussion of what requirements should be used for design. In regulations, worker exposure limits for airborne gases are normally expressed as:

- time-weighted average concentration, TWA, based on a 40-hour work week;
- short-term exposure limit, which can be exceeded for periods of up to 15 minutes; and
- threshold limit value, TLV, a concentration level not to be exceeded any time.

Combinations of different substances in the air might be more harmful than the added effect of the two substances.

Most industrial companies would like to set air quality requirements regarding health hazard at a lower concentration level compared to values found in regulations. For example, when a TWA concentration is given as x kg/m^3 of air for a certain contaminant, the required concentration level for design of general ventilation in an industrial enclosure could be chosen as one-third of that value. This value can be considered the target value for the first design.

Target values must be rough for early design and must be refined for later stages in the design process. The main reasons for this are that details often are scarce and that calculation methods are rough at early design. One example of selection of target values is shown in Table 8.1. Some target values are not possible to consider at early design but should nevertheless be satisfied when production has commenced. General advice

TABLE 8.1 Example of target values for air quality at different stages of design.

Air quality for breathing zone	First design	Main design	Evaluation
Odor from production processes, materials, and humans[a]			
Humans	Use at least $Q = 7$ L/(sp) $N + 1.4$ L/(sm$^2 A_f$), where N is the number of persons and A_f is the floor area	CO_2 level < 500 ppm over outdoor level	
Processes and materials	Avoid release of unpleasant odor from processes and materials	Use local extract	Tracer gas experiment with mathematical or physical model to check efficiency
Humidity	Avoid condensation on surfaces	20% < RH < 60%	
Particles	Avoid release of particles	Use local extract	< 20 μg/m^3 for $PM_{2.5}$[b] < 40 μg/m^3 (d < 0.05 μm) < 90 μg/m^3 (all diameters)
Health-hazardous gases			
Radon	Check ground on construction site		<200 Bq/m^3
TVOC	Avoid moisture and release of fuels and similar contaminants		<400 μg/m^3
Other substances	Use environment-friendly substances Use one-third of regulation TWA	Concentration in breathing zones lower than chosen target value	Regulation concentration limits given by local authority

[a] *European Committee for Standardization. CEN/TC 156 Ventilation for Buildings: Design Criteria for Indoor Environment. CR 1752,1998.*
[b] *$PM_{2.5}$ is the concentration of the fraction of particles where at least 50% (by weight) have an aerodynamic diameter less than 2.5 μm.*
RH, Relative humidity; *TWA*, time-weighted average concentration.

TABLE 8.2 Example of target values of thermal conditions at different stages of design.

Thermal conditions for zone of occupancy	First design	Main design	Evaluation
General comfort[a]		Relative air humidity RH < 70%	RH < 70%
Winter (1.1 m above floor)	Air temperature 21°C	Operative temperature 22.0°C−2.0°C	PPD[b] < 10% (−0.5 < PMV < 0.5)
		Air speed < 0.16 m/s	
Summer (1.1 m above floor)	Air temperature 25°C	Operative temperature 24.5°C ± 1.5°C Air speed < 0.22 m/s	PPD < 10%
Local discomfort			
Draft			
Winter (ankles 0.1 m)		Air speed < 0.16 m/s	PPD < 20%
Summer (neck 1.8 m)		Air speed < 0.20 m/s	PPD < 20%
Vertical temperature gradient		<3°C (0.1−1.1 m above floor)	PPD < 5%
Radiant asymmetry		<10°C for cold wall	PPD < 5% with respect to cold wall, warm wall, cold ceiling, and warm ceiling
		<23°C for warm wall	
		<14°C for cold ceiling	
		<5°C for warm ceiling	
Hot or cold floors		19°C−26°C at surface	PPD < 10%

[a]*ISO EN 7730, 2005. Moderate thermal environments, determination of PMV (predicted mean vote)[and PPD indices and specifications of the conditions of thermal comfort, International Standard Organization, Geneva.*
[b]*PPD—percentage of people dissatisfied.*
RH, Relative humidity.

should be used during the design procedure to avoid such exposure problems at the future work condition.

The company must decide the air quality target values for breathing zones in the work environment after discussion with industrial hygienists and ventilation engineers.

Design criteria with respect to health-hazardous gases could alternatively be given as an accepted daily exposure of contaminant (for each hazardous substance, or a combination of substances).

Hazards from explosions, spills, and extreme working conditions should be considered according to national regulations.

Target values for thermal conditions at different design stages must also be considered. One example, for a scenario similar to regular office work, is shown in Table 8.2. The activity level and clothing insulation of the workers must always be taken into consideration when target values are chosen. Different target values other than the ones used in this table could be required, for example, for very cold and very hot and humid environments.

The definition of the breathing zone and of the zone of occupancy must be revised at each stage in the design process to make the climatization design effective and efficient.

Accepted acoustical conditions should be considered according to national regulations.

8.2.2.2 Conditioning demands other than for human occupants

Industrial processes often require environmental conditions within certain limits.

Air quality requirements may be set as content of particles and as content of chemical substances. Other requirements for temperature, humidity, and air speed could be set. One example is the spray painting process, which is very dependent on a minimum relative humidity level to obtain a high-quality result. Refer to Chapter 2, Terminology, and Chapter 6, Target Levels, for further information.

8.2.3 Architectural design for an industrial enclosure

Once the information on the industrial processes and space requirements, the indoor climate requirements for workers, machinery, and products, and a lot for the factory have been obtained, then the architectural design of the industrial enclosure can be accomplished. A complete factory or industrial building may consist of several

TABLE 8.3 Example of worker behavior during a typical workday.

Time period (h)	Accumulated time (h)	Work description	Breathing zone position			Activity description	Activity level (met)	Pulmonary ventilation[a] rate (kg/h)
			X (m)	*Y* (m)	*Z* (m)			
0.5	0.5	Adjusting machinery	27.7	18.6	1.65	Light work	2.0	1.46
0.5	1.0	Tube-forming quality check	22.9	19.4	1.65	Standing	1.2	0.88
0.5	1.5	Tube-forming quality check	22.9	19.4	1.65	Standing	1.2	0.88
0.5	2.0	Resting outside production hall	–	–	–	Seated	1.0	0.73
0.25	2.25	Checking paint station	17.4	18.6	1.65	Standing	1.2	0.88
0.25	2.5	Checking paint station	17.4	18.6	1.65	Standing	1.2	0.88
0.5	3.0	Checking paint station	17.4	18.6	1.65	Standing	1.2	0.88
0.5	3.5	Checking paint station	17.4	18.6	1.65	Standing	1.2	0.88
0.5	4.0	Checking paint station	17.4	18.6	1.65	Standing	1.2	0.88
0.5	4.5	Checking paint station	17.4	18.6	1.65	Standing	1.2	0.88
0.5	5.0	Checking paint station	17.4	18.6	1.65	Standing	1.2	0.88
0.5	5.5	Furnace adjustment	20.5	18.9	1.65	Light work	2.0	1.46
0.5	6.0	Furnace adjustment	20.5	18.9	1.65	Light work	2.0	1.46
0.5	6.5	Resting outside production hall	–	–	–	Seated	1.0	0.73
0.5	7.0	Resting outside production hall	–	–	–	Seated	1.0	0.73
0.5	7.5	Checking paint station	24.8	19.4	1.65	Standing	1.2	0.88
0.5	8.0	Checking paint station	24.8	19.4	1.65	Standing	1.2	0.88

[a]*Pulmonary ventilation can be found from* $Q_V = 0.0070M$ *(kg/h).* $M = 104.4$ *W for an adult person at 1 met.*

enclosures where industrial processes are run. One heating and ventilation plant may service several enclosures, or there may be a separate system for each enclosure.

The architectural design of an industrial enclosure must primarily satisfy the requirements for space and functions to accomplish the industrial process. Second, the architectural design should support the goals for good indoor environment and low energy consumption. Good indoor environment and low energy consumption can, for example, be supported by use of windows for day lighting and avoiding problems with down-draft and direct solar radiation.

Development in industrial processes and in work environment during the lifetime of an industrial hall should be predicted and planned for during the architectural design.

Utilization of natural ventilation for air conditioning will normally interact with the architectural design, and it should be considered early in the design process.

8.2.4 Worker involvement in the production process

Demand for worker involvement in every part of the industrial process must be outlined. Then zones of occupancy, the related activity level and clothing, and how that is expected to vary over a typical day and year should be described.

Daily work procedure characterization for each worker must be outlined early in the design process.

At a later stage in the design process, when more detailed information is available, the following parameters should be described as a function of time for each worker: breathing zone position, activity level, and clothing value.

One example of how such information can be presented is shown in Table 8.3. Breathing zone location and activity level for one specific worker during a typical workday are listed.

The actual industrial hall has the form of a box of length, width, and height 48.0, 20.0, and 4.4 m, respectively. The origin of the coordinate system is located in the lower northwest corner of the room. The importance of mapping the breathing zone location can be realized when there is a coincidence of a high contaminant level and a high activity level combined with a long period of occupancy. As the body increases its pulmonary rate to absorb more oxygen, an airborne contaminant is likely to be absorbed to a higher degree in the body.

TABLE 8.4 Heat and contaminant release to the air in an industrial hall.

Source	Position X, Y, Z (m)	Power (W)	Contaminant (kg/s)	Comment
Furnace 1	21, 18, 1.4	30.0×10^3	–	Continuous/70% in local extract
Furnace 2	23, 18, 1.4	10.0×10^3	–	Continuous/60% in local extract
Furnace 3	26, 18, 1.4	17.5×10^3	–	Continuous/80% in local extract
Painting station	17, 19, 1.4	–	Paint aerosol 4×10^6	Substance/TLV = x mg/m^3
Press station	28, 18, 1.5	–	Oil vapor 1×10^{-6}	Substance/TLV = y mg/m^3

TLV, Threshold limit value.

8.2.5 Load calculation

8.2.5.1 Heat and contaminant emission

An estimate of heat and contaminant emission to the room air is needed early in the design process. Table 8.4 presents typical parameters of interest. At a later stage in design the position and characteristics of releases are needed such as:

- heat, described as power, as temperature on the described surface, or as advection of heat along with substance flowing out of the production process;
- ratio of radiation and convection of the total heat load;
- momentum and density of flow released from openings and from processes;
- moisture, as emission of water and vapor from the process;
- contaminant emission, such as mass flow of different substances flowing out of the process; and
- detailed list of the actual contaminant, including properties, health effects, and odor.

It is also useful to make a steady-state balance with respect to airflow, heat flow, and mass flow of substances into and out of the enclosure. See Chapter 7, Principles of Air and Contaminant Movement Inside and Around Buildings, for details with respect to calculation of heat and contaminant emission.

Assume the use of mechanical ventilation with full mixing condition in the room, and calculate a rough estimate of ventilation airflow found by assuming dilution of contaminants released to the room air to one-third of the TLV level given for the substance in question.

8.2.5.2 Room envelope characterization

In order to analyze the conditions for air conditioning, the following information must be specified for the industrial enclosure:

- geometric dimensions;
- orientation, *U*-values of envelope, wall and roof, *g*-values of window (solar load), and thermal mass of the different construction elements;
- position of openings for possible infiltration or exfiltration; and
- outline of expected opening of doors, windows, and hatches.

8.2.6 Characterization of room airflow and thermal conditions based on industrial production process and envelope

8.2.6.1 Design winter conditions

In the design of the system for winter conditions:

- Use the winter design temperature at the site along with the required indoor temperature level.
- Estimate the heating power needed to keep that indoor temperature at steady-state conditions without any internal heat gain. Assume the use of mechanical ventilation with full mixing ventilation in the room; that is, use the rough estimate of ventilation airflow given in Section 8.2.

8.2.6.2 Design summer conditions

In the design of the system for summer conditions:

- Check summer design outdoor temperature and humidity.
- Use the summer design room air temperature and humidity.
- Estimate maximum temperature increase in the room, taking into consideration the actual heat release from people and machinery, shading for solar radiation, assuming no thermal mass, and using a typical airflow for ventilation in fully mixed conditions.

8.2.7 Analyses and actions to be considered prior to performing room air conditioning design

The following questions should be considered:

- Is replacement of contaminating substances possible?
- Is containment of production processes possible, to avoid contaminants and heat being released to the room air?
- Is automation of production processes possible, to avoid workers being exposed?
- Is use of local extracts possible, to avoid contaminants and heat being released to the room air, and how efficient are extracts?
- Is use of personal protection gear for workers possible?
- Is use of local fresh-air supply at working stations possible?
- Is use of local thermal conditioning possible?
- Can changed working procedures reduce exposure to contaminants?
- Can clothing be adjusted to correct thermal dissatisfaction problems?
- Can activity level be adjusted by changing working routines?
- In midsummer, when overheating is a problem, can an afternoon shift be changed into a night shift?

8.3 Effective and efficient ventilation

8.3.1 Ventilation efficiency indices

Effective air distribution in ventilated rooms and proper quantity of conditioned air is essential for creating comfortable conditions, removing contaminants, and reducing initial and operating costs of air conditioning and ventilation systems. The degree to which a ventilation system fulfills ventilation requirements is described in the literature in terms of "ventilation effectiveness," "ventilation efficiency," "ventilation performance," etc. Liddament in his review of technical information related to ventilation effectiveness stated that "the subject of ventilation effectiveness is made unnecessarily complex by the lack of uniformity in terminology. Frequently, terms are interchanged or different terms are used to describe the same concepts."[1]

Ventilation efficiency has traditionally been defined as the ratio between contaminant concentration in the occupied spaces and the concentration in the exhaust air.[2] Sandberg and Skåret[3] differentiate between the terms "air change efficiency" and "contaminant removal effectiveness." Air change efficiency is "a measure of how effectively the air present in a room is replaced by fresh air from the ventilation system," whereas "contaminant removal effectiveness" is "a measure of how quickly an air-borne contaminant is removed from the room." A third similar criterion that is used is "contaminant removal efficiency."[3]

In the current review, the term "effectiveness" of air distribution will be used to describe the ratio of the occupied zone area (where thermal comfort and contaminant concentration are within ranges required by standards and codes) to the total occupied zone area. This hygienic criterion allows one to judge how well the HVAC system fulfills its main task—creating thermal comfort conditions and controlling contaminants in the occupied zone.

Industrial halls are typically large enclosures—indoor spaces that typically comprise one or more zones of occupancy. A large height combined with heat sources often results in room airflow patterns controlled by buoyancy flows.

A characteristic of many industrial halls is that zones of occupancy take up only a small proportion of the room volume and height. In addition, the flows are normally buoyancy dominated. This results in a vertical temperature stratification that can be utilized for room air conditioning design in order to achieve effective climatization along with low energy consumption.

A process is found effective when "it produces a desired effect," for example, ventilation in a room is effective when it produces sufficiently good air quality in the breathing zone. However, that air quality may be achieved without the ventilation being efficient. A ventilation process is considered efficient only when it is "effective with little waste of effort,"[4] that is, being cost-effective.

Being cost-effective is normally equivalent to achieving a ventilation design where the indoor climate meets required specifications (target values), applying a low rate of supply air, along with avoidance of heating or cooling. That way, climatization would be effective as well as efficient.

A review of ventilation efficiency indices is given below. Complete mixing ventilation is the comparative basis for all efficiencies, with the value 1.0. That may be useful, as complete mixing ventilation is often assumed in early design phases.

Which ventilation efficiency index should be chosen for assessment depends on the actual scenario in the enclosure. The contaminant removal effectiveness can be used when emission data for contaminant sources are available.

Air exchange efficiency indices can be used for cases where no or little information on sources is available, whereas ventilation efficiency, which concerns workers, can be used where very detailed information is available on sources and activities.

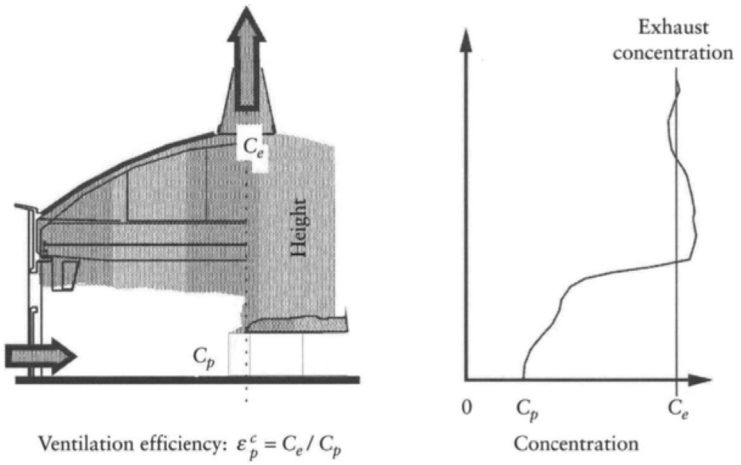


FIGURE 8.1 The definition of "ventilation efficiency."

8.3.2 Contaminant removal effectiveness

As mentioned above, the traditional definition of ventilation efficiency or, in approved terms, "contaminant removal effectiveness," is the ratio between contaminant concentration in the exhaust air and the concentration at a point in the occupied space, that is,

$$\epsilon_p^c = \frac{C_e}{C_p}, \tag{8.1}$$

where ϵ_p^c is contaminant removal effectiveness in the zone of occupancy; C_e is contaminant concentration in the exhaust air; and C_p is contaminant concentration at a point in the zone of occupancy.

In other words, this contaminant removal effectiveness is a measure of how much cleaner the air is in the occupied spaces than in the exhaust. See Fig. 8.1.

When detailed information on heat and contaminant sources is available, assessment of design is improved by evaluating the effectiveness of contaminant removal achieved by space ventilation. The set of contaminant removal effectiveness indices in Table 8.5 is given in accordance with contemporary use of indices.

The basis of comparison is still the complete mixing scenario, where the concentration of the contaminant in question is homogeneous throughout the room, and equal to the value in the exhaust C_e. All concentrations are net values, that is, rated above values at the supply opening. It can be shown that

$$\epsilon^c = \frac{C_e}{\langle C \rangle} = \frac{\tau_n^a}{\overline{\tau}_e^c} \tag{8.2}$$

for steady-state conditions.[3] $\tau_n^a = V_r/Q$ is the nominal time constant of the ventilation airflow, where V_r is the volume of the room, Q is the airflow rate, and $\overline{\tau}_e^c$ is the mean age of the contaminant in the exhaust.

Application of contaminant removal effectiveness indices is relatively simple for scenarios with one or a few dominant contaminants being released. That is often the case in industrial malls. Where there are many polluting substances to consider, the contaminant removal efficiency should ideally be evaluated for each one. Consequently, applications for regular indoor climate—for example, in a restaurant—are limited, except when addressing specific pollutants like smoking and cooking fumes.

TABLE 8.5 Contaminant removal effectiveness indices.

$\epsilon_p^c = C_e/C_p$	Point in room
$\epsilon_b^c = C_e/\langle C \rangle_b$	Breathing zone
$\epsilon_o^c = C_e/\langle C \rangle_o$	Zone of occupancy
$\epsilon^c = C_e/\langle C \rangle$	Complete room

8.3.3 Contaminant removal efficiency

The contaminant removal efficiency can be derived from the contaminant removal effectiveness as follows:

$$\eta^c = \left(\frac{\epsilon^c}{1+\epsilon^c}\right), \tag{8.3}$$

where η^c is contaminant removal efficiency and ϵ^c is contaminant removal effectiveness.

8.3.4 Air exchange efficiency

Application of the "age of air" concept can be justified by the fact that the content of contaminants found in the exhaust air normally rises from the value found in supply air entering the room. On its voyage through the room, the air is likely to pick up more contaminants the longer it stays in the room. This is a very simple assumption. It can be argued, however, that using the age of air concept is the best way to evaluate ventilation design for scenarios where little or no information is available on use of the room and locations and emission rates for heat and contaminant sources.

TABLE 8.6 Air exchange efficiency indices.

$\epsilon_p^a = \tau_n^a/\overline{\tau}$	Point in room
$\epsilon^a = \tau_n^a/(2\langle\tau\rangle)$	Complete room

In order to have effective exchange of air in important locations in a room, the age of the air in those locations should be low. The basis for comparison is the complete mixing scenario. That scenario gives the same age for any air volume selected in the room, identical to the nominal time constant for the ventilation airflow, τ_n^a. A steady-state scenario is assumed. The various air exchange efficiency indices are presented in Table 8.6.

The average age of air for all air molecules in the complete room can be found by performing a step-up tracer gas experiment, and by measuring tracer gas concentration C_e in the exhaust opening. The same procedure can be used for CFD simulations. The definition for average age of air in the room is

$$\langle\overline{\tau}\rangle = \frac{1}{\tau_n^a}\int_0^\infty \left(1 - \frac{C_e(t)}{C_e(\infty)}\right) t dt, \tag{8.4}$$

where

$$\tau_n^a = \int_0^\infty \left(1 - \frac{C_e(t)}{C_e(\infty)}\right) dt. \tag{8.5}$$

8.3.5 Air distribution performance index

Among the commonly used criteria is the air distribution performance index (ADPI),[4] defined as the percentage of locations where a combination of air temperature and air velocity meets comfort requirements. This criterion is based on experimental results of air diffuser performance for specifically tested room configurations. Data on the ADPI are available only for sedentary activity.

8.4 Room air conditioning strategies

8.4.1 Introduction

Traditionally the room air conditioning classification has been based on room air distribution methods. The most used division has been the division into mixing and displacement. ASHRAE classifies air distribution (diffusion) methods into mixing systems, displacement ventilation, unidirectional airflow ventilation, and localized ventilation methods.[5] Tapola et al. made a division between mixing and displacement and additionally divided displacement into submethods: thermal displacement, piston displacement, and mixing displacement.[6] Etheridge and Sandberg suggested that air distribution methods be classified as jet controlled or thermally controlled, which raises the important question of how well the room airflow patterns are controlled by the air distribution method.[7] However, the ventilation effectiveness depends on task of the ventilation system.[8] Four primary tasks have been recognized: contaminant removal, air exchange, heat removal, and occupant's protection. Within these tasks, the most important goal of using ventilation should be the protection of occupants from contaminants and other hazard conditions.

The direct application of air distribution methods to describe the strategies has led to the wild usage of different terms with unclear definitions. In addition, in some cases the same term has been used to describe both the air distribution method and air supply devices or in some cases even the whole air conditioning system. Using a wrong term can also lead to a complete misunderstanding of the physical phenomenon in the room. The results of this inconsistency can also be seen in the everyday construction business, where the customer often does not have any expertise in our technological field.

German guidelines base the division on the resulting airflow pattern within the room rather than distribution methods.[9] They suggest that airflow patterns be divided into four categories: hall-filling mixed flow; zone-wise mixed flow; low-momentum, low-turbulence flow for the air supply in the work region; and zone-wise displacement ventilation.

This chapter describes a strategy approach for room air conditioning based on the classification and terminology presented by Hagström et al.[10] The basis of the classification is different aims or ideas of the temperature, gas, particle, or humidity distributions, and airflow patterns that can be created within the room. The distributions are often described by using contaminant removal and temperature effectiveness coefficients, which are defined in Section 8.3.

The aim of this classification is not to value one strategy over another. They all have their advantages and disadvantages and it is up to the designer to select the most desirable strategy for each case. In practice, a certain type of room air conditioning strategy can be applied by using different kinds of air distribution installations and air supply devices. How well the real situation will fulfill the aim of the ideal strategy depends not only on the physical installation itself but also on the operating parameters as well as the characteristics of other internal sources that influence the supply airflow patterns and the room airflows, such as heat and contaminant sources, cold drafts, and room heating and cooling methods. It is therefore important to separate ideal strategies from practical room air conditioning solutions.

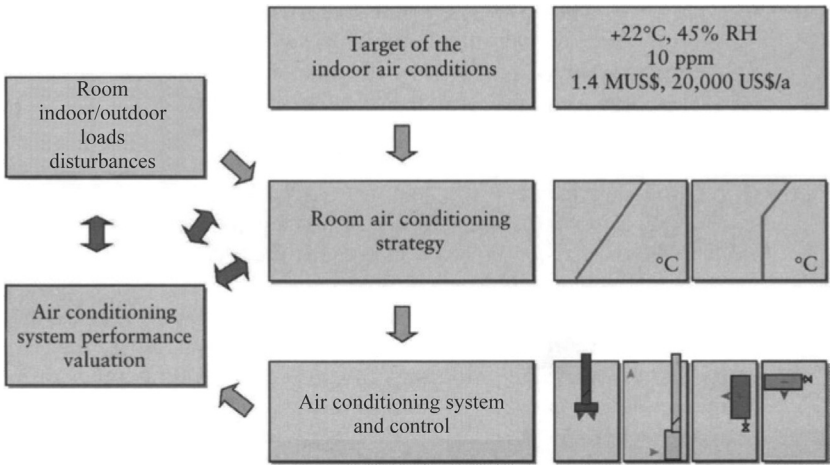


FIGURE 8.2 The room air conditioning design and evaluation process.

TABLE 8.7 Examples of room air distributions, exhausts, and heating and cooling methods.

Air distribution methods	Exhaust methods	Heating methods	Cooling methods
Vertical supply	Local	Convective air heating fan coils	Supply air cooling
Concentrated air jets	General	Radiative infrared	Convective active, e.g., fan coils passive, e.g., ceiling baffles
Low-impulse air supply Inclined jets	Mixed	Mixed local/general	Radiative

A clear classification of the ideal strategies will help the evaluation of the present room air distribution methods in different operating conditions. It also creates a solid base for the development and promotion of new innovations in the field.

8.4.2 Classification for room air conditioning strategies

As the focus of the proposed classification differs from the present practice, it is necessary to explain the terminology used. The aim of room air conditioning is to maintain desired conditions, target levels, in the room during different operating conditions in the most economical way (energy, cost efficiency). Depending on the design criteria, the designer may choose different strategies in order to achieve specified targets. The room air conditioning design and evaluation process is illustrated in Fig. 8.2.

The room air conditioning strategy is a fundamental scheme that describes the targeted temperature, humidity, and contaminant distributions as well as airflow patterns within the air-conditioned room. The room air conditioning system consists of different methods and their controls that together create the system performance. The system performance is evaluated by comparing the reached conditions to the chosen strategy. Both the methods (room air distribution, exhaust, room heating and cooling, etc.) and processes and disturbances inside the room influence the resulting conditions. Different room air distributions, exhausts, and heating and cooling methods are listed in Table 8.7. As an example of the terminology we can use a system consisting of low-impulse air devices supplying directly into the occupied zone (often called displacement ventilation) and cooled ceiling methods.

The ratio between the total area a_o of the air supply openings and the surface area A of wall/ceiling/floor on/in which the supply openings are located, a_o/A, is an important parameter for the air distribution in the room.[11] The ratio, a_o/A, is considered to be small for values smaller than 10^{-3}, medium for values between 10^{-3} and 0.3, and large for values larger than 0.3. The values smaller than 10^{-3} are typical for diffusers designed for mixing ventilation, and the value $6 \cdot 10^{-3}$ is typical for displacement ventilation diffusers.

Fig. 8.3 shows design graph ($q_s - \Delta\theta_s$ graph) for a constant value of a_o/A. The area on the right side of the curve defines momentum driven flow while on the left side defines a flow driven by the buoyancy forces.[12] The curve indicates the position of the critical Archimedes number where the air movement changes between the two different flow types.

The following presentation discusses the room air distribution method as a principal parameter to apply to a certain room air conditioning strategy and heating and cooling as assisting methods. However, it must be noted that in some cases a strategy can be fulfilled

without any mechanical air distribution installations using buoyancy forces. The classification of ideal room air conditioning strategies is summarized in Table 8.8 and explained more in detail in the text. Though the main emphasis of this presentation is on general room air conditioning, the same ideas behind different strategies can be used for local ventilation. In addition, as ideal, the classification is not affected by whether the flow direction is horizontal or vertical (upward or downward).

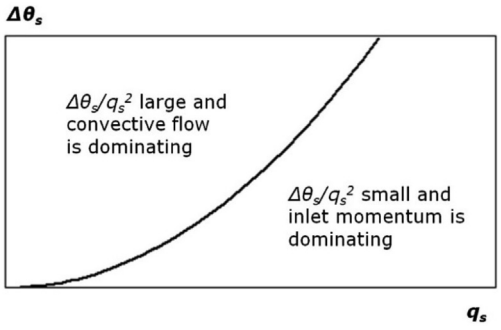


FIGURE 8.3 Principle determination of airflow in a room with a given a_o/A ratio based on the critical Archimedes ratio. Convective flow is dominating in the left side of the graph while inlet momentum flow is dominating at the right side of the graph.

8.4.3 Piston strategy

8.4.3.1 Description

The highest effectiveness can be achieved with the piston strategy. The contaminant concentration, temperature or humidity, and the local effectiveness are functions of the location and the power of the sources in relation to the supply and exhaust openings. With a homogeneous distribution of sources, the contaminant concentration and temperature change linearly between supply and exhaust openings located at opposite ends of the room. With local sources, the concentration upstream of the sources is very low.

Fitzner[13] points out that piston flow from the floor and upward exists for Archimedes numbers less than 360.

$$Ar = \frac{g\,H\,\Delta\theta}{T\,v^2} < 360 \tag{8.6}$$

where g is the acceleration of gravity = 9.81 m/s^2; H is the height of the room (m); $\Delta\theta = \theta_e - \theta_s$ is the temperature difference between exhaust and supply air (K); T is the absolute temperature of the supply air (K); v is the mean air velocity upward; and air volume flow/floor area (m/s). For $Ar > 360$, buoyancy forces will dominate and make a thermally stratified flow.

TABLE 8.8 Ideal room air conditioning strategies.

	Strategy			
	Piston	**Stratification**	**Zoning**	**Mixing**
Description	To create unidirectional airflow field over the room area by supply air	To support flow field created by density differences by replacing the airflow out from the room area with supply air	To control air conditions within selected zone in the room by the supply air and allow stratification of heat and contaminants in the other room areas	To provide uniform conditions throughout the ventilated space
Heat, humidity, and contaminant distribution[a]	Room dimension EX	Room dimension EX	Room dimension EX	Room dimension EX
Main characteristics	Room airflow patterns controlled by low-momentum unidirectional supply airflow, strong enough to overcome disturbances	Room airflow patterns controlled mainly by buoyancy; supply air distribution with low momentum	Room airflow patterns controlled partly by supply and partly by buoyancy	Room airflow patterns controlled typically by high-momentum supply airflow
Ideal efficiency	$\varepsilon_t = (t_{ex} - t_0)/(t_{oz} - t_0)$ $\varepsilon_e = (C_{exh} - C_0)/(C_{oz} - C_0)$			
Typical application (example of a general room air distribution method)				

[a] *x-axis:* °C, *mg/m*^{3}*g/kg; y-axis: room dim. (e.g., height).*
EX, Exhaust.

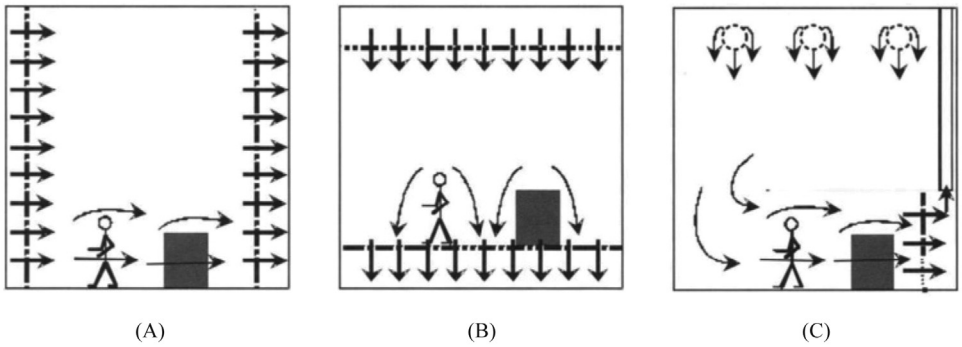


FIGURE 8.4 Examples of air distribution and exhaust methods for the piston strategy. (A) Horizontal piston. (B) vertical piston. (C) partial piston.

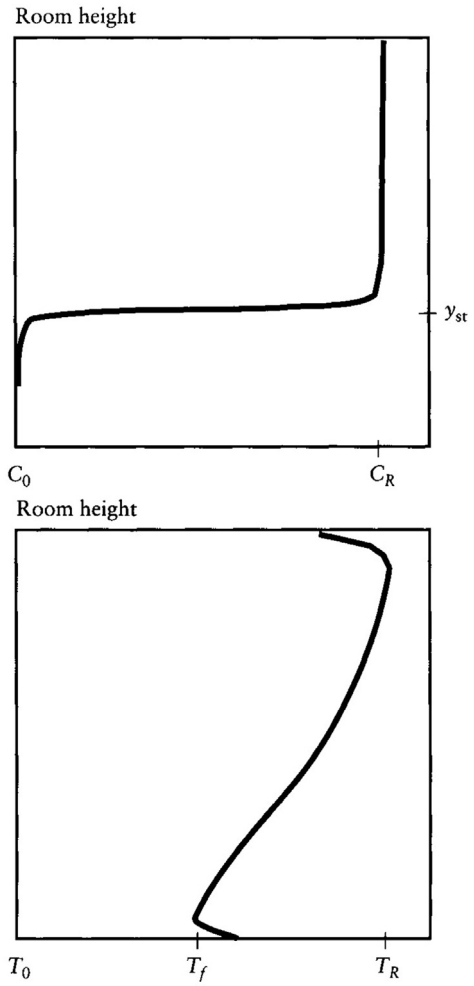


FIGURE 8.5 The height of the contaminant interfacial level in the stratification strategy.

8.4.3.2 *Advantages and disadvantages*

Advantages include the whole flow pattern can be controlled, areas upstream of sources can be kept clean, and high contaminant removal and temperature effectiveness. Disadvantages include the need for high supply airflow rates and large supply areas.

8.4.3.3 *Design criteria*

The design criterion of the piston strategy is to overcome all the air currents opposite to the directional airflow created within the room.

8.4.3.4 *Application*

Piston air conditioning is an expensive strategy due to the high airflow rate that is needed to create a desired airflow pattern inside the room. Thus it is usually used only in applications where it is required, like in the semiconductor industry, where up to about 400 air changes per hour are used. Another example of its application is horizontal piston flow in reinforced plastic plants.[14] Schemes of different ways to apply the piston air conditioning strategy are shown in Fig. 8.4.

8.4.4 Stratification strategy

8.4.4.1 *Description*

A similar temperature and contaminant distribution throughout the room is reached with stratification as with a piston. The driving forces of the two strategies are, however, completely different and the distribution of parameters is in practice different. Typical schemes for the vertical distribution of temperature and contaminants are presented in Fig. 8.5.[15] However, it should be noted that vertical temperature gradient is strongly depending on the type and location of heat load.[16] With heat loads close to floor level, there exists clear mixing layer in the occupied zone. Over the mixing layer the room air temperature could be assumed to be constant or only slightly increase. Thus the major part of the temperature gradient could happen in those cases in low level.

While in the piston strategy the uniform flow pattern is created by the supply air, in stratification it is caused only by the density differences inside the room,

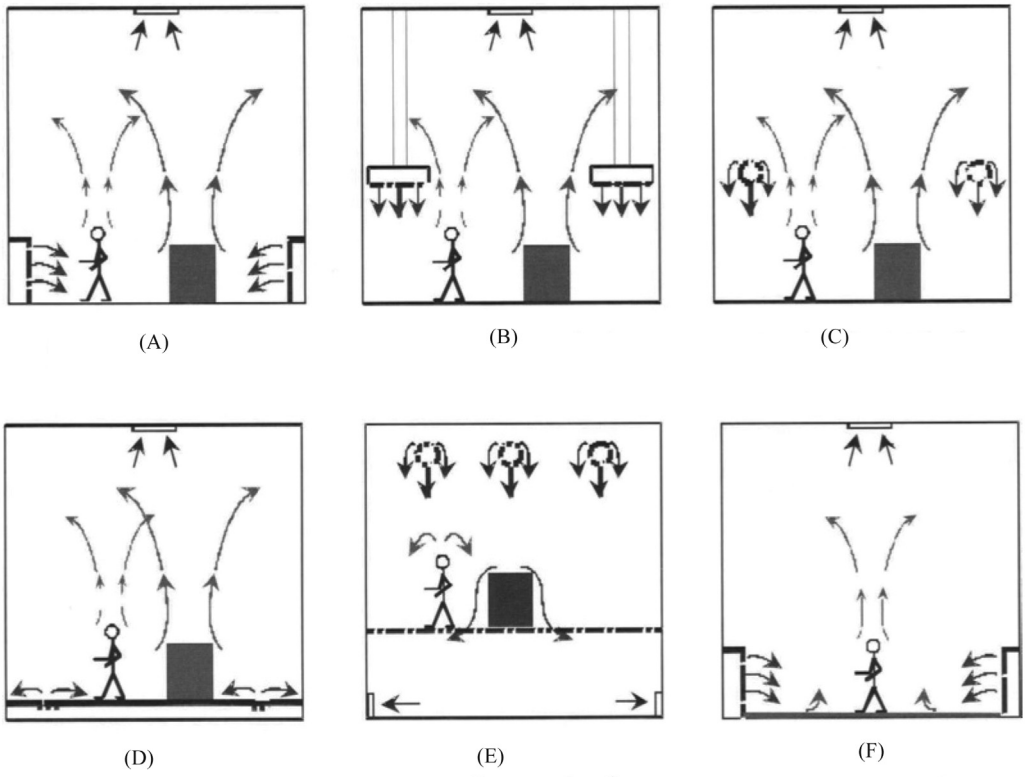


FIGURE 8.6 Different air distribution methods to create thermal replacement. (A) Low-impulse supply from floor level. (B) Low-impulse supply above aisles—1. (C) Low-impulse supply above aisles—2. (D) Spare floor supply. (E) Negative buoyancy. (F) Floor heating.

that is, the room airflows are controlled by the buoyancy forces. As a result, the contaminant removal and temperature effectiveness are more modest than with the piston air conditioning strategy.

8.4.4.2 Advantages and disadvantages

Advantages include low concentration in the ventilated zone can be achieved and relatively high contaminant removal and temperature effectiveness. However, the stratification strategy is sensitive to disturbances and stagnant areas with high local concentrations are possible. It also only functions properly when conditions are favorable.

8.4.4.3 Design criteria

In the stratification strategy the supply air is used to substitute the outgoing air from the ventilated (in most cases occupied) zone, thus preventing circulation patterns between the zones. The supply air has to be distributed in such a way that the buoyancy flows are not disturbed. Exhaust air openings are to be located "downstream" in order to avoid reverse currents within the room. The location of the contaminant sources and the heat sources causing density differences must be the same in order to carry out the contaminants with equal or higher density than air.

8.4.4.4 Application

Stratification is a desirable strategy to provide efficient room air conditioning with much less effort than using the piston strategy. Its main application in room air conditioning is the thermal replacement method. However, it can also be applied for contaminants without any thermal buoyancy sources that have different density from the room air. Examples of different air distribution methods to create thermal replacement are presented in Fig. 8.6.

However, because of its physical nature—ventilating air having very little authority over the room airflow—the stratification strategy is very dependent on the stability of the density differences and the airflow balances and thus very sensitive to disturbances within the room. Also, the selection of the room cooling and heating method can either aid or prevent the creation of the stratification strategy as described earlier in the chapter.

8.4.5 Zoning strategy

8.4.5.1 Description

The idea of the zoning air conditioning strategy is to have control over the certain area or volume of the room, while the rest of the room is left with less attention. In most cases the accumulation of heat, concentration,

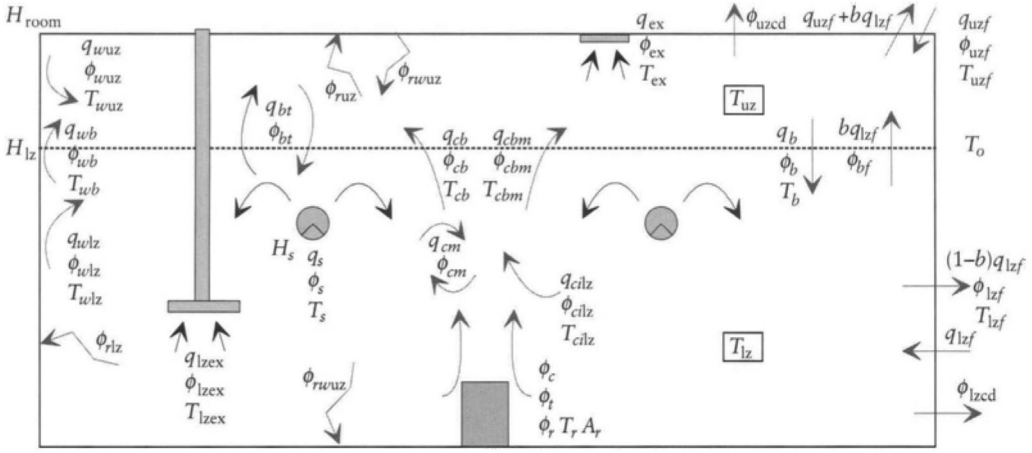


FIGURE 8.7 A general two-zone air mass and heat flow model.

or humidity outside the controlled zone is desired and utilized. The room airflows are controlled both by the supply air jets and the buoyancy sources. The ventilation effectiveness (temperature, contaminant removal, humidity) using the zoning air conditioning strategy is expected to settle between the values of mixing and stratification strategies. However, the effectiveness is strongly dependent on the methods used and the operating conditions. Concentration and temperature are more homogeneously distributed in the controlled zone than with the stratification strategy.

Two-zone models are especially useful for stratification and zoning strategies because of the typical vertical accumulation of heat, contaminants, or water vapor within these strategies. The level of the boundary between the lower and the upper zone is usually determined on the level of the highest temperature or/and concentration gradient.

In the zoning strategy the height of the boundary between lower and upper zones should be determined with the criteria of the air distribution method and devices. The lower zone should be defined high enough to get nearly all the induction air of the supply air devices from the controlled zone, as shown in Fig. 8.7.

Zoning can be either vertical or horizontal. Typically vertical zoning is applied in high rooms, when the supply air is distributed close to the occupied zone near floor level and the exhaust air openings are located close to the ceiling. Horizontal zoning can be applied, for example, using air or portable/partial (plastic, etc.) curtains in order to divide room space into different blocks. Within these blocks it is possible to further apply different strategies in the vertical direction semiindependently.[17]

8.4.5.2 *Advantages and disadvantages*

The zoning method offers better contaminant removal and thermal effectiveness than with mixing, limited control of the flow patterns in the ventilated zone, and the ability to avoid stagnant areas with high local concentrations in the ventilated zone. However, partial mixing of contaminants in the ventilated zone decreases its effectiveness.

8.4.5.3 *Design criteria*

Each method has its own design criteria, but common to most of the methods is that air supply is located close to or inside the controlled zone and the exhaust openings are located inside the uncontrolled zone. The location and power of the buoyancy sources in relation to the supply air jets have a remarkable influence on the accumulations of heat, contaminants, and humidity within the room.

The calculation of the two-zone model is based on the balance equations for air mass flow, contaminant mass flow, water vapor mass flow, and heat flow of both zones.

The air, contaminant, and water vapor mass flow elements in outer boundaries and between the zones are created by the following:

1. supply air,
2. extract air,
3. heat and contaminant sources,
4. local ventilation,
5. plumes of the buoyancy sources through the zone boundary,
6. possible return air from the upper zone into the lower zone,
7. flows along wall surfaces due to temperature differences,
8. infiltration and exfiltration, and
9. mixing between zones due to turbulence and disturbances.

The heat flow elements are created by the following:

1. radiation from heat sources,
2. radiation between room surfaces, and
3. heat transport through surface boundary layers.

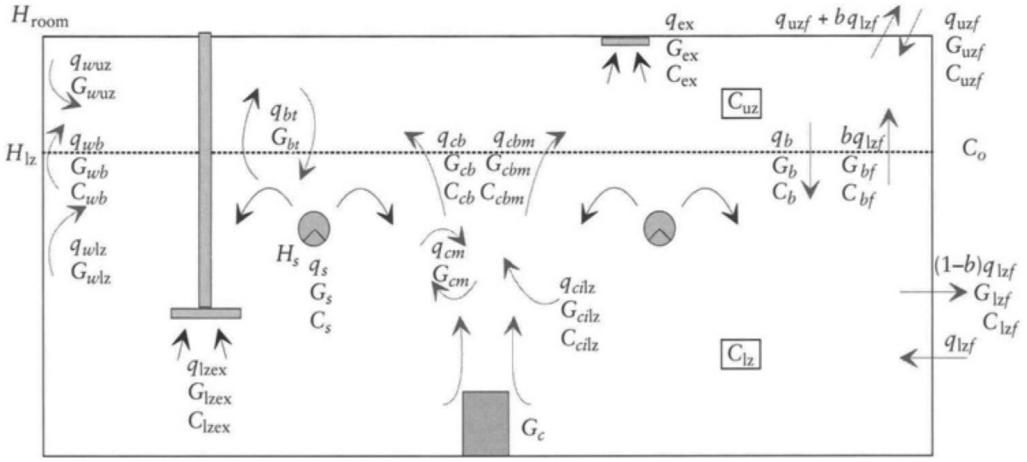


FIGURE 8.8 A general two-zone air and contaminant mass flow model.

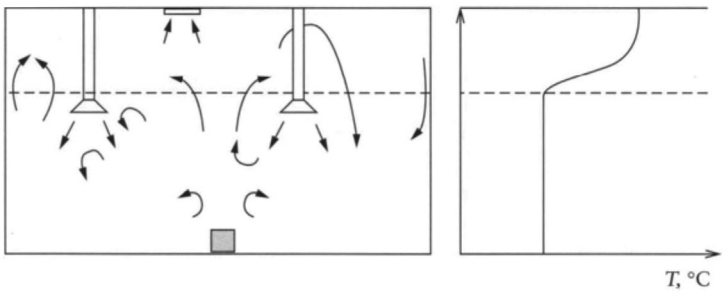


FIGURE 8.9 Vertical temperature and contaminant distribution within the stratification strategy.

A general steady-state balance calculation of a two-zone model is presented in Figs. 8.8 and 8.9 and Eqs. (8.14)–(8.21).

Nomenclature for Figs. 8.8 and 8.9 *and* Eqs. (8.7) and (8.8)

C = concentration (mg/m^3)
H = height (m)
G = contaminant flow rate (mg/s)
q = airflow rate (dm^3/s)
Q = heat flow rate (W)
T = temperature (K)

Subscripts

1 = upper zone
2 = lower zone
b = boundary, through the boundary
c = convection
cd = conduction
ex = exhaust, extract
f = filtration
i = in
m = mixing, mixed
o = out, outside
r = radiation
s = supply
t = turbulent mixing
w = wall

The model equations are determined by writing the balance equations based on the conservation of mass and energy. The balance equations have the following basic form:

$$\Sigma \text{ flow in} + \Sigma \text{ flow sources} - \Sigma \text{ flow out} = 0$$

Air mass flow balance for the lower zone:

$$\rho_{s1}q_{s1} - \sum \rho_{ex1}q_{ex1} + \rho_b q_b - \sum \rho_{\omega b}q_{\omega b} - p_1 q_{fo1} + \rho_o q_{fi1} - \sum p_{cbm}q_{cbm} = 0. \tag{8.7}$$

Air mass flow balance for the upper zone:

$$\rho_{s2}q_{s2} - \sum \rho_{ex2}q_{ex2} - \rho_b q_b + \sum \rho_{\omega b}q_{\omega b} - p_2 q_{fo2} + \rho_o q_{fi2} + \sum \rho_{cbm}q_{cbm} = 0. \tag{8.8}$$

Heat flow balance for the lower zone air:

$$\Phi_{s1} - \sum \Phi_{ex1} + \Phi_b - \sum \Phi_{\omega b} - \Phi_{fo1} + \Phi_{fo1} - \sum \Phi_{cbm} + \sum \Phi_{c1} + \sum \Phi_{c\omega 1} + \Phi_{bt} = 0. \tag{8.9}$$

Heat flow balance for the upper zone air:

$$\Phi_{s2} - \sum \Phi_{ex2} - \Phi_b + \sum \Phi_{\omega b} - \Phi_{fo2} + \Phi_{fi2} + \sum \Phi_{cbm} + \sum \Phi_{c2} + \sum \Phi_{cw2} - \Phi_{bt} = 0. \quad (8.10)$$

Heat flow balance for the lower zone walls:

$$-\sum \Phi_{c\omega 1} + \sum \Phi_{rw1} + \sum \Phi_{rw21} - \sum \Phi_{cd1} = 0. \quad (8.11)$$

Heat flow balance for the upper zone walls:

$$-\sum \Phi_{c\omega 2} + \sum \Phi_{rw2} - \sum \Phi_{rw21} - \sum \Phi_{cd2} = 0. \quad (8.12)$$

Contaminant mass flow balance for the lower zone:

$$G_{s1} - \sum G_{ex1} + G_b - \sum G_{\omega b} - G_{fo1} + G_{fi1} - G_{cbm} + \sum G_{c1} + G_{bt} = 0. \quad (8.13)$$

Contaminant mass flow balance for the upper zone:

$$G_{s2} - \sum G_{ex2} - G_b + \sum G_{\omega b} - G_{fo2} + G_{fi2} + \sum G_{cbm} + \sum G_{c2} - G_{bt} = 0. \quad (8.14)$$

The balance equations for water vapor flows are similar to balance equations for contaminant flows, but in addition possible condensation and evaporation must be calculated. Also, they must be considered in heat flow equations.

The air and wall temperatures and the concentrations in both zones are solved by iteration toward a steady-state situation or by simulating the time-dependent development. In the time-dependent calculation the heat capacity of the walls should be included.

In wall heat balance equations (8.11) and (8.12), the radiation heat flows $\Sigma\ \Phi_{rw1}$ and $\Sigma\ \Phi_{rw2}$ from the heat sources and $\Sigma\ \Phi_{rw21}$ from upper zone wall surfaces to lower zone wall surfaces are assumed to increase the temperature of the walls. In practical cases it is quite complicated to determine how much of the radiation flow rate will be distributed to outer walls and to other surfaces.

Vertical buoyant flows on the wall boundaries $\Sigma\ q_{wb}$, $\Sigma\ \Phi_{wb}$, and $\Sigma\ \Phi_{wb}$ are the sum of several upward and downward flows through the zone boundary, which can be calculated using plume and jet theories.

The convection flows from the heat sources $\Sigma\ \Phi_{c1}$ and $\Sigma\ \Phi_{c2}$ as well as contaminant flows from contaminant sources are flows loading the room. In the sources additional heat and pollutant flows may be generated, which are exhausted directly out by local ventilation and are not included in the balance calculation.

The pollutant sources $\Sigma\ G_{c1}$ and $\Sigma\ G_{c2}$ may be without any buoyancy forces or they may be sinks, in other words negative sources or filters.

The flow rate of the plume through the zone boundary depends on the plume strength and vertical temperature gradient.[18] In the case of a zoning strategy, the plume flow rate may also depend on the air distribution method and device because of the interaction between the plume and the supply air.[19]

The turbulent mixing between the zones depends on the air distribution method and device.[20,21] Specified solutions of the general two-zone model have been presented previously by Skåret.[22] He presents a general air and contaminant mass flow model for a space where the air volume, ventilation, filtration, and contaminant emission have been divided for both the zones and the turbulent mixing (diffusion) between the zones is included. A time-dependent behavior of the concentration in the zones with constant pollutant flow rate is presented.

8.4.5.4 Application

A large variety of methods can be used for zoning, such as inclined jets, horizontal cooled jets, vertical jets, floor jets, nozzle ducts, and vortex. Examples of different methods are illustrated in Fig. 8.10.

8.4.6 Mixing strategy

8.4.6.1 Description

The aim of the mixing air conditioning strategy is to provide uniform conditions throughout the air-conditioned room.

The contaminant removal and temperature effectiveness in the mixing strategy are equal to 1. In practical installations incomplete mixing in the room and unfavorable temperature gradient and location of the exhaust openings in relation to air supply may, however, cause short-circuiting of the supply air into the exhaust openings and the efficiency may remain below 1.

8.4.6.2 Advantages and disadvantages

Using the mixing strategy, stagnant areas with high local concentrations and unfavorable thermal gradients can be avoided during the heating period. At the same time, it has low contaminant removal and temperature effectiveness and its high air velocity may cause drafts.

8.4.6.3 Design criteria

Air jets are used to create enough air movement inside the room to circulate and mix the whole room air. This strategy is often called dilution ventilation as the contaminants created inside the room are mixed into the whole room volume thus reducing the local peak concentrations.

8.4.6.4 Application

The room airflows are controlled mainly by the supply or/and circulation air jets using, for example,

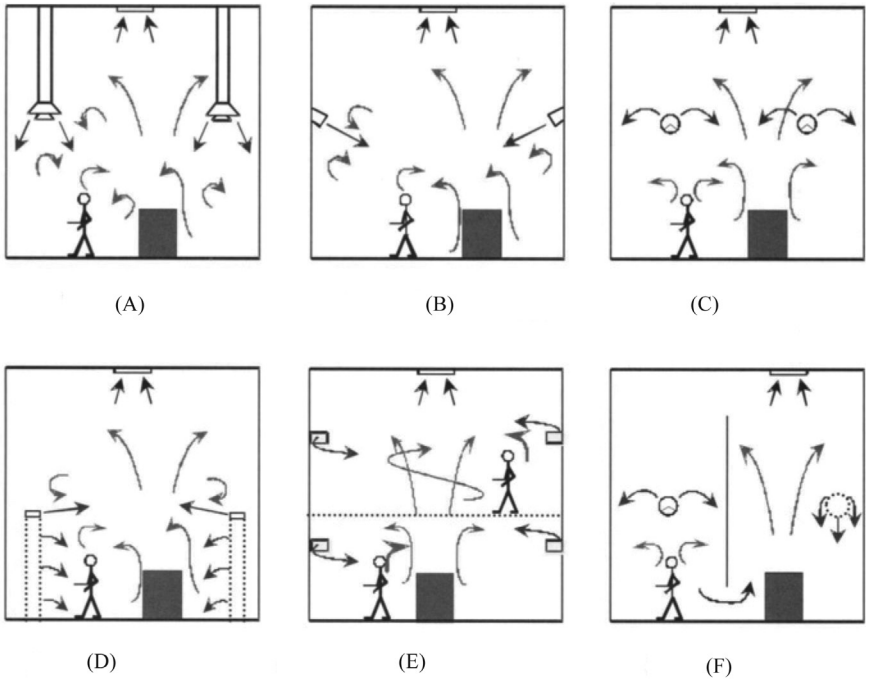


FIGURE 8.10 Examples of air distribution and exhaust methods for the stratification strategy. (A) Vertical air supply. (B) Inclined jets air supply. (C) Activent air supply. (D) Jet-supported low-impulse curtain. (E) Vortex air supply. (F) Horizontal zoning with curtain.

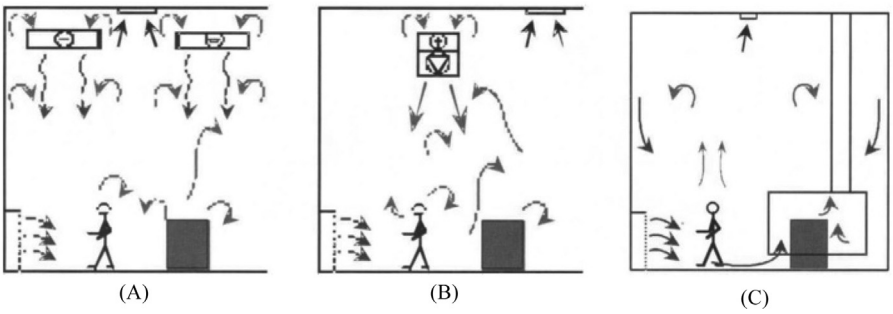


FIGURE 8.11 Examples of air distribution and exhaust methods for the zoning strategy. (A) Replacing air distribution: cooled ceiling. (B) Replacing air distribution: recirculating fan heaters. (C) Replacing air distribution: governing local exhaust.

concentrated jets, ceiling air supply, or high-impulse nozzle systems. Examples of different methods are illustrated in Fig. 8.11.

However, the use of other room air distribution methods together with certain exhaust, heating, and cooling methods will also lead (intentionally or unintentionally) to the application of the mixing air conditioning strategy.[23] Some examples are shown in Fig. 8.12.

8.4.7 Application of the strategy in system selection

The application of the strategies in system selection can be illustrated by using a simple example. Let us think about an industrial hall with some internal heat sources but without any remarkable internal movement that would disturb the stratification.

During summer, Fig. 8.13A, there is a need for cooling in the occupied zone (area up to 2 m from the floor level); thus it is desirable to apply the stratification strategy with vertical temperature and contaminant stratification in the hall in order to save cooling energy costs. This can be done, for example, by using a low-impulse air supply with the devices at the floor level.

During winter additional heating is needed in the occupied zone due to the heat losses. The selection of the heating method depends on the selected air conditioning strategy for the heating season. In order to save heating energy costs the exhaust air temperature should not exceed the temperature in the occupied zone. Thus the desired strategy would be the mixing strategy. An appropriate heating method for that purpose would be, for example, an air and recirculation method with fan heaters located close to the ceiling; see Fig. 8.13B.

If the main reason for the stratification strategy is contaminant control in the occupied zone, the same strategy should be applied in winter conditions, too. Thus the selected heating method has to fulfill two

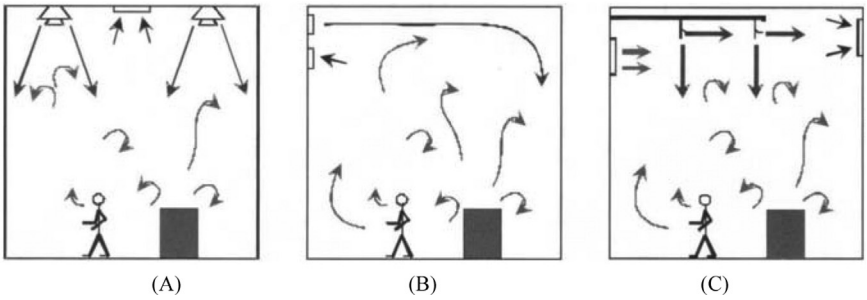


FIGURE 8.12 Examples of air distribution and exhaust methods for the mixing strategy. (A) Ceiling air supply. (B) Concentrated air supply. (C) Air supply with directing jets.

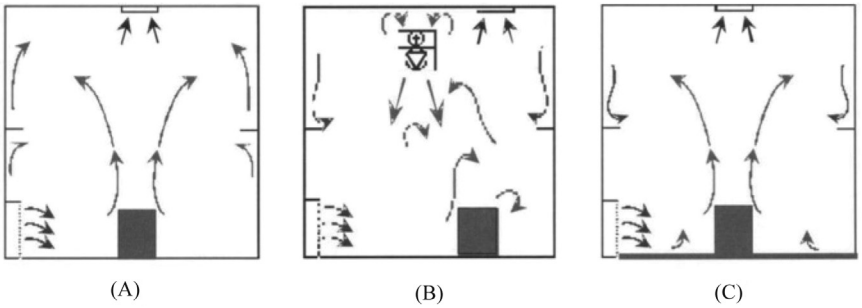


FIGURE 8.13 Application of the mixing and stratification methods in winter and summer. (A) Summer: stratification methods—(1) low-impulse air supply. (B) Winter: mixing methods—(1) low-impulse air supply, (2) recirculating fan heaters. (C) Winter: stratification methods—(1) low-impulse air supply, (2) floor heating.

requirements: to support the creation of the vertical stratification and not to create disturbing airflows into the hall. In this case one option would be the floor heating method; see Fig. 8.13C. In addition, one should consider the prevention of boundary layer flows along the outer walls using, for example, passive methods.[24]

To conclude the example: A different air conditioning strategy can and sometimes should be selected for different seasons or operating conditions. The strategy can be changed by using combinations of available methods.

8.4.8 Summary

The room air conditioning strategy should be used as a target for design and construction of the room air conditioning system. Often it would be desirable to apply several strategies during different operating conditions (e.g., summer–winter). The selection of the system and the set of methods (room air distribution, exhaust, heating, cooling) should be made in such a way that the different strategies can be applied most efficiently.

The clarification of the room air conditioning strategies and their separation from the practical methods at present creates space for creativity and new innovations and their evaluation.

Naming a practical room air distribution method according to a certain strategy may lead to misunderstanding of its performance in varying operating conditions. Though naming probably can't be avoided in business, an understanding of the basic strategies helps customers evaluate offered practical air distribution methods and system solutions.

8.5 Air distribution methods and dimensioning

8.5.1 Selection of air supply method

The choice of room airflow pattern and air supply method is subject to study in each separate case. Table 8.9 presents, however, some guidelines for air distribution methods most commonly applied for various cases.

8.5.2 Mixing air distribution

In this chapter we deal with air distribution in a room, where L is the room length, B is the room width, and H_r is the room height.

See Fig. 8.14.

8.5.2.1 Penetration of horizontal air jets

Room length

If the room is to be ventilated by one single air jet, and the jet shall penetrate across the entire room length, the room length should not exceed[25]:

$$L \leq 0.62 K_1 \sqrt{B H_r}, \tag{8.15}$$

where A_0 is the effective discharge area of the jet (m^2), B is the room width (m), H_r is the room height (m), K_1 is the decay coefficient of the jet, L is the room length (m), u_{max} is the maximum velocity in jet cross-section (m/s), and u_0 is the discharge velocity of the jet.

TABLE 8.9 Guidelines for selection of air distribution method.

Pollutant load	Room height (m)	Heat load	Human activity level	Common air supply methods	Type of process
Insignificant		Small	Light	Mixing, horizontal air jets below ceiling	Mechanical assembly
Moderate, dust and fumes	>5		Moderate	Displacement, low-velocity air supply at floor level	Welding shop
Moderate, solvent vapors	<5		Light	Low-velocity supply from above, local exhausts	Paint shop
High, solvent vapors			Light	Piston flow, vertical from above	Paint booth
Heavy	>10	High	Heavy	Displacement, low-velocity air supply at floor level	Foundry, melting works

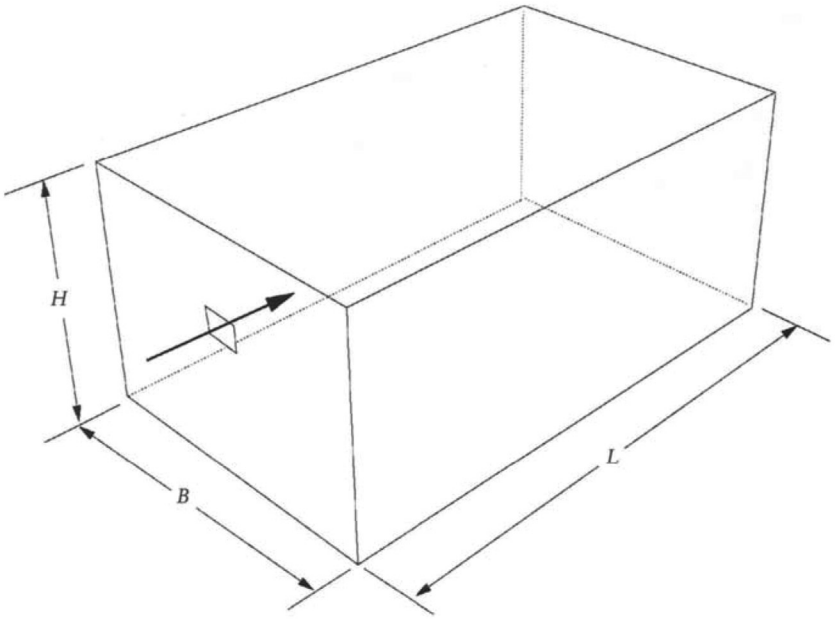


FIGURE 8.14 Room dimensions definition.

The constant K_1 is the decay coefficient used in the formula:

$$\frac{u_{max}}{u_0} = K_1 \frac{\sqrt{A_0}}{x}. \quad (8.16)$$

where K_1 depends on the type of jet, and varies between 2 and 7. Some K_1 values are given in Table 8.10. See also Section 7.4. If the room is longer, one has to take long-room considerations into account. See Section 8.5.2.2.

Room width

The width of the room, B, should be less than or equal to $3H$ in order to be properly ventilated by one jet. Wider rooms require several jets beside each other, as indicated in Fig. 8.15.

8.5.2.2 Reverse flow

Short rooms

When a jet is confined in a short room, the jet will be deflected by the opposite wall, and the flow pattern will be as indicated in Fig. 8.16. Outside the jet will be a recirculation zone which consists of the air that is entrained into the jet and the air that is exhausted through the outlet. The amount of entrained air is far greater than the exhaust air. The maximum airflow rate in the reverse flow in Fig. 8.16 equals the airflow rate in the jet.

The velocity distribution in the reverse flow is assumed to be uniform in the case of isothermal conditions. In most practical cases, however, there is a heat surplus in the room, and the supply air is colder than the room air. In this case the velocity will be higher at floor level.

TABLE 8.10 Decay coefficients K_1^2.

Type of discharge opening	$u_0 = 2-5$ m/s	$u_0 = 8-40$ m/s
Single, circular or square	5.7	7.0
Rectangular opening		
$b/h = 25$	5.3	6.5
$b/h = 40$	4.9	6
Grid, opening >40%	4.7	5.7
Perforated panel, opening 3%–5%	3.0	3.7
Perforated panel, opening 10%–20%	4.0	4.9
Louvers with diverging vanes		
Div. angle 40°	2.9	3.5
Div. angle 60°	2.1	2.5
Div. angle 90°	1.7	2.0

Long rooms

If the room is longer than the length stated in Eq. (8.23), the air jet reaches a point where it decelerates more than in free air, and the reverse flow becomes more dominant. See Fig. 8.17. We define a critical length, x_{crit} as the length where the jet starts getting narrower instead of continuing to widen. This critical length is less than the maximum room length defined in Eq. (8.17). Stensaas gives the following estimate for the critical length[26]:

$$x_{max} \approx 3.3\sqrt{BH}. \tag{8.17}$$

The equation is about 75%–90% of the maximum room length for a compound free jet [decay coefficient of 7.0–5.3 in a decay coefficient equation (8.16)].

If the room is longer there will be rotating cells as indicated in Fig. 8.18 in isothermal conditions.

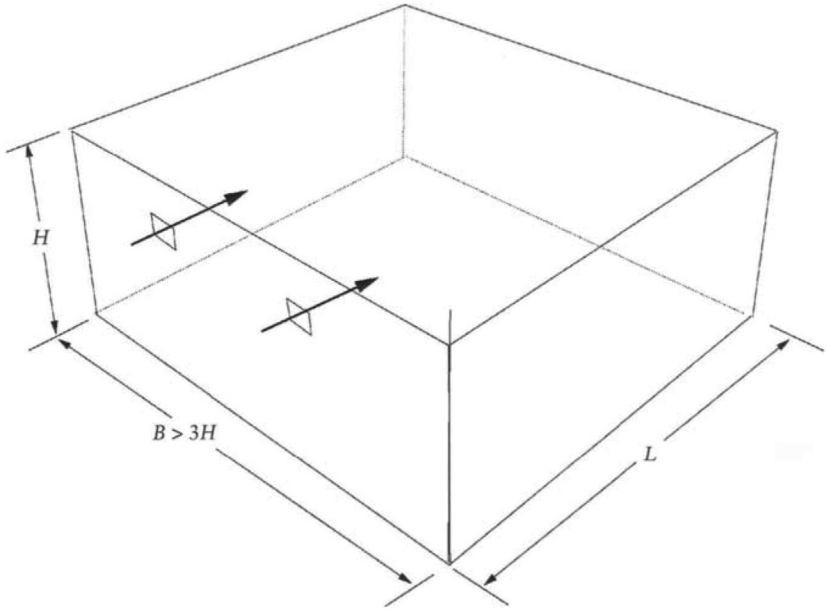


FIGURE 8.15 Wide room.

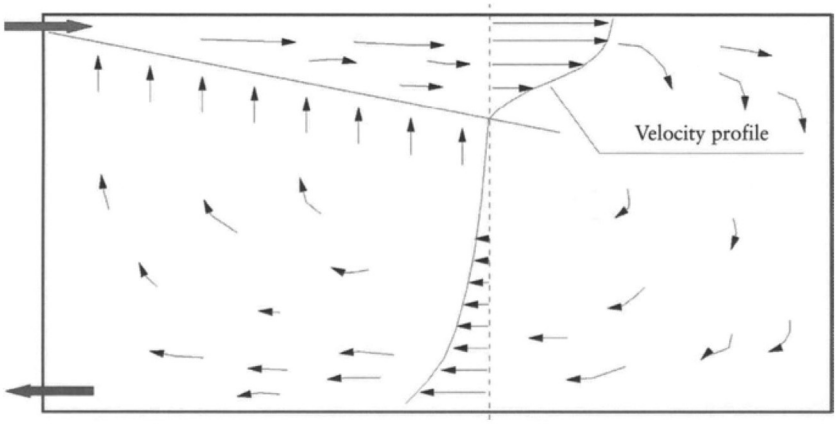


FIGURE 8.16 Reverse flow in a short room.

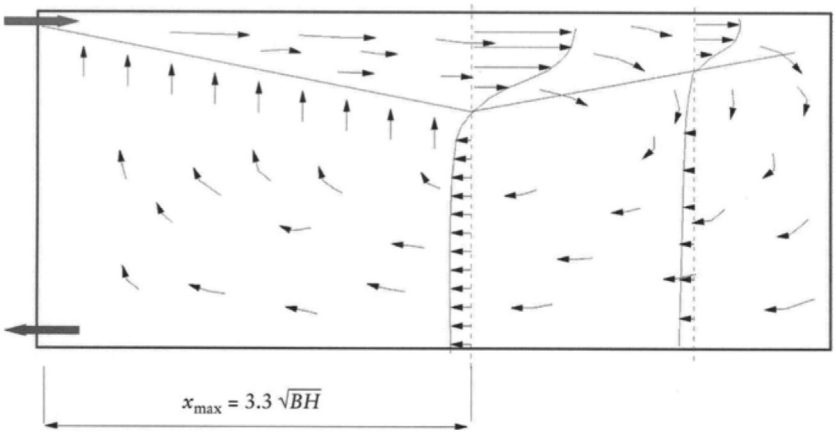


FIGURE 8.17 Maximum penetrating distance for an air jet.

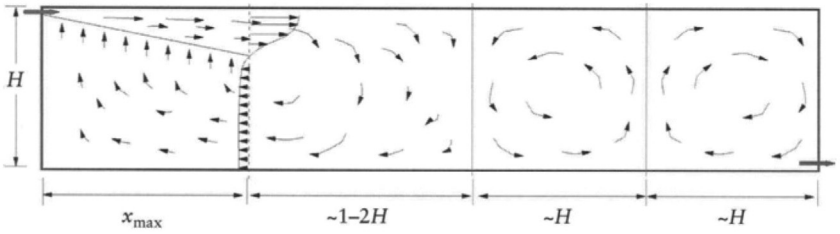


FIGURE 8.18 Multiple rotating cells in a long room: isothermal conditions.

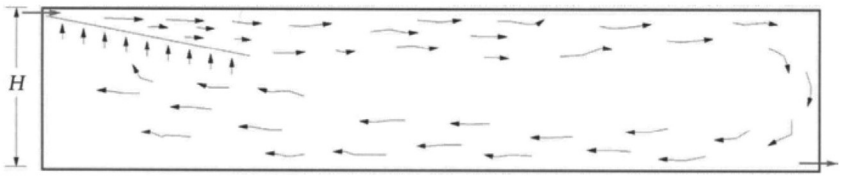


FIGURE 8.19 Long room with thermal stratification.

If the room has a certain amount of heat surplus, this will lead to thermal stratification. The thermal stratification will attenuate the rotation, and eventually lead to a flow pattern as shown in Fig. 8.19.

8.5.3 Piston flow

Piston flow is utilized in several cases where heavily contaminated air needs to be removed, for instance in a spray painting booth (Fig. 8.20).

When there is a very high demand on the cleanness of the air, like in the pharmaceutical industry, piston flow from the ceiling is utilized (Fig. 8.21).

8.5.3.1 Filter mat ceilings

In practice, piston flow is not very easy to establish. A common way, utilized in cleanrooms and paint booths, is to have a filter mat placed all across the ceiling. This gives a good result as long as there is an even pressure distribution above the filter mat. If the supply air jet velocity is high and the pressure difference across the filter mat is low, we may get an uneven velocity distribution along the filter mat, as shown in Fig. 8.22.

When designing air supply through a filter ceiling, one should ensure that the dynamic pressure in the supply air does not affect the static pressure distribution above the filter ceiling too much.

8.5.3.2 Perforated sheet ceilings

In rough environments a filter mat may not be desirable, due to wear or dust in the supply air. In those cases, perforated sheets are used as a means of creating a piston flow.

Perforated sheets are, however, much more vulnerable to uneven pressure distributions and tilted inflow of air, as illustrated in Fig. 8.23. The supply air entering horizontally above the perforated sheet partly maintains its horizontal velocity component when being discharged through the holes in the perforated sheet. Thus it leaves the perforated sheet at an angle less than 90 degrees. The suction between the small outflowing jets also makes the airflow stick to the perforated sheet and flow along the sheet instead of perpendicular to it.

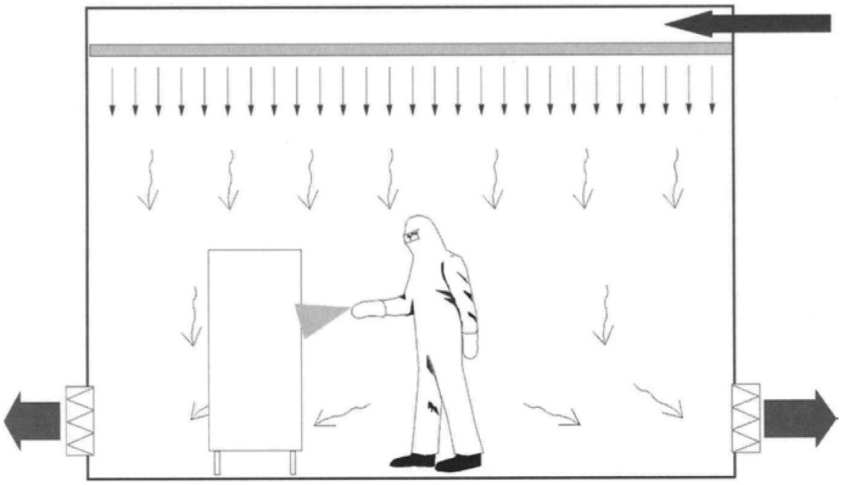


FIGURE 8.20 Piston airflow from a filter mat ceiling in a spray painting booth.

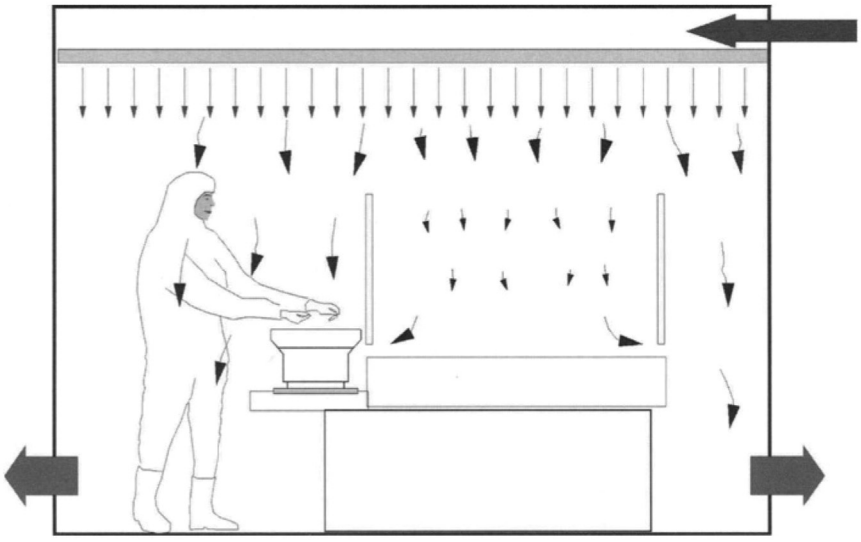


FIGURE 8.21 Piston flow in the pharmaceutical industry.

8.5.3.3 *Thermal instabilities in piston flows*

When designing ventilation systems with piston flow, one should also be aware of the effects of thermal instabilities. Fig. 8.24 shows a room with air supply through a filter ceiling that gives an evenly distributed airflow through the filter mat. However, the supply airflow, being colder than the room air, becomes unstable and turns away from the ascending convection current above the heat source.

Even if there is no concentrated heat source in the room, a flow where colder air is supplied above warmer air always becomes unstable, and tends to turn the flow so that the warmer air goes up and the colder air goes down.

If air is supplied horizontally through a wall, the supply air has to have exactly the same temperature as the room air. A temperature difference of 1°C or more will create flow patterns like those shown in Fig. 8.25.

8.5.4 Displacement flow

Displacement ventilation means that the supply air pushes the older air away without mixing with it. There are several ways of creating displacement flow. When there are no or negligible temperature differences in the room, displacement may be created as piston flow. In industrial ventilation, temperature differences are commonly present. These cases refer to displacement ventilation as a system where colder fresh air supplants warmer old air, or warmer air replaces the colder air using buoyancy as the main driving force of the process.

8.5.4.1 *Warm contaminants*

A simple example of displacement ventilation is shown in Fig. 8.26.

Note that the supply air should be at least 1°C–2°C colder than the air in the lower part of the room in order

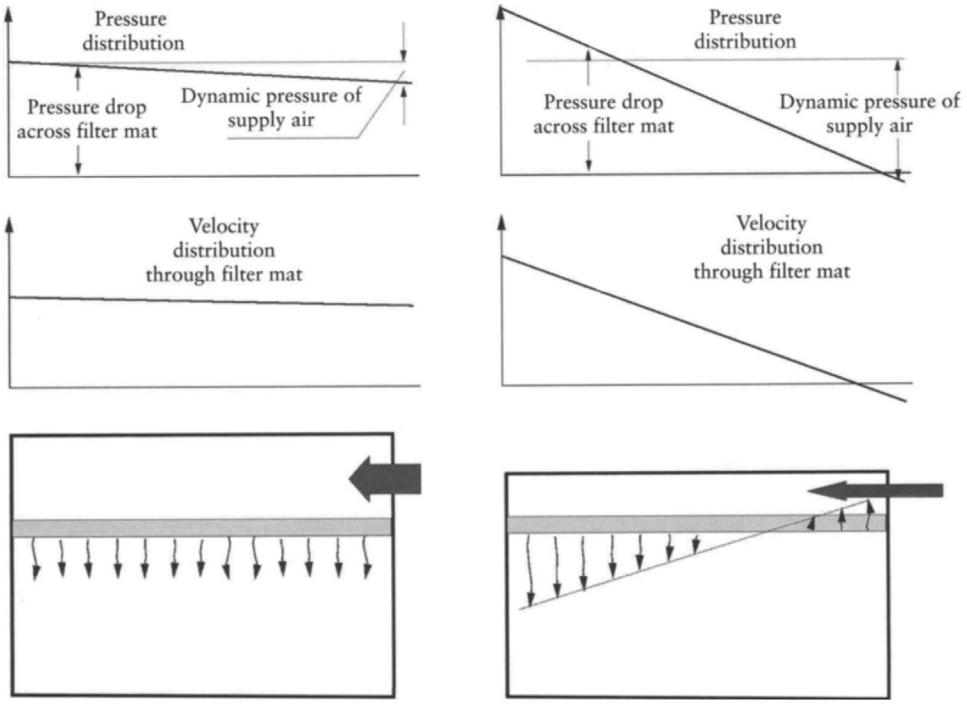


FIGURE 8.22 Velocity distribution along a filter mat.

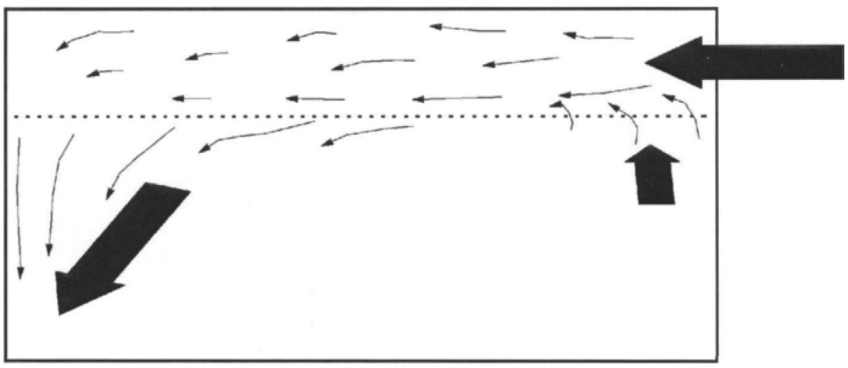


FIGURE 8.23 The outflow from perforated sheets is vulnerable to tilted inflow and to uneven pressure distribution.

to make the supply air layer at floor level. Thus the ventilation air cannot be used for heating the room.

That given above is an example of displacement ventilation with weak thermal stratification. Even though the stratification is weak, the contamination in the lower, cleaner zone is normally on the order of one-third of the contamination in the upper zone.

Fig. 8.27 shows an example of displacement ventilation in a silicon carbide furnace room. The thermal stratification is very strong, as indicated in the graph on the right-hand side of the figure.

8.5.4.2 Cold contaminants

An example of displacement ventilation with cold contaminants is shown in Fig. 8.28. The contaminants, that is, the fumes from the sewer, are colder than the room air, and thus tend to stay low due to the negative buoyancy.

A good arrangement in this case is to locate the air supply below the ceiling. The supply air should not be colder than the room air, in order to layer below the ceiling. The fresh air will fill the room from above.

In this case, heating is possible by means of the supply air.

8.5.5 Zonal air distribution

8.5.5.1 Design requirements for achieving the zoning strategy

The aim of the zoning strategy is to have control of temperature, concentration, or humidity over a certain

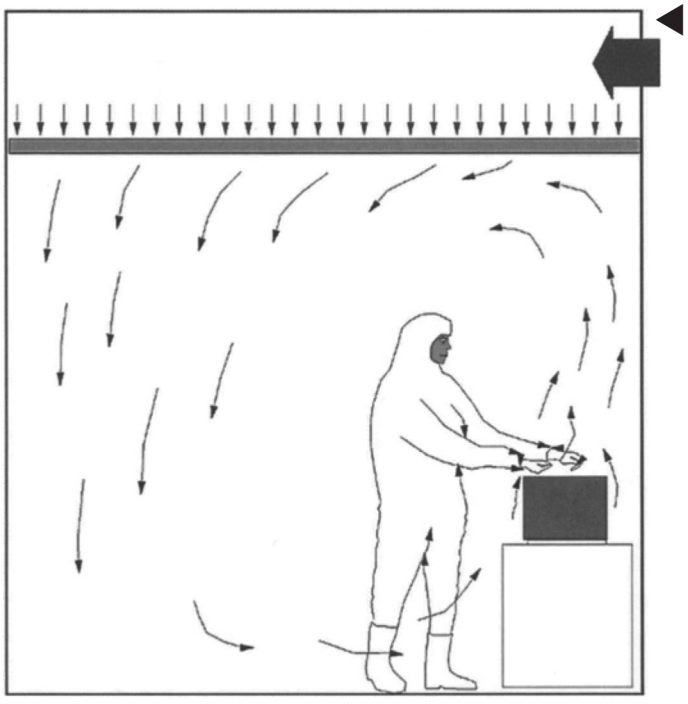
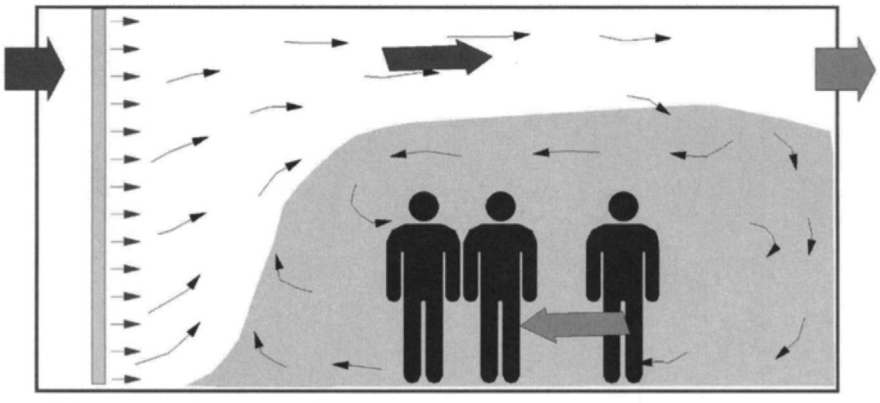


FIGURE 8.24 A cold piston flow from above is unstable, and will make the room air turn upside down.



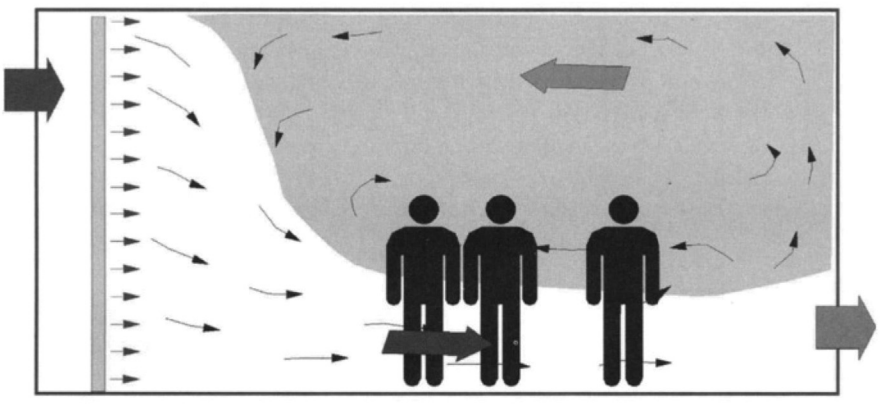


FIGURE 8.25 Horizontal piston airflows are possible only in isothermal conditions.

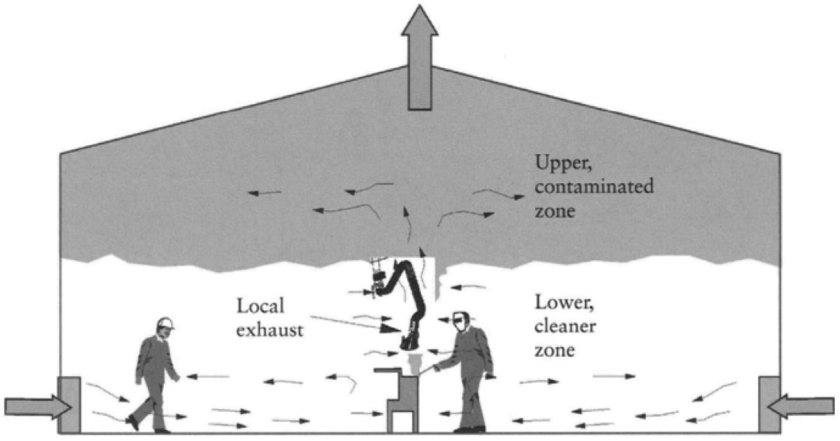


FIGURE 8.26 Displacement ventilation in a welding shop.

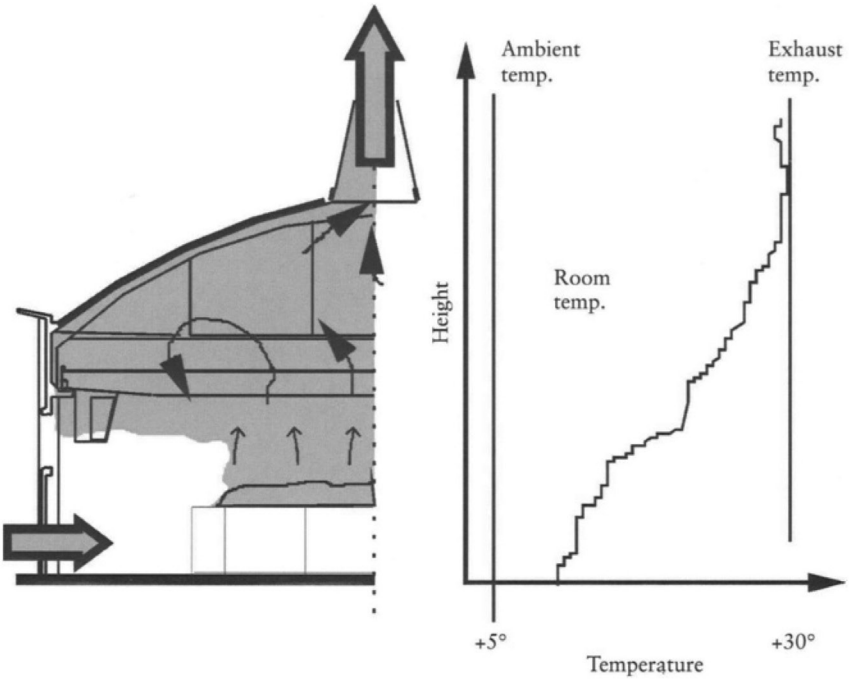


FIGURE 8.27 Displacement ventilation in a silicon carbide furnace room.

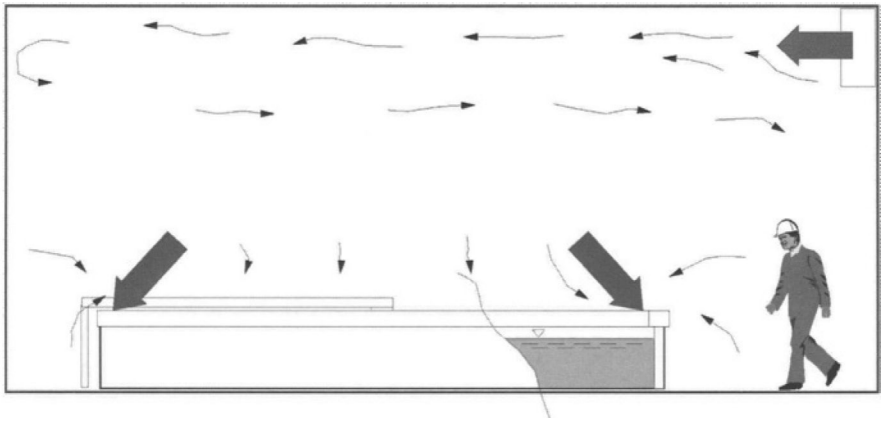


FIGURE 8.28 Sewer treatment plant. The fumes from the sewer are cold, and can be exhausted at the rim of the treatment basin.

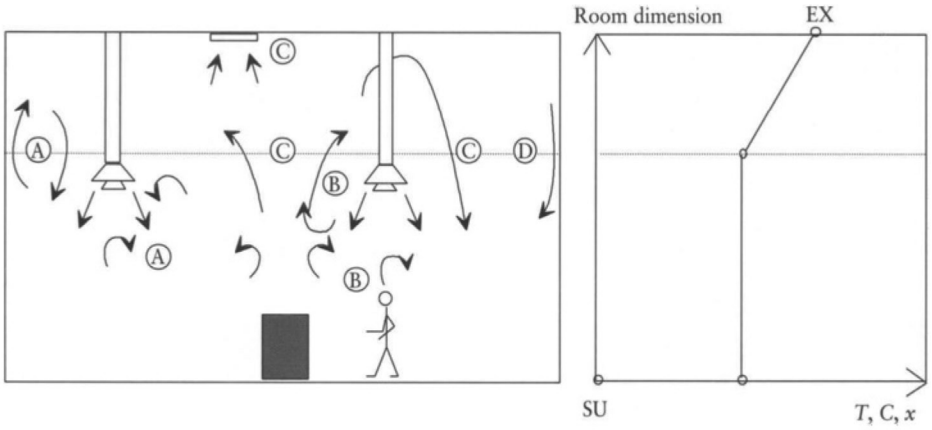


FIGURE 8.29 The key flow elements in the zoning strategy.

volume of the room, while the rest of the room is left with less attention. In most cases the accumulation of heat, contaminants, or humidity outside the controlled zone is desired. The room airflows are controlled partly by the supply air jets, as in the mixing strategy, and partly by the buoyancy sources, as in the stratification strategy. Depending on the air distribution method and dimensioning of the supply air devices, unnecessary mixing is avoided in order to achieve the highest heat and contaminant removal effectiveness possible.

The key flow elements in the zoning strategy are the supply air jets, plumes of the buoyancy sources, buoyant airflows along the surfaces, and turbulent mixing between the controlled and the uncontrolled zones, as in Fig. 8.29. These flow elements have significant influence on the effectiveness of the system.

There are four principal ideas in achieving uniform conditions in the controlled zone and a high heat and contaminant removal effectiveness:

1. Supply air is evenly distributed into the controlled zone. The momentum of the jets is high enough to ensure the uniform conditions but also low enough to avoid mixing in the whole room, that is, the turbulent mixing between the zones is low. This means that usually the number of inlet devices is high.
2. The momentum flux of the plumes is high enough to penetrate the supply airflow patterns. The penetration depends on the location of the plumes in relation to the supply airflow patterns.
3. The plume airflow rate in relation to the extract airflow rate from the uncontrolled zone is low enough to avoid undesired return flow from the uncontrolled zone to the controlled zone.
4. Disturbance flows on the zone boundary should be avoided because of the undesired return flow from the uncontrolled zone to the controlled zone.

The accumulation of heat, contaminants, and humidity is usually vertical in the room, but horizontal zoning is also possible. The same principal ideas should be followed in those cases.

8.5.5.2 *Two-zone model for zoning strategy*

When the zoning strategy is applied, the two-zone model is a useful and simple tool for the determination of the thermal, contaminant, and humidity accumulations.

Relatively uniform conditions in the controlled zone are characteristic for the zoning strategy. By assuming uniform conditions also in the uncontrolled zone, the following two-zone model can be developed. The controlled zone boundary should be defined high enough to get nearly all the induction air of the supply air devices from the controlled zone. This depends on the air distribution method used and the dimensioning of the devices.

Figs. 8.30 and 8.31 describe a two-zone model application of the zoning strategy where all the main variable parameters are presented. Fig. 8.30 (temperature model) describes the accumulation of heat and Fig. 8.31 (concentration model) the accumulation of contaminants. After solving for the temperatures, heat flows, and airflows, contaminant concentrations can be calculated. The models are here determined for stationary loads, airflow rates, and indoor/outdoor conditions, but they can be developed also for dynamic simulations.

The temperature model is based on the air mass flow rate and heat flow rate balances in the lower (controlled) zone (lz) and the upper (uncontrolled) zone (uz).

The turbulent mixing (q_{bt}) between zones and the penetration of the plume airflows (q_{cbm}) through the supply airflow patterns must be determined specially for the air distribution method and devices used as well as the locations of plumes and supply air devices.

The radiation heat transfer (φ_r) from the heat loads such as machines, lamps, persons, and sun has to be determined separately for the lower zone (φ_{rlz}) and upper zone (φ_{ruz}). The radiation between zone wall surfaces (φ_{rwuz}) has to be determined as well.

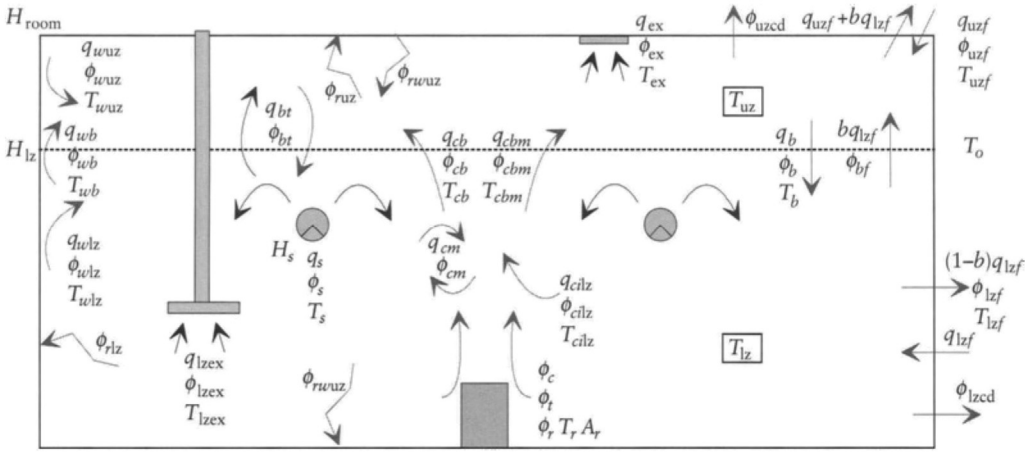


FIGURE 8.30 Two-zone temperature model of the zoning strategy.

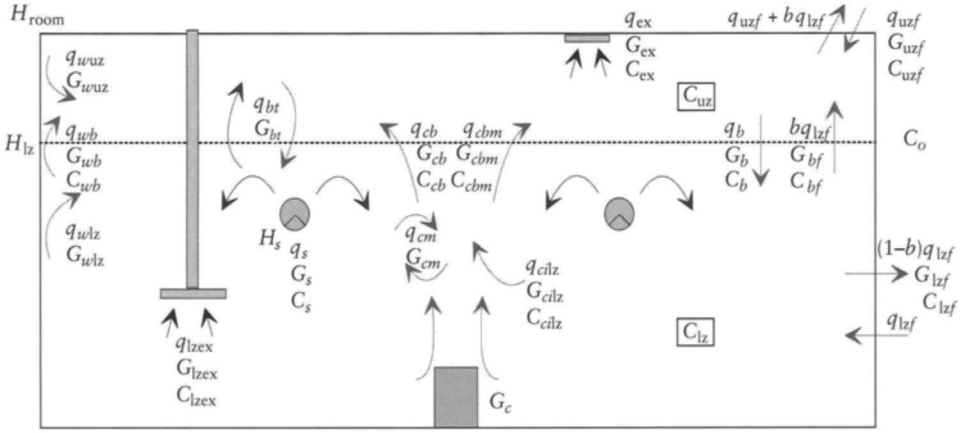


FIGURE 8.31 Two-zone concentration model of the zoning strategy.

Local exhaust airflows from the hoods in the lower zone (q_{lzex}) reduce the heat removal effectiveness, because the return airflow rate (q_b) from the upper zone to the lower zone is increased.

The infiltration airflow into the lower zone (q_{flz}) is assumed to be higher than the exfiltration. The difference (bq_{flz}) describes the airflow rate into the upper zone caused by filtration.

The wall surface temperatures (T_{wuz}) and (T_{wlh}) are calculated separately for each wall and window surface by means of heat transfer through the wall: $T_{wuz} = T_{uz} - (U_{uz}/h_{uz})(T_{uz} - T_o)$ and $T_{wlz} = T_{lz} - (U_{lz}/h_{lz})(T_{lz} - T_o)$, where h is the heat transfer coefficient on the surface and U is the overall heat transfer coefficient. The airflows along the wall surfaces through the zone boundary (q_{wb}) are calculated separately for each wall and window surface.

The concentration model is based on the air mass flow rate and contaminant flow rate balances in the lower (controlled) zone (lz) and the upper (uncontrolled) zone (uz).

The airflows on the wall surfaces and the air filtration through the walls significantly influence contaminant accumulation, and therefore it is essential to carry out the calculation also for the cold season.

Nomenclature for the figures

A = area (m^2)
b = coefficient
C = concentration (mg/m^3)
H = height (m)
G = contaminant flow rate (mg/s)
q = airflow rate (dm^3/s)
φ = heat flow rate (W)
T = temperature (K, °C)

Subscripts

a = air
b = boundary, through the boundary
c = convection
cd = conduction
ex = exhaust, extract

f = filtration
lz = lower zone
m = mixing, mixed
o = outside
r = radiation
s = supply
t = turbulent mixing, total
uz = upper zone
w = wall

8.5.5.3 *Characteristics of the zoning strategy*

The air distribution method and dimensioning of the air supply devices are important factors in determining the accumulation of heat and contaminants. After the behavior of the air distribution method and devices are known, the characteristic effects of the other airflow elements can be calculated.

The simplified examples in Figs. 8.32–8.36 are calculated for the case of an industrial hall with length 40 m, width 25 m, and height 8 m. The zone level is determined at the height of 4 m. The dimensions of the heat (50% convection) and contaminant sources located at floor level are 1, 1, and 1 m. The heat load of the lights located in the upper zone at the level of 6 m is 15 W/m^2. For the walls the U-value is 0.5 W/m^2 K. The turbulent mixing between the zones is estimated as $q_{bt} = 0.5q_s$ or $q_{bt} = 1.0q_s$. The penetration of the plumes through the zone boundary is estimated to be 100%, that is, $\Sigma\, q_{cbm} = \Sigma\, q_c$. Contaminant concentration outside is $C_o = 0$. Other values are listed in the tables of the figures.

The effect of the plume airflow rate and the turbulent mixing airflow rate through the zone boundary is presented in Fig. 8.32. The heat removal effectiveness ϵ_T and contaminant removal effectiveness ϵ_C are presented as functions of the relative airflow rate.

The effect of the supply airflow rate and the heat load is presented in Fig. 8.33. The heat removal effectiveness ϵ_T and contaminant removal effectiveness ϵ_C are presented as functions of the supply airflow rate. The power of one heat source is 500 W.

The effect of the local exhaust airflow rate in the lower zone is presented in Fig. 8.34. The heat removal effectiveness ϵ_T and contaminant removal effectiveness ϵ_C (determined by extract air) are presented as functions of the local exhaust airflow rate. The total heat load is 60 W/m^2 and the power of one heat source is 500 W. The supply airflow rate is 8 L/s m^2.

The effect of the downward airflow along the wall surfaces is presented in Fig. 8.35. The heat removal effectiveness ϵ_T and contaminant removal effectiveness ϵ_C are presented as functions of the outdoor temperature T_o. The total heat load is 60 W/m^2 and the power of one heat source is 500 W. The supply airflow rate is 8 L/s m^2. In

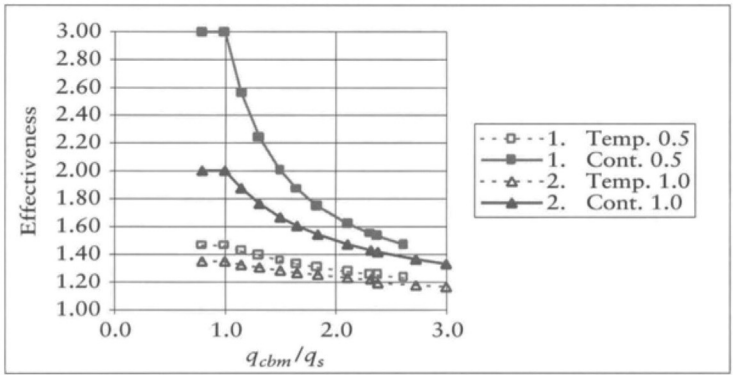


FIGURE 8.32 Effectiveness ϵ_T (temp.) and ϵ_C (cont.) as functions of the ratio q_{cbm}/q_s with the ratio q_{bt}/q_s as a parameter.

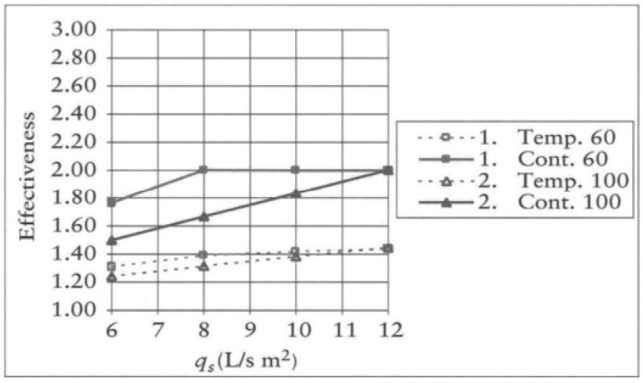
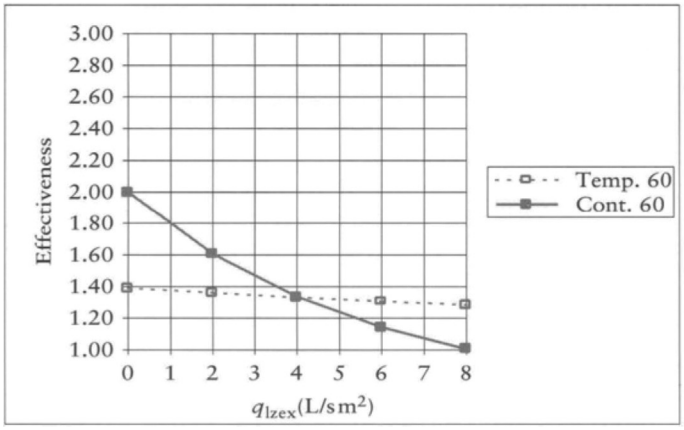
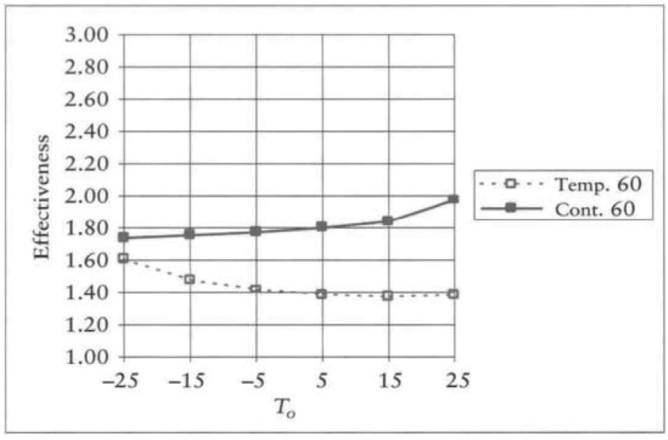


FIGURE 8.33 Effectiveness ϵ_T (temp.) and ϵ_C (cont.) as functions of the supply airflow rate q_s (L/s m^2) with the total heat load φ_t (W m^{-2}) as a parameter.



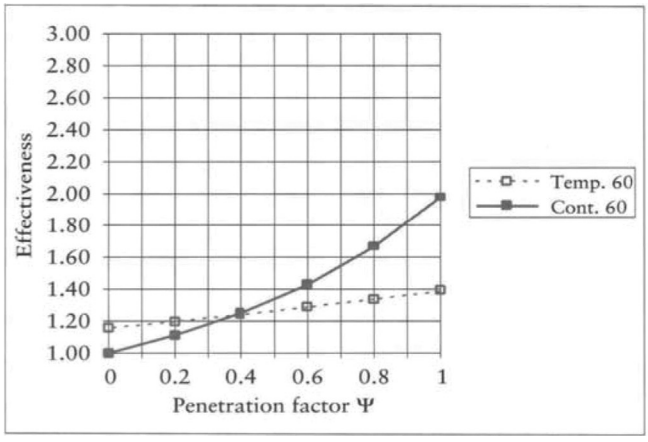
$T_o = 25°C$
$T_{uz} = 26°C$
$q_{wb} = 0$
$q_{flz} = q_{fuz} = 0$
$q_s = 8$ L/s^{-1}m^{-2}
$q_{lzcd} = q_{uzcd} = 0$
$\Phi_t = 60$ W/m^{-2}
$\phi_t = 500$ W/apiece
$q_{bt}/q_s = 1$

FIGURE 8.34 Effectiveness ϵ_T (temp.) and ϵ_C (cont.) as functions of the local exhaust airflow rate from the lower zone.



$T_{lz} = 20–24°C$
$T_{uz} = 21–26°C$
$q_{flz} = q_{fuz} = 0$
$q_s = 8$ L/s m^2
$\Phi_{cd} = 42.0$ W/m^2
$\Phi_t = 60$ W/m^2
$\phi_t = 500$ W/apiece
$q_{bt}/q_s = 1$

FIGURE 8.35 Effectiveness ϵ_T (temp.) and ϵ_C (cont.) as functions of the outdoor temperature T_o.



$T_o = 25°C$
$T_{uz} = 26°C$
$q_{wb} = 0$
$q_{flz} = q_{fuz} = 0$
$q_s = 8$ L/s m^2
$q_{lzcd} = q_{uzcd} = 0$
$\Phi_t = 60$ W/m^2
$\phi_t = 500$ W/apiece
$q_{bt}/q_s = 1$

FIGURE 8.36 Effectiveness ϵ_T (temp.) and ϵ_C (cont.) as functions of the penetration factor of the plume airflow rate.

winter seasons heat losses through the walls and the airflow along the walls increase the relative temperature difference and decrease the concentration difference.

The effect of the disturbance of the supply airflow on the plumes is presented in Fig. 8.36. The heat removal effectiveness ϵ_T and contaminant removal effectiveness ϵ_C are presented as functions of the penetration factor Ψ of the plume, which is the ratio of penetrated plume airflow rate to the whole plume airflow rate. The total heat load is 60 W/m^2 and the power of one heat source is 500 W. The supply airflow rate is 8 L/s m^2.

These case examples illustrate the dependence of the stratification of temperature and contaminants on several parameters, which in some cases increase and in other cases decrease the effectiveness. All the parameters should be included in calculations when designing the system combination of the room air conditioning methods.

Measured results of effectiveness and turbulent mixing are presented in literature by Bach (several air distribution methods)[27] and Hagström et al. (mixing air distribution methods in zoning strategy).[28,29] A typical example of an air distribution method and device in the zoning strategy is the so-called active displacement method, which is based on a nozzle duct device.[30]

8.6 Location of general exhaust

8.6.1 Exhausts in nonstratified room air

In a room with perfect mixing of the air, it theoretically does not matter where the exhaust opening is located (Fig. 8.37). In practice, air seldom mixes as completely as in theory. One reason for this is temperature differences or density differences. The contaminants are often warmer than the room air, and in some cases the density of the contaminant itself differs from the air density. These topics are treated in the later sections. This section will focus on isothermal, nonbuoyant cases.

In long rooms, that is, where the room is slightly longer than the penetrating distance of the supply jet, the best location of the exhaust is opposite to the air supply opening. See Fig. 8.38.

In very long rooms the exhaust should be located at the opposite end of the room. Otherwise, the air exchange in the far end of the room may be small, resulting in an accumulation of contaminants in that part of the room.

8.6.2 Exhaust of buoyant contaminants

8.6.2.1 Exhaust of warm fumes

The general rule is that when contaminants are warm, the exhaust should be located as high as possible in the room. Obviously, this applies in the case of displacement ventilation, as shown in Fig. 8.39.

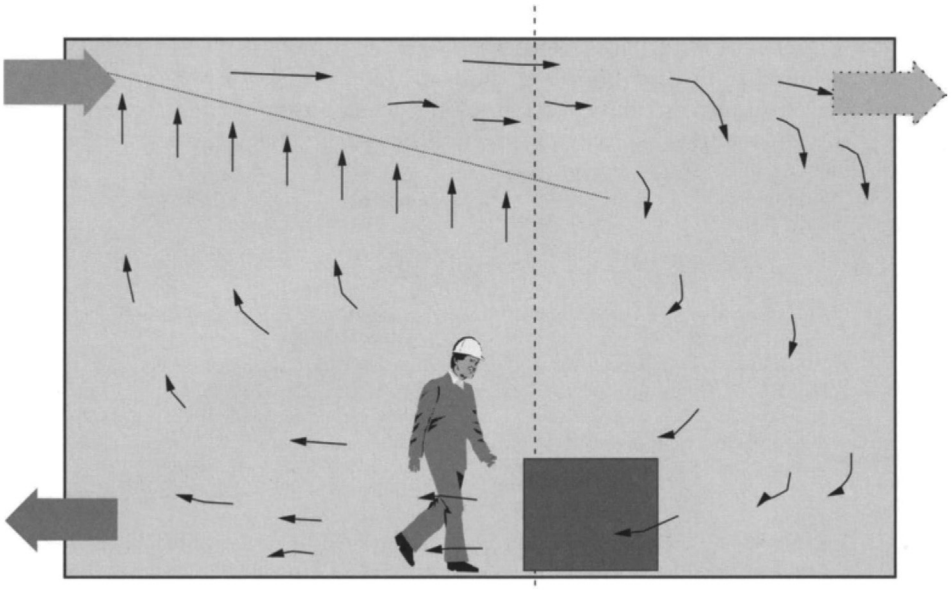


FIGURE 8.37 Exhaust from a room with perfect mixing: the location of the exhaust opening is not important.

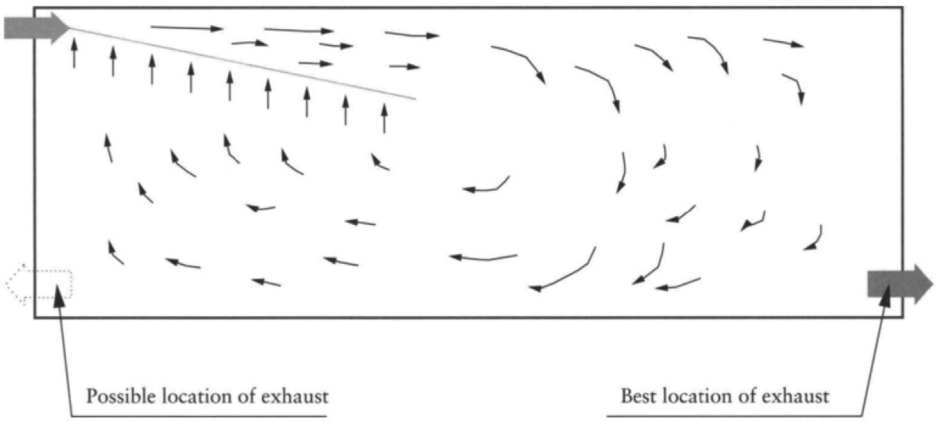


FIGURE 8.38 In very long rooms, the exhaust should be placed opposite to the supply opening.

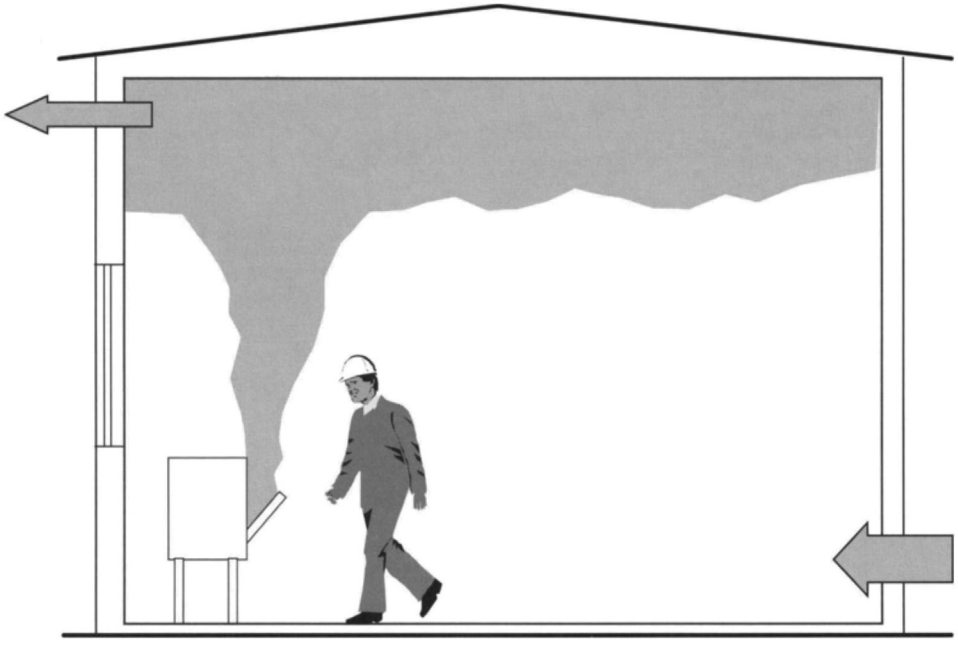


FIGURE 8.39 Warm contaminants should be exhausted below the ceiling, and preferably close to the source.

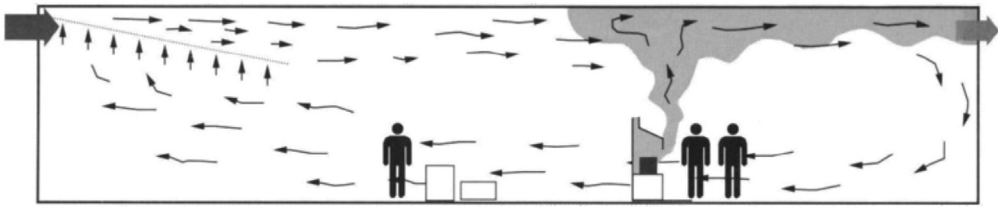


FIGURE 8.40 Buoyant contaminants should be extracted below the ceiling (also in mixing ventilation).

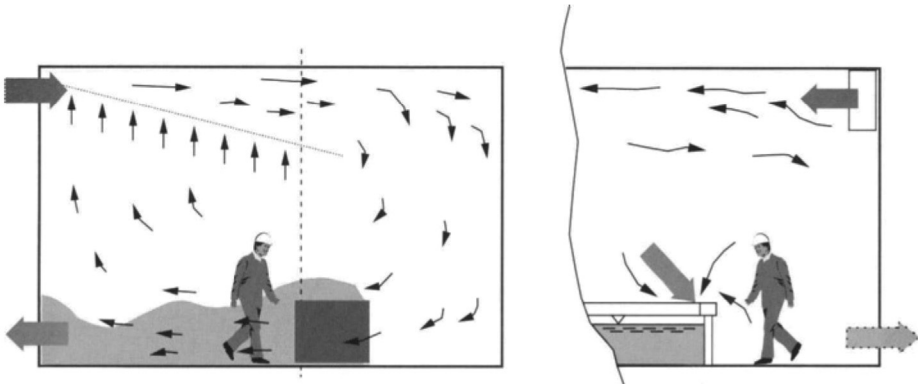


FIGURE 8.41 When the contaminants are heavier than the room air, the exhaust should be at floor level, for displacement ventilation as well as for mixing ventilation.

In mixing ventilation, the exhaust should be located below the ceiling when the contaminants are buoyant, as illustrated in Fig. 8.40.

8.6.2.2 *Exhaust of cold fumes*

Heavy fumes or gases (i.e., negatively buoyant contaminants) should be exhausted at floor level. This applies to displacement ventilation as well as to mixing ventilation (Fig. 8.41).

8.6.2.3 *Exhaust of fumes with unpredictable buoyancy*

In several cases, it is not certain if the contaminants are negatively or positively buoyant. In such cases one

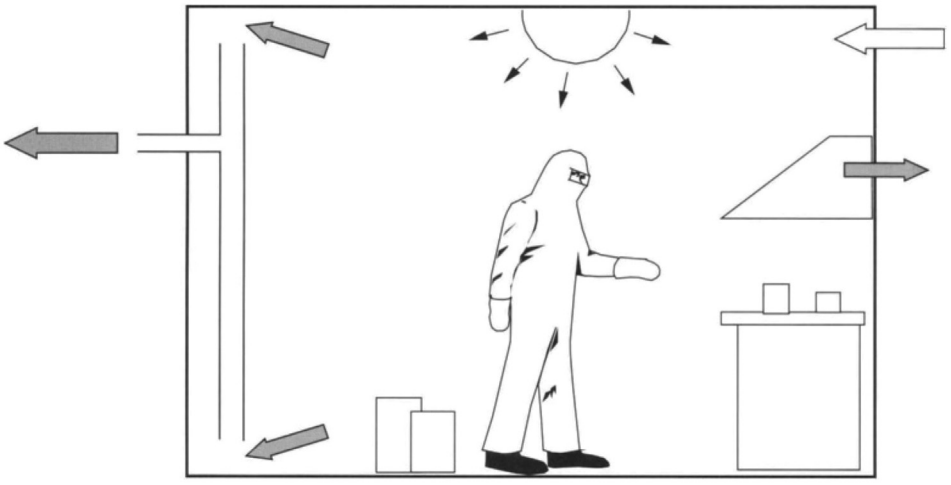


FIGURE 8.42 When contaminants can be both negatively and positively buoyant, exhausts should be located both high and low.

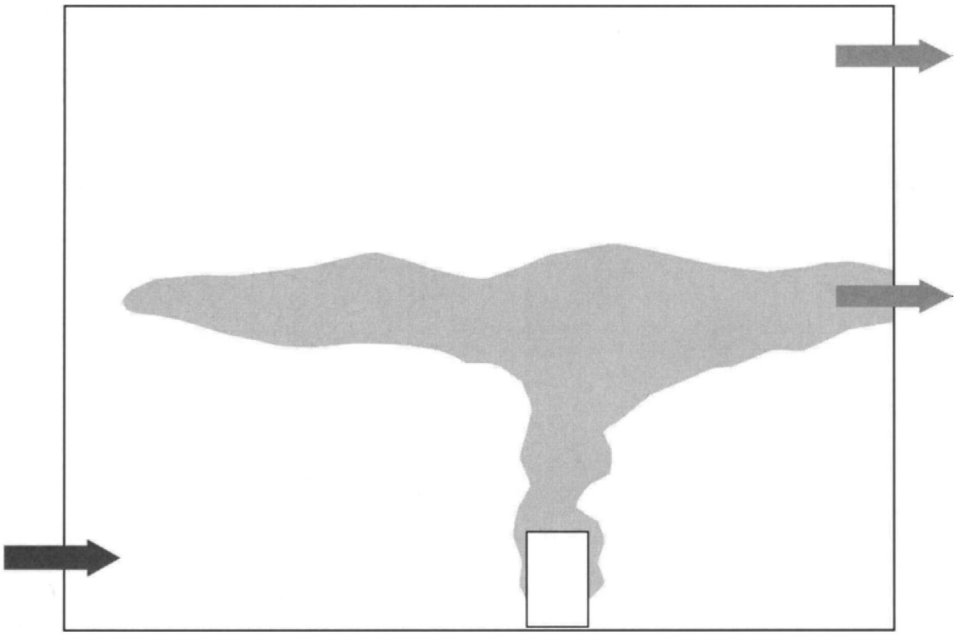


FIGURE 8.43 Thermal stratification should be taken into account when locating the exhaust openings.

should place exhaust openings both below the ceiling and at floor level (Fig. 8.42).

8.6.3 Exhausts in stratified room air

Thermal stratification in the room air greatly influences the spreading and dispersion of contaminants (Fig. 8.43). In such cases, an exhaust opening might be placed at, or close to, the equilibrium height for the main contaminant. In addition, there should be a general exhaust either at ceiling level or at floor level, depending on the buoyancy of the contaminants. A temporary wall (can be canvas) can separate different zones, so that a polluted zone is separated from a clean zone (e.g., the welding zone separated from the mechanical assembly zone, Fig. 8.44).

8.6.4 Location of general exhaust to create displacement flow

When special contaminating processes are located in a semienclosed part of a room like in Fig. 8.45, the exhausts should be located so that a displacement flow is created through the passage into the semienclosure.

With this arrangement, there may be no need for a door, which may be very practical.

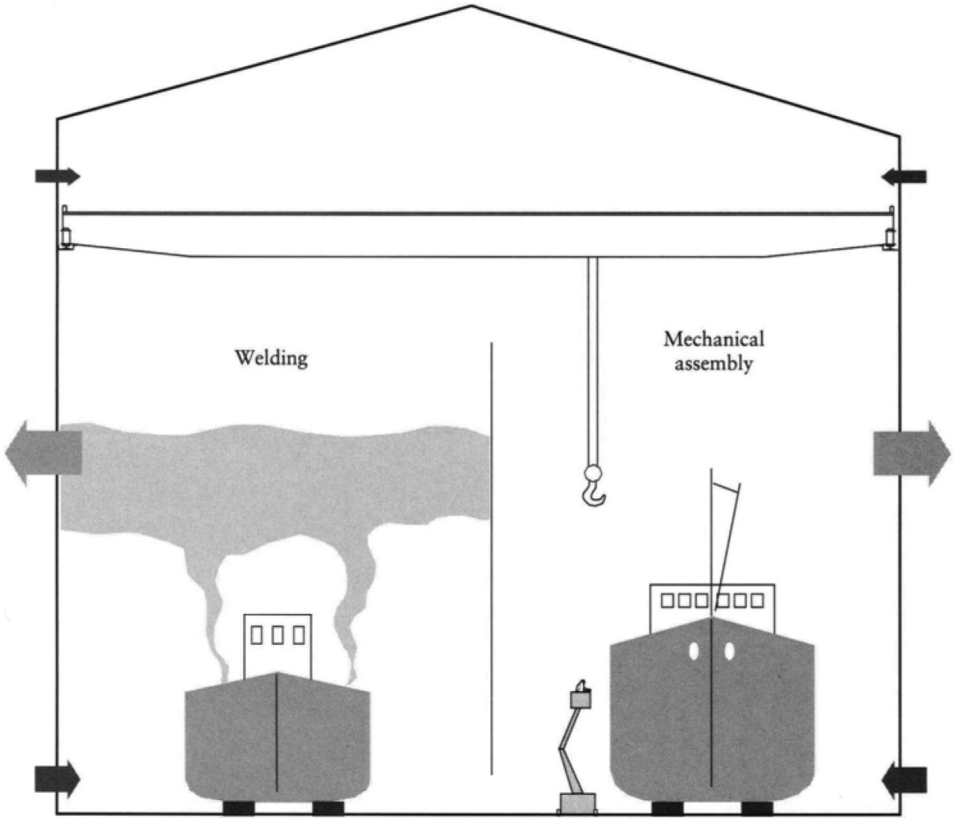


FIGURE 8.44 In stratified air, we can create separate zones in a large enclosure, using the thermal lid as a roof above the zones.

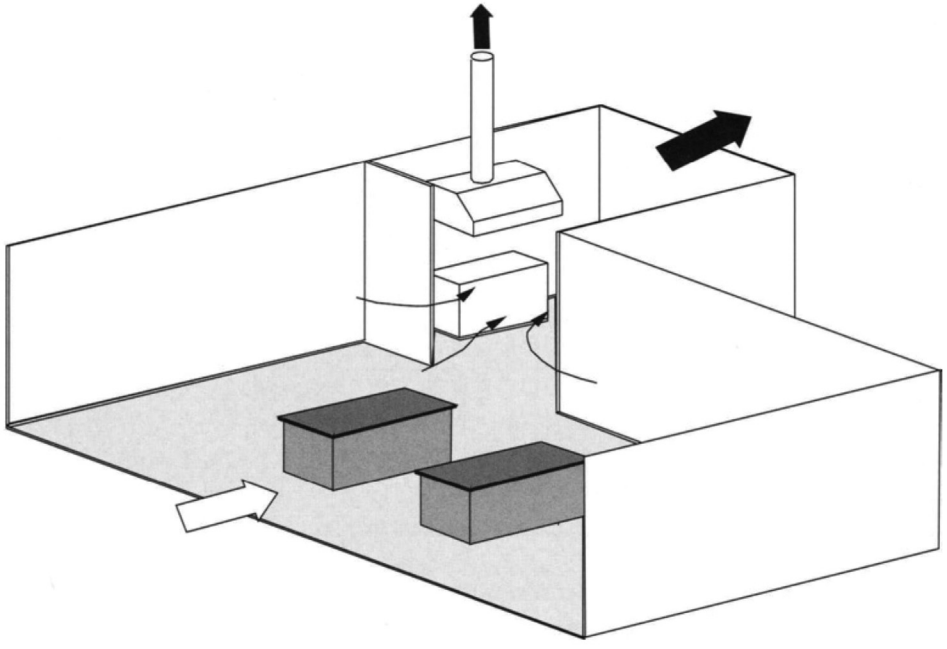


FIGURE 8.45 General exhaust placed to give a displacement flow through a passage.

8.7 Air recirculation

8.7.1 Introduction

One way to reduce the use of outside air—which needs heating, cooling, humidifying, dehumidifying, or cleaning—is to recirculate the air. Other ways to save energy for a ventilation system should preferably be used, such as the use of a heat exchanger, equipment that uses less energy, use of less air, etc. The main difference between other energy-saving processes and air recirculation is its influence on the contaminant concentration. Recirculation always includes some cleaning of the air. If the air is not cleaned, the process is seen as transferring instead of recirculating the air.

Through the years many reports have been published on air recirculation. Mostly they have treated the problems in offices, living rooms, and schools. There have been many reports on air recirculation, which have concentrated on the use of some specific air cleaning system or on possible health risks. In all cases the ventilation should be high enough to remove internal pollutants.[31]

From the literature, it is evident that air recirculation has been used and is used for many different types of ventilated areas. For this reason, there have been some large investigations into the advantages and disadvantages of using air recirculation in industry.[32–37] These reports cover nearly all aspects of using air recirculation in all kinds of industrial premises and together with all kinds of industrial processes: calculation and design models, supervising, economics, planning, health aspects, regulations, environmental issues, and validation.

Air recirculation is quite common in electronic and pharmaceutical industries, where cleanroom techniques are used. In this case the highly cleaned air, after use, is still cleaner than the outside air and is thus cheaper to clean again than to discard. It has also been used in chemical laboratories, from pneumatic transports, from oil mist eliminators, in airplanes, in mines, in hospitals, and for isolation units. Very common air recirculation systems are vacuum cleaners, which from the contaminant point of view whose behave is similar to other recirculation systems. In industry many dust collectors are used as recirculation systems.

Sometimes recirculation systems are used in premises where a health risk could occur if the cleaning system stops working or for some other reason the concentration in the return air becomes higher than presumed. This has resulted in the use of control systems. These can consist of a simple detector or could be a very advanced instrument connected to an alarm and a control system. The use of an instrument to supervise the recirculation system could thus be necessary and must also be checked. Instruments from different manufacturers and for different contaminants have different performances, which makes it difficult to choose and handle the proper instrument. Different countries have different policies regarding the necessity of when and where to use a supervising system.

As a general rule, recirculation should be used only if it does not lead to an increase in contaminant concentration in the workspaces, and recirculation should be avoided if a malfunction could result in dangerous circumstances.

8.7.2 Different recirculating systems

An easy way to divide air recirculation systems is to distinguish between local and central recirculation systems.

Central recirculation means that some part of the air exhausted from a room is passed through an air cleaner and then returned to the room, either directly or by mixing into the supply air. When the main contaminant source is in the supply air, this system should result in cleaner air than when using only outside air. Since a part of the exhausted air is recirculated, the maximum recirculation ratio is 1 (ratio between return air flow rate and total flow rate) and this happens when all air is recirculated.

Local recirculation means, for example, dust collectors and vacuum cleaners, in which all exhaust air is return air, are is distributed into the same room as where they are placed. This type also includes room air cleaners and laboratory fume hoods with immediate recirculation. Local recirculation systems consist of an exhaust hood, an air cleaning system (filter), a fan, and an outlet into the room. Since there could be many such systems in a room, independent of the ventilation system for the room, and the airflow rate in each system is independent of the room ventilation system, the possible recirculation ratio is in principle unlimited. In practice, it is seldom larger than approximately 10. Another difference with central systems is that their performance is heavily influenced by the exhaust hood. If all contaminants generated at a process are captured by the exhaust hood, the performance is similar to a central system, but since this is a very rare phenomenon, the resulting concentration in the room is influenced by the capture efficiency.

The following Eqs. (8.18–8.20) separately outline calculating contaminant concentration inside a room with central and local recirculation. The assumptions for the room are that it has one main ventilation system with supply and exhaust air and that the contaminant

concentration is the same in the whole volume (except very close to the contaminant source or in the ducts, etc.). The contaminant source is steady and continuous. The model for local ventilation assumes also one main ventilation system to which is added one local exhaust hood connected to a local ventilation system from which all the air is recirculated. In the central system the number of inlets and outlets could vary. The flow rates are continuous and steady.

Different models have been published using two or more zones, or where a mixing constant is used. Models also exist for multiple ventilation systems in a room or a building, for varying source rates, and where local and central recirculation take place at the same time.

The most important part of a recirculating system is the cleaning device. Different types of air cleaning devices are described in Chapter 9, Air Handling Processes. The cleaner must be suited for the contaminant to be collected and its concentration. This sounds obvious but is very often forgotten. Cleaners for aerosols (dusts) have efficiencies depending on particle size; for gases, efficiency varies with concentration and type of gas. The efficiency also varies with time, both for particle and gas cleaning systems. For mixtures of gases (vapors) and particles, such as oil mist, it is usually not enough to separate the particles (drops) from the air; the gases must also be collected.

Please note that filtering in dust collectors, vacuum cleaners, etc. does not usually collect the smallest particles. A good rule is not to recirculate such air.

The most common equipment for separating particles from recirculated air is fabric filters, mechanical collectors, electrostatic precipitators, and cleaners and wet collectors. For cleaning of recirculated air from gases, absorbers and adsorbers, such as activated carbon, sometimes with impregnation for specific gases, and impregnated alumina are most common. The performance of different air cleaning equipment is described in many textbooks and handbooks.

The models described in the following use only one parameter for the cleaning efficiency, which is thus a simplification that must be kept in mind when using these models. This works quite well as long as the efficiency value is the smallest one, for example, the efficiency for the most penetrating particle size or the efficiency for the most penetrating gas concentration.

8.7.3 Central recirculation system

Typically, when central recirculation is used the contaminant in the supply air is the main source. This is not the case for industrial use, where the main source is in the ventilated room. This usually results in the concentration being somewhat higher when using recirculation than when not using it. Fig. 8.46 outlines the ventilation system, the contaminant source, and the cleaning system.

In the figure the two flows (supply—exhaust and recirculated) are separated for clarity. Normally they are merged on both the supply and exhaust side.

The following differential equation (or something similar), derived from a mass balance for the room, is solved to find the correlation between flow rates, source rate, contaminant concentrations, cleaning efficiency, and time.

$$(1-\kappa_c)q_{\text{airtor}}c_s + \kappa_c q_{\text{airtot}}c(1-\eta) + q_m = q_{\text{airtot}}c + V\frac{dc}{dt}, \tag{8.18}$$

where the recirculation ratio, κ_c, is defined as

$$\kappa_c = \frac{q_{\text{airret}}}{q_{\text{airtot}}}$$

and κ_c is always less than 1; q_{airret} is the return airflow rate, m^3/s; q_{airtot} is the total flow rate through the room, m^3/s; c_s is the concentration in the supply air, mg/m^3; c is the concentration in the room, mg/m^3; η is the efficiency of the cleaner (0−1); q_m is the source rate, mg/s; V is the volume of the room, m^3; and t is time, s.

It is possible to have a separate recirculating system in addition to the general ventilation system; then there is no restriction on the flow rate. This case is the same as a recirculating local exhaust system (see later).

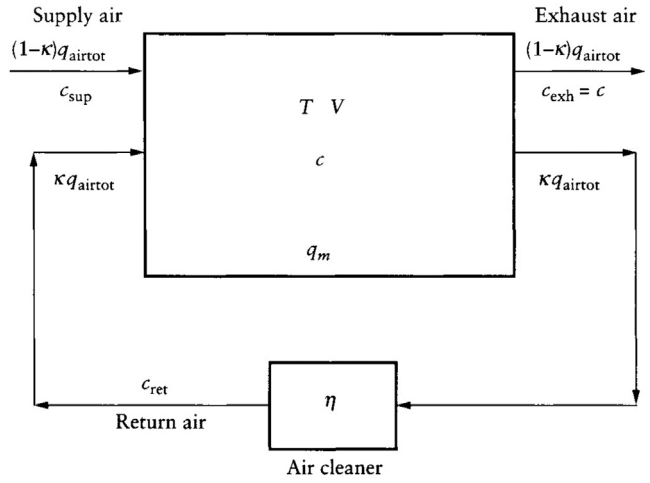


FIGURE 8.46 Model of a central recirculating system used for calculating the connection between contaminant concentrations, airflow rates, contaminant source strength, q_m, and air cleaner efficiency, η. c_{sup} is the concentration in the supply (outside) air, c is the concentration in the room, c_{ret} is the concentration in the returned air, q_{airtot} is the total flow rate through the room, κ is the ratio between recirculated airflow rate and total air flow rate, T is the time constant for the room, and V is the room volume.

By assuming the supply air concentration to be zero, since usually there are quite different contaminants in the outside air and from the source, and that the initial concentration also is zero, the time-dependent solution is the following:

$$c = \frac{q_m}{q_{\text{airtot}}} \times \frac{1}{(1 - \kappa_c + \kappa_c\eta)}(1 - e^{-(1-\kappa_c+\kappa_c\eta)t/T}). \quad (8.19)$$

where T is the time constant for the room, s, equal to V/q_{airtot}.

By assuming a steady state and since the sum of the supply and return flow rates is equal to the total flow rate through the room, it is possible to manipulate this equation to get the following, where q_{airsup} is the supply airflow rate, m^3/s:

$$c = \frac{q_m}{q_{\text{airsup}} + q_{\text{airret}}\eta}. \quad (8.20)$$

This equation shows clearly the influence of the recirculated air. With a source rate of q_m and a general ventilation flow rate of q_{airsup}, the concentration is q_m/q_{airsup}. The addition of recirculated air corresponds to an increase of the supply flow rate by the amount q_{airret} times η and decreases the concentration. On the other hand, if a part of the original supply air without recirculation (that is, the total flow rate before recirculation is equal to $q_{\text{airsup}} + q_{\text{airret}}$) is recirculated, the concentration will increase.

Another solution to the differential equation for steady state is the following, where the concentration in the supply air, c_{airsup}, is included:

$$c = \frac{q_m}{q_{\text{airtot}}} \frac{1}{(1 - \kappa_c + \kappa_c\eta)} \times \frac{c_{\text{airsup}}(1 - \kappa_c)}{(1 - \kappa_c + \kappa_c\eta)} \quad (8.21)$$

This equation is the same as the solution for the steady state without contaminant in the supply air, but with an added multiplication term (the third part on the right) that shows the influence of the recirculation.

Another equation that includes the initial concentration and the concentration in the supply air and the mixing factor has been published:

$$c = c_0 e^{-f(q_{\text{airsup}}+\eta q_{\text{airret}})t/V} + \frac{f\, c_{\text{sup}} q_{\text{airsup}} + q_m}{f(q_{\text{airsup}} + \eta q_{\text{airret}})} \times \left(1 - e^{-f(q_{\text{airsup}}+\eta q_{\text{airret}})^{t/V}}\right) \quad (8.22)$$

where c_0 is the initial concentration at time zero, mg/m^3, and f is the mixing factor, here defined as the portion of the supply airflow that is completely mixed with room air.

This equation can also be used for steady-state conditions, when the exponential terms are taken away and for complete mixing when f is set to 1.

8.7.4 Local recirculation

Local recirculation systems differ from central systems in that all exhausted air is passed back to the room after cleaning and that the flow rate could be larger than the flow rate through the room.

One of the most common systems for cleaning air in homes, offices, schools, etc. is the room air cleaner. Fig. 8.47 outlines a model of a local recirculating system. Usually, these units are situated inside the room if they are small and movable (see Chapter 10). For the model, it does not matter if the unit is placed inside or outside the room with the contaminant source, as long as the exhaust and return air openings are inside.

The room air cleaner consists of a fan and some kind of air cleaner for particles or gases or both, usually mounted together as one unit. This is a local recirculating system and the equation for the contaminant concentration in the room, derived with the same assumptions and in the same way as for central systems, is the following:

$$c = \frac{q_m}{q_{\text{airexh}}} \times \frac{1}{1 + \kappa_1\eta} \times (1 - e^{-(1+\kappa_1\eta)t/T}), \quad (8.23)$$

where c is the concentration in the room, mg/m^3; q_m is the source rate, mg/s; q_{airexh} is the exhaust flow rate from the room, m^3/s; κ_1 is the local recirculation ratio equal to $q_{\text{airexh}}/q_{\text{airexh}}$; q_{airrec} is the flow rate through the unit (cleaner), m^3/s; η is the efficiency of the

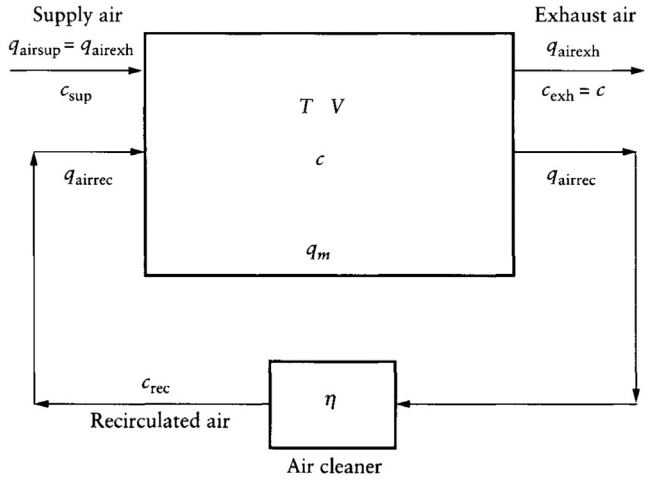


FIGURE 8.47 Model of a local recirculating system (room air cleaner) used for calculating the connections between contaminant concentrations, airflow rates, contaminant source strength, q_m, and air cleaner efficiency, η. c_{sup} is the concentration in the supply (outside) air; c (equal to c_{exh}) is the concentration in the room; c_{rec} is the concentration in the returned air; q_{airtot} is the total flow rate through the room; κ is the ratio between the recirculated airflow rate, q_{airrec}, and total airflow rate; q_{airexh} is the flow rate from the general ventilation system; T is the time constant for the room; and V is the room volume.

cleaner (0–1); T is the time constant for the room equal to V/q_{exh}, seconds; V is the volume of the room, m^3; and t is time, seconds.

By manipulating this equation for the steady state, in the same way as for central systems, the following could be achieved:

$$c = \frac{q_m}{q_{airexh} + q_{iarrect}\eta}, \tag{8.24}$$

which is similar to the equation for central systems. This relation is often expressed as:

$$c = \frac{q_m}{q_{airexh} + q_{airequ}}, \tag{8.25}$$

where q_{airequ} is the so-called equivalent flow rate for the cleaner, because the product of flow rate and cleaning efficiency has the unit of flow rate. In words this could be expressed as follows: the influence of a room air cleaner on the contaminant concentration is the same as if a flow rate of clean air (outside air) equal to q_{airequ} were added to the ventilation flow rate in the room.

From this equation, it is clear that if either flow rate or cleaner efficiency for the recirculation system is zero, there will be no change in contaminant concentration. Also, a low flow rate can only be compensated to a small degree by a higher cleaning efficiency, but a low cleaning efficiency can be compensated to some degree by increasing the flow rate.

This equation makes it quite easy to calculate necessary flow rate and cleaning efficiency for a local recirculation system (room air cleaner).

Local ventilation in industry usually differs from the description above in that it is connected to a local exhaust hood (Chapter 10), which has a capture efficiency less than 100%. The capture efficiency is defined as the amount of contaminants captured by the exhaust hood per time divided by the amount of contaminants generated per each time. Fig. 8.48 outlines a model for a recirculation system with a specific exhaust hood. Here, the whole system could be situated inside the workroom as one unit or made up of separate units connected with tubes, with some parts outside the workroom. For the calculation model, it makes no difference as long as the exhaust hood and the return air supply are inside the room.

The solution to the differential equation at steady state in this case is:

$$c = \frac{q_m}{q_{airexh}} \times \frac{(1 - \alpha\eta)}{(1 + \kappa_1\eta)}, \tag{8.26}$$

where α is the capture efficiency for the local exhaust hood (0–1). To get the time-dependent solution, the right-hand side is multiplied with the same term (the parentheses including the exponential term) as for the system without an exhaust hood.

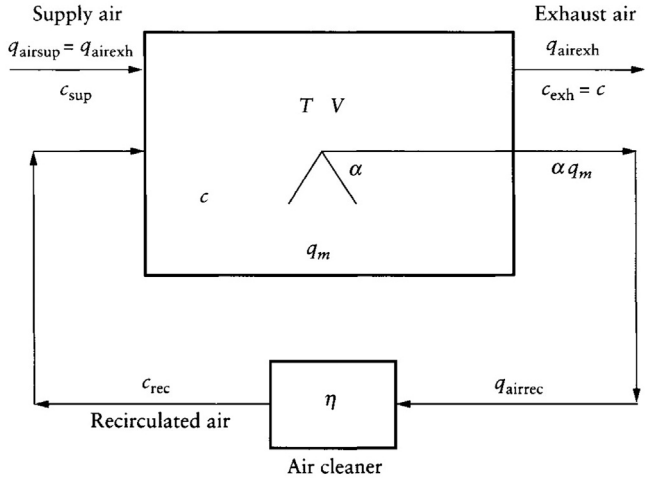


FIGURE 8.48 Model of a local recirculating system with a local exhaust hood, used for calculating the connection between contaminant concentrations, airflow rates, contaminant source strength, q_m, air cleaner efficiency, η and hood capture efficiency, α, c_{sup} is the concentration in the supply (outside) air; c (equal to c_{exh}) is the concentration in the room; c_{rec} is the concentration in the returned air; q_{airsup} is the supply flow rate to the room equal to the exhaust flow rate, q_{airexh}; the recirculated flow rate (through the cleaner) is q_{airrec}; T is the time constant for the room; and V is the room volume.

This latter equation can also be used for systems without a local exhaust hood by setting the capture efficiency to zero. It could also be used to show the result of recirculation from, for example, a laboratory fume hood with immediate recirculation. In such a hood all contaminants are generated within the hood and usually also all generated contaminants are captured, so the capture efficiency is 1. The equation demonstrates that if the cleaning efficiency is zero, there is no change in concentration, and the unit only works as a mixing unit.

8.7.5 Conclusion

To design an air recirculation system, it is necessary to know the performances of fans, air cleaners, and exhaust hoods included in the current system. The equations described here include the source generation rate and the total airflow rate through the room, which could be difficult to measure. The ratio between source rate and flow rate has the unit of concentration and should in fact be equal to the concentration without recirculation. The equations could thus be transformed to include the contaminant concentration without recirculation instead of this ratio. In this way a direct comparison between concentration without and with recirculation is possible. By using the described equations it is then possible to design an air recirculation

system to result in the demanded concentration in a workroom.

8.8 Heating of industrial premises

8.8.1 General

Industrial buildings can generally be divided into three classes with regard to their heating:

1. Buildings that are thermally insulated and airtight. These buildings can be heated with a heating system.
2. Buildings without thermal insulation and/or measures against wind flowing through the outer walls. It is too expensive to heat these buildings totally; local heating or radiant heating at the working places represents a possible solution. An example of buildings of this kind is a warehouse.
3. Buildings where the waste heat from processes is so high that no further heating is needed. This is the case for instance in foundries.

8.8.2 The heating power demand

In the stationary state the heat demand φ for an industrial building is

$$\begin{aligned}\varphi &= G_b(T_i - T_o) + c_{pa}\rho_a q_{Va}(T_i - T_o)\\ &= (G_b + c_{pa}\rho_a q_{Va})(T_i - T_o).\end{aligned} \tag{8.27}$$

where G_b is the conductance of the building describing the totality of conduction through outer walls, roof, and floor. It can be considered constant for each specific building. T_i is the temperature inside the building and T_o the temperature outside the building. c_{pa} is the specific heat of air, ρ_a the density of air, and q_{Va} the volume flow of outside air into the building.

From Eq. (8.27) we also get:

$$\frac{\varphi}{T_i - T_o} = G_b + c_{pa}\rho_a q_{Va} = m. \tag{8.28}$$

If the volume flow q_{Va} is practically the same at every outdoor temperature, the quantity in Eq. (8.28) will also be a constant and specific for each individual building. If the quantity in Eq. (8.28) is constant, we also get from Eq. (8.27), if the inner temperature T_i is held constant:

$$\frac{d\varphi}{dT_o} = -G_b - c_{pa}\rho_a q_{Va} = -m. \tag{8.29}$$

Plotting the heat demand ϕ against T_o should produce a straight line with a negative slope coefficient.

If we look at a measured heat demand (Fig. 8.49) for an industrial building, we see that this is essentially the case. Deviations from the straight line are due to the following three factors:

1. The stationary state is not in reality reached.
2. The inside temperature T_i is not constant.
3. Or the volume flow q_{Va} is not constant.

8.8.3 The heating energy demand

The demand of heating energy for an industrial building can be calculated from Eq. (8.27) as:

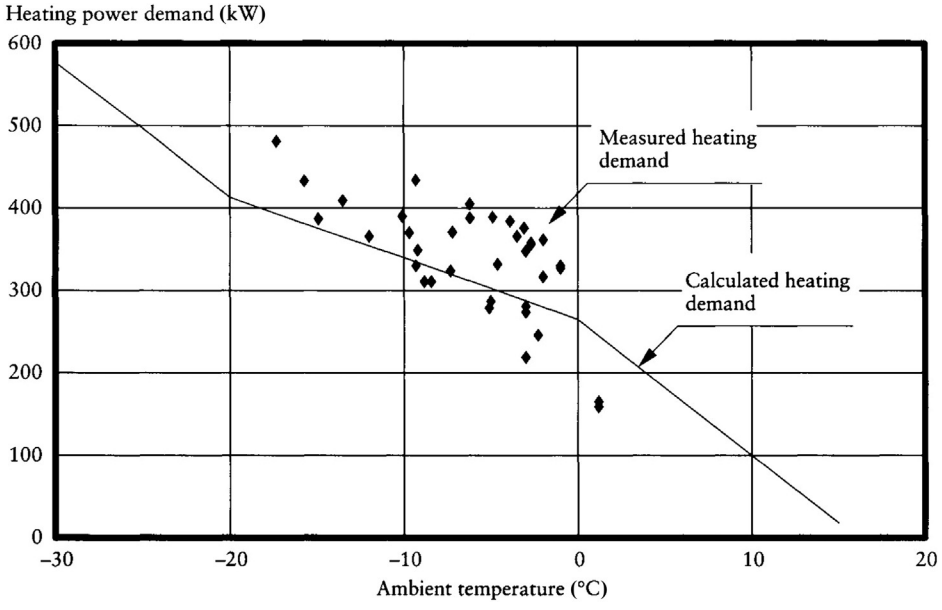


FIGURE 8.49 Heating demand versus ambient temperature for an industrial building. Calculated and measured values compared.

$$Q = \int_{t_1}^{t_2} \varphi dt = \int_{t_1}^{t_2} m(T_i - T_0(t))dt. \quad (8.30)$$

where t is the time and $t_1 - t_2$ the time period under consideration (month, year, etc.). The outdoor temperature T_o now depends on time and the indoor temperature is assumed constant.

The quantity m can be considered constant on condition that the flow of outside air q_{Va} is constant within the considered period. Quite often the ventilation is reduced at night and m is then not constant.

If m can be considered constant or if we use an average value of m over the time range $t_1 - t_2$, then we get from Eq. (8.30)

$$Q = m \int_{t_1}^{t_2} (T_i - T_0(t))dt. \quad (8.31)$$

The quantity $\int_{t_1}^{t_2}(T_i - T_o(t))dt$ is named the "degree day" and is normally calculated for each month but also on a yearly basis. It depends on the climate where the industrial building is situated. This means that different geographic positions have different degree days. From the definition we also see that the degree day depends on the assumed indoor air temperature, which is assumed constant.

Example

How much energy in January is needed for an industrial building if the heat demand is 50 kW at outside temperature −26°C and indoor temperature 18°C? The average outdoor temperature in January at that place is −4.7°C.

Solution From Eq. (8.28) we get:

$$m = \frac{50\ \text{kW}}{18°\text{C} - (-26°\text{C})} = 1140\ W/°\text{C}.$$

The average temperature difference between the indoor air and ambient air is:

$$\Delta\theta_{\text{mean}} = 18°\text{C} - (-4.7°\text{C}) = 22.7°\text{C}.$$

That means that the average heating power demand becomes:

$$\varphi_{\text{mean}} = \text{m}\ \Delta\theta_{\text{mean}} = 1140\ W/°\text{C} \cdot 22.7°\text{C} = 25.878\ \text{W}.$$

If the room is heated the whole day and night, that is, $\dot{m}t = 31$ days = 744 hours, the heating energy demand becomes:

$$Q = \varphi_{\text{mean}}\ \dot{m}t = 25.9\ \text{kW} \cdot 744\ \text{hours} = 19.270\ \text{kWh}.$$

8.8.4 Radiant heating

8.8.4.1 Radiant temperature

The idea of radiant heating in an industrial hall is to install local heating panels with a high surface temperature. Because of the high temperature, the proportion of thermal radiation in the heating power from the panel will increase drastically. But we must remember that the heating panels will also radiate directly on outer surfaces of the buildings, thus raising their surface temperature and increasing the heat flow through the walls.

The heating panels can be heated with electricity, burning gas flames, or hot water.

The temperature that a thermometer with an almost black bulb indicates when it is placed near the heating panel (Fig. 8.50) is practically the same temperature that a human being will feel in the neighborhood of the panel.

The heat balance for the bulb in Fig. 8.50 is

$$F\sigma\varepsilon(T_s^4 - T_b^4) = h_c(T_b - T_a) + (1 - F)\sigma\varepsilon(T_b^4 - T_\omega^4). \quad (8.32)$$

where T_s is the surface temperature of the panel, T_b the thermometer bulb temperature, T_a the air temperature, and T_w the temperature of the walls of the building. F is the view factor from the bulb to the heating panel, ϵ is the emissivity of the thermometer bulb at temperature T_b, σ is the Stefan−Boltzmann constant ($5.67 \times 10^{-8}\ \text{W/m}^2\ \text{K}^4$), and h_c is the convective heat transfer coefficient from bulb to air.

The unknown temperature T_b can be calculated from Eq. (8.32) when the other temperatures T_s, T_a, and T_w are known and also F, h_c, and ε. The temperature of the wall and air is practically the same, so we can set $T_a = T_w$. We also use the heat transfer coefficient $h_c = 4\ \text{W/m}^2\ \text{K}$.

The emissivity e of the bulb is quite insensitive to the material of the bulb, as long as it is not a polished bright metal. Because the bulb radiates longwave heat radiation, we can use the value $\epsilon = 0.95$.

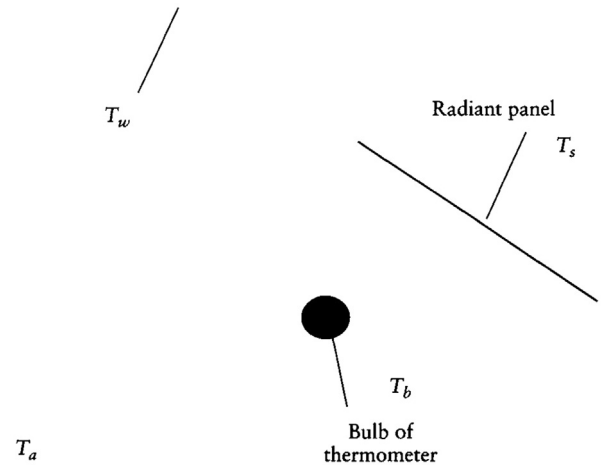


FIGURE 8.50 Heat balance for a black bulb in front of a radiant panel.

The view factor F can be calculated with the aid of Fig. 8.51. With the notations in Fig. 8.51, the view factor for a position for the bulb is:

$$F = \frac{1}{4\pi}\arcsin\frac{BC}{\sqrt{1+B^2+C^2+B^2C^2}}. \qquad (8.33)$$

If the thermometer is situated symmetrically relative to the heating panel, then the view factor will be four times higher than calculated from Eq. (8.33).

Example

Fig. 8.52 gives measured values of the "radiant temperature," which means $T_b - T_a$, caused by an electrical heating panel. The "effective surface" temperature of the panel can be estimated from the curves then used to calculate the temperature of a thermometer bulb at a few other places. These results can be compared to measured results.

Solution The first step is choosing a reference point. At the point 1 m right below the panel, the radiant temperature is 29°C, and if we assume that the temperature of air is 20°C, then $T_b = 49$°C.

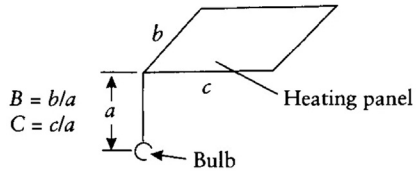


FIGURE 8.51 View factor from thermometer bulb to heating panel.

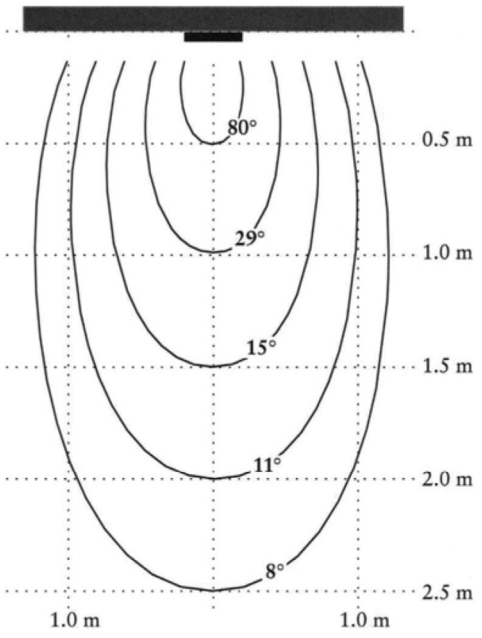


FIGURE 8.52 Measured values of radiant temperature near an electrical heating panel. Length of panel: 1.53 m; width: 0.28 m.

The view factor at that point is:

$$F = 4\frac{1}{4\pi}\arcsin\frac{0.14\times 0.765}{\sqrt{1+0.14^2+0.765^2+0.14^2\times 0.765^2}} = 0.02685.$$

Inserting this value into Eq. (8.32) and also $T_a = T_w = 20$°C = 293 K, we get (by iteration) the temperature $T_s = 740$ K = 467°C. We can use this value for calculating the radiant temperature at other locations. For each location a new view factor has to be calculated. The results are shown in Fig. 8.53.

We see that the agreement between measured and calculated temperatures is fairly good. Only in the right corner near the heating panel is there a big difference between the measured and calculated temperature. However, the measured results cannot be reliable here either, because it is not possible that the radiant temperature is 80°C just near a surface of 467°C.

8.8.4.2 *Radiant heating panels heated by water*

Radiant heating can be realized also by heating the panels by water. The principal construction of such a device is shown in Fig. 8.54.

The temperature of the water in the heating panel is limited by the boiling of the water. The boiling temperature of water increases with increasing pressure. If the pressure of the water is about normal atmospheric pressure, then the temperature of the water can be 90°C–95°C. If we want to raise the temperature of water to 120°C, the absolute pressure of water must be above 2 bar.

The heat balance of a water-heated panel is (Fig. 8.55):

$$\varphi = \varepsilon\sigma A(T_s^4 - T_{sur}^4) + h\,A(T_s - T_a) + \varphi_l, \qquad (8.34)$$

where φ is the total heat delivered to the panel by the flowing hot water, which is cooled, and φ_l is the heat loss through the insulation upward. T_s is the surface temperature of the panel, and we can put:

$$T_s = \frac{1}{2}(T_{wi} + T_{wo}), \qquad (8.35)$$

where T_{wi} is the inlet temperature of the water to the panel and T_{wo} is the outlet temperature. T_{sur} is the temperature of the walls and roofs of the industrial hall; T_a is the temperature of air in the hall, and we can put $T_{sur} = T_a$. A is the area of the heating panel and h is the convective heat transfer coefficient. ϵ is the emissivity of the surface and, in the considered temperature range, we can put $\epsilon = 0.95$. ϕ_l can be estimated if we know the type and thickness of the thermal insulation of the panel. One estimation is that φ_l is 9%–10% of the total heat to the panel.

The convective heat transfer for the panel is free convection from a heated surface faced down. It can be calculated from Incropera and De Witt[38]:

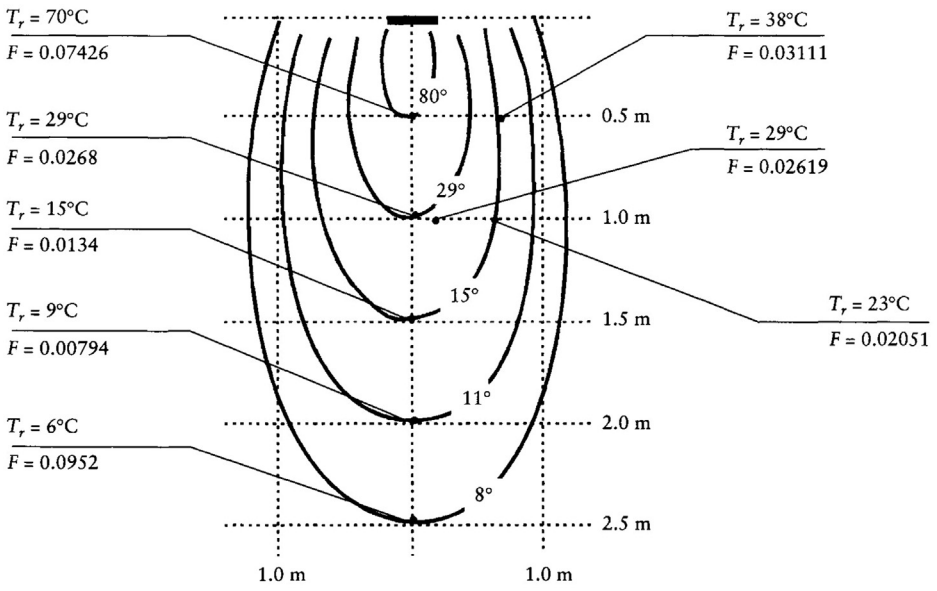


FIGURE 8.53 Measured and calculated radiant temperature at different locations.

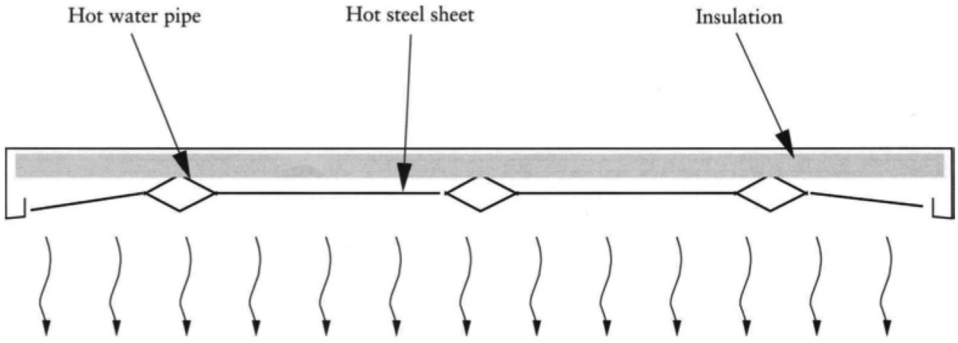


FIGURE 8.54 Radiant heating device using heated panels in water.

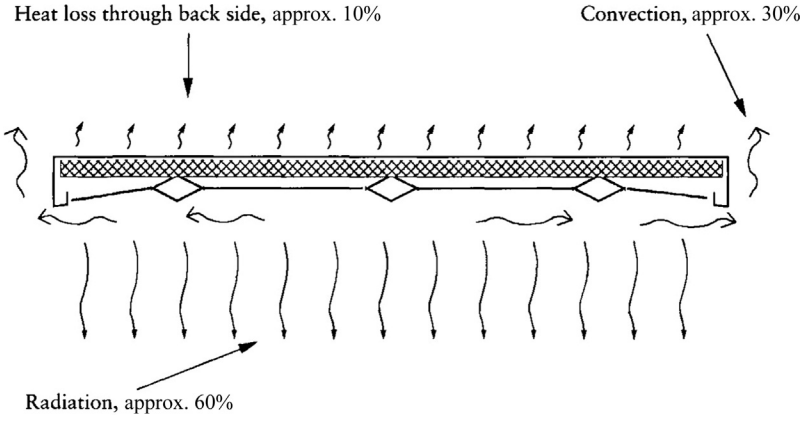


FIGURE 8.55 Heat balance of a heated panel.

$$h = 0.27\frac{k}{L_{ef}}\left[\frac{g(T_s - T_a)}{1/2(T_s + T_a)v\alpha}\right]^{1/4}, \qquad (8.36)$$

where g is the gravitational acceleration, 9.81 m/s^2; k is the thermal conductivity of air; v is the kinematic viscosity of air; and α is the thermal diffusivity of air.

Numerical values for these quantities for a few temperatures are as follows:

Temperature (K)	k (W/m K)	v (m^2/s)	α (m^2/s)
293	0.0257	15.3×10^{-6}	21.6×10^{-6}
313	0.0273	17.2×10^{-6}	24.4×10^{-6}
333	0.0287	19.2×10^{-6}	27.4×10^{-6}
353	0.0302	21.2×10^{-6}	30.4×10^{-6}

When calculating h from Eq. (8.36), the values for k, v, and α should be taken at temperature $1/2(T_s + T_a)$.

The length L in Eq. (8.36) is defined as:

$$L_{ef} = \frac{A}{P}, \tag{8.37}$$

where P is the perimeter of the surface area of the heating panel. In case the length of the panel is much bigger than the width, then Eq. (8.37) becomes

$$L_{ef} = \frac{W}{2}. \tag{8.38}$$

Eq. (8.34) can be written as heat per unit length of the heating panel:

$$\frac{\varphi}{L} = \varepsilon\sigma W T_s^4 - T_{sur}^4 + h\ W T_s - T_s + \varphi_l/L. \tag{8.39}$$

Example

The heat effect per unit length by radiation and convection from a water-heated panel can be calculated theoretically. For example, consider panels of width 0.6, 0.9, and 1.2 m. The room temperature is 3°C and the surface temperatures are 30°C, 50°C, and 70°C of the panel. Let us compare the results with calculations for the room temperature 15°C and the surface temperature 40°C, 60°C, and 80°C of the panel.

Solution One of the most critical and important quantities to calculate in Eq. (8.34) is the convective heat transfer coefficient. It depends on the temperature conditions and also on the width of the panel. Tables 8.11 and 8.12 collect the calculated heat transfer coefficients in different conditions.

We see from the results that in these situations there is no temperature dependence of the heat transfer coefficient. However, the heat transfer coefficient is lower the larger the width of the panel is. It is quite natural that a narrow panel has a higher heat transfer coefficient because it is easier for the air to rise upward when the panel is narrow. Using the heat transfer coefficients mentioned above, the heat effect delivered by the heating panels downward is

$$\frac{\varphi}{L} = \varepsilon\sigma W\left(T_s^4 - T_{sur}^4\right) + h\ \ W(T_s - T_a).$$

Tables 8.13 and 8.14 collect the results and compare them to results given by a manufacturer.

We see that there is a big difference between the calculations and the results given by the manufacturer. The manufacturer's results are about 45%–50% higher. The reason for the differences is not clear. One difficulty is that we do not know in the manufacturer's case how much heat is flowing through the insulation upward. We have assumed that this flow is 9% of the total heat effect of the heating panel.

TABLE 8.11 Heat transfer coefficients for $\Delta T_a = 5$°C.

Surface temperature T_s (°C) for $T_a = 5$°C	Heat transfer coefficient, h (W/m^2 K)		
	$W = 0.6$ m	$W = 0.9$ m	$W = 1.2$ m
30	2.1	1.9	1.8
50	2.4	2.2	2.0
70	2.6	2.4	2.2

TABLE 8.12 Heat transfer coefficients for $\Delta T_a = 15$°C.

Surface temperature T_s (°C) for $T_a = 15$°C	Heat transfer coefficient, h (W/m^2 K)		
	$W = 0.6$ m	$W = 0.9$ m	$W = 1.2$ m
40	2.1	1.9	1.8
60	2.4	2.2	2.0
80	2.6	2.4	2.2

8.8.5 Hot air blowers

A common heating method in industry is using blowers that blow heated air into the room. See Fig. 8.56. An advantage is that the installation costs are low. A possible disadvantage is that it mixes the air in the room, which may not be desirable in combination with displacement ventilation.

When locating hot air blowers, we need to know the penetration depth for the hot air that is being discharged from the blowers. For a given discharge angle the penetration depth, z, can be expressed as:

$$z = k_z l, \tag{8.40}$$

where z is the penetration depth for the centerline of the hot air jet (m); k_z is the penetration factor; and l is the characteristic length for the discharge opening, normally the square root of the discharge area (m).

The Archimedes number is defined as:

$$Ar = \frac{\Delta T g l}{T u_0^2}, \tag{8.41}$$

where g is the acceleration of gravity (9.81 m/s^2); ΔT is the temperature difference between the discharged warm air and the room air (K); T is the absolute temperature in the discharged air (K); and u_0 is the discharge velocity of the warm air (m/s).

To find the penetration depth:

find the Archimedes number from Eq. (8.41)
read the constant k_z from Fig. 8.57
calculate the penetration depth for the centerline of the jet from Eq. (8.40)

Example

A room is heated by hot air blowers. The indoor temperature is +10°C. Each blower has the following data:

TABLE 8.13 Heat transfer from panels: calculated values and manufacturer's data.

Surface temperature T_s (°C) for $T_a = 5$°C	Heat effect per unit length (W/m)					
	$W = 0.6$ m		$W = 0.9$ m		$W = 1.2$ m	
	Calculated	Manufactured	Calculated	Manufactured	Calculated	Manufactured
30	110	150	160	230	212	310
50	220	320	330	480	430	650
70	360	520	520	770	680	1040

TABLE 8.14 Heat transfer from panels: calculated values and manufacturer's data.

Surface temperature T_s (°C) for $T_a = 15$°C	Heat effect per unit length (W/m)					
	$W = 0.6$ m		$W = 0.9$ m		$W = 1.2$ m	
	Calculated	Manufactured	Calculated	Manufactured	Calculated	Manufactured
40	120	150	170	230	230	310
60	240	320	350	480	460	650
80	380	520	560	770	730	1040

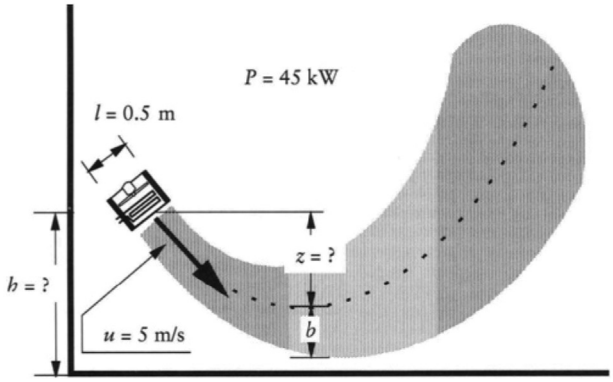


FIGURE 8.56 Hot air blower mounted at the wall.

Discharge opening dimensions: $w = h = l = 0.5$ m
Discharge velocity: $u_0 = 5$ m/s
Heater power: $Q = 5$ kW

The blowers are mounted at the walls, and tilted 45 degrees downward. At what height should the blower be mounted so that the air jet reaches the floor?

Solution First, we find the discharge airflow rate from the blower:

$$q_v = Au_0 = 0.5\text{ m} \times 0.5\text{ m} \times 5\ m/s = 1.25\ \text{m}^3/s.$$

The temperature increase in the blower is:

$$\Delta T = \frac{Q}{q_v \rho c_p}$$

$$= 45\text{ kW}/1.25\ \text{m}^3/\text{s} \times 1.15[\text{kg}/\text{m}^3 \times 1(\text{kW s/kg K})] = 30\text{ K}.$$

The Archimedes number for the jet at the discharge point, Eq. (8.41), is:

$$Ar_0 = 30\text{K} \times 9.81\ m/s^2 \times 0.5\left(m/\left(313\text{K} \times (5m/s)^2\right)\right)$$
$$= 0.0196.$$

From Fig. 8.57, we find:

$$k_z = \frac{z}{l} = 6$$

for a discharge angle of 45 degrees. Thus the penetration depth for the centerline of the jet becomes

$$h = z_{\max} = 6 \times 0.5\text{ m} = 3\text{ m}.$$

The radius of the jet at the lowest point is approximately 1.2 m (calculated from a jet widening angle of 25 degrees and a jet length of 4.2 m from the blower).

This means that the hot air blower should not be placed more than 4.2 m above the floor.

8.8.6 Air jets

An alternative to hot air blowers is air jets that mix the hot air into the occupational zones. See Fig. 8.58.

To avoid generating warmer air layers below the ceiling, air nozzles are utilized to mix the warmer air down into the occupied spaces. This system is well suited for applications where it is important to keep an even temperature throughout the room. It is not suited for rooms with heavy contaminant loads, where zoning is preferable.

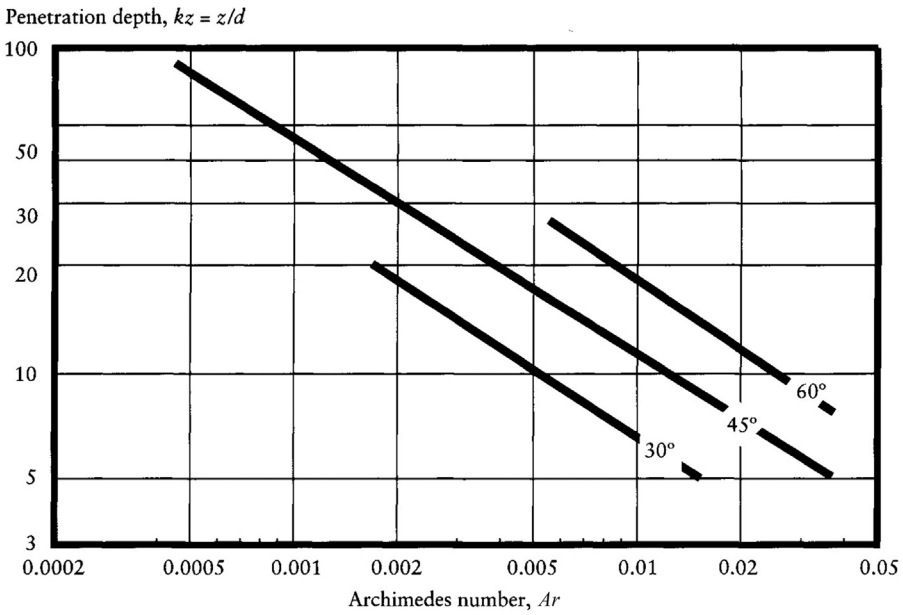


FIGURE 8.57 Penetration depth versus Archimedes number and discharge angle.

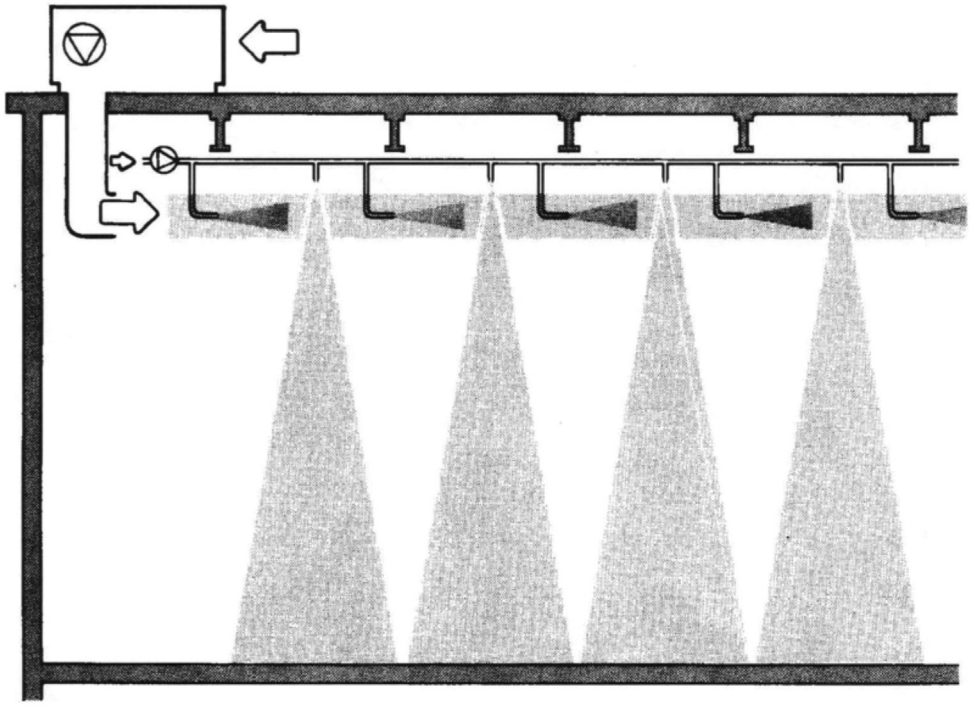


FIGURE 8.58 Mixing jet ventilation. Many nozzles blow the air horizontally and vertically.

8.8.7 Floor heating

8.8.7.1 General

Floor heating in industrial premises usually means hot-water pipes placed inside the concrete floor. (Electric coils or electric sheets are also used in nonindustrial premises; this is, however, not treated here.) Fig. 8.59 shows a typical installation of heat pipes inside the floor. Note that the pipes are placed relatively deep down inside the concrete to help even out the surface temperature.

Floor heating has several advantages:

- The heat is supplied at the floor, where it normally is most needed.
- Putting heating coils or heating pipes into a concrete floor makes a heat reservoir that helps even out temperature fluctuations.

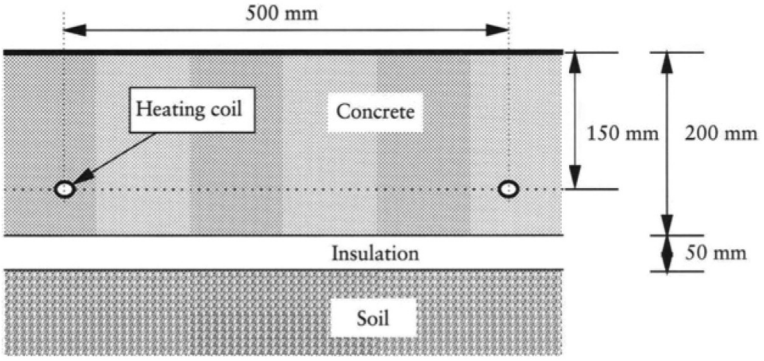


FIGURE 8.59 Arrangement and typical dimensions of hot-water pipes in a concrete floor.

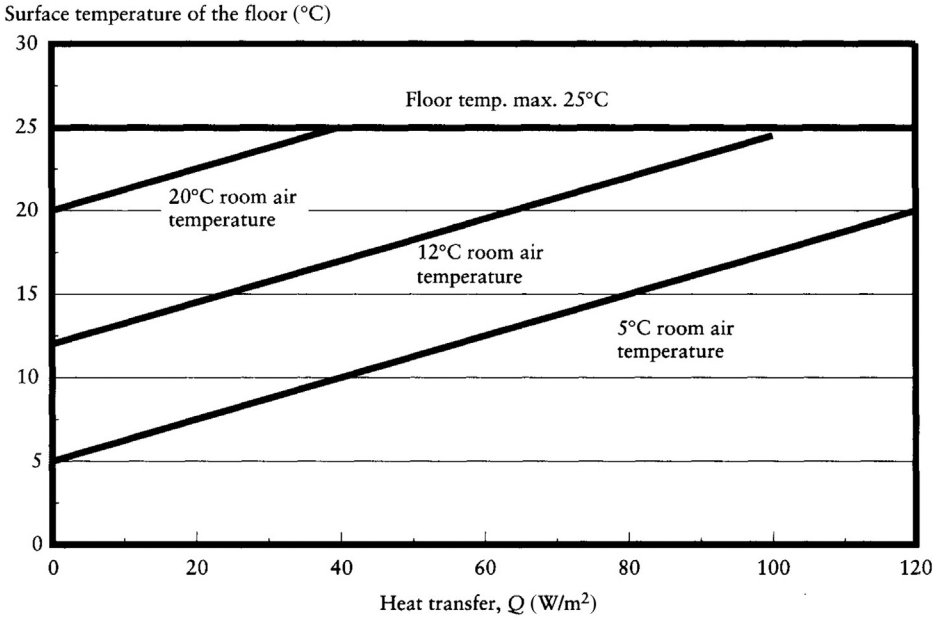


FIGURE 8.60 Heat transfer from a heated floor versus room and floor temperatures.

- When using hot-water pipes in the floor, the water temperature is usually low (30°C–40°C), so the system is well suited for low-temperature heating.
- The system is noiseless and draft-free.

8.8.7.2 *Surface temperature and heat emission*

One limitation in the use of floor heating is the surface temperature of the floor. Most people will find a floor surface temperature of more than 25°C uncomfortable.

Heat is transmitted from the floor to the room by radiation and convection. For practical purposes, we can put the heat transfer coefficient to 8 W/m^2 °C. Based on this assumption, we can make the diagram in Fig. 8.60.

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CHAPTER

9

Air-handling processes

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9.1 Introduction

9.1.1 Scope and purpose

The purpose of this chapter is to present the basic features of air-handling processes, relevant technologies and equipment. The aim is to provide links between the basic theories of air-handling processes, presented in Chapter 4, Physical Fundamentals, and the actual equipment, including air filters, heat exchangers, and fans.

This revised chapter deals with the basic air-handling processes: filtration of particles and gaseous substances from the supply and recirculated air, air heating and cooling, heat-recovery processes, humidification, and dehumidification. In addition, this chapter also describes fans, control systems and ductwork. Moreover, several basic issues have been covered behind energy-efficient design of air-handling systems and equipment. This chapter also deals with the essential factors in the selection of systems and equipment during the design stage: principles of controls, noise-reduction systems, and problems such as erosion, corrosion, maintenance, and equipment cleaning. Beyond this chapter, the following aspects should be taken into account in system design: fan safety; air handling unit (AHU) fire protection issues; safety measures in mines, tunnels, underground car parks, etc.; transportation of chemical and explosives.

Due to safety issues and good design practice. It is prudent when designing industrial ventilation systems that the requirements mainly mechanical, electrical, fire, explosion, health, etc. are carried out to the requirements of the following standards for which country the plant is required for. National standard bodies may be found here:

https://tbtcode.iso.org/sites/wto-tbt/list-of-standardizing-bodies.html

9.1.2 Aims of an air-handling system, including the unit and ductwork

The main objectives are:

- contaminant removal and control from the indoor space and associated processes;
- supply of safe cleaned and/or treated air for the occupied spaces; and
- control of thermal and pressure conditions in treated spaces in a safe, reliable, and energy-efficient manner.

The aims of ductwork design are:

- transport of air to and from treated (conditioned) spaces and
- distribution and control of airflows to and from the treated spaces with optimum life-cycle costs (LCCs).

The air-handling processes should be arranged to take into account the thermal, aerodynamic, and acoustic factors; air quality; moisture control; and cleanliness and other hygiene aspects.

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These issues relate certain requirements of the layout of the air-handling system and individual units, including the question of whether to select centralized or decentralized systems, and the number of units required in the building.

9.2 Air filters

The need to separate impurities from air or other gases has increased as regards both the degree of separation and the necessity to separate both gases and finer particles. Correct dimensioning of filters is a prerequisite for the functioning of ventilation systems, for sensitive production, for protecting humans and the environment, and for improving indoor air quality (IAQ).

9.2.1 Why air filters?

People spend up to 90% of their life indoors: home, work, restaurants, shopping malls, etc. Thus a good IAQ is very important and affects the health, comfort, and performance of the occupants. A remarkable portion of the airborne outdoor pollutants such as particles, pollen, mold spores, and ozone are transported into buildings through the ventilation system and the air infiltration. Heating, ventilation, and air conditioning (HVAC) system should provide a good supply air quality (SUP) and thus ensure healthy indoor environment. A supply air filter can significantly reduce the exposure to ambient air pollutants. It is also a key component to keep HVAC system clean, and thus preventing malfunctioning of air-handling unit.

9.2.1.1 Ventilation system protection

Several major studies of IAQ problems[1] have shown that filters would have been able to prevent blocked ducts, fans, and heat exchangers. Other major IAQ problems such as impurities from outside and microorganisms in the system need not arise given the correct choice of filter.

A precondition for maintaining function for a good number of years is that the system should be effectively protected, both on inlet and outlet exhaust systems. *Impurities must be stopped at the inlet and not be allowed to get into the system.*

9.2.1.2 Hygiene requirement

Every day we breathe about 20–30 kg of air and take in 1 kg of solid and 3 kg of liquid food. We should therefore make the same requirement of the air as we do of food and drink.

9.2.2 Atmospheric air and dust

Air is made up of a mixture of different gases and material from natural processes such as wind erosion, evaporation from the sea, earthquakes, and from human activity in the form of combustion products from processes and vehicles.

9.2.2.1 Size of particles

The size of particles is often indicated in μm—1 μm = 10^{-6} m. Particles in the atmosphere vary, from particles less than 0.01 μm up to leaves and insects. The precise definition of different sizes of the particles such as PM1, PM2.5, and PM10 is quite complex and not necessarily easy to measure. PM stands for particulate matter- solid and/or liquid particles suspended in ambient air. Definition of PM2.5 is particle size fraction less or equal to 2.5 μm. However, in the case of real-life measurements, the definition of PM2.5 is size fraction in the range of 0.3–2.5 μm. The reason for that is the challenges related to the reliable measurements of particles smaller than 0.3 μm.

Studies of atmospheric particles show that their distribution is often bimodal, that is, the particles are made up of two separate fractions, one with fine and one with coarse particles (Fig. 9.1). The coarse particles, from about 2.5 μm upward, are made up of natural dust from the effect of wind, erosion, plants, volcanoes, etc. The finer fraction is made up of particles smaller than 2.5 μm (called PM2.5) and consists primarily of particles from human activity, combustion, traffic, and processes.

Studies have shown a direct connection between the death rate and exposure to the particulate matter. Smaller the particulate matter is, the more deeper it can enter when inhaling. Generally, larger particles have a greater fractional deposition to the nose and to the larynx region, whereas smaller particles can deposit bronchiolar and to the alveolar region of the lungs. More serious health outcomes can occur when particles reach the alveolar region of the lungs where gas exchange of the human body occurs. For example, part of the inhaled PM2.5 can deposit to the alveolar region. According to the WHO,[2] health risks are related especially to the particulate matter of less than 2.5 μm in diameter.

To effectively eliminate these small particles and fulfill official requirements, finer filters are needed, and the requirement for filtering of outdoor air, recycled air, or exhausted air is increasing. With an ePM1 (50%–65%) and ePM2.5 (65%–70%) (former F7 fine filter), or ePM1 (65%–75%) and ePM2.5 (75%–85%) filter (former F8 fine filter), good separation of small particles is achieved.

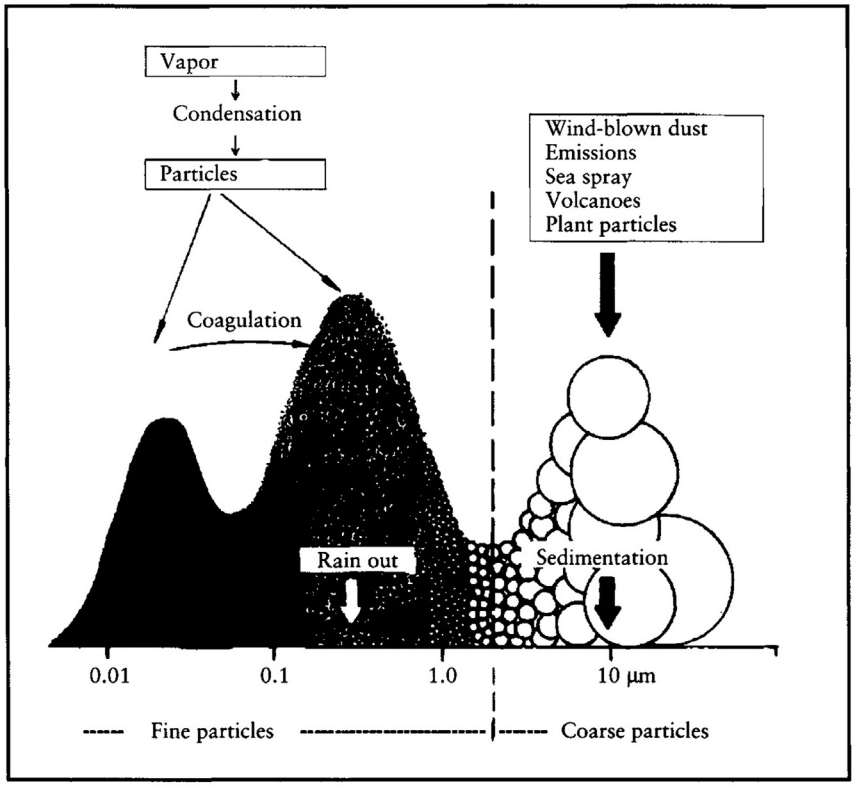


FIGURE 9.1 Size distribution of atmospheric dust.

TABLE 9.1 Number of particles (>0.01 μm) at different locations.

Place (examples)	Particles (m^3)
Clean room	10^3
Arctic	10^7
Countryside	10^9
City	10^{11}
Tobacco smoke	10^{14}

9.2.2.2 *Number of particles*

The number of particles varies considerably with time and place (Table 9.1). To reduce the number of particles in the urban environment to the same number as in the countryside, a filter with 99.9% separation is required. An urban environment or polluted environment thus requires an increasingly better filter quality.

9.2.2.3 *Other aspects*

Allergy

The problem of allergies has increased during the last few decades. The tendency to develop allergies and asthma is not well understood, but a number of pollutants can trigger the reaction, for instance, airborne allergens such as pollen, spores, living or dead bacteria, dust, diesel fume, and cigarette smoke. Tests have shown that an ePM1 (50%–65%) filter (former F7 fine filter) can effectively reduce allergens in the air.[3]

Carcinogenic potential of pollutants

It is known that the urban traffic environment is carcinogenic due to automotive gases and polyaromatic hydrocarbons. Therefore ePM1 (65%–75%); ePM2.5 (75%–85%) category filter will reduce their effect (former F8 filter).

Odors/gases

A large number of odors are borne by particles, but for effective separation, chemical filters are very often required, which can be justified in an urban environment. Activated carbon has been one of the main materials for the effective capture of odorous compounds and ozone as well. Activated carbon has an effective surface area up to 1400 m^2/g. The use of gas filter with complement of particle filtration is justified in the industrial and urban environments where odorous chemicals are present.

9.2.3 Filters and test methods

9.2.3.1 *Test methods*

Efficiency of the filters has been tested according to the several standards. In Europe, ISO 16980 series, adopted in Europe as EN ISO 16980 standards and EN

1822-1:2019,[4] defines the fractional efficiency of HVAC filters and efficient particulate air (EPA) filters, high-efficiency particulate air (HEPA) filters, and ultra low penetration air (ULPA) filters, respectively. In the United States the corresponding test method is ASHRAE 52.2.[5]

ISO 16890 (which is the same as EN ISO 16890) was introduced in 2016 and it has replaced the EN 779:2012.[6] New standard method is considered to give a more realistic value for the efficiency of the air filters in a real ventilation system compared to previous ones. One reason for that is the efficiency values of the filters have been measured in three ranges: PM1 (0.3–1 μm), PM2.5 (0.3–2.5 μm), and PM10 (0.3–10 μm). Test of the filters have been made by optical particle counters, and using atomized diethylhexylsebacate (DEHS) droplets in the range between 0.3 and 1 μm and solid potassium chlorine (KCl) particles in the size range of 1–10 μm. DEHS droplets were also used in previous EN 779:2012[6] standard, but the average efficiency of the tested filter was only determined to the size of 0.4 μm particles. The fine particulate (PM2.5) and coarse particulate matter (PM10) have also international outdoor air quality guidelines given by the European Union, EPA, and WHO.

Guideline values[7]: fine particulate matter (PM2.5):

- 10 $\mu g/m^3$ annual mean
- 25 $\mu g/m^3$ 24-hour mean

Coarse particulate matter (PM10):

- 20 $\mu g/m^3$ annual mean
- 50 $\mu g/m^3$ 24-hour mean

Thus filters are now connected more to the real-life air pollution and guidelines given there. Another important reformation of testing is to define the efficiency of air filters in two parts: before and after the discharging of the filter media. Previously, originally intentionally or unintentionally charged fibrous filters might have played a role in particle collection made to the new filters. Capture of particles to the filter media by electrostatic force can be initially high especially in case of synthetic fiber filters, but it weakens during the use of the filter in real-life air-handling units. However, in EN 779:2012,[6] minimum efficiency was also determined for 0.4 μm particles after discharging the filter media and it was 35%, 55%, and 70% for the F7, F8, and F9, respectively.

In ISO 16890 test, electrostatic charge of the filters has been removed from the filters by isopropanol vapor. It is nondestructive and fibers remain intact. The results of the test will be given an average of initial and discharged efficiency: $E_m = (E_i + E_d)/2$. The reported mean efficiency in the category given (PM1–PM10) needs to be at least 50%, and the discharged efficiency needs to be over 50% as well to pass the test in PM1 and PM2.5. Filters that have ePM10 less than 50% are considered coarse filters.

Table 9.2 has given a tentative comparison of EN 779 and ISO 16890 test methods and category of the filters. Filters with higher efficiencies will be tested according to EN 1822-1:2019 (EPA, HEPA, and ULPA).

The increased need to control the indoor environment and filter efficiency in the actual environment has led to EN ISO 29462:2013,[8] which describes a method of counting ambient air particles of 0.3 to 5.0 μm upstream and downstream of the air filters in functioning air-handling system in order to estimate the behavior of air filters in real life.

9.2.3.2 *Classification of coarse and fine filters*

Depending on the test method and test result, particle filters are classified as ISO coarse, ePM1, ePM2.5, ePM10, EPA, HEPA, and ULPA filters. Filters can be classified as ePM1 or ePM2.5 only if the minimum measured efficiency is at least 50%. Chemical filters are used for gases.

TABLE 9.2 Filter classifications—comparison and classification of EN 779 and ISO 16890 standards.

Standard	**EN 779**	**ISO 16890**	
Size of test particles	**0.4 μm (DEHS)**	**PM1 (0.3–1 μm), PM2.5 (0.3–2.5 μm), PM10 (0.3–10 μm). For coarse: initial gravimetric arrestance**	
Filter classes	G4	ISO coarse	85%–90%
	M5	ISO coarse	95%
		ePM10	50%–70%
		ePM2.5	–
		ePM1	–
	M6	ePM10	60%–75%
		ePM2.5	50%–60%
		ePM1	–
	F7	ePM10	80%–85%
		ePM2.5	65%–70%
		ePM1	50%–65%
	F8	ePM10	>90%
		ePM2.5	75%–85%
		ePM1	65%–75%
	F9	ePM10	>95%
		ePM2.5	90%–95%
		ePM1	80%–90%

DEHS, Diethylhexylsebacate.

To save energy the filter is dimensioned in normal ventilation plants, often with a lower final pressure loss than indicated by the classification, and the filter does not achieve the intended filter classification, where final pressure drop for the coarse filters (ePM10 $<50\%$) is determined to be 200 Pa and for the fine ePM filters (ePM10 $\geq 50\%$) 300 Pa. However, final pressures given in standards are unusually high in real situations, and often for the hygienic reasons, the filters are replaced after a certain period of time, rather than a specific final pressure loss. Typically, filters are changed at least once a year, but in many cases, they should be changed more frequently, for example, after the dusty season in the autumn and after the winter heating season to eliminate stuffy odors originated from the dust loaded filters.

9.2.3.3 EPA, HEPA, and ULPA Filters

To meet today's high requirements within the military, nuclear power industry, hospitals, etc., but especially in the electronics industry, new test methods for HEPA and ULPA filters have been developed. In the EN 1822-1:2019[4] test method the filter's efficiency is determined for the "most penetrating particle size." Depending on the filter's total level of separation and leakage, the filter is classified as E10 to U17 (Table 9.3). MPPS stands for the most penetrating particle size. HEPA filters are commonly used for inlet air in the pharmaceutical, optical, and food industries.

9.2.3.4 Chemical filters

Chemical filters are used to collect gases; these are mainly adsorption filters based on activated carbon. By the addition of chemical substances (impregnation), gases that are difficult to adsorb are adsorbed and retained by means of a chemical reaction.

TABLE 9.3 Classification of US Environmental Protection Agency (EPA), HEPA, and ULPA filters (EN 1822-1:2019).

Filter	EN 1822 class	Efficiency MPPS (%)
EPA	E10	85
EPA	E11	95
EPA	E12	99.5
HEPA	H13	99.95
HEPA	H14	99.995
ULPA	U15	99.9995
ULPA	U16	99.99995
ULPA	U17	99.999995

9.2.4 Filters in operation

9.2.4.1 Outdoor air quality and desired supply air quality

Outdoor air quality and desired SUP determines the correct choice for the good IAQ. Eurovent 4/23—2017[9] gives information to choose correct air filters. Outdoor air quality (ODA) has been categorized in three classes (ODA 1–3) and target quality for the supply air in five classes (SUP 1–5) by EN 16798-3:2017.[10] Tables 9.4 and 9.5 have given outdoor air and supply air categories.

It is important to be aware of the filter's properties in different environments. Outdoor air quality and target supply air class determine the requirements for the average filtration efficiency.

Fig. 9.2 shows how, in the case of new filters, separation varies with particle size and filter class. The filter class is based on the average efficiency, and a new

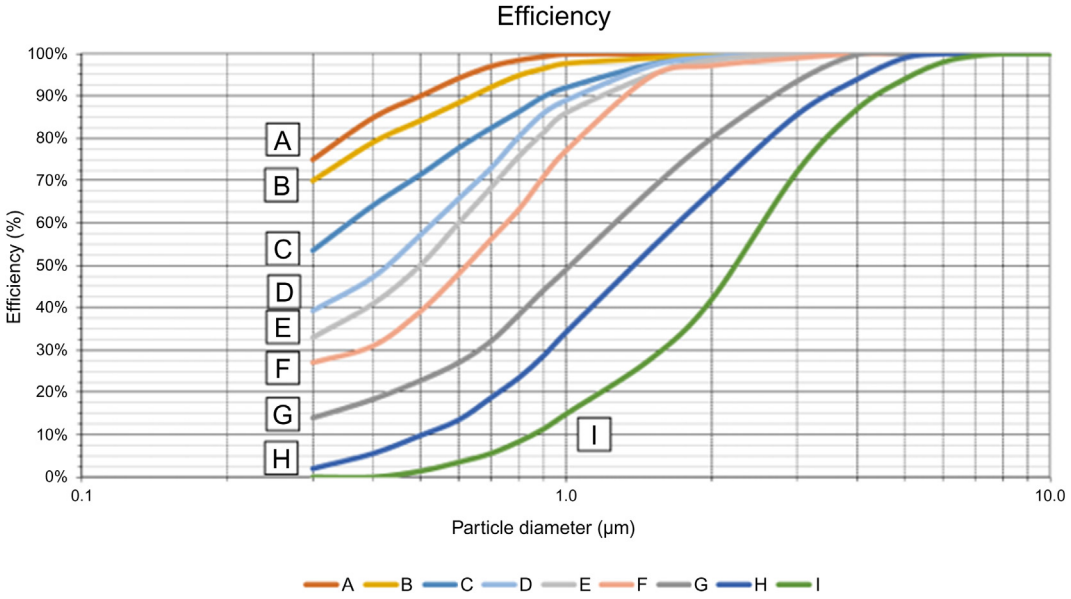
TABLE 9.4 Classification of outdoor air quality.

Category	Description	
ODA 1	Outdoor air, which may be only temporarily dusty	Applies where the WHO[7] guidelines are fulfilled
ODA 2	Outdoor air with high concentrations of particulate matter	Applies where PM concentrations exceed the WHO guidelines by a factor of up to 1.5
ODA 3	Outdoor air with very high concentrations of particulate matter	Applies where PM concentrations exceed the WHO guidelines by a factor of greater than 1.5

TABLE 9.5 Supply air categories.

	Industrial ventilation
SUP 1	Application with high hygienic demands (e.g., hospitals, pharmaceutics, electronic and optical industry, supply air to clean rooms)
	Recommendation $<0.25\times$ WHO[7] guidelines limit values (PM2.5 and PM10)
SUP 2	Recommendation $<0.5\times$ WHO
SUP 3	Recommendation $<0.75\times$ WHO
SUP 4	Recommendation $<1.0\times$ WHO
SUP 5	Production areas of heavy industry
	Recommendation $<1.5\times$ WHO

SUP, Supply air quality.



ISO 16890:2016 ePM efficiency and class

Filter	ePM_1	ISO ePM1	$ePM_{2,5}$	ISO ePM2,5	ePM_{10}	ISO ePM10
A	89 %	85	93 %	90	98 %	95
B	84 %	80	89 %	85	97 %	95
C	72 %	70	80 %	80	94 %	90
D	58 %	55	70 %	70	91 %	90
E	53 %	50	66 %	65	89 %	85
F	42 %	40	58 %	55	87 %	85
G	23 %	-	40 %	-	78 %	75
H	11 %	-	27 %	-	71 %	70
I	3 %	-	14 %	-	61 %	60

FIGURE 9.2 Efficiency of air filters versus particle size. The figures should be the minimum efficiencies in an installation. Original from REHVA GB 11 Air Filtration in HVAC Systems.

filter normally has much lower initial efficiency. In the case of electrostatically charged filters, separation may be significantly higher for new filters. However, in new ISO 16890 test, electrostatic charge has removed by isopropanol, and discharged filter efficiency needs to be over 50% as well (see 9.2.3.1). The figure should be seen as an indication of *minimum separation during actual operation*.

As the filter accumulates dust, the pressure loss increases, and the dust removed improves the normal separation. Another effect can be seen with electrostatically charged filter material. During operation, the impurities neutralize the material, and the filter's capacity to separate is reduced. Fig. 9.3[11] shows examples of filters shown by laboratory tests to be in accordance with ePM1 (50%–65%) (F7). The efficiency drops dramatically from more than 80% to less than 20% after a few weeks' operation in the case of the filter based on electrostatic charge.[12] The effect varies much with fiber size and charge. Table 9.6 shows the required combined average filtration efficiency (ISO 16890) needed to go from an ODA (P) level to a desired SUP level.

9.2.4.2 Average pressure loss

The average pressure loss during operation is dependent on the characteristics of the plant and is often taken to be the average value of initial pressure loss and final pressure loss of the filter. With the lower energy requirement, more and more systems are being dimensioned for constant flow, and average pressure loss is the integrated value. Significant savings can thus be made using filters with a low pressure loss and small increase in pressure during the period of operation.

9.2.4.3 Energy consumption

A filter's energy consumption, E, based on average pressure loss, can be calculated as

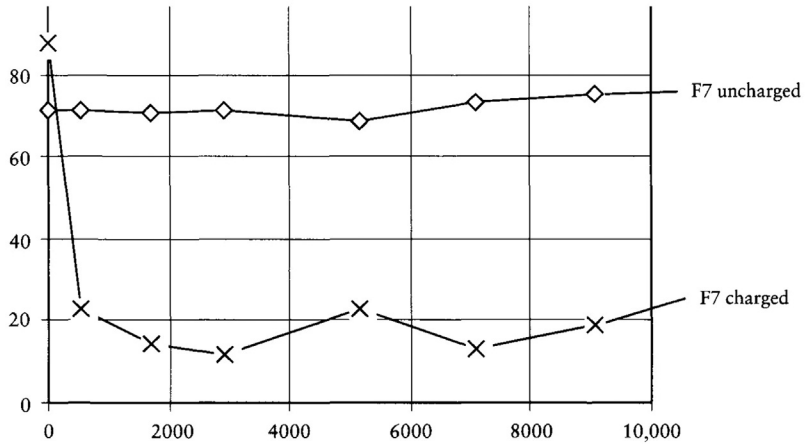


FIGURE 9.3 Example of efficiency changes in an installation with two F7 air filters.

TABLE 9.6 Required average filtration efficiency based on particle outdoor air quality.

	Supply air class (%)					
Outdoor air quality	**SUP 1 (P) ePM_1[a]**	**SUP 2 (P) ePM_1[a]**	**SUP 3 (P) $ePM_{2.5}$[b]**	**SUP 4 (P) ePM_{10}**	**SUP 5 (P) ePM_{10}**	**SUP 6 (P)**
ODA (P) 1	70[a]	50[a]	50[a]	50	50	0[c]
ODA (P) 2	80[a]	70[a]	70[a]	80	50	
ODA (P) 3	90[a]	80[a]	80[a]	90	80	

[a]*Final filter stage should be minimum $ePM_1$50%.*
[b]*Final filter stage should be minimum $ePM_{2,5}$50%.*
[c]*No filtration required.*
Note: The combined filtration efficiency shall be calculated according to ISO 16890. *SUP*, Supply air quality.

$$E = \frac{q\Delta pt}{\eta 1000}(\text{kWh}),$$

where q is airflow (m^3/s), Δp is average pressure loss (Pa), t is operation time (hours), and η is efficiency of fan.

Over 1 year (8760 hours), a 1 m^3/s filter with an average pressure loss of 100 Pa requires 1250 kWh if the fan's efficiency is set at 70%. The energy cost is generally greater than the filter cost, and pressure loss reduction becomes increasingly significant for energy reductions. Lower pressure loss by 10 Pa means 125 kWh less energy in the example above.

9.2.4.4 *Lifetime*

The lifetime of a filter is dependent on the concentration of dust, type of dust, airflow, and, of course, the selected final pressure loss. Filter material and filter construction are often a compromise or combination of filter effects and installation space. Low speed or large filter surface promotes efficiency, low pressure loss, but above all a longer lifetime.

9.2.4.5 *Filter replacement*

Airflow changes in the plant have been the main criterion for changing filters, that is, when pressure loss increases to the extent that the fan cannot maintain a specific minimum airflow. Reduction of maximum effect, energy consumption or economic evaluation, that is, when the energy cost and filter cost reach a minimum, are becoming increasingly significant.

Considerations of hygiene are being applied more and more to filter replacement. Studies[13] have shown that with RH (relative humidity) higher than 75% there is a risk of microbial growth in the filter and in the ventilation system. As it is in many cases difficult to avoid a high RH in the air intake, filtering should take place in two steps. The first filter can often be exposed to high humidity or to rain and snow. Organic impurities also become caught in the filters and could be released later. Particles and endotoxins from microorganisms can become loose in low-quality filters.

9.2.5 Life-cycle issues

9.2.5.1 *Environment: life-cycle analysis*

Global environmental questions have increased in significance during the last few years. A life-cycle analysis (LCA) analyzes the environmental effect with reference to ecological effects, health effects, and consumption of resources.

TABLE 9.7 Example of life-cycle cost (LCC) distribution of an air filter.

Type of cost	Relative cost (%)
Investment	4.5
LCC_{energy}	80.8
$LCC_{maintenance}$	14.2
$LCC_{environment/dumping}$	0.5
LCC_{total}	100

LCA of a filter shows that operation often corresponds to 70%–80% of the filter's total environmental load and is absolutely decisive as regards environmental effect.[14] Raw material, refining, manufacturing, and transports correspond to about 20%–30%, while the used filter contributes at most 1%. Filters of plastic or other inflammable material can render 10–30 kWh energy when burned, which correspondingly reduces the total environmental load from 0.5% to 1%. On the other hand, if the pressure loss in the filter is reduced by 10 Pa, the environmental load is reduced by 125 kWh per year, or approximately 5% decrease in total environmental load. Filters in industrial applications can have quite different figures.

9.2.5.2 *Life-cycle cost*

LCA does not account for economic aspects, and such analysis should therefore be considered together with an LCC analysis,[15] which takes into account the costs of investment, energy, maintenance, and dumping the final waste product throughout the lifetime of a plant.

Future costs of replacement filters and energy are calculated according to the current value method. The final result for a 1 m^3/s filter with average pressure loss of 200 Pa may be as shown in Table 9.7, if the calculation is based on a 10-year period.

The table shows that energy costs account for 80% of the total cost during the plant's period of operation. The actual costs of the filter, investment, and maintenance correspond to about 20%, while the costs of dumping amount to only 0.5%. The calculation is based on filtering outdoor air, and filters in industrial applications can have quite different figures.

LCC analyses provide an excellent tool for minimizing the filter costs of a plant. As in the case of LCA the operation and low pressure loss are absolutely decisive as regards the costs of the filter function.

9.2.6 Summary

The following points should be borne in mind when planning filter installations:

- Great care is required regarding the positioning and design of the air intake to avoid drawing in local impurities and rain or snow.
- Supply air filters are classified according to the ISO 16890:2016 standard and their mean minimum efficiency in PM1, PM2.5, and PM10 category needs to be at least 50%. Filters that have ePM10 less than 50% are considered coarse filters.
- Filters have been selected according to the outdoor air quality (ODA 1–3), and target SUP (SUP 1–5).
- As regards recirculated air, at least ePM10 (50%) M5 quality must be used to cope with contamination of components in the system, but the minimum requirement is for ePM1 (50%–65%) (F7) if the environment in the room is to be improved.
- The exhaust air (EHA) system must be protected from contamination by a filter of at least ePM10 (50%) M5 quality.
- Filters must not be installed directly after the fan outlet or across places where there is a big change in area or flow.
- The final pressure drop is dimensioned and selected with regard to permitted variations in flow, the filter's LCCs, and LCA.
- The risk of microbial growth is low, but to minimize the risk, the plant should be designed so that RH is always below 80% in all parts of the system.
- Dust-holding capacity and test results from laboratory trials differ from performances in actual use.
- The efficiency must not deteriorate or fall below specific minimum values.
- The tightness and condition of the filter are checked regularly by visual inspection of the plant.
- Filter and aggregate must be clearly marked with the type and designation of the filter, date of installation, etc.
- Filters must be replaced when the pressure loss reaches the dimensioned final pressure loss or when the following hygiene interval is reached, if this is earlier.
- Typically, filters are changed at least once a year; however, for hygienic reasons, they should be changed more frequently. It is recommended changing the air filters after the dusty season (spring dust, pollen, spores), in the autumn and after the winter heating season to eliminate odors from combustible products.
- Filters should be replaced carefully by using plastic bag, suitable protection clothing, and a face mask to

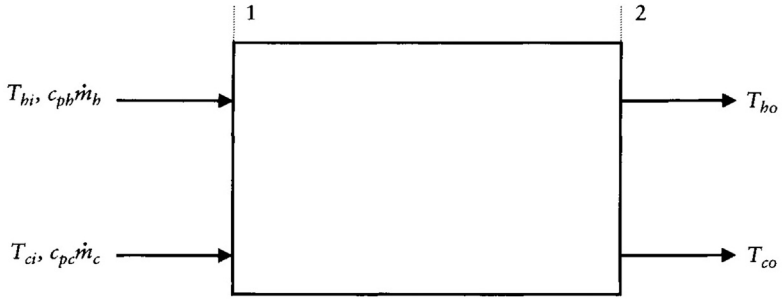


FIGURE 9.4 Heat exchanger with flow notations.

avoid exposure to unhealthy particulate matter released from the filter media.

9.3 Heat exchangers and heat-recovery units

9.3.1 General theory of heat exchangers

9.3.1.1 Introduction

A heat exchanger is a device that transfers heat from one medium to another; the medium may be a solid, liquid, or gas. Some of the most complex engineering design problems relate to heat exchangers.

Heat exchangers are divided into the following types:

1. Recuperator: A heat transfer surface (wall) separates two fluids in a recuperative heat exchanger. The two fluids ideally have no mix and the exchanger core has no moving part. Some examples of the recuperative exchanger are plate and tubular heat exchangers.
2. Regenerator: The hot and cold fluids pass alternately through a space containing solid areas/particles, which provide alternately a heat sink and a heat source. Some examples of the regenerative exchanger are rotary and fixed matrix heat exchangers.

The direction of flow is important, as it has a pronounced effect on the efficiency of a heat exchanger. The flows may be in the same direction (parallel flow, cocurrent), in the opposite direction (counterflow), or at right angles to each other (cross-flow). The flow may be either single-pass or multipass; the latter method reduces the size of the exchanger.

A schematic diagram of a parallel-flow (cocurrent) heat exchanger is shown in Fig. 9.4.

$$\text{Hot fluid: } T_{hi} > T_{ho}$$
$$\text{Cold fluid: } T_{ci} < T_{co}$$

The hot and cold fluids are denoted by the subscripts h and c and the fluid inlet and outlet by i and o, respectively.

In the case of Fig. 9.4, the *relative temperature differences* are defined by

$$\delta T_h = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}}, \tag{9.1}$$

$$\delta T_c = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}}. \tag{9.2}$$

The relative temperature differences can be calculated by dividing the temperature difference of the hot or cold side by the maximum temperature difference, $T_{hi} - T_{ci}$, that occurs in the heat exchanger.

There are two relative temperature changes for a heat exchanger. The greatest of them is the *heat exchanger effectiveness*, ϵ:

$$\epsilon = \max\{\delta T_c, \delta T_h\} \tag{9.3}$$

The maximum temperature difference that takes place in a heat exchanger is $T_{hi} - T_{ci}$. A higher temperature difference cannot occur due to the second law of thermodynamics. The maximum theoretical heat transfer rate in a heat exchanger is

$$\phi_{\max} = \dot{C}_2(T_{hi} - T_{ci}) \quad \text{or} \quad \phi_{\max} = \dot{C}_1(T_{hi} - T_{ci}),$$

where

$$\dot{C} = q_m c_p \tag{9.4}$$

is the *heat capacity rate*, q_m is the mass flow through the heat exchanger, and c_p is the specific heat capacity.

Assuming that the maximum possible temperature difference is on the fluid side with the higher heat capacity rate, then

$$\phi_{\max} = \dot{C}_2(T_{hi} - T_{ci}).$$

The heat exchanger balance is

$$\dot{C}_2(T_{hi} - T_{ci}) = \dot{C}_1 \, \Delta T_1,$$

where ΔT_1 is the temperature difference of the smallest heat capacity rate fluid in the heat exchanger. Then

$$\Delta T_1 = \frac{\dot{C}_2}{\dot{C}_1}(T_{hi} - T_{ci}) > T_{hi} - T_{ci}.$$

This equation, which gives a higher temperature difference than $T_{hi} - T_{ci}$, cannot occur. Then

$$\phi_{\max} = \dot{C}_1(T_{hi} - T_{ci}),$$

where $\dot{C}_1$ is the smaller heat capacity rate. The actual power is

$$\phi = \dot{C}_1\Delta T_1 = \dot{C}_2\Delta T_2,$$

from which

$$\frac{\phi}{\phi_{\max}} = \frac{\dot{C}_1\Delta T_1}{\dot{C}_1(T_{hi} - T_{ci})} = \frac{\Delta T_1}{T_{hi} - T_{ci}} = \frac{\dot{C}_2\Delta T_2}{\dot{C}_1(T_{hi} - T_{ci})} > \frac{\Delta T_2}{T_{hi} - T_{ci}}.$$

From this it follows that

$$\epsilon = \frac{\phi}{\phi_{\max}} = \frac{\text{Temperature difference of smaller heat capacity fluid}}{\text{Initial temperature difference}}. \tag{9.5}$$

The heat exchanger effectiveness shows how close the heat exchanger is operating to the maximum heat transfer performance. Eq. (9.5) is valid for any type of heat exchanger.

9.3.1.2 *Effectiveness–number of transfer units method and counterflow heat exchanger*

Two methods are normally used for sizing and rating heat exchangers: effectiveness–number of transfer units (NTU) method and logarithmic mean temperature difference (LMTD) method. The former one will be introduced for a counterflow heat exchanger in this section. The latter method will be presented in the next section. These two methods yield identical results for the heat exchanger sizing and rating problems within the specified convergence accuracy.

In a counterflow or countercurrent heat exchanger, hot and cold fluids enter the exchanger from opposite sides. The counterflow heat exchanger in Fig. 9.5 serves as a reference for all other heat exchanger configurations. A detailed analysis of this type of heat exchanger is therefore necessary.

The heat is transferred by convection and conduction from the hot to the colder fluid through an infinitesimal surface area dA. The temperature of the hot fluid reduces by an amount dT_h, and the temperature of the cold fluid increases by an amount dT_c.

The heat balance gives

$$G''dA(T_hT_c) = \dot{C}_h dT_h = \dot{C}_c dT_c, \tag{9.6}$$

where G'' is the conductance per unit of surface area of the separating wall, which for a plane wall can be written as

$$\frac{1}{G''} = \frac{1}{\alpha_c} + \frac{1}{\alpha_h} + \frac{\delta}{\lambda}, \tag{9.7}$$

where α_c and α_h are the cold- and hot-side convective heat transfer coefficients, respectively, δ is the wall thickness, and λ is the wall thermal conductivity.

For a thick circular tube the conductance per unit tube length is

$$\frac{1}{G'} = \frac{1}{\pi d_o\alpha_o} + \frac{1}{\pi d_i\alpha_i} + \frac{\ln(d_o/d_i)}{2\pi\lambda}, \tag{9.8}$$

where α_o is the outer convective heat transfer coefficient, α_i is the inner convective heat transfer coefficient, d_i is the inner tube diameter, and d_o is the outer tube diameter.

Eq. (9.6) gives

$$dT_h - dT_c = d(T_h - T_c) = \left(\frac{1}{\dot{C}_h} - \frac{1}{\dot{C}_c}\right)G''dA(T_h - T_c). \tag{9.9}$$

Eq. (9.9), after integrating

$$\int_1^2 \frac{d(T_h - T_c)}{T_h - T_c} = \left(\frac{1}{\dot{C}_h} - \frac{1}{\dot{C}_c}\right)G''\int_0^A dA, \tag{9.10}$$

gives

$$\ln\frac{\Delta T_2}{\Delta T_1} = \left(\frac{1}{\dot{C}_h} - \frac{1}{\dot{C}_c}\right)G''A = G\left(\frac{1}{\dot{C}_h} - \frac{1}{\dot{C}_c}\right) \tag{9.11}$$

or

$$\frac{\Delta T_2}{\Delta T_1} = \exp G''\left(\frac{1}{\dot{C}_h} - \frac{1}{\dot{C}_c}\right). \tag{9.12}$$

Using Fig. 9.5 and Eq. (9.12) gives

$$\frac{T_{hi} - T_{ci} + T_{ci} - T_{co}}{T_{hi} - T_{ci} + T_{ho} - T_{hi}} = \exp\left(G\left(\frac{1}{\dot{C}_h} - \frac{1}{\dot{C}_c}\right)\right). \tag{9.13}$$

For the case of $\dot{C}_h < \dot{C}_c$, the heat exchanger effectiveness is

$$\epsilon = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}}, \tag{9.14}$$

and the heat balance is

$$\dot{C}_h(T_{hi} - T_{ho}) = \dot{C}_c(T_{co} - T_{ci}). \tag{9.15}$$

This then is written as

$$T_{co} - T_{ci} = \frac{\dot{C}_h}{\dot{C}_c}(T_{hi} - T_{ho}) = R(T_{hi} - T_{ho}), \tag{9.16}$$

where $R = \dot{C}_h/\dot{C}_c < 1$.

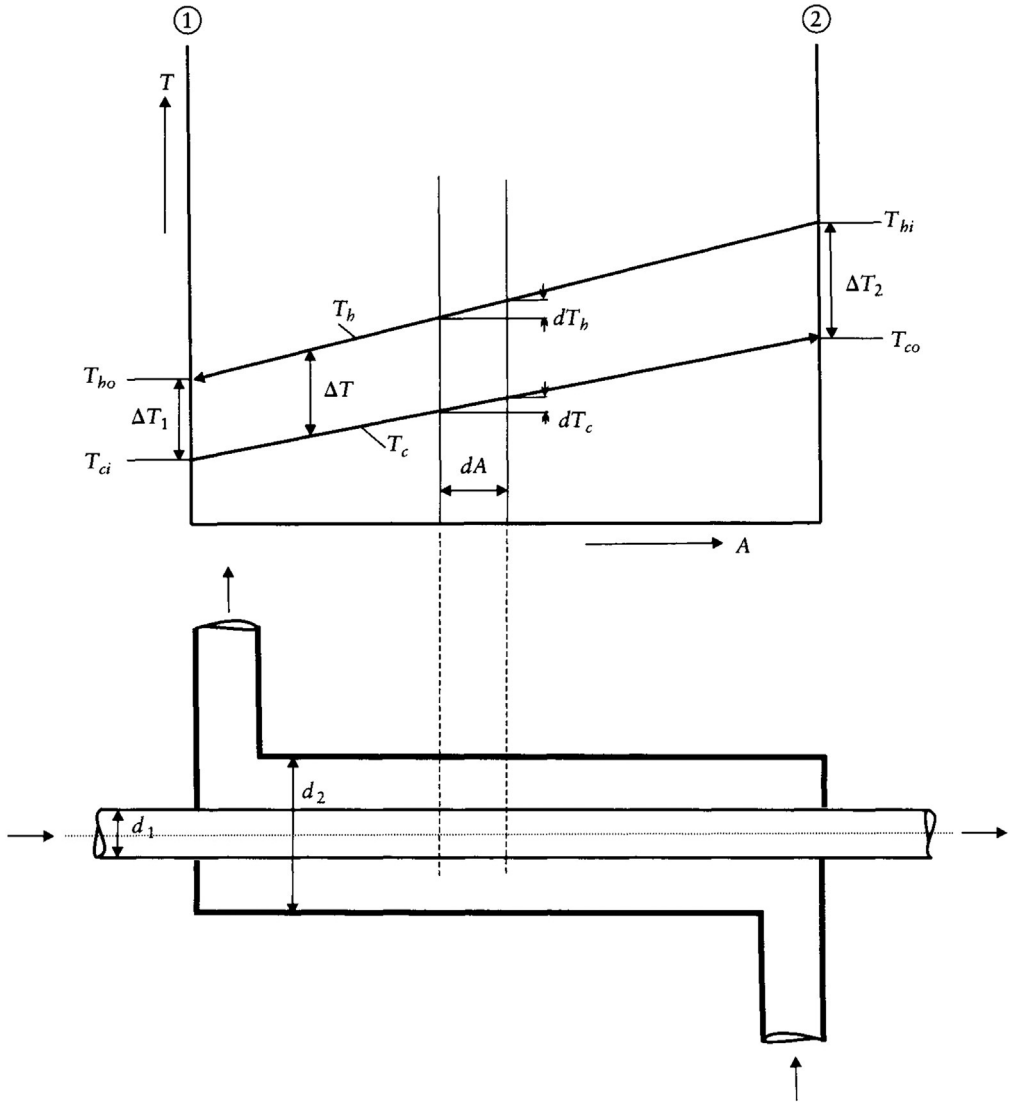


FIGURE 9.5 Counterflow heat exchanger.

Using Eqs. (9.13) and (9.16)

$$\frac{1-R\epsilon}{1-\epsilon}=\exp\left(\frac{G}{\dot{C}_h}(1-R)\right) \tag{9.17}$$

from which

$$\frac{1-\epsilon}{1-R\epsilon}=\exp(-\mathrm{NTU}(1-R)), \tag{9.18}$$

where $\mathrm{NTU}=G/\dot{C}_c$ is the dimensionless conductance, which is also defined as number of heat transfer units.

Taking the case $\dot{C}_c<\dot{C}_h$, it can then be shown that we also get

$$\epsilon=\frac{1-\exp(-\mathrm{NTU}(1-R))}{1-R\exp(-\mathrm{NTU}(1-R))}, \tag{9.19}$$

which is the same as Eq. (9.18).

This heat exchanger effectiveness is one of the important parameters that describe the performance of a counterflow heat exchanger.

In Eq. (9.19) $R=\dot{C}_{\min}/\dot{C}_{\max}<1$ is between the minimum and maximum heat capacity rates. $\mathrm{NTU}=G/\dot{C}_{\min}$ is the heat conductance divided by the minimum heat capacity rate.

Solving Eq. (9.19) for the NTU gives

$$\mathrm{NTU}=\frac{1}{1-R}\ln\left(\frac{1-R\epsilon}{1-\epsilon}\right). \tag{9.20}$$

If the cold and hot fluids heat capacity rates are equal, then $R=1$. Eq. (9.22) gives an indefinite value, and this equation cannot be used directly. Using l'Hopital's rule as $R\to 1$ gives

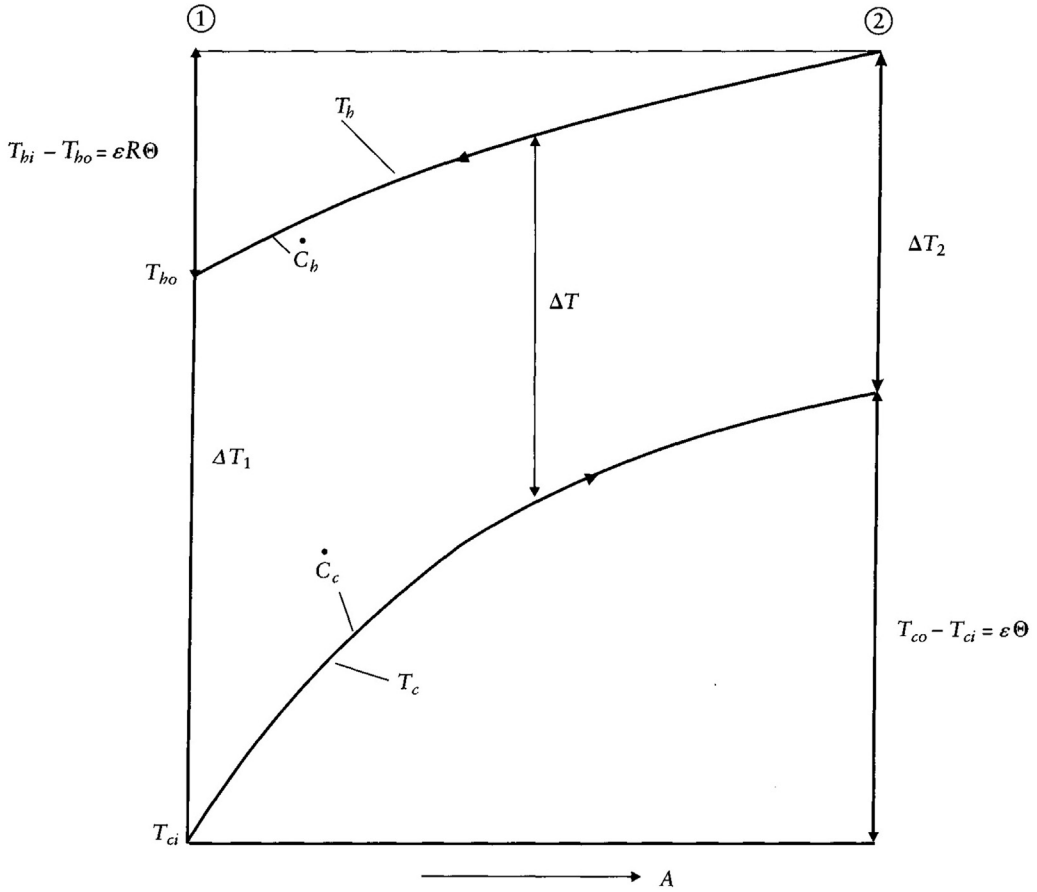


FIGURE 9.6 Counterflow heat exchanger temperature profiles when $\dot{C}_h > \dot{C}_c \cdot \theta = T_{hi} - T_{ci}$.

$$\lim \in = \frac{\lim\left(d/dR\right)(1-\exp(-\mathrm{NTU}(1-R)))}{\lim\left(d/dR\right)(1-R\exp(-\mathrm{NTU}(1-R)))} = \frac{\lim(-\mathrm{NTU}\exp(-\mathrm{NTU}(1-R)))}{\lim(-R\,\mathrm{NTU}\exp(-\mathrm{NTU}(1-R))-\exp(-\mathrm{NTU}(1-R)))} = \frac{\mathrm{NTU}}{1+\mathrm{NTU}}, \tag{9.21}$$

from which

$$\mathrm{NTU} = \frac{\in}{1-\in}. \tag{9.22}$$

In Fig. 9.6 the temperature profiles in a counterflow heat exchanger are shown when $\dot{C}_h > \dot{C}_c$.

9.3.1.3 Logarithmic mean temperature difference

The rate of temperature drop of a fluid as it flows along the length of a heat exchanger is not constant. In order to take account of this nonlinear relationship, the LMTD is used. If the inlet and outlet temperatures do not differ widely, an arithmetic mean can be used, because the relationship is considered to be linear.

In order to calculate the heat transfer from the hot to the cold fluid, the heat exchanger conductance and the temperature of the fluids at both sides of the heat exchanger must be known. The mass flow measurement is often difficult to determine; however, the temperatures are easily measured.

A temperature difference is defined that satisfies the following equation

$$\phi = G\Delta\overline{T}, \tag{9.23}$$

where ϕ is the heat transfer rate in the heat exchanger and G the conductance. For counterflow Eq. (9.12) gives

$$\ln\frac{\Delta T_2}{\Delta T_1} = G\left(\frac{1}{\dot{C}_h} - \frac{1}{\dot{C}_c}\right) = \frac{\phi}{\Delta\overline{T}}\left(\frac{1}{\dot{C}_h} - \frac{1}{\dot{C}_c}\right). \tag{9.24}$$

On the other hand, Eq. (9.15) gives

$$\phi = \dot{C}_h(T_{hi} - T_{ho}) = \dot{C}_c(T_{co} - T_{ci}), \tag{9.25}$$

giving

$$\ln\frac{\Delta T_2}{\Delta T_1} = \frac{1}{\Delta\overline{T}}(T_{hi} - T_{ho} - (T_{ci})) = \frac{1}{\Delta\overline{T}}(T_{hi} - T_{co} - (T_{ho} - T_{ci})) = \frac{1}{\Delta\overline{T}}(\Delta T_2 - \Delta T_1). \tag{9.26}$$

Eq. (9.26) gives the temperature difference defined in Eq. (9.23) as

$$\Delta\overline{T} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} = \Delta T_{\ln}. \tag{9.27}$$

From Fig. 9.6 for a countercurrent heat exchanger

$$\Delta T_1 = T_{ho} - T_{ci}$$

and

$$\Delta T_2 = T_{hi} - T_{co}.$$

For a countercurrent heat exchanger, the logarithmic temperature difference is then

$$\Delta T_{\ln} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln((T_{hi} - T_{co})/(T_{ho} - T_{ci}))}. \tag{9.28}$$

This is the *logarithmic temperature difference* for the counterflow heat exchanger.

The LMTD is defined when $\Delta T_2 \neq \Delta T_1$.

Consider the case where $\Delta T_2 = \Delta T_1$. The logarithmic temperature difference is obtained by applying l'Hopital's rule as $\Delta T_2 \to \Delta T_1$, giving

$$\begin{aligned}\lim \Delta T_{\ln} &= \lim \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} \\ &= \frac{\lim(f/d\Delta T_2)(\Delta T_2 - \Delta T_1)}{d/d\Delta T_1(\ln(\Delta T_2/\Delta T_1))} = \Delta T_2.\end{aligned} \tag{9.29}$$

The LMTD is the same as the temperature difference at the entrance and exit of the heat exchanger, that is, $\Delta T_1 = \Delta T_2 = \Delta T_{\ln}$.

For a counterflow heat exchanger, when $\Delta T_2 = \Delta T_1$

$$T_{hi} - T_{co} = T_{ho} - T_{ci} \tag{9.30}$$

or

$$T_{hi} - T_{co} = T_{co} - T_{ci}. \tag{9.31}$$

Then from Eq. (9.15), $\dot{C}_h = \dot{C}_c$ when $\Delta T_1 = \Delta T_2 = \Delta T_{\ln}$.

Example 1

A brine solution enters a counterflow heat exchanger at $T_{li} = 31.7°C$, and air enters at $T_{ai} = 24.4°C$. The measured outlet temperatures of the brine solution and air are $T_{lo} = 27.2°C$ and $T_{ao} = 30°C$, respectively.

The mass flow rate of the hot fluid is $q_{ml} = 0.382$ kg/s and of the cold fluid, $q_{ma} = 0.9$ kg/s. The specific heat capacity for brine is $c_{pl} = 3.12$ kJ/(kg K) and for air, $c_{pa} = 1.007$ kJ/(kg K).

1. Calculate the total conductance of the heat exchanger.
2. If the liquid mass flow rate is reduced to $q_{ml} = 0.3$ kg/s, calculate the new outlet temperatures of the brine and air.

Solution

1. The total heat conductance can be calculated from Eqs. (9.23) and (9.28) as $T_{hi} = T_{li}$, $T_{ho} = T_{lo}$, $T_{ci} = T_{ai}$, and $T_{co} = T_{ao}$.

$$G_{\text{tot}} = \frac{\varphi}{(T_{li} - T_{ao}) - (T_{lo} - T_{ai})} \ln \frac{T_{li} - T_{ao}}{T_{lo} - T_{ai}},$$

where $\varphi = q_{ma}c_{pa}(T_{ao} - T_{ai}) = 5.1$ kW is the heat flow to the air, which gives $G_{\text{tot}} = 2310$ W/K, $\varphi = q_{ml}c_{pl}(T_{li} - T_{lo}) = 5.36$ kW is the heat flow from the liquid, which gives $G_{\text{tot}} = 2430$ W/K. The difference is due to errors in flow and temperature measurements. The difference between the heat flow in the two cases, 5%, is satisfactory. Then, an average estimation can be obtained as $G_{\text{tot}} = 2370$ W/K.

2. The heat capacity rates of liquid and air are calculated by, respectively,

$$\dot{C}_l = q_{ml}c_{pl} = 0.3 \times 3120 = 936 \text{ W/K}$$

and

$$\dot{C}_a = q_{ma}c_{pa} = 0.9 \times 1007 = 906 \text{ W/K}.$$

Therefore

$$\dot{C}_a < \dot{C}_l.$$

The maximum heat transfer in the heat exchanger is

$$\phi_{\max} = \dot{C}_{\min}(T_{li} - T_{ai}) = 906 \times (31.7 - 24.4) = 6.61 \text{ kW}.$$

The total heat conductance is considered constant and not influenced by the mass flow rate of the liquid.

Then the number of heat transfer units is

$$\text{NTU} = \frac{G_{\text{tot}}}{\dot{C}_{\min}} = \frac{2370}{906} = 2.62.$$

Calculating the heat capacity rates ratio gives

$$R = \frac{\dot{C}_{\min}}{\dot{C}_{\max}} = \frac{906}{936} = 0.968.$$

From Eq. (9.19)

$$\begin{aligned}\epsilon &= \frac{1 - \exp(-z(1-R))}{1 - R\exp(-z(1-R))} \\ &= \frac{1 - \exp(-2.62(1-0.968))}{1 - 0.968\exp(-2.62(1-0.968))} = 0.73.\end{aligned}$$

The actual heat transfer is calculated from

$$\phi = \epsilon\phi_{\max} = 0.73 \times 6.61 = 4.84 \text{ kW}.$$

The exit temperature of the liquid is calculated from

$$\phi = \dot{C}_l(T_{li} - T_{lo}),$$

which gives

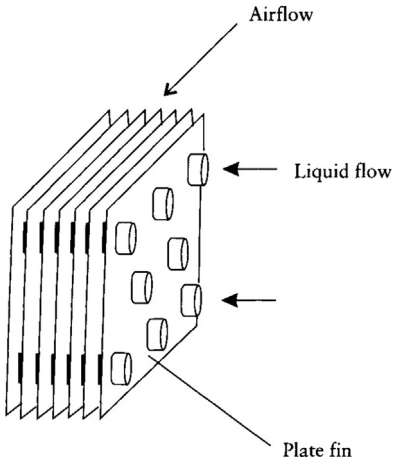


FIGURE 9.7 Plate fin-and-tube heat exchanger.

$$T_{lo} = T_{li} - \frac{\varphi}{\dot{C}_l} = 31.7 - \frac{4840}{936} = 26.5\ ^\circ\text{C}.$$

Similarly, the exit temperature of air is calculated from

$$\phi = \dot{C}_a(T_{ao} - T_{ai}),$$

which gives

$$T_{ao} = T_{ai} + \frac{\phi}{\dot{C}_a} = 24.4 + \frac{4840}{906} = 29.7\ ^\circ\text{C}.$$

A reduction in liquid flow reduces the outlet temperature of the liquid and air.

9.3.2 Plate fin-and-tube heat exchangers

9.3.2.1 *Introduction*

A common type of heat exchanger used in industrial ventilation is the plate fin-and-tube heat exchanger (Fig. 9.7). Liquid or gas flows in the tubes, with a gas or a liquid circulating outside the tubes between the plates.

The plates can be straight or wavy. Wavy plates, due to their greater surface area, enhance the heat transfer between air and the plates but are unsuitable in dusty environments.

9.3.2.2 *Annular fins*

The complete study of *annular fins* is essential in order to calculate the amount of conduction in the plate fin. A schematic diagram of annular fins is given in Fig. 9.8.

The temperature distribution within the annular fin is given by the differential equation

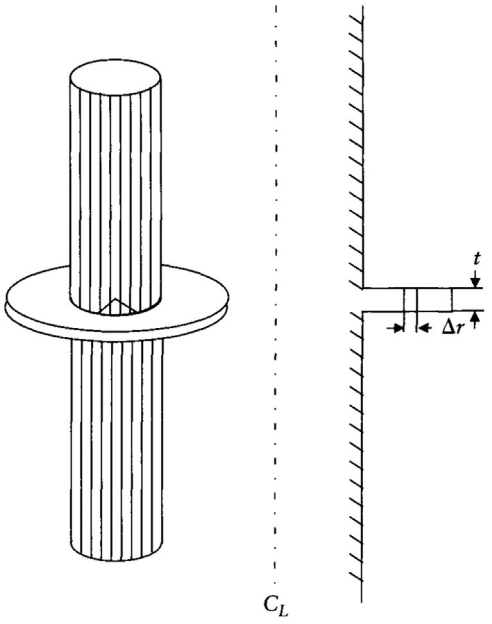


FIGURE 9.8 Annular fin.

$$\frac{d^2T}{dr^2} + \frac{1}{r}\frac{dT}{dr} - \frac{2h}{\lambda t}(T - T_\infty) = 0. \tag{9.32}$$

where h is the heat transfer coefficient between the air and the surface of the annular fin, λ is the thermal conductivity of the fin, and T_∞ is the temperature of air far away from the fin. The differential equation (9.32) together with the boundary conditions is difficult to solve and is not considered in this chapter.

Normally, heat exchanger tubes are arranged in a staggered manner. The continuous plate is considered as being built up of regular hexagons, with a hole in the center for the pipes. This arrangement is shown in Fig. 9.9.

If hexagons are replaced with an annular area of the same size the outer radius of the annular area will be

$$r_o = \sqrt{\frac{\sqrt{3}}{2\pi}}\, l = 0.525l, \tag{9.33}$$

where l is the distance between the pipe centerlines in the heat exchanger. The inner hole diameter in the annular area is the same as the pipe's outer diameter.

9.3.2.3 *Fin efficiency*

The fin heat transfer is determined by using fin efficiency. The fin efficiency is calculated using a theoretical approach where the whole fin is considered to be at the same temperature as the fin base. The required parameters necessary to determine the fin efficiency are shown in Fig. 9.10.

The heat transfer through the fin is

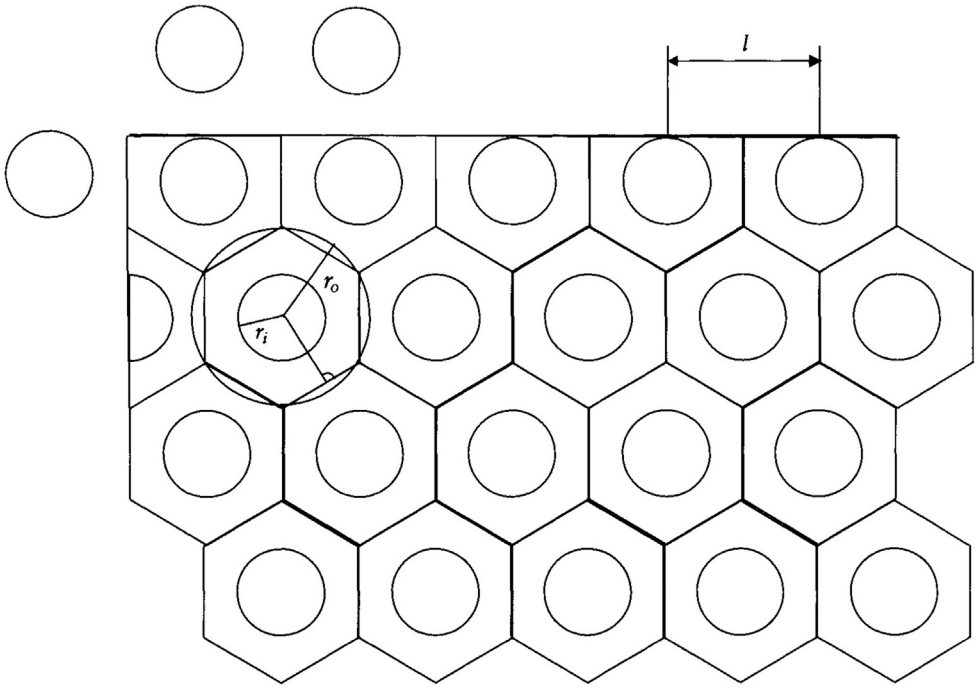


FIGURE 9.9 A continuous plate built up by hexagons.

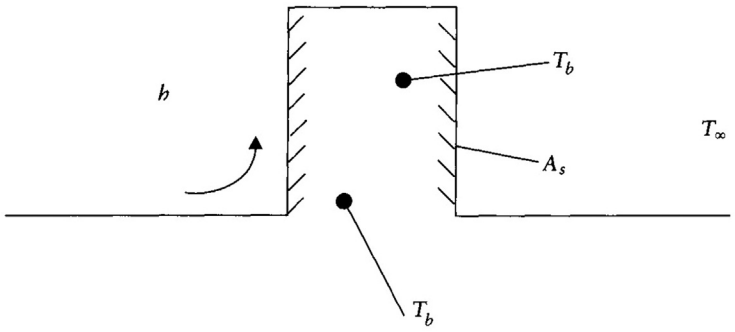


FIGURE 9.10 The comparison for the definition of fin efficiency.

$$\phi_{\text{theor}} = hA_s(T_b - T_\infty) = hA_s\theta_t, \tag{9.34}$$

where A_s is the surface area of the fin.

The fin efficiency is defined by the division of the actual by the theoretical heat transfer, that is,

$$\eta = \frac{\phi_{\text{actual}}}{\phi_{\text{theor}}} = \frac{G\theta_t}{hA_s\theta_t} = \frac{G}{hA_s}. \tag{9.35}$$

The actual heat flow through the fin is

$$\phi_{\text{actual}} = G\theta_t = h\eta A_s\theta_t. \tag{9.36}$$

The actual heat flow is calculated by multiplying the fin outer surface area by the fin efficiency. The outer surface area is easy to determine; hence, if the fin efficiency is known, the heat transfer from the fin is easily calculated.

The efficiency depends on the fin geometrical configuration, the fin thermal conductivity, and the heat transfer coefficient at the fin surface.

The use of Eq. (9.36) to determine the heat transfer of annular fins is a practical approach. The fin efficiency curve is constructed by solving Eq. (9.34) for adiabatic boundary condition of the outer surface. Fig. 9.11 shows the fin efficiency of an annular fin. (The curves G/G_∞ in the figure are not required in this analysis.)

9.3.2.4 *The convective heat transfer coefficient between the plate and flowing air*

To use Fig. 9.11, a means of calculating the heat transfer coefficient between the plate and the flowing

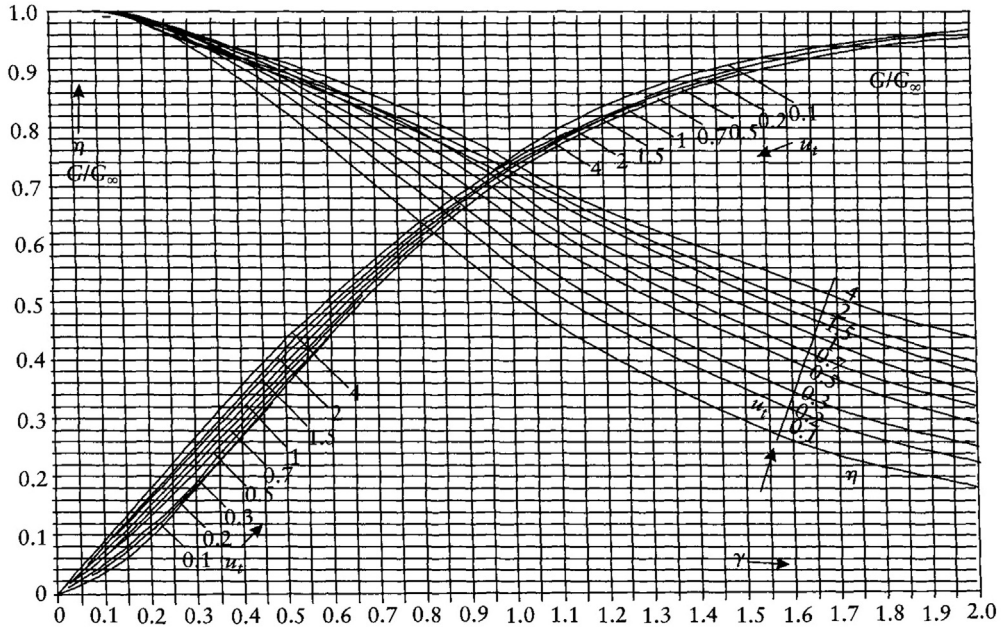


FIGURE 9.11 Fin efficiency of annular fins.

air is required. Consider a straight plate. The heat transfer coefficient is calculated using Fig. 9.12.

The abscissa in Fig. 9.12 is

$$Re = \frac{q''_m \cdot D_h}{\mu} \cdot 10^{-3}. \tag{9.37}$$

Here

$$q''_m = \frac{q_m}{A_{\min}}. \tag{9.38}$$

q_m is the mass flow of air through the heat exchanger, and $A_{\min}$ is the total area between the fins perpendicular to the airflow.

D_h is the hydraulic diameter of one cross-section between the plates, calculated from

$$D_h = 2\frac{A_{\min}}{H \cdot N_{pg}}. \tag{9.39}$$

where H is the height of the plates in the heat exchanger (assuming the plates are in a vertical position, which is normal), and N_{pg} is the number of gaps between the plates. μ in Eq. (9.37) is the dynamic viscosity of the gas.

The ordinate in Fig. 9.12 is $St\ Pr^{2/3}$, where Pr is the Prandtl number of air and is defined as

$$Pr = \frac{c_p \mu}{\lambda}. \tag{9.40}$$

For air at 20°C–40°C, the Prandtl number is 0.71.

St is the Stanton number, defined as

$$St = \frac{h}{q''_m \cdot c_p}. \tag{9.41}$$

Use Fig. 9.12 to calculate the Reynolds number, Re. Then by reading from one of the curves 1 … 8, depending on how many fins there are per centimeter, the value of $St\ Pr^{2/3}$ can be obtained.

The convective heat transfer coefficient is then

$$h = c_p q''_m St = c_p q''_m \frac{St \cdot Pr^{2/3}}{Pr^{2/3}}. \tag{9.42}$$

Next consider the wavy plate with herringbone waves. This arrangement with the relevant geometrical parameters is shown in Fig. 9.13.

The heat transfer coefficient for a herringbone plate is calculated from[16]

$$h = \lambda Re_D Pr^{1/3} i/D. \tag{9.43}$$

In Eq. (9.43) we have

$$Re_D = \frac{q_m \cdot D}{A_c \cdot \mu}, \tag{9.44}$$

where D is the tube outer diameter including the collar of the fin plate, and A_c is the flow area in the heat exchanger where maximum velocity occurs. For a symmetric, staggered configuration, the maximum velocity occurs between the pipes situated vertically on each other.

The factor j is calculated from

$$j = 0.394 Re^{-0.357} \left(\frac{P_t}{P_l}\right)^{-0.272} \left(\frac{s}{D}\right)^{-0.205} \left(\frac{x_f}{p_d}\right)^{-0.558} \left(\frac{p_d}{s}\right)^{-0.133}. \tag{9.45}$$

The relevant geometrical parameters for Eq. (9.45) are given in Fig. 9.13.

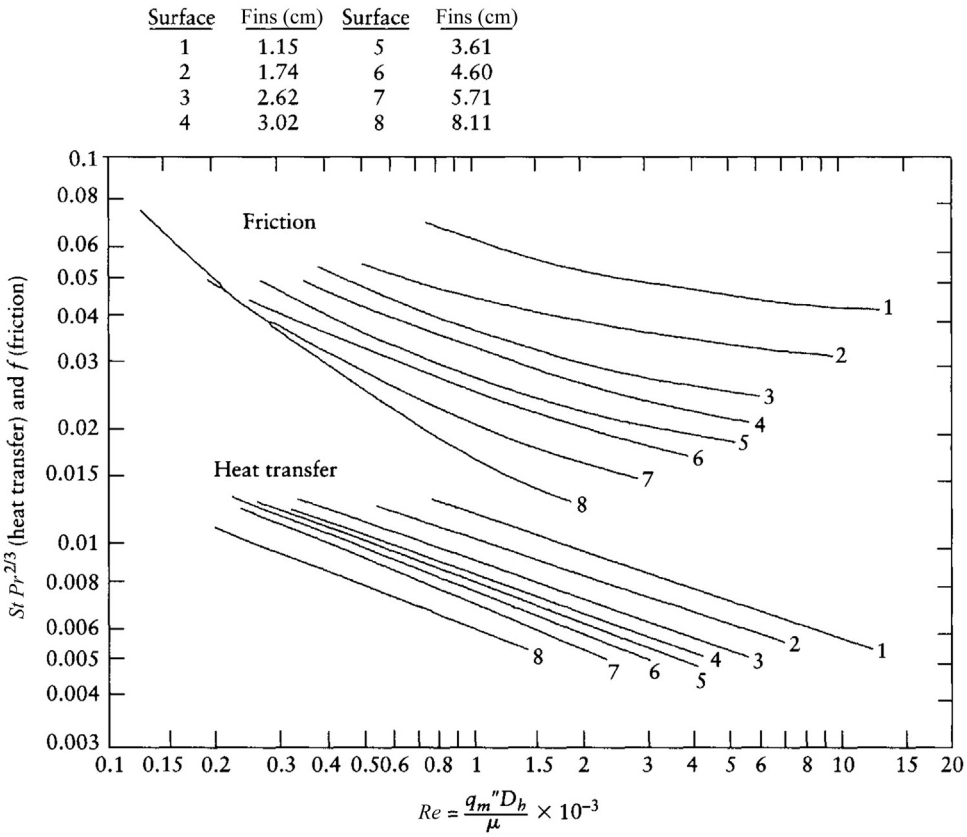


FIGURE 9.12 Diagram for determining the heat transfer coefficient for a straight fin plate.

Example 2

The data of a heat exchanger is given as follows:

- Copper pipes have outer diameter of 12 mm.
- Symmetric, staggered pipe configuration; distance between the pipe centerlines is 32 mm.
- Aluminum plates ASTM 1110 of thermal conductivity 200 W/(m K).
- Plate thickness is 0.3 mm; distance between the plates is 3 mm.
- Number of plates is 200.
- Height of the plate is 600 mm; depth in flow direction is 290 mm.
- Number of pipe rows is 10, 18 pipes in one vertical row.

If the mass flow of air through the heat exchanger is 0.9 kg/s, then calculate:

1. the fin efficiency,
2. the heat conductance or reciprocal of heat resistance on the air side if the plates are straight, and
3. the air-side conductance if the plates have herringbone waves with $x_f = 4.3$ mm and $p_d = 1$ mm.

The physical properties of air can be determined at 20°C.

Solution

1. We calculate first the total heat transfer surface of the air side. The distance between the plates is $s = 3$ mm, plate thickness is $t = 0.3$ mm, and outer diameter of the pipe is $d_o = 12$ mm. The height of plates is $H = 600$ mm, and the depth of the plates is $L = 290$ mm. The number of gaps between the plates is $N_{pg} = 199$, and the number of plates is $N_{pl} = 200$. The total number of pipes is $N_p = 10 \cdot 18 = 180$.

The outer surface area of the pipes between the fins is

$$A_{pg} = N_{pg}\pi(d_o + 2t)sN_p = N_{pg}\pi DsN_p$$
$$= 199 \cdot \pi \cdot 0.0126 \cdot 0.003 \cdot 180 = 4.3 \text{ m}^2.$$

The area of the plate fins is

$$A_{pl} = 2 \cdot N_{pl} \cdot \left(H \cdot L - N_p \cdot \frac{\pi D^2}{4} \right)$$
$$= 2 \cdot 200 \left(0.6 \cdot 0.29 - 180 \cdot \frac{\pi \cdot 0.0126^2}{4} \right) = 60.6 \text{ m}^2.$$

As can be seen, the outer surface area of the pipes between the fins contributes in a small heat transfer area, as expected.

The surface area $A_{\min}$ is

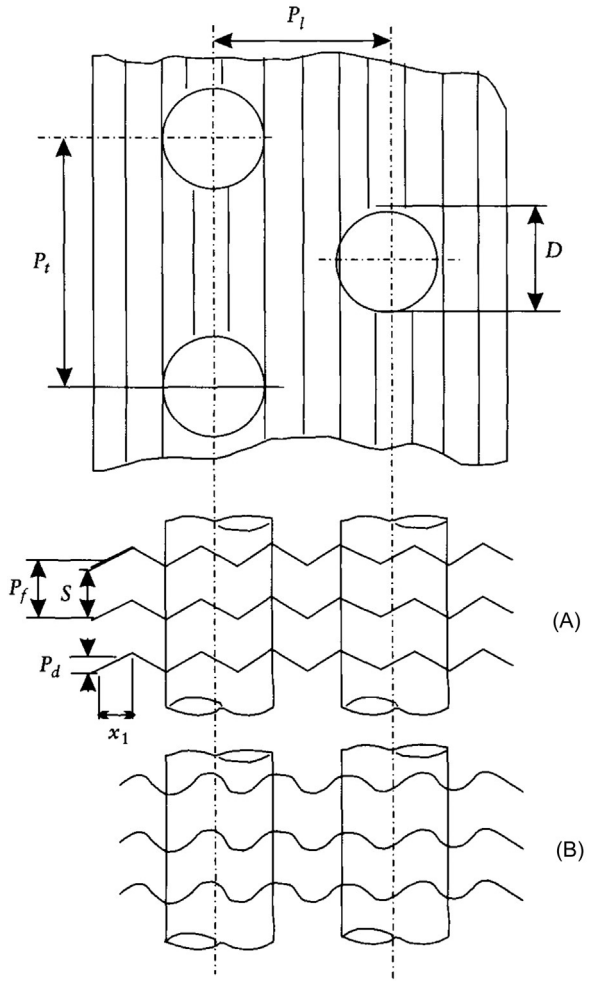


FIGURE 9.13 Wavy fin configuration: (A) herringbone and (B) smooth waves.

$$A_{\min} = N_{pg} \cdot s \cdot H = 199 \cdot 0.003 \cdot 0.6 = 0.358 \text{ m}^2.$$

At 20°C, the physical properties of air are $\mu = 181.1 \cdot 10^{-7}$ N s/m^2, $c_p = 1.007$ kJ/(kg K), $\lambda = 0.0257$ W/(m K), and $Pr = 0.709$.

For $q_m = 0.9$ kg/s Eq. (9.38) gives

$$q''_m = \frac{q_m}{A_{\min}} = \frac{0.9}{0.358} = 2.51 \text{ kg/m}^2/\text{s}.$$

From Eq. (9.39) we get

$$D_h = 2\frac{0.358}{0.6 \cdot 199} = 0.006 \text{ m}.$$

Eq. (9.37) gives

$$Re = \frac{2.51 \cdot 0.006}{181.1 \cdot 10^{-7}} = 832.$$

The plates at 1 cm distance are $10/3.3 = 3.03$. Using Fig. 9.12 we can read $St \cdot Pr^{2/3} = 0.0085$.

From this we obtain

$$h = 1007 \cdot 2.51 \cdot \frac{0.0085}{0.709^{2/3}} = 27.0 \text{ W/m}^2/\text{K}.$$

For $l = 32$ mm, we obtain from Eq. (9.33)

$$r_o = 0.525 \cdot 32 \text{ mm} = 16.8 \text{ mm},$$
$$r_i = \frac{D}{2} = \frac{12 + 2 \cdot 0.3}{2} = 6.3 \text{ mm},$$

and

$$r_o - r_i = 16.8 - 6.3 = 10.5 \text{ mm} = 0.0105 \text{ m}.$$

The thermal conductivity of aluminum is $\lambda = 200$ W/(m K); thus we obtain

$$\gamma = (r_o - r_i)\sqrt{2h/\lambda t} = 0.0105 \cdot \sqrt{\frac{2 \cdot 27}{22 \cdot 0.0003}} = 0.30$$

and

$$u_t = \sqrt{2h/\lambda t} \cdot r_i = \sqrt{\frac{2 \cdot 27}{220 \cdot 0.0003}} \cdot 0.0063 = 0.180.$$

Using the above two parameters, the fin efficiency is obtained from Fig. 9.11 as $\eta = 0.95$.

2. Applying Eq. (9.36), we obtain the heat conductance on the air side as

$$G_a = h(\eta A_{pl} + A_{pg}) = 27.0(0.95 \cdot 60.6 + 4.3) = 1638 \text{ W/K}.$$

3. Because $P_t = l = 32$ mm and the pipe arrangement is staggered and symmetric, we have

$$P_l = \sin 60^\circ \cdot P_t = \sin 60^\circ \cdot 32 = 27.7 \text{ mm}.$$

A_c in Eq. (9.44) is $A_c = 199 \cdot 17(32 - 2 \cdot 6.3) \cdot 3 = 0.197$ m^2.

Then in Eq. (9.44)

$$Re_D = \frac{0.9 \cdot 0.0126}{0.197 \cdot 181.1 \cdot 10^{-7}} = 3180$$

and in Eq. (9.45)

$$i = 0.394 \cdot 3180^{-0.357} \left(\frac{32}{27.7}\right)^{-0.272} \left(\frac{3}{12.6}\right)^{-0.205} \left(\frac{4.3}{1}\right)^{-0.558} \left(\frac{1}{3}\right)^{-0.133} = 0.014.$$

Finally, from Eq. (9.43) we get

$$h = 0.0257 \cdot 3180 \cdot 0.709^{1/3} \cdot 0.0147/0.0126 = 85 \text{ W/m}^2/\text{K}.$$

We see that the wavy profile of the fin plate increases the heat transfer coefficient greatly.

We now have

$$\gamma = 0.0105 \cdot \sqrt{\frac{2 \cdot 85}{220 \cdot 0.0003}} = 0.53$$

and

$$u_t = \sqrt{\frac{2\cdot 85}{220\cdot 0.0003}}\cdot 0.0063 = 0.32$$

From Fig. 9.11, we then get $\eta = 0.86$ and finally

$$G_a = 85(0.86\cdot 60.6 + 4.3) = 4795\ \text{W/K}.$$

9.3.2.5 Liquid-side conductance and total conductance of heat exchanger

The liquid-side conductance is calculated from

$$G_l = 0.023 Re_{D_l}^{4/5}\cdot Pr_l^{0.3}\cdot \lambda_l\cdot \pi\cdot L, \tag{9.46}$$

where λ_l is the thermal conductivity of the liquid, Pr_l the Prandtl number for the liquid, and L the total length of the pipes in the heat exchanger.

For Re_{Dl} we have

$$Re_{D_l} = \frac{4q_l}{\pi D_l \pi_l} = \frac{\rho_l w_l D_l}{\mu_l}. \tag{9.47}$$

where D_l is the inner diameter of the pipes.

For the total conductance G_{tot} of the heat exchanger, we have

$$\frac{1}{G_{\text{tot}}} = \frac{1}{G_l} + \frac{1}{G_c} + \frac{1}{G_a}. \tag{9.48}$$

where $1/G_c$ is the *heat exchanger contact resistance*. The reason for the contact resistance is that there exists a resistance to heat flow between the outer surface of the pipe and the collar of the plate fins. Normally, the fins are attached to the pipes by mechanical expansion of the tubes out into the plate-fin collars. Because of this manufacturing method, the contact will not be ideal. Small gaps between the pipe surface and the collar of the fins will occur.

It is very difficult to estimate the magnitude of the contact conductance G_c. Normally the total conductance of the heat exchanger is determined, and G_c is calculated from Eq. (9.48). Only in the case that the plate fins are welded to the pipes with a metallurgical contact is the contact conductance infinite, leading to zero contact resistance, that is, $1/G_c = 0$.

9.3.3 Additional considerations of using heat exchangers and heat-recovery units

The use of heat-recovery units can generally reduce the energy use and realize an energy-efficient system in industrial ventilation. Nevertheless, they are not without a problem. Several questions should be aware of and answered before the heat-recovery units are employed:

1. The contaminants' level in the extract air (ETA) from the industrial process and the industrial building may be much higher than the comfort ventilation in residential buildings. One of the biggest problems with heat-recovery units for industrial ventilation is maintenance. The effects of the heat exchanger fouling resulting from the particulate matters and gaseous contaminants should be considered.
2. The ETA may contain corrosive substances or water-soluble chemicals that will damage or destroy the heat exchanger. They should be filtered to protect the heat transfer surface in the heat-recovery units. The first two considerations can link to Section 9.2.
3. The heat-recovery units may change the indoor moisture level when the condensation and evaporation occur inside the rotary heat wheel. It should be especially considered in the industrial where the moisture is critical for instance precision electronic processing and assembling. In addition, icing and frosting may occur inside the heat-recovery units in cold climates. Frost control and defrost strategies should be added to maintain the performance of the heat exchangers. The emerging membrane energy exchanger could be an alternative solution to the icing and frosting problem.

Besides the aforementioned types of the heat exchanger, the emerging membrane energy exchanger, run-around heat and energy exchanger, heat pipe and heat recovery with a heat pump are also applied for different industrial ventilation systems. One can find their applications in the open literature. More concrete information and considerations about complex heat exchangers can refer to Refs. [17,18].

9.4 Air-handling processes

9.4.1 Air-heating equipment

9.4.1.1 Introduction

Many different methods of heating the air for industrial ventilation purposes are possible.

In all warm-air design applications, consideration must be given to the effects of stratification in tall buildings. Stratification increases the roof and high-wall fabric losses and the air-change rate by the stack effect, and hence the ventilation loss. These effects may increase the heat loss by 25% over that of a radiant heating system.

In many cases, warm-air heating is the cheapest heating system to install from the initial-cost point of view; the running costs, however, will be higher than for a radiant heating system to provide the same conditions.

The ventilation system can be used to full advantage during the summer months with the heater off to provide outside air and assist in removing the heat gains.

9.4.1.2 Selection

The air may have to be heated for one or more of the following reasons:

- fabric heating
- heating makeup air to take care of the ventilation loss
- comfort heating
- heating to reduce the incidence of condensation
- heating to protect goods from damage in store

The selection of equipment to meet the above needs depends on many factors, which include the following options:

- Direct air warming is achieved by means of air-heater batteries.
- Indirect warming of the air is achieved by the use of radiant heaters.

The heating media used can be classified as:

1. low-temperature hot water (up to 100°C)
2. medium-temperature hot water (100°C–120°C)
3. high-temperature hot water (over 120°C)
4. steam
5. electricity
6. heat recovery from process hot gases
7. direct gas fired
8. heat transfer fluids
9. direct oil fired
10. solid fuel firing

Note: Items 1–3 in the above list are also classed as low-pressure, medium-pressure, or high-pressure hot water.

9.4.1.3 Air-heating coils

These are used in conjunction with a ventilation or air-conditioning system and may be included in one of the following methods of ventilation:

- Natural ventilation.
- Mechanical extract–induced input.
- Mechanical input–forced extract. This arrangement is known as a plenum system.
- Mechanical inlet–mechanical extract.

In order for coils included in any of the above systems to operate efficiently, they must be designed to have a uniform air velocity across the whole of the heater's face area. This is of prime importance, and the manufacturer's specifications regarding the maximum and minimum air velocities must be met.

From the energy and noise point of view, care must be taken to prevent undue airflow resistance. This is achieved by not normally having more than five tube rows.

Typical face-area air velocity across extended-surface finned coils is normally less than 3.5 m/s. However, in the case of a plain heater, a velocity of 4 m/s can be used. In certain instances, it is possible to use higher velocities; in all cases, however, the design requirements of the manufacturer should always be met.

In a dusty industrial environment, finned heaters are more liable than plain tube heaters to become blocked or coated with dust. If this dust is greasy, it will bake on the high-temperature surfaces, reducing the rate of heat transfer and increasing the pressure drop across the coils. Special attention must be paid to the provision of easy access to areas for cleaning and maintenance of the coils; the cleaning is readily achieved by using compressed air or steam.

Air heaters in industrial environments require corrosion-protective finishes that are capable of protecting the coil and case from damage by condensation, acid vapors, or aggressive chemicals, in the air or the primary medium. If air washers are used with coils placed after them, copper or other noncorroding metal tubes should be used.

Full consideration of the thermal expansion of the tubes is necessary, with adequate provision for expansion and contraction. It is a wise policy to fit thermometer wells in the pipes near the inlets and outlets of all air-heating batteries, as these provide a useful means of checking the coil performance.

9.4.1.4 Heat requirements

To determine the heat requirements of makeup air, the following equations can be used:

$$\text{Fabric loss } \Phi_f = \Sigma(\text{AU})(\theta_{ei} - \theta_{ao}) \tag{9.49}$$

$$\text{Ventilation loss } \Phi_v = 0.33\text{NV}(\theta_{ai} - \theta_{ao}) \tag{9.50}$$

Eq. (9.50) is a simplified equation that can be used at normal conditions. For practical purposes,

c (specific heat capacity of air) = 1.01 kJ/kg °C
ρ air density = 1.2 kg/m^3

$$0.33 = \frac{(1.01)(1.2)}{3600}$$

(The number of seconds in 1 hour is 3600.)

where Φ_f is the fabric loss through all elements (kW), Φ_v is the ventilation loss (kW), A is the surface area of elements (m^2), U is the overall thermal transmittance (W/m^2 °C), θ_{ei} is the environmental temperature (°C), θ_{ao} is the outdoor air temperature (°C), θ_{at} is the indoor

air temperature (°C), N is the number of air changes per hour, and V is the volume of room (m^3).

The sum of the two losses, Φ_f and Φ_v, will give Φ_p. This is the plant load. It must be remembered that the plant load is not necessarily the sum of fabric and ventilation loss, as ductwork losses in adjacent areas have to be considered.

For more exact work the temperature ratios of F_1 and F_2 related to the mode of heating can be used. These relate to 100% convective, such as forced warm air to a high-temperature radiant system, which gives 90% radiant and 10% convective.

These factors compensate for the relationship between the inside air and mean surface temperature and provide similar comfort conditions at the center of the space, regardless of the mode of heating employed. These ratios are

$$F_1 = \frac{\theta_{ei} - \theta_{ao}}{\theta_c - \theta_{ao}} \tag{9.51}$$

and

$$F_2 = \frac{\theta_{ai} - \theta_{ao}}{\theta_c - \theta_{ao}}. \tag{9.52}$$

The CIBSE Guide Book A.9 has seven tables that cover these ratios for different heating modes.

A space isolated from external building surfaces will have its heat gains from internal sources, such as process, lighting, and occupants. And as such, these may be considered as being net sensible and latent heat gains throughout the year requiring cooling.

To form a heat balance for the flows, net sensible heat flow from the space = sensible heat absorbed by the ventilating air.

Hence, sensible heat = air-mass flow rate (q_m, kg/s) × specific heat capacity of humid air at constant volume (c_v), which is 1.012 kJ/(kg K).

It is normal in duct design to use volume flow rate rather than the mass flow rate, giving

$$q_v(\mathrm{m}^3/s) = q_m(\mathrm{kg}/s) \times \rho\ (\text{density, } \mathrm{kg}/\mathrm{m}^3). \tag{9.53}$$

The density of air at 20°C and 101.325 kPa can be taken as $\rho = 1.205\ \mathrm{kg}/\mathrm{m}^3$.

Supply air to a space can be at any temperature θ_s.

The general gas laws show that air density is inversely proportional to its absolute temperature, hence

$$pV = mRT$$

where p is the absolute pressure of the air (Pa), V is the volume of air (m^3), m is the mass of air (kg), R is the gas constant (287.1 J/(kg K)), and T is the absolute temperature of the air (K).

and

$$\rho = \frac{m\ (\mathrm{kg})}{V\ (\mathrm{m}^3)} = \frac{p\ (\mathrm{Pa})}{RT},$$

assuming that the values of the gas constant, the moisture content, and pressure remain constant in the heating or cooling process.

$$\rho_1 = \frac{1}{T}$$

Let standard air density be ρ_1 and supply air density be ρ_2.

$$\rho_1 = \frac{1}{T_1} \text{ and } \rho_2 = \frac{1}{T_2}$$

Dividing and inverting gives

$$\frac{\rho_2}{\rho_1} = \frac{T_1}{T_2}$$

and

$$\rho_2 = \rho_1 \frac{T_1}{T_2}.$$

In practice, the air density has to be corrected for the specific supply temperature by

$$\rho_2 = 1.1906 \times \frac{273 + 20}{273 + \theta_s}\ \mathrm{kg}/\mathrm{m}^3.$$

Substituting into the heat balance equation

Sensible heat (Φ_{SH}, kW) = q_v (m^3/s) × ρ_2 (kg/m^3) × c (kJ/(kg K) × ($\theta_r + \theta_s$) K)

$$q_v \times 1.1906 \times \frac{273 + 20}{273 + \theta_s}(1.0048)(\theta_r - \theta_s). \tag{9.54}$$

Hence

$$q_v \times 351 \times \frac{\theta_r - \theta_s}{273 + \theta_s}, \tag{9.55}$$

or, expressed in terms of volume flow,

$$q_v = \frac{\mathrm{kW}(273 + \theta_s)}{351(\theta_r - \theta_s)}. \tag{9.56}$$

If the space temperature is above its design value, cooling is required. If the space temperature is below the design value, the supply air must be heated. Hence

$$q_v = \frac{\Phi_{SH}(\mathrm{kW})(273 + \theta_s)}{351(\theta_s - \theta_r)}\mathrm{m}^3/s \tag{9.57}$$

or

$$\Phi_{SH}(\mathrm{kW}) = \frac{q_v(351)(\theta_s - \theta_r)}{273 + \theta_s}. \tag{9.58}$$

The rates of extract involved in industrial ventilation are by nature of a high volume. It is of interest to consider the energy required to heat one cubic meter

of air from, say, an outdoor temperature of −5°C to be discharged into the space at 20°C.

The basic equation is

$$\Phi_{SH}(\text{kW}) = \frac{q_v(351)(\theta_s - \theta_r)}{(273 + \theta_s)}$$

$$= \frac{1.0(351)(25)}{273 + 20} = 29.94 \text{ kW, say } 30 \text{ kW.}$$

The above answer gives the heat requirement for the air alone, and normally the fabric losses would be added to this figure.

It will be appreciated that if 30 kW is required for such a small air quantity, it is important to reduce the airflow rate to a value as low as possible in order to save energy.

This reduction in energy use can be achieved without affecting the required pollution levels in the environment by the improved collection methods that are covered in these guides.

9.4.1.5 Low-temperature hot-water heating coils

These are used in comfort heating systems and usually have no more than one or two rows of tubes. Various circuit arrangements are possible, depending on the pumping and control methods used.

As well as ensuring that the required design heating capacity is met, the following factors must also be considered.

- The maximum output efficiency must be met with the minimum of air pressure drop.
- The water pressure drop through the coil is as low as economically possible.

The design resistance of hot-water flow through a coil normally never exceeds 4 kPa in accelerated low-pressure hot-water heating installations.

In the case of high-pressure hot-water installations the resistance to the water flow is determined by other factors, such as the balancing of circuits.

The heaters are fed from hot-water flow and return mains, and to ensure uniform distribution of the heating medium, adequate connections to each row or bank of tubes or sections are necessary. To reduce air-locking problems, venting of the heater flow connections should be arranged. Parallel and counterflow are common arrangements with water coils. Counterflow is preferred, as this gives the highest possible mean temperature difference.

9.4.1.6 Steam-heated coils

For a steam coil to operate efficiently, it must have all the latent heat in the steam. This is achieved by the use of a steam trap. The correct trap type must be selected for the particular application in order to prevent waterlogging. All condensate, air, or other noncondensable must be removed from the system without delay; otherwise:

- The rated coil output will fall.
- Corrosion will result, causing premature coil fracture.

The best performance is achieved if the steam is uniformly distributed to the individual tubes. Properly designed and selected steam-distribution tube coils distribute the steam throughout the entire length of all primary tubes. Problems may result with air heaters operating under light load, and these may be overcome by greater sectionalization of the controls. When the entering air temperature is below freezing, the steam supply to the coil should not be modulated. Coils are located in series in the airstream, with each coil sized to be on or completely off in a specific sequence; this depends on the entering air temperature. Low temperatures produce the risk of a coil freeze-up. The use of bypass dampers could be considered, but care should be taken to ensure that cold airstreams do not impinge on the coil through the gaps in the partially closed dampers. Full provision is necessary to accommodate expansion and contraction of the coils.

Provision is necessary for adequate venting of the steam space. Heaters should have pressure gauges fitted at the steam inlet. In order to achieve efficient heat transfer, dry steam must be supplied to the battery; superheated steam is not suitable due to the time necessary for it to lose its latent heat to produce condensate.

9.4.1.7 Electric air heaters

These have the advantage that they are low-cost units to install; the running costs, however, depending on the source of electricity, are generally higher than other energy sources.

The air velocity through a heater battery should be to the manufacturer's rated output within its range of safe temperatures. In large units the electrical load is balanced across the three phases of the electrical supply.

The heaters are normally divided into a number of sections in order to provide step control of the unit. Each section of heater elements may consist of two or more rows, each having its individual busbar connections and adequate provision for it to be withdrawn for repair or cleaning, with the other elements remaining in operation. Care has to be taken with electrical isolation of each section before withdrawing them from the casing. The heaters should be electrically interlocked with the fan motors, allowing the electric heater to be switched off when the fan stops or when the air velocity is reduced to a level below that for which the heater has been designed. The risk of fire under abnormal operating conditions must be

counteracted by the use of a suitably positioned, temperature-sensitive cutoff trip of the manual reset type.

9.4.1.8 Direct-fired air heaters

These may be:

- gas fired
- oil fired
- solid fuel fired

Regardless of the fuel used, the following are the requirements of the flue.

Flues must be of the correct cross-sectional area in order to remove the products of combustion in a safe and efficient manner. The flue terminal should be fitted with a bird guard.

Care has to be taken to ensure that downdrafts do not cause the combustion products to be liberated in the occupied space. To achieve this, it is essential that the stack is of such a height that wind deflecting off adjacent structures does not influence the free passage of the products of combustion.

The appliances must be positioned in a space that has adequately sized combustion air inlet louvers that will not clog with debris and reduce the combustion air supply.

In small installations flueless appliances may be used; the use of these is to be discouraged due to the problems of toxic gases and condensate from the flue gas building up in the space.

9.4.1.9 Gas-fired heaters

Only standard, approved appliances should be used. These may operate on:

- an atmospheric burner or
- a forced jet burner.

In large industrial installations the latter is the most common arrangement. The burner has a profile plate that controls the rate of combustion air. The warm-air delivery fan may be either centrifugal or axial.

In Europe, the gas safety controls must meet the requirements of CEN standards, including flame failure devices, solenoid control valve, pilot controls, ignition, and governor. Overheat-type thermostats and either a pressure switch or an airflow-proving device are fitted to ensure that the burner will cut off in the event of no air flowing through the heater, such as occurs with fan failure.

Thermostatic control of the gas supply to the heater is required so that the air-off temperature can be controlled. This is achieved by a two-stage control that opens a valve partly on low rate and fully on high rate.

The flued appliance is designed to provide maximum heat transfer. The efficiency can be increased by the use of a condensing unit.

The filter banks incorporated require adequate access for cleaning and replacing.

The heat exchanger is normally of the welded type using aluminized steel, stainless steel, or similar materials. It should create the minimum of resistance to the air movement, which has to be turbulent in order to give an efficient transfer of heat.

Depending on the burner system employed, some form of a flue system is required to remove the products of combustion from the appliance to the atmosphere.

In the case of atmospheric type burners a draft diverter is required on the appliance (in addition to the flue outlet).

The positioning of flue terminations must be selected with care and may be subject to statutory or other regulations.

The flueless appliance has gas supplied to the burner head, and air passes outside the baffles at a designed velocity.

The air paths within these baffles are arranged to provide the correct amount of combustion air. Turndown ratios of up to 35−1 are obtainable, providing the correct gas flow for air temperature control.

The control system must satisfy a set sequence, as is required by CEN or other standards. An air switch provides airflow to the unit. A 30-second purge of the unit takes place by the fan; after this period the ignition spark and pilot gas valve are operated.

On proving the pilot ignition the main gas valve opens and the burner ignites. Once alight, the main burner will modulate to the temperature set by the room thermostat.

If, during the ignition sequence or the running sequence, flame failure occurs, a lockout will result, which will require manual resetting.

9.4.1.10 Oil-fired heaters

Burners of the vaporizing type are used only on low-output equipment. The most common type encountered is the pressure-jet burner.

All burners should meet the requirements of CEN or other national standards. The units must be complete with safety devices for ignition failure and main flame failure.

For safe operation the heater should have an overheat thermostat and either a pressure switch or airflow-proving device. This control device ensures that the burner will isolate in the event of restricted or no airflow through the heater.

The removal of the combustion products to the external atmosphere takes place through the flue, which may be:

- a direct flue connection type,
- a direct flue with a stabilizer (the stabilizer must be compatible with the unit), or
- a fan-assisted or balanced flue.

The oil supply to the heater is thermostat controlled by the air-off temperature.

9.4.1.11 Solid fuel–fired heaters

Furnace-type air heaters are manufactured from cast iron or steel, and cased in brickwork or steel. The cast-iron heaters are sectional, cemented, and bolted together. The steel type is welded or riveted. It is essential that the joints are airtight, so that the cold air can pass over the heated surfaces of the furnace and flues without contamination by the flue gases.

To reduce the possibility of combustion products being drawn into the air if the heat exchanger burns through, the air is blown and not induced through the heater.

Adequate draft is necessary, not only for efficient combustion, but also to reduce the risk of soot being deposited on the heating surfaces. Provision must be made for a cleaning door in the flue and for the tubes.

With a fan-furnace heating system, once the temperature is reached, the fan is controlled to start automatically. Should fan failure occur, provision should be made to damp down the furnace automatically to avoid overheating's causing tube damage.

9.4.1.12 Air-heating-coil selection factors

When selecting air-heater batteries, the following factors common to all types must be considered:

1. air volume to be handled (and conditions at which it is measured);
2. entering air conditions (EAT) (dry-bulb temperature, air density), air quality (possibility of dirt, corrosive, or hazardous atmospheres); whether the coils are suitable to operate in temperatures below freezing;
3. required leaving-air dry-bulb temperature (LAT);
4. the heating medium being used, considering:
   a. water and steam entering temperatures,
   b. operating pressure,
   c. permissible temperature drop of the heating medium,
   d. electricity supply, voltage, and phase and whether a standby supply is required, and
   e. maximum element operating temperature;
5. allowable air-side velocity;
6. allowable air-side resistance (as it affects fan power);
7. heating medium connections, size and type, location, circuiting arrangement;
8. provision for air release and drainage;
9. provision for cleaning tubes;
10. electricity: type and location, circuit arrangement; terminal box specification;
11. air-side connections (flanges for duct mounting etc.);
12. construction: materials for heaters, coils, and extended surface; number of rows and arrangements; type of extended surface and spacing;
13. division of coil to suit control, manufacturing, or installing arrangement, that is, number of sections and rows;
14. air pressure at point of installation (and air leakage through coil casting), particularly when located at fan discharge on a high-pressure system; and
15. economy of coil selection as related to the system, such as flow rates, temperature rise, pressure drops, and fan and pump power.

If a preheater coil is provided and positioned before the air filter, the heater should be either a plain tube or have widespread fins for easy cleaning

9.4.1.13 Selection of direct-fired air heaters

Flued heaters

This method of heating is the recommended one, and the following factors should be considered:

1. Air volume to be handled (q_v), m^3/s.
2. Entering-air conditions (θ_{ao}) (EAT).
3. Leaving-air condition (θ_{lat}) (LAT).
4. Will leaving-air temperature cause discomfort or other problems?
5. Allowable air-side resistance.
6. Is the unit provided with all the necessary safety controls?
7. Is the flue outlet suitable for easy discharge of the products of combustion into the atmosphere?
8. Are the ductwork connection flanges provided?
9. Is an access door provided for inspection of the pilot and cleaning the main burner?
10. Are sight glasses or observation ports provided for inspection of pilot and main burner flames?
11. Is a suitable temperature control provided?
12. Does the fan within the unit have sufficient pressure to overcome the system ductwork resistance?
13. Will moisture from the combustion products adversely affect the space humidity requirements?
14. The concentrations of combustion products liberated into the occupied space must be well below the accepted tolerance levels. This is the factor that limits the use of flueless heaters for makeup air supply.

9.4.2 Humidification and dehumidification

9.4.2.1 Introduction

Humidification and dehumidification (drying) of air are required in many commercial and industrial applications for the following reasons:

TABLE 9.8 Influence of moisture content in air.

Factors	High humidity	Low humidity
Ill health and associated allergies	Encourages the growth of spores and fungi.	Lowers the body's resistance to infection and respiratory disease.
Electrostatic shocks	Eliminated at high moisture content.	Increased at low moisture content, causing discomfort to occupants and damage to electronic components. High explosion risk.
Eye conditions	Possible bacteria buildup and health effects.	A high rate of evaporation of the eyeball oil film results in dry, itchy eyes with an increase in eye damage in dusty environments.
Respiratory tract (nose and throat) problems	Increase in bacterial infection.	Resistance to throat, nose, and lung infection is increased, due to the breakdown of the protective mechanisms in the body.
Thermal stress	Reduction in evaporative body cooling. In hot industries, this results in body overheating.	Increase in body cooling with an increase in air movement.
Skin effects	Bacteria growth in skinfolds. Clothing type has to be considered.	As moisture is required to keep the skin supple, low humidity causes cracking of the skin, increasing the risk of chemicals and bacteria entering the body
Materials	Rusting of steel, breakdown of timber and fabrics. Damage of stored goods by insects and mold growth.	Rapid drying out of paper, timber, and fabrics. This is an important factor for storing antiques.
Pollution		The dust concentration is higher in a dry atmosphere, causing health and discomfort problems. Odors, irritant gases, and vapors are more noticeable at low humidity.
Vegetation effects	Various diseases, depending on produce.	Various diseases, depending on produce.

- the control of air moisture content within the occupied space to ensure the well-being of human, animal, or plant life and
- the control of air moisture content within a space for process control or to protect products in store.

Table 9.8 indicates some of the factors that have to be considered when dealing with the moisture content in air. In the case of inert gases used in various industrial processes, other factors have to be considered.

The process of air humidification is achieved by means of a device called a humidifier.

9.4.2.2 *Humidifier types*

Many different types of humidifiers are in common use; the humidification process is simply achieved by adding moisture into the air to be conditioned.

The selection of a particular method depends on the size of the installation, availability of water or steam, the degree of control required, and the methods used for conditioning and distributing the air.

Humidification is achieved by one of the following methods:

1. By producing a fine mist or spray
2. By evaporation from a pad of absorbent material within the airstream
3. By vaporizing water
4. By steam injection

Fig. 9.14 shows the processes of humidification and dehumidification in a skeleton psychrometric chart, while Table 9.9 explains the cycle.

Note: The actual final air conditions given in Fig. 9.14 and Table 9.9 must always be related to the slope of the line, as in some cases the specified conditions may not be met.

Humidifiers complete with water storage

In this type of humidifier the water required for the humidification process is stored within the unit and is normally fed automatically from the water mains. However, smaller portable units do not have this automatic facility and have to be manually filled with water as required.

To prevent contamination of the water mains by back-siphonage the water inlet to the appliance should incorporate either:

- an air gap,
- an approved type of pipe interrupter, or
- a combined check and antivacuum valve.

The national and local water bylaws and codes should be consulted to ensure that the device fitted is of an approved type.

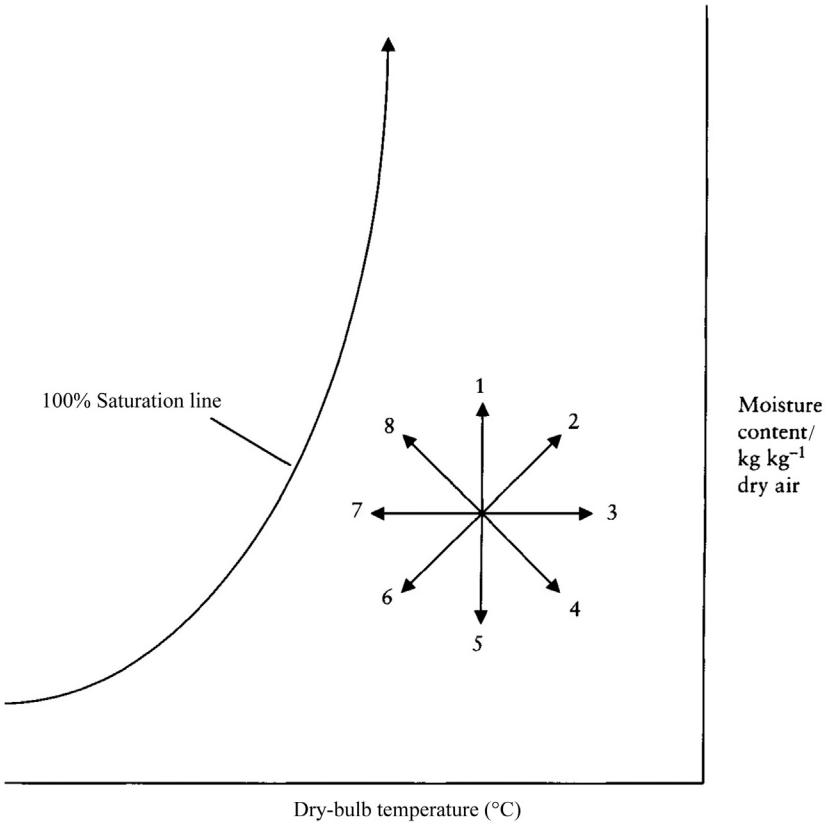


FIGURE 9.14 Humidification and dehumidification processes.

TABLE 9.9 Humidification and dehumidification processes.

State line: center point to	Process	Final air conditions	Equipment used
1	Humidification only	Increase in moisture content, specific volume, dry and wet bulb temperature, specific enthalpy, and percentage saturation.	Air washer with heated water
2	Heating and humidification	Increase in moisture content, specific volume, dry- and wet-bulb temperature, and specific enthalpy: decreased percentage saturation.	Steam humidifier or recirculated hot-water spray
3	Sensible heating only	Constant moisture content; increase in specific volume, dry- and wet-bulb temperature, and specific enthalpy; decreased percentage saturation.	Steam, hot-water coils, or electric heating
4	Dehumidification and heating	Decreased moisture content; increase in specific volume and dry- and wet-bulb temperature; decreased specific enthalpy and percentage saturation.	Chemical dehumidification
5	Dehumidification only	Decreased moisture content and specific volume; constant dry-bulb temperature; decreased wet-bulb temperature, specific enthalpy, and percentage saturation.	Not practical
6	Cooling and dehumidifying	Decreased moisture content and specific volume, dry and wet-bulb temperature, specific enthalpy, and percentage saturation.	Chilled-water washer
7	Sensible cooling only	Constant moisture content; decreased specific volume, dry- and wet-bulb temperature, and specific enthalpy; increased percentage saturation. When point 7 reaches the 100% saturation line, the air is saturated; further cooling will result in moisture being removed from the air.	Cooling coil and washer at the dew point
8	Evaporative cooling only	Increased moisture content and specific volume; decreased dry- and wet-bulb temperature; increased specific enthalpy and percentage saturation.	Washer

Spray-type humidifier

Water is pumped from a storage tank located at the base of the unit to one or more spray nozzles that inject a fine water spray into the airstream. In large units the air movement through the ductwork system ensures efficient mixing of the moisture and the air. Units used in small enclosures or rooms use a fan with the pump on the same shaft.

Pan-type humidifier

In this type an absorbent material is partially immersed in a water pan store at the base of the unit. The evaporation of the water into the airstream takes place from the wetted surface of the absorbent material.

Mechanical pan

In this case an absorbent material is fixed to a drum or disk. Part of this drum is immersed in the water. When the drum is rotated in the water pan, it absorbs moisture, which is then transferred into the airstream by evaporation.

Steam-generating pan

This type consists of an enclosed water tank, which is connected to the main air duct.

High-temperature water coils or direct-fired gas or electrical elements heat the water in the tank. The tank water vaporizes, and the moisture is entrained into the airstream as it passes over the tank.

Humidifiers without water storage

Water need not be stored within the unit; this is often the case where the equipment used for humidification forms part of a central station plant or where a suitable source of steam is available.

Spinning-disk humidifier

In this case a controlled quantity of water is discharged against a rapidly rotating disk where centrifugal force spins the droplets radially against circumferentially placed baffles, producing a fine water mist which is discharged into the airstream.

Steam jet

Steam generated from an external source positioned close to the unit is injected into the airstream. It is essential that the steam supply is uncontaminated and odorless. In order to stop scale buildup, it is essential that some means of water treatment be provided.

Ultrasonic atomization

Full use can be made of ultrasonic frequencies in producing a fine, atomized jet, which is liberated into the airstream.

Air washer

A device called an air washer is also used for humidification and dehumidification. It consists of a chamber incorporating a water spray system, a collection tank, and an eliminator section. The eliminator plates are necessary to reduce the incidence of water droplets that are carried out of the plant into the duct run.

The air, in passing through the chamber at velocities in the range 1.5–3.5 m/s, comes into intimate contact with the water, and depending on the conditions required, mass transfer of moisture into the airstream occurs. This transfer produces either addition or removal of moisture; hot or chilled water is used in this process.

The spray bank consists of a series of standpipes with nozzles connected to a horizontal header. The nozzles are arranged to ensure that the spray gives good coverage of the spray chamber without causing any interference with the adjacent nozzles. The pressure through these nozzles is normally between 140 and 280 kPa.

The spray water requirements vary from 0.3 L/s for a single bank to 0.8 L/s per 0.5 m^3.

Depending on spray pressure and the strainer, pipeline, and valve resistance, the pumping head is normally in the 16–24 m range.

By having more than one spray bank discharging into the direction of airflow, the mixing efficiency is improved. The nozzles provide water atomization by means of forcing the water under pressure through a small orifice, producing a wide-angle rotating spray.

In the past, air washers were used for air cleaning; today, however, more efficient methods are available for the cleaning process.

The continuous recirculation and spraying result in dirty water building up in the sump. In order to reduce the incidence of infection to occupants and fouling of the nozzle, water treatment with biocides and softening of the water supply are required. The sump is complete with strainers in a position that allows easy access for cleaning. See Fig. 9.15.

Another type of air washer is the capillary air washer; see Fig. 9.16. It consists of many thousands of small glass or plastic filaments (cells) providing a large, wetted surface area. These are placed to allow a parallel path of air through them. The surface of each of these strands is covered with the water that is discharged from sprays.

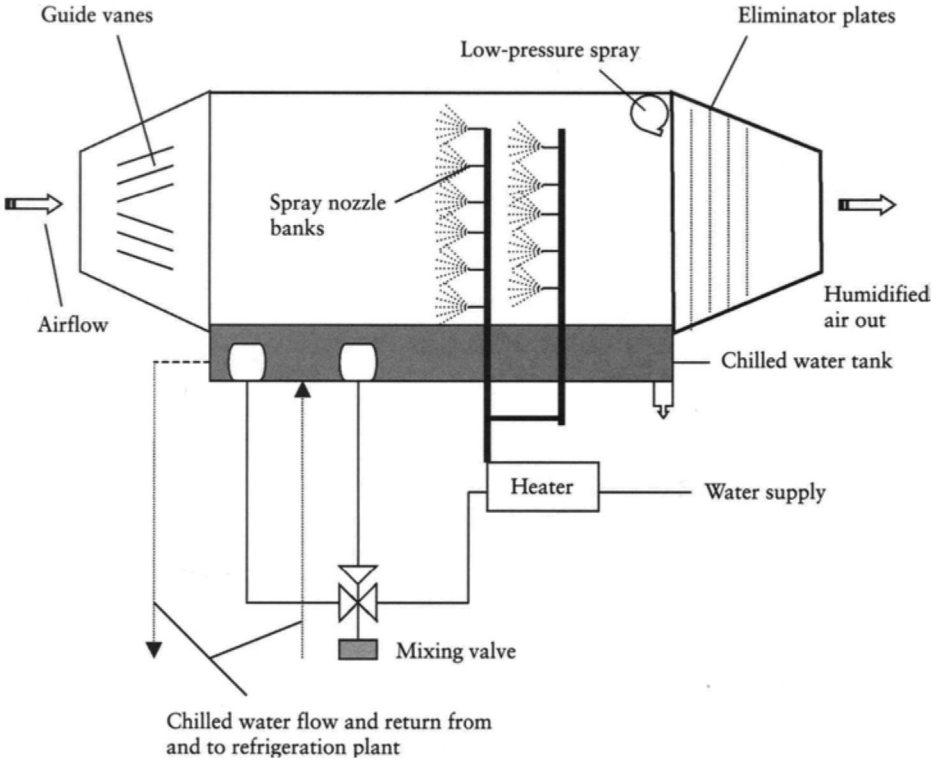


FIGURE 9.15 Air washer.

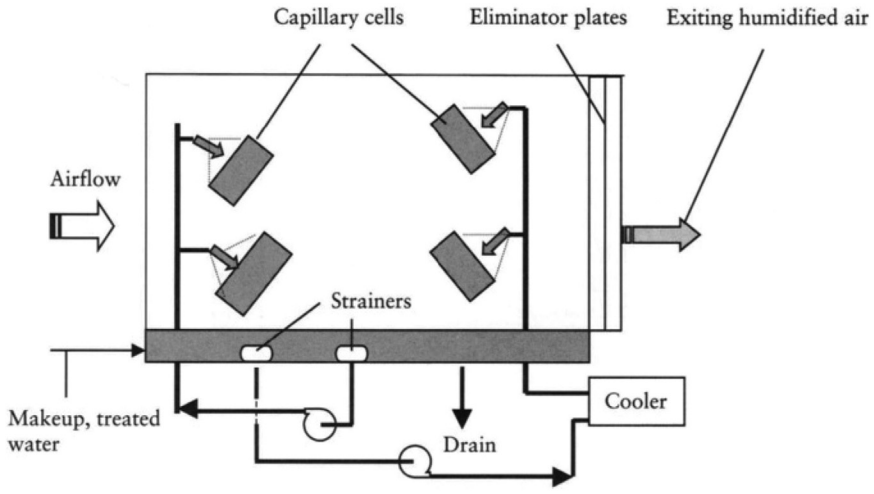


FIGURE 9.16 Capillary air washer.

9.4.2.3 Selection factors

Humidification can be achieved by placing separate humidifiers directly in the conditioned space. In order to maintain the design conditions, however, humidity control should be incorporated into the system. When positioning humidifiers, care has to be taken to ensure that the leaving moisture does not impinge on adjacent surfaces, forming lime or algae deposits.

Regardless of the type of unit used, full consideration should be given to provide a suitable method of water treatment on the following counts:

- To prevent solids from being carried over into the airstream.
- To stop deposits from fouling the unit and the associated heat exchanger.
- To stop a fine dust of the salts from being discharged into the conditioned space. This is

particularly important in the case of spinning-disk or spray-type humidifiers.

For all humidifiers, full consideration should be given to the possibility of the growth of fungi, algae, bacteria, and in particular *Legionella pneumophila*, the microbiological contamination that causes humidifier fever etc.

Dehumidification

The reverse of the humidification process is that of dehumidification. In this process the water content of air, gases, or fluids is reduced.

In many industrial applications the moisture extraction takes place at atmospheric pressure; however, certain applications may require a reduction in the atmospheric pressure in order to achieve the maximum efficiency.

The process may be required for the following reasons:

1. reducing the moisture content of gas to aid the manufacturing and handling of hygroscopic materials;
2. for comfort or process air conditioning in combination with cooling to reduce the latent load;
3. providing protective atmospheres to reduce the oxidization of metals; and
4. controlling set humidity conditions in warehouses.

Dehumidification can be achieved by one or more the following methods:

1. compression
2. refrigeration
3. liquid sorption
4. solid sorption
5. a combination of the above

Fig. 9.14 and Table 9.9 cover the dehumidification processes.

Compression

If a gas is compressed, its absolute moisture content is reduced, generally resulting in a saturated gas at the elevated pressure. Due to the high cost of providing compression, it is used on its own only in limited applications. It is, however, used as the first stage with one of the other three methods. If a high-pressure gas is allowed to expand, the increase in volume and reduction in pressure results in a lower dew point.

Refrigeration

By allowing moisture-laden, relatively warm gas to come into intimate contact with a cold surface which is below the dew point of the gas, moisture is condensed from the gas.

The term *refrigeration* refers to the gas coming into contact with evaporator coils on a dx vapor-compression cycle, coils on an absorption cycle, vortex tube, thermoelectric cycle, chilled water coils, or even water mains passing through coils.

The required refrigeration capacity is $\phi h = q_m(h_1 - h_2)$ kW.

Chemical dehumidification

Certain chemicals (sorbents) have the ability to absorb moisture from a gas; they may be either solid or liquid. Performance of a chemical dehumidification device depends on the sorbent used. The sorbent must be able to attract and remove the sorbate, such as water, from the gas stream. Sorbents absorb water on the surface of the material by adsorption or by chemically combining with water (absorption). If the unit is regenerative, the process is reversible, allowing water to be removed. This is achieved by a sorbent such as silica gel, alumina gel, activated alumina, lithium chloride salt, lithium chloride solution, glycol solution, or molecular sieves. In the case of nonregenerative equipment, hygroscopic salts such as calcium chloride, urea, or sodium chloride are used.

An absorbent material is one which changes either chemically, physically, or both during the sorption process. Certain chemicals, in absorbing moisture during this process, will dissolve into the water from the initial crystalline structure. Further added water results in a phase change from solid to liquid. An adsorbent is another material in which there are no chemical, phase, or physical changes during the sorption process.

Liquid sorption. If a moist gas is passed through sprays of a liquid sorbent, such as lithium chloride or an ethylene glycol solution, moisture is removed from the air at a rate depending on the vapor pressure difference. This is a function of the absorbent concentration and is maintained at the required level by a regeneration cycle. The regeneration process is continuous and is achieved by allowing a percentage of the chemical into the exhaust-heated air.

If the vapor pressure of the sorbent in its active state is below that of the gas being dehumidified, moisture is absorbed from the gas stream. As the process continues, the sorbent becomes diluted due to the moisture increase. See Fig. 9.17.

Solid Sorption. A material such as silica gel or activated aluminum through which the moisture-laden gas is passed achieves solid sorption; see Fig. 9.18. The mass transfer that takes place depends on the relative conditions of the chemical and the air conditions. As the material becomes saturated, heating the bed and applying a vacuum to it will reverse the process. Once sufficient water is removed from the material, it can be reused.

The intermittent operation used in the above method is time-consuming; the undesirable effects are

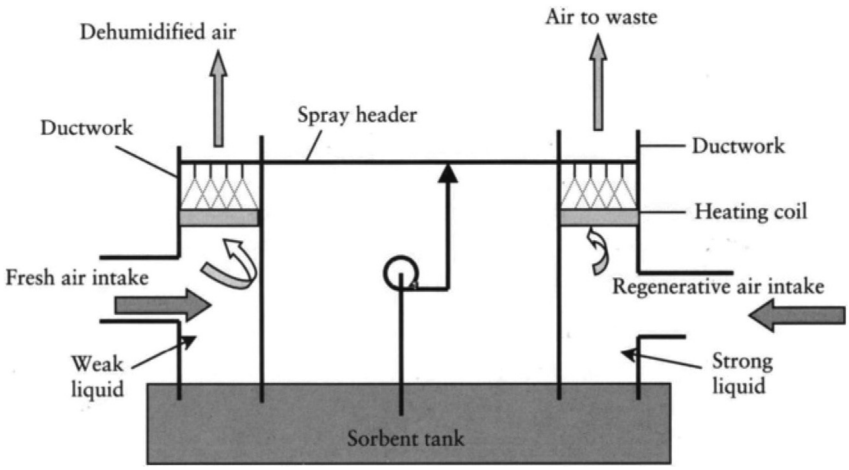


FIGURE 9.17 Liquid sorption.

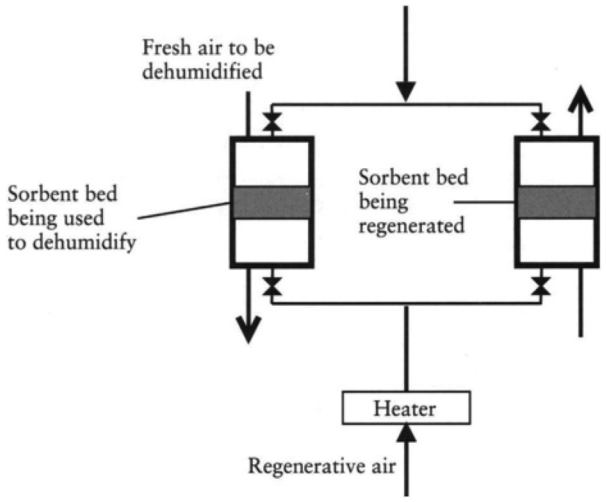


FIGURE 9.18 Solid sorption.

overcome by the use of a continuous regeneration process, as shown in Fig. 9.19.

9.4.2.4 Summary

The selection of humidifiers or dehumidifiers must not be taken lightly, as many design factors have to be considered to ensure economic owning and operating costs.

9.4.3 Air distribution

9.4.3.1 Introduction

The aim of this section is to provide a basic introduction to the methods by which air may enter a space and be distributed and to consider the governing equation for the determination of the air quantity and temperature. The governing equation relating to airflow patterns in a space is not covered in this section, as it is discussed elsewhere in the guides.

Ventilation

Ventilation is required in buildings for many different reasons. In this section the emphasis is on industrial environment; however, the general method of approach is common to all systems for the following reasons:

- to provide adequate oxygen to support life;
- to remove all odors by dilution;
- to reduce the bacteria count by providing fresh air to the space;
- to reduce any toxic gases, vapors, and dusts;
- to remove explosive gases and dusts;
- to ensure that adequate combustion air is provided to any combustion process;
- to lower the moisture content in the air, thus reducing the risk of condensation and mold growth; and
- to add to or remove heat from the space.

All the abovementioned reasons must be carried out in the most efficient manner.

The air that enters a space has to be distributed in a manner suitable for the particular application. Regardless of the method selected, the air entering the space must achieve one or more of the above requirements in the following ways:

- efficient distribution with no stagnation of the air;
- air velocities that will not cause thermal discomfort or disturb papers or manufactured goods such as light powders and fibers;
- low noise transmission level;
- low owning and operating costs; and
- capable of maintaining the internal design conditions, regardless of the external environmental conditions.

The abovementioned factors can be grouped under the following headings:

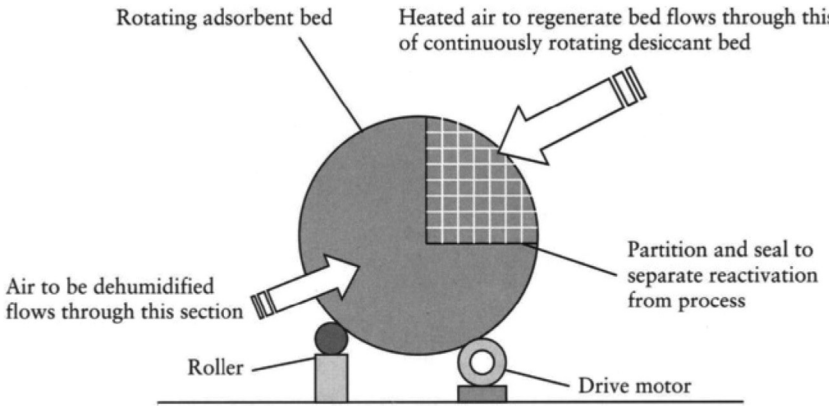


FIGURE 9.19 Automatic rotary regeneration.

- health ventilation
- safety ventilation
- comfort ventilation
- structural heating or cooling

The distribution and the associated extract of air from a space may be from:

- high level (the roof or ceiling)
- the walls, at any level
- the floor
- benches

Ventilation of a space can be achieved by the following methods:

- natural ventilation
- mechanical extract–induced inlet
- mechanical input–forced extract
- mechanical inlet–mechanical extract

The following sections consider each of these methods.

9.4.3.2 *Ventilation methods*

Natural ventilation

This is the usual method of ventilation in domestic dwellings and many small office buildings and workshops. New standards, however, require buildings to have set ventilation rates, which require mechanical ventilation systems. However, as covered later, use is made of natural ventilation to control the air-change rate, regardless of the external conditions. This approach is not practical for industrial applications.

The term *natural ventilation* relates to the airflow in a building that is caused by three natural factors:

1. temperature differences (thermal density) between the inside and the outside of the building
2. wind forces around the building
3. a combination of (1) and (2)

Natural ventilation has the advantage that no power supply is required; hence, there are no fans and maintenance costs. It has the disadvantage that during the summer months, the temperature difference between the inside and outside is small, and with low wind forces, poor ventilation will result when it is most required. During the winter months, however, the temperature difference between indoors and outdoors is high, with corresponding strong wind forces, resulting in high ventilation and heat-loss rates.

It is not possible to design a system according to a given natural ventilation rate, as this will vary with weather conditions. The cold outdoor air has a greater density for a given volume than the warmer indoor air; it enters through low-level openings, displacing the warmer indoor air and creating air changes. The displaced warmer air escapes through high-level openings (see Fig. 9.20). Ventilation caused by this method is commonly called the *stack effect*.

The greater the distance between the low-level inlets and the high-level outlets, the greater the resulting air-change rate will be. The resulting airflow patterns in this arrangement will not ensure satisfactory air distribution in many industrial environments.

The science of building aerodynamics considers the influence of wind forces over buildings and the associated mechanics of fluids; these are complex in nature and are not considered here. It is sufficient to briefly consider Fig. 9.21, which shows how wind passing over a building produces a positive pressure on one side and a negative pressure on the other. It is this pressure difference that produces airflow through openings. The combined wind and stack effects vary with the seasons.

Fig. 9.21 considers the air movement in elevation; the aerodynamic flow in plan is shown in Fig. 9.22. It can be appreciated that adjacent structures to the simple block considered result in more complex flow patterns that flow over roof ridges and the geometric shape of the building in relation to the prevailing wind direction. Care has to be taken in the positioning of input or extract fans in relation to the wind direction, as it may have an effect on the fan performance.

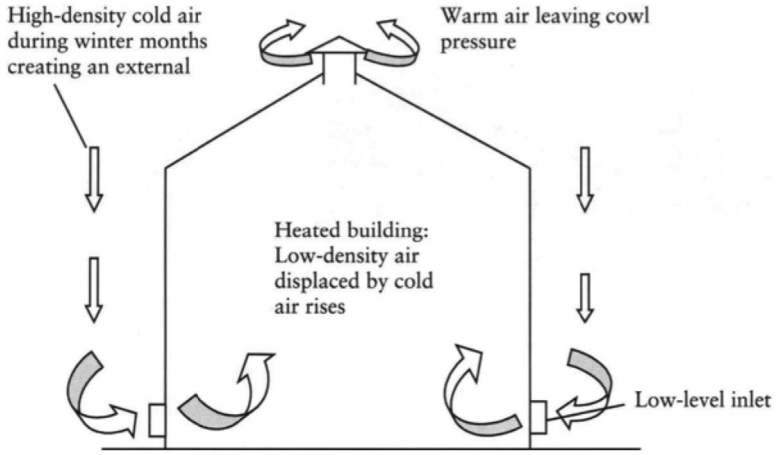


FIGURE 9.20 Stack effect.

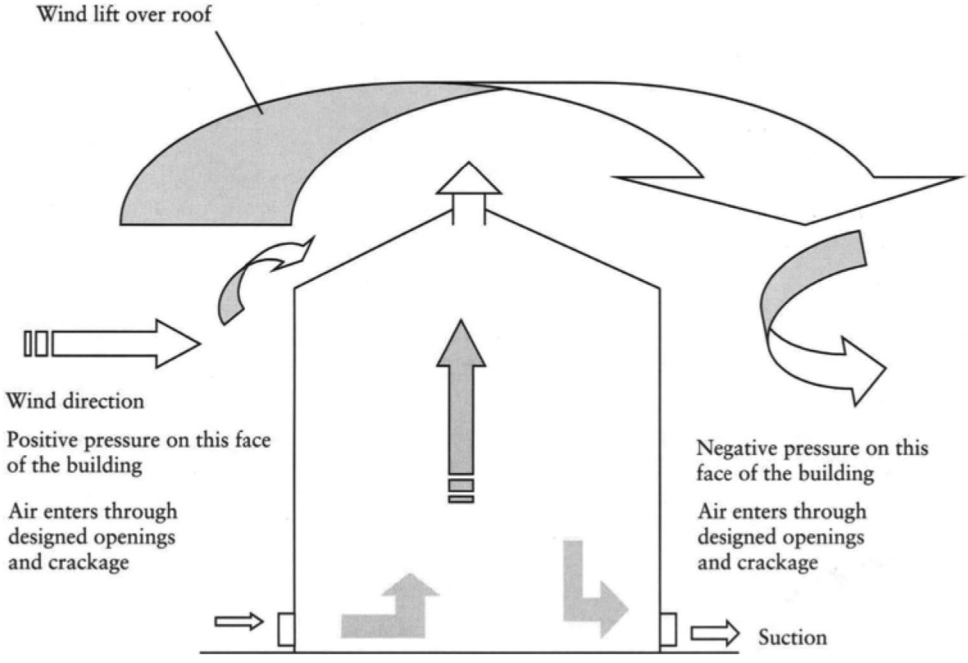


FIGURE 9.21 Wind forces around a building in elevation.

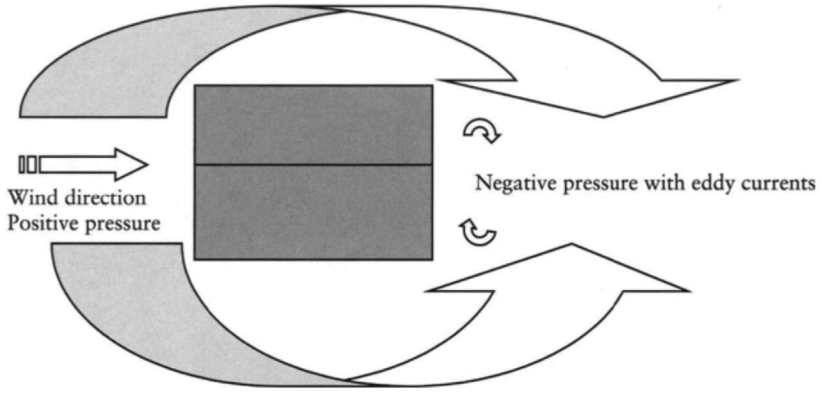


FIGURE 9.22 Wind forces around buildings in plan.

Mechanical extract–induced input

This method of ventilation uses extractor fans. A fan will create a negative pressure within the space. With this method a set flow rate can be achieved, as the fan will overcome the stack and wind effects; hence, the system is not at the mercy of the weather. This system causes the inside of the building to be held at a negative pressure, so air will be drawn in from outside or from surrounding spaces that are at a high pressure.

A space may need to be held at negative pressure, as in lavatories, kitchens, and process areas for example. In such cases, air will flow into the negative-pressure zone; hence, odors or toxic gases will not escape to other areas. The fan, with its running and maintenance costs, makes this system more expensive than the natural method.

Fig. 9.23 shows a typical arrangement of this method of ventilation. The resulting airflow patterns in this arrangement will not ensure satisfactory air distribution in many industrial environments.

Mechanical input–forced extract

If heating is provided, this is known as a *plenum system*. This is a ducted system, which may provide air to a space in one or more of the following conditions:

- untreated (no filter);
- tempered by means of a heater battery (heated to near room conditions); and
- warm, at a temperature high enough to take care of the fabric and ventilation losses from the building.

With the air being forced into the space the pressure is positive; hence, all leakages are outward.

More sophisticated applications of a pressurized system combined with a suitable extractor are found in hospital operating theaters. The positive pressure produced by sterile air entering the room ensures that all leakages are outward. This outward leakage ensures that contaminated air at a lower pressure in the surrounding rooms will not enter the space.

Fig. 9.24 shows a typical plenum system of mechanical input and forced extract. The outlets in Fig. 9.24 may be subdivided into two categories:

1. diffused air distribution
2. tangential air distribution

In the case of diffused air the associated characteristics are as follows:

- The jet has a high degree of entrainment.
- There are no circular air patterns in the occupied zone.
- Air movement is ideal in the working zone, both for thermal comfort and pollution control.
- Temperature gradients (stratification) are at a minimum.

The above are achieved by the use of:

- linear air outlets with highly inductive jets discharging directly downward or alternate jets discharging in opposite directions;
- twist outlets with adjustable twist vanes;
- specially designed air outlets; and
- whirl outlets, either radial or linear, that have geometrically designed aperture chambers.

The tangential method of air distribution has the following features:

- The room airflow patterns are circular.
- Stagnant pockets form between the circular room air patterns.
- The jets come into contact with surrounding surfaces.

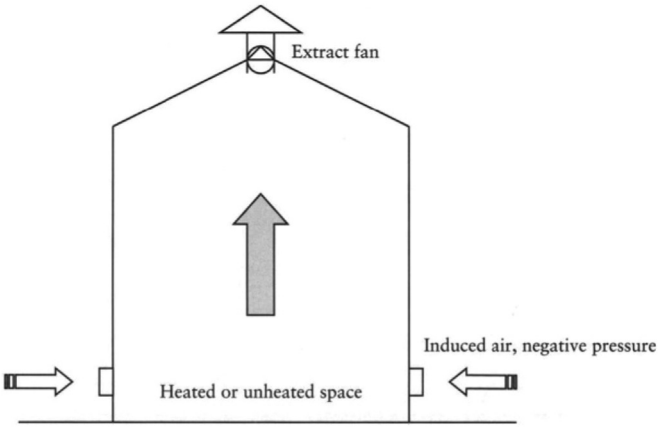


FIGURE 9.23 Mechanical extract–induced input ventilation.

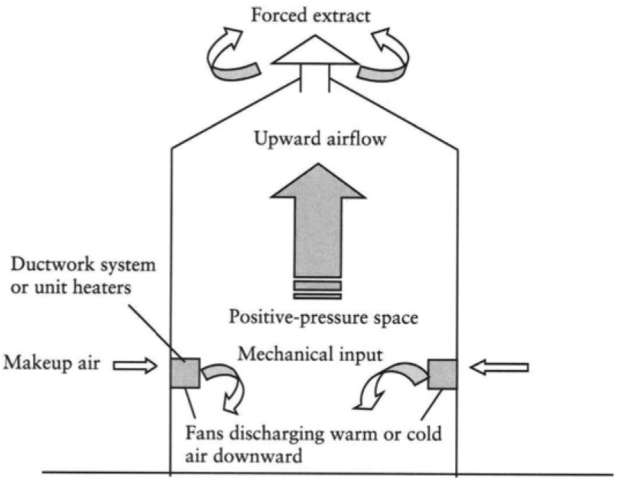


FIGURE 9.24 Plenum system.

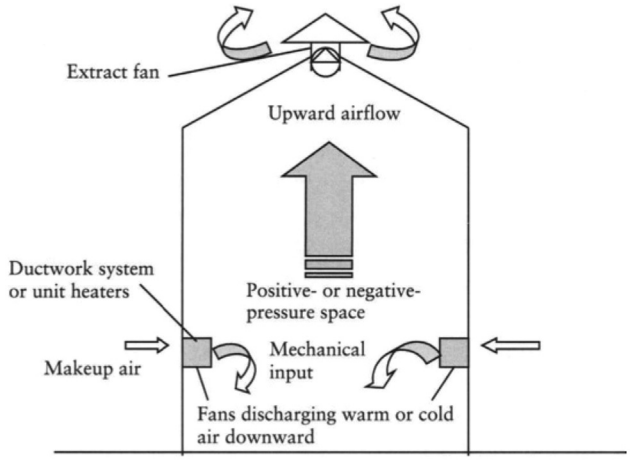


FIGURE 9.25 Mechanical input–mechanical extract.

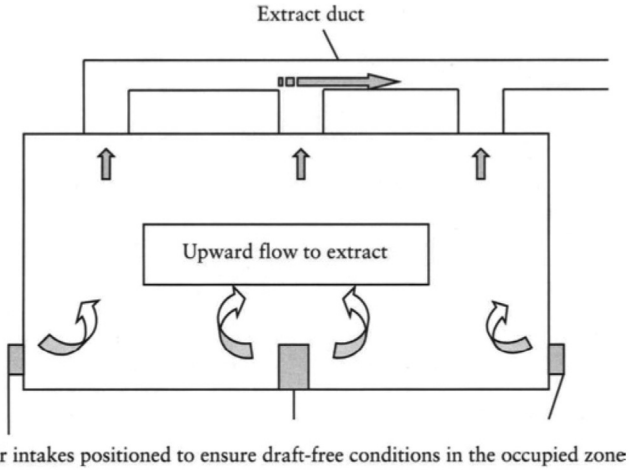


FIGURE 9.26 Upward ventilation, large hall.

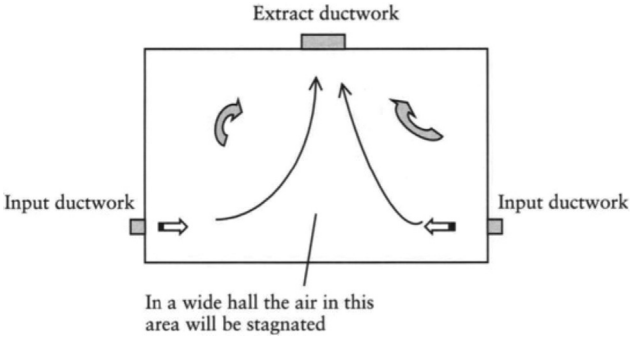


FIGURE 9.27 Upward ventilation, narrow room.

In the application of this method of distribution, full use is made of the Coanda effect (wall jet), in which the jet adheres to a surface due to the fact that no entrainment can occur from a solid surface.

The outlets may be square, rectangular, or circular without any air twist being produced.

All outlets must be positioned at such a height that they do not create drafts in occupied zones.

Mechanical input–mechanical extract

In this method of ventilation, powered extract and input systems are provided. The ratio of the supply to extract will control the pressure produced within the space.

If the input fans are handling the same amount of air as the extract fans, a neutral condition will result, with the disadvantages of the natural system.

Fig. 9.25 shows a typical layout of this type of system. From the foregoing statements the advantages of such a system are obvious.

This section has considered methods of ventilation. These should not be considered the same as methods of air distribution, which are now covered.

9.4.3.3 Methods of air distribution

When air is supplied to a space by means of a fan forcing it along the ductwork, it has to be distributed into the space without causing drafts, noise, or poor air distribution, which would result in stagnant conditions in the space. Many variations of the following air-distribution methods are possible.

Upward ventilation (displacement)

This arrangement is suited for warm air, as on leaving the grill, hot air will rise. If this air is too hot, it will rise rapidly, and the jet will not reach the middle of the room. Therefore the leaving velocity and the temperature from the grill are critical.

If the velocity is too high, the occupants will experience drafts, while if it is too low, the occupants in the middle of the room may not receive adequate fresh air. Hence, the arrangement shown in Fig. 9.26 is not suitable for wide spaces. This problem is overcome by using the layout shown in Fig. 9.27.

Downward systems

Air-conditioning systems make use of this method, as the cooler, dense air supplied from high level will drop to low level, picking up the space heat gains before extraction at low level.

It is essential that the entering air is no more than, say, 9°C cooler than the room air, for if it is, the cold air will drop, causing complaints from the occupants. In the case of air cooling of hot environments, the temperature difference can be greater.

Full use is made of ceiling diffusers, which ensure that the cold air spreads out over a wide area before

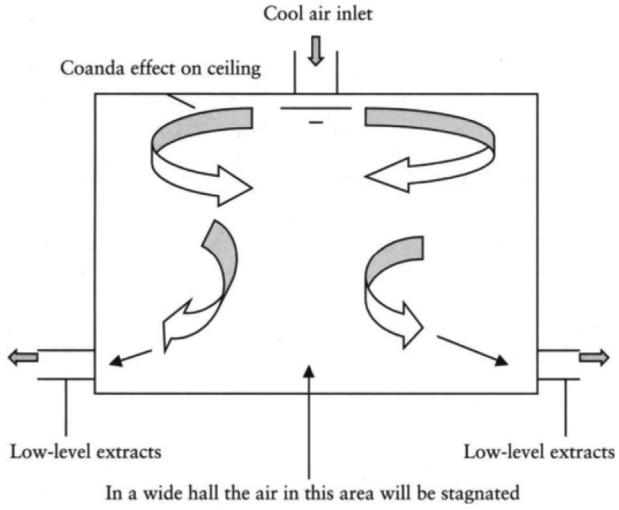


FIGURE 9.28 Downward system.

dropping. Perforated ceilings may be used, with the ceiling void being the plenum chamber. A typical downward system is shown in Fig. 9.28.

The arrangement shown in Fig. 9.28 is used in industrial halls or auditoriums. Clean rooms have complex systems using laminar flow to ensure that the room is fully ventilated.

Fig. 9.29 shows a typical arrangement of a laminar-flow clean room. The flow may be vertical through a perforated ceiling and floor.

Full use can be made in the industrial environment of long jet throws, which entrain room air.

It is essential that the velocity envelope in the occupied zone be in the range of 0.2–0.25 m/s to avoid drafts. However, in hot industrial environments, these velocities are frequently exceeded in order to provide adequate body cooling.

Mixed upward–downward system

The working of this system is shown in Fig. 9.30, from which it will be seen that good air mixing takes place. Care has to be taken to ensure that the high-level inlets and extract are positioned so that short circulating of the air does not occur.

The low-level extract is normally about 25% of the total extract; the actual value is selected to suit the design conditions.

9.4.3.4 Air-handling equations

Air mixing

As the outside air conditions vary from hour to hour, day to day, and month to month, to economize on the heating and cooling loads, the recirculated and fresh airstreams are mixed in varying proportions. If the air contains toxic or inflammable gases, vapors, or dusts, no direct mixing occurs, and a heat-recovery device is used.

In order to appreciate the mixing arrangement, consider the layout shown in Fig. 9.31.

In this instance, the following abbreviations are used:

q_m = mass flow rate of the gas
m_w = mass of water vapor present
h = specific enthalpy of the gas
θ = dry-bulb temperature

Consider any two gas streams A and B which, when combined, produce a mixed condition C.

For mass flow

Assuming no flow losses occur, the total mass of a gas introduced into and out of the system must be constant. Hence

$$q_{mA} + q_{mB} = q_{mC.}$$

For moisture content

Similarly, the total quantity of moisture in the system will be the same before and after mixing. However, the amount of moisture per unit mass of the gas will change: hence, in this case the equation must include mass.

$$q_{mA} \times m_{wA} + q_{mB} \times m_{wB} = q_{mC} m_{wC}$$

For enthalpy

Provided that no gain or loss of heat takes place during the mixing process, the total heat in the two airstreams before mixing must equal that of the combined airstream after mixing.

$$q_{mA} \times h_A + q_{mB} \times h_B = q_{mC} \times h_c$$

For temperature

When the dry-bulb temperature of a gas is altered, a positive or negative sensible heat transfer takes place, in terms of both the dry bulb and its associated moisture content.

Because the specific heat capacity of the water vapor is different from that of the dry air, the true dry-bulb mixed-stream air temperature can be determined only by means of a heat balance.

Heat gained or lost by stream A = Heat gained or lost by stream B

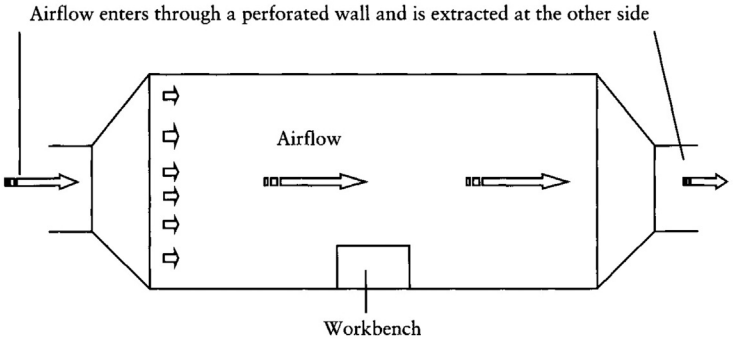


FIGURE 9.29 Laminar-flow room.

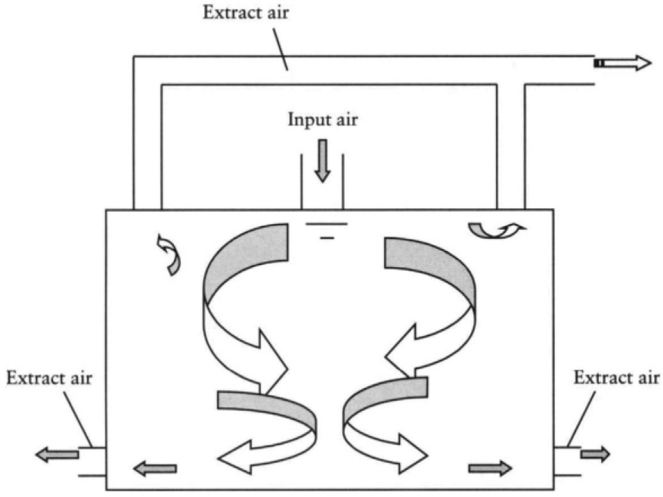


FIGURE 9.30 Mixed upward–downward system.

$$\begin{bmatrix}\text{Heat gained or lost in stream } A + \\ \text{heat lost or gained by associated moisture}\end{bmatrix} = \begin{bmatrix}\text{Heat lost or gained in stream } B + \\ \text{heat lost or gained by associated moisture}\end{bmatrix}$$

$$\begin{bmatrix} q_{mA} \times c_{pg}(\theta_A - \theta_C) + \\ q_{mA} \times m_{wA} x c_{pW}(\theta_A - \theta_C)\end{bmatrix} = \begin{bmatrix} q_{mB} \times c_{pg}(\theta_C - \theta_B) + \\ q_{mA} \times m_{wB} x c_{pW}(\theta_C \theta_B)\end{bmatrix}$$

$$q_{VA}(\theta_A - \theta_C)(c_{pg} + c_{pw} m_{wA}) = q_{VB}(\theta_C - \theta_B)(c_{pg} + c_{pw} m_{wB})$$

Total room air movement

It is essential to ensure that the air distribution in a space provides satisfactory air movement for the occupants' thermal comfort.

When a jet is released into a free space, it induces into its body the room air. The air leaving the grille is the primary air, while that entrained from the room into the body of the jet is the secondary air. The degree of entrainment that takes place is related to the velocity of the leaving jet. The combined airstream is known as the total air.

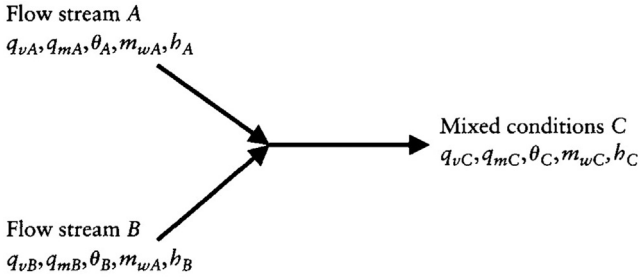


FIGURE 9.31 Mixing arrangement.

The induction is expressed by the momentum equation as

$$q_{mP} v_P + q_{ms} v_s = (q_{VP} + q_{VS})_{VT},$$

where q_m is the mass flow rate of the primary air (kg/s), v_p is the entering velocity of the primary air (m/s), q_{ms} is the mass flow rate of the secondary air (kg/s), v_s is the velocity of the secondary air (m/s), q_{VP} is the volume flow of the primary air (m^3/s), q_{Vs} is the volume flow of the total air (m^3/s), and v_T is the velocity of the total air.

The induction ratio (R) is the ratio of the total air to the primary air.

$$R = \frac{\text{Total air}}{\text{Primary air}} = \frac{\text{Primary air + secondary air}}{\text{Primary air}}$$

Since the throw of a jet is a function of the velocity, and since the rate of velocity decrease is dependent on the rate of induction that occurs, the quantity of air inducted into the discharge from an outlet is a direct function of the perimeter of the cross-section of the primary airstream.

In the case of two outlets, each of the same area, the outlet having the largest perimeter will induce the greatest amount of secondary air; hence, the throw will only be short. For a given air quantity and pressure discharged into a space, the minimum induction

and a single outlet of circular cross-section obtain the maximum throw, while the greatest induction and shortest throw occur with a single, long, narrow slot.

The spread of a jet is the angle of divergence of the airstream after it leaves the outlet. The outlet design may be for horizontal or vertical spread or both.

Example 3

The following example provides some indication of the effects of induction.

Primary and secondary air are mixed, respectively, at the rates of 0.5 m^3/s, the primary air velocity at outlet being 5.0 m/s, with the secondary air velocity assumed to be zero.

Determine the velocity and the area of the total airstream when complete mixing of the primary and secondary airstreams has taken place.

Solution. The area of the primary airstream before induction is

$$\frac{q_{V1}}{v_1} = \frac{0.5}{5.0} = 0.1\text{m}^2.$$

Substituting in the momentum equation

$$(0.5 \times 5) + (0.5 \times 0) = (0.5 + 0.5)v_3.$$

Hence, $v_i = 2.5$ m/s.

The area of the total airstream is

$$\frac{q_{m1} + q_{m2}}{v_3} = \frac{0.5 + 0.5}{2.5} = 0.4\ \text{m}^2.$$

In basic ventilation design, if a given air change is required, it is a simple matter to determine the capacity of the fan required from

$$q_v = \frac{\text{Volume of space } (m^3) \times \text{number of air changes per hour}}{3600}.$$

Depending on the application, the air-change rate may range from 0.5 to 100 air changes per hour. It must be remembered, however, that adequate provision must be made for the makeup air to enter the space without creating discomfort or other problems.

In the case of sensible and latent heat the cooling procedure becomes a little more complicated.

In the case of summer cooling the heat balance is

Heat gain to space =
Heat absorbed by chilled air flowing through the room,

or

Sensible heat gain (kW) =
Air mass flow × gain in specific enthalpy,

or

$$\phi_s(\text{kW}) = q_m(\text{kg/s}) \times (h_r - h_s)(\text{kJ/kg}).$$

The sensible heat gain is determined by taking into account all the heat gains in a space, such as solar, fabric, infiltration, machines, processes, occupancy, and lighting.

The specific enthalpy (h_s) of the room-air design condition is obtained from tables of the property of air or the psychrometric chart.

For deciding to what dry-bulb temperature (θ_s) the incoming supply air has to be cooled, the following approach is applied.

Sensible heat ϕ_s(kW) = Air mass flow rate $q_m(\text{kg/s}) \times$ specific heat capacity $(\text{kJ/(kg } K)) \times$ air temperature rise

$$\Phi_s(\text{kW}) = q_m \times c_p \times (\theta_r - \theta_s)$$

$$q_m = q_v \times \rho$$

$$\Phi_s(\text{kW; SH}) = q_v \times \rho \times c_p \times (\theta_r - \theta_s)$$

As the air density depends on both the temperature and moisture content of the air, it is necessary to apply the general gas equation:

$$p_a V = mRT$$

where p_a is the absolute pressure of the gas (Pa), V is the gas volume (m^3), m is the mass of gas (kg), R is the gas constant (287.1 J/(kg K)), and T is the absolute temperature of the gas (K).

$$\rho = \frac{m}{V} = \frac{p}{RT},$$

assuming that the pressure and moisture content are constant in the flow along the air supply duct.

$$\rho_1 \approx \frac{1}{T}$$

If the standard air density is ρ_1 and the supply air density is ρ_2, then

$$\rho_2 \approx \frac{1}{T_2}.$$

The above gives

$$\frac{\rho_2}{\rho_1} = \frac{T_1}{T_2}$$

and

$$\rho_2 = \rho_1 \frac{T_1}{T_2}.$$

Correcting the air density for other air supply temperatures

$$\rho_2 = 1.1906 \times \frac{273 + 20}{273 + \theta_s}\text{kg/m}^3$$

where 1.1906 is the density of air at 20°C and 50% saturated (specific volume 0.8399 m^3/kg).

Substituting this into the heat balance equation for sensible heat only gives

$$\Phi_s(\text{kW}) = q_v \times \rho_2 \times c_p \times (\theta_r + \theta_s)$$
$$\Phi_s(\text{kW}) = q_v \times 1.1906 \times \frac{273+20}{273+\theta_s} \times 1.0048 \times 1.0048 \times (\theta_r - \theta_s)$$
$$q_v \times 351 \times -\frac{\theta_r - \theta_s}{273+\theta_s},$$

where 1.0048 is the specific heat capacity of air (kJ/(kg K)).

On rearranging to obtain q_v

$$q_v = \frac{\Phi_s(\text{kW}) \times (273+\theta)}{351(\theta_r - \theta_s)}.$$

Example 4

The sensible gain in an industrial complex is 1000 kW, and the space temperature has to be controlled at 20°C.

If the supply air leaves the cooling plant at 11°C and gains 1°C in the ductwork before entering the space, determine the air supply volume that is required.

Solution

$$\Phi_s = 1000 \text{ kW}; \quad \theta_r = 20^\circ C; \quad \theta_s = 11 + 1^\circ\text{C gain} = 12^\circ\text{C}$$

so $\Delta\theta = 20 - 12 = 8^\circ$C.

Hence

$$q_v = \frac{1000 \text{ kW} \times (273+12)}{351 \times (20-12)} = 101.5 \text{ m}^3/\text{s}.$$

Answer: 101.5 m^3/s.

Question

A workshop is 30 m × 15 m × 4 m and has to be maintained at 20°C with six air changes per hour with a supply air temperature of 14°C.

Determine the maximum cooling load that can be met.

Solution

$$q_v = \frac{N \text{ air changes per hour} \times \text{volume}}{3600}$$
$$\frac{6 \times 30 \times 15 \times 4}{3600} = 3.0 \text{ m/s}$$
$$\Phi_s = \frac{q_v \times 351(\theta_r + \theta_s)}{273+\theta_s}$$

Substituting known values

$$\frac{3.0 \times 351(20-14)}{273+14} = 22 \text{ kW}.$$

Answer: 22 kW.

In order to determine the summer supply air temperature for a given volume of air necessary to remove a given sensible heat gain, the following equation is used

$$q_v \frac{\Phi_s(\text{kW}) \times (273+\theta_s)}{351(\theta_r - \theta_s)}$$
$$q_v \times 351(\theta_r - \theta_s) = \Phi_s(273+\theta_s)$$
$$q_v \times 351\theta_r - \Phi_s \times 351\theta_s = \Phi_s \times 273 + \Phi_s \times \theta_s$$
$$q_v \times 351\theta_r - \Phi_s \times 273 = q_v \times 351\theta_s + \Phi_s \times \theta_s$$
$$q_v \times 351\theta_r - \Phi_s \times 273 = \theta_s(q_v \times 351 + \Phi_s)$$

Hence

$$\theta_s = \frac{q_v \times 351\theta_r - \Phi_s \times 273}{q_v \times 351 + \Phi_s}$$

Example 5

A process plant occupies a space of 30 m × 20 m × 4 m. The area has to be maintained at 21°C with 10 air changes per hour of chilled air. The refrigeration plant is capable of 35 kW of sensible cooling. Determine the required supply air temperature.

Solution

$$q_v = \frac{30 \times 20 \times 4 \times 10}{3600} = 6.66 \text{ m}^3/\text{s}$$
$$\theta_s = \frac{q_v \times 351\theta_r - \Phi_s \times 273}{q_v \times 351 + \Phi_s},$$

substituting the known values.

$$\theta_s \frac{6.66_v \times 351 \times 21 - 35 \times 273}{6.66 \times 351 + 35} = 16.6^\circ\text{C}$$

Answer: 16.6°C.

The above examples have dealt with cooling. In the case of heating applications, the supply air temperature must be greater than the room air, hence $\theta_s > \theta_r$.

Example 6

A workshop has a fabric and ventilation sensible heat loss of 30 kW, and the process requires that the room be maintained at 22°C dry bulb. Determine the supply air volume required at 34°C in order to maintain the space at design conditions.

Solution

$$q_v = \frac{\Phi_s \times (273+\theta_s)}{351(\theta_s \sim \theta_r)}$$
$$q_v = \frac{30 \times (273+34)}{351(34-22)}$$
$$= 2.187 \text{ m}^3/\text{s}$$

Answer: 2.187 m^3/s

Example 7

A plenum system provides two air changes per hour of makeup air at 33°C to a process area of 20 m × 15 m × 3.5 m in order to maintain the space temperature at 21°C.

Determine the maximum heat loss the above parameters are capable of meeting.

Solution

$$q_v = \frac{2 \times 20 \times 15 \times 3.5}{3600} = 0.583 \text{ m}^3/\text{s}$$

$$q_v = \frac{\Phi_s \times (273 + \theta_s)}{351(\theta_s - \theta_r)}$$

$$\Phi_s = \frac{q_v \times 351(\theta_s - \theta_r)}{273 + \theta_s}$$

$$= \frac{0.583 \times 351 \times (33 - 21)}{273 + 33}$$

Answer: 8.02 kW.

The supply air temperature in the case of a heating system is determined from

$$\theta_s = \frac{351 \times q_v \theta_r + 273 \times \varphi_s}{351 q_v - \theta_s}.$$

Question

Determine the air temperature that is necessary to maintain a volume of 1200 m^3 with six air changes per hour at 20°C, if the total heat loss is 45 kW.

Solution

$$\frac{1200 \times 6}{3600} = 2.0 \text{ m}^2/\text{s}$$

$$\theta_s = \frac{351 \times 2 \times 20 + 273 \times 45}{351 \times 2 - 45} = 40.07^\circ\text{C}$$

Answer: 40.07°C.

When calculating for heater and cooler batteries, it is the mass flow rate q_m that is required, not the volume flow rate q_v as used above. This is determined by using the appropriate specific volume or density of the air at the actual design temperature and moisture content.

Moisture content

The water vapor present in the air exists at its own partial pressure. The molecular mass of water vapor is 18.015 kg/kmol. This is less than dry air, which is 29 kg/kmol.

From the above it will be appreciated that the greater the concentration in the air, the less the mixture weighs.

As the moisture content in a dry air mixture increases, the specific volume increases and the density reduces. The specific volume is v (m^3/kg), and the density is ρ (kg/m^3).

The relationship between the two parameters is simply $\rho = 1/v$.

The base for air density is taken as 101.325 kPa at 0°C.

Using the general gas law, the standard density is

$$p_a v = mRT,$$

hence density

$$\rho = \frac{m}{v} = \frac{p_a}{RT}.$$

Taking the vapor pressure of the air supply from hygrometric tables, the value of specific density can be found.

$$v = \frac{287.1 \times (273 + \theta_s)}{101.325 - p_s} \text{m}^3/\text{kg}$$

Example 8

Air is supplied to a space at the rate of 2.0 m^3/s at 12°C and 70% saturated, at which condition the vapor pressure is 0.9852 kPa. Determine the density and the mass flow rate.

Solution

$$v = \frac{287.1 \times (273 + 12)}{101325 - 0.9852 \times 1000} \text{m}^3/\text{kg} = 0.8154 \text{ m}^3/\text{kg}$$

The mass flow is then

$$q_m = \frac{q_v}{v} = \frac{2.0}{0.8154} = 2.452 \text{ kg/s}.$$

Answer: 2.452 kg/s.

9.5 Fans

9.5.1 General

The fan is the heart of a ventilation system. The industrial engineer requires a comprehensive working knowledge of the subject of fan engineering for the following reasons:

1. The designer must fully appreciate the correct fan selection, by ensuring that its duty meets the requirements of the task for which it is intended.
2. In many cases, an existing fan system has to be adapted or modified to meet new demand requirements.
3. The owning and operating costs must be fully considered.

A fan is a rotodynamic device and is the driving part of all mechanical ventilating systems. The energy of rotation applied to the fan shaft is converted into a pressure difference, causing the air, gas, or particulate matter to flow through the ductwork or discharge into a free space.

To be classed as a fan, the work per unit mass on the gas must be less than 25 kJ/kg; above this value it is called a turbo compressor.

Hence, the fan pressure must not exceed the standard air density of $1.2\,kg/m^3 \times 25\,kJ/kg$, giving 30 kPa, with the pressure ratio not exceeding 1.30, taking the atmospheric pressure as 100 kPa.

ISO gives the classifications for fans shown in Table 9.10.

9.5.1.1 *Fan types*

Fans can be divided into four general categories: propeller, axial, centrifugal, and special purpose. These are defined according to the direction of gas flow.

Propeller

This type consists of a propeller or disk-type wheel within a mounting ring panel or cage. The wheel or housing is constructed from either sheet metal, cast aluminum, plastic, or plastic-coated material. It may be a direct drive with the wheel on the motor shaft or belt driven.

Advantages and typical uses:

- Wide range of volumes
- A minimum operating cost per m^3/s
- Minimum space and weight per m^3/s
- Blast or man cooling for hot processes

TABLE 9.10 Category according to work per unit mass.

Description	Code	Work per unit mass (kJ/kg)	Maximum fan pressure (kPa)	Class
Low pressure	LB	0–0.6	0–0.7	0
		0.6–0.833	0.7–1.0	1
		0.833–1.333	1.0–1.6	2
		1.333–1.667	1.6–2.0	3
Medium pressure	MP	1.667–3.00	3.0–3.6	4
		3.0–5.25	3.6–6.3	5
		5.25–8.33	6.3–10	6
High pressure	HA	8.33–13.33	10.0–16.0	7
		13.33–18.67	16.0–22.4	8
		18.67–25.00	22.4–30	9
Turbo compressor	–	25 +	30 +	–

- Dilution ventilation for toxic and odor removal

Disadvantages/limitations:

- Limited to resistances of 250 kPa
- Sound-level problems with high speeds
- Not suited for corrosive or abrasive applications, bearings to be protected
- Direct-drive fans should not be used in the area in which explosive or flammable gases or vapors are handled by the fan
- Operating temperature limitations

Axial fan

A tube axial fan (see Fig. 9.32) is essentially a propeller fan located in a short cylinder housing, the gas flowing in an axial direction. A vane axial fan incorporates specially designed vanes, which are positioned either upstream or downstream of the fan.

The axial fan consists of an impeller fitted with airfoil blades mounted on a rotating hub. The hub is positioned in a cylindrical casing in line with the gas flow direction. If safe gases are being handled, the motor is located in the airflow. If, however, explosive, abrasive, flammable, or corrosive gases are conveyed, a bifurcated fan is used, with the motor positioned outside the gas stream. A motor located outside the casing allows the fan to be belt driven, providing easy speed changes if necessary. Due to advances in motor electronic speed control, the use of belts for speed control is on the decline.

An airfoil fan may have an efficiency as high as 80%. It has the advantages of being compact and able to fit in line with the ductwork. Its disadvantage is that it may

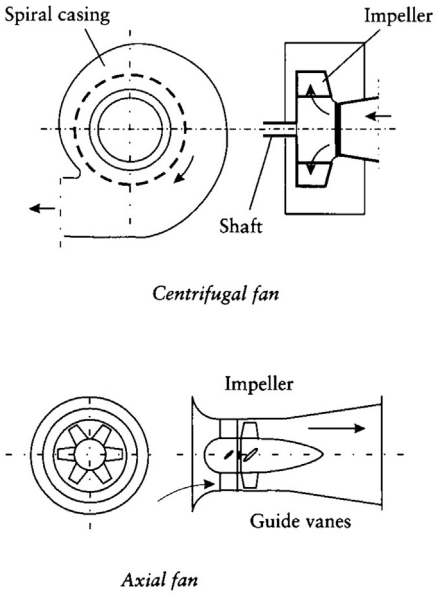


FIGURE 9.32 Centrifugal and axial fans.[19]

not be capable of developing the high pressures required for many industrial ventilation applications.

The purpose of the blades is to reduce the degree of flow spiraling and to convert some of the velocity component into useful static pressure.

Vane axial fans develop a greater static pressure than tube axial fans. They are constructed from a variety of materials, depending on the application. They may use either a direct drive or a belt drive. More expensive models are fitted with adjustable pitch blades, allowing a direct-driven fan to cover the same capacities as belt-driven fans of the same diameter.

Advantages and typical uses:

- Operates from low-to-high volume flow rates.
- The actual pressure range of some vane axials is similar to high-efficiency backward-curved centrifugal fans. By fitting the fans in series the operating pressure can be increased.
- Compact, low-space environment, and low weight per unit volume handled make it second only to the propeller fan.
- Applications include comfort, dilution, man cooling, and paint-spray-booth extract.

Disadvantages/limitations:

- Inherently higher sound levels than most efficient centrifugal fans for the same duty
- Unsuitable in abrasive or corrosive atmospheres
- Problems in protecting bearings
- Unsuitable for flammable and explosive gases and vapors unless a bifurcated fan is used
- Fan curve problems with damper closing

Centrifugal fan

This is the most common type of fan encountered in industrial ventilation systems. These fans are similar to a water wheel, with blades mounted on backplates. The impeller is positioned in a volute or scroll casing. The air enters with the line of the driving shaft through the eye and discharges at 90 degrees to the entering air.

The centrifugal fan may have one of four impeller designs:

1. airfoil
2. backward inclined-backward curved
3. radial (paddle)
4. forward curved

Airfoil, backward curved

These low-cost fans are of simple construction and have static efficiencies up to 80%.

This fan design is capable of the highest efficiency with the lowest sound level of all the centrifugal fans for a given duty, particularly those with airfoil-shaped double blades.

They are normally fitted with 10–12 blades, the blade tips inclined away from the angle of rotation. They have the advantage of being very efficient, with fan static efficiencies in excess of 80% for airfoil blades. This type of fan has a nonoverloading characteristic curve.

Advantages and typical uses:

- Handles moderate-to-high airflow rates
- Static pressure up to the 7.5 kPa range
- Highest efficiency of any fan type
- Lowest noise level of any fan type for a given duty
- Self-limiting power characteristics, nonoverloading characteristic
- Variable air volume systems

Disadvantages/limitations:

- For equal pressure and volume, axial fans' weight and size are greater for the backward-curved fan.
- Unsuitable for high dust loading, due to particulate buildup on the impeller causing imbalance and vibration.
- Wheels are difficult to clean and paint.

Radial blade, straight paddle blade

This fan group, sometimes referred to as industrial exhaust fans' group, is characterized by its simple, rugged construction.

They may or may not have flat, radial blades. The wheel configuration may be from simple paddle blades to the semiopen and enclosed types. They may be belt or direct driven.

Advantages and typical uses:

- Ease of maintenance and cleaning
- Ease of repair due to simple construction
- Can handle any type of gas or dust
- Used for pneumatic conveying
- Suitable for high-temperature operation
- Suitable for corrosive and abrasive material if correctly designed

Disadvantages/limitations:

- Lowest efficiency of all centrifugal fans
- Highest sound level of all centrifugal fans for a given duty
- Shaft power increases as the fan approaches the maximum volume

Forward-curved blade, centrifugal

These are made up of a large number of wide, shallow blades with a very large inlet area relative to the wheel diameter. For equal duty the speeds are lower than other centrifugal fans. They are sometimes called multivane; the operating efficiencies are in the 65%–75% range. They consist of a large number of relatively small blades mounted on the impeller. The blade tips are inclined toward the direction of rotation. The actual flow

rate can be 2.5 times as high as the same size backward-curved fan.

Advantages and typical uses:

- Ideal for any volume at low to medium static pressures.
- Due to low speeds they are quiet in operation.
- The space and weight requirements are about the same as backward-inclined blade fans.

Disadvantages/limitations:

- Lower efficiency than backward-curved fan.
- Greater space requirement for a given duty than an axial flow fan.
- Unsuitable for high dust loading.
- Shaft power increases as fan approaches maximum volume, unlike the backward-curved fan, where it decreases.

The types of fans are given in Fig. 9.32.

As the gas density handled by a fan is relatively constant, the pressure increase is small, and the pressure ratio is less than 1.1.

9.5.2 Centrifugal fan

In this type of fan the gas flows initially in an axial direction toward the impeller. After this, in a part of the impeller blade, the gas flow becomes radial.

To force the air to flow through the impeller blades of a centrifugal fan, a tangential force is needed. According to the momentum law this force is

$$F_t = \frac{d}{dt}(mc_{1t}) = q_m(c_{2t} - c_{1t}). \tag{9.59}$$

where c_{1t} is the tangential component of the air velocity at the inlet to the impeller blades, and c_{2t} is the same at the exit. q_m is the mass flow of air through the impeller.

Mechanical power is generally equal to force times velocity, so we get from Eq. (9.59)

$$P_u = q_m(u_2c_{2t} - u_1c_{1t}). \tag{9.60}$$

where u_1 is the velocity of impeller at the inlet to the blades, and u_2 is the velocity of impeller at the outer edge.

Generally the velocity u can be calculated from

$$u = \pi Dn \tag{9.61}$$

where D is the diameter at the point in consideration and n is the speed of revolution of the impeller.

Consider the thermodynamic process in the fan (Fig. 9.33). As the fan is a stationary flow system, consideration is directed to the total enthalpy change. As the suction openings are often at the same, or almost the same level, the potential energy change can be neglected.

Two cases are studied:

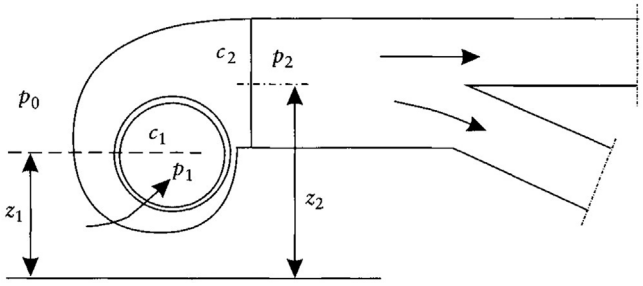


FIGURE 9.33 Fan as a stationary flow system.

- adiabatic reversible (isentropic) process and
- adiabatic irreversible process in which entropy is generated.

In both the cases the mass flow, mechanical work, and density are constant. The flow is denoted by subscripts 1 before the fan and 2 after the fan. If there is no entropy generation, subscript 2 is replaced by subscript 2s.

The mass conservation equation gives

$$q_m = \rho c_1 A_1 = \rho c_2 A_2 = \rho c_{2s} A_2, \tag{9.62}$$

where A_1 is the cross-sectional area of the fan entrance and A_2 is the cross-sectional area of the fan exit.

Considering the air density to be constant yields $c_2 = c_{2s}$, and using

$$Tds = dh - vdp, \tag{9.63}$$

the total enthalpy difference for the isentropic case can be written as

$$\begin{aligned} \Delta h_{\text{tor},s} &= h_{2s} - h_1 + \frac{1}{2}c_2^2 - \frac{1}{2}c_1^2 \\ &= v(p_{2s} - p_1) + \frac{1}{2}(c_2^2 - c_1^2) \\ &= \frac{1}{\rho}(p_{2s} - p_1) + \frac{1}{2}(c_2^2 - c_1^2). \end{aligned} \tag{9.64}$$

In an adiabatic process, an entropy change occurs, giving

$$\int_1^2 T\,ds > 0. \tag{9.65}$$

Thus

$$\begin{aligned} \Delta h_{\text{tot}} &= h_2 - h_1 + \frac{1}{2}c_2^2 - \frac{1}{2}c_1^2 \\ &= \frac{1}{\rho}(p_2 - p_1) + \int_1^2 Tds + \frac{1}{2}(c_2^2 - c_1^2) \\ &= \frac{1}{\rho}(p_2 - p_1) + \int_1^2 Tds + \frac{1}{2}(c_2^2 - c_1^2). \end{aligned} \tag{9.66}$$

As the mass flow and power requirements are the same in both cases, $P = q_m\ \Delta h_{\mathrm{tot},s} = q_m\ \Delta h_{\mathrm{tot}}$ or $\Delta h_{\mathrm{tot},s} = \Delta h_{\mathrm{tot}}$.

Comparing Eqs. (9.64) and (9.66) and relating to Eq. (9.65), the following is obtained:

$$p_{2s} > p_2 \tag{9.67}$$

In the isentropic case the static pressure increases more than for the irreversible case, with the fan operating under the same power input and mass flow.

The total isentropic enthalpy difference can be expressed in terms of the total isentropic pressure difference by

$$\Delta p_{\mathrm{tot},s} = \rho \Delta h_{\mathrm{tot},s} = \Delta p_s + \frac{1}{2}\rho\left(c_2^2 - c_1^2\right). \tag{9.68}$$

Denoting the total pressure by

$$p_{\mathrm{tot}} = p + \frac{1}{2}\rho c^2 \tag{9.69}$$

gives the total pressure difference as

$$\Delta p_{\mathrm{tot}} = \Delta p + \frac{1}{2}\rho(\Delta c^2). \tag{9.70}$$

In Eq. (9.67), p is the *static pressure*, and $1/2\rho c^2$ is the *dynamic pressure*.

Using Eq. (9.65) gives

$$\Delta p_{\mathrm{tot},s} > \Delta p_{\mathrm{tot}} \tag{9.71}$$

for a fan in which the opening suction pressure, mass flow, and power are the same.

Eqs. (9.60), (9.68), and (9.70) give

$$\text{Isentropic case: } u_2 c_{2u} = \frac{1}{\rho}\Delta p_{\mathrm{tot}}. \tag{9.72}$$

$$\text{Adiabatic irreversible case: } u_2 c_{2u} - u_1 c_{1u} > \frac{1}{\rho}\Delta p_{\mathrm{tot}}. \tag{9.73}$$

The isentropic or hydraulic efficiency of the impeller is

$$\eta_s = \frac{\Delta p_{\mathrm{tot}}}{\Delta p_{\mathrm{tot},s}} < 1. \tag{9.74}$$

Using Eqs. (9.60), (9.72), and (9.74) gives

$$P_u = q_m(u_2 c_{2u} - u_1 c_{1u}) = \frac{q_m}{\rho}\frac{\Delta p_{\mathrm{tot}}}{\eta_s} \tag{9.75}$$

where $\eta_s = 1$ if the flow through the impeller is isentropic.

For isentropic or nonisentropic flow, the fan shaft power is greater than the power required for the flow through the impeller.

A proportion of the shaft power is used to overcome the bearing friction. This is allowed by using the mechanical efficiency. Thus the required axial power is

$$P_a = \frac{P_u}{\eta_m} = \frac{q_v \Delta p_{\mathrm{tot}}}{\eta_s \eta_m} = \frac{q_v \Delta p_{\mathrm{tot}}}{\eta}, \tag{9.76}$$

where η is the total efficiency of the fan and $qv = qm/\rho$ is the air volume flow through the fan.

By using the vector equation

$$\begin{aligned} w^2 &= \overline{w}\cdot\overline{w} = (\overline{c} - \overline{u})\cdot(\overline{c} - \overline{u}) = c^2 + u^2 - 2\overline{u}\cdot\overline{c} \\ &= c^2 + u^2 - 2uc_2 \end{aligned} \tag{9.77}$$

the velocity expression of Eq. (9.73) can be written as

$$\begin{aligned} u_2 c_{2u} - u_1 c_{1u} &= \frac{1}{2}\left(c_2^2 - c_1^2\right) + \frac{1}{2}\left(u_2^2 - u_1^2\right) + \frac{1}{2}\left(w_1^2 - w_2^2\right) \\ &= \frac{1}{\rho}\left(p_{2s} - p_1\right) + \frac{1}{2}\left(c_2^2 - c_1^2\right), \end{aligned} \tag{9.78}$$

from which

$$\Delta p_s = p_{2s} - p_1 = \frac{1}{2}\rho\left(u_2^2 - u_1^2\right) + \frac{1}{2}\rho\left(w_1^2 - w_2^2\right). \tag{9.79}$$

The centrifugal force produced increases the impeller static pressure [the first term of the right-hand side of Eq. (9.79)] and reduces the relative velocity w. To increase the static pressure by a change in the relative velocity, the following relationship is necessary:

$$w_1 > w_2.$$

The mass flow through the blade passage is constant; hence, $w_1 > w_2$, and the flow across the surface is greater for the leaving edge than the incoming edge. The cross-sectional area of the passage increases in the direction of flow.

Eq. (9.79) shows that the static pressure increases with an increase in the tangential speed and the distance between the incoming and leaving edges of the impeller blade. This will not influence the nature of the flow process in the impeller.

Up to this stage, only the fan characteristics have been considered, without investigating the influence of different impeller blade shapes. Consider the flow at the edge of the impeller blade. Normally, for cost reasons, leading devices are not installed in front of the fan propeller, resulting in radial gas flow into the propeller, with the tangential velocity component $c_{1u} = 0$.

The velocity triangles of the incoming and leaving flow at the blade edges of a centrifugal fan are shown in Fig. 9.34. The flow to the blade is in a radial direction.

From Fig. 9.34, it can be seen that at the incoming edge of a backward-curved blade, the relative velocity w_1 is in the blade direction when $c_{1u} = 0$.

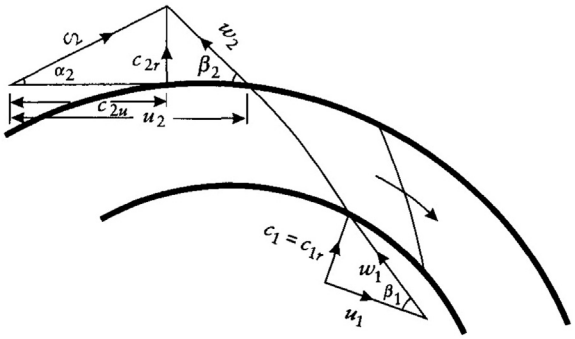


FIGURE 9.34 Velocity triangles at the blade edges of a backward-blade centrifugal fan.[20]

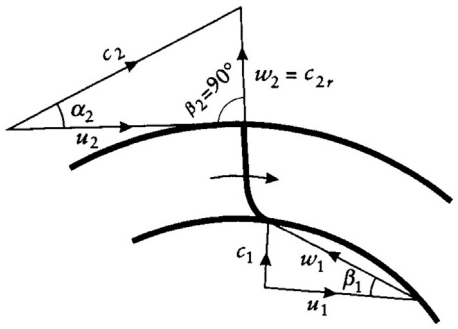


FIGURE 9.35 Straight-blade centrifugal fan impeller. The blade is backward curved at the entrance.[20]

The blade leaving edge may be

1. backward curved ($\beta_2 < 90$ degrees), Fig. 9.34;
2. straight ($\beta_2 = 90$ degrees), Fig. 9.35; or
3. forward curved ($\beta_2 > 90$ degrees), Fig. 9.36.

The *reaction ratio* of the fan is defined by

$$r = \frac{\Delta p}{\Delta p_{\text{tot}}}, \tag{9.80}$$

by which the static pressure increase is related to the total pressure increase. The reaction ratio can be determined for different blade shapes for the isentropic case as

$$r_s = \frac{\Delta p_s}{\Delta p_{\text{tot},s}}. \tag{9.81}$$

When $c_{1u} = 0$ and $\eta_s = 1$, Eq. (9.75) gives

$$\Delta p_{\text{tot},s} = \rho u_2 c_{2u}. \tag{9.82}$$

Using Eq. (9.79) gives

$$r_s = \frac{u_2^2 - u_1^2 + w_1^2 - w_2^2}{2u_2 c_{2u}}. \tag{9.83}$$

The radial velocity is approximately constant; thus

$$c_1 = c_{1r} = c_{2r}. \tag{9.84}$$

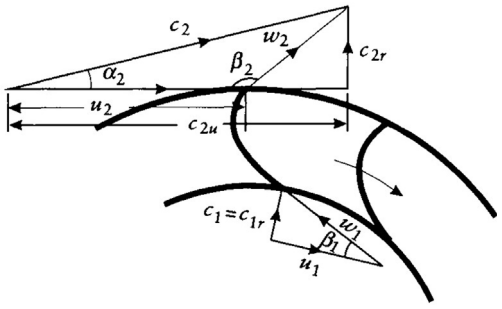


FIGURE 9.36 Centrifugal fan impeller, where the blade is forward curved at the leaving edge and backward curved at incoming edge.[20]

At the entrance edge of the blade the velocity triangle is a right triangle and $c_{1u} = 0$, so Eq. (9.77) gives

$$c_1^2 = w_1^2 - u_1^2. \tag{9.85}$$

Apply Eq. (9.77) to solve for the reaction ratio in isentropic flow

$$\begin{aligned} r_s &= \frac{u_2^2 - w_2^2 + c_1^2}{2u_2 c_{2u}} = \frac{2u_2 c_{2u} - c_2^2 + c_{2r}^2}{2u_2 c_{2u}} \\ &= 2u_2 c_{2u} - c_{2u}^2 = 1 - \frac{1}{2} \cdot \frac{c_{2u}}{u_2}, \end{aligned} \tag{9.86}$$

in which the result $c_2^2 = c_{2r}^2 - c_{2u}^2$ was used.

For backward-curved blade $\beta_2 < 90$ degrees from Fig. 9.34, it is seen that $c_{2u}/u_2 < 1$. The reaction ratio must be less than 1. In straight-curved blade $\beta_2 = 90$ degrees of Fig. 9.35, $c_{2u}/u_2 = 1$ and the reaction ratio is $r_s = 1/2$. In forward-curved blade $\beta_2 > 90$ degrees of Fig. 9.36, $c_{2u}/u_2 > 1$, and the reaction ratio is always less than 1/2.

The reaction ratio of a straight-blade impeller is poor when compared with a backward-curved impeller. The radial (paddle) fan has the advantage that when the air contains much dust, the straight blades remain cleaner than curved blades.

The reaction ratio of forward-curved impeller blades is the lowest. This type is used when a small static pressure increase is sufficient to transport the air. An example of such type is the *drum fan*, with short blades. A large suction opening is required, which allows large volumes of air to be handled. With backward-curved blades, a better reaction ratio is achieved, but with a reduction in total pressure increase. This is because u_2 is only slightly larger than u_1, and for that reason, c_{2u} will be small. By using forward-curved blades, c_{2u} increases, resulting in a larger total pressure differential.

The previous discussion assumes that the gas flows parallel to the blades. This is not the case, as the propeller rotation causes the air to rotate between the blades, against the impeller rotational direction at the same angular velocity. Due to this, the velocity c_{2u} decreases to c'_{2u}; hence

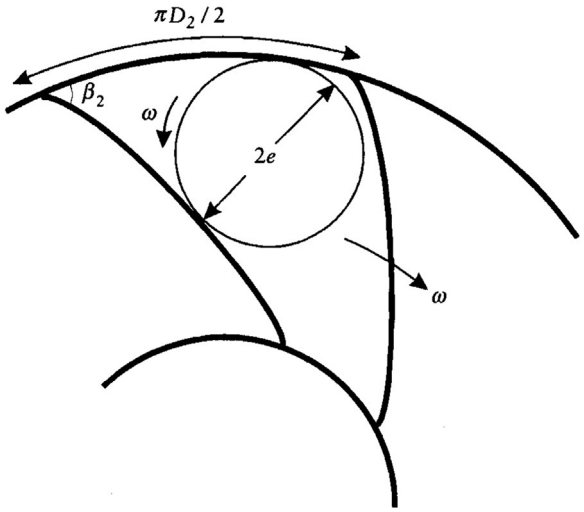


FIGURE 9.37 Secondary flow between the blades of a centrifugal fan.[19]

$$c'_{2u} = c_{2u} - e\omega \tag{9.87}$$

where ω is the impeller angular velocity, and e is the effective radius of the vortex at the outer edge of the impeller (Fig. 9.37).

Fig. 9.37 gives

$$\sin\beta_2 \approx \frac{2e}{\pi D_2/z} \tag{9.88}$$

where

$$e = \frac{\pi D_2 \sin\beta_2}{2z} \tag{9.89}$$

where z is the number of blades.

The fan slip factor is

$$\sigma = \frac{c'_{2u}}{c_{2u}} = \frac{c - (\pi D_2 \sin\beta_2/2z)\cdot(2u_2/D_2)}{c_{2u}} = 1 - \frac{\pi\sin\beta_2}{z}\cdot\frac{u_2}{c_{2u}}, \tag{9.90}$$

in which $\omega = 2u_2/D_2$ is used.

In Eq. (9.90), c_{2u} is the tangential component of the absolute velocity at the exit if the flow is exactly in the blade direction. Since the slip factor is less than 1, the total pressure increase will decrease according to Eq. (9.72) for the same impeller and isentropic flow.

The reaction ratio is improved, as shown in the following expression

$$r_s = 1 - \frac{1}{2}\frac{c'_{2u}}{u_2} = 1 - \frac{1}{2}\sigma\frac{c_{2u}}{u_2}. \tag{9.91}$$

The mass flow through the impeller is constant. For constant flow density and volume flow, the volume flow is

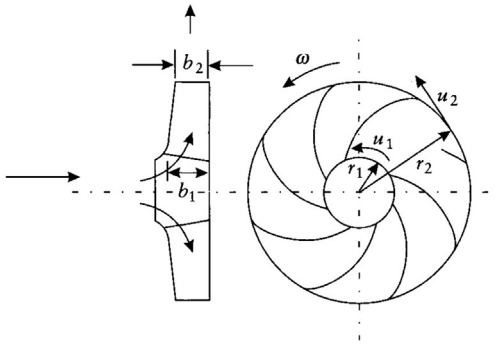


FIGURE 9.38 Dimensions of centrifugal fan impeller.[21]

$$q_v = \pi D_1 b_1 c_{1r} = \pi D_2 b_2 c_{2r} \tag{9.92}$$

where b_1 is the blade width at the entrance (Fig. 9.38), b_2 is the blade width at the exit (Fig. 9.38), c_{1r} is the absolute radial velocity component at the entrance, and c_{2r} is the absolute radial velocity component at the exit.

Because $c_{1u} = 0$, we have $c_{1r} = c_1$.

Example 9

The blades of the impeller in a centrifugal fan are backward curved. At the blade entrance $\beta_1 = 22$ degrees, and at the blade exit $\beta_2 = 50$ degrees. The outer and inner diameters of the impeller are $D_2 = 0.8$ m and $D_1 = 0.4$ m, respectively. The width of the blade at the entrance is $b_1 = 22$ cm and at the exit is $b_2 = 12$ cm. The impeller has 15 blades. The impeller rotational speed is $n = 960$ rev/min $= 16\ s^{-1}$. Calculate the volume flow, total pressure increase, static pressure increase, and reaction ratio. Take $c_{1u} = 0$, and air density is 1.22 kg/m^3.

Solution. The velocities of the circumferences are

$$u_2 = \pi D_2 n = \pi\cdot 0.8\cdot 16 = 40.2\ \text{m/s}$$
$$u_1 = \pi D_1 n = \pi\cdot 0.4\cdot 16 = 20.1\ \text{m/s}$$

The velocity triangle at the entrance, taking into consideration that $c_{1u} = 0$, is shown in Fig. 9.39A.

The triangle gives

$$c_{1r} = c_1 = \tan\beta_1 = 20.1\cdot\tan 22^\circ = 8.12\ \text{m/s}$$
$$w_1^2 = \sqrt{c_1^2 + u_1^2} = \sqrt{8.12^2 + 20.1^2} = 21.68\ \text{m/s}.$$

Eq. (9.92) gives

$$c_{2r} = \frac{D_1 b_1}{D_2 b_2}\cdot c_{1r} = \frac{0.40\cdot 0.22}{0.8\cdot 0.12} = 7.44\ \text{m/s}$$

Using

$$c_2^2 = c_{2u}^2 + c_{2r}^2$$

gives

$$c_{2u} = \sqrt{c_2^2 - c_{2r}^2} = \sqrt{34.8^2 - 7.44^2} = 34.0\ \text{m/s}.$$

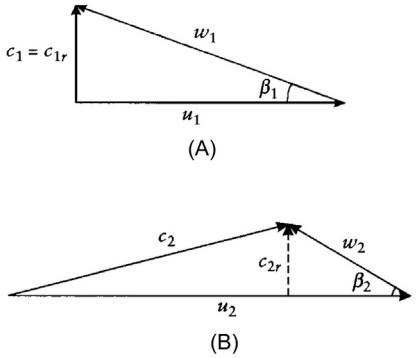


FIGURE 9.39 Velocity triangles: (A) at the entrance, (B) at the exit.

For $z = 15$, Eq. (9.90) gives

$$\sigma = 1 - \frac{\pi \sin \beta_2}{z} \cdot \frac{u_2}{c_{2u}} = 1 - \frac{\pi \cdot \sin 50^\circ}{15} \cdot \frac{40.2}{34.0} = 0.810,$$

from which $c'_{2u} = \sigma c_{2u} = 27.55\ m/s$, which is the tangential component of the absolute velocity parallel to the circumference velocity component when the blade number is taken into consideration. The flow process is still assumed isentropic. The velocity triangle at the exit is shown in Fig. 9.39B.

Using the velocity triangle gives

$$c_2 = \sqrt{c_{2u}^2 + c_{2r}^2} = \sqrt{27.55^2 + 7.44^2} = 28.5 \text{ m/s}$$

c_2 and w_2 are now the velocities when the slip factor is $\sigma = 0.81$.

$$w_2 = \sqrt{(u_2 - c'_{2u})^2 + c_{2r}^2} = \sqrt{(40.2 - 27.55)^2 + 7.44^2} = 14.7 \text{ m/s}$$

Using $c_1 = 8.12$ m/s, $w_1 = 21.68$ m/s, and $u_1 = 20.1$ m/s, the total pressure increase is

$$\Delta p_{tot,s} = \frac{1}{2}\rho\left(u_2^2 - u_1^2 + w_1^2 - w_2^2\right)$$

$$= 0.5 \cdot 1.22(82.5^2 - 8.12^2 + 40.2^2 + 20.1^2 + 21.68^2 - 14.7^2) = 1349 \text{ Pa},$$

and the static pressure increase is

$$\Delta p_s = \frac{1}{2}\rho\left(u_2^2 - u_1^2 + w_1^2 - w_2^2\right)$$

$$= 0.5 \cdot 1.22(40.2^2 - 20.1^2 + 21.68^2 - 14.7^2) = 894 \text{ Pa}.$$

The reaction ratio is

$$r_s = \frac{\Delta p_s}{\Delta p_{tot,s}} = \frac{894}{1349} = 0.663.$$

For an isentropic process and $c_{1u} = 0$, Eq. (9.75) gives

$$\Delta p_{tot,s} = \rho u_2 c_{2u}. \tag{9.93}$$

Using Fig. 9.33 gives

$$\frac{u_2 - c_{2u}}{c_{2r}} = \cot \beta_2, \tag{9.94}$$

from which

$$c_{2u} = u_2 - c_{2r} \cot \beta_2. \tag{9.95}$$

Fig. 9.35 gives

$$\frac{c_{2u} - u_2}{c_{2r}} = \cot(180^\circ - \beta_2) = \cot \beta_2, \tag{9.96}$$

from which

$$c_{2u} = u_2 - c_{2r} \cot \beta_2. \tag{9.97}$$

For all blade shape exits, we can write

$$c_{2u} = u_2 - c_{2r} \cot \beta_2 \quad \text{when} \quad 0 < \beta_2 < 180^\circ. \tag{9.98}$$

Using Eq. (9.92), the total pressure rise is expressed by

$$\Delta p_{tot,s} = \rho u_2 \left(u_2 - \frac{q_v \cot \beta_2}{\pi D_2 b_2}\right). \tag{9.99}$$

Eq. (9.99) shows the total pressure increase as a function of volume flow for given impeller dimensions and rotational speed at fixed u_2. The blade angle β_2 at the exit is taken as parameter. This function is a linear one. When $\beta_2 > 90$ degrees, $\Delta p_{tot,s}$ increases; for $\beta_2 = 90$ degrees, $\Delta p_{tot,s}$ remains constant; and for $\beta_2 < 90$ degrees, $\Delta p_{tot,s}$ decreases for other parameters given in Eq. (9.97).

The dependency of total pressure increase on volume flow is demonstrated in Fig. 9.40.

Fig. 9.40 is valid only for a single rotational speed. In addition, it is assumed that in the velocity triangle at the leaving edge of the impeller, the velocity component w_2 has the same direction as the angle β_2. The volume flow q_v changes when c_{2r} changes, which means, in the case of constant u_2 and constant direction of w_2, that the absolute velocity c_2 changes. When c_{2r} changes, so does c_{1r}, and the direction of w_1 changes because u_1 is constant. This means that in Fig. 9.40 only for one volume flow is the direction of the velocity w_1 the same as the angle β_1 for the impeller blade. For other values of q_v this is not the case. We still think that the flow process between impeller blades is isentropic, but it is likely that in reality the deviation from an isentropic process is larger the more the direction of w_1 deviates from β_1. This will change the curves in Fig. 9.40 so that we get lower increases in the total pressure.

If the flow is isentropic but does not follow the blade shape exactly, then the slip factor is smaller than 1, and this will directly affect what is shown in Fig. 9.40. Denoting $\Delta p'_{tot,s}$ the total pressure rise when

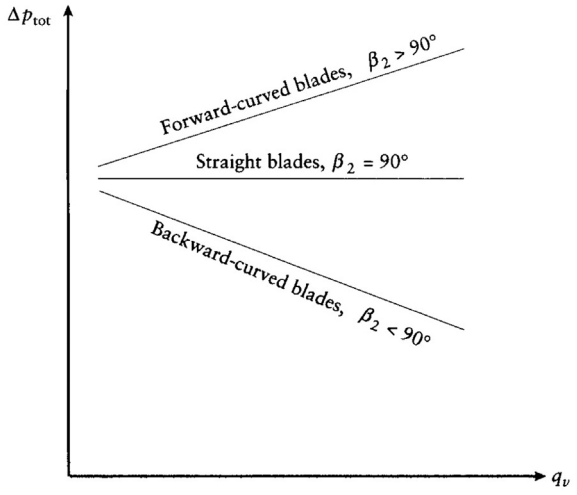


FIGURE 9.40 Total pressure increase versus volume flow for different exit blade angles in isentropic flow. The flow follows the blade shape exactly.

the flow does not follow the blade shape and calculating the slip factor from Eq. (9.90) give

$$\begin{aligned}\Delta p_{tot,s} - \Delta p'_{tot,s} &= \rho u_2 c_{2u} - \rho u_2 c'_{2u} \\ &= \rho u_2 c_{2u}(1-\sigma) = \rho u_2 c_{2u} \cdot \frac{\pi \sin \beta_2}{z} \cdot \frac{u_2}{c_{2u}} \\ &= \rho u_2^2 \cdot \frac{\pi \sin \beta 2}{z}.\end{aligned} \tag{9.100}$$

The total pressure decreases by the same quantity regardless of volume flow; the change is smaller for larger blade number z.

After the impeller the flow is flowing into the fan casing. The purpose of the casing is to collect the flow coming from the impeller and take the flow into the fan exit duct. Air flows to the casing everywhere from the exit edge of the impeller. For that reason the casing has a spiral shape (Fig. 9.41).

If there is no large duct after the fan which needs a large static pressure, then there is no need for spiral casing. The air then leaves steadily from the impeller circumference, and the air is not collected into one point. At the exit side in ventilating systems of buildings the need for static pressure is small if the leaving air is directly sent to the atmosphere.

Real fan total pressure difference is smaller for the same volume flow than that of an isentropic, theoretical fan. This is a result of the fan losses. These arise from the entropy generation in adiabatic systems. We investigate the losses separately, that is, entropy generation in the impeller and casing.

At the design point the volume flow is such that the relative velocity is parallel to the blade at the inlet of

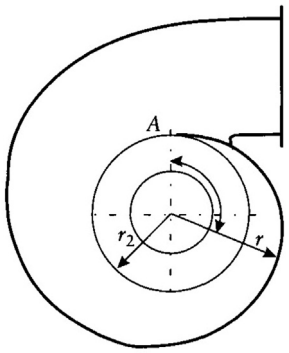


FIGURE 9.41 Centrifugal fan impeller and spiral casing.

the impeller. The impeller losses at the design point are as follows:

- Friction losses between the flow and blades.
- Leakage losses when the gas is transported from pressure side to suction side. Since the phenomenon is equalizing pressure in an adiabatic system, this will increase the entropy in the system.
- Friction in bearings, gasket, and impeller side walls. These losses are taken into consideration by the mechanical efficiency η_m.

The casing losses are due to friction and mixing. The frictional losses and their reasons are the same as those in the impeller channel. The mixing losses develop because the velocity of the impeller exit is not the same at every point as in the spiral casing; it is an average velocity. Mixing of two different flow velocities leads to acceleration and deceleration and pressure difference, whose equalizing increases the entropy.

Outside the design point the direction of the relative velocity is not parallel to the blade, and shock losses are generated.

The theory of losses is very complex. The effect of mixing on the total pressure has been investigated.[22] The calculated results qualitatively match the measured results. Friction in impeller channel and casing decreases the total pressure.

In practice, the fan characteristic curve—that is, the total pressure difference dependency on the volume flow—is determined experimentally. The measured results are then for the impeller and casing together. Since the losses are greater outside the design point, the fan efficiency is high at the design point.

Fig. 9.42 shows the typical characteristic curve of a centrifugal fan, where the blades are backward curved. The figure also shows the theoretical characteristic curve when the slip factor is 1 and when it is smaller than 1. Characteristic curves for a real fan are closer to the isentropic one at the design point. At this point the efficiency is maximum.

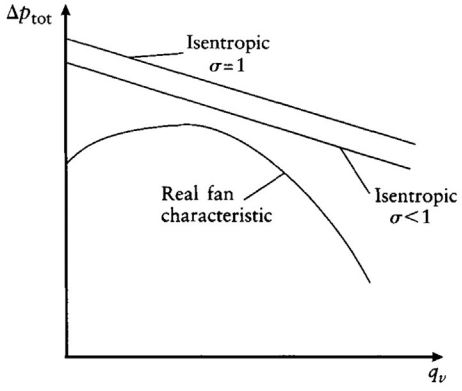


FIGURE 9.42 Typical performance curve of centrifugal fan, $\Delta P_{tot} - q_v$ chart.

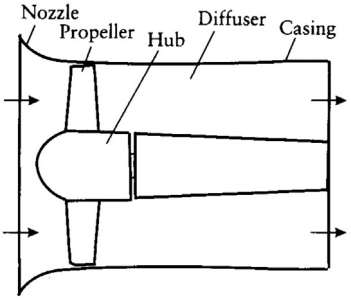


FIGURE 9.43 Schematic diagram of axial fan.[20]

9.5.3 Axial fans

Fig. 9.43 shows the schematic diagram of an axial fan system. The air flows through a nozzle toward the impeller, where static pressure rises. The impeller is attached to a hub. The impeller is also called the propeller. The propeller is followed by a diffuser.

When the aim is to build a cheap fan system and have a small pressure rise, then the nozzle and diffuser are not usually installed in the system.

The performance of an axial fan is based on the external force to drive the propeller, whose blades change the direction of airflow when flowing from the inlet edge to the outlet edge.

Eq. (9.59) is valid, but when $u_1 = u_2$, the power needed is

$$P_u = q_m u(c_{2u} - c_{1u}) = q_m u \Delta c_u. \tag{9.101}$$

Since P_u is positive, it must be that $c_{2u} > c_{1u}$ when the circumference speed is chosen as the positive direction. Eq. (9.101) is valid for every circumference speed between the hub circumference and the outer impeller circumference.

In the following discussion, u is the average circumference speed. The absolute velocity in the shaft direction is denoted by c_a.

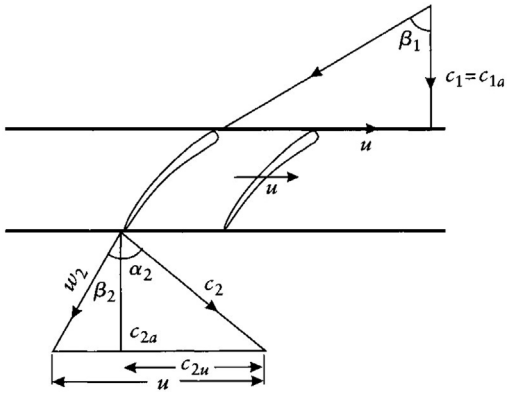


FIGURE 9.44 Velocity triangles of an axial fan impeller.

The mass flow is constant in the shaft direction, so

$$\rho \cdot \frac{\pi}{4}(D_1^2 - D_2^2)c_{1a} = \rho \cdot \frac{\pi}{4}(D_1^2 - D_2^2)c_{2a} \tag{9.102}$$

where D_1 is the impeller diameter and D_2 is the hub diameter.

Eq. (9.102) gives $c_{1a} = c_{2a}$. The axial component does not change, but when $c_{2u} > c_{1u}$, then $c_2 > c_1$. Thus the axial fan increases the absolute velocity of airflow.

Generally, Eq. (9.80) is valid, but when $u_1 = u_2$, it gives the static pressure increase for an axial fan as

$$\Delta p_s = p_{2s} - p_1 = \frac{1}{2}\rho(w_1^2 - w_2^2). \tag{9.103}$$

Eq. (9.79) shows that the rise in circumference velocities at the entrance and exit leads to the static pressure $1/2\rho(u_2^2 - u_1^2)$ increase but not in the axial fan. This results in the axial fan's not having the same pressure rise as that of the centrifugal fan. Axial fans are applicable for large airflows, when the needed pressure increase is relatively small.

The reaction ratio of an axial fan for the isentropic case based on Eqs. (9.75), (9.81), and (9.103) is

$$r_s = \frac{w_1^2 - w_2^2}{2u(c_{u2} - c_{u1})}. \tag{9.104}$$

We restrict ourselves to investigate an axial fan, where there is only an impeller. The incoming air has the direction of the axis of the propeller.

The rotating of the impeller increases the absolute velocity of air. In order to increase the static pressure of the gas flow owing to the fan, the velocity relative to the blades should decrease according to Eq. (9.103).

The flow velocity diagrams on both sides of the impeller are shown in Fig. 9.44. The axial direction is the datum for all angles.

From the conditions $c_{2a} = c_{1a} = w_{2a} = w_{1a}$ and $w_1 > w_2$, the relative velocity is smaller at the exit than at the inlet. There are two possibilities: the leaving flow direction angle is on the same axis side as that of the coming flow,

or the leaving flow direction angle is on a different axis side. Fig. 9.44 shows blades where both the relative velocity direction angles β_1 and β_2 are on the same axis side.

The velocity triangle at the entrance gives

$$w_1^2 = u^2 + c_a^2 \tag{9.105}$$

and at the exit gives

$$w_2^2 = (u - c_{2u})^2 + c_a^2. \tag{9.106}$$

From Eq. (9.103) the isentropic pressure rise is determined by

$$\Delta p_s = \frac{1}{2}\rho(2uc_{2u} - c_{2u}^2) = \rho u^2\left(\frac{c_{2u}}{u} - \frac{1}{2}\frac{c_{2u}^2}{u^2}\right). \tag{9.107}$$

The exit velocity triangle gives

$$\tan\beta_2 = \frac{u - c_{2u}}{c_a}, \tag{9.108}$$

from which

$$\frac{c_{2u}}{u} = 1 - \frac{c_1}{u}\tan\beta_2, \tag{9.109}$$

and from Eq. (9.107)

$$\Delta p_s = \frac{1}{2}\rho u^2\left(1 - \frac{c_a^2}{u^2}\tan^2\beta_2\right). \tag{9.110}$$

Using Eqs. (9.103) and (9.104), the reaction ratio is determined by

$$r_s = \frac{2\Delta p}{2\rho u c_{2u}} = \frac{u^2\left(1 - (c_a^2/u^2)\tan^2\beta^2\right)}{2u^2\left(1 - (c_a/u)\tan\beta_2\right)} = \frac{1}{2}\left(1 + \frac{cL}{u}\tan\beta_2\right) = \frac{1}{2}\left(1 + \frac{\tan\beta_2}{\tan\beta_1}\right). \tag{9.111}$$

Because $0 < \beta_1 < \pi/2$ and $0 < \beta_2 < \pi/2$, then $\cot\beta_1 > 0$, $\tan\beta_2 > 0$, and $r > 1/2$.

From the exit velocity triangle, it can be seen that the gas flow has a tangential velocity component. The gas rotates when it leaves the fan. Normally, the tangential velocity component is of no benefit if a duct is attached to the fan, since it disappears due to friction.

Example 10

The diameter of an axial fan impeller is $D_1 = 0.6$ m, the hub diameter is $D_2 = 0.3$ m, and the rotational speed is $n = 960$ rev/min. The axial velocity of airflow is $c_{1a} = 5.5$ m/s, and the blade angle is $\beta_2 = 10$ degrees (average) at the blade exit. Calculate the power, isentropic static pressure increase, and reaction ratio. The pressure of the coming air is 1 bar, and the temperature is 45°C.

Solution. The average diameter is calculated by

$$D = \frac{1}{2}(D_1 + D_2) = \frac{1}{2}(0.6 + 0.3) = 0.45 \text{ m}.$$

The average circumference speed is $u = \pi Dn = 22.6$ m/s. The velocity triangle on the entrance side as shown in Fig. 9.45A gives

$$\tan\beta_1 = \frac{u}{c_{1a}} = \frac{u}{c_a} = \frac{22.6}{5.5} = 4.11 \Rightarrow \beta_1 = 76.3^\circ.$$

The velocity triangle at exit has $\beta_2 = 10$ degrees, $w_{2a} = c_{2a} = 5.5$ m/s, which gives

$$\tan\beta_2 = \tan 10^\circ = \frac{w_{2u}}{w_{2a}}$$

$$w_{2u} = w_{2a}\cdot\tan\beta_2 = 5.5\tan 10^\circ = 0.97 \text{ m/s}$$
$$c_{2u} = u - w_{2u} = 22.6 - 0.97 = 21.6 \text{ m/s}.$$

Using ideal gas law, the air density is calculated as

$$\rho = \frac{Pm}{RT} = \frac{10^5\cdot 28.964}{8314.31\cdot(273 + 45)} = 1.095 \text{ kg/m}^3.$$

The flow area is

$$A = \frac{\pi}{4}(D_1^2 - D_2^2) = \frac{\pi}{4}(0.6^2 - 0.3^2) = 0.212\text{m}^2.$$

The mass flow through the fan is

$$q_m = \rho c_{1a} A = 1.095\cdot 5.5\cdot 0.212 = 1.277 \text{ kg/s}.$$

Eq. (9.101) gives the power as

$$P_u = q_m u(c_{2u} - c_{1u}) = q_m u c_{2u} = 1.277\cdot 22.6\cdot 21.6 = 623 \text{ W}.$$

The isentropic static pressure increase is obtained from Eq. (9.107) as

$$\Delta p_s = \rho u^2\left(\frac{c_{2u}}{u} - \frac{1}{2}\frac{c_{2u}^2}{u^2}\right) = 1.095\cdot 22.6^2\left(\frac{21.6}{22.6} - \frac{1}{2}\cdot\frac{21.6^2}{22.6^2}\right) = 267 \text{ Pa}.$$

Eq. (9.111) gives the reaction ratio as

$$r_s = \frac{1}{2}\left(1 + \frac{\tan\beta_2}{\tan\beta_1}\right) = \frac{1}{2}\left(1 + \frac{\tan 10^\circ}{\tan 76.3^\circ}\right) = 0.521.$$

At the exit the absolute velocity has velocity component c_{2u} on the large circumference parallel to the shaft of

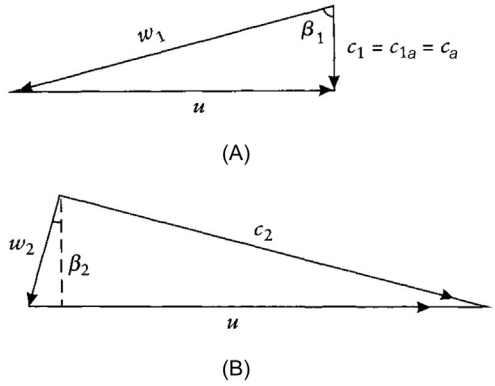


FIGURE 9.45 Velocity triangles: (A) at the entrance, (B) at the exit.

Example 11. Component c_{2u} is of no advantage if a duct is connected to the axial fan, since it disappears due to the friction between the walls of the duct and gas flow.

9.5.4 Effect of speed of revolution

In the previous sections the centrifugal and axial fans were investigated for a constant rotational speed.

In this section the rotational velocity is directly proportional to the rotational velocity n according to the equation $u = \pi Dn$. The impeller blade angles remain the same regardless of the rotational velocity of the impeller. Hence, the inlet and exit velocity triangles have the same form. The axial velocity of an axial fan changes directly proportionally to the circumference velocity u. This is also valid for the radial velocity at the outer circumference of a radial impeller fan. These velocities are directly proportional to the fan flow volume; hence

$$\frac{q_{v2}}{q_{v1}} = \frac{n_2}{n_1}, \tag{9.112}$$

where n_1 is the rotational velocity of impeller, q_{v1} is the volume flow at rotational velocity n_1, n_2 is another rotational velocity of the impeller, and q_{v2} is the volume flow at rotational velocity n_2.

As the velocity triangles' shape remains the same

$$\frac{u_{21}}{u_{22}} = \frac{u_{11}}{u_{12}} = \frac{c_{u21}}{c_{u22}} = \frac{c_{u11}}{c_{u12}} = k. \tag{9.113}$$

where subscript u indicates the velocity to the tangential velocity component, the first subscript 2 indicates impeller exit, the first subscript 1 indicates impeller inlet, the second subscript 2 stands for n_2, and the second subscript 1 stands for n_1. The proportionality constant k is

$$k = \frac{n_1}{n_2}. \tag{9.114}$$

Even though the rotational velocity changes, the flow is still parallel to the blades, and the hydraulic efficiency remains the same regardless of rotational speed.

Using Eqs. (9.75), (9.113), and (9.114) gives

$$\frac{\Delta p_{tot2}}{\Delta p_{tot1}} = \frac{\rho_2}{\rho_1}\cdot\frac{u_{22}c_{u22} - u_{12}c_{u12}}{ku_{22}kc_{u22} - ku_{12}kcu_{12}} = \frac{\rho 2}{\rho 1}\cdot\frac{1}{k^2} = \frac{\rho 2}{\rho 1}\cdot\left(\frac{n_2}{n_1}\right)^2. \tag{9.115}$$

Normally, with fans $\rho_1 = \rho_2$. Then

$$\frac{\Delta p_{tot2}}{\Delta p_{tot1}} = \left(\frac{n_2}{n_1}\right)^2. \tag{9.116}$$

If the rotational velocity change is within reasonable limits and the mechanical efficiency does not change, Eqs. (9.76), (9.112), and (9.115) give the shaft power relation as

$$\frac{P_{a2}}{P_{a1}} = \frac{\rho_2}{\rho_1}\cdot\left(\frac{n_2}{n_1}\right)^3. \tag{9.117}$$

Example 11

The fan is tested at an air pressure of 102.9 kPa, temperature of 10°C, and a rotational speed of 970 rev/min. Under these conditions the volume flow is 0.7 m^3/s, total pressure difference is 250 kPa, and shaft power is 250 kW. If the operating conditions change to handle an air temperature of 16°C and pressure of 100 kPa and the efficiency remains unchanged, calculate under the new operating conditions the volume flow, total pressure difference, and shaft power.

Solution

Operating conditions 1: $p_1 = 102.9$ kPa and $T_1 = 10°C = 283K$.

Operating conditions 2: $p_2 = 100$ kPa and $T_2 = 16°C = 289K$.

Using ideal gas law

$$\rho_1 = \frac{p_1 M}{RT_1} = \frac{102.9\cdot 10^3\cdot 28.964}{8314.31\cdot 283} = 1.267\ \text{kg/m}^3$$

$$\rho_2 = \frac{p_2 M}{RT_2} = \frac{100\cdot 10^3\cdot 28.9864}{8314.31\cdot 283} = 1.205\ \text{kg/m}^3.$$

Using $n_1 = 970$ rev/min, $q_{v1} = 0.73\ m^3/s$, $\Delta P_{tot1} = 250$ Pa, $P_{a1} = 250$ W, and $n_2 = 500$ rev/min, Eq. (9.112) gives

$$q_{v2} = \frac{n_2}{n_1}\cdot q_{v1} = \frac{500}{970}\cdot 0.7 = 0.361\ \text{m}^3/\text{s}.$$

Eq. (9.115) gives the total pressure difference as

$$\Delta p_{tot2} = \frac{\rho_2}{\rho_1}\cdot\left(\frac{n_2}{n_1}\right)^2\Delta p_{tot1} = \frac{1.205}{1.267}\left(\frac{500}{970}\right)^2\cdot 250 = 63.2\ \text{Pa},$$

and Eq. (9.117) gives the shaft power as

$$P_{a2} = \frac{\rho_2}{\rho_1}\cdot\left(\frac{n_2}{n_1}\right)^3 P_{a1} = \frac{1.205}{1.267}\left(\frac{500}{970}\right)^3\cdot 250 = 32.6\ \text{W}.$$

Example 12

The total pressure is 500 Pa, and volume flow is 2.4 m^3/s for a fan working at standard air conditions (pressure 1 bar and temperature 20°C). The speed of revolution is 14 rev/s, and fan efficiency is 60%.

1. Calculate the total pressure, speed of revolution, and power if the volume flow is increased to 3.5 m^3/s. The fan efficiency is assumed to remain constant.
2. Calculate the total pressure and power at the first volume flow if the temperature is increased to 82°C and the speed of revolution remains constant.

Solution

1. Denote $q_{v1} = 2.4\ \text{m}^3/\text{s}$, $\Delta p_{\text{tot1}} = 500$ Pa, $n_1 = 14$ rev/s, and $q_{v1} = 3.5\ \text{m}^3/\text{s}$. From Eq. (9.112), we get

$$n_2 = \frac{q_{v2}}{q_{v1}} n_1 = \frac{3.5}{2.4}.14 = 20 \text{ rev/s}.$$

From Eq. (9.116), we get

$$\Delta p_{\text{tot2}} = \left(\frac{n_2}{n_1}\right)^2 \cdot \Delta p_{\text{tot1}} = \left(\frac{20}{14}\right)^2 \cdot 500 = 1020 \text{ Pa}.$$

From Eq. (9.76), we get

$$P_{a2} = \frac{q_{v2} \cdot \Delta p_{\text{tot2}}}{\eta} = \frac{3.5 \cdot 1020}{0.6} = 5950 \text{ W}.$$

2. At standard conditions, we have $p_1 = 1$ bar and $T_1 = 20°\text{C} = 293\text{K}$.

In the new situation we have $p_1 = p_2 = 1$ bar and $T_2 = 82°\text{C} = 355\text{K}$. From ideal gas law, we have $\rho_1 = p_1 M/RT_1$ and $\rho_2 = p_2 M/RT_2$ and because $p_1 = p_2$, we get

$$\frac{\rho_2}{\rho_1} = \frac{T_1}{T_2}.$$

From Eq. (9.115), we then get, for $n_1 = n_2$

$$\frac{\Delta p_{\text{tot2}}}{\Delta p_{\text{tot1}}} = \frac{\rho_2}{\rho_1}$$

$$\Delta p_{\text{tot2}} = \frac{\rho_2}{\rho_1} \Delta p_{\text{tot1}} = \frac{T_1}{T_2} \Delta p_{\text{tot1}} = \frac{293}{355} \cdot 500 = 413 \text{ Pa}.$$

From Eq. (9.76), we get

$$P_{a2} = \frac{q_{v1} \cdot \Delta p_{\text{tot2}}}{\eta} = \frac{2.4 \cdot 413}{0.6} = 1652 \text{ W}.$$

9.5.5 Fan and duct network

In some instances the fan has a free discharge. Typical is the axial fan installed in a wall opening (wall fan). In most cases the fan is connected to a duct run; in this instance the total pressure difference and volume flow are determined from both the fan and duct network characteristics.

Consider the thermodynamic analysis for incompressible gas flow in a duct. Denote the point at the beginning by 1 and at the end by 2. Since there is no mechanical power, for steady-state flow the energy balance equation is

$$h_1 + \frac{1}{2}c_1^2 + gz_1 = h_2 + \frac{1}{2}c_2^2 + gz_2 \tag{9.118}$$

where c is the air velocity and z is the height.

Eq. (9.63) gives

$$h_2 - h_1 = \int_1^2 T\,ds + \frac{1}{\rho}(p_2 - p_1), \tag{9.119}$$

for which Eq. (9.118) can be written as

$$\begin{aligned} p_1 + \frac{1}{2}\rho c_1^2 + \rho g z_1 &= p_2 + \frac{1}{2}\rho c_2^2 + \rho g z_2 + \rho \int_1^2 T\,ds \\ &= p_2 + \frac{1}{2}\rho c_2^2 + \rho g z_2 + \Delta p. \end{aligned} \tag{9.120}$$

The term $\Delta p = \rho \int_1^2 T\,ds$ is the entropy generation given as pressure loss.

In a straight duct where $z_1 = z_2$ and the cross-sectional area is uniform, then $c_1 = c_2$. This gives $\Delta p = p_1 - p_2 > 0$. Δp is the duct pressure loss, which is a result of entropy generation. Entropy generation is due to the flow friction, which is the reason for pressure loss.

Consider the pressure loss in a duct with straight, uniform cross-sectional area. The pressure loss is caused by friction. When different air sheets move against each other, friction is generated. The velocity and thermodynamic properties of air influence the friction. The duct wall has an overall roughness, which causes vortices to be formed with resulting friction in gas. The velocity has a pronounced effect; in flow with low velocity, the vortices are small. For a straight duct the pressure loss Δp_k can be determined from

$$\frac{\Delta pk}{L} = \frac{f}{D} \cdot \frac{1}{2}\rho c^2, \tag{9.121}$$

where L is the duct length, D is the inner diameter, and f is the flow friction factor.

In velocity and gas property effects the Reynolds number, Re, is taken into consideration as

$$Re = \frac{cD}{v}, \tag{9.122}$$

where v is the kinematic viscosity.

The kinematic viscosity is related to density by

$$\mu = \rho v \tag{9.123}$$

where μ is the dynamic viscosity.

The friction factor depends on the Reynolds number and duct wall relative roughness e/D, where e is the average height of the roughness in the duct wall. The friction factor is shown in Fig. 9.46. For a large Reynolds number, the friction factor f is considered constant for rough pipe surfaces. The friction pressure loss is $\Delta p_k \propto c^2$.

For a smooth duct surface,[17] when $Re > 2000$, the friction factor is

$$f = 0.184\, Re^{-0.2}. \tag{9.124}$$

Eq. (9.121) then shows that $\Delta p_k \propto c^{1.8}$. In practice, $\Delta p_k \propto c^2$ can be used with a high accuracy. For straight, uniform duct cross-sectional area, the static

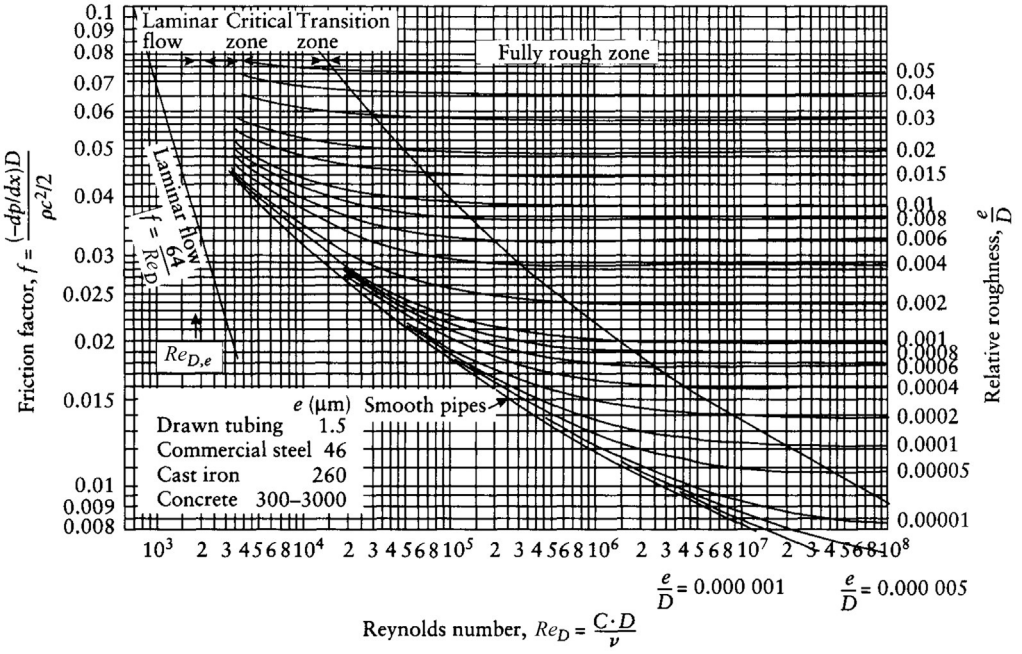


FIGURE 9.46 Friction factor f of duct flow.[23]

pressure drop in the flow direction is directly proportional to c^1.

The internal friction due to vortices occurs in rapid expansion, diverging, and regulation valves. The entropy generation due to those vortices is taken into consideration in the local resistance. The entropy generation is directly proportional to c^2; thus

$$\int_1^2 T\,ds = \zeta \frac{1}{2} c^2 \tag{9.125}$$

where ζ is the minor loss coefficient.

$$\Delta p_{kv} = \rho \int_1^2 T\,ds = \zeta \frac{1}{2} \rho c^2. \tag{9.126}$$

The minor loss coefficient depends on the Reynolds number; because this dependency is weak, it is normally ignored. The minor loss coefficients for different resistances are given in textbooks.

Consider next the fan with its connected air duct characteristics when both are operating together. We indicate the fan leaving air by subscript 2 and the suction side by subscript 1.

Using Eq. (9.120) gives

$$p_2 + \frac{1}{2}\rho c_2^2 = p_1 + \frac{1}{2}\rho c_1^2 + \Delta p, \tag{9.127}$$

where $\Delta p = \Delta p_k + \Delta p_{kv}$, the pressure difference in straight ducts, bends, etc. caused by the entropy generation.

Eq. (9.127) can be rewritten as

$$\Delta p = p_2 - p_1 + \frac{1}{2}\rho\left(c_2^2 - c_1^2\right) = \Delta p_{\text{tot}}, \tag{9.128}$$

which means that the generated pressure loss is large—as much as the fan total pressure difference.

For a special fan situation, a straight air duct of uniform cross-sectional area is used on the leaving side. The outgoing velocity c_3 is the same as the fan leaving velocity c_2. The only minor loss is the outgoing loss, that is, $1/2\rho c_3^2$. Another part of the pressure drop is the frictional pressure drop Δp_k. Eq. (9.127) gives

$$\Delta p = \Delta p_k + \frac{1}{2}\rho c_3^2 = \Delta p_k + \frac{1}{2}\rho c_2^2 = \Delta p_{\text{tot}}, \tag{9.129}$$

from which

$$\Delta p_k = \Delta p_{\text{tot}} - \frac{1}{2}\rho c_2^2 = p_2 - p_1 - \frac{1}{2}\rho c_2^2 = p_k. \tag{9.130}$$

The pressure term p_k in Eq. (9.130) is called the obtainable fan pressure.

In steady-state conditions the mass and volume flow are constant through the fan and duct. If the duct consists of branches of different cross-sectional area A_i, then

$$q_v = c_i A_i \quad i = 1, \ldots, n \tag{9.131}$$

where n is the number of different cross-sectional areas A_i.

The duct pressure drop can be obtained from Eqs. (9.121) and (9.126) as

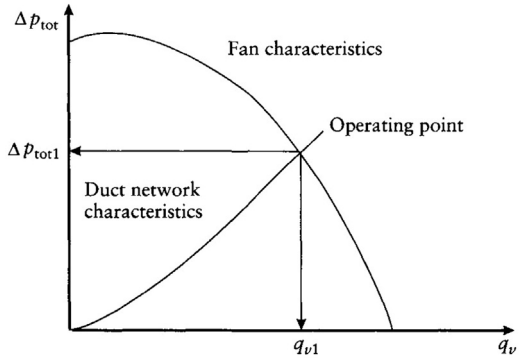


FIGURE 9.47 Fan and duct operating point (Δp_{tot}, q_{v1}).

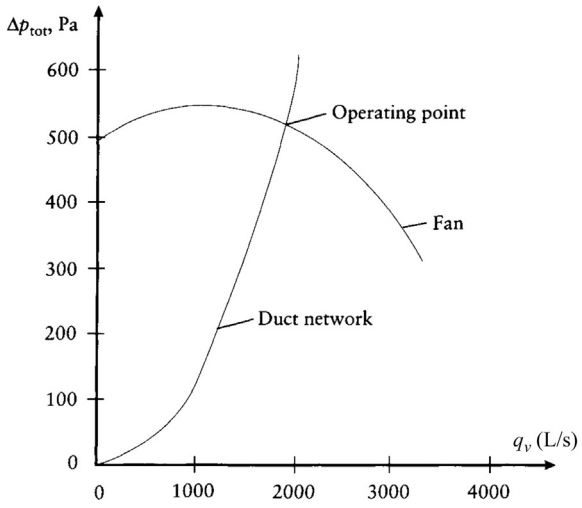


FIGURE 9.48 Fan characteristic curve.

$$\Delta p = \sum f_i \frac{L_i}{D_i}\frac{1}{2}\rho c_i^2 + \sum \zeta_i \frac{1}{2}\rho c_i^2. \qquad (9.132)$$

The fan total pressure difference Δp_{tot} also depends on the volume flow. In practice, dependency is determined experimentally, $\Delta p_{tot} = f(q_v)$. Eqs. (9.128) and (9.132) give

$$\Delta p_{tot} = f(q_v) = \sum f_i \frac{L_i}{D_i}\frac{1}{2}\rho c_i^2 + \sum \xi_i \frac{1}{2}\rho c_i^2. \qquad (9.133)$$

The fan volume flow q_v and its corresponding Δp_{tot} can be found when a $\Delta p_{tot} - q_v$ chart is drawn; the duct parabola and experimental Δp_{tot} both equal $f(q_v)$ (Fig. 9.47). The experimental curve $\Delta p_{tot} = f(q_v)$ is called the *fan characteristic curve* (Fig. 9.48), and the duct static pressure drop dependency on the duct volume flow is the *characteristic curve*. The characteristic curve intersection point is called *fan operating point*.

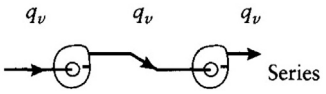


FIGURE 9.49 Series fan connection.

Example 13

The fan characteristic curve is given by

q_v (L/s)	0	556	1111	1667	2222	2778	3333
Δp_{tot} (Pa)	491	535	549	535	491	417	314
P_a (kW)	0.40	0.63	0.90	1.20	1.53	1.7	1.75

The duct pressure drop is 589 Pa; airflow is 1944 L/s. Determine the fan total pressure, volume flow, shaft power, and total efficiency.

Solution. Draw into the $\Delta p_{tot} - q_v$ diagram the characteristic curve of the fan and the duct-pressure-drop volume flow dependency. The latter is a parabola passing through the origin with the following equation:

$$\Delta p = \frac{589}{1944}\cdot q_v^2.$$

(Here Δp_{tot} is in Pa and q_v in L/s.) At the curve's intersection point, $\Delta p = \Delta p_{tot} = 520$ Pa and $q_v = 1900$ L/s $= 1.9$ m^3/s. By interpolation from the above table, the shaft power is 1.34 kW. The total fan efficiency is

$$\eta = \frac{\Delta p_{tot}\cdot q_v}{\text{Pa}} = \frac{520\cdot 1.9}{1340} = 0.737 = 73.7\%.$$

It may be necessary in a given system to use more than one fan. The fans may be connected either in series or parallel.

9.5.6 Series fan connection

In this case the outlet of the first fan is connected to the inlet of the second fan. With this arrangement, the total head developed at a given volume is equal to the sum of the total heads developed by the individual fans. Sometimes more than one fan is used in a duct arrangement. In the series connection (Fig. 9.49) the flow through the fans has the same volume flow, and the leaving flow from the first fan is connected to the suction side of the second fan.

For the series connection, we indicate by 1 and 2 the suction and leaving sides of the first fan, respectively, and by 3 the leaving side of the second fan. The suction side of the second fan is the same as the leaving side of the first fan, which is indicated by 2.

The common total pressure difference is

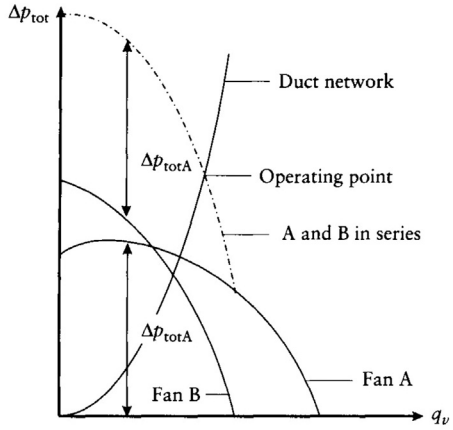


FIGURE 9.50 The determination of the operating point for two fans in series connected a ductwork.

$$\Delta p_{tot} = p_3 + \frac{1}{2}\rho c_3^2 - p_1 - \frac{1}{2}\rho c_1^2 = p_3 + \frac{1}{2}\rho c_3^2 - p_2 - \frac{1}{2}\rho c_2^2$$

$$p_2 + \frac{1}{2}\rho c_2^2 - p_1 - \frac{1}{2}\rho c_1^2 = \Delta p_{tot1} + \Delta p_{tot2} = f(q_{v1}) + f(q_{v2}), \quad (9.134)$$

where q_v is the volume flow through both fans and $f(q_v) = \Delta p_{tot1}$ is the total-pressure-difference volume flow dependency of the first fan. The points (qv, $f_1(q_v)$) are the characteristic curve of the first fan, and the points are the characteristic curve of the second fan.

Eq. (9.128) gives

$$\Delta p_{tot} = \Delta p = \Delta p_{tot1}(q_{v1}) + \Delta p_{tot2}(q_{v2}). \quad (9.135)$$

The network and fan combination operating point can be obtained from Eq. (9.134).

In a $\Delta p - q_v$ chart the characteristic curve of the ducts is drawn. In the same chart is drawn the characteristic curve of both fans. At each volume flow q_v, the total pressure of each fan is added. In this way, we get a new $\Delta p_{tot} - q_v$ curve. The intersection point of this new curve and the characteristic curve of the ducts is the operating point (Fig. 9.50).

9.5.7 Fan volume flow regulation

If outlet volume flow on the outer side of the attached network of the fan must be adjusted or changed, this will happen by changing the fan characteristic curve or the network impedance. The fan and network then settle in a new operating point, and the volume flow changes.

The impedance of the network can be influenced by the change of the minor loss resistance number of a damper.

When the minor loss resistance changes, the impedance of the network and hence the fan and network common operating point also change. The choking regulation with a damper is a bad regulation mode from the point of view of fan energy consumption. The fan is chosen so that in normal conditions the fan is running at the design point. At this point, air absolute velocity, and circumference velocities of the impeller and blade angles are such that the velocity triangles are in the correct form, that is, the flow relative velocity is parallel to the blades, so shock losses are not generated and efficiency is at its highest value.

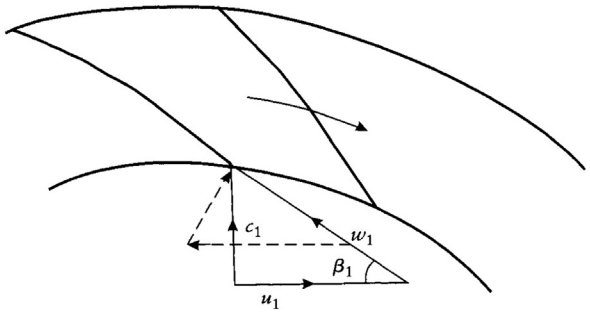


FIGURE 9.51 Leading blades' influence on the inlet velocity triangle of a centrifugal fan. Solid line: velocity triangle without leading blades. Dashed line: velocity triangle with leading blades.

When choking, neither circumference velocity nor blade angles change. The direction of the flowing gas's absolute velocity toward the impeller does not change, but the magnitude changes. The velocity triangles are now such that the relative speed is not parallel to the blade, and shock losses are generated. This causes a decrease in the efficiency.

A change in the fan characteristic curve can be made by:

- adjustable leading blades (in front of suction opening of centrifugal fans),
- regulation of blade angles of impeller (axial fans), or
- regulation of rotational velocity of impeller.

With adjustable leading blades for a centrifugal fan the volume flow changes by changing the velocity triangles' form at the impeller inlet. By giving the coming airflow a tangential velocity component, the inlet velocity triangle can be changed in such a way that the absolute velocity radial component changes and the relative velocity remains parallel to the blade. Then the fan efficiency is high. The leading blades change the coming flow's absolute velocity in such a way that the tangential velocity component is in the same parallel direction as the circumference velocity. Leading blades are generally used if the blades are backward curved (Fig. 9.51).

The regulation of axial fan blade angle also influences the inlet and exit velocity triangles in such a way that the axial velocity and thus the volume flow

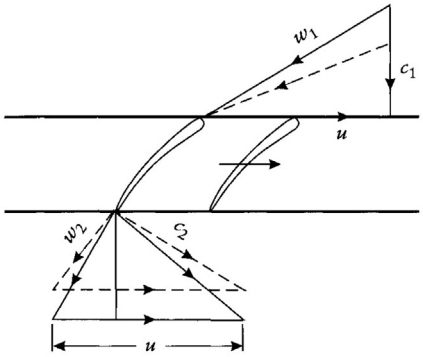


FIGURE 9.52 Influence of axial fan blade angle on volume flow and velocity triangles.

change. When the relative velocity remains parallel to the blade, the efficiency remains high (Fig. 9.52).

Recently, the regulation of impeller rotational velocity has become a popular regulation mode for volume flow. Electric-motor rotational velocity is regulated by a frequency changer, and its price has dropped lately. Changing the rotational speed also affects the circumference velocity of the impeller. The volume flow can be changed by the same ratio as rotational speed. The form of the velocity triangles and the efficiency remain the same.

Consider the fan characteristic curves for two different rotational velocities n_1 and n_2. Select the operating point for the characteristic curve n_1 as $(\Delta p_{tot1}, q_{v1})$. The corresponding point is $(\Delta p_{tot2}, q_{v2})$ for characteristic curve n_2.

Based on Eqs. (9.112) and (9.116)

$$\frac{q_{v2}}{q_{v1}} = \frac{n_2}{n_1} \text{ and } \frac{\Delta p_{tot2}}{\Delta p_{tot1}} = \left(\frac{n_2}{n_1}\right)^2. \tag{9.136}$$

From Eq. (9.136), the characteristic curve of n_2 can be drawn when the characteristic curve of n_1 is known.

Denoting $(\Delta p_{tot}, q_v)$ as the corresponding point of the operating point $(\Delta p_{tot1}, q_{v1})$ and the corresponding rotational velocity $n \neq n$, Eq. (9.136) gives

$$\frac{\Delta p_{tot}}{\Delta p_{tot1}} = \left(\frac{q_{v2}}{q_{v1}}\right)^2 \text{ or } \Delta p_{tot} = \frac{\Delta p_{tot1}}{q_{v1}^2} \cdot q_{v2}^2. \tag{9.137}$$

The corresponding points to $(\Delta p_{tot1}, q_{v1})$ compose a parabola, the affinity parabola in the $\Delta p_{tot} - q_v$ chart. The parabola passes through the origin and point $(\Delta p_{tot1}, q_{v1})$.

At the corresponding points of the parabola the rotational velocity and volume flow change by the same ratio. If the efficiency is high at the point $(\Delta p_{tot1}, q_{v1})$, it is also high at the corresponding point. The fan is connected to a network where there is no static height, so the network $(\Delta p_{tot}, q_v)$ curve is a parabola that passes through the origin.

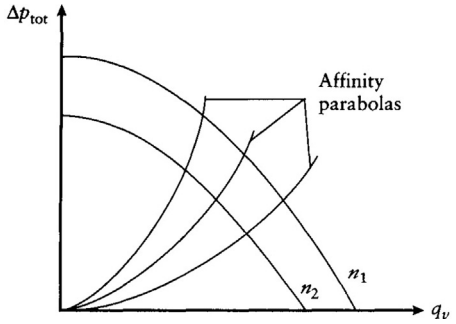


FIGURE 9.53 Fan characteristic curve for two rotational velocities, n_1 and n_2, and three fan affinity parabolas.

The fan operating point is the intersection point of the characteristic curve and the network $(\Delta p_{tot}, q_v)$ curve. Both parabolas have two common points; the origin and the operating points. If the volume flow is changed, increasing or decreasing the rotational velocity, the new operating point will be on the affinity parabola. If the efficiency is high for the original operating point, then it will also be high for the new one (Fig. 9.53). Hence, rotational speed regulation is a good mode for energy consumption.

Example 14

A fan delivers air to a ventilating system at total pressure difference 500 Pa. The fan is running at 10 rev/s, and the shaft power needed in these conditions is 7.46 kW. Determine the volume flow, total pressure difference, and shaft power if the fan speed is increased to 12.5 rev/s.

Solution. Denote $q_{v1} = 1.55\ \text{m}^3/\text{s}$, $\Delta p_{tot1} = 500\ \text{Pa}$, $n_1 = 10\ \text{rev/s}$, and $P_{a1} = 7.46\ \text{kW}$. The new fan speed is $n_2 = 12.5\ \text{rev/s}$.

The new operating point is on the affinity parabola through $q_{v1} = 1.55\ \text{m}^3/\text{s}$, and $\Delta p_{tot1} = 500\ \text{Pa}$. From Eq. (9.136) we have for corresponding points on the affinity parabola

$$q_{v2} = \frac{n_2}{n_1} q_{v1} = \frac{12.5}{10} \cdot 1.55 = 1.937\ \text{m}^3/\text{s},$$

$$\Delta p_{tot2} = \left(\frac{n_2}{n_1}\right)^2 \Delta p_{tot1} = \left(\frac{12.5}{10}\right)^2 \cdot 500 = 781\ \text{Pa}.$$

Finally

$$P_{a2} = \frac{\Delta p_{tot2} \cdot q_{v2}}{\eta} = \frac{\Delta p_{tot1} (n_2/n_1)^2 \cdot (n_2/n_1) \cdot q_{l/1}}{\eta}$$

$$= \frac{\Delta p_{tot1} (n_2/n_1)^3 \cdot q_{v1}}{\eta} = P_{a1} \cdot \left(\frac{n_2}{n_1}\right)^3$$

$$= 7.46 \cdot \left(\frac{12.5}{10}\right)^3 = 14.6\ \text{kW}.$$

9.6 Automatic control of HVAC systems

The first technological revolution, where mechanical production was powered by steam, lasted until the second half of the 19th century. The most important was the use of steam engine invented in 1763 by James Watt. The industrial development has intensified the evolution of heating, ventilation and air conditioning (HVAC) controls on hydraulic, pneumatic, and steam systems. First ideas date back to the 17th century, and among them are thermostat for controlling boilers, a device for temperature control in incubators, feedback loop concepts (one of the first feedback-controlled devices was invented by Cornelis Drebbel in 1620). The first proportional control for steam engine was used in 1788. The Second Industrial Revolution, in which mass production was powered by electricity, was initiated by another series of inventions from the turn of the 19th and 20th centuries. The most important was the development of oil refining methods by Łukasiewicz in 1852, which allowed to use oil as an energy source. We owe the propagation of electricity, lighting, and machine drive to Edison's invention from 1879.

Diesel engine construction was a breakthrough in the field of transportation. Chemical industry was also associated with crude oil processing. Ports that enabled oil transport and large urban agglomerations, which represented large markets, played a special role. The Second Industrial Revolution allowed the use of the first automatic controllers powered by electricity, with relay logic, to represent the manufacturing program for On/Off, Normal/Fault, Over limit/Standard events. Till the end of the first half of the 20th century, control systems were mainly analog-based.

The economy of processes and maintaining stable climate conditions for technological processes were expected to improve. It was possible thanks to the use of automation and this ability contributed to its further development. Central control rooms started to be crucial for monitoring and adjusting processes. Logic was improved with closed-loop control and automation systems were perceived as nonlinear, containing errors from measurements and noise.

The Third Industrial Revolution, which began in the second half of the 20th century, relies on the automation and computerization of production. The use of the achievements of science and technology such as transistors, semiconductors, integrated circuits, and optical fibers was decisive.

The high technology industry is developing. For the Third Industrial Revolution the scientific background and qualified workforce creating the high technology industry play a role. Further development of HVAC automation is related to communication and the possibility of using processing control computers. In the 1970s, Programmable Logic Controllers (PLC) are introduced, which can simulate relay ladder logic. Over time, PLCs are becoming more powerful due to improving computing power and memory size. Electronic systems start to replace analog systems. Communication standards have been introduced to allow PLCs to talk with each other and to support real-time applications. The Supervisory Control and Data Acquisition (SCADA) software is introduced, which can monitor and manage increasingly complex systems within long-distance communications networks. Filed devices provide input information (e.g., sensors, transducers) and perform local operations (e.g., valve actuators, damper actuators).

Since 2003, when controllers with embedded web server were introduced, industry did not record any major automation event. This shows that progress takes place in a different field, namely, at the level of management, software development, communication, cybersecurity, and business models. Besides wired, wireless systems began to emerge and pass information over distances.

The fourth revolution is called industry 4.0. A cyber physical systems' network is emerging, often referred to as the Industrial Internet of Things (Industrial IoT). The changes also include solutions for the automation of industrial HVAC. The principles of physics remain unchanged, but HVAC control technology is changing faster than ever and the users' needs ultimately shape it. Types of control (analog, digital), ways of information collection and processing based entirely on real-time data from the plant (data-based control), data analysis tools and forecasting algorithms, are features which decide on precision and reliability of HVAC automation systems for production and operations quality. Digitizing the economy and creating a network of systems enables remote energy monitoring and optimization of its consumption as well as proactive maintenance of HVAC systems. Industrial HVAC automation systems are more and more often modular, based on TCP/IP protocols and use open-source operating systems. They are easier to integrate within different brands; however, issues with systems' cybersecurity and evolution to IT-centric solutions are still to be solved.

Higher value jobs are created to support ongoing need for innovation in HVAC control solutions. The most important features of the innovative HVAC automation systems are:

- interoperability—which enables communication between machines, objects, and people and
- virtualization—creating a virtual copy of a system for business continuity and disaster recovery.

Also, the nature of a system operating in the Industry 4.0 environment is changing: mesh networks

of wireless sensors, which provide accurate data on temperature, pressure, light levels, voltage, current, schedules, energy consumption, etc., transfer data to the edge controller that performs initial analysis and sends it to a cloud that uses external data sources, such as information from weather stations, dynamic electricity prices, other websites, databases, and workflows, for data analysis. The latter, workflows, determine how and when the HVAC system should be controlled, which is becoming more and more of a business type and dependent on demand. In addition, users need to be able to interact with the system. Thus IT security is seen as the most demanding and key attribute of the innovative HVAC automation system. Private information and connected systems must remain secure.

9.6.1 Methods for automation control

There are unfortunately no established rules or methods for how an industrial process can be controlled. The methods are usually the necessary and desirable functions and requirements for accuracy, which is necessary for an economical and quality product.

In providing a correctly optimized and accurate control system, all sections of the professional team involved in the project have to work together as a fully cooperating group. This includes persons who have a working knowledge of the process and those who are responsible for the automation and the electrotechnical aspects of the project.

The following control methods are recommended:

- Initially roughly design the systems.
- Survey the functions of the control system.
- Check the system requirements for the required accuracy.
- Select the control system and the user interface.
- Select the process subproduct.
- State the degree of responsibility for delivery and commissioning.

The user interface and the simplicity of usage are important issues. Likewise, stabilized and qualified control for some of the control loops must be as simple as possible.

Also, important is the interplay between different sensors, controllers, automation equipment, and objects regulated by the control equipment. The requirements of pumps, fans, batteries, heat exchanger, valves, motors, etc. in standard sizes may greatly differ from the theoretical calculations. Because of this fact, the control equipment, in addition to satisfactory control, must be capable of correcting the differences between the calculated and delivered subproducts.

An automated technical installation is normally no better than the weakest subproduct in the process, system, or installation.

9.6.2 Main types of control equipment and automation level

The type of control system and instrumentation to be selected will depend on its location (e.g., the danger of fire, explosion, pollution, moisture, the conditions of temperature, and its variations). The control equipment can be classified as:

1. electromechanical and electronic,
2. data-based control by PLC, widely used in industrial automation, or Direct Digital Controller (DDC), which is dedicated for building automation,
3. pneumatic control, and
4. mixed electronic-pneumatic system.

The level of automation depends on the requirements for the technical operation of the attendance system, the need for communication with other processes and data systems, and the requirements for electrical and pneumatic components and cabling methods.

9.6.3 General technical requirements

The control and subproducts for the instrumentation are governed by the general technical requirements for the other subproducts in the system. In addition, to fulfill all the general technical requirements, the encapsulation and protection of the environments in which the subproducts are used have to be considered, including danger of explosion, pollution, moisture, temperature, vibration, and influence on heating and cooling. See Table 9.11.

9.6.4 Automation equipment and instrumentation

Control and instrumentation can be defined as subproducts for measuring, sending messages, controlling, and regulating. These are installed locally in each system, in the installation as a whole, or in the control technical room.

To ensure flexibility for possible later additions or changes, the instrumentation has to be selected based on standardized subproducts using national or international norms. These standards are mainly based on the measuring areas and signal types. Typical signals for input and output to the data-based control (for DDC and PLC) are:

- measuring and control signals as 0–10 VDC, 0(4)–20 mA, or 0.2–1 bar and

TABLE 9.11 Important information and requirements for control and instrumentation.

Power supply to buildings	3-phase 400 alternating current voltage (VAC) ±10% 50 Hz
Type of control	Data-based
Input and output modules	0(2)–10 direct current voltage (VDC) or 0 (4)–20 mA
Climate surroundings	0°C to +50°C and 10%–90% relative humidity
Electronic norms/ requirements	IEC 60364/CENELEC HD384
Level of encapsulation	IP54 (local components)

- sensors with receptor of thermal element, platinum or nickel (Pt 100, Pt 1000, Ni 1000).

Other important parts of the instrumentation are:

- local electromagnetic and pneumatic components, damper, and valve actuators;
- fuses, relays, and contactors for control of pumps, fan, light, energy, etc.; and
- electrical safety.

9.6.5 Process

The term *process* is normally used in connection with the control of a plant in industry; the term, however, also describes the HVAC installation in industry and buildings.

The word *process* is from Latin and defines many different aspects. The automation process relates to the automatic regulation required to control the physical conditions in a system. The term *process* relates to both the optimal conditions within which the operation is maintained and when the operation varies with time or by a predetermined plan.

9.6.6 Controller

Until about 1995 mainly the analog controller was used for controlling an HVAC process. Since then, the digital and data-based controller has taken over (Fig. 9.54). In principle, there are no major control technical differences between an analog and similar digital controller. The digital data-based controller has the following advantages:

- small size, requiring a relatively small installation area
- accurate reading of the measuring signals
- keyboard for accurate adjustment of different parameters
- signals for internal defects and external status and alarms
- a possibility for communication with the computer system

Depending on the control system and components selected, the process variable from the sensors and signals to the actuator control unit is connected either directly or through transducers. Normally, the input and output are adjusted from pneumatic sensors and control units. The signals produced then are transformed into standard electronic values before connecting to the controller's input and output modules.

DDCs and PLCs are used for data-based control and management of energy, lights, cooling, heat, and ventilation in buildings and industrial plants respectively. The term *DDC* was used at the beginning of the computer age and derives from American English, and is an abbreviation of direct digital control. This term was used to separate the original industrial controller PLC (program logic control) from more specialized DDC controllers used for HVAC processes in buildings.

The modern DDC controller can realize three types of control: P (proportional), PI (proportional-integral) and PID (proportional-integral-derivative). PLC controllers used in process installations may contain more complex regulation functions, for example, fuzzy or auto-tuning of PID functions. Most DDC controllers are self-sufficient and independent of the controllers or computer programs that are used for system configuration.

They are suited for the technical (HVAC) processes in buildings and include installed and tested computer programs, which cover the most common forms of controlling and regulating in a typical HVAC plant, such as:

- feed-forward (compensating) of air or water temperature from outdoor temperature or for input or extract systems;
- cascade controllers used where the supply and ETA temperatures or return water from heater or cooler batteries requires complete control;
- frost protection of heating coil (where water is used as an energy carrier for heating or cooling);
- functions for selecting an average of the lowest or highest of different measuring values
- calculations and mathematical functions;
- PLC functions for controlling lights, electrical motors, heating coil, boilers, refrigerating machines, etc.;
- controlling from internal time programs (daily, weekly, or monthly clock with extra program for vacations);

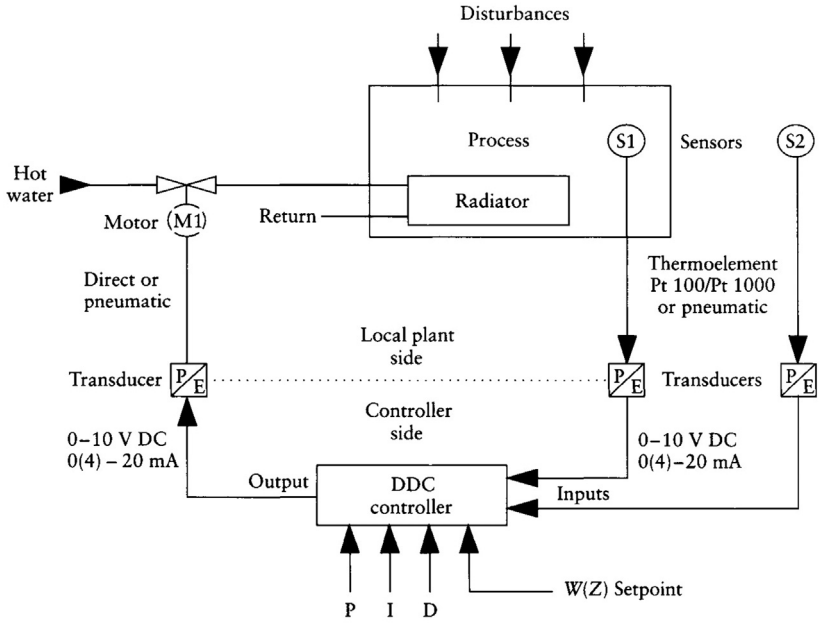


FIGURE 9.54 The controlling loop with sensors, actuators, and controller.

- optimally starting and stopping for the override of heating, cooling, and ventilation;
- protector for storage of hour rating and measuring values (trend logs);
- theft-proof admission cards and alarms for water leaks, etc.; and
- integrated network for connecting to BAS systems (central operation control).

Analog inputs and outputs in DDC controllers are usually standardized and record the signals from sensors or transducers as 0(2)−10 VDC or 0(4)−20 mA. The inputs module can also be standardized for resistances, such as Pt 100 DIN, Pt 1000, or Ni 1000 DIN (Pt = platinum, Ni = nickel).

DDCs or PLCs controllers with universal inputs and outputs can be used. Depending on the manufacturer, these inputs and outputs can be configured before system commissioning or adjusted after via software. In some cases the input may be used only for temperature measurement from special types of thermistors. (Thermistors are constructed from semiconductor materials where the resistance changes reversibly proportional to the temperature, that is, a negative temperature coefficient.)

9.6.7 The choice of controllers

Controllers for decentralized control and regulation are usually positioned locally to the process and connected to the control room.

They attend to and control different technical plants and systems, and they have to deal with all processes required. In data-based plants, controllers usually communicate with one or several local or central computers.

If there are special requirements for communication at all technical data levels, the local controller must still be chosen from the same factory as the other technical operating programs in the central computer or personal computer (PC). If not, there will be a considerable reduction in the amount of data sent to and from the controller.

In addition to controlling and regulating HVAC plants, DDC controllers have considerable internal memories for the storage of important system data and trend tables. Other important requirements are the interface for using and attending. If the controller is equipped with a graphic dynamic display, service selectors (keyboard) or light diodes for input and output modules, separated operation selectors, and operating or error lights in the electrical distribution may be ignored. The costs for both the new system and future maintenance can be considerably reduced.

The market consists of a considerable assortment of PLC and DDC controllers. Most of the factory types are capable of controlling separate technical plants in a satisfactory manner.

The PLC controller mentioned earlier is used basically for regulating industrial processes or to replace relay controllers (latching) in electrical processes. Plants with local or central computers are used for operating and controlling energy requirements of HVAC plants. It is impossible to state that the PLC is capable of giving the required results, especially when they are connected in a network and function as self-operating controllers.

TABLE 9.12 Physics behind the measurements.

Parameter	Physics behind the measurements
Temperature (°C)	Resistance thermometer or RTD (e.g., Pt 100; platinum 100 Ω), semiconductor-based sensor, thermistor, NTC, PTC, thermocouple, thermo camera, and many more
Humidity (%)	Capacitive, resistive, thermal, optical hygrometer.
Pressure (Pa)	Hydrostatic (hydrostatic gauges, piston, liquid column as manometer), aneroid (bourdon gauge, diaphragm, bellows, magnetic coupling), spinning-rotor gauge
Sound (dB)	Dynamic, electrostatic, and piezoelectric according to their conversion system
CO_2 (ppm)	NDIR and chemical gas sensor with sensitive layers based on polymer or heteropolysiloxane
PM ($\mu g/m^3$)	Optical particulate counter, gravimetric sampler, surface area diffusion charging sensor
Airflow (m^3/h)	Thermal mass flow meters or mass flow controller. Pressure-based meters: Venturi meter or Dall tube, orifice plate, Prandtl tube, multihole pressure probe, linear resistance meters, and ultrasound Doppler

NDIR, Nondispersive infrared sensor; *NTC*, negative temperature coefficient; *PTC*, positive temperature coefficient; *RTD*, resistance temperature detector.

Many PLCs can have few and simple data programs, together with a limited internal memory capacity. Due to this fact, it is necessary in many cases to provide extra software and greater memory capacity in the central data station than for more specialized and autonomous DDC controllers. Another reason for selecting the DDC controller is that the skills of the firms who are producing and using DDC controllers are probably better adjusted to the special processes in the HVAC plants which are generally installed in industry and buildings.

9.6.8 Sensors

Sensors are placed in the process for measuring status and changes in their environment. The purpose could be for controlling processes such as space heating or for measuring conditions like room temperature. In industrial ventilation the most common sensors are used for measuring temperature, humidity and pressure of fluids, vibrations and noise levels, current and fluid flows, light level and its color, detecting presence and motion, gases, such as CO_2, refrigerants, and particulate matter. Depending on what physical factors are being measured a wide range of principles are used for the sensors. When choosing the right sensors, it is especially important to look at the accuracy of the measurement, hysteresis, the reaction ability (time constant or response time), linearity, and repeatability (see Table 9.12).

Mounting and housing for protection is a critical factor when the sensor is placed in an aggressive environment. Even if the sensor by itself has high accuracy, it may be unable to execute the measurement in a defined place. Quality and total accuracy depend on the combination of sensor, the converter for measured values (transducer), and mechanical protection. The mechanical protection can take the form of pockets in water and fluid or assembly boxes that protect against pollution, humidity, and temperature in the surroundings or against electromagnetic transmissions and noise from power-supplied pipes and cables. Protection may affect the accuracy of the measurements and should be considered with care.

9.6.9 Placing of sensors in HVAC systems

The sensors must be installed in a correct and representative place in the process. Determining an optimal installation of sensors for measuring environmental conditions in large halls is not a simple task. Many different factors have to be taken into account. The main place where a certain climate parameter is to be maintained is given priority. Second, the influence of disturbances like for instance air infiltration and energy radiation from surrounding surfaces must be considered. Good recommendations are available in international and national standards for measurements in different parts of a ventilation system and should be followed. For instance ISO 5167−2:2003 Measurement of fluid flow or ISO 5801:2017Fans—Performance testing using standardized airways or ASHRAE STD 41.1 2013 Standard Method for Temperature Measurement and many others.

The influence of airflows from mechanical ventilating systems (jets) and as well as natural stack effects or crossflows must also be considered to choose the placement of the sensor. Stratification should also be considered in halls ventilated with displacement ventilation at the time to measure temperatures and concentrations of pollutants. Processes using liquid or gasses of different physical qualities when mixed may experience separation into different layers. The level of fluid in containers and tanks is due to stratification of horizontal temperature layers, while airflow after heating

and cooling coils, heat-recovery systems, and humidifiers or dehumidifiers might separate into parallel layers. Thus depending on where one should be measuring, one or several measurement points are required to achieve a correct measurement.

Depending on the accuracy required, the pressure difference across components in a flowing gas or fluid is measured by sensors for pressure difference. The quantity of flowing medium can be determined by measuring the flow velocity in pipes or ducts. Sensors for measuring the velocity of gases or liquids are dependent on the nature of the flow and its turbulence. If accurate measurements of the velocity of gas or liquid are required, the sensors have to be installed in straight runs of the pipe or duct and possibly after straighteners so that the flow is fully developed. Regarding measurements of airflow rates in a duct, they should be done after a straight section from the last bend or other disturbance. The straight length is dependent on the pipe/duct size and use of flow straighteners, normally longer than five times the diameter of the pipe or the duct. One should measure in a series of points representing equal area so that the real airflow rate can be calculated (traverse readings). Both velocity and pressure differences can be transformed into respective values by mathematical formulas in the function modules of the DDC controller.

Modern ventilation systems are often built in a compact way. Frequently, due to the smaller-space installation, inspection, and maintenance of the sensors can be difficult. In ducts, deviations from the real values of measurements occur due to the sensor being exposed to moisture or heat or cold radiation from the surroundings. Choosing the incorrect sensor or wrongly positioning it can lead to instability and oscillations in the control system. The oscillations influence the process and make control difficult. In offices the airflow rate to each room is often controlled by a damper operated as a function of the room temperature. Sensors that measure the CO_2 concentration in the room air provide an extra mode of control, in addition to that of air temperature. CO_2 sensors should be always placed at breathing height. When used for demand-controlled ventilation (DCV), a single sensor per room should be placed and a common CO_2 sensor in the return flow to avoid inaccuracy.

Sensors or receptors installed in ducts or pipes are placed after the filter sections of the air or fluid to avoid sensor pollution. Temperature sensors for liquids in pipes must be installed in pockets that are directed against the flow. At least two-thirds of the sensor pocket has to be in contact with the fluid. If the pipe diameter is less than the sensor length, the pocket can be installed at an angle less than 90 degrees against the direction of flow to provide the necessary contact area.

In buildings that are divided into zones with a central heating system, it is common to adjust the water temperature depending on the outdoor temperature. In this example a function called feed-forward or outdoor temperature compensating is used. Fig. 9.55 shows how the water temperature changes as a function of the outdoor temperature.

Sensors TS 1−2-4 regulate the heating and cooling coils in a sequence to achieve the required temperatures (Fig. 9.56). Valves for heat recovery are controlled by a frequency converter RC1 (room criterion 1) for the pump motor. When a greater output is required from the heating coil, the pump motor speed increases before the valve MV2 opens. If the extract temperature is lower than the outdoor temperature, the speed of the pump motor increases before valve MV1 opens. To avoid ice formation at low outdoor temperatures the sensor TS 7 operates on a lower limit, depending on the demands of the battery in the exhaust.

Nowadays, it is more and more common to achieve required climate conditions with a feedback closed-loop, realizing demand-response strategy.

Saturated air is difficult to measure accurately; a deviation of 3−5 K must be accepted. In some air-conditioning systems, very humid air may condense on surfaces below the dew-point temperature of the air. Poorly insulated ducts containing humid air can cause serious problems. See Fig. 9.57. Dew-point sensors help to detect these situations.

The speed of the fans is controlled by sensors PS1 and PS2 and frequency converters RC1 and RC2 (Fig. 9.58). The sensors measure the difference in pressure between duct and atmosphere outside and maintain constant pressure in the ducts.

9.6.10 Changing speed by using frequency converters

The speed of an electric motor can be changed by altering the frequency of the electric current. This is because the ratio is the same as 60 or $50\,f/p$ (f = the frequency of the current, p = the number of poles in the stator). Frequency converters are built of electronic components, frequently combined with microprocessors. They provide good motor protection and are superior to the traditional bimetal protection. The characteristic curve for a pump and fan motor is also quadratic, making lower demands to the frequency converters

When the frequency of the electrical current is changed in the frequency converter, the main AC supply is transformed into DC. The DC is then treated in the frequency converter's components, before being transformed back into AC at the required frequency.

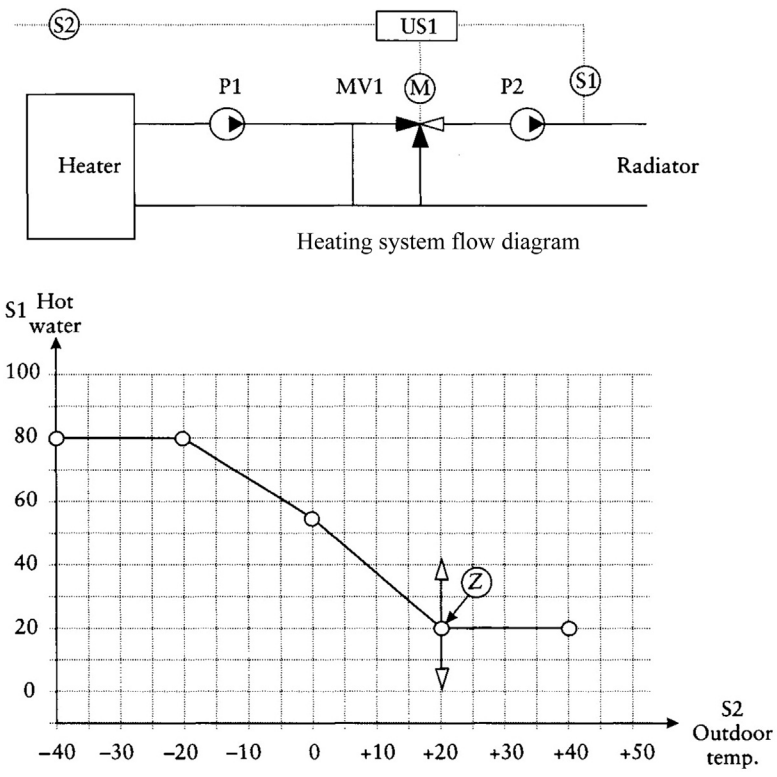


FIGURE 9.55 A central heating system supplying low-pressure hot-water radiators in a building.

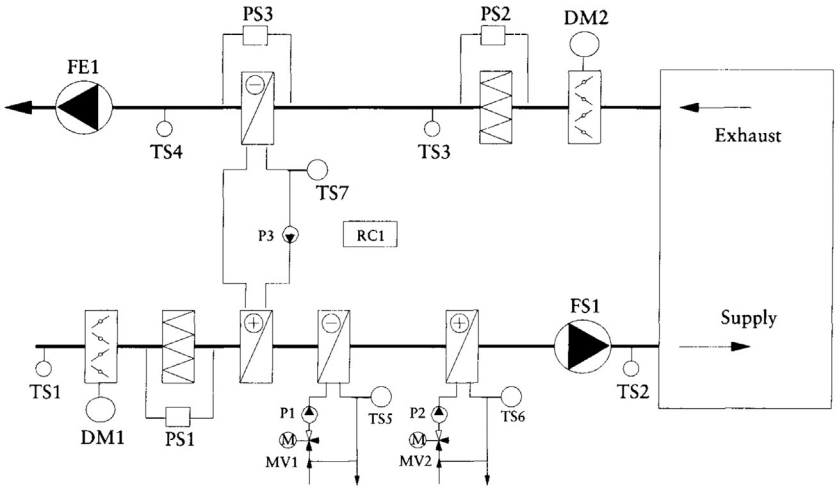


FIGURE 9.56 A ventilation system with provision for heating and cooling batteries.

Frequency converters constructed for use with three-phase motors less than 4 kW can therefore be delivered with a single-phase supply from the electrical mains. For greater power requirements, it is most common to use a three-phase power supply. Through switches and potentiometers or the programing module of the frequency converter, the important parameters of electrical motors can be adjusted as needed.

The advantage of using frequency converters is that the possibility exists to use a ramp function when starting and stopping the motor (soft start). By using this function, it is possible to avoid starting both fans at full speed with closed dampers; it also reduces stresses on the fan transmission (belts) at the start.

9.6.11 Building the control station

When local or central computers are used for controlling the operation of HVAC installations in buildings, they are described as building automation

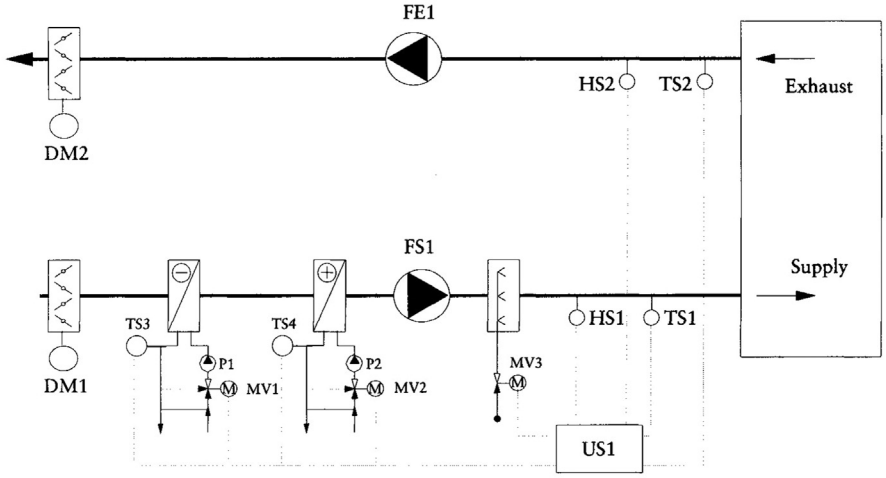


FIGURE 9.57 A ventilation system for controlling the air temperature by humidifying and dehumidifying the air.

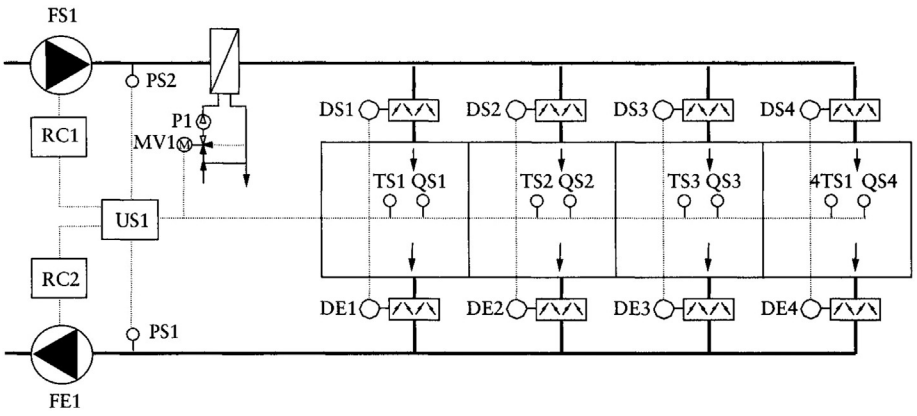


FIGURE 9.58 A VAV ventilation system. *VAV*, Variable air volume.

systems. In the control station, operators communicate with control installations, which are connected through the computer plant. Computers for individual use are called PCs. The use of PCs for technical and economic tasks in the past few years has increased considerably. The business overflows with PCs, which get greater data power and memory at lower prices.

Building control systems are typically classified as:

- local instrumentation (control system);
- data-based local controllers; and
- control station with computers, mainframe, or PC.

Because of these solutions, important technical data can be transferred from local instrumentation (control system) through data-based controllers to a control station with computers. The operator may use the many variations that the software data system provides. Technical data operation may be digital off/on messages such as the status of operation and the performance of alarms or analog measurements such as temperature, humidity, pressure, velocity, and energy usage.

From the operator's PC in the control station the operator should be able to perform the following operations:

- Switch on and off electrical lights.
- Control the heating and ventilating systems.
- Switch on and off and control the speed of pump and fan motors.
- Control valves and dampers.
- Detect the condition of and operate filters and dust-cleaning devices.
- Receive and register status and alarm messages.
- Read measured values (temperature, energy, etc.).
- Change set-point for regulation.
- Receive and register energy information, and record the trend log.
- Control and regulate the power consumption.

9.7 Air distribution system, ductwork

9.7.1 Friction loss calculation

These are the general principles:

- No internal friction.
- Noncompressible gas.
- Isothermal.
- Stationary.
- Bernoulli theorem is applicable.

$$p + \frac{\rho v^2}{2} + \rho g h = \text{constant} \tag{9.138}$$

Because the influence of gravitation ρgh is negligible for horizontal (distribution) systems, this term is ignored in the equations.

Friction losses in ductwork are

$$\Delta p = \frac{\lambda L}{d} + \frac{\rho v^2}{2}, \tag{9.139}$$

in which λ is friction factor, dimensionless; L is length, m; d is internal diameter, m; ρ is density, kg/m^3; and v is average air velocity, m/s.

The friction factor λ for laminar flow ($Re \leq 2300$) is

$$\lambda = \frac{64}{Re},$$

where Re is Reynolds number: $Re = ud/v$, in which u is the average air velocity and v is the kinematic viscosity.

For turbulent flow the empirical formula of Colebrook–White applies:

$$Re > 3500$$
$$\frac{1}{\sqrt{\lambda}} = -2\log\left(\frac{\epsilon}{3.72D} + \frac{2.51}{Re\sqrt{\lambda}}\right). \tag{9.140}$$

This expression is difficult to use, as iteration is required. A simplified expression can be used with sufficient accuracy:

$$\frac{1}{\sqrt{\lambda}} = -2\log\left(\frac{\epsilon}{3.72D} + \frac{5.74}{Re^{0.901}}\right). \tag{9.141}$$

The formulas are represented in the Moody diagram, which allows a quick solution.

In the transient field where $2300 < Re < 3500$, the flow may be laminar or turbulent, and λ is expressed by the following formula:

$$\lambda = \frac{\lambda_{2300}(3500 - Re) + \lambda_{3500}(Re - 2300)}{3500 - 2300},$$

in which λ_{2300} and λ_{3500} are the calculated λ values at $Re = 2300$ and 3500, respectively.

9.7.1.1 *The surface roughness factor* ϵ

This factor is material dependent. The values in Table 9.13 could be applied.

The value for flexible plastic ducts (* in the table) can be estimated by

$$\lambda = \frac{1}{5}\sqrt[10]{(s/d)^6(S/s)^7} \quad \text{if } Re \leq 5\cdot 10^4, \tag{9.142}$$

in which d is the internal diameter, m; s is the depth of the winding, m; and S is the distance of the windings, m.

Hydraulic diameter

The hydraulic diameter is four times the flow area divided by the duct perimeter.

The formulas given before show the diameter d.

For rectangular and oval ducts a corrected hydraulic diameter should be used.

$$d_h = 4\frac{A}{P}, \tag{9.143}$$

in which A is surface area of the duct, m^2 and P is perimeter per unit length of the duct, m.

Pressure loss due to local resistance

$$\Delta p = \xi \cdot \frac{\rho v^2}{2} \text{ (Pa)} \tag{9.144}$$

where ξ, the local friction resistance factor, depends on the geometrical shape of the ductwork and flow path through the various fittings used in duct systems.

Values for ξ are given in standard handbooks and are based on experimental measurements.

For computer applications, it is useful to have the friction factor in a mathematical expression (empirical).

Theoretical background of ξ. One example for the case of a round collecting T-piece at 45 degrees is shown in Fig. 9.59.

The impulse balance along the x axis is

$$\rho q_{v1} v_1 \cos\alpha + A_2 p_2 + \rho q_{v2} v_2 = A_3 p_3 + \rho q_{v3} v_3 \tag{9.145}$$

in which ρ is density, kg/m^3; q_v is volume flow rate, m^3/s; v is air velocity, m/s; and A is surface area, m^2.

Because

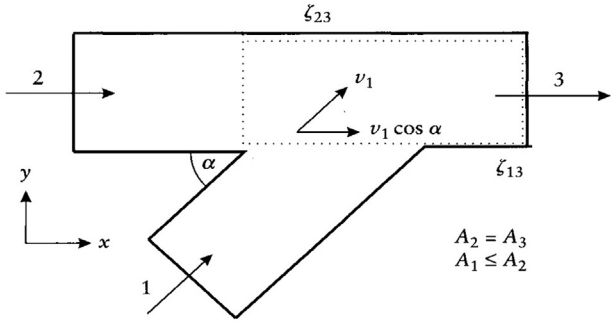


FIGURE 9.59 Round collecting T-piece, 45 degrees.

$q_{vi} = q_{v3} - q_{v2}$
$A_2 = A_3$
$p_1 = P2,$
$p_2 - p_3 = \rho v_3^2 - \rho v_2^2 - \rho v_3 v_1 \cos\alpha + \rho v_2 v_1 \cos\alpha.$

Bernoulli's law gives

$$p_2 - p_3 = \frac{\rho v_3^2}{2} - \frac{\rho v_2^2}{2} + \Delta p \tag{9.146}$$

$$\Delta p = \frac{\rho v_3^2}{2} - \frac{\rho v_2^2}{2} - \rho v_1 v_3 \cos\alpha + \rho v_1 v_2 \cos\alpha \tag{9.147}$$

The friction loss in this branch is expressed as

$$\Delta p = \zeta_{23} \frac{\rho v_3^2}{2} \tag{9.148}$$

$$\zeta_{23} = 1 - 2\cos\alpha\left(\frac{v_1}{v_3}\right) - \left(\frac{v_2}{v_3}\right)^2 + 2\cos\alpha\frac{v_1}{v_3}\frac{v_2}{v_3}. \tag{9.149}$$

Using the conservation law, a similar expression can be derived for other connection pieces. The general formula structure is

$$\varsigma = a_0 + a_1\left(\frac{v_1}{v_3}\right) + a_2\left(\frac{v_1}{v_3}\right)^2 + a_3\left(\frac{v_2}{v_3}\right)^2 + a_5\left(\frac{v_1}{v_3}\frac{v_2}{v_3}\right). \tag{9.150}$$

where a_1–a_5 are regression factors, calculated on the basis of the measured values.

9.7.2 Design methods

- Constant friction method
- constant velocity method
- gradual velocity reduction method
- pressure recovery method
- balanced pressure loss method

The design methods are normally a combination of one or more of the above methods.

- Constant friction method. The principle is that the friction losses per meter run of the duct are taken as being constant. Values in the 1–5 Pa/m range are typical. The friction losses of connecting elements are expressed as an equivalent length of the straight ductwork runs. This is a simple method. It is ideally suited as a preliminary design method where it is combined with another approach.
- Constant velocity method. This is a simple but not very cost-effective approach for systems with a wide range of duct diameters.
- Gradual velocity reduction method. This method is a variation of the constant friction approach, where a maximum velocity is used for the main and branch ducts. This procedure provides a reasonable solution and choice between the velocity, diameter, and resistance. The method is not useful to provide the same static pressure at each outlet.
- Static pressure recovery method. The diameters are selected in such a way that the same static pressure is available before every connection. The duct reduction is selected in such a way that the gain of static pressure is in balance with the friction losses up to the next connection point. This method may result in fewer control devices at connection points or outlets. Low velocities and large diameters at the end of the system may be the result of this design approach.
- Balanced pressure loss method. This is a method that is used to improve the performance of a preliminary design of a duct system and reduce the number of control dampers.

9.7.2.1 Boundary conditions

Apart from the energy constraints due to the friction losses, the main reason for limiting the air velocities in ducts is to reduce noise production. Some recommended air velocities are given in Table 9.9.

9.7.3 Thermal losses by transmission

9.7.3.1 Circular ducts

$$\varphi_i = U^*(\theta_{ad} - \theta_a)$$
$$\frac{1}{U^*} = \frac{1}{\pi h_i D_i} + \frac{1}{2\pi\lambda_{is}}\ln\frac{D_i + 2d_{is}}{D_i} + \frac{1}{\pi h_u(D_i + 2d_{is})}, \tag{9.151}$$

in which φ_i is the linear heat flux, W/m; U^* is the linear U-value, W/(m K); θ_{ad} is the air temperature in duct, °C; θ_a is the surrounding air temperature, °C; h_i is the heat transfer coefficient inside, W/(m^2 K); D_i is the internal diameter, m; λ_{is} is the heat conduction coefficient of insulation, W/(m K); d_{is} is the insulation thickness, m; and h_u is the heat transfer coefficient outside, W/(m^2 K).

9.7.3.2 Rectangular ducts

$$\frac{1}{U_A^*} = \frac{1}{\pi h_i a} + \frac{1}{2\pi\lambda_{is}}\ln\left(\frac{a + d_{is}}{a}\right) + \frac{1}{\pi h_u(a + 2d_{is})}, \tag{9.152}$$

in which a is the internal width of the duct.

This formula is valid for rectangular ducts. For the parallel extension part, one could calculate

$$\frac{1}{U_B^*} = \frac{1}{h_i} + \frac{d_{is}}{\lambda_{is}} + \frac{1}{h_u}. \tag{9.153}$$

The total U^* is calculated as

$$U^* = \frac{2aU_A^* + (b - a)U_B^*/P}{a - b}, \tag{9.154}$$

in which b is the internal height of the duct and $P = 2a + 2b$ is the perimeter length, m.

The energy loss calculation for air temperature losses is

$$\theta_{end} = \theta_a + (\theta_{start} - \theta_a)e^B, \quad (9.155)$$

in which θ_{end} is the air temperature at the end of the duct system, °C; θ_{start} is the air temperature at the start of the duct system, °C; θ_a is the surrounding air temperature, °C; and B is calculated as

$$B = -\frac{U^* L}{q_v \rho c_p}, \quad (9.156)$$

where U^* is the U value of duct, W/(m K); L is the length of duct, m; q_v is the volumetric airflow, m^3/s; ρ is the density of air in duct, kg/m^3; and c_p is the specific heat of air in duct, J/(kg K).

9.7.4 Air leakage from ductwork

It is of prime importance to keep air and gas leakage from ductwork at a minimum, as it represents increased fan running cost and the waste of treated air. Leakage into extract ductwork reduces the efficiency of the collection system.

9.7.4.1 Leakages of air distribution systems

Testing of installed systems in situ is described in EN 12599.[24] The testing of ductwork and components is listed in the product standards. The classification and testing of airtightness of round ducts are defined in EN 12237,[25] of rectangular ducts in EN 1507,[26] of flexible ducts in EN 13180,[27] of dampers and valves in EN 1751,[28] of nonmetallic ducts made from insulation ductboards in EN 13403.[29]

The minimum requirements shall be specified according to Table 9.10.

The leakage class of the duct system shall be classified according to Table 9.10. The leakage class shall be verified according to EN 12599.[24] In large systems and systems in which all components are tested and classified, testing only parts of the system is sufficient according to EN 12599.[24]

All leakages will increase the energy use of the air distribution system. This increase depends also on the ductwork surface area and average pressure difference (positive or negative). Table 9.13 gives an example of the fan power increase in a typical air distribution ductwork. The ductwork surface area can be estimated according to EN 14239.[30]

TABLE 9.13 The surface roughness factor ϵ.

Duct type	Material	ϵ (10^{-3} m)
Seamless ducts	Steel	0.045
	Aluminum	0.045
	Plastics	0.01
Spiral-type ducts	Galvanized steel	0.15
	Stainless steel	0.15
	Aluminum	0.15
Ducts with beams	Galvanized steel	0.07
	Stainless steel	0.07
	Aluminum	0.07
Flexible ducts	Metal	0.5–3
	Plastics	–
Internally insulated ducts	Coated mineral wool	0.25
Masonry ducts	Concrete	2
	Brick	3

9.7.5 Ductwork components for safety in ventilation

The aim of this chapter is to briefly cover the areas of fire and explosion which can occur in Industrial environments. It is essential that the ventilating engineer designs the system to current standards and mandatory requirements. Three areas are covered cover:

1. fire dampers
2. hazardous areas
3. pressure relief dampers

9.7.5.1 Fire dampers and smoke control dampers

Fire dampers

Many installations will have ducts running through them for ventilation purposes. These ducts will generally be steel but may be of other materials (e.g., PVC) depending on the atmosphere or application. These ducts (even those made from steel) have no inherent fire resistance and would be expected to collapse in a fire. In the past there have been "deemed to satisfy" allowances for say 0.8 mm thick steel, but as it is not simply the duct that fails during a fire test, but also the hanging system. Consequently, these "deemed to satisfy" allowances are no longer acceptable (see fire-resisting and smoke control ducts in the later sections).

Where the ventilation ducts pass through a compartmentation structure such as a wall or floor, the compartmentation must be protected with a fire damper. The fire damper protects and maintains the compartment, not the duct.

Generally, the fire damper should be installed in the plane of the wall or floor. In all cases, extreme care must be taken to follow the manufacturer's prescribed installation methods. Care must also be taken when

substituting one manufacturer's product for another's, at a later stage in a project, because installation methods vary by manufacturer.

Some products may have been tested to be mounted on the surface of the wall or floor and (extremely rarely) may have been tested to be mounted in the duct away from the wall. If there is no method for mounting on the wall or away from the wall, then this should not be specified or used.

For dampers installed away from the wall, simply insulating the duct in some way is, again, not acceptable. This is a method for making the duct fire resistant itself and the most important detail here will be how the duct is strengthened and penetration sealed at the compartment wall or floor (see fire-resisting ducts in a later section).

Fire dampers for sale in the EU and the United Kingdom should be CE marked and come with a Declaration of Performance (DoP). This follows the requirements of a harmonized standard EN 15650. As fire dampers are a fire safety product, they must have third party testing to EN 1366-2 and product certification from notified bodies to be allowed on to the market.

The fire dampers may then be classified (EN 13501-3) and various parameters stated. Any fire damper should have an integrity (E) time and may be combined with an insulation (I) characteristic and/or a reduced leakage (S) classification where this is a requirement. Any damper without an S classification will have no specific requirement to have a known ambient leakage characteristic. The ability for the damper to be used either vertically (horizontal duct) or horizontally (vertical duct) or both will be made clear, together with the ability for it to be used either way round or with fire exposure form just one side (not generally likely). Finally, there may be several operations that may be expected, so if the damper is expected to change position at least one the $C_{10,000}$ classification should be selected. Table 9.14 shows typical classification of fire dampers.

Some changes to the original products may be allowed under extended field of application (EXAP—EN 15882-2) rules. This is a simplified example (for explanation only) to show what should be proposed. More information should be sought directly from manufacturers, noting that manufacturers without third party certification, not offering CE marking or a DoP should be avoided, as they are not demonstrating competency in these products.

TABLE 9.14 Classification of fire dampers.

Fire damper: typical classification	**EI 90S v_{ex} h_{ox} $i \leftrightarrow o$ $C_{10,000}$**
	E—Integrity—minimum requirement
	I—Insulation—optional to suit application or local requirements
	90—Time (min)—60, 90, 120
	S—Reduced leakage—optional to suit application or local requirements
	v_e—vertical mounting
	h_o—horizontal mounting
	$i \leftrightarrow o$—tested both ways round—options
	$C_{10,000-\text{number of operations}}$—options 300, MOD (modulating)
Product standard	EN 15650
Test standard	EN 1366-2
Classification standard	EN 13501-3
EXAP standard	EN 15882-2

Note: Fire dampers for use in hazardous areas should be separately ATEX certificated and note that the actuator forms part of the assembly and the assembly needs to have been fire tested and CE marked—actuators cannot simply be exchanged.

Smoke control dampers

Smoke control dampers differ from fire dampers in several ways. Until an incident occurs it is not known where the smoke will be. At the smoke signal, all the smoke control dampers along the route of extract must open and stay open, all other smoke control dampers must close and stay closed to maintain compartmentation and stop smoke and fire spread into other compartments. Consequently, smoke control dampers must be under the control of a system with protected poser supply and have no fusible links or other devices to allow them to move from their smoke position, whichever it is.

Single compartment smoke control dampers working up to 600 degrees. C may be available, but these can only be used in systems where the ducting or shat goes directly to the outside without crossing a compartment barrier.

Smoke control dampers may be used in conjunction with smoke control or fire-resisting ducts as well as in the plane of walls or floors. In all cases, extreme care must be taken to follow the manufacturer's prescribed installation methods. As with fire dampers, care must also be taken when substituting one manufacturer's product for another's, at a later stage in a project, because installation methods vary by manufacturer.

Smoke control dampers for sale in the EU and the United Kingdom should be CE marked and come with a DoP. This follows the requirements of a harmonized standard EN 12101-8. As smoke control dampers are a fire safety product, they must have third party testing

to EN 1366-10 and product certification from notified bodies to be allowed on to the market.

The smoke control dampers may then be classified (EN 13501-4) and various parameters stated. Any smoke control damper should have an Integrity (E) time and may be combined with an Insulation (I) characteristic and/or a reduced leakage (S) classification where this is a requirement. The ability for the damper to be used either vertically (horizontal duct) or horizontally (vertical duct) or both will be made clear, together with the ability for it to be used either way round or with fire exposure from just one side (not generally likely).

A guide to what pressure the smoke control damper may be subjected to at ambient is given and whether it must respond to alarms automatically and immediately or they can be left for a certain time, as the actuators are protected against fire temperature rises.

Finally, there may be several operations that may be expected, so if the damper is expected to change position at least one the $C_{10,000}$ classification should be selected. Table 9.15 shows the typical classification of smoke control dampers.

This is a simplified example (for explanation only) to show what should be proposed. More information should be sought directly from manufacturers, noting that manufacturers without third party certification, not offering CE marking or a DoP should be avoided, as they are not demonstrating competency in these products.

(*Note*: Smoke control dampers for use in hazardous areas should be separately ATEX certificated and note that the actuator forms part of the assembly and it is the assembly that needs to have been fire tested and CE marked—actuators cannot simply be exchanged.)

TABLE 9.15 Classification of smoke control dampers.

Smoke control damper: typical classification	**EI 90S v_{ex} h_{ox} 1000 AA $i \leftrightarrow o$ $C_{10,000}$ multi**
	E—Integrity—minimum requirement
	I—Insulation—optional to suit application or local requirements
	90—Time (min)—60, 90, 120
	S—Reduced leakage—optional to suit application or local requirements
	v_e—vertical mounting
	h_o—horizontal mounting
	1000—Ambient pressure—options 500, 1500
	AA (option—MA)
	$i \leftrightarrow o$—tested both ways round—options
	$C_{10,000-\text{number of operations}}$—options 300, MOD (modulating)
	Multi—multicompartment (option—single)
Product standard	EN 12101-8
Test standard	EN 1366-10
Classification standard	EN 13501-4
EXAP standard	None

No EXAP document is available at present but might become available in the future. *AA*, Automatic activation; *MA*, manual activation. *AA*, Automatic activation; *MA*, manual activation.

9.7.5.2 *Fire-resisting ducts and smoke control ducts*

Fire-resisting ducts

Some ventilation ducts may be specified as fire resisting and may also be used to maintain compartmentation. They should be tested from inside to outside and outside to inside. The most important parameter again is installation using the correct instructions and making sure that strengthening of the duct at the wall, the penetration seal at the wall around the duct, and the correct hangers are used. Different suppliers/installers/manufacturers have differing details due to the testing that has been done.

If supplied in the form of sections there may be a requirement for the sections to be CE marked at some point, but at present the standard is not yet published. However, testing should have been performed to EN 1366-1 and a classification made to EN 13501-3. Table 9.16 shows typical classification of fire-resisting duct.

This is a simplified example (for explanation only) to show what should be proposed. More information should be sought directly from manufacturers, noting that manufacturers without third party certification, and not offering a DoP should be avoided, as they are not demonstrating competency in these products.

Smoke control ducts

As an extension to fire-resisting ducts, there are special ducts to allow the extract of smoke from compartmented areas. When a smoke control damper opens, it causes the smoke control duct connected to it to become part of the compartment, so the smoke control ducts have some additional requirements, not least of which is that they must not collapse.

Multicompartment smoke control ducts should be first tested to EN 1366-1 from inside to outside and outside to inside and then further tested to EN 1366-8. The same rules apply, and the most important parameter again is installation using the correct instructions and making sure that strengthening of the duct at the

TABLE 9.16 Typical classification of smoke control dampers.

Fire-resisting duct—typical classification	**EI 90S v_{ex} h_{ox} $i \leftrightarrow o$**
	E—Integrity—minimum requirement
	I—Insulation—optional to suit application or local requirements
	90—Time (min)—60, 90, 120
	S—Reduced leakage—optional to suit application or local requirements
	v_e—vertical mounting
	h_o—horizontal mounting
	$i \leftrightarrow o$—tested inside to outside and vice versa
Product standard	Under development—will be EN 15871—at some point
Test standard	EN 1366-1
Classification standard	EN 13501-3
EXAP standard	EN 15882-1

An EXAP document is under development and will become available in the future.

TABLE 9.17 Typical classification of smoke control duct.

Smoke control duct—typical classification	**EI 90S v_{ex} h_{ox} $i \leftrightarrow o$**
	E—Integrity—minimum requirement
	I—Insulation—optional to suit application or local requirements
	90—Time (min)—60, 90, 120
	S—Reduced leakage—optional to suit application or local requirements
	v_e—vertical mounting
	h_o—horizontal mounting
	$i \leftrightarrow o$—tested inside to outside and vice versa
Product standard	EN 12101-7
Test standard	EN 1366-8 (after EN 1366-1) (single compartment to EN 1366-9 only)
Classification standard	EN 13501-4
EXAP standard	None—may appear at some point in the future

No EXAP document is available at present but might become available in the future.

wall, the penetration seal at the wall around the duct, and the correct hangers are used. Different suppliers/installers/manufacturers have differing details due to the testing that has been done.

If supplied in the form of sections, there is a requirement for the sections to be CE marked following the requirements of EN 12101-7, third part certification etc. Following testing a classification should be made to EN 13501-4. Table 9.17 shows the typical classification of smoke control duct.

This is a simplified example (for explanation only) to show what should be proposed. More information should be sought directly from manufacturers, noting that manufacturers without third party certification, not offering CE marking for duct sections (only) or a DoP should be avoided, as they are not demonstrating competency in these products.

Single compartment smoke control duct tested to 600°C is available but must not be used where the duct crosses a compartment boundary.

9.7.5.3 Hazardous areas

It is essential that the Industrial Ventilation engineer fully appreciates the cause of Explosions that can occur in hazardous areas with atmospheres containing certain dusts, vapors, and gases. A hazardous area is a location where there is a risk of an explosion. Each specific requirement is different depending on the nature of the atmosphere and the combustible products in it.

Zone classification

The risk level is expressed by classifying hazardous areas into three zones as Zone 0, Zone 1, or Zone 2 (for gas, vapor, and mist atmospheres) or Zone 21 or Zone 22 for dust atmospheres.

Three components for an explosion or fire to occur are:

- *Flammable substance*: Certain quantity of this must be present to produce an explosive mixture of the gas, vapors, mists, or/and dusts.
- *Oxygen:* The correct quantity in the room air must be reached in combination with the flammable substance to produce an explosive atmosphere.
- *Ignition source*: From a spark or high surface temperature.

The above three factors are normally shown as the combustion triangle. If any one of the sides of the triangle is removed, combustion or an explosion will not occur.

This is further complicated by the "3Ts":

- time
- temperature
- turbulence

If one of these is removed, combustion will not occur. Where a potential for an explosive atmosphere exists, it is essential that special precautions are taken to prevent fires and explosions.

All electrical appliances and plant in hazardous areas require to be purpose designed to stop sparks from igniting flammable substances. Each zone and application must be considered and specified for each hazard and classified as a given "zone." Hence all hazardous areas are designed with zone classifications in mind. The "zone" governs the protection level precautions necessary.

Zone classification

For gases, vapors, and mists the zone classifications are Zone 0, Zone 1, and Zone 2 areas.

- *Zone 0:* The area in which an explosive atmosphere is present continuously or will frequently occur.
- *Zone 1*: An area in which an explosive atmosphere may occur occasionally during normal operation. Reasons maintenance or leakage.
- *Zone 2*: A space in which an explosive atmosphere is unlikely in normal operation. However, if it does, will be only for a short period. Causes an accident or some abnormal operating condition.

9.7.5.4 Pressure relief dampers

Pressure relief damper or pressure vent is a device typically used for venting the overpressure created when a gas suppression system discharges into an enclosure. These pressure relief valves (PRVs) must be fire rated if installed in a fire wall and conform in operation to suit the calculations for free vent area provided by the installers or manufacturers of the gas suppression systems.

There are two types of gas suppression agents that require two types of venting. Inert gas suppression agents are typically nitrogen based and aim to add around 50% more volume to an enclosure for the purpose of diluting the oxygen content to around 11% making it still possible for people to breath and function whilst making it impossible for flames to be created from any heat source or combustible material that would otherwise start a fire.

This extra volume is typically added in 60 or 120 seconds and it is critical that the excess gas volume is promptly and safely vented from the enclosure. This is where the correctly designed pressure relief venting is required to avoid any structural damage and to be able to maintain the correct suppression agent concentration in the enclosure for the specified time. Chemical gas suppression systems produce both a negative and positive pressure in the enclosure a so require either two one-way vents, one for each direction of flow or a single two-way vent that can vent in both directions.

9.8 Sound reduction in air-handling systems

9.8.1 Basic concepts

The subject of acoustics involving sound transmission is of prime importance in industrial ventilation. Correct system design will ensure that the designer provides a system that will not give rise to complaints regarding noise levels.

Consider the continuous oscillations of a tuning fork. These oscillations generate successive compressions and rarefactions outward through the air. The human ears, when receiving these pressure variations, transfer them to the brain, where they are interpreted as sound. Therefore the phenomenon of sound is a pressure variation in a fixed point in the air or in another elastic medium, such as water, gas, or solid.

This pressure variation can be considered as the transfer of a pressure wave in space. In the same way, when a stone is thrown into a lake, the ripples generated move radially from the point of entry of the stone. But this observation is only apparent, because a floating buoy will stay in the same horizontal position. It does not move radially in the space; the perturbation, however, moves.

These phenomena can be considered as work processes. In fact, pressure is the force per surface unit, and work is the product of force and displacement of the force. Work is equivalent to energy, and it cannot be created or destroyed.

Therefore when a noise is to be reduced, the sound energy must be converted into another form of energy, such as kinetic energy of a medium or heat.

This issue must be understood in order to reduce noise problems.

In a fixed point in space, the magnitude of pressure will change according to the nature of the sound source.

A tuning fork gives a pure tone, that is, a pressure variation represented by a sinusoid curve (Fig. 9.60).

The sound speed c, m/s, is the velocity of propagation of the pressure variations. This depends on the physical properties of the medium and increases with the density of the medium. In air, for example, it is 344 m/s, while in water, 1410 m/s and in concrete, 3000 m/s. The elapsed time between successive compressions is called the period time T.

Frequency is the repetition rate of pressure variations, and it is described by the reciprocal of the period time T

$$f = \frac{1}{T}(\text{Hz}). \tag{9.157}$$

Its unit is hertz (s^{-1}), corresponding one cycle per second. Pure tone consists of one frequency, but

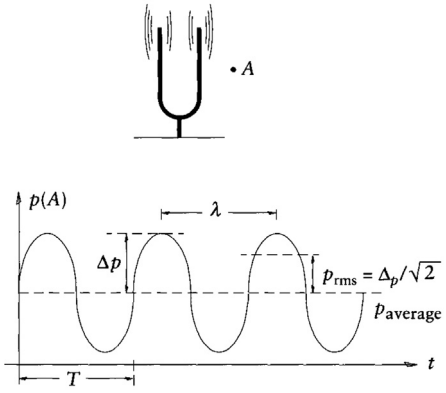


FIGURE 9.60 Sound-waves generation by a tuning fork.

normally all sounds are a mixture of many frequencies. In the audio range, the frequency varies normally from 20 to 16,000 Hz. The size of the audio range depends on the sensitivity of the listener's ears. When the frequency is below 20 Hz, it is called infrasound, while for frequencies over 16,000 Hz, it is called ultrasound.

Wavelength λ is defined as the distance between successive compressions or rarefactions and is

$$\lambda = \frac{c}{f}(\text{m}). \tag{9.158}$$

A sound can be periodic, hence steady, or random (Fig. 9.61).

A sound is generally not a pure tone, as the latter is only emitted from particular sources. It can be demonstrated that a sound can be divided into different pure tones (superposition method). The waves at different frequencies give the spectrum of the sound, which also describes its energy distribution. In frequency analysis, the spectrum is divided into octave bands. An octave band is defined as the frequency range with its upper boundary twice the frequency of its lower boundary. For every octave band, a central band frequency (f_c) is defined as follows

$$f_c = \sqrt{f_l \cdot f_u} \tag{9.159}$$

where f_l and f_u are the lower and upper boundary frequencies, respectively. In ventilation technology the normally interesting frequency area is 63–8000 Hz. In this area, we have eight octave bands. Sometimes also the frequency band 31.5 Hz is under consideration.

For example, the lowest octave band corresponds to a frequency range between 22 and 45 Hz. Its central value is

$$f_c = \sqrt{22 \cdot 45} = 31.5\text{Hz}.$$

In Table 9.18 the standardized octave band series are shown.

Octave bands are divided, on a logarithmic frequency scale, into three equally wide one-third octave bands. This is done often when more exact data of sound spectra are needed. Table 9.19 shows the standardized one-third octave band series.

The sound generation mechanism involves the transmission of acoustic energy in space. Therefore the power of a sound source is the energy emitted in time units and is measured in W.

On the basis of a sound wave equation, it is shown that the power of a noise source is equal to

$$\prod = \frac{p^2}{\rho c} A(\text{W}), \tag{9.160}$$

where A is the orthogonal surface to the wave path (measured in m^2), ρ is the density of the medium (kg/m^3), c is the sound speed in the medium (m/s), and p is the pressure amplitude of the sound (Pa).

If the noise source is a point source and the emission propagation is spherical (Fig. 9.62), then the source power can be written as

$$I = \frac{p^2}{\rho c} 4\pi r^2. \tag{9.161}$$

Therefore with constant (steady-state) sound power, p decreases when r increases, not linearly, but quadratically.

Perception of sound by the human ear is not related to sound power but to sound intensity, defined as

$$I = \frac{\prod}{A} = \frac{p^2}{\rho c}. \tag{9.162}$$

that is, sound power for surface unit W/m^2. As the listener moves away from the sound-generating source, A increases and intensity decreases.

The lowest intensity audible by human ear is $10^{-12}\,W/m^2$, while the maximum value is $1\,W/m^2$. Therefore the scale range is very large and inconvenient for technical calculations: it is more convenient to make use of a logarithmic unit, defined as a function of the ratio between the sound intensity and the intensity I_0 of the hearing threshold

$$L_p = 10\log\left(\frac{I}{I_0}\right) = \log\left(\frac{p^2}{p_0^2}\right). \tag{9.163}$$

where I_0 is fixed equal to $10^{-12}\,W/m^2$, and its corresponding pressure is 2×10^{-5} Pa. The new unit is *decibel* (symbol dB) and L_p is called the sound pressure level. In a similar way, we can define sound power level L_w as

$$L_W = 10\log\left(\frac{\prod}{\prod_0}\right) = 10\log\left(\frac{\prod}{10^{-12}}\right). \tag{9.164}$$

This unit is also a *decibel*. Note that the decibel is a pure number.

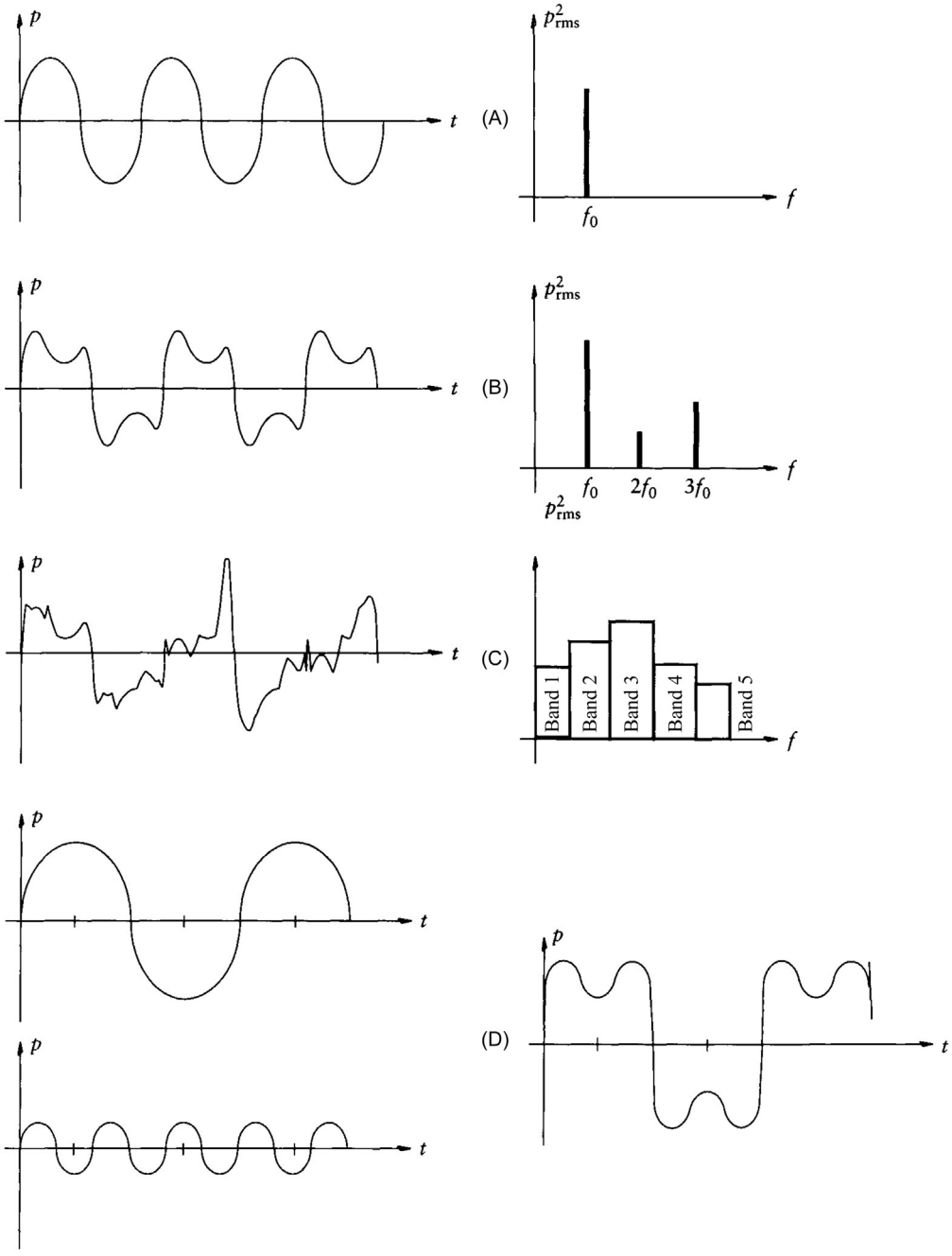


FIGURE 9.61 Different kinds of sounds: (A) pure tone, (B) periodic sound, (C) random noise, and (D) effects overlapping principle.

When two sound sources are added, it must be remembered that the decibel is a logarithmic unit. Therefore they cannot be added arithmetically but must be combined as follows:

Step 1. They must be divided by 10.
Step 2. Then the antilogarithm must be calculated.
Step 3. The obtained values are added.
Step 4. The logarithm is calculated and then multiplied by 10 to obtain the combined value.

In general, when two or more sounds reach the listener at the same time, a composed sound will be received, with pressure level L equal to

$$L = 10\log[10^{L_1/10} + 10^{L_2/10} + 10^{L_3/10} + \cdots + 10^{L_n/10}], \tag{9.165}$$

where L_1, L_2, L_3,..., L_n are the pressure levels of each source. This formula can also be applied for the composition of the pressure levels at different frequencies for the same source.

TABLE 9.18 Standardized octave band series (ISO 226).

Octave 1			Octave 2			Octave 3		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
22	31.5	45	45	63	90	90	125	180
Octave 4			**Octave 5**			**Octave 6**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
180	250	355	355	500	710	710	1000	1400
Octave 7			**Octave 8**			**Octave 9**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
1400	2000	2800	2800	4000	5600	5600	8000	11,200
Octave 10								
Lower	Central	Upper						
11,200	16,000	22,400						

As an alternative to the above Eq. (9.167), it is easier to add to the higher pressure level a term depending on the difference between the considered levels, as shown in Table 9.20.

Example 15

Two noise sources produce 69.2 and 69 dB pressure levels respectively (Table 9.21) at the same space point. The composed sound at the said point will have a total pressure level of 72.1 dB, calculated by adding 2.9 dB to 69.2 dB. In a similar way, at 250 Hz, for example, if each of the sources has a pressure level of 60 dB, the total level will be 63 dB (sum of 3 and 60 dB).

To determine the levels of each frequency for the same source, it is advisable to proceed considering two levels at a time, starting from the lower levels. In this way, for example, by determining the pressure levels of source 1, the total level is 69.2 dB. Thus the same result can be achieved by applying Eq. (9.138) or the method revealed in Table 9.20.

Example 16

In a work area, a machine has a power level of 50 dB. A second machine in the same room has a power level of 50 dB. Determine the final combined power level.

The total power level will be 53 dB.

In work areas the noise is usually generated by different sources, such as air-handling units and refrigerating plants (especially extract and supply air fans).

It is important to quantify the sound pressure levels in dB generated by each source and for each frequency (31.5–8000 Hz) in order to establish which noise will be masked or prevalent. It must be noted that when the pressure levels of two noises differ by more than 10 dB, the resulting level is equal to that of the higher-level source; in other words, the noise at the higher level masks the noise at the lower level, which will not be perceptible to the listener (or the phonometer). In this case, it is useless to reduce the latter noise, as the composed noise will remain the same, being influenced by the higher level noise only.

It is important to remember that the response by a human ear to sound is different from that detected by scientific instruments, as the human ear is more sensitive in the middle frequency range than at the low and high frequencies at the same level.

Therefore acoustic science has introduced different kinds of weighting curves in order to correlate as accurately as possible the sound level with the level really perceived by a generic listener. In practice, A-weighting correction is used. It consists of a series of coefficients, shown in Table 9.22. These are added to sound levels expressed in dB for each frequency.

The results are expressed in dB(A). The suffix (A) means that the new level has been calculated referring to the A-weighting correction. Phonometers have a special filter called the A-filter, which automatically introduces this correction at measured values, allowing the reading of pressure level in dB(A) directly.

Example 17

Referring to Example 15, the pressure level of the A-weighted combined noise is 68.8 dB(A). Therefore the acoustic perception of a listener will be 68.8 dB(A), not 72.1 dB (Table 9.23).

9.8.2 Free-field noise transmission

When the noise transmission takes place in a free field (no reflective surfaces), it is possible to calculate the pressure levels at different distances from the source.

TABLE 9.19 Standardized one-third octave band series (ISO 226).

First third of octave 1			Second third of octave 1			Third third of octave 1		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
22	25	28	28	31.5	35.5	35.5	40	45
First third of octave 2			**Second third of octave 2**			**Third third of octave 2**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
45	50	56	56	63	71	71	80	90
First third of octave 3			**Second third of octave 3**			**Third third of octave 3**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
90	100	112	112	125	140	140	160	180
First third of octave 4			**Second third of octave 4**			**Third third of octave 4**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
180	200	224	224	250	280	280	315	355
First third of octave 5			**Second third of octave 5**			**Third third of octave S**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
355	400	450	450	500	560	560	630	710
First third of octave 6			**Second third of octave 6**			**Third third of octave 6**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
710	800	900	900	1000	1120	1120	1250	1400
First third of octave 7			**Second third of octave 7**			**Third third of octave 7**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
1400	1600	1800	1800	2000	2240	2240	2500	2800
First third of octave 8			**Second third of octave 8**			**Third third of octave 8**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
2800	3150	3550	3550	4000	4500	4500	5000	5600
First third of octave 9			**Second third of octave 9**			**Third third of octave 9**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
5600	6300	7100	7100	8000	9000	9000	10,000	11,200
First third of octave 10			**Second third of octave 10**			**Third third of octave 10**		
Lower	Central	Upper	Lower	Central	Upper	Lower	Central	Upper
11,200	12,500	14,000	14,000	16,000	18,000	18,000	20,000	22,400

For spherical propagation, the following formula can be used

$$L_p = L_w - 20 \cdot \log r - 11, \tag{9.166}$$

where r is the distance (in m) from the source and L_w is its noise power level. The following formula can be used in the case of hemispherical propagation:

$$L_w - 809,809(20) \cdot \log, \tag{9.167}$$

and in general

$$L_p = L_w - 20 \cdot \log r + 10 \cdot \log Q - 11, \tag{9.168}$$

where Q assumes the values indicated in Fig. 9.62, which is a characteristic of the geometry of floor and walls around the source. Fig. 9.62 shows how the sound propagates from a point shaped source.

If we place the sound source in the middle of the room (Fig. 9.62A), and assume even propagation of the sound, the propagation shape will be spherical.

If we place the sound source in the middle of a wall (Fig. 9.62B), we assume that the sound will propagate to the room and not to the wall. Then the propagation

will be a half of a sphere. The concentration will double.

If we place the sound source between the ceiling and the wall, or between the floor and the wall (Fig. 9.62C), the sound will spread in a quarter of a sphere. The concentration will be four times of a sphere.

If we place the sound source in a corner (Fig. 9.62D), the propagation will be an eighth of a sphere, and the concentration eight times of a sphere.

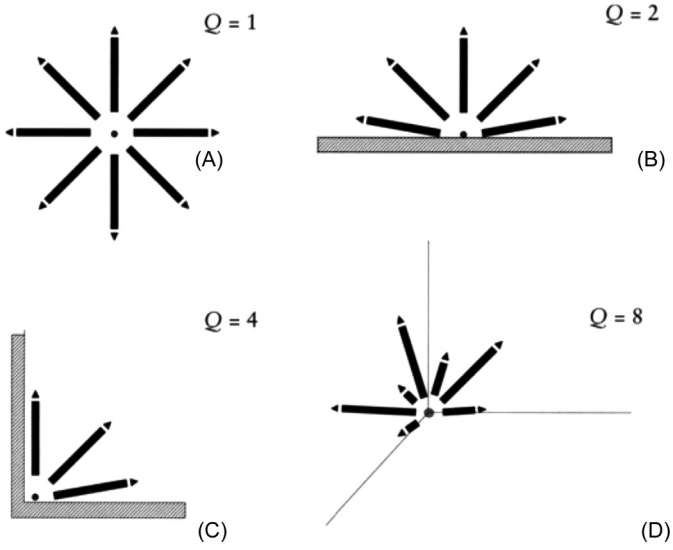


FIGURE 9.62 Different kinds of sound propagation: (A) the propagation shape is spherical, (B) the propagation shape is a half sphere, (C) the propagation shape is a quarter of a sphere, (D) the propagation shape is an eighth of a sphere.

9.8.3 Criteria for acceptable air-handling units and HVAC system noise levels

The disturbance caused by a noise depends on its intensity [equivalent pressure level L in dB(A)], its frequency spectrum (that is its energy distribution), and the acoustic characteristics of the medium in which the listener is located.

Concerning the sound pressure level, when a noise generated by an HVAC system or an air-handling unit increases the ambient background noise by 3 dB, the noise increase is just perceptible. On the contrary, an increase of 5 dB or more is clearly perceptible.

As regards the noise spectrum, the different situations can be analyzed approximately with NC (noise criterion) and NR (noise rating) curves (Fig. 9.63). NC and NR curves define the octave band limits of an acceptable background noise: each of them is characterized by a number representing the sound pressure level at 1000 Hz.

The procedure must be carried out in this manner: the noise spectrum is superimposed on NC or NR

TABLE 9.20 Coefficients for the composition of pressure levels of different sources or of different frequencies of the same source.

Difference in dB between two pressure levels	0	1	2	3	4	5	6	7	8	9	10 or more
Term to add at the higher level	3	2.5	2	2	1.5	1.2	1	1	0.5	0.5	0

TABLE 9.21 Composition of pressure levels.

Frequency (Hz)	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**	**Total level**
L_1 (dB)	60	62	60	65	60	55	54	48	69.2
L_2 (dB)	60	62	60	60	60	55	60	60	69.0
L total (dB)	63.0	65.0	63.0	66.2	63.0	58.0	61.0	60	72.1

TABLE 9.22 A-weighting corrections.

Frequency (Hz)	25	31.5	40	50	63	80	100	125	160
A-wt. correction (dB)	−44.7	−39.4	−34.6	−30.2	−26.2	−22.5	−19.1	−16.1	−13.4
Frequency (Hz)	200	250	315	400	500	630	800	1000	1250
A-wt. correction (dB)	− 10.9	− 8.6	− 6.6	− 4.8	− 3.2	− 1.9	− 0.8	0	0.6
Frequency (Hz)	1600	2000	2500	3150	4000	5000	6300	8000	10,000
A-wt. correction (dB)	1	1.2	1.3	1.2	1	0.5	− 0.1	− 1.1	− 2.5

TABLE 9.23 A-weighted combined noise.

Frequency (Hz)	63	125	250	500	1000	2000	4000	8000	Total level
L total (dB)	63.0	65.0	63.0	66.2	63.0	58.0	61.0	60.0	72.1
A-wt. correction (dB)	− 26.2	− 16.1	− 8.6	− 3.2	0.0	1.2	1.0	− 1.1	
L total in dB(A)	36.8	48.9	54.4	63.0	63.0	59.2	62.0	58.9	68.8

diagrams, and the highest intercepted NC or NR curve represents the noise. For example, when a noise is represented by an NC 50 curve, it means that its spectrum does not exceed the NC curve, in correspondence of which at 1000 Hz the pressure level is equal to 50 dB (A). If the intercepted point is placed between 250 and 1000 Hz, the noise is classified as neutral; under 250 Hz is called rumbly, while over 1000 Hz is classed as hissy.

A more recent method of analysis uses the RC curves. In this case, it must be calculated the arithmetical mean of sound pressure level at 500, 1000, and 2000 Hz. The obtained value identifies the specific RC curve. The noise is classified as rumbly (with excess of energy at low frequencies) if it is under 500 Hz and its sound pressure level exceeds the RC value by 5 or more dB. The noise is classified as hissy (with excess of energy at high frequencies) if it is over 500 Hz and its sound pressure level exceeds the RC value by 3 or more dB.

The acceptability of a noise is fixed from the designer on the basis of specific data for the area depending on the state and country laws. In some countries there are regulations requiring that the workers cannot be exposed to a noise exceeding set limits.

Unsteady noise can be evaluated by a phonometer, which measures the sound pressure level for a time period of noise fluctuation and gives the time-weighted average value.

The actual noise levels produced by HVAC systems can vary considerably, and it is not possible to generalize the problems that may be encountered. From a safety point of view, it is advisable to start hearing conservation programs for workers. Permanent hearing damage will result when the noise levels exceed 80 dB(A) for a given time period. Whenever possible, it is desirable to control noise pressure levels to meet the requirements of speech communication; in this case noise should not exceed 65–70 dB(A).

9.9 Fundamentals of energy system optimization in industrial buildings

This section briefly introduces the energy aspects that have to be taken into account in system design.

In this section, attention is paid to the energy demand of ventilation and air-conditioning systems as a whole. On the system level, considerable efforts have been made to reduce the heating and cooling demands, and the electrical energy consumption has been regarded as a marginal part. But after the first energy crisis in the mid-1970s, the situation started to change, so rapidly that in many modern air-handling installations, the share of electrical energy can be as high as about 90%. This figure has repeatedly been reported in cold climates, too.

One simple example describes the problem: the efficiency of a heat-recovery device can be increased from approximately 50% up to approximately 75% by using a double unit. This will double the pressure drop from, say, 100 to 200 Pa, resulting in the fan energy's increasing respectively.

Measures undertaken to improve the IAQ have the same effect by upgrading the filter class, increasing the air change rate, etc. These small improvements have grown both in number and in size, little by little.

Example: A ventilation system (Fig. 9.64) handling 20 m^3/s of air needs to heat the supply air from 10°C to 20°C. Doubling the number of heat exchangers from one to two increases the heat-recovery efficiency from 50% to 75% and introduces an extra pressure drop of 300 Pa. As we can see from Table 9.24, this is probably a cost-efficient measure.

Therefore it is essential also in industrial environments to pay serious attention to energy usage of components and systems. This will bring a need to develop standards, guidelines, and other tools for practitioners to achieve the optimum. Some criteria to estimate the total energy consumption for the building and for individual systems, air-handling units, and fans have been developed.

Space requirements for air-handling systems are briefly described in European Standard EN 16798-3,[10] which, however, is targeted to commercial, public, and office buildings. These requirements also take into account the need for service and maintenance. In industrial applications the same principles apply, but the location of air-handling units, service routes, and many other factors have to be considered more from the production-process point of view than for nonindustrial applications.

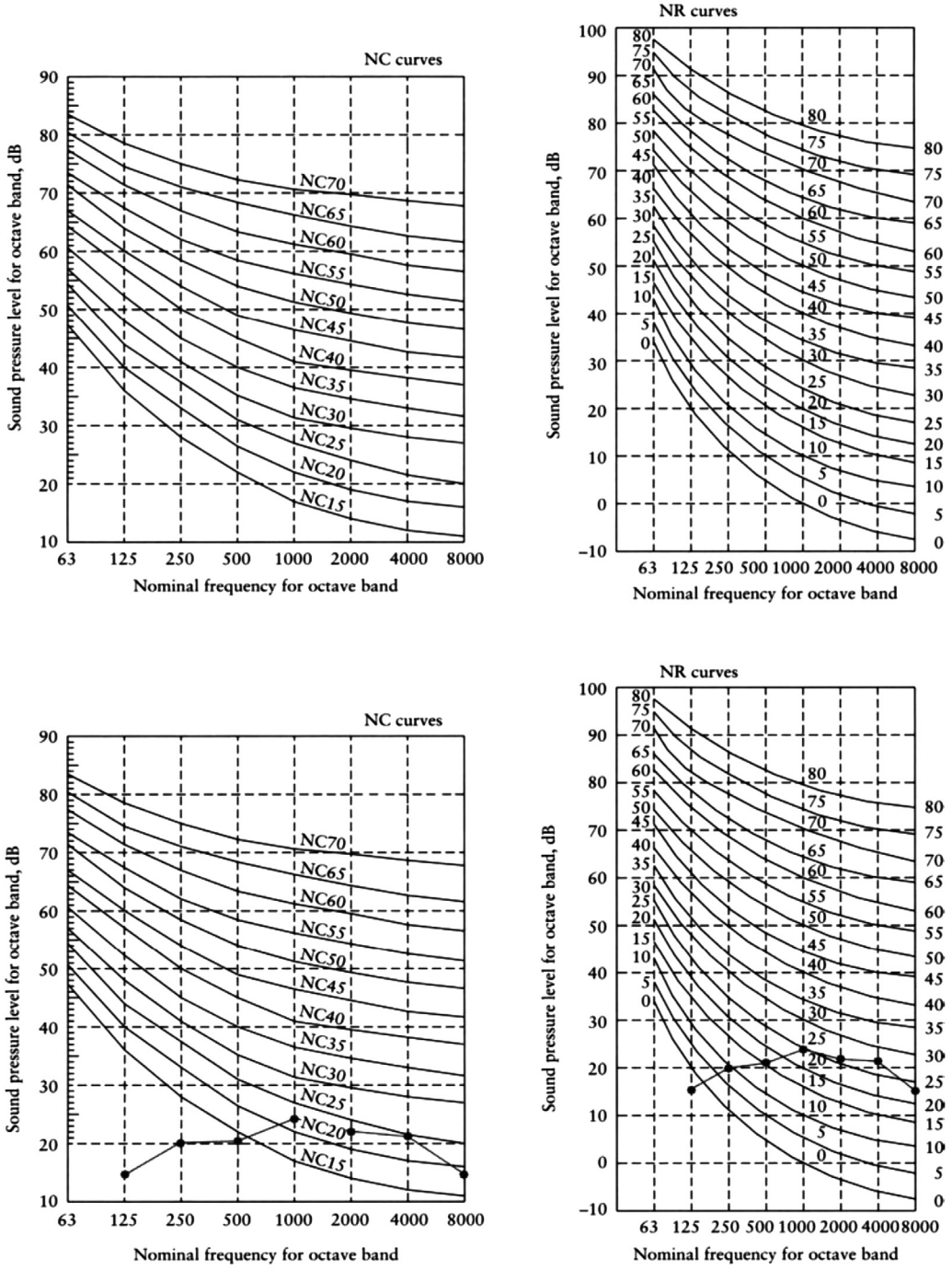


FIGURE 9.63 NC and NR curves and examples of application—the examined noise has an NR index of 28 and an NC index of 25. *NC*, Noise criterion; *NR*, noise rating.

In ductwork design, attention shall be paid to reduce energy losses using properly designed fittings and proper duct sizing. Balancing also becomes a critical factor in order to ensure that the system operates at its most efficient designed condition. Systems that are designed for easy balancing are good from the energy-consumption point of view, provided that the pressure drops in dampers and terminal devices are within reasonable values. Special attention to equal pressure drops in different duct branches is necessary in design of such ductworks for contaminated air where the use of dampers is not possible. See also Section 9.7.

DCV is one approach to reduce energy consumption due to ventilation, which is gaining popularity in both

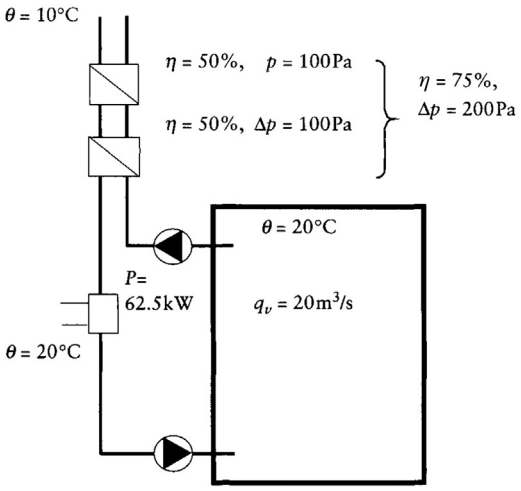


FIGURE 9.64 Ventilation system (example).

TABLE 9.24 Ventilation system characteristics (example).

Air volume flow	q_v	20 m^3/s
Air mass flow	Δp	25 kg/s
Extra pressure drop	ΔT	300 Pa
Temperature differential	$\Delta\eta$	10°C
Extra recovery efficiency	ΔT gain	25%
Extra temperature recovery		2.5°C
Extra heat recovery		62.5 kW
Fan power increase		6.0 kW
Net gain		56.5 kW

TABLE 9.25 Recommended maximum pressure drops for specific components in supply air systems.

	Pressure drop (Pa)[a]		
Component	**Low**	**Normal**	**High**
Ductwork	120	200	300
Heating or cooling coil	40	80	120
Heat-recovery unit	100	150	200
Air filter per section[b]	100	150	250
Silencer	30	50	80
Terminal device	30	50	100
Air inlet	20	50	70

[a]*Values for individual components may be exceeded then the overall target can be achieved by lower pressure drops of other components.*
[b]*Final pressure drop before replacement.*
Note: The values are somewhat different from prEN 13779,[32] which has been superseded by EN 16798-3:2017[10] (currently under revision).

TABLE 9.26 Recommended maximum pressure drops for specific components in extract air systems.

	Pressure drop (Pa)[a]		
Component	**Low**	**Normal**	**High**
Ductwork	120	200	300
Heat-recovery unit	100	200	300
Air filter per section[b]	100	150	250
Air outlet	20	40	60

[a]*Values for individual components may be exceeded then the overall target can be achieved by lower pressure drops of other components.*
[b]*Final pressure drop before replacement.*
Note: The values are somewhat different from EN 13779[32], which has been superseded by EN 16798-3:2017[10] (currently under revision).

industrial and nonindustrial applications. It is used in cases where ventilation requirements vary with time, regularly or irregularly. The control is based on a specified level of IAQ by means of continuous measurement of the parameters, which are expected to primarily determine the IAQ, such as the concentration of the main contaminant liberated from the production process. The principle is thus similar to the one in some better-known nonindustrial applications, for example, CO_2 levels in rooms with dense human occupancy (theaters, classrooms, etc.) or nicotine concentration in smoking rooms. See also Section 9.6.

9.9.1 Design aspects of energy-efficient systems

In the design of air-handling systems, a reasonable energy efficiency can be achieved by relatively simple means, just paying attention to some basic principles.

To give the designer some advice, many recommendations exist in literature. Some general recommendations can be found in EN 16798-3[10] and CEN/TR 16798-4[31] for designing the air distribution system for low energy consumption. These include giving a certain target level for power consumption [the so-called SFP class. SFP means specific fan power, the ratio between the input power of fan motors and the total airflow, in $W/(m^3/s)$].

Tables 9.25 and 9.26 present recommendations for maximum pressure drops of components in the supply and ETA systems in order to achieve a certain target level for power consumption in the whole system. In category "low," this target level is 1500: in "normal," 2500; and in "high" 4000 $W/(m^3/s)$.

The selection of components to match the target level can be based on the default maximum pressure drop for each component. If a certain component with

TABLE 9.27 Extract air (ETA) categories.

Pollution level code	Description	Example
ETA with low pollution levels		
ETA 1	Air of the same quality as outdoors with respect to humidity. From rooms with pollutant sources from humans and building material.	Offices, storage rooms, public service places. No major pollution sources, including smoking.
ETA with moderate pollution levels		
ETA 2	Air from occupied spaces that have impurities in excess of ETA 1.	Smoking lounges, eating areas.
ETA with high pollution levels		
ETA 3	Spaces in which moisture, chemical processes, etc. substantially lower air quality.	Toilets, kitchens, garages, tunnels, car parks, solvent areas, laboratories.
ETA with very high pollution levels		
ETA 4	Air containing impurities and odors detrimental to health, in concentrations higher than regulations allow.	Industrial processes, laboratories, smoking lounges.

a higher pressure drop is selected (e.g., because of its higher filter class), then the overall target can be achieved by lower pressure drops of other components. For industrial applications, however, the figures in EN 16798-3[10] and CEN/TR 16798-4[31] are not necessarily valid due to the special demands of the production processes, but still their relevant parts can be taken as a design basis. The design process will end up in SFP categories for each individual air-handling unit (SFP_V)—not only as a target value, but as a measurable design value or even as a guarantee value.

9.10 Special considerations and system design aspects

This section addresses special considerations and aspects very briefly. Details and specific issues could be added later on.

9.10.1 Aspects related to the quality of extract or exhaust air

In many of these aspects, the ETA classification, summarized in Table 9.27, is a useful tool to assist in system design. Applications of these classifications are described in EN 16798-3.

Categories for EHA are also defined in EN 16798-3. Normally, the EHA category for air to be discharged outdoors is the same as ETA, but if the system includes reliable and monitored cleaning equipment, the category can be one lower. With these provisions, air from industrial processes with very high pollution level, in category ETA 4 can be cleaned to EHA 3 levels.

9.10.1.1 Examples of ETA and EHA classification applications—reuse of extract air

The quality of ETA is crucial for its use as recirculated air. The classification described in EN 16798-3 is also valid for industrial applications. Air even possibly containing any carcinogenic compound is automatically of category ETA 4 (very high pollution levels) and must not be used, not even after cleaning, as recirculated air (should be applicable normally only in category ETA 1, and in some cases ETA 2, after cleaning) or transfer air (applicable in categories ETA 1 and ETA 2). This means that reuse of ETA in any industrial application is limited to relatively clean industrial environments (ETA in category ETA 1 or 2), and even then the quality of air must be continuously controlled and monitored in a reliable way.

Distances and locations of openings

Rough principles are presented in EN 16798-3. The general principle is to discharge air with high pollutant levels (category EHA 3 or 4) from openings above the roof level, and upward. Only relatively clean air can be discharged through outer walls. Openings for supply air should be located at the highest practicably possible distance from contamination sources outdoors or from EHA.

Pressure conditions

As a general rule the system must be designed so that contaminated extract or EHA cannot mix with

supply air through leakages. This entails requirements for pressure conditions, especially for heat-recovery units. For category ETA 1, no limitations are needed: for ETA 2 and 3 using air-to-air heat exchangers for heat recovery, a positive pressure in the supply air side is required (ETA 2, in average; ETA 3, throughout). When ETA is of category ETA 4, only systems using an intermediate heat transfer medium should be applied. Positive pressure should be avoided in air ducts at least for category ETA 3 and must be avoided altogether for category ETA 4.

9.10.2 Other questions

More system- and equipment-specific questions will be dealt with in more detail in the *Systems and Equipment* volume. Examples include the following:

System hygiene and cleanliness, and maintenance issues: General requirements presented in EN 12097,[33] EN 15780 and EN 16798-3 apply.
Dampers in air-handling units and air distribution systems: Attention has to be paid to reducing the leakages, in accordance with EN 1751.[28] Dampers used in mixing sections of air-handling units must fulfill additional requirements, in accordance with EN 13053.[34]
Mixing in air-handling units and air distribution systems: Some main principles are presented in Section 9.4. However, air mixing is a complicated process under strong development.
Air terminals and functional factors affecting the air-handling systems: Some basic principles are presented in Chapter 8, Room Air Conditioning. However, more attention has to be paid to minimizing the leakages and to sound attenuation.
Materials and corrosion protection: From the hygienic point of view, this issue is growing in importance. Applications with high hygiene requirements are rapidly increasing due to automation of processes.
Flexible ducts (or tubes) and their pressure drops at various flow rates both when extended and when compressed; connection to metal sheet ducts.
Ducts of other materials than metal sheet (plastic, glass fiber wool and aluminum sheet, ceramics, glass, etc.)
Movable or swinging ducts and associated pressure drops in different positions.
Dampers for air ducts handling dust.
Fan selection for air and gases that are dust laden or contain corrosive gases or vapors.
Silencers in air-handling units and air distribution systems.

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Physical Factors, Units, Definitions and References

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Mean molecular weight of dry air $M_a = 28.969$ kg/kmol

Mean molecular weight of water $M_v = 18.02$ kg/kmol

Density of dry air at 101.325 kPa and 0°C = 1.293 kg/m^3

Density of water at 4°C = 1000 kg/m^3

Density of water at 20°C = 998.23 kg/m^3

Barometric pressure at standard temperature and pressure = 101.325 kPa.

Standard temperature and pressure (STP) = 0°C at 101.325 kPa (also known as normal temperature and pressure)

Universal gas constant $R_{gas} = MR = 8.3143$ J/mol/K

Volume of 1 mol of the permanent gases (at 101.325 kPa and 0°C) = 22.4136 m^3

Characteristic gas constant for dry air $R_a = 287$ J/kg/K

Characteristic gas constant for steam $R_v = 462$ J/kg/K

Mean specific heat of air at constant pressure $c_{pa} = 1005$ J/kg/K

Mean specific heat of air at constant volume $c_{va} = 718$ J/kg/K

Mean specific heat of steam of air at constant pressure $c_{pv} = 4210$ J/kg/K

Mean specific heat of steam at constant volume $c_{vv} = 1810$ J/kg/K

Adiabatic index for air at room temperature and pressure = 1.4

Latent heat of steam at 0°C = 2500 kJ/kg/K

Standard gravity = 9.806 65 m/s^2

Velocity of sound in air at normal temperature and pressure c = 331.46 m/s

Stefan's Constant $\sigma = 5.67 \times 10^{-8}$ W/m^2/K^4

Dimensionless numbers

The following dimensionless numbers may be expressed in various forms due to the use of other relevant parameters.

Archimedes number

$$\text{Ar} = \sqrt{\frac{\Delta\rho g L}{\rho v^2}} = \frac{\text{Buoyancy force}}{\text{Inertia force}}$$

The ratio $\Delta\rho/\rho$ can be replaced by $\Delta T/T$. Ar relates the influence of velocity and temperature of a jet when discharged into an environment of a different temperature. In some instances, the Froude number, the Galileo number, or the Grashof number may replace the Archimedes number.

Colburn j-factor

$$\text{Colburn } j\text{-factor} = \frac{\text{Nu}}{\text{Re Pr}^{0.33}} = \text{St}\cdot\text{Pr}^{0.66}$$

Used in heat-transfer applications.

$$\text{Colburn j-factor} = \frac{\text{N}}{\text{Re Se}}\text{Sc}^{0.66}$$

Used in mass-transfer applications.

Condensation number

$$\text{Co} = h\left(\frac{\eta}{\rho^2 g\ \lambda^3}\right)^{0.33}$$

Euler number

$$\text{Eu} = \frac{p}{\rho v^2} = \frac{\text{Pressure force}}{\text{Inertia force}}$$

Fraude number

$$\mathrm{Fr} = \frac{v^2}{gL} = \frac{\text{Inertia force}}{\text{Gravity force}}$$

See the Archimedes number.

Graetz number

$$\mathrm{Gz} = \frac{q_m c}{\lambda l}$$

The same as the Peclet number except $\frac{d_w}{L}$ considered (entrance region).

Grashof number

$$\mathrm{Gr} = \frac{\beta g \rho^2 l^3 \Delta\theta}{\eta^2} = \frac{\left(\text{Buoyancy forces}\right)\left(\text{Inertia force}\right)}{\left(\text{Viscous forces}\right)^2}$$

Used for free convection.

Knudsen number

$$\mathrm{Kn} = \frac{l_1}{0.5d} = \frac{\text{Molecular mean free path}}{\text{Characteristic length}}$$

Used for particulate movement in a gas.

Lewis number

$$\mathrm{Le} = \frac{\mathrm{Sc}}{\mathrm{Pr}} = \frac{\kappa}{D}$$

where

$$\kappa \equiv \frac{\lambda}{c\rho} = \frac{\text{Thermal diffusivity}}{\text{Mass diffusivity}}$$

Used for calculations involving the vaporization of a fluid.

Mach number

$$\mathrm{Ma} = \frac{\rho l^2 v^2}{Kl^2} 2 = \frac{v}{\frac{K}{\rho}} \text{or} \frac{v}{\sqrt{\frac{K}{\rho}}} = \frac{\text{Inertia force}}{\text{Compressibility force}}$$

Nusselt number

$$\mathrm{Nu} = \frac{\phi l}{\lambda \Delta\theta} = \frac{hl}{\lambda}$$

Ratio of temperature gradients, used for heat transfer taking place with fluid flow.

Peclet number

$$\mathrm{Pe} = \frac{v l C_p \rho}{\lambda} = \mathrm{Re} \cdot \mathrm{Pr} = \frac{\text{Heat convection}}{\text{Heat conduction}}$$

Used for convective flow in heat-transfer applications.

$$\mathrm{Pe} = \frac{lv}{D} = \frac{\text{Mass transfer}}{\text{Mass diffusivity}}$$

Used in mass transfer applications involving aerosols.

Prandtl number

$$\mathrm{Pr} = \frac{c\eta}{\lambda} = \frac{v}{\kappa} = \frac{\text{Molecular diffusivity of momentum}}{\text{Molecular diffusivity of heat}}$$

Used for heat transfer with fluid flow.

Reynolds number

$$\mathrm{Re} = \frac{v\rho l}{\eta} = \frac{vl}{v} = \frac{\text{Inertia force}}{\text{Viscous force}}$$

Relates the nature of the fluid flow in and around bodies.

Richardson number

$$\mathrm{Ri} = \frac{-g\left(\frac{\partial T}{\partial z}\right) + G}{T\left(\frac{\partial v}{\partial z}\right)^2} = \frac{-g\left(\frac{\partial \theta}{\partial z}\right)}{\left(\frac{\partial v}{\partial z}\right)^2} = \frac{\text{Buoyancy}}{\text{Momentum gradient}}$$

Schmidt number

$$\mathrm{Sc} = \frac{v}{D} = \frac{\text{Momentum diffusivity}}{\text{Mass diffusivity}}$$

Used for mass transfer = Pr number for mass transfer = Colburn number.

Sherwood number

$$\mathrm{Sh} = \frac{\beta}{\frac{D}{l}} = \frac{\beta l}{D} = \frac{\left(\text{Mass transfer coefficient}\right)\left(\text{Length}\right)}{\text{Diffusion coefficient}}$$

This is the Nusselt number for mass transfer.

Stanton number

$$\mathrm{St} = \frac{\mathrm{Nu}}{\mathrm{Re}\cdot\mathrm{Pr}} = \frac{\phi}{V\rho c\Delta\theta} = \frac{h}{V\rho c} = \frac{\text{Wall heat-transfer rate}}{\text{Heat transfer by convection}}$$

Used for convective heat transfer applications.

Stokes number

$$\mathrm{Stk} = \frac{\rho d^2 v C_f}{18\eta L} = \frac{\text{Stopping distance}}{\text{Characteristic length}}$$

where C_f = Cunningham's factor. Used for particulate settling calculations.

Glossary

A

A weighting scale, dBA The unit of sound intensity expressed as a logarithmic scale, related to a reference level of 10^{-12} W/m^2. The A weighting scale is the most commonly used scale, as it reduces the response of sound meters to very high and low frequencies and emphasize those within the range audible by the human ear.

Abatement The reduction in air or water pollution from all sources by preventive measures.

Abrasive blasting The process of product cleaning by the use of an abrasive material.

Abrasive dusts Dusts used for abrasive blasting, grinding, polishing, and buffing.

Absolute filters Strictly dry filters used in installations that require a very clean environment, classified according to efficiency.

Absolute humidity The mass of water vapor present in a unit mass of dry air.

Absolute pressure The pressure recorded relative to the absolute zero value of pressure.

Absolute radiant heat flow The radiant energy emitted per unit area in one direction.

Absolute roughness The roughness of a pipe or duct wall, normally expressed as a dimensionless ratio of the linear measure of the internal roughness k_s divided by the diameter.

Absolute total pressure Sometimes called stagnation pressure, the algebraic sum of the velocity and static pressure at a given location in a moving system, in Pa.

Absolute ventilation efficiency A value that provides a means of determining the actual ability of a ventilating system to reduce the concentration of a pollutant, compared to that theoretically possible.

Absolute zero The temperature at which a perfect gas kept at constant volume exerts no pressure; it is equal to −273.16°C (0K).

Absorber The material that is used in an absorption process.

Absorption

1. The process of diffusion in which molecules are transferred from the gas phase into a solid or liquid sorption medium.
2. The taking up of radiant energy by a material as it encounters the body, or as it passes through. A physical change or chemical change or both may accompany it.

Absorption coefficient Measure of the amount of sound or heat absorption provided by a material.

Absorption spectroscopy Analytical technique involving measuring the amount of energy absorbed by a compound.

Absorptivity The fraction of the energy incident on a body that is absorbed by that body (absorbance). Relating to thermal radiation and acoustics.

Acceleration loss The energy input necessary to accelerate air to a higher velocity.

Acceptable air quality When workspace air has no known harmful contaminants present which may influence the immediate or future health of the occupants.

Access door An opening providing access to some plant item or ductwork to allow for inspection, cleaning, or maintenance.

Accessibility Ease of access to a plant item.

Acclimatization The state of the human body becoming accustomed to a given thermal environment, either heat or cold.

Accuracy The degree of closeness of a measurement to the true reading of the value being measured.

Acid A substance that liberates hydrogen ions in solution.

Acid cleaning The use of acids to clean scale or other deposits from pipes or tanks.

Acid dew point The temperature at which a vapor containing an acid appears as condensate on a cool surface, causing corrosion.

Acid fumes Particles in the air generated by the condensation of acid vapors. They may also arise from sublimation, condensation, or chemical reaction.

Acid mists Mists resulting from a process in which an acid is produced or used.

Acidic smuts Solid and liquid conglomerates formed by the condensation of water vapor and sulfur trioxide on a cold surface. A typical case is combustion products in a flue, which come into contact with surfaces at temperatures below the flue gas dew point temperature. These products contain metallic sulfate and carbon aqueous particles approximately 1−3 mm in size.

Acoustics The science and understanding of the behavior of sound in the environment.

Acoustic environment The level of sound in an internal or external environment, related to a standard prescribed level.

Acoustic force An acoustic field used to enhance the evaporation, coagulation, or condensation of particulate matter.

Acoustic insulation A material that has the ability to absorb sound energy.

Acoustic leak detection A technique used to detect cracks in a building structure through which infiltration or exfiltration is taking place. A constant source of high frequency sound is produced within the building and a detector microphone positioned outside the building detects the weak spots by recording a volume increase.

Acoustic muff (muffler) Sound-absorbing material that is placed around a noisy item in a plant.

Acoustic pod Sound-absorbing material inserted in ductwork to absorb sound.

Action level A term describing the airborne concentration that triggers certain provisions of a regulation; generally, but not always, it is 50% of the PEL value.

Activated alumina Hydrated aluminum oxide, a granular desiccant activated at high temperature that absorbs moisture and gases.

Activated carbon or activated charcoal Carbon in the form of charcoal granules, which has an affinity to adsorb many gases and vapors and, in so doing, removes odors. It is manufactured by exposing coal, coconut shells, or peat to steam at 800°C−900°C.

Activated carbon filter A canister filter containing activated carbon.

Active site The position on an adsorbate surface where adsorbate molecules are trapped.

Activity The nature of the work carried out by a person, measured in Met units. Also, the decay rate of radioactive particles.

Actuator An automatic device providing valve or damper control of fluid flow by means of an electrical, hydraulic, or pneumatic motor.

Acute The immediate influence of given concentrations of an air pollutant on the health of a person.

Acute effects Symptoms of injury or other physical manifestations that follow an acute exposure.

Acute exposure Exposure to a high level or concentration of a pollutant for a relatively short time.

Adhesion The phenomenon of particles sticking to the fiber surfaces in a filter.

Adhesion of particles Small particles experience adhesion forces, allowing them to attach to surfaces. These forces may be made up from surface tension of liquid films, or London (Van der Waals) forces.

Adiabatic A process that takes place without loss or gain of heat, for example, when air rises it expands adiabatically in the atmosphere.

Adiabatic lapse rate The adiabatic temperature change that takes place with height of a rising (or falling) parcel of air, approximately $-1^{\circ}C/100$ m.

Adiabatic mixing A mixing process that takes place without the removal or addition of heat.

Adiabatic saturation temperature The temperature attained after an adiabatic process.

Adjustable flow rate A controlled state of flow that is achieved by means of a damper or valve.

Adjustable grill See Grill.

Adjustable pattern air diffuser An air diffuser incorporating a device that allows the direction of the leaving air to be adjusted.

Adjustable pitch fan A fan in which the pitch angle can be set to provide the required airflow rate. The pitch angle may be preset or controlled with the fan running.

Administrative controls The working method that allows workers to be exposed to set exposures of contaminants in the workplace. Lower exposure levels are achieved by the use of work assignment, job rotation, with set periods working away from the hazard zone.

Adsorbate The contaminant collected by an adsorber.

Adsorbent A medium that traps vapor or gaseous contaminants on its surface by chemical and physical properties.

Adsorbent, regenerable An adsorber, which is treated when fully contaminated in order to restore its original collection properties.

Adsorber An adsorbent material used in the adsorption process.

Adsorption A physical process in which a molecule of a vapor or gas (adsorbate) is condensed on and taken up by the surface of a porous material (adsorbent) such as silica gel or activated carbon.

Adventitious opening Any unintended openings in a building structure, such as cracks through which infiltration or exfiltration can occur. The terms unintentional opening, fortuitous leakage, and crackage are also used.

Aerobic microbes Microbes used in a biofiltration process in which gaseous pollutants are removed from a process gas stream by aerobic digestion.

Aerodynamic diameter The diameter of a unit-density sphere that has the same settling velocity in air as the particle in question.

Aerosol A special class of particulate consisting of colloidal suspensions larger than molecular size, but not large enough to settle under gravity.

Aerosol, monodisperse An aerosol with a size-distribution function described by a geometrical standard deviation less than 1.15. If the deviation is between 1.15 and 1.5, it is classified as a quasimonodisperse aerosol.

Aerosol, polydisperse An aerosol with a geometric standard of deviation of size-distribution greater than 1.5.

Aerosolize Mixing a gas and a liquid to form microscopic ($0.5-10\ \mu m$) airborne droplets.

After filter A filter located in the flow stream after another filter to remove any particulate matter that may have passed the upstream unit.

After heater An air or water heater positioned in a long duct or pipe work run to remedy distribution heat losses.

Afterburner A unit installed after the main combustion zone in a process to further provide combustion to reduce the emission of certain pollutants. It may be

1. A direct-flame afterburner, in which an auxiliary fuel burner provides all the heat from a flame, or
2. A catalytic afterburner, in which the surface action of catalysts allows incineration to take place at a temperature lower than a direct flame, reducing the auxiliary heat required, or
3. A recuperative afterburner, a heat exchanger combined with a direct flame unit that preheats the combustion gases.

Age of air A statistical measure of the air in a space.

Agglomeration The process of the clustering or adhering together of a number of small particles.

Aging Any chemical process that reduces the effectiveness of the properties of a material.

Agitator A device used to stir or agitate a fluid, reducing stagnation or stratification.

Air

Air The composition of gases that make up the earth's atmosphere, approximately 79% nitrogen and 21% oxygen. "Pure air" has no definite meaning regarding the proportion of these gases; this term is used to imply the absence of industrial particulate matter.

Air after-treatment Treatment of the supply air after the main treatment.

Air barrier A device that provides a jet barrier between two zones in a building. See Air curtain.

Air bleed-in Openings at the end of a branch duct that provide increased flow rates for the transportation of heavy particulate matter.

Air change coefficient The ratio of air volume flow rate and the volume of the space.

Air change efficiency The ratio of the nominal time constant and the time taken to change the air within a space.

Air change rate The ratio of the volume of air supplied or extracted to the volume of the space usually measured in air changes per hour (ach) and normally related to the fresh air change rate.

Air cleaner A device that removes airborne contaminants from air.

Air cleaning equipment Equipment that removes airborne contaminants, either

- Equipment to clean the ambient air, normally classified as air filters, or
- Equipment that will collect large concentrations of particulate matter from industrial processes, called dust collectors.

Air conditioning The process of air treatment in which the temperature, moisture content, purity, and distribution are controlled to set conditions. The resulting conditions may be chosen for human thermal comfort or to meet the requirements of a manufacturing process.

Air conditioning installation Any plant assembly which can achieve the conditions outlined above.

Air conditioning, partial The process of air conditioning which is lacking one or more of the ideals, that is, no moisture control, no heating, or no cooling.

Air contaminants Aerosols, gases, vapors, or dusts which may cause adverse effects if discharged into the indoor or outdoor atmosphere.

Air core area The gross area of the openings of an air terminal device (ATD).

Air core velocity The air flow volume divided by the core area of an air terminal device (ATD).

Air current Air movement in a space produced by either thermal or mechanical means.

Air curtain A high-velocity air jet that provides an air barrier between two different building or work zones.

Air diffuser A circular, linear, rectangular, or square device from which supply air is discharged in a controlled direction.

Air-diffusing ceiling The process of air distribution into a given space through a perforated ceiling.

Air diffusion The process of air distribution into a space by means of an air terminal unit. The components of air diffusion are

- Air terminal devices (ATD), components of the installation that are designed for the purpose of achieving the predetermined movement of air into or from the treated space, for example, grills and diffusers;
- Complementary accessories for ATD components of the installation that are used in conjunction with, and in several cases form an integral part of, the air terminal device for the purpose of achieving the predetermined profile or rate of flow into or from the air terminal device (e.g., air flow controllers, dampers, flow equalizers, and baffles); and
- Fixed accessories for ATD, components of the installation that assist the fitting and fixing into place and/or maintenance of the air terminal devices and their complementary accessories (e.g., plaster frames and snap-in fasteners).

Air discharge velocity The average velocity of the air discharged from the opening of an ATD, $V_c/(C_d A_{rf})$.

Air distribution The mechanical or natural delivery of air into a space in a designed manner, or the transport of air in ductwork. The components of air distribution are

- Elements of air distribution, the components involved in ensuring air distribution from the plant room to the space, for example, ductwork and damper;
- Air terminal units (ATUs), item at the end of a duct run to control velocity, pressure, flow rate, and/or temperature;
- Accessories of distribution, any components to keep the unit in place, or allow maintenance to be carried out.

Air douches Versions of fresh air jets in which air is discharged from a wide nozzle or plenum at a relatively low velocity close to the worker.

Air, dry A mixture of air with no moisture content present.

Air duct An enclosure that conveys air from one location to another.

Air ejectors (jet pumps) A device injecting high-velocity (pressurized) primary air into the secondary air. It allows air to be moved without passing through a fan, it may be a simple or Venturi-type ejector.

Air envelope The surface of a jet produced by an ATD with the same velocity.

Air excess Air provided to a combustion or ventilating process in excess of that theoretically required for the combustion process.

Air exfiltration The uncontrolled rate of air interchange from a space to outdoors due to density variations, or by space pressurization by means of a fan.

Air exhaust The volume or mass flow of air that is mechanically rejected to outdoors due to its unsuitability for further use within the space. See also Extract air (EA) classification.

Air extraction The removal of contaminated air from a space, either directly to outdoors or recirculated back to the space after suitable treatment. See also Extract air (ETA) classification.

Air extraction cooker hood A range hood positioned above kitchen cooking equipment, designed to collect without spillage the plume generated at the range.

Air filters

Air filter A device that removes particulate matter from a gas flowing through it. These are classified as

- Absolute: A high-efficiency particulate air filter that is at least 99.79% efficient in the removal of thermally generated monodisperse dioctylphthalate smoke particles with a diameter of 0.3 μm, also known as a HEPA filter.
- Activated carbon: An adsorption filter that makes use of an activated carbon bed to remove odors and various gases from a ventilating or gas cleaning system.
- After: A filter that is positioned after a coarse filter, to ensure that the air entering a space is at the required degree of cleanliness.
- Automatic roll: A motor-driven continuous roll of material, either viscous or fiber, which ensures a set pressure drop in the airflow. Its operation depends on sensors that record the pressure differential on either side of the filter.
- Blow-off: A filter, such as a bag filter, that has the dust burden (cake) on the surface blown off by strong blasts of compressed air.
- Cleanable: A filter that can be cleaned by manual or automatic washing.
- Coarse: A filter fitted before a HEPA filter to remove the larger particulate matter to ensure that the HEPA filter has a longer life without clogging up.
- Dry: A filter manufactured from cotton wool, glass−fiber fabric, pleated paper, foamed polyurethane, cellular polythene, or other suitable materials. As the name suggests, the filter is dry in nature and has no oil coating.
- Electrostatic: A filter in which a high voltage is used to collect the particulate matter onto earthed plates.
- Fabric: A dry throwaway filter in panel or wedge form mounted in supporting frames.
- Grease: A filter used in kitchen air extraction systems to prevent the contamination of the ductwork system with grease, which would be a fire hazard.
- HEPA: See Absolute.
- Impingement: The process by which particulate matter is stopped in its path by colliding with a surface, normally a surface coated with oil.
- Panel: Any type of filter material mounted in a rigid frame. The frame must allow only the absolute minimum of particulates to pass between the frame and the sides of the filter material.
- Pre: A prefilter is placed before the main filter to extend the life of the main filter.
- Rotary viscous: A continuously rotating oil-coated roll of material. The filter resistance is kept low by a motor drive that ensures a clean surface is always presented to the airflow.
- Self-cleaning: A filter similar to the rotary viscous in which the filter plates are passed through an oil bath to remove the collected dust.
- Terminal: The ultimate filter at the end of a run, which ensures that the airflow entering the room is at the required design purity.
- Throwaway: A filter which is thrown away after collecting its dust burden, no attempt being made to clean it.
- Viscous: As the rotary viscous or fixed-cell type, in which the particulate matter sticks to the oil-coated material.
- Wet: The various forms of washers and scrubbers.

Air filter cell The material that forms the active part of a filter.

Air filter dust-holding capacity The amount of dust, by weight, retained by a filter under a standard test such as EN779.

Air filter efficiency The ability of a filter to remove dust from the air, expressed in terms of the contaminant concentrations upstream and downstream of the filter. It may be obtained by

1. Weight test (gravimetric),
2. Dust spot test,
3. Arrestance tests, or
4. Particle counting.

Air filter medium The material type from which the filter is constructed.

Air filter performance An overall assessment of the collection efficiency, pressure drop, flow rate, fire rating, health aspects, and behavior of a filter in the environment in which it is used.

Air filter resistance The pressure drop created between the upstream and downstream faces of a filter increases as the dust burden.

Air filter resistance, final The maximum pressure drop allowed across a filter to ensure that the design airflow rate is achieved.

Air filter test Any test to determine filter efficiency, flow rates, or other characteristics by some preset method.

Air free area (A_F) The sum of the smallest areas of the cross-section of the opening of an ATD.

Air free area ratio (A_{RF}) The ratio of free area to the core area.

Air grill An entry or exit in a duct consisting of a mesh or lattice opening; it may be fixed (unadjustable) or adjustable.

Air grill, adjustable An air grille with manual or automatic adjustable louvers.

Air handling function The ability of an air handling plant to perform a given type of air treatment.

Air handling unit A self-contained unit for the introduction or removal of air from a space. It consists of one or more of the following plant items: heating or cooling coils, filters, fans, damper provision for mixing and recirculation, humidification controls, and acoustic treatment.

Air heating and cooling coil As tube or plate heat exchanger in which air, the secondary fluid, is either heated or cooled with a primary medium.

Air humidity The moisture content of air. See Absolute humidity and Relative humidity.

Air infiltration The uncontrolled air interchange through structural imperfections and other openings into a space, due to natural convection, rising currents, or wind forces over a building.

Air infiltration, balanced Infiltration that takes place at a constant or balanced rate.

Air infiltration, unbalanced Infiltration that takes place at an unbalanced or variable rate.

Air inlet Any opening that provides air for any purpose, either from outside or inside the building.

Air leakage See Air infiltration.

Air leakage rate The leakage of air through an enclosure such as a building or ductwork, expressed as air loss in L/s per meter run, or as a percentage loss of the total volume.

Air, make-up The air quantity necessary to replace air extracted from the space, this may be required for combustion or process work and may be obtained by mechanical or natural means.

Air mass flow rate The mass flow of air or any other fluid, expressed in kg/s.

Air monitoring The process of continuous sampling and measuring of the quantity of pollutants present in indoor or outdoor air.

Air movement The direction and velocity of air within an enclosure.

Air pollutant Any undesirable element present in indoor or outdoor air, such as air contaminants and moisture.

Air pollution The presence of foreign matter (gaseous or particulate or combinations of both), bacteria, sound, or other undesirable elements in air, which is detrimental to the health or welfare of man, animals, plants, or materials.

Air pretreatment See Air treatment.

Air pressure Atmospheric air pressure or static, velocity or total pressure in a ventilating system, Pa.

Air, primary The actual quantity of air injected into a space before secondary air induction occurs.

Air-purifying respirator A respirator that removes airborne contaminants, such as particulates, gases, vapors, and fumes, from ambient air through filtration, absorption, adsorption, or chemical reactions on the media contained in the cartridge or filter.

Air quality The concentration of one or more pollutants in the air, measured in either ppm or $\mu g/m^3$. It may be related to pollution sources inside an enclosure or outdoor air. Temperature and moisture conditions may also be considered.

Air quality standard (AQS) A standard providing a level beyond which air pollutants in the atmosphere can cause damage to plants, animals, or materials. The concentration and time factors have to be considered.

Air, recirculated Any air that is mechanically extracted from a space that, after suitable treatment, is returned to the space.

Air recirculation The process of returning exhaust air to the air-treatment plant and after treatment returning it to the space.

Air relief The release of air from a space, either by pressure difference or by mechanical extraction.

Air resource management The enforcement of set standards to reduce contamination supported by control regulations, planning, and quality testing facilities.

Air, return The air returned from a space for processing (also called recirculated air).

Air, sample The air obtained for analytical testing.

Air sampling The process of collection of an air sample to determine its contamination level and dust concentration.

Air, secondary Air entrained in a primary airflow, or the additional air supplied to a combustion process.

Air sedimentation A measuring technique for particulate matter using a micromeorograph, to determine the Stokes number of fall.

Air space The gap between two surfaces containing either still or moving air.

Air spread The maximum horizontal distance between two vertical or horizontal planes of equal velocity in a jet, tangential to the supply envelope from the ATD.

Air stratification The layering of air within a space due to density differences.

Air stream A defined air current within an enclosure, resulting from natural thermal air movement or from mechanical air movement produced by a jet.

Air supply The quantity and condition of treated or untreated air discharged into a space.

Air terminal device (ATD) A specially designed component in an installation, which provides a predetermined air movement into or from a treated space. They are classified as

- Automatically controlled, if they have moving parts to change the local conditions of temperature, humidity, pressure difference, airflow rate, and levels of pollution,
- Fixed, if they are preset to achieve the desired results and have no moving parts, or
- Manually adjusted, if adjustments are made manually by the service engineer or the occupants.

Air terminal unit Air distribution equipment that provides set conditions by the mixing of primary and secondary air. The device

may be fixed, having no control or means of manual adjustment, automatically controlled or manually controlled. If automatically controlled, a sensor is used to indicate any required change in temperature, humidity, airflow rate, pressure, or CO_2 entering the room.

Air terminal unit assembly The air terminal unit, consisting of casing, mixing section, flow control, heating, cooling, filters, fans, and sound attenuators.

Air throw The maximum distance from the outlet of an ATD to a plane tangential to the jet envelope and perpendicular to the initial jet section where the velocity is reduced to a predetermined level.

Air transfer Air that is allowed to pass either mechanically or naturally from one zone to another. In the mechanical case, the transfer volume is controlled by means of suitable control valves.

Air transfer device A mechanical device that allows controlled flow of air from one room or zone to another.

Air treatment Any technique used to control the temperature, moisture content, or levels of dusts, gases, vapors, pollens, bacteria or viruses in air.

Air treatment, thermodynamic Relating to the various thermodynamic changes that occur in the specific volume, enthalpy, and wet and dry bulb temperatures of treated air.

Air turning vane A vane fitted in a ductwork bend to reduce pressure loss.

Air type Classification of air at a specific point in its passage through an air conditioning or ventilation system, either in the duct or the space, for example, outdoor air, supply air, treated air, recirculated air, and extract air.

Air, vitiated Contaminated or polluted air unsuitable for a given application.

Air volume flow rate (q_v) The volume of air or gas moved per unit of time.

Air washer One of the following devices that adds water to an airstream to increase the moisture content of the leaving air.

- Capillary cell: An arrangement of porous pads sprayed with water, through which the air to be treated is passed.
- Saturation efficiency: Relating to air washers, the percentage of water added to the flowing air supply compared to that which must be added for the air to leave fully saturated.
- Spinning disk: A rotating disk onto which a fine jet of water impinges, resulting in fine droplets that add moisture to the air supply.
- Spray Banks of water sprays that inject atomized water particles into the airstream.

Airflow Steady or unsteady air mass or volume movement past a fixed point, either in a duct or in a free space.

Airflow model A mathematical or computer model of the airflow path within an enclosure.

Airflow pattern The air distribution within a space resulting from discharges from a fan or ductwork.

Airflow rate The speed of volume or mass airflow that takes place in a duct or space.

Airflow rate controller Any mechanical, pneumatic, or electrical device that controls the air flow rate into or out of a space by increasing or decreasing the flow resistance. See Dampers and Valve.

Airflow straighteners Vanes in a ductwork section positioned to reduce turbulence after a change in section.

Airlock A two- or three-door enclosure providing access to a clean room that reduces the air leakage of external polluted air into the clean room.

Airlock, active An airlock connected to an air treatment device.

Airlock, passive An airlock that is not connected to the air conditioning system.

Airtightness ductwork class A description of the quality of a ductwork system and its ability to contain air with the minimum of air or gas leakage, classified based on air leakage factor f, expressed in $L/s/m^2$ or $m^3/s/m^2$, a function of pressure p in Pa.

Airtightness class	f (maximum)
A	$0.027\ p_s^{0.65}$
B	$0.009\ p_s^{0.65}$
C	$0.003\ p_s^{0.65}$
D	$0.001\ p_s^{0.65}$

Aitken nuclei Particles, generally with diameters less than 0.1 μm, that are true aerosols when they form the nucleus for condensation or ice formation.

Algorithm A method of calculation that produces a control output by operating on an error signal or a time series of error signals.

Alkali, alkaline A substance that neutralizes acids.

Allergens Pollutants that may cause an allergy, including pollens, dusts, animal dander, insect debris, mold, and fungi spores.

Allergic alveolitis An allergic response to inhalation of organic particles that involves inflammation of the small terminal branches of the bronchioles. Symptoms include coughing, increased production of mucus, fever, fatigue, and muscle aches.

Allergic contact dermatitis Skin condition that occurs in response to exposure to sensitizing material. It is characterized by redness, swelling, and cracking, and sometimes more severe reactions involving the entire immune system.

Allergic reaction A body reaction due to exposure to an allergen.

Allergic response The release of antibodies by the immune system in response to foreign molecules in the body.

Allergy A hypersensitivity reaction of the human body due to a particular substance.

Allowable exposure time (AET) The recommended maximum exposure time allowed for an operator in a workspace when subject to a physical or biological pollutant.

Alveolar fraction Particles with approximate aerodynamic diameters of 0.5–3 μm.

Alveoli The small terminal air sacs in the lungs, through which gas exchange between the blood and the inhaled air takes place.

Amber zone A ventilation containment zone used in the atomic energy industry.

Ambient air Air from the surroundings; used to describe the air pollution concentrations in the open air as compared to the point of generation within the workspace.

Amorphous silica The noncrystalline forms of silica or quartz.

Amplification The reflection of a greater amount of sound than originally impacted the surface.

Analog A continuously variable function in a control system ranging from off to full flow.

Analyte The components of an air sample to be measured directly or indirectly.

Anechoic A room that has a low degree of acoustic reverberation, such as an anechoic test chamber.

Anemometer An instrument used to measure the velocity of air or gas.

Aneroid gauge A gauge used for the measurement of static, velocity, or total pressure with a pitot tube.

Aneroid gauge, electronic An aneroid gauge with the advantage of being able to integrate the velocity pressure directly into velocity.

Anesthetic gases Narcotic gases which when inhaled give a feeling of well-being followed by unconsciousness.

Angle factor The geometrical shape factor used in calculating radiation exchange between surfaces *i* and *j*.

Angle of divergence The leaving angle of an air jet from an outlet, or the angle of change in a ductwork section.

Angle of transformation piece The angle between the opposite faces of a converging or diverging ductwork section.

Anisokinetic Not isokinetic; a sample collected at a velocity different from that of the airflow in the ventilation system.

Antibodies Specific substances within the human body formed in response to invasion by an antigen.

Anti-C Anticontamination protective clothing used in the radiation industry.

Antisneakage baffles Baffles in an electrostatic precipitator that prevent untreated gas from bypassing the active treatment zone.

Antivibration mounting A designed support placed between a rotating or reciprocating machine and the building structure to reduce the transmission of vibration.

Antidegeneration clause A term used in the U.S. Clean Air Acts implying not only that air quality standards must be achieved, but also that nowhere should air quality be allowed to worsen, even if standards are exceeded.

Antoine's equation An equation used for calculating saturation vapor pressure.

Apparatus dew point For practical purposes, the average temperature of a cooling coil surface.

Appliance A functional device such as a fan or a combination of package units.

Approach velocity The velocity of airflow into a filter bank or heat exchanger.

Approved codes of practice (ACOPs) Legislation in the United Kingdom dealing with the safety aspects of dangerous materials.

Arc A high-voltage discharge to ground.

Archimedes number A dimensionless number that relates the ratio of buoyancy forces to momentum forces, expressed in many forms depending on the nature of the Reynolds number.

Area, actual filter face The area of a filter perpendicular to the flow direction.

Area, body surface (A_{Du}) The total surface area of a nude person, determined by the Du Bois equation, $A_{Du} = 0.203\ W^{0.425}\ H^{0.725}$

Area, duct cross-sectional The duct area perpendicular to the direction of gas flow.

Area, face The frontal face area of a filter or coil through which a gas passes.

Area, filter medium The total area of the filter medium that is available for gas flow.

Area, filter surface The area formed by the filter medium normal to the direction of flow.

Area, nominal filter face The frontal face area of a filter including the header frame.

Area samples Samples taken by placing the sampling train in a fixed location in the workplace.

Arrestance A measure of the ability of a filter to remove a standard test dust from the air under test conditions.

Arrestance, average The weighted percentage ratio of the total amount of standardized test dust retained by a filter to the total amount of dust fed into the filter in order to achieve a specified final pressure drop.

Arrestance, initial The arrestance value recorded during the first loading cycle in a filter test.

Arrestment The removal of a pollutant from a gas stream by an arrester or a cleaning device.

As low as reasonably achievable (ALARA) A standard for controlling and reducing worker exposure to pollutants.

Asbestos A natural fibrous form of several silicate minerals of the following types.

- Chrysotile (white),
- Amosite (brown),
- Crocidolite (blue).

Each of the above forms has different OEL requirements.

Asbestosis A disease caused by the inhalation of asbestos fibers.

Ash The nonvolatile inorganic residue left when a fuel is fully combusted.

Aspect ratio The ratio of the length of a grill or duct to the breadth.

Asphyxiant Simple asphyxiants are inert gases which deplete the oxygen supply in the breathing air to below the critical value of 18% by volume, such as gaseous fuels or nitrogen. Chemical asphyxiants, such as carbon monoxide and hydrogen cyanide, have a direct biological effect.

Assigned protection factor (APF) The minimum level of respiratory protection that a respirator can be expected to provide, assuming it is properly fitted, worn, and functioning. APFs are assigned by NIOSH.

Assmann psychrometer An apparatus that uses a clockwork or electrical fan located above wet and dry bulb thermometers, positioned in a cylinder to provide a set airflow rate over the thermometer bulbs for accurate temperature readings of both bulbs.

Asthma A diseased condition of the lungs caused by pollution and other factors.

Atmosphere The gaseous envelope of air that surrounds the earth, held together by gravitational attraction. It consists of 79.1% nitrogen and 20.9% oxygen by volume, with approximately 0.03% CO_2, traces of the noble gases (argon, krypton, xenon, neon, and helium), water vapor, organic matter, ammonia, ozone, various salts, and suspended solid particulates.

Atmospheric dust concentration The dust burden present in atmospheric air measured in mg/m^3.

Atmospheric dust spot efficiency A test to measure the extent to which a filter paper is soiled after dust-laden air has passed through it.

Atmospheric pressure The pressure due to the air column above sea level at 45° latitude at 0°C is 101.325 kPa.

Atmospheric stability The state of the atmosphere in which vertical air movement is restricted.

Atom The smallest particle of an element that retains the characteristics of the element.

Atomic absorption An analytical method in which the sample is converted into a vapor by passing it through a flame or other energy source and the absorbance at a particular wavelength is measured and compared with that of a reference substance. The absorbance measured is proportional to the concentration of that substance in the sample.

Atomic number The number of protons in the nucleus of an atom of an element. Also, the number of electrons in a neutral atom of the element.

Atomic weight The relative atomic mass of a substance.

Atomization The process by which a solid or liquid is reduced to very small particles or droplets, as in a fine spray.

Attenuation The reduction of sound or radiant energy.

Attenuator A unit fitted in air-handling or other noise-producing equipment to absorb sound.

Audiogram The report produced by an audiometry test, showing measured hearing threshold levels at frequencies of 500, 1000, 2000, 3000, 4000, and 6000 Hz.

Audiometry The assessment of damage to a person's hearing.

Automatic control system A control system that reacts to a change or imbalance of a variable. It controls the imbalance by adjusting other variables to restore the system to the set control condition.

Auxiliary fuel firing Combustion of an auxiliary fuel to provide additional heat to an incinerator in order to either dry or ignite the waste material, to maintain ignition, and to ensure complete combustion of solids, liquids, and gases in the incinerator.

Averaging time The time period over which the measuring procedure provides a single value.

Avogadro's hypothesis States that equal volumes of different gases at the same pressure and temperature contain the same number of molecules. Hence, the volume occupied by any gas whose mass is numerically equal to its molecular weight is a constant quantity.

Axial-flow fan A fan positioned in a cylindrical casing in which the air enters and leaves the impeller in a direction parallel to the casing axis. The fan may have fixed-pitch blades or variable-pitch blades.

B

Bacharach smoke scale A scale of 10 shades of white to black for a smoke stain formed in a prescribed manner by a pumping action through a filter paper, used for the assessment of smoke from combustion appliances.

Back corona (back ionization) The discharge phenomenon that takes place from a particulate layer in an electrostatic precipitator.

Backdraft dampers Dampers installed in a system to prevent reversed flow of gases.

Backdrafting The action of extracted air or the products of combustion returning to a space due to adverse high pressures at the point of extraction.

Background level The concentration of a particular substance present in the air without any local source of the pollutant, or the concentration of the pollutant at some distance from its source.

Bacteria The collective name for cellular microorganisms.

Bacteria count The count of bacteria collected on a slide within a duct run or in a space.

Bad air Air that contains any form of contamination.

Baffle A device fitted in equipment or ductwork runs that changes the flow direction.

Baffle chamber A chamber consisting of baffles (slats) that present an obstruction to the flow of dust-laden air. These obstructions slow down the heavy particulate matter, which then settles by gravity into an adjacent hopper.

Bag house An enclosure containing a series of frame-mounted bag filters and a hopper collection unit.

Balanced design method A method of duct sizing to ensure the correct airflow rate in all branches, also known as the static pressure balancing method.

Balancing The process of obtaining the design airflow in hoods and ductwork by the use of static pressure balancing or blast gates. In the case of a liquid, it is achieved by means of balancing valves.

Bar The unit of pressure equal to 100 kPa, approximately one atmosphere. It is not strictly an SI unit due to its magnitude being 10^2 not 10^3 as is required.

Barograph An instrument for recording atmospheric pressure.

Barometer An instrument used to measure the barometric pressure at a given location in the earth's atmosphere.

Basal metabolic rate The rate of oxygen consumption by a person at rest.

Basic thermal insulation of a garment The thermal resistance provided by a given item of clothing.

Battery, heat exchange coil A device used to either heat or cool air in a duct run.

Bauxite lung See Shaver's disease.

Beaufort scale The scale used for estimating and reporting wind forces, in which 0 is calm (velocity less than 0.5 m/s) and 12 is a hurricane.

Bed depth The depth of adsorbent material through which the gas being treated passes.

BEI See Biological exposure index.

Bend A duct or pipe fitting which changes the direction of flow through a specified angle.

Benign Not associated with negative health effects, self-limiting.

Bernoulli effect At any point in a conduit through which a liquid is flowing, the sum of pressure energy, potential energy, and kinetic energy is constant.

Bias A consistent deviation from the true value in the results of a measurement.

Bifurcated fan An axial flow fan that directs the airstream around the motor, which is enclosed in a protective casing. It is used for handling corrosive, high-temperature, and explosive dusts, vapors, and gases.

Bimetal thermometer A thermometer that uses two dissimilar bars of metals (with different rates of linear thermal expansion) riveted together. A variation in temperature produces a bending moment on the bar, which is magnified by a lever to record temperature on a dial.

Bio-aerosol A general term relating to airborne viruses, bacteria, pollen, and fungus spores.

Bioclean classes The various standards of contaminant control to which clean rooms are held.

Biofiltration The process by which gaseous pollutants are removed from a gas stream by aerobic digestion.

Biological agent Any of a range of microorganisms which have an adverse effect on human health, including those genetically modified cell cultures and endoparasites.

Biological exposure index (BEI) Reference values developed by ACGHI as guidelines for the evaluation of potential health hazards.

Biological half-life The time required for one-half of the material accumulated in a tissue to be removed.

Bioprocess The facility necessary in a biotechnology clean room to protect the worker, the product, and the environment.

Biosafety The safety requirements necessary in biotechnology clean rooms, fume cupboards, stores, etc.

Black body A hypothetical body that has an absorptance and an emissivity of unity, that is, it absorbs all the radiation falling on it.

Blackness test A filter test for air or the products of combustion, in which a dust spot is formed on filter paper, the degree of reflection is determined by a light meter, and the result is related to a standard.

Blade shape Relating to the shape of a fan blade, either airfoil, solid, radial, forward-curved, and backward-curved.

Blank-corrected Data that have had trace contamination amounts (detected in a nonexposed sample) deducted from the total amount of contaminant detected in the sampling media.

Blast gates Adjustable sliding gates (dampers) on extract ductwork used to balance the system.

Blender An enclosure in which two different fluid streams are mixed.

Blower A device producing air movement.

Blow-out panels Safety panels provided to some process, such as an oven, that will protect the structure in case of an explosion by releasing the force of the explosion.

Blow-through unit An air-handling unit downstream of the supply fan.

Body core temperature increase The increase in body core temperature that takes place due to the inability of the body to get rid of heat.

Body heat gain or loss The positive or negative change in the heat content of the human body caused by an imbalance between heat production and heat loss.

Body heat storage The heat stored in the body due to metabolism.
Body height The standing height of a human body, H_b, in m.
Body mass The mass of the unclothed human body in kg, which is a measure of its inertia, or resistance to any alteration in its motion. The mass of a given body is the same anywhere on the earth or in space.
Body mass loss, gross The reduction in body mass over a given period of time, Δm_g, in kg.
Body mass loss, respiration The loss of body mass due to respiratory evaporation, in kg.
Body mass loss, sweat The body mass loss due to sweating, Δm_{sw}, in kg.
Body mass variation for solids The variation in body mass due to food intake and subsequent excretion, Δm_{sol}, in kg.
Body mass variation for water The body mass variation due to intake of water and excretion of urine, Δm_{wat}, in kg.
Body surface area The total surface area of the human body, determined from the Du Bois Index.
Body surface area covered by clothing The percentage of the body surface area covered by clothing.
Body temperature The temperature of a human body, either the body core temperature, the mean temperature of the body, or the temperature at some point on the skin. Also, the temperature of a surface which is radiating, conducting, or convecting heat.
Body thermal sensation The response of the body to changes in the thermal environment, relating to moisture, air movement, or temperature.
Boiling point The temperature at which the vapor pressure of a liquid is equal to the external pressure, and the liquid changes phase into the vapor state.
Boundary conditions The actual environmental conditions within a controlled zone.
Boundary layer A layer of fluid, extending from the boundary into the bulk of the fluid, in which fluid motion is influenced by the frictional drag at the boundary.
Boundary layer insulation The thermal resistance at the boundary of the skin or clothing, I_a, in Clo m^2/W.
Bourdon gauge A pressure gauge in which the sensing element is constructed from a coiled flattened tube closed at one end.
Boyle's law See Laws of perfect gases.
Bracket A support for any item of equipment either on the main building structure or on another item of equipment.
Branch duct entry The position at which a branch duct enters the main duct run.
Breakthrough A condition that exists when the backup section of a sorbent tube is found to contain 20%–25% of the total amount of contaminant captured in the front section.
Breakthrough time In the context of chemical protective clothing, the time between initial contact of the chemical on the barrier material surface and the analytical detection of the chemical on the other side of the material.
Breathing zone The space around an operator in which breathing occurs, normally taken as being a hemisphere of radius 0.3 m circumscribing the ears, the top of the head and the larynx.
Breathing zone sample The sample of a contaminant or contaminants collected in the working operator's breathing zone.
BRI See Building-related illness.
Bronchi The larger air passages of the lungs (bronchus, singular).
Bronchioles The very small airways of the lungs that terminate in the alveoli.
Bronchitis Inflammation of the lining of the bronchi, which may be caused infection or by the effect of pollution.
Bronchogenic carcinoma A lung cancer associated with asbestos exposure.
Brownian diffusion (Brownian motion) The diffusion of particles due to the erratic random movement of microscopic particles in a disperse phase, such as smoke particles in air.
Bubble plate A plate or cap used in absorption equipment.
Buffer zone The zone of clean air in a room adjacent to a clean room; an air lock.
Building automation system Sometimes called building management system, a system that controls the mechanical and electrical (M&E) services within a building.
Building related illness (BRI) Any health problem related to poor air quality, due to equipment malfunction or contaminants in buildings. See also Sick building syndrome (SBS).
Building Services All of the mechanical, air, water, electrical, and transport services required to provide satisfactory environmental conditions within a building, with due consideration of the health and safety of the occupants and energy conservation.
Bulging or caving of a duct The maximum deflection that occurs in the sides of a duct due to negative (caving) or positive (bulging) pressure differences. The reference plane is that existing with no pressure difference.
Bulk density Apparent density of bulk solids, kg/m^3.
Buoyancy The upward force exerted on any object immersed in a fluid of greater density. Hot pollutant gases rising in cooler air have positive buoyancy. A volume of gas denser than the surrounding air has negative buoyancy.
Buoyancy effect See Buoyancy.
Butt connection The interface between two pieces of metal that are joined together by welding.
Butterfly damper or valve See Dampers and Valve.
Bypass The provision for a secondary flow path from the main flow in duct or pipe flow.
Bypass damper A damper positioned in the airstream which allows a given quantity of air to be diverted to another run or rejected to outdoors.
Bypass factor The ratio of the secondary flow to the sum of the main flow and secondary flow.
Byssinosis Reactive airway disease associated with inhalation of organic textile fibers, such as cotton, flax, linen, and hemp.

C

Ca A notation used by NIOSH to indicate that a substance is considered a known or potential occupational carcinogen.
Calibration The adjustment of a measuring instrument to ensure that it is giving the correct readings.
Calm A meteorological term describing a wind speed of less than 0.5 m/s.
Calorific value The measure of the heating capacity of a fuel, usually expressed as the available heat resulting from the complete combustion of that fuel in kJ/kg or kJ/m^3. Gross calorific value includes the heat of condensation of the water vapor in a hydrogen fuel; net calorific value excludes this.
Canopy hoods A capture hood located above a process, designed to provide a suitable capture velocity to ensure the safe removal of the contaminant produced by the process.
Capacity The actual duty of a fan, heater battery, filter, or other item of equipment.
Capacity, total lung (TLC) The volume of gas contained in the lungs at full inhalation.
Capacity, vital The maximum gas volume that can be expired from the lungs following maximum inhalation.
Capture velocity The air velocity necessary at a point in order to capture and transport to the exhaust opening the contaminants being emitted from a process.
Carbon dioxide (CO_2) The gas formed by complete combustion of carbon-containing substances. Also a product of the metabolic process.

Carbon dioxide production The quantity of carbon dioxide exhaled from the human body depends on the metabolic rate.

Carbon monoxide (CO) A colorless odorless gas formed by incomplete combustion. Highly toxic if allowed to accumulate in the blood.

Carboxyhemoglobin A molecule formed by the combination of carbon monoxide and hemoglobin.

Carcinogen Any substance that has been shown to cause cancer in animals or humans.

Carnot cycle The cycle of a perfect heat engine, in which the heat is and rejected at constant temperature and the whole cycle is perfectly reversible.

Carnot principle States that no engine can be more efficient than a reversible engine when both operate between the same temperature limits.

Carrier gas An inert gas that moves the sample through the column of a gas chromatograph.

Cartridge filters Filters normally consisting of nonwoven V-pleated filter paper made into flat panels or cylindrical cartridges.

Cascade impactor An instrument used to sample and separate particulates into a number of successive fractions of different sizes.

Casing An external cover of any plant item, covering the whole of the item or part of it.

Catalyst A substance used to speed up a chemical reaction, including the transformation of certain pollutants present in a combustion process.

Catalytic combustor A device used to remove various solid, liquid, or gaseous pollutants from air or another gas, in which the gas is heated by an open burner between 250°C and 500°C and passed through a catalyst bed in which the organic contaminants are oxidized into harmless by-products.

Catalytic oxidizer Used to promote oxidization of a combustible pollutant.

Ceiling exposure limit The maximum allowed concentration of a contaminant to which a worker may be exposed, set by legislation.

Celsius A temperature scale that sets the freezing point of pure water at atmospheric pressure at 0°C and the boiling point of pure water at atmospheric pressure at 100°C.

Cenosphere Small, hollow, spherical ash particles formed from the combustion of liquid or solid fuels.

Central chambered system A combination of components in a dedicated chamber.

Central nervous system (CNS) The system of the body composed of the brain and spinal cord, which controls important body functions.

Central station A central ventilation or air-conditioning unit that provides treated air to various zones within a building.

Centrally recirculated Exhaust air from one or more treated spaces that is reintroduced through a central unit before air treatment occurs.

Centrifugal A driving force that causes a body to move in a circular path, for example, a centrifugal fan, a centrifugal separator, or a centrifugal pump.

Centrifugal collectors Separators for the removal of particulate matter from gas streams, classified as

- Straight-through cyclones with fixed impellers,
- Straight-through cyclones with moving impellers,
- Scroll collectors,
- Induced-draft fans combined with collectors,
- Reversed-flow cyclones, or
- Multiple cyclones.

See also Cyclone.

Centrifugal fan See Fan.

Chain of custody Documentation necessary to trace sample possession from the time of collection throughout the time of analysis.

Chamber A gas- or dust-tight enclosure.

Chambering Components contained in a chamber erected on-site.

Charcoal cloths A filter manufactured from pretreated woven cellulose fiber cloth.

Charged particles Particles that have a positive or negative electrical charge. The nature of this charge effects the collection of the particles in a precipitator.

Charles' law See Laws of perfect gases.

Chemical agent Any chemical element or compound.

Chemical asphyxiant A substance that interferes with the absorption or utilization of oxygen in the body, for example, carbon monoxide.

Chemisorption An adsorption process in which the solute chemically reacts with the adsorbent to form a new compound.

Chilled water Water that is cooler than ambient temperature, obtained from a refrigeration plant, cooling tower, or a well.

Chiller A heat exchanger in which heat is removed from the warmer water or air.

Chilling temperature See Temperature, chilling.

Chimney A free-standing or built-in structure used to remove the products of combustion from a process or to provide natural ventilation.

Chimney effect See Stack effect.

Chlorinated hydrocarbons Hydrocarbons in which some or all of the hydrogen atoms are replaced by chlorine.

Chlorofluorocarbons (CFCs) Aliphatic carbon compounds containing both chlorine and fluorine atoms.

Chromatography A method of analysis using separation of mixtures based on selective adsorption.

Chronic exposure Long-term repeated exposure to low levels of pollutants, which may cause damage to the health of occupants.

Chrysotile One of the types of fibrous mineral asbestos.

Chute A device used for removing waste material or charging a furnace.

Cilia Fine hairlike structures found in the membranes that line the respiratory tract that assist in particulate removal.

Circulating fan See Fan.

Circulation time The time necessary for complete mixing of a tracer gas in a space.

Class I–III Cabinets Containment cabinets used for various requirements. In a high-risk area, a Class III cabinet would be used. Class I and II cabinets are used in low-risk areas.

Classes for clean rooms Many national standards are in use for the classification of clean rooms. It is recommended that the standard of the country in which they are to be installed be referred to.

Clean in place (CIP) A system used in clean rooms, consisting of tanks, piping, pumps, and associated controls for the distribution of wash and rinse solutions.

Clean-out door A door in an item of equipment or ductwork that provides access to the inside of the unit for cleaning purposes.

Clean room A room in which the environmental conditions of air purity, temperature, moisture content, and air movement are maintained to high and accurate standards.

Clean tunnel A tunnel providing access of operators or production components from one clean area to another.

Clean zone Any area, such as a clean room, in which set standards of air purity and other environmental conditions are maintained.

Cleaning system A system or device that cleans a fluid or solid medium to a given standard.

Clearance samples Samples taken following a lead, asbestos, or other removal action, which must indicate the contaminant concentration to be at or below a specific level before the area can be cleared for normal occupation and work activities.

Cleat A device used to connect two or more items together.

Clo The SI unit of insulation value of clothing; 1 Clo = 0.155 m^2 K/W. The term tog may be seen in the literature but tog is not an SI unit and should not be used.

Closed face sampling Sampling performed through a small hole in the top of a filter cassette.

Closed system The primary containment of a process in a biotechnology clean room.

Cloth-filter collectors A mechanical method of filtration of particulate matter from a gas stream by the use of a number of cloth bags. Its operation is similar to a vacuum cleaner method of removal.

Clothing area factor The ratio f_{cl} of the surface area of a clothed body to the surface area of a nude body.

Clothing insulation The resistance of a uniform layer of clothing covering the entire body that has the same effect as the actual clothing worn on the sensible heat flow under still-air conditions, I_{cl}, in Clo m^2 °C/W.

Clothing insulation, effective The increased body insulation due to clothing, compared to the nude state, the difference between the total insulation and the boundary layer insulation, I_{cle} in Clo m^2 °C/W.

Clothing insulation, minimum requirements The minimum clothing insulation required to maintain body thermal equilibrium at a subnormal level of mean temperature, $IREQ_{min}$, in Clo m^2 °C/W. This represents the highest admissible body cooling in occupational work.

Clothing insulation, neutral requirements The thermal insulation of clothing necessary to provide conditions of thermal neutrality $IREQ_{neutral}$ in Clo m^2 °C/W. It represents the state of no or absolute minimum cooling of the body.

Clothing insulation, required The required clothing insulation to ensure a given body thermal balance.

Clothing insulation, resultant The true level of thermal insulation provided by clothing under given conditions, I_{clr}, in Clo m^2 °C/W.

Clothing surface temperature The actual mean surface temperature of clothing.

Clouds A mass of droplets of water or other liquids remaining at a more or less constant height. Clouds are usually formed by condensation after warm moist air rises by convection into cooler regions and cools by expansion to below its dew point.

Coagulation The process of particulates sticking together on coming into contact. As the process continues, the particle size distribution becomes coarser and settles out.

Coalescence The merging of small drops of a liquid into a larger droplet.

Coanda effect When a jet becomes and remains attached to a surface due to static pressure differences, as in the case of a wall jet.

Coarse solid particles Any solid particle larger than 50 μm, and solid particles contained in or on any liquid particle.

Coated/treated filters Filters that have been coated with a chemical specific for the contaminant to be collected. The coatings enhance the collection by chemically reacting with the contaminant as the air is drawn through the filter.

Cochlea A snail-shaped fluid-filled organ of the inner ear, lined on its inner surface with specialized hair cells that convert sound pressure vibrations into nerve impulses.

Coefficient of discharge A coefficient describing the actual discharge of a fluid jet compared to the theoretical discharge.

Coefficient of hood entry The coefficient describing the pressure drop that occurs when gases flow through a collecting hood or other enclosure.

Coefficient of velocity The coefficient describing the actual velocity of a jet, compared to the theoretical value.

Cohort A group of individuals that share a particular statistical or demographic characteristic, for example, exposure.

Coincident error An error due to the presence of more than one particle in the measurement volume of an optical particle counter.

Cold-generated DOP A cold-generated aerosol dioctylphthalate (DOP) test, used to measure the efficiencies of high-efficiency filters. A hot DOP test may also be used.

Cold stress Physiological stress on the body created by excessive loss of body heat.

Collar A connecting piece used to connect two components of duct or pipe.

Collecting surface area The actual surface area of a filter on which particulate matter is collected, normally greater than the filter face area.

Collection efficiency The efficiency of a collection process, expressed as a percent of theoretical (100%) collection.

Collectors Any device used to collect particulate matter, preventing it from entering the environment.

Combined section of air-handling plant A section of equipment that consists of two or more items of equipment necessary for its operation.

Combustion The chemical process that occurs when a given combination of fuel and oxygen is heated to a given temperature at which the combustible matter burns, with an increase in temperature.

Combustion air The air quantity that has to be supplied to a combustion process to ensure complete combustion. The air quantity may be either theoretical or excess.

Comfort conditions The environmental conditions in a space that will ensure statistically the majority of occupants are comfortable. Relates to thermal, acoustic, and visual conditions.

Comfort indices Relates to the many comfort scales, either empirical or calculated, that are in common use.

Comfort ventilation The minimum amount of air that must be provided to a worker to ensure

- Relative air velocity for comfort;
- Body heat removal; and
- Removal of body odor and cigarette smoke, etc.

Comfytest A measuring instrument used to predict thermal comfort in a space.

Commissioning The process of setting to work an HVAC system in order to meet the operating design requirements.

Compensating control The process of automatically adjusting the control point of a controller to compensate for changes in a second measured variable.

Component Any functional element of an HVAC system. See also Air diffusion and Air distribution.

Components of ventilating or air conditioning A single functional element forming a part of a ventilation or air conditioning system.

Compound hood An extraction hood that has two or more points of appreciable entry loss.

Compressed air Air at a pressure greater than the atmospheric pressure at that location. In the case of a fan, if the outlet pressure exceeds 30 kPa it is classified as a turbo compressor.

Computational fluid dynamics (CFD) The technique of using computers to provide an assessment of the flow of air and other fluids.

Concentration The amount of a substance present in a given volume of a gas or liquid, in parts per million (ppm) or $\mu g/m^3$. In the case of gases, the ppm is proportional to the molecular concentration; hence, the relationship between ppm and $\mu g/m^3$ depends on the molecular weight of the gas concerned.

Condensate The liquid formed from condensation of a vapor generally on a cool surface.

Condensation Formation of a liquid from a gas, when its temperature is lowered at constant pressure.

Conductive hearing loss Hearing loss that is caused by blockage or other interference in the path by which sound energy is transferred to the inner ear.

Conductive heat exchange Heat flow that takes place by thermal conduction between two surfaces in contact or along or across a solid body due to temperature difference, in W/m^2.

Confined space As defined by OSHA, any space that is large enough for an employee to enter and perform work that has limited or restricted means for entry or exit and is not designed for human occupancy.

Connector Any device that joins two or more components together.

Consensus standards Existing standards that are voluntarily being followed by industry, typically containing the minimum requirements for materials, procedures, and applications.

Constant dryness or constant quality lines Lines on a steam, gas, or psychrometric chart passing through all the "state points" of equal dryness fraction.

Constant volume lines Lines on a steam or psychrometric chart passing through the "state points" representing an equal volume of steam or air (dry or wet).

Containment The process of ensuring that all contaminants are contained in a room or cabinet.

Contaminant The same as pollutant, but usually used to describe indoor conditions.

Contamination Any unwanted material, such as radioactive material, present in a location. Also refers to loose radioactive materials that can be easily removed from surfaces.

Continuity relation The mass flow rate in unit time is the product of the density multiplied by the volume flow in unit time.

Continuous-flow mode The mode of air supply in which a regulated amount of air is supplied to the facepiece at all times.

Continuous sampling The uninterrupted sampling of air or pollutant from a space at a fixed rate.

Contrarotating fan See Fan.

Contraction The size reduction of a duct allowing it to fit into tight spaces.

Control A manual or automatic device allowing the regulation of pressure, temperature, and moisture content of volume flow of a system.

Control agent The medium in which the manipulated variable exists. In a steam process, the control agent is steam and the manipulated agent is the flow of steam.

Control device A manual, mechanical, pneumatic, or electrical device that controls any component, for example, fan and thermostat.

Control point The actual value of the controlled variable (set point plus or minus offset).

Controlled medium The medium in which the controlled variable exists. In controlling space temperature, the controlled variable is the space temperature and the controlled medium is the air in the space.

Controlled variable The quantity or condition that is measured and adjusted.

Controller Any device that senses changes in the controlled variable and provides the corrective output.

Convection The mechanism of heat transfer due to different temperatures, and hence different densities in fluids. It may be natural, dependent only on thermal forces, or forced, when use is made of a rotodynamic device to improve the rate of heat exchange.

Convective appliance A room-mounted device that transfers hot or cold air into the space mainly by convection.

Convective heat exchange The heat interchange by convection between the clothing surface or skin and the surrounding environment, C, in W/m^2.

Convective heat exchange coefficient A complicated factor involving the surface geometry, fluid velocity, and the various fluid properties. Its magnitude governs the rate of heat exchange.

Convective heat exchange, globe The convective heat exchange that takes place between the surface of a globe thermometer and the surrounding air, C_g, in W/m^2.

Convective heat exchange, respiratory The heat exchange that takes place by convection in the respiratory tract, C_{res}, in W/m^2.

Conveying system A system of ductwork used to convey powder, dust, or granular material from the point of generation to a collection chamber.

Cooker hood A device to collect cooking fumes from above a kitchen range and discharge them to the outside. It may incorporate a grease filter, fan, and fire damper or nonreturn flow damper.

Cooler See Chiller.

Cooling The removal of sensible or latent heat from a medium by means of a chiller.

Cooling coil A heat exchanger (battery) in which the chilling of a fluid occurs by means of a heat-transfer medium.

Cooling, direct A cooling system in which the medium to be cooled is not affected by another medium situated between it and the refrigerating apparatus.

Cooling effect Heat removed by a refrigerating appliance or heat exchanger.

Cooling, indirect A cooling system in which the medium to be cooled is affected by another medium situated between it and the refrigerating apparatus.

Cooling load The quantity of heat to be removed from a process or a space in order to meet the plant design or environmental conditions.

Cooling load, latent The quantity of heat to be extracted from a medium, without temperature change, in order to produce a given mass of fluid or condensate from the vapor.

Cooling load, sensible The quantity of heat to be removed from a fluid stream to maintain a desired fluid temperature.

Cooling tower A packed tower in which a warm liquid is allowed to fall by gravity, cooling it to within 1°C of the wet bulb temperature of the entering air.

Cooling water Water used as the cooling medium in refrigerating plant condensers. It may be from a well or river or recooled.

Core area of air terminal device That part of an air terminal device located within a convex closed surface of the minimum area required to include all of the air terminal device openings inside the surface.

Core area of sand trap louver The product of minimum height h and minimum width b of the front opening in the sand trap louver assembly with the louver blades removed.

Core temperature The deep core temperature of a living body resulting from metabolism.

Corona The luminous discharge that appears at the surfaces of a conductor in an electrostatic precipitator due to air ionization.

Corrected effective temperature An empirical comfort index that uses the dry bulb, wet bulb, and globe temperatures and the relative air velocity in a space.

Corrective action A control action that provides a change in the manipulated variable.

Corrosion The process of a material being destroyed by chemical, electrochemical, or microbiological action.

COSHH (Control of Substances Hazardous to Health) UK legislation regulating toxic dusts, vapors, and gases.
Cotton filter cloths Woven cotton cloth stretched on a frame to produce a filter medium.
Counting rate The number of counting events per unit time.
Cowl A roof-mounted device designed to provide airflow out of a building with the minimum of flow reversal.
Crackage Cracks in the building structure through which infiltration or exfiltration can occur by means of wind forces or temperature and pressure differences.
Criteria Set values used to establish guidelines for air or water quality standards.
Criterion level The 8-hour TWA limit for noise exposure, used for determining the noise dose.
Critical pressure The pressure at which the gas starts to liquefy at its critical temperature.
Critical temperature The temperature above which a given gas cannot be liquefied, regardless of the pressure.
Crocidolite See Asbestos.
Cross-drafts The unwanted or wanted movement of air within a space, which may be natural or mechanical. See Cross ventilation.
Cross-sectional area of a duct The area of a duct perpendicular to airflow, A_c. In the case of a circular duct, $A_c = \pi D^2/4 \approx 0.7854\ D^2$, and for a rectangular or square duct $A_c = a \cdot b$. For oval and other shapes of ductwork, the appropriate equations are used.
Cross ventilation Ventilation that takes place by the circulation of air introduced at one side of the room and extracted at the other side.

Cumulative Additive.
Cumulative effectCumulative errors Errors in a measuring process consistently in the same direction, either positive or negative.
Cumulative frequencies Accumulated sums of frequency values in a frequency distribution.
Cumulative sampling A system in which the sample is accumulated over the time, either by being taken continuously or for periods at regular intervals to a given single sample. Its composition is regarded as representative of the whole period of its accumulation.
Cunningham correction factor A factor used as a refinement to the Stokes equation for falling particles of small diameter. These tend to slip between the air molecules and, as a result, fall faster.
Cup anemometer A device used by meteorologists for the measurement of wind speed.
Curie (Ci) A unit of radioactivity, related to the emission from 1 g of radium, it is equal to 3.7×10^{10} disintegrations per gram per second. This unit has been replaced by the Becquerel (Bq); 1 Bq = 27.03 pCi.
Cuvette Small cylinder (test tube) used to hold a sample in a spectrophotometer.
Cycle The sequence of events in a heat engine, refrigerating machine, or any process where, during the performance of mechanical work, heat is supplied to and rejected from the working fluid, which is returned to its original condition.
Cycling A periodic change in the controlled variable from one value to another. If uncontrolled, this is known as hunting.
Cyclone A device for the removal of particulate matter from gas streams by centrifugal force and a velocity reduction. There are three main patterns of operation:

1. Descending spiral flow.
2. Ascending spiral flow.
3. Radial inward flow.

Cyclones may be subdivided into

- Dry dynamic precipitator,
- Gravity,
- High efficiency,
- Inertial separation, and
- Wet centrifugal.

See also Centrifugal collectors.

D

Daily noise dose (DND) The allowable noise exposure for an 8-hour workday.
Dalton's law of partial pressure States that the total pressure of a gas mixture is equal to the sum of the pressures which each component gas would exert if it occupied the same space alone.
Dampers Devices fitted in ductwork to provide a flow resistance to control the air supply. Dampers may be

- Butterfly: A plate that turns on a diametrical axis inside a duct, or a pair of flaps hinged to a common spindle to allow flow in one direction only.
- Plate (single blade): A hinged flap that, by virtue of its position relative to airflow, creates a flow variation. This simplest form of damper, only used on small duct sizes, does not provide accurate control.
- Horizontally opposed: A multileaf damper in which adjacent blades rotate in opposite directions.
- Iris: A circular damper with moving leaves that forms a variable orifice.
- Parallel-blade: A damper that allows a gas to flow, in which the blades rotate in the same direction.

Damper control fan See Fan control methods.
Damper section A section of HVAC equipment containing a damper or valve.
Data Information collected in a given test.
Data acquisition The identification and collection of information relating to the performance of a particular piece of equipment.
DC pressurization A test of the air-tightness of a building in which a fan is used to pressurize the building to a uniform static pressure. The pressure differential between indoors and outdoors is measured to determine the air-tightness of the building structure.
DCV See Demand controlled ventilation.
Dead band A range of the controlled variable in which no corrective action is taken.
Decay The spontaneous disintegration of an unstable atomic nucleus to form another more stable element or isotope of a lower atomic mass.
Decay method The time from the liberation of a given concentration of a tracer gas into a space before its concentration decreases to a set value in the air.
Decay rate The rate at which the concentration of an air pollutant decreases with time, due to absorption or precipitation.
Decipol A unit derived in an attempt to quantify odor concentration by the perception of odor.
Decontamination factor A logarithmic scale used to measure the collection efficiency of a particulate collection device.
Decontamination index The logarithm to the base 10 of the decontamination factor.
DDC Direct digital control. See Digital control.
Deflection of a duct The largest deformation of a duct subjected to an imposed load, given as the measured difference in distance between a plane through the points of support and a plane through the lowest point of the duct under a load.
Deflection of a joint The largest deformation of a joint subjected to a positive or negative pressure, given by the measured difference in distance from a reference plane outside the joint to the joint with and without pressure.

Degasser A packed tower through which a fluid to be degassed flows. Air is forced through the fluid stream, stripping the gas from the liquid.

Degree-days Temperature data recorded over a 24-hour period as deviation from a certain base temperature used to determine the operating costs of a heating or air conditioning system depending on the external climatic conditions.

Degree of enclosure The actual protection offered by an enclosure in the containment of a generated contaminant.

Dehumidification The removal of water vapor from a gas.

Dehumidification load The mass of water vapor to be removed from a space or a process in order to meet design conditions.

Demand-controlled ventilation (DCV) A ventilation system in which the room airflow rate is governed by an automatic control that depends on the level of a given pollutant within the space. A typical example is allowing CO_2 in a space to reach a certain level before the extract fans come into operation. However, in many industrial environments other pollutants control fan operation.

Demand mode The mode of air supply in which inhalation creates a negative pressure inside the facepiece, causing the regulator to release air into the facepiece. Respirators that operate in this mode are not recommended and have been largely replaced by respirators operated in the pressure-demand mode, in which the facepiece is maintained under a slight positive pressure at all times.

Demand ventilation A system that is capable of supplying varying amounts of fresh air in response to either manual or automatic control. See Demand controlled ventilation (DCV).

Density The measure of the amount of mass in a unit volume. The density of a gas is a function of its pressure and temperature. It can be determined by using the ideal gas laws.

Density factor A factor used to correct the density of standard air or other gases for altitude, temperature, or moisture content.

Deposit gauge A collection device that records the deposition rate of solid or liquid particulate matter from the air.

Deposition Relating to either the dry or wet deposition of a particulate.

- Dry: The interception and retention by surfaces of gases or particulate matter by diffusion, gravitational settling, or thermal forces.
- Wet: The process of gas or particulate interception by sprays.

Depressurization A fan test used to determine the rate of air leakage from a building by creating a negative static pressure.

Depth loading The deposition of particles mainly within the filter interstices, rather than on the filter surface.

Desiccator A sealed container containing a water-absorbing substance such as silica gel or calcium chloride used to dry test materials in the laboratory.

Design standards The appropriate industrial, national, or international standards covering

- Air,
- Air filters,
- Clean rooms,
- Comfort,
- Contamination levels,
- Fans,
- Health,
- Noise and vibration, and
- Pharmaceutical.

Desorption The removal of adsorbed materials from a solid sorbent by the use of a solvent or the application of heat.

Desulfurization The removal of sulfur from flue or other sulfur-containing gases.

Detector tube A direct method for identifying airborne contaminants, also known as length-of-stain tube. It is a convenient tool for detecting and quantifying contaminants in field or emergency situations.

Determinant A chemical metabolic product of the change in the body's chemistry caused by exposure to a pollutant. The level of determinant is measured in a biological sample collected from the exposed worker, and compared to the biological exposure index (BEI).

Determination The analytical measurement of a pollutant.

Detoxification The process of decomposition of toxic substances in the body to produce harmless substances, which are duly eliminated from the body.

Deviation Difference between a set point and the controlled variable in a control device at any instant.

Dew point The temperature at which a particular gas starts to condense on a cool surface.

Dew point, acid The temperature at which acid vapor in a gas stream condenses out of the flow onto a cold surface or in a cold gas stream.

Dew point depression The difference in temperature between the wet and dry bulb readings.

Differential pressure The difference in pressure between two locations in a fluid.

Diffuser A device of variable cross-sectional area used to spread airflow into a space.

Diffusion The mixing of substances by molecular motion to equalize a concentration gradient. Applicable to gases, fine aerosols and vapors. (See Brownian diffusion.)

Diffusion effect The capture of particles due to Brownian motion.

Diffusion of particles The transfer of small particles and gas molecules into the surrounding air due to concentration difference.

Diffusiophoresis See Stephan flow.

Digital control A control loop in which a microprocessor-based controller directly controls equipment by means of sensors. Its operation depends on a series of on-off pulses arranged to convey information.

Dilution The reduction in concentration of a solute by the addition of solvent.

Dilution equations Mathematical equations that allow the determination of the decay rate of a pollutant in a space due to mechanical or natural ventilation.

Dilution ventilation or general exhaust ventilation A mixed airflow designed to dilute the contaminants within a space to required safe concentration limits. The air is extracted from the space as a whole rather than from the zone of pollution generation.

Direct-fired heater A heat generator that allows the combustion products to be mixed with the air to be heated.

Direct flame incineration A fume control device in which organic pollutants in the waste gas stream are oxidized to form nonpolluting by-products.

Direct interception Particle removal from a gas stream by a filter with geometry such that the particulate matter does not deviate from the fluid flow lines.

Direct reading A sampling approach that provides immediate or very fast feedback such as a meter or colorimetric method.

Directivity The characteristic associated with sound energy in the form of waves moving in a straight line from the source.

Discharge or entry loss of a louver The reduction in airflow caused by a louver. The discharge loss coefficient is equal to the actual airflow rate divided by the theoretical airflow rate at a given pressure difference across the louver. If tested with the airflow in the reverse direction, the coefficient becomes the entry loss coefficient.

Discharge coefficient A dimensionless number describing the energy loss that occurs when a fluid is discharged from an orifice.

Discharge stack A stack that conveys combustion products or other pollutants from a space directly to outdoors. The pollutants may be removed before the remaining gas is allowed to discharge into the atmosphere.

Discharge system A system that discharges unwanted gaseous, solid, or liquid products.

Dispersion The manner in which a pollutant spreads from its point of generation, becoming diluted with distance from the source.

Dispersoid The particles involved in the act of dispersion.

Displacement air diffusion Air diffusion where the mixing of supply air and room air external to the air terminal device is at a minimum. See also Air diffusion and Air terminal devices.

Displacement flow, actual Actual flow pattern in an enclosure, resulting in uniform air distribution with virtually no mixing.

Displacement flow, ideal Ideal flow pattern in an enclosure, in which uniform air diffusion is provided without mixing.

Displacement ventilation Room ventilation created by room air displacement, by introducing air at a low level in a space at a lower temperature than the room air.

Disposal The action involved in the disposal of waste matter.

Distance to the v isovel (displacement air) The maximum horizontal distance L from the center of an air terminal device to the rectangle circumscribing the specific isovel. It is independent of the distance from the floor.

Distribution The act of conveying a medium from one point to another.

Distribution ducting The supply or extract air ductwork, which conveys air from the plant room to the conditioned space and vice versa.

DND See Daily noise dose.

Door and inspection panel Sealed openings in air-handling plant and ductwork providing access for cleaning or maintenance.

Door air leakage The leakage due to pressure differences through the crack around a door.

DOP (Dioctylphthalate) Generated particles of this chemical are used in filter efficiency tests.

Dose

1. The level or amount of exposure to a hazardous chemical or physical agent.
2. The level or amount of a chemical or ionizing radiation that has been absorbed, usually expressed as amount per weight of the exposed organism, for example, mg/kg.

Dose rate The amount of a pollutant taken or received by an individual per unit of time.

Dose-response relationship The toxicological concept that the toxicity of a substance depends not only on its toxic properties, but also on the amount of exposure or dose.

Downdraft A natural or mechanical downward airstream, either that may, due to its temperature and/or velocity, cause thermal discomfort. In the case of a stack discharge, the term downwash may be used for the downward air current in the lee of the chimney that takes the smoke and other emissions below the emission discharge level causing ground-level pollution.

Downdraft hood A hood positioned under a process that receives gases, vapors, or dusts from the source above.

Downstream Relating to a position after a filter has treated a gas, or some distance away from a measuring device.

Downwash See Downdraft.

Draft An airstream within an occupied zone that causes thermal discomfort of the occupants due to its temperature and/or velocity. Also, the thermal uplift caused by density differences required to provide adequate air both for the combustion process and for the removal of the products of combustion.

Draft proofing The process of filling in air gaps around a structure to reduce the rate of air interchange.

Draft risk (DR) The percentage of people dissatisfied by a particular combination of air movement and temperature.

Draft risk rating The percentage of people predicted to be dissatisfied due to draft.

Drag anemometer An instrument used for the measurement of wind velocity by measuring the drag forces.

Drag coefficient The coefficient relating to the influence of drag over a surface in either laminar or turbulent flow.

Drain plug or cock A removable plug or valve that allows water, condensate, or other liquids to be drained from a system.

Drift velocity The velocity of the air as it drifts from a high-pressure zone to a low-pressure zone in a building.

Drop of jet The vertical distance between the lowest horizontal plane tangent to a specified isovel and the center of the core of an air jet.

Droplet A very small particle of a liquid suspended in a gas stream.

Dry bulb temperature See under Temperature.

Dry heat loss The sensible heat loss from the body that takes place by raising the temperature of the air around it.

Dry scrubber An absorption system that uses a dry solvent directly injected into the gas stream.

Drying The process of fluid removal from a medium, either by heat or by vacuum.

Drying oven An oven or stove used for the drying of a product.

Drying time The time necessary under given ambient or artificial conditions to remove a required amount of water or solvent from a manufactured product.

Dryness fraction of wet steam The mass of pure saturated steam contained in unit mass of wet steam.

Dual-circuit heat exchanger Combined air heater and air cooler battery, with independent pipework or ductwork circuits for the heating and cooling media.

Dual duct unit An air terminal unit assembly consisting of two-ducted air inlets and a means of automatically adjusting the mixing ratio of the two air streams.

Du Bois area The total body surface area of a person,

$$A_{\mathrm{Du}} = 0.203\ W^{0.425}\ H^{0.725}$$

See Area, body surface.

Duct A conduit to distribute supply air or to extract air from a space, or a boxed run in which pipework or electrical cables are carried.

Duct board A rigid board of insulation material with one or both sides faced with a finishing material. The outer face is normally a vapor barrier and air barrier.

Duct branch Used to subdivide the flow from one or more ducts into two or more ducts, or, conversely, to unite the flow from two or more ducts into one duct (T pieces, Y pieces, cross pieces, etc.). It may or may not include diverting elements. Note: Rigid components of ductwork allow sound and vibration transmission; the fixing of flexible sleeves on branches reduces the magnitude of this transmission.

Duct connection component Items intended to facilitate the joining of two components of ductwork, including

- Collars,
- Flanges,
- Connectors,
- Cleats, and
- Slip joints.

Duct design The process of sizing ductwork to ensure the optimum performance in initial costs, running costs, and the distribution of air in a duct distribution system. The design techniques may be

- Constant pressure drop,
- Constant velocity, or
- Static regain.

Duct fitting Component of ductwork incorporating a change in one or several of the following:

- The length of the duct.
- The orientation of the duct.
- The shape of the straight length of the duct.
- The area of the cross-section of the duct.
- The direction of the duct, for example, bend, elbow, or tee.

Transformation affects a change of area and/or the form of the cross-section. If the transformation is continuous, then an area reduction is termed convergent and an area increase is termed divergent. If the transformation is abrupt, then an area reduction is termed an abrupt contraction and an area increase is termed an abrupt enlargement.

Duct sealing Measures taken to ensure that the air distribution system has an airtight seal.

Duct support The spacing of hangers or supports on a duct run to ensure that it is capable of self-support with any imposed load.

Duct transformation See Duct fitting.

Ducted fan See Fan.

Ductwork components Individual elements of ductwork, which are intended to be joined together at the time of installation. These components are of various types. See also Duct connection component and Duct fitting.

Duration limited exposure (DLE) The recommended maximum time of exposure, in h.

Dust The solid particulate matter formed by the breaking up of larger particulates by mechanical action. The particles range up to 75 μm in diameter; larger particles are classified as grit.

Dust cake The dust layer that builds up on a fabric filter, initially improving its collection efficiency.

Dust collection mechanisms The various means by which particulate matter is collected, which may be classified as

- Dynamic dry: The dust is collected under dry conditions.
- Dynamic wet: The dust stream is exposed to a liquid such as water to improve the collection efficiency.

Dust disposal The method for safely disposing the collected dust.

Dust explosion An explosion caused by the ignition of certain dusts allowed to exceed a given volume for volume concentration in air.

Dust fall The deposit rate of grits and dusts collected from the air in a measuring instrument.

Dust-holding capacity The weight of dust retained by a filter under specified test conditions.

Dust porosity The porosity of the dust cake, which has a direct influence on the filter pressure drop.

Dust suppression The preventive measures taken to eliminate or reduce the spread of dust generated by a process into surrounding areas.

Dwelling The portion of a building in which people live.

Dynamic pressure The pressure equivalent of a fluid velocity at a given point.

E

Ecology The interactions between living organisms and their environment.

Economic velocity The velocity at which gases or fluids are conveyed to ensure that running costs are kept at an economic minimum and that damage is not caused by erosion.

ED_{25} The dose of a toxic product which has an effect on 25% of the exposed population.

Eddy A current in a fluid that moves in a direction contrary to that of the main stream, often having a rotary motion.

Eddy diffusion The interchange of liquids, gases, or vapors that takes place in an eddy current.

Effective area, air terminal The net area determined aerodynamically from an A_k factor.

Effective drift velocity The velocity resulting from air flowing from one zone to another due to a pressure differential.

Effective length, duct The dimension that a straight duct contributes to the length of an air distribution installation.

Effective length, fitting The dimension that a duct fitting contributes to the length of an air distribution installation.

Effective mechanical power The energy spent in overcoming external mechanical forces on the body, in W, normally ignored for most activity.

Effective radiant heat flow The heat exchange by radiation between the walls of the enclosure and the human body, E_{eff}, in W/m^2.

Effective radiating area of a body The net effective radiating area of a body exposed to its surrounding.

Effective specific gravity The true specific gravity of an extract gas stream, as opposed to the specific gravity of the air alone.

Effective stack or Chimney height The sum of the stack height and the effective plume rise, determined for the buoyancy plume and its associated efflux velocity.

Effective temperature (ET) See Temperature, operative.

Efficiency The useful energy output of a device divided by the energy input into the system.

Efficiency average The efficiency of an item of equipment, such as a filter, boiler, or heating or cooling coil over its life cycle.

Efficiency, counting The proportion of particles in a volume or mass flow that are counted as they pass through the sensing element of an optical particle counter.

Efficiency, dust spot The capability of a filter to remove the staining portion of atmospheric dust from a gas under set test conditions, expressed as a percentage.

Efficiency, filter The ratio of the number of particles retained by a filter or other air cleaning device to the number of particles entering, expressed as a percentage.

Efficiency, fractional See Efficiency, particle size.

Efficiency, initial The efficiency determined prior to the first loading cycle in a filter test.

Efficiency integral See Efficiency, overall.

Efficiency, local The efficiency of a given point on a filter at set operating conditions.

Efficiency, minimum The minimum efficiency obtained during the performance classification of a filter.

Efficiency, overall The average efficiency of a filter or other item of equipment under set operating conditions.

Efficiency, particle size The ability of a collection device to remove particles of a specified size or size range.

Effluent Any unwanted material, such as water or exhaust gases, discharged into the environment.

Efflux velocity The discharge velocity of waste gases from the top of a stack.

Egg crate straighteners A lattice device inserted in a ductwork run to straighten the airflow vortex after a bend or other directional change.

Ejector A device used to provide a primary airstream into which the contaminated air is entrained for subsequent removal. Used

when corrosive products, high temperatures, fan blockage by particulate matter, or fire or explosion risk make a fan unsuitable.

Electric control An electrical device that controls some mechanical function, such as a damper control.

Electrical properties Relating to the resistance, electrical capacity, and insulating characteristics of a conductor or electrical device.

Electromagnetic interference Interference created by rotating electrical equipment causing problems in areas such as microelectronics clean rooms.

Electronic air cleaner A device used to clean particulate matter from a gas stream, consisting of a fan and an electrostatic precipitator.

Electronic control A control system operating on low voltage, making use of solid-state components to amplify input signals from which the control functions are performed.

Electronic filter A filter incorporating an electrostatic precipitator. A fibrous filter that has its collection efficiency electrostatically enhanced.

Electrostatic The properties of electrically charged bodies, and the resulting associated electrical phenomena that occur in the immediate vicinity of these materials.

Electrostatic charging The creation of a different electrostatic charge between particulate matter and droplets. An increased efficiency in contact between the dust particle and the droplets is achieved.

Electrostatic filter See under Air filter.

Electrostatic force A field in which stationary electrically charged particles are subjected to a force of attraction or repulsion, as the result of another stationary electric charge.

Electrostatic precipitator A filtering system for the removal of particles from an air stream by giving them an electrical charge. The charged particles are attracted to plates of opposite polarity onto which they adhere. The precipitators are classified as

- Dry,
- Wet,
- Rapping,
- Low voltage,
- Medium voltage, or
- High voltage.

Electrostatic shocks Electric shocks experienced by occupants due to a static discharge. Increasing the humidity and using non-static materials reduce the frequency of such events.

Element of distribution See Air distribution.

Eliminator plate A plate that mechanically separates droplets of moisture from a gas passing through it.

Elutriation The separation of particles in a fluid by gravity, which allows those with the greater falling speed settle as the fluid flows through an elutriator.

Emission The undesirable liberation of a dust, gas, or vapor from a process, either indoors or outdoors.

Emission factor A value representing the average amount of a pollutant that is emitted from a particular source in relation to the amount of product.

Emission generation The volume or mass of a pollutant liberated in unit time.

Emission limit The maximum design or statutory values of the given emission of a given pollutant in a work area.

Emission rate The rate of pollutant discharge into the surrounding atmosphere.

Emission standard The allowable quantity of a pollutant that can be discharged from a particular process. It may be expressed as

- Mass discharge over a given time period, kg/h,
- Mass of pollutant per mass of processed material, g/kg,
- Parts of the pollutant in a unit volume of air, ppm, or
- Mass of pollutant per unit volume of the gas in which it is discharged, mg/m^3.

Emissivity (ε) The ability of a surface to emit radiant heat transfer.

Emphysema, pulmonary The swelling and breaking down of the air sacs in the lungs. This reduces the area available for oxygen and carbon dioxide exchange within the lungs.

Enclosure A box, cupboard, or room in which a toxic process is carried out in safety.

Enclosing hood An extract hood that partially or completely encloses the point of pollution generation.

End-of-service-life indicator (ESLI) A warning system that alerts a respirator user that the cartridge or canister is approaching the end of its usefulness.

Energy The capacity of a body for doing work. Mechanical energy may be either potential (by virtue of the body's position) or kinetic (by virtue of its motion).

Energy balance The arithmetic relationship between the energy input and output of a system.

Energy conservation Measures taken to reduce the use of energy.

Engineered control The removal or reduction of a hazard through implementation of an engineered solution, such as material substitution, process change, or installation of an exhaust ventilation system.

Enthalpy The total heat content of a gas.

Entrainment velocity The velocity in a jet stream that effectively entrains the dust or gas particles that surround it.

Entropy A function of the state of a substance related to order or disorder. The entropy increases as the substance receives heat.

Entry The breaking of the plane by any part of the body at the opening of a space that requires an entry permit.

Entry loss The loss of energy in the moving stream that takes place as a fluid enters an opening.

Environment Indoor or outdoor conditions, including pollutants, thermal conditions, moisture, noise, and light.

Environment impact statement (EIS) A document used in the United States that details the influence of proposed Federal legislation on the environment.

Environmental temperature A design value used by the CIBSE in heating and cooling calculations, equal to $0.33\,\theta_a + \theta_r$.

Epidemiology The study of the development and cause of diseases.

Epidermis The outermost skin of the body, through which some dangerous chemicals can be absorbed into the body.

Epigenetic carcinogen A substance that causes cancer through a mechanism other than interaction with the genetic material.

Equilibrium dust content The amount of dust held on a clean filter cloth, which after a period of time remains approximately constant.

Equivalent diameter of a duct The diameter d_e of a duct that will cause the same pressure drop at the same friction factor at equal flow as a given straight rectangular duct.

Equivalent diameter of a particle The diameter of a standard-density sphere whose motion would be similar to a given real particle, which may not be spherical.

Equivalent evaporation The evaporation that takes place in a boiler at or above 100°C.

Equivalent exposure The exposure to a harmful product experienced by a worker. It is the sum of the exposure fraction of each component in a mixture.

Equivalent leakage area (ELA) The specified design pressure difference that allows a given air quantity to pass through a set orifice area.

Equivalent pressure The pressure corresponding to standard density,

Equivalent temperature A synthetic comfort scale that takes into account the effects of dry bulb temperature, air movement, and mean radiant temperature.

Equivalent warmth An early UK thermal comfort scale devised by Bedford.

Ergonomics The science dealing with the application of information on physical and psychological characteristics to workplace design.

Erosion control The protection of the containing material, which stops the fluid flow, from gradual removal by particulates or gas bubbles.

Error An individual or cumulative mistake made in any experiment or test.

Eupatheoscope An early device used in the United Kingdom for the assessment of thermal comfort.

Evacuated container A sealed vacuum sampling container, which is opened for the collection of a given material.

Evaporation The process of conversion of a liquid to a vapor, without necessarily reaching the boiling point.

Evaporative cooling The cooling of a body by virtue of latent heat removal to the surrounding environment.

Evase An increase in the size of a ductwork section.

Excess air Air supplied to a combustion process over and above the air theoretically required for efficient combustion.

Exchange rate The doubling of sound energy for each increase of 3 dB.

Excursion limit A time-weighted average of pollutant exposure over a length of time specified by OSHA that cannot be exceeded during the working day. See Peak limit.

Exfiltration The uncontrolled leakage of air from a building, either by natural or by mechanical means.

Exhausting The process of removing gases or vapors from a space or process and discharging them safely to outdoors.

Exhaust air See Air, exhaust.

Exhaust air (EHA) classification CEN/TC 156 classifies exhaust air (EHA) into four categories related to the extract air (ETA) classifications.

EHA classification

Pollution level	Description
EHA 1: Low pollution	ETA 1 or ETA 2 after cleaning
EHA 2: Moderate pollution	ETA 2 or ETA 3 after cleaning
EHA 3: High pollution	ETA 3 or ETA 4 after cleaning
EHA 4: Very high pollution	ETA 4

In industrial ventilation, the boundaries selected must be clearly stated for each application. See Extract air (ETA) classification.

Exhaust rate The controlled quantity of air, gases, vapors, and particulate matter that is removed from a space or process.

Exhaust ventilation The removal of polluted air from either a point source or a number of positions in a space direct to outdoors.

Expansion joint A flexible joint in a run of pipework or ductwork that allows expansion or contraction.

Experimental variance Permission granted by OSHA for the use of an alternative method of worker protection during an approved experiment to demonstrate or validate new safety and health techniques. The variance terminates upon study completion unless another type of variance is issued by OSHA.

Expired air temperature The air temperature of the breathe on leaving the nose.

Explosion, dust See Dust explosion.

Explosive limits The maximum and minimum concentrations of a mixture of gas, dust, or vapor in air or another gas which will explode if ignited.

Explosiveness A measure of the liklihood of a material to explode. For example, aerosol particles provide a very large surface area, accelerating the oxidation reaction, resulting in a high explosion risk.

Exposed area The particle capture area of a filter medium free from obstructions, through which a gas flows.

Exposure The period of time an organism has been in contact with a certain concentration of a pollutant.

Exposure by inhalation The toxic exposure of the body due to breathing contaminated air.

Exposure limits Guidelines for worker exposure to physical agents and hazardous chemicals, usually expressed as an allowable time of exposure or an air concentration below which health hazards are unlikely to occur among most exposed workers.

Exterior hood A hood that is located close to but does not enclose the point of pollution generation.

External fan pressure difference The difference between the total gauge pressure at the outlet of an air handling unit and the total gauge pressure at the inlet.

External fan pressurization See DC pressurization.

External work Energy used in overcoming external mechanical forces on the body, or the fraction of metabolic energy related to mechanical efficiency.

ETA classification

Pollution level	Description	Examples
ETA 1: Low pollution	Air of the same quality as outdoors, with respect to humidity. From rooms with pollutant sources from humans and building materials only	Offices, storage rooms, public service places with no pollution sources, including smoking
ETA 2: Moderate pollution	Air from occupied spaces that have impurities in addition to ETA 1	Rooms with smoking, eating areas, etc.
ETA 3: High pollution	Spaces in which moisture, chemical processes, etc., substantially lower air quantity	Toilets, kitchens, garages, tunnels, car parks, solvent areas, laboratories, etc.
ETA 4: Very high pollution	Air containing impurities and odors detrimental to health in concentrations higher than regulations permit	Industrial process areas, laboratories, etc.

Externally mounted air terminal device A unit such as a louver that prevents the ingress of rain, snow, birds, etc., into the ductwork.

Extinguishing system A system designed to extinguish fires by means of certain chemicals, gases, or water, either manual or automatic.

Extract air (ETA) classification Treated or untreated air that is removed from a space and discharged to outdoors. CEN/TC 156 classifies extract air into four categories. (See top of page.)

See also Exhaust air (EHA) classification.

Extract duct Any duct through which air or another gas is removed from a space and expelled to outdoors.

Extract temperature differential The increase or decrease in temperature between the supply air and the extract air.

Extract terminal device The grille or other device that is positioned in the main or branch duct that extracts air from a space.

Extract ventilation The mechanical ventilation arrangement to extract polluted air away from a space, either directly or by means of ductwork.

Extractor Any fan used for the extraction of air from a space.

F

Fabric arrester The collection of particulate matter by means of a suitable fabric material.

Fabric collector A filter manufactured from various fabrics expanded on a frame or formed into a bag or sock.

Fabric filter See Fabric collector.

Face loading The weight of dust collected by a filter divided by the effective filter medium area.

Face velocity The average velocity across an opening or item of equipment, such as a hood, fume cupboard, heating or cooling coil, or filter.

Fans

Fan A rotodynamic bladed device that conveys air at a given pressure and quantity in ductwork or a space by means of mechanical energy supplied.

Fan, abrasion-resistant A fan designed to minimize abrasion of the parts by the use of suitable materials.

Fan alignment The positioning of the fan shaft and its associated pulley in a line with the driving motor shaft and pulley.

Fan-assisted balanced ventilation Supply or extract ventilation within a space designed to provide the correct ratio of supply to extract air by means of one or more fans.

Fan control method	Description
Variable speed control	Speed change may be achieved by belts, pulleys, gear box sliding couplings, or a variable-speed motor
Damper control	A damper positioned in the fan inlet or outlet will either increase or decrease the flow resistance and result in a variation in flow rate. This may be achieved either manually or automatically
Vane control	Fan inlet vanes alter fan performance by controlling the swirl
Variable blade pitch control	The impeller blade angle is varied when the fan is in operation, normally only for axial flow fans
Adjustable pitch	The blade angle can be altered
Fixed pitch	The blade angle cannot be altered from the manufacturer's setting

Fan-assisted exhaust ventilation Extract ventilation of a space by means of a fan, with the supply air provided by induced leakage into the building.

Fan-assisted induction terminal unit CEN/TC 156 defines these as constant flow or variable flow.

- Constant flow: An assembly within which the primary airflow rate is modulated and mixed with air induced from the surroundings by means of a fan (also known as *series type*).
- Variable flow: An assembly within which the primary airflow rate is modulated and mixed with air induced from the surroundings by means of a noncontinuous running fan that provides a variable flow in response to thermal loads (also known as *parallel type*).

Fan-assisted supply The ventilation of a space achieved by means of powered air movement components in the supply air.

Fan, conveying A fan used for the conveying of particulate matter entrained in the air stream.

Fan, corrosion-resistant A fan constructed from materials designed to withstand the corrosive properties of the gases being carried.

Fan curve A curve relating the total pressure and airflow rate for a fan.

Fan, dust A purpose-designed fan that is capable of extracting a dust-laden gas.

Fan energy The energy required by a fan in order for it to provide a given airflow rate against a set resistance.

Fan, flameproof A fan with a flameproof motor and bearings.

Fan, gas-tight A fan with a casing that will provide a set air leakage rate at a given operating pressure.

Fan, general-purpose A fan used to handle air that will not affect its working life, that is, with no special requirements for temperature, moisture, or corrosive, abrasive, or flammable properties.

Fan, hot gas A fan capable of handling hot gases of a given temperature for a set operating time, normally designed to be installed with its motor outside the gas stream.

Fan, impeller tip diameter The maximum diameter measured over the tips of the fan blades.

Fan inlet The opening through which the air enters the fan casing, either rectangular or circular.

Fan installation

Fan laws The equations that describe the relationship between fan flow rate, pressure, density, power, size, rotation speed, and noise levels.

Fan motor systems The methods by which an electric motor drives a fan or pump system.

Fan, nonclogging A fan with an impeller designed to reduce the clogging of the material being handled.

Fan outlet The opening through which air leaves the fan, either rectangular or circular.

Fan pressure

- Static The fan total pressure minus the dynamic pressure corresponding to the mean air velocity at the fan outlet. The fan static pressure is the bursting or collapsing pressure on the enclosure.
- Total The algebraic difference between the mean total pressure at the fan outlet and the mean total pressure at the fan inlet.
- Velocity (or dynamic) Pressure associated with the kinetic energy in the air stream in the fan exerted in the direction of flow.

Fan section of air handling unit A unit in which one or more fans are housed.

Fan, spark-resistant A fan designed to reduce the risk of spark generation from stationary or moving parts.

Fan, special-purpose Any fan selected to overcome the shortcomings of a general-purpose fan.

Fan types Classifications of fans based on specific properties.

Fan tables Data provided by the manufacturer that describes the relationship between the volumetric output of a fan, energy requirements, and noise level for a given fan operating at different static pressures.

Fanger's comfort equations The various equations devised by Professor Fanger relating to activity, clothing, vapor pressure, mean radiant temperature, air temperature, and air velocity.

Federal standards Standards laid down by federal governments covering certain control aspects.

Female connection A circular sleeve used to join two duct or pipe components together. The male ends of the two components are inserted into the female connection.

Fiber counting A microscopic technique which is of particular relevance to asbestos, where the fibers are counted on a filter paper.

Fiberglass A filler having a glass fiber medium.

Fibrous filter Any filter consisting of a mass of fibers as opposed to a mesh.

Type	Description
Centrifugal	A fan in which air enters the impeller with an axial direction substantially parallel to the radial plane
	The impeller is defined as backward curved, inclined, radial, or forward curved depending on whether the outward direction of the blade at the periphery is backward, inclined, radial, or forward relative to the direction of the rotation
Axial flow	A fan in which the air enters and leaves the impeller axial to the fan
Contra-rotating	An axial flow fan that has two impellers arranged in series and rotating in opposite directions
Reversible axial flow	An axial flow fan, specially designed to rotate in either direction
Propeller	A fan with an impeller with a small number of broad blades of uniform material and thickness designed to operate in an orifice
Plate-mounted	An axial flow fan mounted in an orifice or spigot
Bifurcated	A fan in which the direct drive motor is separated from the airstream, reducing corrosion rate, allowing higher operating temperatures, and reducing wear and tear on bearings

Fick's law States that the molecular diffusion of water vapor in a gas without appreciable displacement of the gas is analogous to the conduction of heat and is governed by a similar type of law.

FID See Flame ionization detector.

Field blanks Sample media that are exposed to the same conditions as the media use for the actual sampling but are not connected to a sampling pump. See also Laboratory blanks.

Film badge A personal dosimeter containing a photographic film that is darkened by ionizing radiation, used to evaluate the degree of ionizing radiation exposure in comparison to a control film.

Filters

Filter Any medium used for the separation of solid, gaseous, or liquid contaminants from a gas or fluid stream. The collection efficiency depends on the materials used. Some types of filters are listed here.

- Activated carbon: A canister filter containing a porous form of pure carbon, which is capable of adsorbing gases and removing odors.
- Automatic roll: A roll filter that constantly or intermittently provides a clean portion of filter in the air stream by means of a pressure switch activating an electric motor, which winds the filter from a clean spool to a dirty spool.
- Bag; An extended surface filter in the form of a pocket or bag. A typical example of a bag filter is a mineral fiber bag made up of three layers, in which the first layer acts as a prefilter, the second is for fine filtration, and the third prevents fiber migration from the material used.
- Brush: An air filter constructed from a medium of intermeshing brushes.
- Cartridge: A replaceable in-line filter.
- Cellular: Replacement filter elements, which may be installed in a multiple, bark, or wall structure.
- Cleanable: A filter that, having collected a given amount of particulate matter, can be removed from the filter frame, cleaned, and reused.
- Coarse: A filter positioned before a fine filter to remove the larger particulates in order to extend the operating time of the fine filter.
- Cylindrical: A filter contained in a cylindrical form.
- Disposable: The opposite of a cleanable filter, which after collecting a certain dust burden is thrown away.
- Dry cell panel: A dry filter mounted in a rigid frame. In the past, these were manufactured from woven fabrics and felts; however, synthetic fibers are replacing these. They have fiber diameters of 20 μm with an average spacing of 300 μm and allow air velocity in the 2 m/s range.
- Electret: A type of filter that does not require a power supply and depends on the use of a filter medium with a permanent charge. Best performance is achieved with dry air.
- Electronic: A fibrous filter that is electrostatically enhanced.
- Electrostatic: The filter in an electrostatic precipitator.
- Fabric: A filter made of either woven or felted textile.
- Final: The last filter in a system of a multiple array of filters.
- Fine: A filter made up of fibers about 1 μm in diameter, with spacings of about 10 μm, and air velocity in the 0.02–0.1 m/s range.
- HEPA: A high-efficiency particulate air filter designed to deal with particles below 1 μm with efficiencies of 99.95 or better. Air velocities are 0.03 m/s or less.
- Insertable: A freely removable filter fitted in a frame.
- Louver: A filter pleated in a louver form to increase the face surface area.
- Membrane: A filter that incorporates a membrane as the collection medium.
- Metal: A filter constructed from metal mesh, fibers, or sintered porous metal.
- Panel: A shallow parallel-faced filter element or cell.
- Passive electrostatic: A mechanical filter in which the medium is electrostatically charged without the aid of a continuous external power supply.
- Pocket: An extended-surface filter in which the medium is formed into pockets or bags through which the dust-laden gas flows. They may be supported by the air pressure or self-supporting.
- Primary: A filter that removes airborne particles 5 μm and larger, normally supplied in panel types.
- Roll: A roll of filter medium on a drum that advances clean new material as the filter becomes clogged that may be manually or automatically driven.
- Second-stage: Filters for particulate matter from 0.5 to 5 μm. They have an extended face area in order to reduce the through velocity.
- Self-cleaning: An air-cleaning device that has the ability to be mechanically or chemically cleaned.
- Sorption: A filter that removes gaseous or vapor contaminants from a gas stream by an adsorptive or absorptive process.
- Ultra-low-penetration air filter (ULPA): A filter that has a penetration of less than 0.0005%, measured under CEN test conditions.
- Viscous: Filters constructed of a metal or synthetic mesh wetted with oil to retain dust particles.

Filter, air See Air filter.

Filter class A certain range of filter performance characteristics as specified by international standards.

Filter element The supporting housing of a filter medium.

Filter insert The replacement part of a filter medium that is inserted in the filter housing.

Filter medium A material used for filtering particulate matter from gases or liquids.

Filter medium face velocity The volume flow rate divided by the effective area of the filter element.

Filter pack A filter medium uniformly folded and interleaved with spacers.

Filter section of an AHU The section of an air-handling unit that contains a filter.

Filtration The process of removing particulate or gaseous matter from a fluid stream.

Final control element Any device that changes the value of a manipulated variable, such as a damper.

Fire damper See Fire and smoke damper.

Fire point The temperature at which a liquid gives off sufficient flammable vapor to produce sustained combustion.

Fire and smoke damper A device used to isolate one area of a building from fire or smoke in another, or one part of a duct run from fire or smoke in another. The device is mechanically or electrically operated in the case of fire.

Fit test A method for evaluating how well a respirator seals against the wearer's face. OSHA requires that a respirator be found to have a satisfactory fit before it can be used.

Fixed air terminal device A component that has fixed (nonadjustable) parts.

Fixed contamination A term used in the context of radiation for nonremovable contamination.

Fixed-direction grill A grill in which the discharge direction of air velocity is nonadjustable.

Fixed nondirectional grill A grill in which the discharge direction may be fixed to ensure the correct discharge pattern.

Fixing accessory of an ATD Any component that provides easy fitting and removal of air terminal devices.

Flame ionization detector (FID) A nonspecific air-sampling instrument used to identify the amount of a substance by measuring the absorption of electrons resulting from its ionization as it passes through a hydrogen flame. It is generally used for detecting organic compounds, specifically hydrocarbons.

Flame-retardant Any item in an HVAC system that will slow the passage of flames, should a fire occur.

Flammable atmosphere Any atmosphere that represents a fire or explosive hazard by virtue of gases, vapors, or dusts contained in it.

Flammable limits The minimum and maximum concentrations of a gas or vapor in air which can be ignited and sustain a self-propagating flame.

Flange A connection device that allows two sections of duct or pipe to be bolted together.

Flare A device used to burn rich mixtures of combustible waste gases containing pollutants.

Flash chamber A chamber provided to allow the burning of a flammable gas in a process. In a refrigeration system, it is the separating tank between the expansion valve and the evaporator.

Flash point The lowest temperature at which a heated liquid fuel will ignite.

Flexible duct A nonrigid duct that can be bent, expanded, or compressed within set limits without fracturing its cloth, metallic, or plastic covering.

Floor temperature dissatisfaction risk The degree of dissatisfaction experienced by occupants in a space due to the floor surface temperature.

Flow The movement of a vapor, fluid, sludge, or gas in a conduit. The flow may be forced or due to gravity.

Flow coefficient The constant K used in a typical flow equation, $V = K\,(\Delta P)^n$.

Flow equalizer A component used in a conduit to reduce turbulence or eddies in the flow.

Flow exponent The exponent n of the pressure difference in the flow equation. Its value ranges from 0.5 for turbulent flow to 1.0 for laminar flow.

Flow meter A device used to measure gravimetric or volumetric flow in a conduit.

Flow rate controller An item of equipment that will control the flow rate at a fixed value for a given pressure difference.

Method of control	Description
Mechanical, constant flow rate	Self-actuating and deriving its energy from the airstream to maintain the constant flow rate function
Mechanical, variable flow rate	Self-actuating and deriving its energy from the airstream to maintain the constant flow rate function and having facilities for resetting the required value depending on an external system
Pneumatic or electric	Deriving the energy for maintaining the constant flow rate function from an external source. It can be either of the constant or variable type
System-powered	Deriving its energy from the dynamic pressure in the airstream to maintain its constant flow rate function and can be either a constant or variable type

Flow rate pressure characteristic The relationship between the flow rate and a given pressure differential.

Flow reversal The backflow that occurs when a fluid changes its flow direction due to an imposed pressure gradient.

Flue A tube through which the gases generated during the combustion process are discharged to the external environment.

Flue gas The mixture of gases produced during the combustion process.

Fluidized bed A bed of solid particles floating on air or any other gas, on which combustible matter is burnt.

Fluorescence spectroscopy Analysis in which the intensity and wavelength of the energy that are emitted from excited atoms is used to indicate the presence of certain compounds.

Fly ash Fine ash particulate matter found in flue gases.

Foam scrubber A cleaning device that uses foam as a collecting medium for particulate matter in a gas stream.

Fog A naturally occurring aerosol of water vapor containing water droplets less than 100 μm in diameter, typically 15–35 μm.

Forced draft The forcing of air by means of a fan into a closed chamber for combustion or other purposes. The pressurization of the chamber forces the air and combustion products up a stack.

Form view factor A factor that describes the effects of the relative area of two surfaces, the geometry of the surfaces in relation to each other, and the two emissivities on radiation heat exchange between the surfaces.

Foul air Air that is unsuitable for respiration.

Fractional efficiency The efficiency of a device expressed for different fractions, for example, the efficiency of a filter for particles of different sizes.

Free area velocity The velocity in a device where the flow is not influenced by changes in section.

Free delivery The actual volume flow from a fan outlet with no imposed system pressure.

Free-falling diameter Also known as sedimentation or Stokes diameter, the diameter of a sphere with the same terminal settling velocity and density as a nonspherical or irregular particle.

Freezing point The temperature at which a liquid solidifies. The same as melting point.

Freons The trade name for the series of chlorofluorocarbons (CFCs).

Fresh air Air taken directly from the external atmosphere, on the assumption that its purity is superior to that within the space to which it is being supplied.

Freshness Relating to the sensation of air entering the nose, creating a feeling of freshness rather than stuffiness.

Free area of an ATD The area available in an air terminal device for the discharge of air, as opposed to the actual area.

Free area ratio The ratio of an actual opening to the obstructed portion of that opening.

Friction The property possessed by two bodies in contact which prevents or reduces the motion of one body relative to the other.

Friction factor Describes the relationship between the wall roughness, Reynolds number, and pressure drop per unit length of duct or pipe run.

Friction loss The pressure energy loss that takes place in duct or pipe flow. It is related to the Reynolds number, boundary layer growth, and the velocity distribution.

Frictional resistance The resistance to fluid flow resulting from the friction between the fluid and the surrounding solid surface.

Frit A porous structure that breaks an airstream entering a solution into small bubbles, maximizing the surface area of air in contact with the solution and increasing the amount of contaminant dissolved in the airstream.

Full-face respirator A respiratory protective device that covers the entire face from hairline to under the chin.

Fully adjustable air diffuser An air diffuser that has the provision of adjusting the discharge flow direction through a wide angle.

Fuel A substance suitable for the rapid and economic supply of heat by combustion.

Fume cupboard Cupboards of various efficiency classifications in which dangerous gases, dusts, and vapors are contained.

Fumes Small solid particulate matter normally spherical in shape and ranging in size from 0.001 to 1 μm.

Fumigation The result of a pollutant being trapped under or in an inversion layer, or the process of using poisonous gases to kill insects.

Functional check A check on the performance of an item of operating equipment.

Functional measurement Relating the check on the performance of an item of equipment to the design specification.

Fungus A simple organism that contains no chlorophyll, which may consist of one cell or of many cellular filaments called hyphae. If allowed to grow in HVAC equipment, it may cause allergic reactions.

G

Galvanized steel A zinc-coated steel sheet or plate with good corrosion resistance properties used for ductwork and other applications.

Gamma ray The shortest wavelength and highest energy type of all electromagnetic radiation. It originates in the nucleus of radioactive isotopes along with alpha particle, beta particle, or neutron emissions.

Garment insulation The degree of resistance to heat flow to and from the human body that a particular clothing arrangement will provide.

Gas A state of matter in which a substance completely fills the region in which it is contained, no matter how small the amount. Or any fuel in a gaseous form for use in an atmospheric or forced-draft burner.

Gas absorption The process of absorption of gases that takes place in certain solids or liquids.

Gas adsorber A device for the removal of gaseous impurities from a gas or liquid phase, two methods are in use, physical adsorption and chemisorption.

Gas chromatograph (GC) An analytical instrument with an internal tube or column that contains a solid sorbent, which allows some components of an injected sample to pass more quickly than others, separating the substances in the sample.

Gas collectors A sampling bag used to collect a sample for analysis.

Gas constant The coefficient R used in the ideal gas law, 8.3143 J/mol/K.

Gas contaminants Any matter that contaminates a pure gas or air above a given concentration level.

Gas monitoring The use of measuring or recording instruments to determine the concentration of a given gas within a space.

Gas physics The study of the various laws relating to the behavior of air or other gases.

Gas sensors Electrical or chemical devices that record the presence or level of a certain gas.

Gas scrubber A device for the removal of particulate matter from a gas stream by scrubbing the gas with a liquid.

Gaseous ion diffusion A method of charging particles in an electrostatic precipitator.

Gasket A semirigid or flexible sealing material fitted in the connection between two surfaces.

Gauge (or gage) A measuring instrument used for the determination of pressure, flow, temperature, moisture, or the thickness of materials.

Gauge, altitude A pressure gauge that displays the force per unit area in terms of the height of a column of a named liquid required to exert that force.

Gauge, Bourdon See Bourdon gauge.

Gauge, compound A device that allows pressures below and above atmospheric pressure to be measured.

Gauge pressure The pressure of a system over and above atmospheric pressure.

Gauge pressure of a space The positive or negative pressure in a space with respect to its surroundings, due to wind or thermal forces or the relationship of supply air to extract air.

General-duty clause A clause in the OSH act that requires the employer to provide a workplace that is free from recognized hazards likely to cause death or serious physical harm.

General exhaust ventilation (GEV) See Dilution ventilation.

Geometrical standard deviation A measure of the range of particulate sizes present in a collection of particles.

Globe temperature The temperature of the surroundings (mean radiant temperature) as recorded by a black globe thermometer.

Glove box A sealed enclosure used for handling toxic products by means of long impervious gloves sealed to form part of the enclosure.

Grab sample A sample of air collected over a short time period in the workspace.

Grade efficiency See Fractional efficiency.

Gravimetric analysis The chemical analysis of materials by the separation of the constituents and their measurement by weight. This describes the gas mixture by giving the percentage by weight of each component gas. See also Volumetric analysis.

Gravitational settling The fallout of particulate matter from a gas stream due to the gravity forces being predominant over the flow velocity forces.

Gravimetric efficiency The efficiency of a dust collector to remove a given weight of particulate matter related to the total weight present in the air stream.

Gravity settling device A chamber in which a change in velocity and/or direction of a dust-laden airstream allows the dust to settle into a collection hopper.

Grease absorption efficiency The ratio by weight of the quantity of grease retained by a grease filter to a reference quantity.

Green zone A compartment of secondary containment used in the atomic energy industry.

Greenhouse effect The retention of heat by the earth and the atmosphere due to certain gases being transparent to incoming solar radiation but opaque to the longer-wave radiation back from the earth.

Grill An air terminal device with designed outlets for airflow and distribution.

Grit Particulate matter with a diameter greater than 75 μm.

Ground level concentration The pollution concentration at the ground level resulting from the emission of a pollutant from a stack or other extraction point.

Grill type	Description
Adjustable	Grill intended to vary the direction or directions of the air delivered to the treated space consisting of one or more series of adjustable parallel ribs
Fixed directional	Grill intended to diffuse the air in one or more fixed directions, consisting of one or more series of fixed parallel ribs
Fixed nondirectional	Grill not intended to change the direction of air, consisting of parallel lamina ribs, perforated metal grid, and wired grid

Grounding requirements All components that handle flammable gases and dusts are electrically grounded to provide a safe path for an electrostatic charge to leak away.

Guards A cover to provide occupant safety from any exposed moving parts or live electrical parts.

Guideline, air quality Any guideline that indicates the level and duration of a substance above which it is assumed that the effects produced will adversely influence animal or vegetable life.

H

Half-face respirator See Half-mask respirator.

Half-life The time required for the concentration of a pollutant to decay to half its original value.

Half-mask respirator A respiratory protective device that covers roughly half of the face, from under the chin to the bridge of the nose.

Halocarbon (HCFC) A class of refrigerants that contain fluorine, chlorine, carbon, and hydrogen.

Halogenated compound A compound that contains one or more of the elements chlorine, bromine, fluorine, or iodine as a part of its structure.

Haze The presence of particulate matter in the air, reducing visibility.

Header An element of a ductwork or pipework run which circuit branches are taken from.

Hearing threshold level (HTL) The lowest level at which a person can detect a sound or tone. Test frequencies used to establish HTLs include 500, 1000, 2000, 3000, 4000, and 6000 Hz.

Heat balance The thermal balance that occurs in a building when the heat gains equal the heat losses. Also known as balance point or break-even point.

Heat cramps Muscle cramps, usually of the legs and abdomen, caused by heavy sweating that results in an imbalance in the salts and minerals in the muscles.

Heat drop The difference between the heat contained in a working fluid at any two points in a cycle.

Heat engine A machine that enables mechanical work to be done from heat energy, usually by changes in the volume of a working fluid.

Heat exchanger A device that transfers heat from one fluid to another without allowing the fluids to come into contact with each other.

Heat exchanger, cross flow A heat exchanger in which the fluid flow direction in the shell is perpendicular to the direction of flow in the tubes.

Heat exchanger, counter flow, or counter current A heat exchanger in which the flow inlet of one fluid is adjacent to the outlet of the second fluid and vice versa; the fluids flow in opposite directions.

Heat exchanger, parallel flow A heat exchanger in which the fluids enter the same end of the heat exchanger and leave at the opposite end; the fluids flow in the same direction.

Heat exchanger, pipe A heat exchanger in which the transport medium changes between gaseous and liquid states.

Heat exchanger, plate A heat exchanger in which the fluids are separated by a thin plate as opposed to a tube.

Heat exchanger, unidirectional A heat exchanger that is used to provide a heat exchange in one direction only.

Heat exhaustion The physiological condition resulting from the body suffering heat stress and loss of body fluids.

Heat island Relating to an area where the average air temperature is higher than the surroundings.

Heat load The heat input necessary to ensure that a treated space provides the internal design temperature at a given external temperature.

Heat load, latent The heat input necessary to ensure that a treated space is maintained at a given moisture content. This process is assumed to take place at a constant temperature.

Heat load, sensible The heat input necessary to bring about a temperature change.

Heat loss, dry The heat exchange that takes place from the human body to the surroundings by convection, radiation, and conduction, but not by evaporation.

Heat output The useful heat output from a heat generator or a heat exchanger.

Heat pump A "reversed" heat engine or refrigerator that takes in heat from a body at low temperature and by the expenditure of mechanical work rejects heat to a body at a higher temperature.

Heat rash A rash that appears as small red spots on hot, moist skin. The spots are inflamed sweat glands.

Heat recovery The process of collecting waste heat from a gas or liquid and utilizing it for space heating or a process.

Heat recovery section of an AHU The part of an AHU in which a sensible or latent heat gain or loss takes place by means of a heat-transfer medium.

Heat removal luminaire A light fitting provided with an extract duct from which the heat generated within the fitting, and a portion of that generated in the space, is extracted either directly to outside or for recirculation.

Heat, specific The amount of heat required to raise the temperature of a unit mass of a substance 1K.

Heat storage, body The amount of heat that can be stored in a body due to its temperature, mass, and specific heat capacity.

Heat stress index An index devised by Belding and Hatch to determine the effect of extreme heat stress.

Heat stroke A serious acute condition caused by the elevation of the body temperature above the danger level. Symptoms can include redness of the face, reduced sweating, erratic behavior, confusion, dizziness, collapse, or unconsciousness.

Heat syncope Fainting that occurs in some people after standing for a long period of time.

Heat transfer coefficient A proportionality factor used in an equation for determining the rate of heat transfer.

Heating The process by which sensible heat is added to one medium from another.

Heating capacity The capacity of a heat emitter, heat generator, or heat exchanger.

Heating coil A heat exchange coil (battery) containing the primary heat transfer fluid positioned in a run of ductwork where it passes its heat to the secondary fluid.

Heating, direct Any heating system that does not have a heat-transfer medium, for example, an electric fire or a gas fire.

Heating, indirect A heating system that makes use of a heat-transfer medium to convey heat from the heat generator to a heat emitter.

Heating water Water in a heat exchanger used for space and process heating. It can be low, medium, or high temperature, from 30°C to 160°C.

Heavy metal A general term relating to a specific group of metals, which as suspended and deposited particulates can contaminate the environment.

HEG See Homogenous exposure group.

Height allowance A percentage added to heat loss calculations to compensate for the vertical temperature gradient.

Hemoglobin The protein in red blood cells that binds with and transports oxygen.

Henry's law States that the mass of a gas dissolved in a definite volume of liquid at constant temperature is proportional to the partial pressure of the gas.

HEPA (high-efficiency particulate air) filters Also known as absolute filters, the large collection filter surface area provides a high collection efficiency for particulate matter.

Hertz (Hz) The frequency with which sound pressure changes. One oscillation per second is equal to 1 Hz.

High potential hazard The health, fire, or explosion risk resulting from the presence of certain materials in excess of a certain limit.

High-volume sampler A device used for extracting particulates from the air for analysis that requires a shorter sampling period than a low-volume sampler.

Histogram A diagram of the frequency of occurrences of values of a variable, grouped according to value in a number of separate ranges.

Hit and miss damper or valve A damper or valve consisting of two or more slotted slides operating in parallel.

Homogeneous exposure group A population or group of workers with similar exposure.

Hood A device that is located over a working area to collect any emissions generated at the source.

Hood, capturing A hood that has a sufficient flow rate to ensure that most contaminants are drawn into the hood.

Hood entry loss The pressure drop (energy loss) that occurs due to turbulence at the entrance to the extraction system.

Hood, receiving A hood intended to receive generated contaminants at some distance from the source.

Hood static pressure The negative static pressure available at a hood.

Hopper A collection bin for the supply or disposal of powdered or granular material.

Hot air column A rising column of low-density air inside or outside a building.

Hot film anemometer An instrument for the measurement of fluid velocity similar to the hot wire anemometer, but more robust as it consists of a thin quartz rod covered with a film of platinum rather than a wire.

Hot grid anemometer An instrument for the measurement of fluid velocity similar to the hot wire anemometer, but with the heating and sensing elements are separated.

Hot wire anemometer An instrument for the measurement of fluid velocity by measuring the resistance of a fine platinum or nichrome wire, which may or may not be shielded by a silica tube. The wire resistance is proportional to the temperature and the fluid flow rate.

Hot wire microphone anemometer An instrument for the measurement of fluid flow.

Housing A device or enclosure that contains any item of HVAC equipment.

HTL See Hearing threshold level.

Human factors A term sometimes used synonymously with ergonomics, it may also refer to psychological and sociological aspects of ergonomic issues.

Humid air Air that is high in moisture content at a given temperature.

Humidification The process of adding moisture to air by spinning disk, ultrasonic, steam, direct water injection, or other methods.

Humidification efficiency The ratio of the actual mass of water evaporated by a humidifier to the theoretical mass of water needed to achieve saturation at a given temperature.

Humidification load The mass of water that must be added to the airsteam in order to ensure that the design conditions are met.

Humidifier fever An illness caused by the growth of microorganisms in air cooling coils. These microorganisms or their generated toxins may be carried in the airstream to the conditioned space, causing an allergic response in susceptible people.

Humidifier section of an AHU The part of an AHU in which water vapor is added to the air.

Humidistat A measuring and control device used to control the humidity of a space.

Humidity The water vapor content present in atmospheric air.

Humidity, absolute The actual mass of water vapor present in a unit mass of air.

Humidity ratio The ratio of the mass of water vapor present in air to the mass of dry air.

Humidity ratio, saturation The humidity ratio of a gas at saturation.

Humidity ratio, expired air The mass ratio of water vapor to dry air in expired air.

Humidity, relative The ratio of the mole fraction of water vapor in moist air to the mole fraction of water vapor in saturated air at the same temperature and pressure.

Humidity, specific The mass of water vapor per unit mass of dry air.

HVAC The heating, ventilation, and air conditioning system that circulates and delivers filtered, humidified, or dehumidified, and cooled or warmed air to the interior of a building.

Hybrid (mixed mode) ventilation A system that makes use of a mix of natural and mechanical ventilation. May be further subdivided into seasonal hybrid, for example, natural ventilation in summer and mechanical ventilation in winter or spatial hybrid, for example, mechanical ventilation in core areas and natural ventilation at the perimeter.

Hydraulic efficiency The ratio of the actual head to the ideal head.

Hydrocarbons Chemical compounds containing only hydrogen and carbon.

Hydrogen sulfide A highly toxic gas with the characteristic smell of rotten eggs.

Hydrograph An instrument that measures and records relative humidity.

Hydrophobic Water-resistant or having a lack of affinity for water, usually said of a substance or material that does not absorb moisture.

Hydrostatıc pressure The pressure at a point of a fluid at rest, due to the weight of the fluid above.

Hygrometer An instrument used for measuring the moisture content of the air.

Hygroscopic The ability of a material to absorb moisture.

Hyperbolic expansion The expansion of a fluid according to the law $pV = c$.

Hypothalamus The temperature control center at the base of the brain, which regulates body temperature.

Hypothermia The physiological state resulting when the deep core body temperature drops below 35°C. It results in vasoconstriction and shivering in an attempt to conserve body heat.

Hypoxia A condition characterized by a deficiency of oxygen reaching the tissue.

Hz See Hertz.

I

IAQ See Indoor air quality.

Ideal gas A gas that obeys the ideal gas law.

Ideal gas law This relates to the properties of a gas and can be represented in the form $pv = nRT$.

IDLH See Immediately dangerous to life and health.

Ignition source A source that is of a high enough temperature or has enough energy in a spark to cause the ignition of a gas or a material being conveyed.

Immediately dangerous to life and health (IDLH) A condition that poses a threat of exposure to airborne contamination likely to cause death or immediate or delayed permanent adverse health effects, or that prevents escape.

Immission The rate at which a receptor of pollution encounters a pollutant.

Impact damage Damage caused to lungs, surfaces of ductwork, or fans by particulate matter.

Impaction Collection mechanism where the contaminants collide with the surface of the filter by inertia, interception, or Brownian diffusion.

Impactor A general term for instruments that sample particles in the air by allowing them to impact on a retaining plate.

Impactor, cascade An instrument consisting of stages for producing successively increasing air velocities for collecting particles by size range.

Impeller tip diameter (fan or pump) The maximum diameter measured over the tips of the blades of the impeller.

Impingement The collision of a dust particle upon a surface, leaving the particle on, for example, a filter fiber.

Impinger A sample collector that resembles a graduated cylinder with a long tube fitted into a stopper. The inlet tube extends nearly to the bottom of the outer tube, which holds the solution. The sampling pump is connected so that negative pressure is created inside the impinger, drawing air through the inlet tube into the solution, allowing the air to bubble up through the solution.

Implementation plan A plan that makes practical provision to ensure that set environmental standards are met.

Impulse noise Noise of short duration, that is, 3 seconds or less. Also called impact noise.

Incineration The process of burning solid, liquid, or gaseous combustible wastes, leaving a sterile residue containing little or no combustible matter.

Inclined manometer A manometer in which the vertical movement of the liquid column is amplified by inclination of the U-shaped reading tube.

Index number A number that is used to indicate general trends in a quantity.

Indoor air (IOA) classification Categories defined by CEN 156 to classify the quality of indoor air.

Category	Description
IDA 1	Excellent air quality
IDA 2	Typical air quality
IDA 3	Low but acceptable air quality

Indoor air is also classified based on CO_2 concentration.

Classification	CO_2 (ppm)	Percentage dissatisfied (%)
IDA-C1	800	15
IDA-C2	1000	20
IDA-C3	1500	30

Indoor air quality can be controlled using several methods.

Classification	Description
No control	A constantly running system
Simple control	A system that only runs at the dictate of a sensor, such as an infrared sensor detecting movement
Direct control	A system controlled by a sensor that detects levels of indoor contaminants within the space

Indoor air quality The actual quality of air within a space compared with a given sample or standard, related to temperature, moisture, biological content, and contaminant levels.

Indoor climate The actual temperature, moisture content, and air velocity within a space.

Indoor pollution Pollution inside a building due to internally generated pollutants as well as external pollutants entering the building.

Induced air The quantity of secondary air entrained into a primary airstream.

Induced air temperature See Temperature.

Induced draft Air drawn through a fuel bed, a furnace, or a space by means of a fan situated beyond the item.

Induced leakage Leakage into an enclosure to equalize a pressure difference due to natural or mechanical ventilation.

Induction The process by which secondary air is entrained into the primary jet air. The mixing process is due to the momentum forces in the primary air jet.

Induction effect See Induction.

Induction ratio The ratio of entrained air to primary air.

Induction supply ATD An air terminal device in which the primary air from the duct induces secondary airflow from the treated space in such a way that a high rate of mixing between the air from these two sources takes place within the device.

Induction terminal unit An air terminal assembly which by virtue of the configuration of the primary air inlet(s) within the unit can induce secondary air from the surrounding atmosphere before being discharged to the treated space. The flow rate of the

primary air may or may not be variable. The inlet aperture(s) for the secondary air may be fixed or adjustable by means of manual remote control.

Industrial hygiene The science and art devoted to the anticipation, recognition, evaluation, and control of those environmental factors or stresses arising from the workplace which cause sickness, impaired health and well-being, or significant discomfort and inefficiency among workers or among the citizens of the community.

Industrial hygienist A person qualified in the associated sciences dealing with the industrial environment.

Inert gas A gas that does not react with other substances under ordinary circumstances, for example, nitrogen.

Inertia The resistance of a body to a change in its momentum or direction of motion.

Inertia bases Bases for the mounting of fans, pumps, or other rotating machines that are designed to eliminate the transfer of inertial forces to the structure.

Inertial deposition The deposition of particulate matter that occurs under the influence of an inertial force.

Inertial effects The force due to inertia equal in magnitude but opposite in direction to the accelerating force.

Inertial impaction This is the predominant mechanism used in all particle collection devices.

Inertial separator A device used for separating bodies from one another, or from a fluid in which they are contained, by virtue of inertial differences.

Infiltration The leakage of air through the imperfections in a building structure, due to thermal or wind forces.

Infiltration rate The rate at which outdoor air enters into a room through the imperfections in the building structure, expressed in air changes per hour or L/s.

Influx The rate at which a gas enters a space.

Influx velocity The velocity of a gas as it enters an opening.

Ingestion The absorption of substances into the gastrointestinal tract.

Inhalable fraction Particles with aerodynamic diameters up to 10 μm, which can enter the lungs.

Inhalation The process of breathing in, when the air enters the respiratory tract.

Inlet The opening that allows air or a contaminant into a system by either natural or mechanical forces.

Inlet bells or boxes Aerodynamically shaped inlet ducts for a fan.

Inlet vane dampers Dampers inserted in the airstream at the inlet of a fan.

Inlet vanes Specially designed adjustable vanes inserted in the airstream entering a fan inlet to control fan performance by producing a swirl of the gas in the direction of the rotation of the impeller.

Insertion length See Overlap length.

Insertion loss, weather louver The difference in simulated rain penetration between the test specimen and the calibration plate at the same test conditions.

Inspection panels See Door and inspection panel.

In-stack filters A filter positioned in the discharge stack to remove pollutants before the waste gases are discharged to outdoors.

Installation The complete plant arrangement.

Instantaneous Relating to the measurement of a variable, it may be either a peak reading or a measure at any other time during sampling.

Instrumentation A sensor that either simply or automatically controls HVAC equipment or that records some particular function of the plant operation, such as temperature, pressure, or flow.

Insulation of clothing The resistance to sensible heat transfer provided by a clothing ensemble, reflecting the intrinsic insulation between the skin and the surface of the clothing, excluding the resistance provided by the layer of air surrounding the clothed body.

Integrated sampling Samples taken by drawing the air to be tested through the sampling medium, which is then analyzed by a laboratory to determine the amount of contaminant transferred.

Interception A special case of impingement, in which a particle is trapped on a fiber due to the effect of Van der Waals forces rather than inertia. The interception of a particle in a particle collection device occurs when the particle follows a gas streamline round a collector at a distance less than the radius of the particle.

Interferent Any undesirable component in a sample to be analyzed that will adversely influence the instrument reading.

Interim order An official statement issued by OSHA allowing an employer to continue operations under existing conditions while an application for a variance is being considered.

Intermittent sampling Any sampling process carried out for limited periods of time rather than continuously.

Internal energy The energy contained in a substance, which is its ability to do work.

Internal heating load Heat gains that occur in a space from process loads, lighting, solar gain, occupants, machines, etc.

Internal leakage The leakage that takes place into an enclosure or ductwork from outside.

Internal pressure The pressure inside a space or container, as opposed to the pressure outside.

Internal temperature The temperature inside a space, as opposed to the external temperature.

Internally induced airflow rate of an ATD Volume of air induced into the primary airflow inside the air terminal device in unit time.

Internally mounted air transfer device See Air transfer device.

Interstitial Situated between the cells of a structure or part.

Interstitial condensation Condensation that occurs within the interstices of a material when the dew point is reached.

Intoxication The general state of the body caused by the effects of a toxic substance.

Intrinsic clothing thermal efficiency Reduction of sensible heat exchange due to wearing clothes.

Intrinsically safe Instruments that can be safely operated in an explosive or corrosive atmosphere.

Intermittent duty The duty of a device when operating on a part-time basis, as opposed to continuous duty.

Inversion The condition that occurs when the lapse rate is positive, that is, temperature rises with height at a rate greater than the adiabatic lapse rate 3°C per 300 m. In these conditions, stagnant air pollution builds up and is trapped under this layer.

Involute A geometrical curve as used in a centrifugal fan casing.

Ionization The critical voltage at which gas molecules are separated into positive and negative ions in an electrostatic precipitator.

Ionizing radiation Radiation that is capable of causing ionization to occur, either directly or indirectly through interaction with matter.

Iris damper or valve See Dampers and Valve.

Irradiance The radiant flux striking a unit area of a surface.

Irradiation The exposure to radiation of any kind.

Isentropic operation A change in conditions at constant entropy.

Isokinetic A process in which the velocity at the entrance to the sample probe in a gas stream is the same as the velocity at a given point in the duct or stack at a given time.

Isokinetic sampling The sampling of a gas such that the motion of the gas entering the sampling device is identical to that of the gas being sampled.

Isolation The process of disconnecting a supply, electricity, fuel, or air.

Isotherm A line in a flow system or on a graph connecting points of equal temperature, or a mathematical or graphical relationship between two variables at constant temperature. Or a display using lines on a drawing to show constant-temperature contour lines, as from thermal imaging with infrared techniques.

Isothermal change A process that takes place at constant temperature, such as the isothermal expansion of a gas.

Isotopes Atoms of the same element (having the same atomic number) that differ in mass number.

Isovel A line in a flow system or on a graph connecting points with constant velocity.

J

Jet A gaseous or liquid stream issuing from a slot, orifice, or nozzle.

Jet angle The angle at which a jet or array of jets diverges into a free space.

Jet, Coanda A jet attached to a surface.

Jet drop The downward change in the direction of a jet due to the difference between its velocity or temperature and that of the ambient air. Also called jet fall.

Jet, enclosed A jet that is allowed to expand within a channel or duct and then is constrained by the channel or duct walls.

Jet envelope The boundary between a jet and the surrounding air.

Jet fan See Fan functions.

Jet, free A jet that, on leaving an orifice, is allowed to expand freely without coming into contact with any surfaces.

Jet, isothermal An air jet of the same temperature as the space it is entering.

Jet, nonisothermal An air jet of a different temperature from that of the space it is entering.

Jet rise The upward change in the direction of a jet due to the difference between its velocity and temperature and that of the surrounding air.

Jet spread The angle of divergence of a jet from its point of origin.

Jet, wall A jet that attaches itself to a surface. See also the Coanda effect.

K

Kata cooling power The rate of cooling of a silvered or unsilvered kata thermometer due to the relationship between the air temperature and the air velocity over the bulb.

Kata thermometer A thermometer that allows the air velocity to be determined by its cooling power.

Katharometer A device that compares the thermal conductivity of two gases, used to detect the presence of impurities in air.

Kelvin effect The electrical potential gradient caused by a temperature gradient along a conducting wire. Also known as the Thomson Effect.

Kinematic coagulation The scavenging of small particles by large particles, which increases the speed of the larger particles by differential settlement, such as in rain drops or from spray nozzles.

Kinetic energy The energy a body possesses by virtue of its motion.

Kinetic theory A mathematical explanation of the behavior of gases on the assumption that gases consist of molecules in ceaseless motion in space. The molecular kinetic energy depends on the temperature of the gas.

Kirchhoff's law The relationship that exists between the absorptivity and emissivity of radiating bodies. It is the capacity of a body to absorb radiation, which varies with the wavelength of the incident radiation and the angle of incidence.

L

Laboratory blanks Sample media that is not sampled on but is analyzed by the laboratory to detect contamination or other problems associated with preparation and analysis of the samples. See also Field blanks.

Lag A delay between a change in a condition at one point in a system and its effect.

Lambert–Beer law The mathematical description of the attenuation of a light beam by absorption and scattering by dust particles in the airstream.

Laminar flow Fluid flow in which the fluid particles move in straight lines parallel to the axis of the pipe or duct.

Land breeze The air movement that takes place after sunset, when the land cools and air currents flow from the land to the cooler sea.

Langmuir equations The mathematical expressions that describe vapor adsorption equilibria.

Lapse rate The rate of temperature increases with height.

Laser Light amplification by stimulated emission of radiation.

Laser anemometer See Laser Doppler anemometer.

Laser Doppler anemometer An instrument for determining fluid velocity by measuring the difference in frequency between the incident beam and that scattered from particles moving with the flow.

Latent heat The quantity of heat that is absorbed or released in an isothermal transformation of phase, in kJ/kg/°C. Latent heat of vaporization The heat added during an isothermal change of phase from liquid to gas.

Laws of perfect gases

1. Boyle's law: The volume of a gas is inversely proportional to its pressure, at constant temperature.
2. Charles' law
   a. The volume of a gas is proportional to its absolute temperature, at constant pressure.
   b. The pressure of a gas is proportional to its absolute temperature, at constant volume.
3. Joule's law: The internal energy of a given quantity of gas depends only on its temperature and is independent of its pressure and volume.

Laws of thermodynamics

a. First law: Heat and work are mutually convertible.
b. Second law: Heat will not pass from a colder to a hotter body spontaneously.

LCL See Lower confidence limit.

LD_{50} (median lethal dose) A standard measure of toxicity indicating the dose of a substance that will kill 50% of a group of test organisms.

Lead dioxide candle A device for determining the amount of sulfur dioxide in the air. SO_2 reacts with a film of lead dioxide to produce lead sulfate, which is measured to determine the concentration.

Leakage The rate of fluid loss from an enclosure due to a pressure difference between the inside and outside of the enclosure.

Leakage function The relationship of the leakage occurring in a building to the pressure difference, measured in $m^3/h/Pa$.

Leakage path leeward Leakage of building air that takes place due to structural openings on the downwind or sheltered side of a building.

Legionella pneumophila (LD) Infections, particularly pneumonia, caused by inhaling *Legionella pneumophila* and other bacteria from the family *Legionellaceas* in water droplets drifting from cooling towers, showers, etc.

LEL See Lower explosive limit.

Length-of-stain tube See Detector tube.

LEV See Local exhaust ventilation.

Lewis relationship The ratio of the convective heat-transfer coefficient to the evaporative heat-transfer coefficient.

Lesion An injury to the body due to the intake of certain atmospheric pollutants.

LFL See Lower flammable limit.

LIDAR An instrument that uses a laserradar to study the concentration and location of particulate matter by the reflection or absorption of a laser beam.

Life cycle The design of any combination of plant items that considers the owing and operating costs of the plant.

Life cycle assessment (LCA) An analysis defined by ISO 14040 as "compilation and evaluation of inputs and outputs and the potential environmental impacts of a production system throughout its life cycle."

Limestone scrubbing A process using a ground limestone and water mix to neutralize sulfur dioxide in waste gas products.

Limit, opening The maximum sash opening that can be allowed on a laboratory fume cupboard to ensure safe working conditions.

Limit of detection (LOD) The smallest amount of contaminant that can be reliably detected by a particular analytical method.

Limit of quantification (LOQ) The smallest amount of contaminant that can be reliably quantified using a particular analytical technique.

Limit value A reference figure giving the allowable concentration of a chemical or biological agent in the air.

Linear air diffuser An air terminal device with single or multiple slots, each of which has an aspect ratio not less than 10:1. Each slot may consist of a number of separate elements and may or may not have an adjustable member, which allows the directions of the air delivered to the treated space to be varied.

Linear grillGrill with an aspect ratio not less than 10:1.

Liquefied natural gas (LNG) Natural gas cooled to $-162^{\circ}C$ to achieve a significant volume reduction, then stored under pressure.

Liquefied petroleum gas (LPG) Paraffin hydrocarbon gases comprising propane, butane, and pentanes derived from natural gas wells and from the petroleum refining process that remain as liquids when stored under pressure in tanks and bottles.

Liquid entrainment separator Any device that removes and collects moisture present in an airstream.

Load In a ventilating or heating system, the magnitude of heat, airflow, or cooling the system must provide to meet the design conditions. The work the system must perform. The heating, ventilating, or cooling load requirement of a space or appliance.

Load, connected The sum of all the individual loads related to an HVAC system.

Load factor The ratio of the average demand to the maximum demand; may relate to electrical, heating, or cooling load.

Load pattern The load change over time.

Load utilization factor Ratio of the effective load in a given space to the load supplied.

Loading dust The selected synthetic dust used to determine the dust-holding capacity of a filter.

Local air velocity The air velocity in the zone in which the design conditions have to be met. Or, the air velocity recorded at a specific location in a space or in a jet stream.

Local exhaust ventilation (LEV) The removal of a contaminant at or near the point of its generation.

Local cooling The cooling of a given area in a space by means of chilled air jets or chilled water panels.

Local mean velocity The magnitude of the time-averaged vector of velocity at a point in an airstream.

Local response The response of an occupant or control device to changes in the local environment.

Local ventilation The transportation of air into or from a space near its point of use.

Long-term exposure limit (LTEL) An exposure limit requirement based on the assumption that the total body intake of a pollutant below this limit over an 8-hour working day will have no harmful effect on the worker over a working life. See also Maximum exposure limit (MEL), Occupational exposure limit (OEL), and Short-term exposure limit (STEL).

Louver An opening that allows air to enter or leave a space, which has inclined vanes to provide protection from the entry of rain, snow, and animals.

Low-leakage seal A seal on a fluid-handling device that ensures system leakage is at or below a given level.

Low potential hazard A concentration of pollutants within a space, which will present a very low hazard to the occupants or the plant.

Low-velocity ATD An air terminal device that is designed for thermally controlled ventilation, for example, displacement flow applications. See also Air terminal device.

Low-volume high-velocity (LVHV) The method of local exhaust using small hoods, which exhaust contaminants from a process at velocities of 50–100 m/s.

Lower confidence limit (LCL) A statistical procedure to estimate whether the true value is lower than the measured value.

Lower explosive limit (LEL) The lowest concentration of a substance in air at ambient temperature that will explode if ignited, expressed as a percentage of the substance in the room air by volume. See also Upper explosive limit (UEL).

Lower flammable limit (LFL) The lowest concentration of a substance in air that will sustain combustion.

Lower limit of a duct The algebraic difference between the minimum limit of size and the corresponding nominal size.

Lubrication The use of oil or grease on moving parts in order to reduce the friction force.

Lung function Relating to the transfer of oxygen from air into the blood and the disposal of carbon dioxide from the blood to the air.

M

Macropores Pores of diameter greater than 0.005 mm in the structure of an adsorbent medium.

Magnehelic dial gauges A measuring device recording the static pressure of a fluid.

Make-up air Air introduced into a space to replace air that is being extracted.

Main air treatment The part of the treatment of air that, by virtue of the number of air handling functions involved or of the effect achieved, is considered the principal treatment.

Male connector A short circular sleeve to join two pieces of spiral duct together. The ends of the male connectors are inserted into the spiral tube ends.

Manifold A section in the exhaust air ductwork of an air treatment system into which exhaust air enters from a number of orifices or ducts, or a header pipe in a fluid flow system that has branches.

Manometer An instrument that measures pressure by fluid displacement in a U-shaped tube.

Manometer pressure The pressure recorded on a manometer, measured in Pa or mm water gauge.

Manual damper or valve A metal or plastic flap used to control the rate of airflow in a duct manually.

Manually adjusted ATD See Air terminal device.

Marginal irritant A material that is capable of causing an irritation response after repeated exposures.

Masking Relating to an additive introduced into the air supply in an attempt to neutralize or conceal an odor.

Masks A protective device complete with an approved filter designed to keep dangerous chemicals from being inhaled into the lungs.

Mass flow rate Mass of matter, which crosses a given surface, divided by time.

Mass spectrometer (MS) An instrument that identifies substances by causing them to be ionized and subjecting the resulting ions to a strong electromagnetic field.

Mass transfer The transfer of mass across a boundary, similar to heat transfer.

Maximum allowable concentration (MAC) An old American definition used before the term TLV came into use (the term is still used in Germany). See Threshold limit value (TLV).

Maximum body heat storage (Q_{max}) The maximum value of the body heat gain achievable by the subject such that the resulting increase in body core temperature does not induce pathological effect, in W h/m^2.

Maximum exposure limit (MEL) The maximum concentration of an airborne substance, averaged out over a reference period to which employees may be exposed by inhalation.

Maximum penetrating particle size The particulate size for which a filter has minimum removal efficiency under test conditions.

Maximum-risk employees Workers who are most likely to be exposed to the highest levels of hazardous agents.

Maximum use concentration (MUC) The maximum atmospheric concentration of contaminants in which a respirator cartridge or filter is recommended for use. Can be approximated by multiplying the PEL for the contaminant of concern by the assigned protection factor.

Mean, arithmetic The sum of a set of n numbers divided by n, often called the average.

Mean air temperature See Temperature.

Mean diameter The geometric mean diameter of the size range.

Mean free length The mean free length of a particle (μm) used in particle scrubbing equations.

Mean free path The average distance traveled by a particle between collisions. In a gas, it is inversely proportional to the pressure.

Mean, geometric The nth root of the product of n terms.

Mean particle diameter The mean value of the particle size distribution of the test aerosol.

Mean radiant temperature The average temperature of the six surfaces of a cubicle enclosure, used in thermal comfort work and in other heat-transfer applications. It is the sum of all the surface areas multiplied by the temperature of the surface divided by the total surface area.

Mean skin temperature The average temperature of the skin exposed to a given environment.

Means, best The best practical means for preventing the escape of noxious or offensive gases, smoke, grit, and dust from a process into the atmosphere (U.K. Alkali Regulations).

Measured variable A variable that is measured, and may be controlled.

Measurement station Element inserted in ductwork or pipework to facilitate the determination of temperature, humidity, flow rate, and/or pressure.

Measuring procedure Procedure for sampling and analyzing one or more chemical agents in the air, including the storage and transportation of the sample to the laboratory.

Mechanical constant flow rate controller See Flow rate controller.

Mechanical diffusion Eddy diffusion caused by mechanically produced turbulence.

Mechanical efficiency The actual work possible by a machine, related to the work put into that machine.

Mechanical variable flow rate controller See Flow rate controller.

Mechanical rapping The mechanism that vibrates electrostatic precipitators or bag filters in order to remove the dust burden.

Mechanical shakers A table that vibrates at a given frequency in order to remove particulate matter from a casting.

Mechanical turbulence Any turbulence produced by means other than natural, such as fans. The term is also used incorrectly to define wind currents set up in and around buildings.

Mechanical ventilation Ventilation created by fans or other air-moving devices within a building, which can be divided into the following classifications:

- Mechanical extract—induced inlet.
- Mechanical inlet—forced outlet.
- Mechanical inlet—mechanical outlet.

Mechanism An arrangement that allows rotary movement to be converted to linear movement or vice versa. A linkage for dampers, etc.

Media filters Filters that collect particulate matter on individual filter elements.

Median The central value of a series of observations ranked in order of magnitude.

Medical surveillance program The evaluation of an employee's health status, performed on a regular periodic basis by a health professional, to detect problems associated with exposure to health hazards, so that appropriate steps can be taken to prevent permanent or debilitating injury. Medical surveillance programs may also be used to ensure that an employee's health status will allow the continued safe use of protection equipment or the continued safe performance of work.

Membrane A film used for collection of particulates in which the size of the microscopic pores is controlled.

Mesopores Pores of diameters from 0.00005 to 0.005 mm that form the internal structure of an adsorbent material.

MEL See Maximum exposure limit.

Melanoma A skin tumor containing dark pigment.

Melting point The temperature at which a solid liquefies. The same as freezing point.

Mercaptans Organic compounds containing sulfur, which have an unpleasant odor.

Mesh A metal fiber or other material formed into a woven lattice, used to strain or filter out particulate matter from a fluid or gas.

Met unit The metabolic rate of a sedentary person at rest, 1 met = 58.2 W/m^2.

Metabolic energy transformation Metabolic rate.

Metabolic heat production The production of body heat due to the intake of oxygen and carbohydrates.

Metabolic rate (M) The rate of transformation of chemical energy into heat and mechanical work by aerobic and anaerobic metabolic activities within an organism, usually expressed per unit area of the total body surface, in met or W/m^2.

Metabolic rate, basal (BM) Metabolic energy transformation calculated from measurements of heat production or oxygen consumption in an organism in a rested, awake, fasting, and thermoneutral state, in W/m^2.

Metabolic rate, seated (M_s) The heat liberated from a body when the occupant is seated at rest, in met or W/m^2.

Metal fume fever A fever suffered by workers who inhale metal fumes from a process.

Metal poisons Certain metals that cause illness or death when inhaled or ingested.

Methods of air distribution Can be classified into the following groups.

- Crosswise: Airflow that takes place from one side of a space to the other. This may be achieved by one or more jets or by allowing the air to enter the whole of one side surface and extracting the air by the whole area of the opposite side. The latter arrangement provides a piston effect, ensuring good air and contaminant transport.
- Downward: The supply air enters at ceiling level or high wall level and is extracted at low level. A perforated ceiling may be used to provide a piston effect. Good air mixing is achieved if cool air enters at high level at the correct temperature and velocity.
- Mixed upward and downward: Downward supply with a small proportion of high-level extraction. The largest proportion of the extraction occurs at low level. This arrangement provides good mixing of the room air, if care is taken to ensure that short-circuiting of the high-level input and extraction does not take place.
- Upward: Air enters the space at low level and is extracted at high level, ideally suited for warm air supply.

Microclimate The distinctive pattern of temperature, humidity, air movement, and purity within a relatively small zone either inside or outside a building.

Microclimate suit A suit worn to protect an operator who is working in adverse conditions of either heat or cold.

Micrometer (μm) The SI unit of measurement of particulate matter, equal to 1×10^{-6} m. The non-SI term is the micron (μm).

Microorganisms Organisms that can only be seen with the aid of a microscope, such as bacteria, viruses, and some fungi.

Micropores Pores of diameter less than 0.0005 mm that form the internal structure of an adsorbent material.

Migration The movement of dust collected in a filter in the direction of flow.

Migration velocity The electrophoretic velocity of a charged particle in an electric field.

Millibar (mbar) A unit of pressure equal to 100 Pa.

Minimum air change rate The lowest possible air change rate that can be used in a space in order to attain the recommended air purity standards.

Minimum filter efficiency The value of a filter's efficiency relating to its performance classification under specified operating conditions. See also Maximum penetrating particle size.

Minimum ventilation requirements The lowest possible airflow rates that will ensure all obnoxious products are removed from the air by the introduction of fresh air.

Mist The suspension in air of small droplets of materials that are liquids at normal pressure and temperature.

Mist elimination The removal of a mist from a gas stream either by condensation or by the use of baffles.

Mitered elbows A bend in pipe or ductwork formed by a series of flat sections.

Mixed flow, actual An actual flow pattern in an enclosure resulting in the air being mixed to such an extent that conditions are almost the same at every point in the occupied zone.

Mixed flow, ideal The flow pattern in an enclosure in which the air is completely mixed and has the same conditions at every point.

Mixed air Air that contains two or more streams of air.

Mixing The process by which fluids of different density are combined naturally or by some mechanical device. One or more of the following mechanisms may produce the mixing process:

- Displacement,
- Piston flow,
- Spot (local),
- Laminar flow, and
- Layering.

Mixing actuator Component designed to mix two airflows without controlling the volume.

Mixing air diffusion Air diffusion where the mixing of supply air and room air is intended.

Mixing controller Component designed to mix two airflows while controlling the volume flow.

Mixing factor A factor used in air distribution relating to the actual degree of mixing that takes place between the room air and a contaminant generated in that room.

Mixing height The height above an internal or external pollutant source within which emitted pollutants are dispersed and mixed with the surrounding atmosphere. In meteorological terms, this is the area below the inversion layer.

Mixing section of an ATD A section in which two air streams of different temperatures or moisture content are damper controlled to provide a given flow rate before mixing occurs.

Mixing section of an AHU A section where outdoor airflow and the recirculation airflow are mixed in a controlled manner.

Mixture rule A mathematical expression applying to workers simultaneously exposed to chemicals that act on the same organ or organ system. The exposure level for each chemical must remain at a fraction of the permissible exposure level (PEL) so that the sum of the fractional exposures does not exceed unity.

Model A model of a particular flow problem, which may be either

- Mathematical A mathematical simulation of the emission, dispersion, and chemical process relating to the concentration of pollutants.
- Physical A model system in which tests are carried out on the emission and dispersion of a pollutant, for example, a wind tunnel.

Modulating The control action of minute increments and decrements of adjustment in a system, such as in automatic control valves.

Moisture content The mass of water vapor present in a unit mass of dry air.

Moisture recovery Measures taken to prevent the moisture present in the air from leaving the treated space.

Molar diagram A plot of the thermodynamic properties of a substance that has specific enthalpy as one of its coordinates.

Mold diseases Diseases produced by the concentration of mold or fungi spores within a space.

Mole The SI unit of quantity; the amount of a pure element or chemical compound that contains the same number of atoms or molecules. It is often simpler to use moles rather than volume or mass when working with gases. Moles are given by

$$n = \frac{m}{\mathrm{MM}}.$$

For example, 32 kg of $O_2 = 1$ mol of oxygen and 16 kg of $O_2 = 0.5$ mol of oxygen.

Molecular diameter of air Air at atmospheric pressure is 0.00037 μm at 20°C.

Molecular sieve Zeolites used for ion exchange in water treatment.

Monitoring The continuous or regular observation of a fixed or variable parameter.

Morbidity The incidence of disease in a community or a working group.

Multiple-leaf damper or valve An air damper or valve that has more than one damper, arranged to provide low flow loss.

N

Narcotic gases Gases that produce sleep, stupor, or insensibility when inhaled in certain concentrations.

Natural atmospheric dispersoids The American Meteorological Society classifies natural dispersoids by size:

- Haze 1.0 μm,
- Mist 1–5 μm,

- Cloud or fog 5–200 μm,
- Drizzle 200–500 μm, and
- Rain 500–8000 μm.

Natural circulation Circulation occurring in a fluid due to temperature changes.

Natural ventilation Ventilation achieved by means of wind forces or density differences or a combination of the two, as opposed to mechanical ventilation, which depends on a rotodynamic device.

Natural ventilation system Ventilation of a space by the influence of thermal forces and wind forces over and around a building. Under certain conditions, only one of these applies, however, in the majority of cases it is assumed that both apply.

Necrosis The death of any cell tissue by the action of pollutants.

Negative pressure A pressure less than the ambient pressure, which may be created due to stack effect or by mechanical means.

Negative rated operating pressure The tested maximum negative pressure at which a duct is rated.

Nephelometer A device used to determine the suspended particulate size and concentration by the scattering of light.

Neutral clothing insulation See Clothing insulation, neutral requirements.

Neutral solution A chemical solution that is neither acidic nor alkaline.

Neutral zone The physical state within a building where no pressure difference exists between inside and outside the building. Also used in relation to the effect of a chimney in removing the products of combustion.

Nitrogen oxides A number of different compounds of nitrogen and oxygen, normally referred to as NO_x.

Noise An unwanted sound that causes annoyance or distraction.

Nominal length, flexible duct The actual length of a flexible duct after decompression in an unstressed state.

Nominal length, rigid duct The actual length of a rigid duct without fittings or components.

Nominal size of an ATD The nominal value of dimensions of the prepared opening (duct) into which the air terminal device is to be fitted. For an air diffuser, the nominal size is generally defined as the duct size into which the neck of the device is fitted.

Nominal size of a duct or fitting The reference dimension used for the designation, calculation, and application of ducts and fittings.

Nomogram A chart consisting of variables, and provided two of these are known others can be determined.

Nonoverloading fan A fan with backward-curved blades, which has power characteristics that tend to flatten with increasing flow rates so that as the maximum volume flow rate is approached the power consumed may become constant or even decrease. The power characteristics of a fan with forward-curved blades steepen at high-volume flow rates, and so such fans are overloading.

Nonspecific When an instrument responds to more than one contaminant that is known to be present in the air.

Normal temperature and pressure (NTP) See Standard temperature and pressure (STP).

Nosocusis Hearing loss resulting from causes other than noise, such as disease and heredity.

Noxious A term relating to any chemical that is harmful to the occupants of the space in which it exists.

Nozzle An air terminal device used to obtain the maximum conversion of static pressure to dynamic energy with minimum entrainment.

NTP Normal temperature and pressure. See Standard temperature and pressure (STP).

Nucleation A cleaning process using a humidification and cooling cycle, causing water or another fluid to condense on submicrometer particles. This process increases particle size until impingement on packing is possible.

Null point The distance from a generated pollution source at which the initial energy or velocity of the contaminants is dissipated, and collection by a hood is possible.

Nuisance dusts Any dust that creates a nuisance, rather than being a health risk, such as dusts that cause sneezing, coughing, eye irritation, etc.

O

O Ring A device used to seal a shaft of a device or a pipeline that is conveying a fluid.

Obscuration The concealing from sight; lack of visibility due to dust, fumes, or smokes.

Occupied zone The volume of air confined by horizontal and vertical planes defined to include space occupied by persons.

Occupational exposure limits (OEL) The maximum time a person can work in a given polluted environment.

Occupational exposure standards (OES) U.K. standards relating to the concentration of an airborne substance that can be tolerated without harmful effects on workers over a reference period. See Long-term exposure limit (LTEL) and Short-term exposure limit (STEL).

Octave band frequency The band in frequency scale that is split into bands, each assigned a sound power level, that is twice the power level of the lower limit.

Odor Relating to the sense of smell, a substance that stimulates the olfactory organ, allowing us to detect if a smell is pleasant or unpleasant.

Element	Distance from the inner surface of the elements, m	
	Typical range	Default value
External windows, doors, and radiators	0.5–1.5	1.0
External and internal walls	0.25–0.75	0.5
Floor (lower boundary)	0.00–0.2	0.1
Floor (upper boundary)	1.30[a]–2.0[b]	1.8

[a] *Mainly seated occupants.*
[b] *Mainly standing occupants.*

It will be appreciated that in the industrial environment, each case will have to be considered in its own rights. Except when agreed otherwise, the default values shall be applied.

Odor control The elimination of odor in a space by the use of masking chemicals or special filters.

Odor dispersion time Time taken to reduce an odor to a defined level from a given concentration in a standard test.

Odor reduction factor The efficiency of odor reduction by a medium capable of removing odors.

Odor threshold The minimum concentration of an odorous gas which 50% of a panel of trained sniffers can detect.

Off-gassing of materials The liberation of volatile organic compounds (VOCs) and other gases from building products or from a manufacturing process.

Offset A sustained deviation between the control points and the set point of a proportional control system.

Oil slant gauge An inclined manometer tube using oil as the measuring fluid to record the pressure.

Olfaction The sense of smell or the act of smelling.

Online Relating to either

- The measurement of a variable in a process that is in operation, or
- A computer arrangement to process input data without delay and to present an output result.

On/off control A simple two-position control system that is only capable of performing these two functions (on and off).

Once-through scrubber system A system seldom used due to the problem of using large quantities of fresh water and the resulting discharge of a large volume of polluted water.

Opacity The degree to which a plume of exhaust gases obscures the view of an observer, measured in terms of percentage obscuration, with 100% meaning that the plume completely obscures the line of sight through the plume.

Open face Sampling with the top portion of a filter holder or cassette removed to ensure the full filter surface is exposed.

Open rotor An open air lock with open ends between the rotor blades and the blade seal.

Operating point The static pressure and volumetric airflow that a fan is capable of producing.

Operative temperature See Temperature.

Opposed-blade damper A damper in which the adjacent blades rotate in opposite directions so that the leading edges seal against each other and the trailing edges seal against each other.

Optical anemometer An instrument for measuring gas flow rate using a laser, in which small frequency shifts are visualized as interference fringes.

Optical particle counter An optical-electronic instrument for measuring the number of airborne particles in different size ranges.

Optimum droplet size The ideal size of a water droplet in a centrifugal spray scrubber or spray tower to ensure the highest possible cleaning efficiency.

Oral temperature The temperature in the mouth recorded by a thermometer or thermocouple.

Organic Relating to or derived from living matter that has organs or an organized physical structure. Organic chemistry is the study of carbon compounds.

Organic contamination The contamination of products by organic matter, particularly in clean room applications.

Organic material A material from an organic source.

Organic solvents An organic liquid capable of dissolving an organic material or compound.

Organoleptic A material that influences a sensory organ, as in the perception of odor by the human nose.

Orifice meter See Orifice plate.

Orifice plate A metal plate with a hole of diameter smaller than the pipe or duct run in which it is fitted. The pressure drop that takes place across the plate is used to calculate the fluid velocity.

Category	Description
ODA 1	Pure air, which may be temporarily dusty (e.g., pollen)
ODA 2	Air with a significant concentration of dust
ODA 3	Air with significant concentrations of gaseous pollutants
ODA 4	Air with significant concentrations of gaseous pollutants and dust
ODA 5	Air with very high concentrations of gaseous pollutants or dust

Orifice venturi A measuring device used to determine the flow rate of a fluid by means of the pressure drop across the device.

Outdoor air Air introduced into a building from a source external to the building.

Outdoor air (ODA) classification This classification covers five categories of air quality from ODA 1 to ODA 5 as shown at the top of the next page.

Outdoor pollution Natural or man-made pollution produced by sources external to a building.

Outlet An opening through which air or effluent is discharged, either by natural or by mechanical means.

Outlet damper A device fitted in a duct that will allow the flow of gas to be controlled either manually or automatically.

Overall heat-transfer coefficient The heat flow per unit area for a given construction for an overall temperature difference of 1K.

Overall uncertainty of a measuring procedure or of an instrument The quantity used to characterize the uncertainty of results given by an apparatus or a measuring procedure, expressed on a relative basis by a combination of bias and precision, according to a formula.

Overbreath In using a respirator, when the wearer's breathing rate exceeds the ability of the respirator to provide a volume of air sufficient to ensure that a positive pressure is maintained inside the face piece.

Overlap length The length by which a fitting or duct overlaps a connecting duct.

Oxidants Substances present in air, such as nitrogen dioxide, and ozone, that are capable of oxidizing other chemicals or elements in oxidation-reduction type chemical reactions.

Oxygen consumption The rate at which the lungs take up oxygen.

Oxygen-deficient An atmosphere consisting of less than 19.5% oxygen.

Oxygen-enriched An atmosphere containing more than 23% oxygen.

P

P4SR Predicted 4-hour sweat rate. A scale used to predict the evaporation rate from a body under hot conditions.

Package units Air-handling equipment containing all the components together in a common casing.

Packed beds An absorption separator that employs a fluidized bed of plastic spheres constrained between horizontal screens.

Packed-tower wet scrubber A gas scrubber that removes gases and vapors, by using either water or a chemical liquid method. Efficient pollutant removal depends on the contact time between the entering gas stream and the wetted surface of the pack in the tower. This type of scrubber can be classified as

1. Concurrent flow,
2. Cross-flow, or
3. Countercurrent flow.

Paddle wheel impellers A radial blade impeller on a fan, used for dust conveying due to its self-cleansing properties.

Panel fans A simple form of axial fan with its impeller mounted in a ring or diaphragm; it discharges air both axially and radially.

PAP See Photochemical air pollution.

Parallel flow Referring to the operation of two or more fans or pumps connected in parallel with each other. May also relate to a standby fan in a system.

Parameter A quantity that serves to determine a measurable or quantitative characteristic, or the variable feature of a measurement.

Partial air treatment Treatment that involves one or more of the possible methods of treatment but is not complete.

Partial enclosure An enclosure used for work with toxic dusts, gases, or vapors in which one or more of the sides may be open to the remainder of the work area.

Partial pressures See Dalton's law of partial pressures.

Partial ventilation system A local exhaust system designed to provide an airflow less than that required for all the hoods that form part of the system. The application is to provide system diversity.

Particle The nature of an aerosol or other pollutant.

Particle aerodynamic diameter The diameter of a sphere of density 1 g/cm^3 that has the same terminal velocity due to gravitational force in still air at set conditions of temperature, pressure, and relative humidity as the particle in question.

Particle counters A manual or automatic device used to determine the particulate concentration of a given gas sample.

Particle migration velocity The velocity at which a charged particle moves in a given direction in an electric field.

Particle scrubbing A gas-cleaning device that generates large particles that can be easily collected by combining them with liquid droplets issued from fine jets.

Particle size distribution A method of relating the size or weight of particulate matter, for example, 50% with diameters in the 0.1−1.0 μm range and 25% in the 1−5 μm range.

Particulate concentration The concentration of one or more particulates in a given quantity of a gas.

Particulate matter Matter consisting of particulate liquid and solid substances ranging in size from 0.0002 to 500 μm in diameter.

Parts per billion (ppb) Parts of a contaminant in a billion parts of air or water. Care has to be taken in ensuring the term billion is the correct one. In the past in the United Kingdom, a billion was 10^{-12}, but in the United States, a billion is 10^{-9}. It is now assumed that current practice relates to the latter. Other terms encountered are parts per hundred million (pphm) and parts per million (ppm).

Parts per million (ppm) The number of parts of a contaminant by volume in a million total parts. Volume ratio = −mole ratio = -pressure ratio in the case of ideal gases.

Partition fan See Fan.

Passive sampling Sampling that depends on the diffusion of the contaminant into a solid sorbent.

Pathogen A material that is capable of producing disease in living organisms.

Pathology The study of the causes and results of disease.

Peak-above-ceiling exposure limit The short-term exposure peak permitted above the OSHA standard ceiling exposure level.

Peak limit A pollutant or noise level that exceeds the ceiling exposure limit but is allowed for a specific limited time during the work shift.

Penetration The distance particles of a particular size will travel into a given filter before coming to rest.

Per capita air rate Volume intake of outdoor air per occupant.

Percentage saturation The ratio of the moisture content of moist air at a given temperature to the moisture content of saturated air at the same temperature. Also known as the degree of saturation.

Perforated plate See Flow equalizer.

Performance The operating characteristics of a device, as compared with those of the original design. Or the performance of an item of plant as stated by a manufacturer.

Peripheral nervous system Nerve tissues lying outside the brain and spinal cord, functions include the transmittal of sensory information such as touch, heat, cold, and pain, and the motor impulses for limb movement.

Permeation efficiency Reduction factor for latent heat exchange through clothing.

Permissible exposure limit (PEL) The maximum exposure level allowed by OSHA, expressed as an 8-hour time-weighted average. These are legally enforceable in the United States.

Permissible range The range of a physical quantity that satisfies the different parameters for each of the categories of the specified environment.

Permit space As defined by OSHA, a confined space that contains a hazardous atmosphere, a material that could engulf an occupant, a configuration that could trap an occupant, or any other recognized safety or health hazard.

Personal protective equipment (PPE) Devices and apparel worn by employees to prevent or reduce exposure to health and safety hazards in any adverse environment. Examples include respirators, gloves, chemical-resistant overalls, earplugs, and safety glasses.

Personal sample The result obtained from the products collected during the process of personal sampling.

Personal sampler A collection device attached to a person that obtains samples of air to be tested for radioactive, chemical, or biological agents.

Petri dish A shallow dish used to culture bacteria.

pH The unit used to relate the acidity or alkalinity of a liquid. Pure water is neutral and has a pH of 7.0, values below this denote acidity, and those above this denote alkalinity.

Phase equilibria The relationship between contaminant solubility in the gas and liquid phases at equilibrium, which must be known for absorption separator design.

Photoallergic A reaction similar to other allergic reactions of the skin.

Photochemical Relating to a chemical reaction brought about when sunlight encounters certain gaseous mixtures.

Photochemical air pollution (PAP) Pollutants such as nitrogen oxide and certain hydrocarbons that cause photochemical reactions in the air.

Photometer An analytical instrument containing a light source on the one side and a light detector on the opposite side that measures the amount of light that passes through the sample.

Photophoresis Particle motion that takes place in the direction of radiation due to the absorbed radiation warming one side of the particle more than the other.

Physical absorption The process of collecting a gas in water or another fluid.

Physical adsorption An exothermic physical process that gives off less heat than chemisorption.

Physical testing Any process involving an actual test in order to obtain results.

Physiology Study of the function of the human body.

P iezometer tube An open-ended calibrated glass or plastic tube that measures the pressure in a pipe or vessel full of a fluid.

Piping A metallic, plastic, glass, etc., enclosure, for the conveying of a fluid, a vapor, or a gas.

Piston effect The ideal method of air distribution, in which uniform airflow occurs over the whole of a room, such as when the air is injected into the room over the whole surface area of one wall and extracted from the opposite wall.

Pitch The spacing of holes in a flange, or the angle of fan blades. That attribute of auditory sensation depending primarily on the frequency of the sound in terms of which sounds may be ordered on a scale extending from low to high.

Pitot-static traverse The set positions of a Prandtl tube in a duct run required to provide a statistically valid set of readings. A series of measurements of the total and static pressure taken across an area of a duct to determine the air velocity at that point. The sampling distance should be at least 7.5 times the diameter of the duct away from any disturbances of air flow.

Pitot-static tube A measuring device consisting of two concentric tubes used to measure the total and static pressures in a duct run, known as a Prandtl tube.

Plane radiant temperature See Temperature.

Plaster frame See Fixing accessory of an ATD.

Plate-mounted axial flow fan See Fan types.

Plate-type design (space heaters) Type of heat exchanger characterized by a substantial proportion of its heat output being by way of radiant energy.

Plenum box A component forming an interface between ductwork and one or more air terminal devices. By virtue of its design or by the inclusion of accessories, it can also be used to equalize the pressure/velocity across air terminal devices.

Plenum chamber Any air compartment connected to one or more ducts or to a slot in an air distribution hood.

Plenum system A ventilation system that holds a space at positive pressure.

Plume Effluent discharged from a chimney or exhaust duct, composed of gases alone or gases and particulate matter. The plume shape depends on temperature difference and turbulence. The flow of visible hot gases or vapor from an outlet.

Pneumatic Any device operated by or filled with compressed air or a liquid.

Pneumatic control A control system that operates on compressed air as the operating medium for the control of valves and dampers, etc.

Pneumatic conveying The conveying of dusts, powders, or granular materials in ducts or pipes by means of a difference in air pressure.

Pneumoconiosis A lung disease experienced by miners caused by industrial dust.

Pollen Small particles of the male fertilizing seeds of plant life, which may cause various allergic reactions especially of the respiratory tract, known as hay fever.

Pollutant Any unwanted liquid, solid, or gaseous product, resulting from the activity of man. It can be further divided in the case of air into

- Primary A pollutant that is discharged into the ambient air.
- Secondary A pollutant formed in the air as a result of reactions of primary pollutants.

Polluter pays principle (PPP) The term that relates to either the industry or the individual being responsible for the cost of all pollution-control measures.

Pollution Relating to any environmental constituent present in air or water to such an extent that it presents a hazard to the present or future health of humans or any ecosystem.

Polonium chamber Any air compartment connected to two or more ducts, or a slot in an air distribution hood.

Polychlorinated biphenyls (PCBs) Highly toxic organic compounds used in the electrical industry, use of which is now restricted.

Polydisperse aerosol See Aerosol, polydisperse.

Pore diffusivity The ability of a material to diffuse gas through its pores, trapping the contaminants.

Porosity The presence of spaces within a material that can absorb gases or moisture.

Positive rated operating pressure The tested maximum positive pressure at which the duct is rated.

Potential temperature The temperature an air envelope would acquire if brought adiabatically from its initial or actual pressure to a standard pressure of 1000 mb.

Powder A substance or combination of substances in the form of fine dry particles.

ppb See Parts per billion.

Powered air-purifying respirator (PAPR) A type of air-purifying respirator that utilizes a battery-powered fan to draw contaminated air through the cartridge or filter into the facepiece.

Prandtl tube See Pitot static tube.

Precipitator, electrostatic A device for collecting particulate matter from a gas stream by using electric forces to impart a negative charge to the particulate matter in the gas stream. These charged particulates are attracted to collecting surfaces which have the opposite polarity.

Precipitation When a solid is formed from a solution, or the separation of particles from a fluid by the process of precipitation.

Precursor A substance involved in the formation of new air pollutants, that is, a hydrocarbon is the precursor to the formation of ozone.

Precleaners A device to remove a contaminant from a liquid or gas stream before the main cleaner.

Precision The degree of agreement between independent test results obtained under the same conditions.

Predicted mean vote index (PMV) An index used to predict the mean value of thermal sensation votes of a large group of persons, expressed on a seven-point scale.

Predicted percentage dissatisfied An index that predicts the percentage of a large group of people who are likely to feel thermally dissatisfied, that is, feel either too warm or too cold.

Prefilter A rough filter positioned before a fine filter to reduce clogging of the fine filter.

Preheat coil A heating coil in a ductwork run sized either to temper the air or to stop the filter from freezing.

Pressure The force per unit of area exerted in all directions by a gas or liquid on the walls of its container.

Pressure, atmospheric The pressure of the atmosphere as indicated by a barometer, in kPa.

Pressure blowers Variations of radial-bladed fans with narrow housings and impellers, developing pressures in excess of 25 kPa at low volumes.

Pressure burst A failure in a filter, ductwork, or an air handling unit due to high differential pressure.

Pressure depressions The differences in air pressure between zones necessary to stop contaminant drift.

Pressure drop The static pressure difference due to friction or turbulence between two locations in a ventilation system.

Pressure drop, final The value to which the filtration performance is measured in order to classify the filter.

Pressure drop, final recommended The manufacture recommended operating pressure drop of a filter at rated flow conditions.

Pressure drop, initial The pressure drop obtained on a test on a clean filter.

Pressure factor The test ratio between the suction effect and air velocity passing over a cowl or roof outlet, represented by

$$-\xi = \frac{\Delta p_s}{p_d}$$

Pressure gradient The rate of change in the pressure in a fluid at any given time.

Pressure loss See Pressure drop.

Pressure maintenance Maintenance of specified (differential) pressures in spaces or pipe/duct systems.

Pressure sensor A recording or measuring device that corrects the pressure of a fluid when it varies by a given amount from its design set point.

Pressure, static The potential pressure that is exerted in all directions by a fluid at rest.

Pressure, total The algebraic sum of the velocity pressure and static pressure.

Pressure, velocity The kinetic pressure exerted in the direction of flow that is necessary to cause a fluid at rest to flow at a given velocity.

Pressure ventilation The ventilation of a space by providing air movement from a high-pressure region to a low-pressure region.

Pressure, water vapor partial The pressure that the water vapor in air would exert if it alone occupied the volume of the humid air at the same temperature, in kPa.

Pressurization The process by which a space is held at a pressure greater than the surrounding areas.

Prevailing wind direction The direction from which the wind is most frequent for a given period.

Primary air The air provided to a combustion process to ensure efficient combustion, or the air leaving an outlet, forming a jet.

Primary airflow rate The mass or volume of air entering a supply air terminal device in unit time from an upstream duct or a plenum box. Or the air leaving through an opening and entering a space.

Primary air induction system Air introduced via an air duct to the induction unit, generally in the form of treated outdoor air.

Primary air temperature See Temperature.

Primary calibration standard A calibration standard based on direct measure of a reference value.

Primary collector An air-cleaning device that removes the larger particulate matter before a HEPA filter.

Primary containment The enclosing structure around a red zone used in the atomic energy industry, which has a specific leak tightness.

Priming The carryover of particles of water or other fluid in gas flow, mainly in steam boilers.

Probability A quantitative measure having values between 0 and 1 inclusive.

Process The general term that describes a method of manufacture or control.

Process requirements The conditions, for a given process, including the boiler or cooling.

Process ventilation Ventilation provided primarily for process purposes, the requirements of the occupants being secondary.

Product recovery The recovery of the waste products from a process reducing pollution, saving energy or materials.

Promoter A chemical substance or specific set of conditions that is favorable to, or triggers the development of, cancer.

Propeller anemometer A device used for measuring airflow in which the air velocity revolves a propeller, the shaft of which is connected to a gearbox and measuring dial.

Propeller fan See Fan types.

Proportional bandProportional control A control algorithm in which the final control element moves to a position proportional to the deviation of the value of the controlled variable from the set point.

Proportional integral (PI) control A control algorithm that combines the proportional response and integral response control algorithms.

Proportional integral derivative (PID) control A control algorithm that enhances the PI control algorithm by adding a component that is proportional to the rate of change of the deviation of the controlled variables.

Protective equipment, personal See Personal protective equipment (PPE).

Psychrometric chart A chart constructed so that knowing any two air properties the remaining thermodynamic properties of air at that condition can be determined.

Psychrometric coefficient The coefficient in the equation for the determination of the water vapor partial pressure from the wet bulb depression.

Psychometrics The study of the properties of moist air.

Pulse The regular beating of the blood in the main blood vessel, a pulse of electricity, the vibrating of a mechanical or electrical device.

Pulsed hot wire anemometer A device used for gas flow measurement, similar to the hot grid anemometer, in which measurements are made by pulses of hot air at a downstream sensor.

Pump A device that moves a fluid by mechanical or electrical means. The movement being created by a pressure differential between the inlet and the outlet.

Pumping The act of forcing a fluid along a conduit by overcoming frictional resistance.

Push nozzle A push jet, located either on a working surface or above a process.

Push-pull exhaust See Push-pull hood.

Push-pull hood A protecting hood around a process with the air supply on the one side of the contaminant source and the extract on the other side.

PVC Polyvinyl chloride, a polymeric plastic material.

Pyrolysis The chemical decomposition of a material caused by the application of heat.

Q

Quartile If a set of observations are ranked in order of magnitude, then the quartiles are those three values which divide the observations into four equal parts, that is, the lower quartile is that value below which one quarter of the observations lie, and the upper quartile is that value below which three quarters of the observations lie.

Quenching Heat removal from a solid by means of a fluid, either oils or water.

Quick disconnecting ducts Ductwork provided with a quick-release mechanism for disconnection providing easy access for cleaning the ductwork.

Quick-release access doors Doors located in strategic areas of a plant to allow access for maintenance or cleaning.

R

Radial Arranged like a wheel with lines (spokes) radiating from a center point.

Radial acceleration Rate of velocity change with respect to time in a radial direction.

Radial-bladed fan See Paddle wheel impellers.

Radiant cooling The cooling of a human body or other surface by means of a radiant panel.

Radiant heating The heating of a human body or other surface by means of a radiant panel.

Radiant temperature The temperature of a surface emitting heat to surrounding bodies by means of electromagnetic radiation.

Radiant temperature asymmetry The difference between the plane radiant temperature and the temperatures at the sides of an element.

Radiation Energy provided to a body by electromagnetic waves.

Radiation shape factor The angle factor representing the fraction of the angular field of view from which energy exchange is trading places.

Radiative heat exchange The heat exchange by radiation between the clothing surface, including uncovered skin, and the environment, in W/m^2.

Radiative heat exchange, globe The heat exchange by radiation that takes place from a black globe thermometer, in W/m^2.

Radiative heat transfer coefficient The heat-transfer coefficient wholly attributed to radiative heat transfer.

Radiator A body warmer than its surroundings that emits its heat to the cooler surroundings.

Radioactive materials Elements that have unstable nuclei that spontaneously disintegrate, releasing radiation in the form of subatomic particles and energy.

Radioactivity The property of spontaneous disintegration possessed by certain unstable nuclides.

Radiometer A sensitive instrument for the measurement of heat radiation.

Radiometric forces Weak forces that cause the motion of particulate matter, including diffusiophoresis, thermophoresis, and photophoresis.

Radon A radioactive element, the heaviest of the noble gases, formed by the radioactive decay of radium.

Radon daughters The series of unstable isotopes that are formed as radon atoms undergo radioactive decay.

Rain louver or weather louver A fresh air inlet to a duct designed to reduce the ingress of driving rain.

Random particle motion See Brownian diffusion.

Range The interval defined by the largest and smallest values in a set of observations.

Range hood An extraction hood positioned above a cooking range to provide the best possible capture velocity of the fumes.

Raoult's law States that at equilibrium, the partial pressure of a solute vapor over a liquid mixture is equal to the vapor pressure of the pure solute at the given temperature times the mole fraction of the solute liquid component in the mixture.

Rapping A method used on bag or other filters to dislodge the dust cake from the fabric when the airflow resistance exceeds a certain value.

Rated airflow The flow rate specified by the manufacturer for an item of equipment.

Rating conditions A set of operating conditions over which specified performance will result.

Rating plate The plate fitted to an item of equipment showing the operating characteristics.

Raynaud's syndrome Diminished blood flow or loss of blood to the fingers, caused by vibration, also known as white or dead finger.

Reactive metal A metal that readily enters into a chemical reaction.

Receiving hood An extract hood in the immediate vicinity of a process that extracts the generated pollutants in an effective manner.

Receptor A body (e.g., a person or animal) that absorbs a pollutant from the air, which may suffer from some adverse effects.

Receptor system A system that contaminants enter without inducement.

Recirculated air Exhaust air returned as supply air to the air treatment system from which it is originated.

Recirculation The process of returning air to the space from which it has been removed, with or without treatment.

Recovery The collection of the products of a process for reuse.

Recycling The reuse of scrap material for pollution control and conservation purposes.

Red zone The primary air containment zone in ventilating systems in the atomic energy industry.

Re-entrainment The induction of air that has been discharged back into a jet, at a position upstream from the discharge.

Reference method A method of analysis to which other methods of analysis are compared.

Reference period A specified time period allowed for human exposure to a specific concentration of a biological agent or chemical.

Refractories A special heat-resistant heatretaining brick used in furnaces, chimneys, and boilers.

Reflectance A measure of the extent to which a surface is capable of reflecting radiation, defined as the ratio of the intensity of reflected radiant flux to the intensity of the incident flux.

Refrigeration The controlled removal of heat from a substance.

Regenerable adsorbers See Adsorbent, regenerable.

Regenerative blowers A fan that generates pressure by centrifugal force, in which, in contrast to a centrifugal fan, the airflow is around the circumference of the impeller.

Regenerative thermal oxidizers A series of beds made from heat-resistant material which alternately store heat from the combustion chamber exhaust gases and release heat into the cooler gases entering the combustion chamber.

Register A combined grille and damper assembly.

Regression A statistical method used to investigate the dependence of one variable on one or more other variables.

Regulations Any legislation or statutes enforced by law.

Relative humidity The ratio of the partial pressure of the water vapor in moist air at a given temperature to the partial pressure of water vapor in saturated air at the same temperature.

Relative ventilation efficiency A quantity describing how the collection efficiency of a ventilation system varies between different parts of an enclosure.

Relaxation time The time necessary for a moving particle to adjust from one given steady-state velocity to another, for example, the time for a falling particle to reach its terminal velocity. It is independent of the nature of the force applied to the particle.

Rem (Röntgen equivalent man) The unit of measure of the radiation dose to the internal tissues.

Replacement The process of air entering a space to fill the void created by either natural or mechanical extraction.

Replacement air Air supplied to a space to replace the air removed by a combustion process or by natural or mechanical ventilation.

Reproducibility The extent to which any measurements taken during tests performed under the same conditions will provide statistically similar results.

Residence time The time a pollutant remains in a space after it is released.

Resistance The opposition to flow caused by friction as air passes through a ductwork or pipework system.

Resistivity A property of a material equal to the reciprocal of its conductivity.

Resistivity of dust The electrical resistivity of a dust, which is one of the factors that influences the practical efficiency of an electrostatic precipitator.

Resonant frequency The sound frequency for which a particular system provides the maximum absorption. The amount of sound absorption in a system depends on the degree of damping achieved; this depends on the mass and the associated air space.

Respirable particles Particulate matter of such a size that it can pass through the body defences and into the lungs, where, depending on its nature, it will either deposit itself or be exhaled.

Respiratory protection The use of face masks, respirators, or separate air supply to reduce the intake of pollutants into the lungs.

Response A reaction of a living organism caused by a pollutant. Also, the reaction time of a measuring instrument or control device to perform an action.

Respirable particulates Particulates in the size range that can pass through the defence mechanisms in the human body and enter the lungs during inhalation.

Resultant temperature See under Temperature.

Return air Air that has been removed from a space that is returned to the space either mechanically or naturally with or without treatment.

Reverberation The continuation and enhancement of a sound caused by rapid multiple reflections between the surrounding surfaces.

Reverberation time The time required for a sound to fall to a given level in an enclosure.

Reverse air cleaning filters Filters that become self-cleaning by dislodging the impacted dust when the gas flow is reversed.

Reverse pulse The use of jets of high-pressure air to dislodge material from the exterior of a bag filter.

Reversible heat engine A heat engine, which will convert a certain quantity of heat into an amount of work (W), that will produce the original quantity of heat if the same amount of work is expended in driving the engine backwards.

Reversible process A process that may be performed in the opposite direction to return to the initial state of the working fluid.

Reynolds number A dimensionless parameter that represents the ratio of the inertia forces to the viscous forces in a flow. Its magnitude denotes the actual flow regime, such as streamline (laminar), transitional, or turbulent.

Ringelmann chart A method for the visual comparison of smoke from a chimney. The estimation is made by the comparison of the shade of smoke against shade cards.

Riser A vertical section of duct or pipe in a distribution system.

Risk assessment The sequence of events necessary to ensure that a system is designed to provide the safest possible working arrangement.

Röntgen The amount of X-ray or gamma radiation that produces one unit of charge in 1 cc of dry air.

Roof ventilator A natural or mechanical unit positioned in the roof to provide air extraction from the space.

Room air conditioner A package air-conditioning unit for the air treatment of the space in which it is located.

Room air conditioner (self-contained) A room air conditioner complete with a direct expansion (dx) system condenser and evaporator fans, filtration, and thermostatic control.

Room air distribution system The method by which air is distributed within a space.

Rotating anemometer An instrument used to measure gas flow that depends on the rotation of vanes mounted on a spindle.

Rotation The angular displacement of a body about a specified axis in a specified direction in unit time.

Rotor The rotating portion of an electric motor; the stator is the stationary part.

Route of entry Path by which toxins and other substances may enter the human body. These include inhalation, ingestion, and absorption through the skin. Less common routes include injection and absorption through moist surfaces surrounding the eyes and ear canal.

Runaround coils The coils in a heat recovery system that collect heat from the hot waste steam and supply it to the cold incoming stream via heat exchangers.

S

Safety cabinets A protective air enclosure (workstation) used in microbiological laboratories.

Safety factor An uncertainty factor that is used in combination with the no-adverse-effect level data to estimate the safe human dose.

Safety guards Devices to protect operators from moving machine parts.

Sample badge A small clip-on device that contains solid sorbent and is used for the collection of a variety of airborne materials. It is a passive sampler that is typically clipped to the worker's lapel and worn throughout a shift.

Sample bag A bag made from an inert polymer such as Teflon, complete with a fitting for connecting to an air-sampling pump.

Sampling The action of selecting a set of items for measurement of a given parameter.

Sampling and analytical error (SAE) A numerical factor used in analytical methods to account for uncontrollable errors. Its value is taken into consideration in the determination of whether the exposures are within acceptable limits.

Sampling box An enclosed chamber in which sampling is carried out or samples are protected after collection for dispatch to the testing laboratory.

Sampling duration The period of time that sampling takes place at a specified sampling rate.

Sampling medium The device or material through which contaminated fluids are drawn in order to collect the contaminants for analysis.

Sampling train The assembly of sample medium in its holder, with connecting tubing and sample pump.

Sampling volume flow rate The induced flow rate into a sampling system.

Saturated steam Steam that has the same temperature and pressure as the water from which it is formed.

Saturated vapor A vapor that can exist in equilibrium with its liquid.

Saturated vapor pressure The pressure exerted by a saturated vapor. This pressure is a function of the temperature.

Saturation efficiency A measure of the performance of an air washer. It is the amount of water added to the air leaving the washer expressed as a percentage of the amount of water that would have been added if the air had left the washer in a fully saturated condition.

Saturation tables Tables that relate the dryness properties of a gas to its temperature and pressure.

Saturation vapor density The density of a saturated vapor.

Sauter mean diameter The average ratio of the volume to the surface area used in the determination of the pressure drop in a scrubber.

Scales of sensation A simple numerical scale used to report the response of a person to temperature, humidity, air velocity, air purity, noise, light, taste, etc.

Scan test A test used to determine the local efficiency of an air filter.

Scatter diagram A graph in which values of one variable are plotted against the corresponding values of another property.

Scavenging The removal of an unwanted product.

Screening The separation of particulate matter by the use of filters, mesh screens, or other devices. Also, the covering of a plant item to improve the visual impact.

Screw conveyor A conveyor that uses an Archimedes screw to convey granular material from a hopper to point of use.

Scroll collectors A centrifugal collector in which a scroll imparts a centrifugal motion to the dust stream, concentrating the dust in the peripheral layer, from where it is passed to a secondary collector, and then to exhaust.

Scrubbers and absorbers Wet systems used for the removal of aerosols and other gaseous pollutants from an airstream.

Scrubbing The process of cleaning contaminated gas by passing it through a water spray or cascade.

Seals A device on a container or conduit run that ensures that the internal products do not escape from any joints.

Secondary air Air that is introduced above a combustion process in addition to the primary air in an attempt to obtain complete combustion.

Secondary containment The enclosing structure around a green zone in the atomic energy industry.

Secondary filter A filter to provide final cleaning of the air after the main filter in a system.

Sedimentation The process of settling solid particulates out of suspension in a fluid or a gas.

Selectivity The degree of independence from interference.

Sensible heat The portion of heat supplied to a substance that produces a change in temperature without changing the state of the substance.

Sensing element A component that measures of the value of a variable and provides the input for control devices.

Sensitization The development of an adverse immune response following more than one exposure to a substance.

Sensitizer A substance that stimulates a response from the immune system.

Sensitivity The ability of a chemical analysis to detect low levels of the analyte. Also, any abnormal reaction of the human body to chemical substances.

Sensor A device used to measure flow, temperature, pressure, or another property of a medium.

Sensory hearing loss Irreversible hearing loss resulting from damage to the inner ear tissue that translates sound pressure into nerve impulses.

Series operation The connection of two fans or pumps in succession in order to increase the available pressure in a system.

Serpentine An asbestos mineral with a wavy appearance, such as chrysotile.

Set point The value on the scale of a controller at which the design conditions are set.

Settling The process of particulate matter falling out of a gas stream or a fluid.

Settling chamber A chamber in which large particulate matter settles out of the air due to gravity. By increasing the cross-sectional area of this chamber the air velocity decreases, allowing settling to take place.

Settling velocity The velocity that has to be attained to ensure that particles of a particular size settle at a given distance from the generation source due to the influence of gravity.

Shaver's disease A disease of the lungs found in workers exposed to fumes or dusts containing aluminum oxide. It is a type of pneumoconiosis and results in interstitial fibrosis and decreased lung function.

Shedding The loss of fibers from a filter.

Shivering The process of metabolic regulation against cold by muscular action.

Shock losses The energy loss in a moving fluid stream due to one or more of the following:

- Violent mixing producing eddy formation.
- Separation occurring due to contractions or expansions in duct flow.

Shock ventilation The purging of a space prior to or after a process or occupancy.

Shock wave A pressure wave resulting from the rapid closure of a valve or damper in a pipeline or ductwork system, or from an explosion.

Short circuiting The process in which air from an inlet is extracted from the space before it has chance to mix, caused by the extract being positioned too near the input grille.

Shot blasting room A ventilated room devoted to the cleaning of castings by the use of various grades of shot.

Short term exposure limit (STEL) The permissible exposure limit for a given contaminant averaged out over any 10-minute period during an 8-hour working shift. See also Occupational exposure limit (OEL) and Long-term exposure limit (LTEL).

Sick building syndrome (SBS) The group of symptoms related to poor air quality, including headaches, irritation of mucous membranes of the eyes and nose, breathing difficulties, etc., experienced by occupants in poorly ventilated buildings. See also Building-related illness (BRI).

Siderosis A benign reddish discoloration resulting from deposits of iron oxide in the lungs.

Sieving The use of sieves for the collection of particulate matter or for the grading by size of particulate matter for classification purposes.

Significance A statistical term relating to tests made to ascertain the probability of an effect or correlation.

Silicates Chemicals used as adsorbents.

Silicosis Pneumoconiosis resulting from inhalation of crystalline silica (quartz).

Simple asphyxiant Substance that displaces air, producing an oxygen-deficient atmosphere.

Sink A storage device for a fluid or heat.

Sintered filters A self-supporting sintered element used in gas cleaning systems.

Size-selective sampling Industrial hygiene sampling methods that collect particles with a specific range of aerodynamic diameters.

Skew distribution Any set of values measured during a test that is not symmetrically distributed.

Skin notation The word skin included as part of an exposure limit. It is used for those substances for which absorption through the skin is considered to be a significant route of entry into the body.

Slant gauge An inclined calibrated manometer tube.

Sling psychrometer An instrument used to measure the dry-bulb and wet-bulb temperatures of the air, from which the humidity of the air can be determined by means of calculations, tables, or charts.

Slip (Cunningham factor) A factor used in particle physics to predict the behavior of small particles.

Slot venturi A device used to adjust the pressure drop in a scrubber.

Sludge The waste product formed by a wet cleaning process.

Smell See Odor.

Smog A mixture of smoke and fog that arises from nitrogen oxides and hydrocarbons and the photochemical action of sunlight.

Smoke Aerosols formed from minute solid or liquid particles, mostly less than 1 μm in diameter, generated by the incomplete combustion of a fuel or by sublimation.

Smoke bomb A firework type of device that produces smoke used for the observation of airflow within a space.

Smoke damper A damper installed in a ductwork system designed to close in the case of fire by means of a fusible link and electrical or magnetic device to stop the spread of smoke.

Smoke extraction A mechanical or natural means by which smoke generated during a process or a fire is removed from the space to outdoors.

Smoke generator A device that electrically heats oil-producing smoke. The smoke is liberated from a nozzle by either thermal forces or by means of a fan and used to observe airflow patterns within a space or to observe leakage from ductwork, etc.

Smoke stain When a certain quantity of dirty air is passed through a filter paper, the degree of staining on the paper is measured and expressed as a concentration of equivalent standard smoke by means of an optical reflectometer.

Smoke tube A tube or a canister containing a smoke-generating chemical used to observe air movements within a room.

Smuts Unburned carbon emitted from chimneys. If sulfur is present in the fuel, these smuts will be acidic.

Sociocusis Hearing loss that results from exposure to the noises of everyday life.

Sodium flame test A test of HEPA filter efficiency using small particles generated from NaCl.

Solid phase The condition of a body being a solid, such as ice.

Soiling index The degree of soiling of a filter paper fitted in a sampling device through which a contaminated gas has been passed.

Soot The aggregates leaving a combustion chamber due to incomplete combustion of a carbonaceous fuel.

Sorbent Any agent that is used in a sorption process.

Sorbent tubes Small glass tubes that contain sampling media such as silica gel or activated charcoal.

Sorption Adsorption (a surface process) or absorption (a volume process).

Sound A physiological sensation received by the ear, which may or may not cause annoyance.

Sound intensity The sound power distributed over unit area, in units of W/m^2.

Sound power The rate at which sound energy is produced at the source, given in W.

Sound pressure The average variation in atmospheric pressure caused by a sound, given in Pa.

Sound rating The manufacturer's rating of the noise level produced by an item of rotating or reciprocating plant.

Source Where any form of pollution is generated.

Source sampling The testing and measurement of an emission at its point of generation.

Space requirements The footprint (m^2) or volume for a given item of equipment.

Sparking The production of sparks; occurs under certain conditions on electrostatic precipitators.

Speciation The process of determining the different species present in a chemical agent.

Species The different forms in which a biological or chemical agent may be present.

Specific flow The volumetric air change rate within a space, denoted by n, the flow volume rate into and out of the space divided by the volume of the space.

Specific gravity The weight of a material in kg that would occupy one cubic meter under a definite state of conditions.

Specific heat The amount of heat (or mechanical work) required to raise the temperature of a unit mass of a substance 1°C. In the case of gases, there are two specific heats, according to whether the heating takes place at constant pressure or at constant volume.

Specific humiditySpecific leakage Relating to the leakage that takes place from an enclosure due to a pressure difference created either mechanically or naturally. It is the ratio of leakage area in m^2 divided by the floor area in m^2.

Specific measuring range The range of concentration values for which the overall uncertainty of a measurement procedure is intended to lie within specific limits.

Spectrophotometry A photometer for comparing two light radiations, wavelength by wavelength.

Spirometry A test method used to evaluate lung function that measures volume of exhaled air passing through a tube during a given time.

Splitters Turning vanes inserted in a fluid flow system to reduce frictional losses.

Spontaneous combustion Any combustion that takes place without a source of ignition due to the nature of the material and its packing arrangements.

Spread The angle of divergence of a jet or plume from its origin.

Squirrel cage fan A fan with a squirrel cage rotor, which has the advantages of low cost, low maintenance, and sturdiness.

Stability The state of being stable.

Stable Relating to a chemical that is not readily decomposed, or a stable condition in a working environment.

Stack effect The pressure forces set up in space due to the density differences between the hot and cold columns.

Stack solids The solid content of a gas stream leaving a chimney or exhaust duct.

Stagnation A situation in which flow is assumed not to occur.

Standby fan A fan or fans included in a system to provide a backup in the case of failure, and/or to provide a rapid air change in the case of a chemical spill.

Standard The maximum level of an air contaminant allowed in workplace or external air as defined by a legal authority. Or any national or international standard relating to a product or code of practice.

Standard air Air at standard temperature and pressure (STP), which is the same as normal temperature and pressure (NTP) 0°C and 101.325 kPa, with the force due to gravity 9.80665 N/kg.

Standard conditions Relating to normal conditions such as standard temperature and pressure (STP), which is the same as normal temperature and pressure (NTP).

Standard deviation A statistical measure of the scatter of a series of numbers or measurements about their mean value.

Standard temperature and pressure (STP) This is the same as normal temperature and pressure (NTP), 0°C and 101.325 kPa.

Standard threshold shift (STS) An increase of 10 dB or more in a person's HTL in the 2000–4000 Hz range, significant because it represents a loss of a significant proportion of hearing.

Static efficiency The ratio of fan static pressure to fan total pressure.

Static electricity Phenomena associated with electric charges at rest, due purely to the electrostatic field produced by the charge.

Static head The difference between the total fluid pressure and the dynamic pressure.

Static pressure curve A graphical representation of the static pressure and volume flow of a fan at a set speed.

Static pressure regain The increase in static pressure that takes place due to a decrease in the air velocity causing the velocity pressure to be converted into static pressure.

Static sample The result of the process of static sampling.

Static sampler A device not attached to a person that samples air in a particular location.

Static sampling The use of a static sampler to determine a particular property.

Stator The stationary portion of an electric motor.

Steam Water in gaseous state above its boiling point.

Steam tables Tables containing the thermodynamic properties of steam over a range of pressures and superheat.

STEL See Short-term exposure limit.

Step control A control method in which a multiple switch assembly sequentially switches on or off various stages of a device, such as a heater battery.

Stephan flow The flow of molecules toward or away from the surface of a volatile liquid due to either evaporation or condensation.

Stokes diameter The equivalent spherical diameter of the particle being considered.

Stokes law This relates to the factors that control the passage of a spherical particle through a fluid. The Stokes diameter of a particle is the diameter of a sphere of unit density, which would move in a fluid in a similar manner to the particle in question, which may not be spherical.

Stoichiometric The exact quantity of reactants required to completely react according to a particular chemical equation. If the reaction were complete, only products and no reactants would remain.

Stopping distance The maximum distance a moving particle will travel in still air after all the external forces are removed. In the Stokes region, it is the velocity of the particle times the relaxation time.

STP See Standard temperature and pressure.

Straighteners Vanes fitted in ductwork or air handling units before or after a change in section to produce a reduction in pressure drop.

Stuffing box A box that provides a seal for a fan shaft.

Subcutaneous The deepest layer of skin, containing fatty and connective tissue that provides a cushion and insulative base for the skin and also binds the skin to the underlying tissues.

Superheated steam Steam that has a temperature above that corresponding to boiling temperature, corresponding to the pressure at which it exists.

Superheated vapor See Superheated Steam.

Supersaturation An unstable condition in which the concentration of a solution or a vapor is greater than that corresponding to saturation.

Supply air Treated or untreated air entering the space. For the purpose of drawings, it is color-coded to show the various thermodynamic treatments.

Number of thermodynamic treatments	Color code for drawings
None	Green
1	Red
2 or 3	Blue
4	Violet

Supply air (SUP) classification

Category	Description
SUP 1	Supply air containing only outdoor air
SUP 2	Supply air containing a mixture of outdoor and recirculated air

Supply system An arrangement that provides the distribution of a fluid, vapor, gas, electricity, or another medium.

Surface contaminant Any contaminant that adheres to a surface in a clean room.

Surface tension A characteristic of a liquid surface, with effects at liquid–gas or liquid–liquid interfaces.

Suspended matter (particulates) Particles that remain in suspension in a gas or a fluid for a sufficient time in to be detected by physical means.

Sutherland's equation An equation that allows the effect of temperature on the viscosity of a gas to be determined.

Swirl nozzles Nozzles used to distribute primary air into a space by creating a swirl. This arrangement provides good entrainment and mixing of the air.

Synergism The phenomenon in which the effect produced by two causes together is greater than the sum of the effects that would be produced by the causes separately.

System curves The graphical representation of the resistance (static pressure) that occurs in a ventilation or pump system at different flow rates.

System effect The effect that system components and/or the room have on the air quantity and pressure delivered to the space.

System life The duration of the life cycle of an item of equipment or a complete system.

System specification The engineering specification produced by the manufacturer of the equipment, or by the system designer for the plant as a whole, stating what the system is capable of achieving.

System toxin A substance that affects target organs or entire organ systems.

T

Table exhaust Mechanical extractions of pollutants generated on a worktable during a process. The extraction takes place through a perforated workbench or from the sides or back of the table.

Tachometer An instrument used to determine the speed of rotation of a shaft, normally in revolutions per second (rps).

Tackifier A substance applied to a particulate collection device to increase its efficiency in dust retention.

Take-off Any pipework or ductwork branch taken from a main run.

Target A desirable air quality or temperature to aim for. Also items affected by pollutants.

Tangential acceleration Acceleration of a fluid tangentially to a vane or impeller due to rotary motion.

Target level The predetermined concentration of a dominant contaminant to be achieved by air technology using chemical or other control methods. It may relate to an entire room volume or a building zone.

Target organ A specific organ where the toxic effect of a substance is manifested.

Teflon filter A chemical-resistant hydrophobic filter composed of polytetraflouroethylene (PTFE) used for industrial hygiene sampling.

Telemetering Signal transmission from a measuring instrument by telephone or radio to a distant point for recording or display.

Temperature

Temperature The degree of molecular activity in a body; high activity gives a high temperature, low activity a low temperature. The degree of activity is based on the assumption that absolute zero has no molecular movement at all. The following are some specific temperatures:

- Absolute: The temperature relative to absolute zero, expressed in K. Also called thermodynamic temperature.
- Asymmetry (radiant): The difference between the plane radiant temperatures of the two opposite sides of a small plane element.
- Dew point: The temperature of a mixture of air and water vapor at which further cooling or the addition of more water vapor will cause moisture to condense from the air.
- Difference (vertical air): The difference in air temperature measured at 1.1 and 0.1 m above the floor.
- Differential in occupied zone: The largest value of the difference between the measured air temperatures in the occupied zone.
- Dry bulb: The air temperature recorded by a dry bulb thermometer, a sensory device excluding any effects of moisture or radiation.
- Effective: See Operative
- Environmental: The sum of two-thirds of the mean radiant temperature and one-third of the air temperature.
- Equivalent: The temperature as recorded by an Eupatheoscope, taking into consideration air temperature, air velocity, and thermal radiation.
- Globe: The temperature recorded by a 100 mm black bulb (globe) thermometer.
- Gradient risk: The percentage of people predicted to be dissatisfied due to a difference in air temperature between the ankle and the head.
- Induced: The temperature of the internally induced air flow.
- Jet: The leaving temperature of a jet from an opening or the average jet temperature at a given cross-sectional area of a jet.
- Mean in occupied zone: The arithmetic average of the measured values of air temperature in the occupied zone.
- Mean radiant: The theoretical uniform surface temperature of an enclosure in which an occupant would exchange the same amount of radiant heat as in the actual nonuniform enclosure.
- Operative: The theoretical uniform temperature of an enclosure in which an occupant would exchange the same amount of heat by radiation and convection as in the actual nonuniform space.
- Optimum operative: The temperature that satisfies the greatest possible number of people at a given clothing and activity level.
- Plane radiant: The uniform temperature of an enclosure where the radiance on the one side of a small plane element is the same as in the nonuniform actual environment.
- Primary; The temperature of the primary airflow.
- Primary difference: The difference between the primary air temperature and the reference air temperature in the reference zone.

- Resultant: The temperature recorded by a thermometer in the center of a 100 mm diameter blackened globe, corrections for the air velocity.
- Resultant temperature: For air velocities less than 0.1 m/s, it is given by

$$\theta_{res} = 0.5\ \theta_r + 0.5\ \theta_a$$

and if the air velocity is above 0.1 m/s,

$$\theta_{res} = \frac{\theta_r + \theta_a\sqrt{10v}}{1 + \sqrt{10v}}$$

where θ_{res} = dry resultant temperature, θ_r = mean radiant temperature, θ_a = air temperature, and v = velocity in m/s.

- Room reference: The average of at least five measurements of the air temperature at a height of 1.1 m from the floor and outside the area directly influenced by the device.
- Thermodynamic: A Kelvin is 1/273.16 of the triple point of water.
- Total: The air temperature of a total flow from an ATD.
- Wet Bulb: The temperature as recorded by a thermometer bulb covered by a wet wick.

Tempering The process of heating or cooling make-up air being supplied to a space in order to provide the required temperature limits.

Temporary threshold shift (TTS) A temporary shift in the hearing threshold level that goes away after the person has been in a quiet environment for a few hours. It is confirmed by an audiogram retest following a suspected standard threshold shift (STS) after at least 14 hours away from high levels of noise.

Temporary variance An OSHA variance issued to an employer who is unable to comply with a standard by its effective date for reasons beyond their control. The employer must demonstrate a plan for coming into compliance within a period not to exceed 1 year and provide all available measures to protect the employees in the interim.

Teratogens Toxins that cause abnormal development or birth defects.

Terminology Relating to current word usage of technical terms.

Terminal Falling Velocity See Terminal settling velocity.

Terminal Settling Velocity The maximum velocity attained by falling particulate matter, at which the gravitational force is balanced by the viscous drag created by the fluid in which it is falling. Terminal free-fall velocity of a sphere moving through a fluid, derived by using Stokes' law, Newton's drag coefficient, and Reynolds number, is proportional to the radius of the sphere squared and to the density, and is inversely proportional to the fluid viscosity.

Test aerosols Aerosols for testing the efficiency of filters generated from various grades of dusts.

Test airflow rate The rated airflow in equipment testing.

Test dust Various grades of dust used to test the collection efficiency of filters.

Test procedures The arrangements of the various sequences of events necessary for carrying out a test on a system during commissioning or for any other testing.

Test volume flow rate See Test airflow rate.

Theoretical air quantity The stoichiometric quantity of air required for complete combustion of a given quantity of a specific fuel.

Thermal anemometer An anemometer that employs the principle that the quantity of heat removed by a gas stream passing a heated element has a direct relationship to the velocity of the gas stream.

Thermal coagulation The process by which Brownian movement causes particulate matter to collide and adhere.

Thermal comfort That state in which the human body is in a state of thermal equilibrium, also called thermal neutrality.

Thermal currents Natural convection currents set up in a fluid due to density differences.

Thermal discomfort Discomfort experienced due to excessive heat loss or gain from or to the human body due to radiation, convection, conduction, evaporation, or air movement.

Thermal ignition sources A source that will cause the ignition of a flammable gas, vapor, or dust, such as an electric spark, flame, or hot surface.

Thermal load The heating or cooling requirements of a space due to structural gains or losses and the air infiltration and mechanical ventilation. The load produced by a process in a working environment.

Thermal oxidizers Devices used for the destruction of hazardous or toxic gases by oxidation at elevated gas temperature, producing primarily carbon dioxide and water.

Thermal pollution The effect of the emission of high emission temperature waste products into the environment, creating a significant temperature change.

Thermal updraft The air movement that is created by a thermal plume.

Thermocouple An instrument for the measurement of temperature consisting of two wires of different metals joined at each end. An electrical electromotive force is generated, the magnitude of which allows the temperature to be measured.

Thermodynamics The branch of engineering that deals with the relationship between heat, power, and gases.

Thermograph A device for measuring and recording air temperature.

Thermography The use of a tube or IR film to determine surface temperatures.

Thermohydrograph A mechanical or electrical device that records simultaneously the relative humidity and temperature of the air throughout the day.

Thermometer A device used to determine the temperature of a medium.

Thermometer anemometer An instrument that measures the heat removed by an air stream passing over a heated element or bulb to determine the air velocity. See Kata thermometer.

Thermophoresis The motion of particulate matter in the direction of a cooler gas or surface due to the hot side of the particles exerting more force than the cool side.

Thermostat An instrument used to detect temperature changes and provide corrective output.

Thoracic fraction Particles with aerodynamic diameters of 5–10 μm, which enter the lungs but not the alveoli.

Threshold A level or dose of a pollutant below which it is assumed to have no effect on life.

Threshold dose A dosage or exposure level below which the adverse effects of a substance are not realized or expressed by the exposed population.

Threshold limit concentration (TLC) The concentration of an air pollutant allowed in the work space.

Threshold limit value (TLV) The limits of airborne concentration of chemical substances that are allowed in workplaces published by the American Conference of Governmental and Industrial Hygienists (ACGIH). Also known as MAC.

Threshold limit value-ceiling (TLV-C) The concentration that should not be exceeded during any part of the working exposure.

Throttling The expansion of a fluid through a constricted passage (across which there is a pressure difference), during which no external work is done. The initial and final velocities of the fluid are equal, and there is no heat exchange with external sources. A change in entropy will, however, take place.

Throttling range In a proportional controller, the control point range through which the controlled variable must pass to move the final control element through its full operating range.

Through ventilation Ventilation that takes place through a space due to wind forces, normally resulting in poor mixing of the room air due to short circuiting.

Throw The distance a fluid stream travels on leaving an outlet before its velocity is reduced to a specific value.

Tidal volume The volume of gas inhaled or exhaled during each cycle of breathing.

Tight building syndrome See Sick building syndrome (SBS).

Time constant The time required for a dynamic component, such as a sensor, to reach 63.2% of the total response to a change in its input.

Time-weighted average (TWA) exposure The average exposure to contaminant that an operator is exposed to over a work period.

Tinnitus The perception of high-pitched noises in the ears, such as ringing, roaring, or hissing caused by a bodily condition rather than an eternal source.

Tip speed The velocity of the tip of a fan or pump impeller.

Titration An analytical method involving the quantitative addition of reagents to a solution until an endpoint is reached as indicated by a color change or a precipitate.

TLC See Capacity, total lung or threshold limit concentration.

Tolerance The ability of a person to withstand adverse conditions of air quality, infectious agents, noise, vibration, or light without showing signs of infection or disease.

Total airborne particles All the particles surrounded by air in a given volume of air.

Total efficiency The ratio of the power added to the airstream to the power put into the fan or other device at the shaft.

Total energy The sum of the internal energy, pressure energy, and kinetic energy of a fluid or substance.

Total lung capacity (TLC) See Capacity, total lung.

Total pressure The algebraic sum of the velocity and static pressure, recorded at a set position within the system.

Toxicity The ability of a substance to be poisonous or injurious to living organisms.

Toxicology The study of the body's responses to toxic substances.

Tracer gases Gases used with an instrument to determine the air change rate within a space.

Tracer technique The use of a tracer gas in air for the study of air movement within a space.

Tracheobronchial region The middle region of the respiratory system, comprised of the trachea and the bronchi.

Trajectory The actual path taken by air or particulate matter due to its velocity and density when allowed to enter a space either naturally or mechanically.

Transition A change in a duct from square or rectangular to round or vice versa.

Transitional flow The nature of flow in the zone between laminar and turbulent flow.

Transport velocity The air velocity required in an extract duct conveying dust to ensure that the particulate matter remains in suspension and does not settle.

Traversing The process of moving across a grid line in a duct or on a hood with a Pitot tube in order to determine the velocity or pressure distribution.

Trickle valves Manual or automatic valves or openings that allow a given quantity of air or other gas to pass from one space to another.

Trigger finger An industrial injury caused by a constriction of the tendon characterized by the inability to bend or straighten a finger.

Troubleshooting The sequence of events carried out by a service engineer to determine the reason for faulty operation of a system.

True value A theoretical value that exactly relates a quantity for specific conditions.

Tube-axial fans An axial flow impeller mounted in a tubular housing, which contains the rotational velocity.

Turbulence Fluid motion made up of random eddies as opposed to streamline flow.

Turbulence loss The energy loss that takes place in a ventilation system through air turbulence.

Turbulent ventilated rooms The method of air distribution within a space that ensures the maximum amount of air mixing.

Turn-down ratio The lowest percentage of capacity at which a fluid flow device can be set in order to obtain suitable design flow conditions.

Turning vanes Curved vanes added to ductwork elbows in an attempt to ensure streamline flow and, by so doing, reduce turbulent losses.

Tyndall lamp A parallel light beam projected onto a cloud of dust particles generated from a process to produce scattering of the light, allowing an assessment of the magnitude and path of the cloud.

U

U-Tube manometer An instrument used for measuring pressure differences in a fluid or a gas by means of a U-shaped tube containing a fluid such as mercury or oil.

Ultraviolet (UV) analyzer An instrument using the wavelength of UV light to determine the properties of a gas or vapor.

UV radiation Electromagnetic radiation in the wavelength range of approximately 4×10^{-7} to 5×10^{-9} m, that is, between visible light waves and X-rays.

UV sterilization Sterilization of air or water by means of UV rays.

U value The overall heat-transfer coefficient, in $W/m^2\,^\circ C$.

Ultraclean room A high-efficiency clean room used in surgical operations.

Uniform mixing The mixing of two or more fluids that ensures complete uniformity of the mix.

Unit A quantity or dimension adopted as a standard of measurement.

Unit collector A particulate-collection device that is self-contained with fans, filters, etc.

Unidirectional flow See Laminar flow.

Unidirectional jet Airflow in one direction only. It is essential that the temperature differential between the jet air and the ambient air is small and the leaving air velocity is high.

Unstable Relating to a fluid in a state that is not stable. See Stable.

Upstream The location of a fluid before entering a process.

Upper confidence limit (UCL) A statistical procedure used to estimate whether the true value is higher than the measured value.

Upper explosive limit (UEL) The highest concentration of a substance in air that will explode if ignited. See also Lower explosive limit (LEL).

Upper flammable limit The highest concentrations of a substance in air that will sustain combustion.

V

V belts Belts of a V shape used to convey power from a motor pulley to one on a fan or pump shaft.

Vacuum cleaning The cleaning of a product or space by using suction.

Vacuum gauge A gauge used to measure the pressure depression below atmospheric pressure.

Validation The process of evaluating the performance of a measuring procedure and checking that the performance meets certain preset criteria.

Valve A manual or automatic device for controlling the flow of a fluid or a gas, which may provide either complete isolation or modulated flow.

Van der Waals forces Relatively weak molecular forces that attract together atoms or molecules of almost all organic solids and liquids. They assist in the arresting of low-inertia particles in a fiber filter medium.

Vane anemometer A device for measuring the low range of air speeds. It is simply a windmill constructed of a number of lightweight vanes attached to a spindle. Gearing from the spindle converts the vane rotation to a pointer on a dial, from which the air speed is determined.

Vapor The gaseous form of substances that are either solid or liquid at ambient temperatures. Increasing the pressure without altering the temperature can liquefy it.

Vapor barrier Any impervious layer applied to the surfaces of a structure, plant, or duct run that reduces moisture migration. It is normally applied to thermal insulation.

Vapor pressure The pressure exerted by the high-energy molecules in a liquid. The amount the liquid leaving the liquid surface (evaporation) depends on the opposing pressure above the surface.

Variable air volume (VAV) A method of ventilation that maintains the space air temperature by either increasing or decreasing the airflow rate, as opposed to varying the supply air temperature.

Variable speed A device that allows an increase or decrease in the rotational speed of a fan or a pump to achieve the control requirements.

Variance An alternative to an OSHA requirement that ensures that the employer's workplace is as safe as it would be if the employer did comply with the OSHA requirement.

Vasoconstriction The decrease in the diameter of the blood vessels that occurs when the body is losing more heat than it is producing.

Vasodilation The increase in the diameter of the blood vessels that occurs when the body is attempting to lose heat.

Velocity, capture The air velocity necessary to capture pollutants released by a process and draw them effectively into a hood or a duct for safe disposal.

Velocity, face The actual air velocity at the opening plane of an enclosure.

Velocity to prevent back diffusion The minimum air velocity at any point in an air stream necessary to prevent back diffusion of contaminated particles.

Velocity profile The distribution of fluid velocities across a duct or an opening within a duct, hood, or in free flow.

Velometer Any instrument used for measuring air velocity.

Vena contracta The section of a jet in which the streamlines first become parallel after leaving an orifice or a diameter reduction in a duct or hood entry.

Ventilation (1) The process of replacing contaminated air by fresh air. (2) The movement of air within a space. (3)The volume of air entering or leaving the lungs in one respiratory cycle.

Ventilation, dilution See Dilution ventilation.

Ventilation effectiveness The ability of a ventilation system to remove the pollution generated within a space.

Ventilation efficiency Indices that provide a method of assessing the mixing characteristics of incoming air with the room air. It presents a means of determining the pollutant distribution within the space.

Ventilation heat loss or gain The quantity of sensible and latent heat lost or gained from an enclosure due to natural or mechanical ventilation.

Ventilation, local exhaust See Local exhaust ventilation.

Ventilation, mechanical Air movement created in a space by a fan or other air-moving device.

Ventilation, natural Air movement created by wind forces, thermal forces, or a combination of both.

Ventilation rate The actual mechanical or natural air change rate within a space, expressed in L/s or air changes per hour. The supply air may be all fresh air, or a mixture of fresh and recirculated air.

Ventilation strategy The method of planning to ensure that the best method of ventilation is provided to a space.

Ventilation system Any natural or mechanical system that provides some form of ventilation.

Ventilation thermal load The heating or cooling load required to compensate for thermal losses resulting from natural or mechanical ventilation.

Venturi A tube that is constricted and then opens out again, used to either measure the flow of a fluid or to assist the scrubbing of a gas by a liquid.

Venturi meter A measuring instrument used to determine the fluid velocity, achieved by the comparison of pressure differentials across its throat.

Venturi scrubber A scrubber with water velocities of between 60 and 100 m/s or higher, which create shear stresses, breaking up a gaseous air stream to provide effective particulate removal.

Vibration The rapid oscillating movement of a solid body due to an alternating force, for example, a rotating piece of machinery that is out of balance.

Vibration isolator A flexible cloth or plastic connection placed between the source of a vibration, such as a fan housing, and a potential conductor of the vibration, such as ductwork.

Viruses The causative agent of many infectious diseases.

Viscosity The resistance to fluid flow caused by the shear forces between layers in the fluid.

Viscosity, dynamic Sometimes called absolute viscosity, the shear stress in a fluid divided by the velocity gradient.

Viscous flow See Laminar flow.

Viscosity, kinematic The ratio of dynamic viscosity to density.

Visibility The measure of the clearness of air.

Vital capacity (VC) The volume of air that can be taken in and pushed out of the lungs.

Vitiated air Bad or polluted air.

Volatile organic compounds (VOCs) A varied group of pollutants that are liberated from certain synthetic building materials and fabrics. They are assumed to be responsible for some of the aspects of sick building syndrome.

Volumetric analysis The determination of the amount of a particular gas in a mixture of gases, as the percentage of the total volume. See Gravimetric analysis.

Volumetric efficiency The ratio of the net volume flow rate handled by the machine to the volume flow rate handled by the impeller.

Volumetric heat The heat required to raise the temperature of a unit volume of a gas 1°C.

Volumetric flow rate The flow of a fluid expressed in m^3/s or L/s.

Vorticity Rotary motion, such as in a tornado.

Vortex Fluid flow that takes place with rotary motion, such as that observed in the wakes of buildings.

Vortex breakers (1) A device used to straighten out rotary flow in a duct a short distance after a fan. (2) A device found in a cyclone discharge fitted to reduce shell erosion by particulate abrasion.

Vortex gas cleaners A gas pollutant separator that makes full use of a vortex for the separation of particulate matter.

Vortex shedding anemometer A device for measuring air velocity by placing an obstruction in a gas flow and measuring the frequency of vortex is formation downstream of the obstruction.

W

Walk-in fume cupboard A fume cupboard that has the sash opening from the floor of the laboratory.

Walk-through survey An examination or inspection of a workplace involving a review of hazardous materials present and/or used, observation of work practices, and consversations with individuals to identify all of the actual or potential chemical, physical, biological, and ergonomic hazards.

Wall function See Boundary conditions.

Wall ventilator A wall air outlet or inlet with a weatherproof cover to provide air exchange between inside and outside by natural forces.

Warning properties The physical and chemical characteristics of a substance that allow it to be tasted or smelled at unsafe concentration levels.

Warmth The state or quality of the body being warm; it is not necessarily a state of thermal equilibrium.

Washing See Scrubbing.

Washer, air A device for adding moisture to an air stream by means of spray or capillary action. This term should not be used to relate to any wet cleaning process of a gas or other air stream.

Washout Equipment attached to dust collectors, that allows the collected particulate matter to be washed away.

Waste energy Any waste liquid, solid, or gas that contains useful energy after a process.

Waste heat Heat from a process that is surplus to requirement; this heat may be supplied to a heat recovery device for use in another part of the plant or process.

Water chiller A water-cooling device operating either by the direct expansion of a refrigerant by an absorption system or by evaporative cooling.

Water hardness The hardness of a water sample due to the salt content.

Water make-up The extra treated or untreated water required to replace water lost by evaporation or absorption into dust particulates.

Water purification The treatment necessary to ensure that water used for a process or discharged from a process is of an approved standard.

Water raw Water as supplied from a source before any treatment.

Water, softening A chemical process used to change the chemical composition of water by the removal of hardness salts.

Water, vapor Water in the gaseous state that can be liquefied by increasing the pressure without altering the temperature.

Weather strip A purpose-made strip fitted around doors and windows to reduce infiltration.

Weight, arrestance Tests carried out with synthetic dusts to determine the weight of dust collected on a filter during testing.

Weighting A The frequency-selective device on a sound level meter used to measure the A frequency network.

Weighting scale The A, B, or C weighting scales used to approximate the response of the human ear at different ranges of sound pressure levels.

Weld fumes The fine fumes that are produced and liberated into the room air during the welding process.

Wet air filter Any filter that depends on water or another fluid to improve the efficiency of collection of contaminants in a gas stream.

Wet bulb globe temperature (WBGT) A heat stress index that provides information on comfort conditions.

Wet bulb temperature See under Temperature.

Wet centrifugal A dust collector that uses water or other fluid in a centrifugal action in order to improve the particulate collection efficiency.

Wet collector A gas-cleaning device that uses a fluid for the removal of a pollutant from an air or gas stream.

Wet chemical processes A gas-cleaning process that uses certain chemicals to ensure the maximum retention of a given pollutant in the cleaning fluid.

Wetted wall column An experimental apparatus used to determine the mass transfer that takes place through laminar boundary layers.

White zone A ventilation containment zone used in the atomic energy industry.

Wide-open volume The maximum flow volume a fan is capable of delivering with no resistance on the outlet side.

Wind Air motion relative to the earth's surface caused by thermal forces and the earth's rotation.

Windbreak A natural or artificial barrier that protects a building from the prevailing winds.

Wind chill index An empirical scale that correlates well with the sensation of bare dry skin due to the chilling effect of the outdoor air temperature and wind speed.

Wind infiltration Infiltration of outdoor air into a building caused by the pressure difference across the faces of the building.

Wind pressures The resulting positive or negative pressures due to the wind velocity set up on the walls and roof of a structure.

Wind rose A graphical method of showing the direction, velocity, and frequency of wind for a given location over set time periods.

Wind shear The change in the wind velocity and direction with height above some reference plane.

Wind speed The wind velocity measured at a point in open undisturbed country 10 m from the ground. Corrections have to be made for other locations, and for heights above 250 m.

Wind stop A flat plate or a cone fitted over an outlet or inlet duct in order to reduce the possibility of flow reversal due to wind pressure.

Wind tunnel A fan-assisted test rig used to determine the air forces and flow patterns acting on model buildings or components.

Wind vane A revolving instrument used to indicate the direction in which the wind is blowing.

Windward The side of a building or duct opening on which the wind is blowing, also known as upwind.

Work benches A bench on which a process is carried out.

Work field analysis The actual analysis of a given task or the determination of the concentration of pollutants within a work region.

Work region Any area in which an allotted work task is carried out.

Work room A room in which an operator carries out some allotted process.

Work pattern The sequence of activities carried out by the worker over a given work period.

Workplace The defined working area or areas in which work is carried out.

Work platform A platform provided around a plant item for safe maintenance of the unit.

Work space The area or volume of a space in which work on a given process is carried out. The term is also used for the clearance area provided around equipment for maintenance.

Work station Any area in which some aspect of work activity is carried out.

Working level (WL) The allowable level of exposure of a person to an atmosphere that contains any combination of Radon daughters.

Working level month (WLM) An exposure of 1WLM can be taken to be received by a person working in a Radon daughter concentration of 1WL for 170 hours.

Working procedures Set requirements on how an industrial work process is carried out.

X

X-rays Short-wavelength electromagnetic energy that originate outside the nucleus as electrons suddenly move from higher to lower energy levels, giving up energy.

Y

Yield point The point at which material extension is no longer proportional to tension applied. Up to this limit, elastic deformation occurs; after this limit, plastic deformation occurs.

Z

Zeolite Pellets or granules of aluminum silicate, used in water treatment or air-cleaning applications.

Zero count rate The number of counts recorded in unit time by an optical particle counter when a particle-free gas is passed through the measuring chamber.

Zero exposure standards Relating to carcinogens, which have essentially zero allowable exposure levels.

Zone A set area or volume, either indoors or outdoors, in which work may or may not be carried out, depending on pollutant levels.

Zone, capture The area or volume in which a capturing device contains the generated emissions around a process. The capture velocity in this zone must be high enough to ensure the efficient collection of pollutants.

Zone, comfort The area or volume of a space that has its thermal and acoustic environment held at a set standard for the comfort of occupants.

Zone, control The area or volume of a space that has its acoustic, visual, thermal, and air purity conditions controlled to specified levels.

Zone, dead The part of an environment in which the influence of air interchange can be considered negligible.

Zone, exterior Any area or volume outside a building or process.

Zone, interior The most central portion inside a building or process.

Zone, local The area or volume in which the air is controlled locally. The control requirements may be for

- Worker protection and comfort,
- Process control, or
- Product protection.

Zone, main Normally the largest area, often the same as the occupied zone, in which the specified pollutant levels must be maintained.

Zone, multi Any area or volume within a building that may have more than one requirement for the set environmental conditions.

Zone, occupied The zone of a building in which humans or animals are housed.

Zone, perimeter The inside portion of a building or process that is nearest to the outside environment.

Zone, pressure A zone within a building or a process that is held at a given positive, negative, or neutral pressure in order to contain process contaminants.

Zone, protection The area or volume in a working space in which protection of the process or occupants is maintained to set conditions.

Zone, trapped air Any area in the working environment in which the air movement is inadequate to remove the pollutants generated within the space.

Zone, uncontrolled Any area in which the space conditions are not specified or controlled.

Zone, working The volume or area of a building in which a given activity is being carried out.

Zoning The practice of dividing a building into sections for heating and cooling control, in order to ensure that one controller is capable of dealing with the requirements of one zone alone.

International and National Bodies

Lists of CEN and ISO members link to website
https://standards.cen.eu/dyn/www/f?p=CENWEB:5
https://www.iso.org/members.html

AAIH
American Academy of Industrial Hygiene
ABIH
American Board of Industrial Hygiene
ABOK
Association of Engineers in Heating, Ventilation, Air Conditioning, Heat Supply, and Building Thermal Physics (Russia)
ACCA
Air Conditioning Contractors of America
ACGIH
American Conference of Governmental Industrial Hygienists
ADC
Air Diffusion Council (USA)
AGA
American Gas Association
AIHA
American Industrial Hygiene Association
AMCA
Air Movement and Control Association, Inc. (USA)
ASHRAE
American Society of Heating, Refrigeration, and Air Conditioning Engineers
ASTM
American Society for the Testing of Materials
BSRIA
The Building Services Research & Information Association (UK)
CEC
Commission of the European Communities
C Eng.
Chartered Engineer (UK)
CIBSE
Chartered Institute of Building Services Engineers (UK)
COSTIC
Comite Scientifique et Technique des Industries de Chauffage (French Heating Research Organization)
CSTB
Centre Scientifique et Technique des Batiment (French Building Research Organization)
D.
EPA
Environmental Protection Agency (USA)
EU
European Unn
EUROVENT
The European Committee of Air Handling, Air Conditioning and Refrigeration Equipment Manufacturers. See eurovent.eu
Finnish Institute of Occupational Health
HEVAC
Heating, Ventilating and Air Conditioning Manufacturers Association (UK)
HSE
Health and Safety Executive (UK)
IEA
International Energy Agency
IEC
International Electrochemical Commission
I Chem. E
Institute of Chemical Engineers (UK)
EI
Energy Institute (UK)
ILO
International Labor Organization
I Mech. E.
Institute of Mechanical Engineers (UK)

INRS
Institute National de Recherché et de Securie (The French Research Organization dealing with the Prevention of Occupational Accidents and Diseases)
INVENT
The Program covering Industrial Ventilation Financed by Various Organizations in Finland
NAPCA
National American Pollution Control Administration
NBFU
National Board of Fire Underwriters (USA)
NFPA
National Fire Protection Association (USA)
NIOSH
National Institute for Occupational Safety and Health (USA)
NIWL
National Institute of Working Life (Sweden)
NSF
Occupational Safety and Health Administration (USA)
REHVA
Federation of European Heating and Air Conditioning Associations
HASE
The Society of Heating, Air Conditioning and Sanitary Engineers (Japan)
VDI
Verein Deutscher Ingenieure (The Association of German Engineers)
VDMA
German Machinery and Plant Manufacturers Association
WHO
World Health Organization (The United Nations Agency that deals with the Relationship between Human Health and Pollution)
WMO
World Meteorological Organization (The United Nations Agency concerned with the Atmospheric Composition)

Further reading

Some of the following standards are included in the various chapters; however, it is of benefit to include them at this stage.

ISO/TC 205. The Built Environment all parts.

CEN/TC 156. Ventilation national standards and EPA (USA) and HSE EH and other publications (UK). For a comprehensive list of definitions dealing with ventilation se CEN/TC 156 WG 1 document. pR. EN 12792. The latest edition will be published early in 2020.

ISO 6944-2:2009. Fire containment—Elements of building construction—Kitchen extract ducts.

ISO 10121-1:2014. Test method for assessing the performance of gas-phase air cleaning media and devices for general ventilation—Gas-phase air cleaning media.

ISO 10121-2:2013. Test method for assessing the performance of gas-phase air cleaning media and devices for general ventilation—Gas-phase air cleaning devices (GPACD).

ISO 10294-5:2005. Fire resistance tests—Fire dampers for air distribution systems—Intumescent fire dampers.

ISO 15714:2019. Method of evaluating the UV dose to airborne microorganisms transiting in-duct ultraviolet germicidal irradiation devices.

ISO 15858:2016. UV-C Devices—Safety information—Permissible human exposure.

ISO 15957:2015. Test dusts for evaluating air cleaning equipment.

ISO/AWI 15957. Test dusts for evaluating air cleaning equipment.

ISO 16170:2016. In situ test methods for high efficiency filter systems in industrial facilities.

ISO 16890-1:2016. Air filters for general ventilation—Part 1: Technical specifications, requirements, and classification system based upon particulate matter efficiency (ePM).

ISO 16890-2:2016. Air filters for general ventilation—Part 2: Measurement of fractional efficiency and air flow resistance.

ISO 16890-3:2016. Air filters for general ventilation—Part 3: Determination of the gravimetric efficiency and the air flow resistance versus the mass of test dust captured.

ISO 16890-4:2016. Air filters for general ventilation—Part 4: Conditioning method to determinet he minimum fractional test efficiency.

ISO/CD 16890-2. Air filters for general ventilation—Part 2: Measurement of fractional efficiency and air flow resistance.

ISO/CD 16890-4. Air filters for general ventilation—Part 4: Conditioning method to determine the minimum fractional test efficiency.

ISO 16891:2016. Test methods for evaluating degradation of characteristics of cleanable filter media.

ISO 21083-1:2018. Test method to measure the efficiency of air filtration media against spherical nanomaterials—Size ranges from 20 to 500 nm.

ISO/TS 21083-2:2019. Test method to measure the efficiency of air filtration media against spherical nanomaterials—Size ranges from 3 to 30 nm.

ISO/TS 21805:2018. Guidance on design, selection, and installation of vents to safeguard the structural integrity of enclosures protected by gaseous fire-extinguishing systems.

ISO/DIS 22031. Sampling and test method for cleanable filter media taken from filters of systems in operation.

ISO 29462:2013. Field testing of general ventilation filtration devices and systems for in situ removal efficiency by particle size and resistance to airflow.

ISO/CD 29462. Field testing of general ventilation filtration devices and systems for in situ removal efficiency by particle size and resistance to airflow.

ISO 29463-1:2017. High-efficiency filters and filter media for removing particles from air—Part 1: Classification, performance, testing, and marking.

ISO 29463-2:2011. High-efficiency filters and filter media for removing particles in air—Part 2: Aerosol production, measuring equipment and particle-counting statistics.

ISO 29463-3:2011. High-efficiency filters and filter media for removing particles in air—Part 3: Testing flat sheet filter media.

ISO 29463-4:2011. High-efficiency filters and filter media for removing particles in air—Part 4: Test method for determining leakage of filter elements-Scan method.

ISO 29463-5:2011. High-efficiency filters and filter media for removing particles in air—Part 5: Test method for filter elements.

ISO/AWI 29463-5. ISO 29463-5 High-efficiency filters and filter media for removing particles in air—Test method for filter elements.

ISO 29464:2017. Cleaning of air and other gases—Terminology.

EN 779. Particulate air filters for general ventilation—Determination of the filtration performance.

EN ISO 5801. Fans—Performance testing using standardized airways (ISO 5801).

prEN 17166. Fans—Procedures and methods to determine the energy efficiency for the electrical input power range of 125 W up to 500 kW.

EN ISO 12759. Fans—Efficiency classification for fans (ISO 12759).

EN 1886. Ventilation for buildings—Air handling units—Mechanical performance.

EN ISO 5801. Fans —Performance testing using standardized airways (ISO 5801)

EN ISO 13349. Fans — Vocabulary and definitions of categories (ISO 13349).

ISO 13347 (all parts). Industrial fans—Determination of fan sound power levels under standardized laboratory conditions.

The following list of ISO/TC 80000 parts is an ideal reference for some of the terminology of the industrial ventilation engineering field.

Part and date	Title	Replaces	Status
ISO 80000-1 2009	General	ISO 31-10 IEC 6027-1 and IEC 60037-3	Under review
ISO 80000-22019	Mathematics	ISO 31-11, IEC 60027-1	Published
ISO 80000-4 2019	Mechanics	ISO 31-3	Published
ISO 80000-5 2019	Thermodynamics	ISO 31-4	Published
IEC 80000-6.2008	Electromagnetism	SO 31-5	Under review
ISO 80000-7.2019	Light and radiation	ISO 31-6	Published
ISO 80000-8.2007	Acoustics	ISO 31-7	Under review
ISO 80000-9.2019	Physical chemistry and molecular physics	ISO 31-8	Published
ISO 80000-10.2019	Atomic and nuclear physics	ISO 31-9 and ISO 31-10	Published
ISO 80000-11.2008	Characteristic numbers	ISO 31-12	Under review
ISO 80000-12.2019	Condensed matter physics	ISO 31-13 subclausers 3.8 and 3.9 of IEC 60027	Published
EC 80000-13. 2008 I8	Information science and technology	ISO 31-13	Published
IEC 80000-14.2008	Telebiometrics related to human physiology	IEC 60027-7	Withdrawn

(*Continued*)

A further glossary covering further definitions and terms associated with Industrial ventilation.
Is too large for this Appendix is in preparation. For further information contact, eric.f.curd@btinternet.com.
For Air Conditioning and refrigeration, see NPTEL that covers a wide range of issues.
On the web go to NPTEL then to Mechanical Engineering and select the topic.

TABLE A.1 SI basic units.

Term	Name	Symbol
Length	meter	m
Mass	kilogram	kg
Time	second	s
Electric current	ampere	A
Temperature	kelvin	K
Luminous intensity	candela	cd
Amount of substance	mole	mol

TABLE A.2 Multiples.

Magnitude	Term	Symbol
10^{-12}	pico	p
10^{-9}	nano	n
10^{-6}	micro	μm
10^{-2}	milli	m
10^{-2}	centi	c
10^{-1}	deci	d
1	—	—
10	deca	da
10^{2}	hecto	h
10^{3}	kilo	k
10^{6}	mega	M
10^{9}	giga	G
10^{12}	tera	T
10^{15}	peta	P
10^{18}	exa	E

TABLE A.3 Heat.

Quantity	Name	Symbol	Dimension
Heat, work, or energy	joule	J	Nm
Heat flow rate, power	watt	W	J/s
Temperature (thermodynamic unit)	kelvin	K	K
Temperature (customary unit)	Celsius	θ	°C
Heat flow rate		φ	W/m^2
Thermal transmittance	Transmittance coefficient	U	$W/m^2/K$
Thermal conductivity		λ	W/m/K

TABLE A.4 Force and pressure.

Quantity	Name	Symbol	Dimension
Force	newton	N	$kg\,m/s^2$
Pressure, stress	pascal	Pa	N/m^2

Note: 1 bar = 10^5 Pa.

TABLE A.5 Viscosity.

Quantity	Name	Symbol	Dimension
Kinematic viscosity	stokes	St	$10^{-4}\,m^2/s$
Dynamic viscosity	poise	P	$10^{-1}\,N\,s/m^2$

TABLE A.6 Mass and density.

Quantity	Name	Symbol	Dimension
Mass	kilogram	m	kg
Density	—	ρ	kg/m^3

Note: 1 metric ton = 10^3 k.

TABLE A.7 Electrical.

Quantity	Name	Symbol	Dimension
Potential	volt	V	W/A
Resistance	ohm	Ω	V/A
Magnetomotive force	ampere	A	A
Charge	coulomb	C	A s
Capacitance	farad	F	A s/V
Inductance	henry	H	V s/A
Frequency	hertz	Hz	1/S

TABLE A.8 Light.

Quantity	Name	Symbol	Dimensions
Luminous flux	lumen	lm	cd sr
Illuminance	lux	lx	lm/m^2

TABLE A.9 Length (L).

	km	M	mm	mile	foot	inch	μm
km	1	10^3	10^6	6.214×10^{-1}	3.281×10^3	—	—
m	10^{-3}	1	10^3	6.214×10^{-4}	3.281	3.937×10	10^6
mm	10^{-6}	10^{-3}	1	—	3.281×10^{-3}	3.937×10^{-2}	10^3
mile	1.609	1.609×10^3	—	1	5.28×10^3	—	—
foot	3.048×10^{-4}	3.048×10^{-1}	3.048×10^2	—	1	1.2×10	—
inch	—	2.54×10^{-2}	2.54×10	—	—	1	2.54×10^4
μm	—	10^{-6}	10^{-3}	—	—	—	1

TABLE A.10 Area ($A = L^2$).

	m^2	$ft.^2$	$in.^2$
m^2	1	1.076×10	1.55×10^3
$ft.^2$	9.29×10^{-2}	1	1.44×10^2

TABLE A.11 Volume ($V = L^3$).

	m^3	L	$ft.^3$	gallon
m^3	1	10^3	3.531×10	2.200×10^2
L	10^{-3}	1	3.531×10^{-2}	2.200×10^{-1}
$ft.^3$	2.832×10^{-2}	2.832×10	1	6.229
gallon	4.546×10^{-3}	4.546	1.605×10^{-1}	1

Note: The US gallon is 3.785×10^{-3} m^3.

TABLE A.12 Mass (m).

	kg	g	lb	Grains
kg	1	10^3	2.205	—
g	10^{-3}	1	—	1.543×10
lb	4.536×10^{-1}	4.536×10^2	1	7.0×10^3

TABLE A.13 Mass per unit length (m/L).

	kg/m	lb/ft.
kg/m	1	6.720×10^{-1}
lb/ft.	1.488	1

TABLE A.14 Mass per unit area (m/L^2).

	kg/m^2	$lb/ft.^2$
kg/m^2	1	2.048×10^{-1}
$lb/ft.^2$	4.882	1

TABLE A.15 Force ($N = kg\ m/s^2$).

	N ($kg\ m/s^2$)	kN	lbf
N	1	10^{-3}	2.248×10^{-1}
kN	10^3	1	2.248×10^2
lbf	4.448	4.448×10^{-3}	1

Note: 1 kgf = 9.807 N = 2.205 lbf.

TABLE A.16 Power ($W = J/s$).

	w (J/s)	Horsepower
W	1	1.341×10^{-3}
Horsepower	7.457×10^2	1

TABLE A.17 Quantity of heat (W).

	MJ	kWh	Btu
MJ	1	2.778×10^{-1}	9.478×10^2
kWh	3.6	1	3.412×10^3
Btu	1.055×10^{-3}	—	1

Note: 1 kcal = 4.187 kJ = 3.968 Btu and 1000 kcal = 1 Thermie.

TABLE A.18 Specific heat (c).

	kJ/kg	Btu/lb/°F
kJ/kg	1	2.388×10^{-1}
Btu/lb/°F	4.187	1

TABLE A.19 Heat flow rate (ϕ).

	W	Btu/h	Refrigeration (Ton)
W	1	3.412	2.843×10^{-4}
Btu/h	2.931×10^{-1}	1	8.333×10^{-5}
Refrigeration (Ton)	3.517×10^3	1.200×10^4	1

Note: 1 kcal/h = 1.163 W = 3.96 Btu/h.

TABLE A.20 Heat emission or gain.

	W/m^2	$Btu/ft.^2/h$
W/m^2	1	3.170×10^{-1}
$Btu/ft.^2/h$	3.155	1

TABLE A.21 Heat transfer coefficient, U.

	$W/m^2/K$	$Btu/ft.^2/h/°F$
$W/m^2/K$	1	1.761×10^{-1}
$Btu/ft.^2/h/°F$	5.678	1

TABLE A.22 Heat flow per unit volume.

	W/m^3	$Btu/ft.^3/h$
W/m^3	1	9.662×10^{-2}
$Btu/ft.^3/h$	1.035×10	1

TABLE A.23 Heat flow per unit length.

	W/m	kW/m	Btu/ft./h
W/m	1	10^{-3}	1.040
kW/m	10^3	1	1.040×10^3
Btu/ft./h	9.615×10^{-1}	9.615×10^{-4}	1

TABLE A.24 Thermal conductivity (λ).

	W/m/K	Btu in./$ft.^2/h^2/°F$
W/m/K	1	6.933
Btu in./$ft.^{-2}$/h/°F	1.442×10^{-1}	1

TABLE A.25 Mass calorific value, latent heat.

	kJ/kg	Btu/lb
kJ/kg	1	4.299×10^{-1}
Btu/lb	2.326	1

TABLE A.26 Volume calorific value.

	MJ/m^3	$Btu/ft.^3$
MJ/m^3	1	2.684×10
$Btu/ft.^3$	3.726×10^{-2}	1

TABLE A.27 Pressure (p).

	kN/m^2 (kPa)	MN/m^2 (MPa)	b (bar)	$lbf/in.^2$	atm	ft. head
kN/m^2 (kPa)	1	10^{-3}	10^{-2}	1.450×10^{-1}	9.869×10^{-3}	3.346×10^{-1}
MN/m^2 (MPa)	10^3	1	10	1.450×10^2	9.869	3.346×10^2
b (bar)	10^2	10^{-1}	1	1.450×10	9.869×10^{-1}	3.346×10
Lbf/in^2	6.895	6.895×10^{-3}	6.895×10^{-2}	1	6.805×10^{-2}	2.307
atm	1.013×10^2	1.013×10^{-1}	1.013	1.470×10	1	3.390×10
ft. head	2.989	2.989×10^{-3}	2.989×10^{-2}	4.335×10^{-1}	2.950×10^{-2}	1

Note: 1 kgf/cm^2 = 98.07 kN/m^2 (kPa) = 14.22 $lbf/in.^2$.

TABLE A.28 Pressure (p).

	$N\ m^{-2}$ (Pa)	mb	in. Hg	in. H_2O
N/m^2 (Pa)	1	10^{-2}	2.953×10^{-4}	4.015×10^{-3}
mb	10^2	1	2.953×10^{-2}	4.015×10^{-1}
in. Hg	3.386×10^3	3.386×10	1	1.360×10
in. H_2O	2.491×10^2	2.491	7.356×10^{-2}	1

TABLE A.29 Density (p).

	kg/m^3 (g/L)	kg/L	$lb/ft.^3$	lb/gal
kg/m^3	1	10^{-3}	6.243×10^{-2}	1.002×10^{-2}
kg/L	10^3	1	6.243×10	1.002×10
$lb/ft.^3$	1.602×10	1.602×10^{-2}	1	1.605×10^{-1}
lb/gal	9.978×10	9.978×10^{-2}	6.229	1

TABLE A.30 Specific volume.

	m^3/kg (L/g)	L/kg	$ft.^3/lb$	gal/lb
m^3/kg	1	10^3	1.602×10	9.978×10
L/kg	10^{-3}	1	1.602×10^{-2}	9.978×10^{-2}
$ft.^3/lb$	6.243×10^{-2}	6.243×10	1	6.229
gal/lb	1.002×10^{-2}	1.002×10	1.605×10^{-1}	1

TABLE A.31 Concentration, mass per unit volume.

	g/m^3	$grain/ft.^3$	oz/gal
g/m^3	1	4.370×10^{-1}	1.604×10^{-4}
$grain/ft.^3$	2.229	1	3.670×10^{-4}
oz/gal	6.236×10^3	2.725×10^3	1

TABLE A.32 Concentration, mass per unit mass.

	kg/kg	g/kg	grain/lb
kg/kg	1	10^3	7000×10^3
g/kg	10^{-3}	1	7.0
grain/lb	1.429×10^{-4}	1.429×10^{-1}	1

TABLE A.33 Mass fluid flow (q_m).

	kg/s	kg/h	lb/h
kg/s^{-1}	1	3.6×10^3	7.937×10^3
kg/h	2.778×10^{-4}	1	2.205
lb/h	1.260×10^{-4}	4.536×10^{-1}	1

TABLE A.34 Volumetric flow of fluids (q_v).

	m^3/s	m^3/h	$ft.^3/min$	L/s	L/h	gal/min	gal/h
m^3/s	1	3.6×10^3	2.119×10^3	10^3	3.6×10^6	1.320×10^4	7.919×10^5
m^3/h	2.778×10^{-4}	1	5.886×10^{-1}	2.778×10^{-1}	10^3	3.666	2.20×10^2
$ft.^3/min$	4.719×10^{-4}	1.699	1	4.719×10^{-1}	1.699×10^3	6.229	3.737×10^2
L/s	10^{-3}	3.6	2.119	1	3.6×10^3	1.320×10	7.919×10^2
L/h	2.778×10^{-7}	10^{-3}	5.886×10^{-4}	2.778×10^{-4}	1	3.666×10^{-3}	2.220×10^{-1}
gal/min	7.577×10^{-5}	2.728×10^{-1}	1.605×10^{-1}	7.577×10^{-2}	2.728×10^2	1	6.0×10
gal/h	1.263×10^{-6}	4.546×10^{-3}	2.676×10^{-3}	1.263×10^{-3}	4.546	1.667×10^{-2}	1

TABLE A.35 Velocity (v).

	m/s	ft./s	ft./min
m/s	1	3.21	1.968×10^2
ft./s	3.048×10^{-1}	1	6.0×10
ft./min	5.080×10^{-3}	1.667×10^{-2}	1

TABLE A.36 Pressure drop per unit length.

	Pa/m	mm H_2O/m	in. H_2O/ft.	in. H_2O/100ft.	$lbf/in.^2$/100 ft.
Pa/m	1	1.020×10^{-1}	1.224×10^{-3}	1.224×10^{-1}	4.421×10^{-3}
mm H_2O/m	9.807	1	1.200×10^{-2}	1.200	4.335×10^{-2}
in. H_2O/ft.	8.172×10^2	8.333×10	1	10^2	3.613
in. H_2O/100 ft.	8.172	8.333×10^{-1}	10^{-2}	1	3.613×10^{-2}
$lbf/in.^2$/100 ft.	2.262×10^2	2.307×10	2.768×10^{-1}	2.768×10	1

TABLE A.37 Absolute (dynamic) viscosity (μ).

	P (poise) = 10^{-1} N s/m^2	cP (centipoise)	lbf s/$ft.^2$	lbf h/$ft.^2$
P (poise)	1	10^2	2.089×10^{-3}	5.802×10^{-7}
cP	10^{-2}	1	2.089×10^{-5}	5.802×10^{-9}
lbf s/$ft.^2$	4.788×10^2	4.788×10^4	1	2.778×10^{-4}
lbf h/$ft.^2$	1.724×10	1.724×10^8	3.600×10^3	1

TABLE A.38 Kinematic viscosity (v).

	St (stokes) = 10^{-4} m^2/s	cSt (centistokes)	$ft.^2$/s	$ft.^2$/h
St	1	10^2	1.076×10^{-3}	3.875
cSt	10^{-2}	1	1.076×10^{-5}	3.875×10^{-2}
$ft.^2$/s	9.290×10^2	9.290×10^4	1	3.600×10^3
$ft.^2$/h	2.581×10^{-1}	2.581×10	2.778×10^{-4}	1

TABLE A.39 Symbols.

Term	Units	Symbol
A		
Absolute radiant heat flow	W/m^2	φ_{abs}
Absolute static pressure	Pa	p_{sa}
Absolute total pressure (stagnation pressure)	Pa	P_{ta}
Acceleration	m/s^2	a
Acceleration due to gravity	m/s^2	g
Air, gas, vapor, or fluid flow rate, Mass flow	kg/s	q_m
Volume flow	m^3/s	q_v
Air leakage factor	$m^3/s/m^2$	F
Air leakage rate	m^3/s	q_{vl}
Air temperature	°C	θ_a
Air velocity	m/s	v_a
Air velocity at time t	m/s	v_t

Term	Units	Symbol
Allowable exposure time	h	AET
Angle (plane)	Radian (rad) or degree (°)	α
Angle (solid)	Steradian (Sr)	Ω
Angular acceleration	rad/s^2	a_a
Angular velocity	rad/s	ω
Approach velocity	m/s	v_{app}
Area	m^2	ω
Area, actual (filter face)	m^2	ω_f
Area, duct cross-section	m^2	A_{dcs}
Area (filter medium)	m^2	A_{fm}
Area (filter surface)	m^2	A_{fs}
Atmospheric pressure	Pa	p_a
B		
Basal metabolic rate	W/m^2	BM
Blade (fan) tangential velocity	m/s	U
Body heat storage	W/m^2	S
Body height	m	h_b
Body mass	kg	m_b
Body surface area	m^2	A_{Du}
Body surface area covered with clothing	%	A_{cov}
Boundary layer insulation	Clo	I_a
Breadth	m	B
Bulge or sag of a duct or enclosure	m	S
C		
Capacity (dust-holding)	kg/kg	C_{dh}
Carbon dioxide production	L CO_2/h	$q_{v,\,CO_2}$
Cartesian coordinates	—	X, y, z
Celsius temperature	°C	Θ
Chilling temperature	°C	θ_{ch}
Coefficient of cubical expansion	1/K	β
Counting rate	1/s$^-$	N
Clothing insulation	m^2 °C/W	Clo
Clothing mass variation	kg	$\Delta\, m_{clo}$
Clothing surface temperature	°C	θ_{clo}
Coefficient of thermal conductivity	W/m °C	λ
Component of air velocity along the X axis	m/s	v_x
Component of air velocity along the y axis	m/s	v_y
Component of air velocity along the z axis	m/s	v_z

Term	Units	Symbol
Compressibility factor of a gas	—	Z
Conductive heat exchange	W/m^2	φ_{cond}
Convective heat exchange	W/m^2	φ_{conv}
Convective heat exchange ($\pm$) from globe thermometer to air	W/m^2	φ_g
Convective heat transfer coefficient	W/m^2/K	h_c
Core temperature	°C	θ_c
Cross-sectional area	m^2	A_c
D		
Darcy friction factor	—	λ
Deflection	M	δ
Density	kg/m^3	ρ
Dew point temperature	°C	θ_d
Diameter ratio of a flow measuring device	—	β
Diameter		
Outer	—	D
Inner	—	d
Differential pressure	Pa	Δp
Distance to v m/s isovelocity line	m	L_v
Draft rating	%	DR
Drop of air jet from its leaving center line	m	h_v
Dry heat loss	W/m^2	φ_{dry}
Dryness fraction, steam	%	X
Duration, limited exposure	h	D_{LE}
Dynamic pressure	Pa	P_d
Dynamic viscosity	N s/m^2	μ
E		
Effective area of a device	m^2	A_{eff}
Effective clothing insulation	m^2 °C/W	I_{clo}
Effective length	M	l
Effective mechanical power	W/m^2	W
Effective radiant heat flow	W/m^2	$\varphi_{r\ eff}$
Effective radiating area of a body	m^2	A_r
Efficiency	—	H
Efficiency average	—	η_{av}
Emissivity of a surface or sensor	—	ϵ_s
Emissivity of black globe	—	ϵ_g
Energy	J	E
Energy loss per unit mass	J/kg	Δy
Enthalpy per unit mass	J/kg	h

Term	Units	Symbol
Entropy per unit mass	J/kg/K	s
Equivalent diameter of a rectangular duct	M	d_e
Evaporative heat transfer coefficient	W/n^2/Pa	h_e
Exposed area	m^2	A_{exp}
F		
Face loading (filter)	kg m^2	—
Fan air power	W	P_f
Fan or pump efficiency	—	η_r
Fan equivalent orifice	m^2	o_{fe}
Fan or pump head	m, Pa	H
Fan or pump impeller power	W	P_{fi}
Fan or pump work per unit mass	J/kg	Y
Fan pressure	Pa	P_f
Fan or pump shaft power	W	P_s
Flow coefficient of leakage	m^3/(s Pa^n)	C_l
Flow coefficient of subsonic flow in an orifice	—	α
Flow mass	m^3/s	q_m
Flow volumetric	m^3/s	q_v
Fluid density upstream of a measuring device	kg/m^3	ρ_u
Force	N	F
Frequency	1/s	f
G		
Globe temperature	°C	θ_g
Gross body mass loss	kg	Δm_g
H		
Heat capacity	J/K	C
Heat flux	W	ϕ
Heat flux density	W/m^2	φ
Height	M	H
Height above datum	M	Z
Height of v m/s isovelocity line	M	h_v
Humidity ratio	kg water/kg dry air	W_a
Humidity ratio at saturation	kg water/kg dry air	W_{as}
Humidity ratio expired air	kg water/kg dry air	W_{ex}
Humidity ratio inhaled air	kg water/kg dry air	W_a
Hydraulic diameter	M	d_h
I		
Impeller tip diameter of a fan	M	D
Impeller tip radius of a fan	M	R
Increase in body core temperature	°C	$\Delta\theta_{co}$
Internal diameter of a pipe or duct	M	D
Insulation of clothing	m^2 K/W	I_{cl}
Internal energy per unit mass	J/kg	u
Isentropic exponent	—	k
J		
Jet angle	°	α, β
Jet drop	M	h_d
Jet rise	M	h_r
Jet spread	°	β
Jet temperature	°C	θ_J
Jet throw	m	L_J
K		
Kinematic viscosity	m^2/s	V
Kinetic energy (mass)	J/kg	e_K
L		
Latent heat (mass)	J/kg	l
Length	m	L
Lewis relationship	°C/kPa	LR
Limit value for body heat gain or loss	W h/m^2	Q_{lim}
Local skin temperature	°C	θ_{sk}
M		
Mach number	—	Ma
Mass	kg	m
Mass of dry air	kg	m_{da}
Mass flow rate (gas or fluid)	kg/s	q_m
Mass of water vapor	kg	m_{wv}
Maximum body heat storage	W h/m^2	Q_{max}
Maximum evaporative heat transfer from skin	W/m^2	E_{max}
Mean penetration (filter)		P_m
Mean pressure drop	Pa	Δp_m
Mean skin temperature	°C	θ_{sk}
Mean velocity of flow in a conduit	m/s	v_m
Metabolic rate	W/m^2	M (met)
Molar mass	kg/mol	MM
Momentum	kg m/s	p
Motor input power	W	P_E
Motor output fan efficiency	—	η_M
Motor power output	W	P_M
N		
Natural wet bulb temperature	°C	θ_{wb}
Nominal volume air flow	L	$q_{v\ nom}$

Term	Units	Symbol
O		
Operative temperature	°C	θ_{op}
Overall fan efficiency	—	η_F
Overall heat transfer coefficient	W/m^2/K	U
Overlap length (ductwork)	m	l_p
P		
Partial pressure	Pa	p_v
Particle production rate	1/s	Q_p
Particle size	μm	d_p
Percentage dissatisfied	%	PD
Periodic time	s	t
Permeability index for clothing layer	—	I_{clo}
Plane angle	Rad or °	α, β, γ
Plane radiant temperature	K	T_{pr}
Polytropic coefficient	—	n
Position of control setting	% or °	s
Power	W	P
Predicted mean vote	—	PMV
Predicted percentage dissatisfied	%	PPD
Pressure difference between points	Pa	Δp_t, Δp_s, etc.
Pressure loss coefficient	—	ξ
Pressure total	Pa	p_t
Primary air flow rate	m^3/s or 1/s or kg/s	q_{vp} or q_{mp}
Q		
Quantity of heat	J	Q
R		
Radiation heat transfer coefficient	W/m^2/K^4	h_r
Radiation temperature asymmetry	°C	T_{as}
Radiative heat exchange	W/m^2	φ_r
Radiative heat exchange between globe thermometer and surroundings	W/m^2	φ_g
Radiative heat transfer coefficient	W/m^2/K	h_r
Radius		
Inner	m	r
Outer	m	R
Radius of curvature	m	r_m
Ratio of specific heat	—	γ
Capacities		
Relative fluid velocity to an impeller	m/s	W
Relative humidity	—	ϕ
Reverberation time	S	t
Rotational speed	1/s	n
S		
Saturation pressure of a vapor	kPa	p_{sat}
Saturated water vapor pressure at skin temperature	kPa	p_{sks}
Saturated water vapor pressure at wet bulb temperature	kPa	$p_{as,w}$
Secondary air flow rate		
Mass flow	kg/s	q_{vs}
Volume flow	m/s or 1/s	q_{ms}
Shaft fan power efficiency	—	η_A
Solid angle	sr	Ω
Sound power level	dB	L_w
Sound pressure level	dB	L_p
Specific heat capacity	J/kg/K	c
Specific heat capacity at constant pressure	J/kg/K	c_p
Specific heat capacity at constant volume	J/kg/K	c_v
Spread of a jet	m	b_v
Stagnation pressure	Pa	p_{ta}
Static gauge pressure	Pa	p_s
Stefan-Boltzmann constant	W/m^2/K^4	σ
STPD reduction factor	—	F
Surface area	m^2	A_s
Surface heat transfer coefficient	W/m^2/K	h
Surface temperature	°C	θ_s
Surface tension	N/m^2	σ
T		
Tangential component relating to a fan, or pump impeller, or fluid	m/s	c_u
Temperature difference	K or °C[a]	ΔT or $\Delta\,\theta$[a]
Thermal diffusivity	m^2/s	A
Thermodynamic (absolute) temperature	K[a]	T[a]
Thickness	m	t or d
Thickness of dynamic boundary layer	m	δ
Thickness of thermal boundary layer	m	δ_T
Throw of a jet	m	L_J
Time	s	t
Time constant, exponential change	s	τ
Tip Reynolds number of a fan impeller	—	Re_u

Term	Units	Symbol
Tip speed of a fan impeller	m/s	U
Torque	N m	T
Total gas or air flow rate		
Mass flow	kg/s	q_{mt}
Volume flow	m^3/s; L/s	q_{vt}
Total gauge pressure	Pa	p_t
Total heat transfer coefficient	$W/m^2/°C$	H
Turbulence intensity	%	T_u
U		
Universal gas constant	J/kg/K	R
V		
Velocity	m/s	V
Velocity components in the x, y, z directions	m/s	u, v, w
Velocity of sound	m/s	c
Volume	m^3	V
Volume flow rate	m^3/s or L/s	q_v
W		
Water vapor latent heat of vaporization	W/m^2	φ_L
Water vapor partial pressure	kPa	p_a
Water vapor pressure at skin temperature	kPa	p_{sk}
Wave length	m	λ
Weight	N	G
Weighted sound pressure level	dB A	L_{pA}
	dB B	L_{pB}
	dB C	L_pC
Wet bulb globe temperature	°C	θ_{wbg}
Wetted duct perimeter	m	χ
Width	m	b
Wind chill index	W/m^2	WCI
Work	J	W
Y		
Young's modulus	N/m^2	E

[a] *In normal work, °C is used in preference to the absolute temperature K. However, it is essential that K be used when working with the gas laws, radiation, and the coefficient of cubical expansion. The symbol for normal temperature is θ followed by a suffix, while T always denotes absolute temperature.*

TABLE A.40 The Greek alphabet.

Name	Symbols	
Alpha	A	A
Beta	β	β
Gamma	Γ	γ
Delta	Δ	δ
Epsilon	E	ϵ
Zeta	ζ	ζ
Eta	H	η
Theta	θ	θ
Iota	I	ι
Kappa	K	κ
Lambda	Λ	λ
Mu	M	μ
Nu	N	ν
Xi	χ	χ
Omicron	O	O
Pi	Π	π
Rho	P	ρ
Sigma	Σ	σ
Tau	τ	τ
Upsilon	Y	υ
Phi	Φ	ϕ
Chi	χ	χ
Psi	Ψ	ψ
Omega	Ω	ω

TABLE A.41 Symbols for operations.

Symbol	Definition
$\cong$	is identical to
$\neq$	does not equal
$\cong$ or $\approx$	is approximately equal to
$\propto$	is directly proportional to
$\rightarrow$	tends to
$<$	is less than
$>$	is greater than
$\leq$	is less than or equal to
$\geq$	is greater than or equal to
Δx	finite increase in x
Δx	variation in x
Dx	total differential in x
Grad	gradient
Div	divergence
Curl	curl

Symbol	Definition
∇^2	Laplacian
!	factorial
( )	parentheses
exp or e^x	exponential of x
ln x	logarithm to base e of x
$\log_{10} x$	logarithm to base 10 of x
[ ]	brackets
[1]	one-dimensional
[3]	three-dimensional
Σ	summation

TABLE A.42 Abbreviations.

Meaning	Abbreviation
About	ca.
Absolute	abs
Alternating current	a.c.
Apparatus dew point	Adp
Atomic weight	At.wt.
Boiling point	b.p.
Boundary layer	bd.
Centerline	c.l.
Compare	cf.
Direct current	d.c.
Dry bulb temperature	d.b.t.
Electromotive force	emf
Equation	Eq.
For example	e.g.
High pressure	H.P.
Hydrogen ion concentration	pH
Liquid (specified)	L (followed by the appropriate chemical symbol)
Liquid oxygen	LOX
Liquefied petroleum gas	LPG
Melting point	m.p.
Molecular weight	mol. wt.
Namely	viz.
Note well	n.b.
Outside diameter	OD
Parts per million	ppm
Percent	%
Relative humidity	RH
Research and development	R&D
Specific	Sp.
That is	i.e.
Latin terms	
In the place cited (reference to an earlier quote)	*loc. cit.*
In the work cited (a further reference to a book previously mentioned, but this time in a different passage)	*op. cit.*
In the same place (a reference to a topic covered in a preceding reference)	*ibid.*
And another or and others (e.g., Burgess et al. rather than Burgess, Ellenbecker, and Treitman)	et al.

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