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Air Curtains for Buildings and Industrial Processes

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Abbreviations and Acronyms

ACC   Air Curtains for Cooled Spaces
ACH   Air Curtains for Activated Exhaust Hoods
ACHI   Air Curtains that Supply Heated Indoor Air
ACPE   Air Curtains for Emission Prevention from Process Equipment
ACUHI   Air Curtains with Unheated Indoor Air
ACUHO   Air Curtain Using the outside Unheated Air
CAC   Combined Air Curtain
CACL   Combined Air Curtain with a Lobby
HPAC   Heat Piping/Air Conditioning
MAC   Mixing-Type Air Curtain

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Chapter 1
Introduction

Air curtains are local ventilation devices used in large commercial, industrial buildings, warehouses, and maintenance facilities to reduce the leakage of airflow through dedicated apertures (gates, doors, windows, etc.) in building enclosures. The operation of air curtains is based on the damping effect of air jets installed and operated near the open aperture. The use of air curtains offers several advantages; they improve working conditions near open apertures, reduce the consumption of heat and electric energy used to heat the building, and reduce the loss of usable working space near the apertures due to ingress of cold air into the building. Well-designed air curtains can improve the indoor air environment of the premises, reduce construction costs, and reduce energy consumption during the building operation by 10–20%.

Traditional air curtains, which use only indoor air heated inside the curtain assembly, are not always economical since they consume considerable thermal energy. This heat consumption can be reduced by using air curtains that use unheated indoor or outdoor air and by using combined air curtains that heat only some of the supplied air. Air curtains that use unheated air conserve 30–70% of thermal energy compared to air curtains supplying heated air.

This book provides an overview of air curtains used for commercial and industrial buildings and processes, including methods of their design and criteria for their selection.

Chapter 2
Classification of Air Curtains and Requirements for Their Performance

This book discusses the following types of air curtains:

- Air curtain that supply heated indoor air (ACHI).
- Air curtains that supply unheated indoor air (ACUHI).
- Combined air curtains (CAC).
- Air curtains with recirculation of unheated outdoor air (ACUHO).
- Air curtains for cooled spaces (ACC).
- Mixing-type air curtain (MAC).
- Combined Air Curtain with a Lobby (CACL)
- Air curtains for emission prevention from process equipment (used in industrial buildings) (ACPE).
- Air curtains for activated exhaust hoods (ACH).

Around the world, many different types of air curtains, in various configurations, are used to supply heated and unheated indoor and outdoor air [1–6]. The book describes different designs of air curtains, discusses the advantages and disadvantages of each type, and provides examples of their applications. In many industrial buildings, one or several apertures equipped with air curtains may be found (Fig. 2.1).

Shutter-type air curtains create resistance to airflow through the aperture and thus reduce the incoming airflow. They direct air toward the incoming outside air at an angle ranging from 20 to 40°. Shutter-type air curtains may be single-sided or double-sided and may project air upward, downward, or in a lateral direction (Figs. 2.2 and 2.3). Lateral shutter-type air curtains are often double-sided because to better deter airflow. Upward projected air curtains are typically used when the gate width is greater than its height. The jets can be supplied through a slot air diffuser (Figs. 2.4 and 2.5) or a system of nozzles (Fig. 2.6).

Typically, ACHI are used for doors smaller than 3.6×3.6 m and for process apertures that are frequently opened, e.g., more than five times or longer than 40 min during an 8 h shift), and that are in regions with design outdoor winter temperatures of ($- 15$ °C) and lower. This type of air curtains provides desired air temperatures

Fig. 2.1 Examples of protected apertures of industrial facilities equipped with air curtains:
a Machining shop of Ford Werke, Koeln-Neil, Germany; **b** Scania Cabin Plant in Oskarshamn,
Sweden. *Source* A. Zhivov [2, 3]

Fig. 2.2 Air curtain using
the indoor heated air with a
two-sided lateral supply
through slot air diffusers
(ARWUS, Germany)
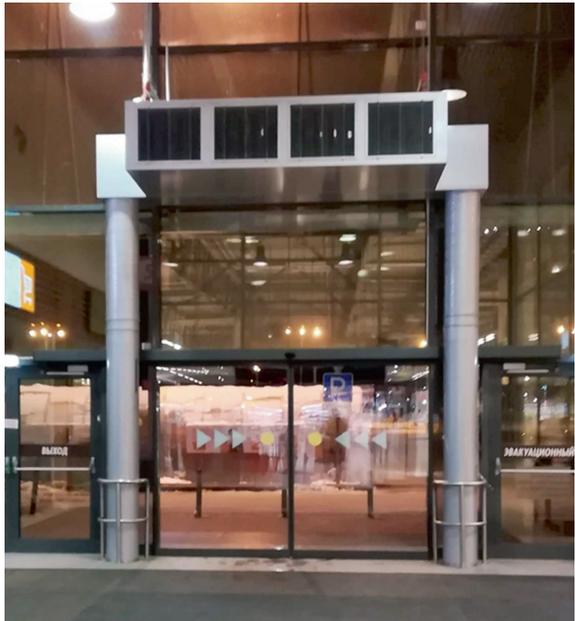
Fig. 2.3 Air curtain using
the indoor heated air with a
two-sided lateral supply
through slot air diffusers
(Frico, Sweden)

Fig. 2.3 Air curtain using the indoor heated air with a two-sided lateral supply through slot air diffusers (Frico, Sweden)

Fig. 2.4 Example of air curtain with a slot air diffuser

for workstations located near apertures; however, their use is associated with high heat consumption.

Air curtains supplying unheated indoor air (ACUHI) have a similar air supply arrangement as those that supply heated air. Only the design of the air intake duct might differ; it may be extended to allow the use the warm air, for example, from the upper overheated zone of the room. Air curtains with unheated indoor air are recommended where the temperature of the air entering the building through the

Fig. 2.5 Example of air curtain with multiple rectangular air diffusers (ARWUS, Germany)
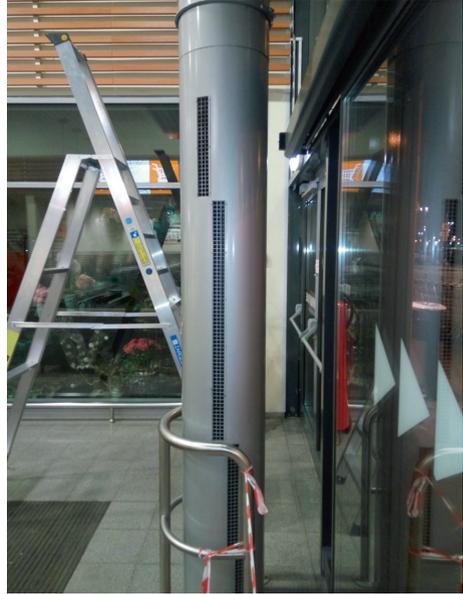
Fig. 2.6 Example of air curtain with multiple air supply air nozzles at Scania Cabin Plant in Oskarshamn, Sweden. *Source* A. Zhivov [2]
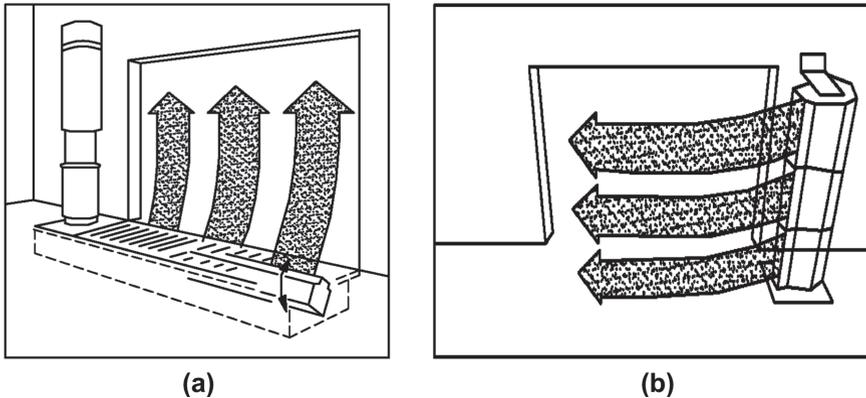
Fig. 2.7 Schematic of **a** ACUHI air curtain with vertical upward air supply and **b** with a horizontal one one-sided air supply

aperture of the open gate will be sufficient without heating the air in the curtain, for example:

- Spaces with an overheated upper zone of over 2 °C (if it is possible to use the air from the upper overheated zone of the room).
- Spaces with excess heat (if it is technically possible and reasonable to use it).
- Spaces with no permanent workplaces in the area near the aperture or those allowing lower air temperature in the working zone near the gates (8 °C) and below).

The advantage of ACUHI is that they do not need air heaters although their application is limited compared to the use of ACHI. These air curtains usually supply air from the bottom of the opening (Fig. 2.7), and they are commonly used with an air supply duct at the bottom of the opening.

CAC that supply heated indoor and unheated outdoor air (Fig. 2.8) are used in regions with very cold climates (winter temperatures as low as − 65 °C for doors larger than 3.6 × 3.6 m and for spaces with several doors). CAC systems supply air through two pairs of linear jets either parallel to each other or diverging at a slight (5–15°) angle.

Jets formed by nozzles in the duct that are located close to the door supply unheated room air. Jets supplied from nozzles in the duct located further from the door opening supply heated air. The heat loss resulting from the air curtain jet's contact with the outside air is reduced due to a decrease in its average temperature. To reduce the heat transfer between the external and indoor (more heated) jets of the CAC, it is recommended to leave a gap between them, which ensures that the leakage of the internal air of the room carries only into the space between the "outside" and "indoor" jets. The optimal size of this gap, the distance between the outlet nozzles, and the angles of air supply are determined during the design process of the CAC.
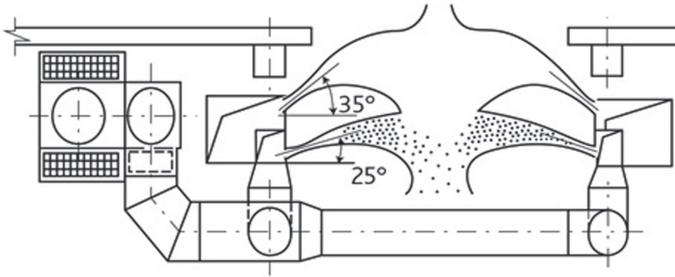
Fig. 2.8 Example schematic of horizontal cross-section of CAC
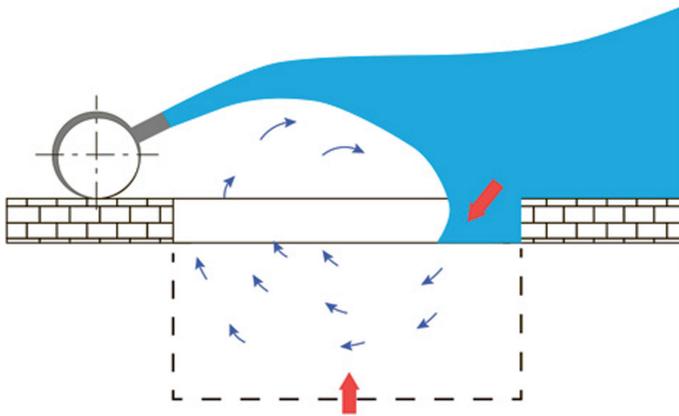
Fig. 2.9 Schematic with horizontal cross-section of ACUHO air curtain using the outside unheated air

The disadvantages of CAC are its relatively high cost and complex design, which limit the scope of its application.

CAC can reduce energy consumption by 25–60% depending on the size of the gate and the location (climate) of the building and can reduce the energy costs by 30–70%.

The main advantages of air curtains that supply unheated outdoor air (ACUHO) are their simplicity and cost-effectiveness. Because they do not heat the supplied air, they incur no costs for thermal energy and associated equipment (heat exchangers, etc.) (Figs. 2.9 and 2.10).

At the same time, ACUHO have several disadvantages that limit the scope of their application:

- Occupants experience discomfort resulting from high air velocity in the vicinity of the jet when workplace is located close to the gate or people have to walk/drive through the jet.

Fig. 2.10 Schematic and an example of ACUHO air curtain application

- ACUHO is unsuitable for use in rooms with moisture release due to the potential for fogging.

Depending on the design of the ACUHO, air circulation is created in one of two areas of the room, either closed (restricted to the area of the gate) or open to the volume of the room. Closed circulation is preferable if the purpose is to thermally protect the opening; open circulation will allow heat exchange between the air of the room and the air of the circulation zone due to the absence of a directed averaged airflow and the mechanism of turbulent exchange.

The ACC curtain design is used for all types of premises that use artificial cooling (refrigerating chambers, air-conditioned warehouses, etc.). ACC helps to reduce the load placed on the refrigeration equipment, reduces energy costs, and increases the equipment service interval. ACC also reduces the potential for emergency situations associated with the formation of condensate and ice. As a rule, ACC units are placed above the gate (top supply), outside the cooled room. Air supply is directed at a slight angle to the plane of the door opening. The selection of the optimal air curtain scheme can be a complex process that requires a calculation of technical parameters and an economic comparison of different options. Figure 2.11 shows the airflow during ACC operation, and Fig. 2.12 shows an example of ACC use in a food store.

Mixing-type air curtains (MAC) are designed to improve thermal conditions near the entrance doors during the cold season. They are commonly used in public buildings, e.g., office centers, shopping and multifunctional complexes, sports and entertainment enterprises, and transport infrastructure facilities (train stations, airports, etc.). Figures 2.13 and 2.14 show examples of MAC.

The performance of air curtains in combination with a lobby is based on transition of the supply jet momentum into counter pressure, which prevents outdoor airflow into the shop (Fig. 2.15). Air is supplied in a direction opposite to that of the outdoor airflow or at a small angle to it. Air jets propagate along the walls of the channel/lobby, slow down, make a U-turn, and create a reverse flow moving along the axis of the lobby. A closed circulation contour is created, stable against external effects.

Fig. 2.11 Schematic of vertical cross-section showing airflow during ACC operation according to [7]
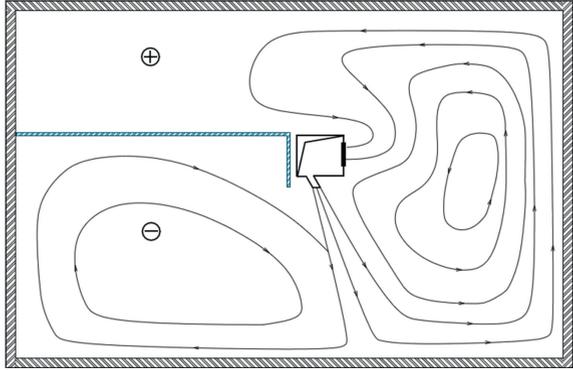
Fig. 2.12 Example of ACC operation in a food store (Frico, Sweden)

The lobby length is selected such that the supply air is not forced outside. To shorten the length of the lobby, the air can be supplied with an incomplete fan jet from a specially constructed outlet. The air curtain can include medium pressure axial fans mounted on top of air distribution ducts.

Fig. 2.13 Example of mixing air curtain with a vertical air supply use at a shopping mall (Frico, Sweden)

Fig. 2.14 Example of the mixed air curtain incorporated into concave profile of the body of the revolving door (Frico, Sweden)

The lobby can be located outside (Fig. 2.16) or inside (Fig. 2.17) the building. Walls of the external lobby can be manufactured from the corrugated steel to reduce the wind pressure on the gate (Fig. 2.18). Air can be supplied through a slot-type air diffuser installed along the duct (Fig. 2.15) or through a row of small size nozzles (Fig. 2.16). Special measures should be considered to prevent negative pressure in the building when the wind pressure is less than the design value used to calculate the supply jet momentum. For locations with a severe climate, i.e., low outside air temperatures and strong winds, the authors developed a combined air curtain with a specially designed lobby shown in Fig. 2.18. Air curtains in the lobby supply untreated outdoor air, while air curtains inside the building supply heated air. Studies
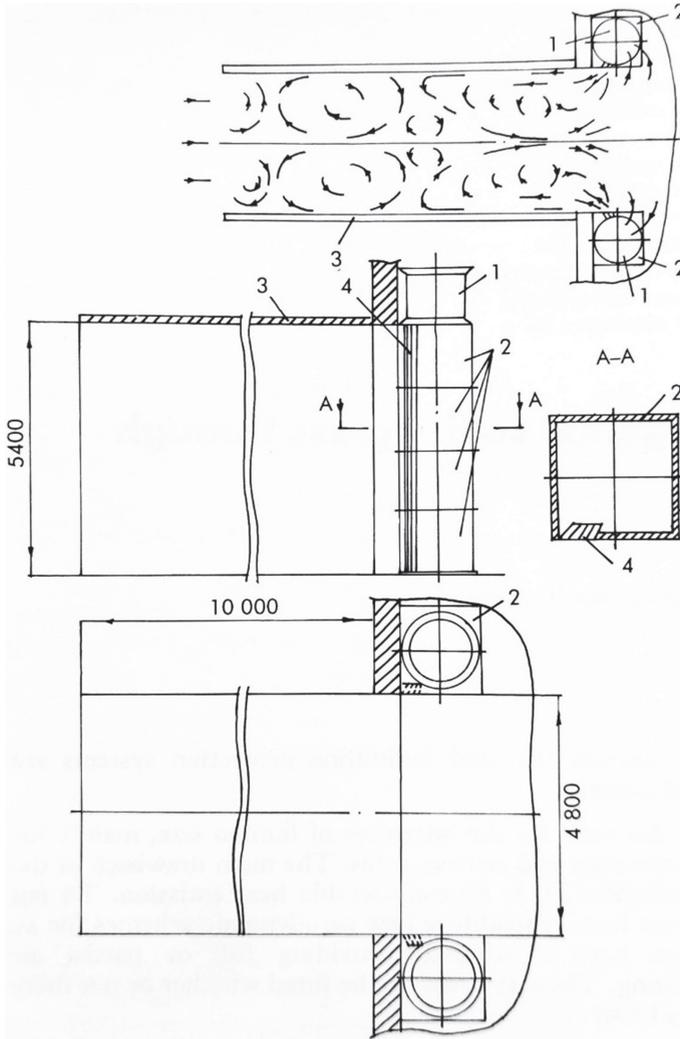
Fig. 2.15 Air curtain for medium size door with lobby: (1) fan; (2) distributive duct; (3) lobby; (4) outlet slot

of this design conducted in a wind tunnel have shown that such configurations significantly reduce wind pressure with any wind direction. Calculations have shown that air curtains of this design use 2 to 3 times less thermal energy to operate than traditional air curtains.

When possible, individual process equipment or parts of the automated process conveyors may be encapsulated by enclosures to prevent the ingress of heat and environmental contaminants such as gases, vapors, smoke, and aerosols into the building [7, 8] (Fig. 2.19). These enclosures may have minimal fixed openings for

(a)   (b)

(c)
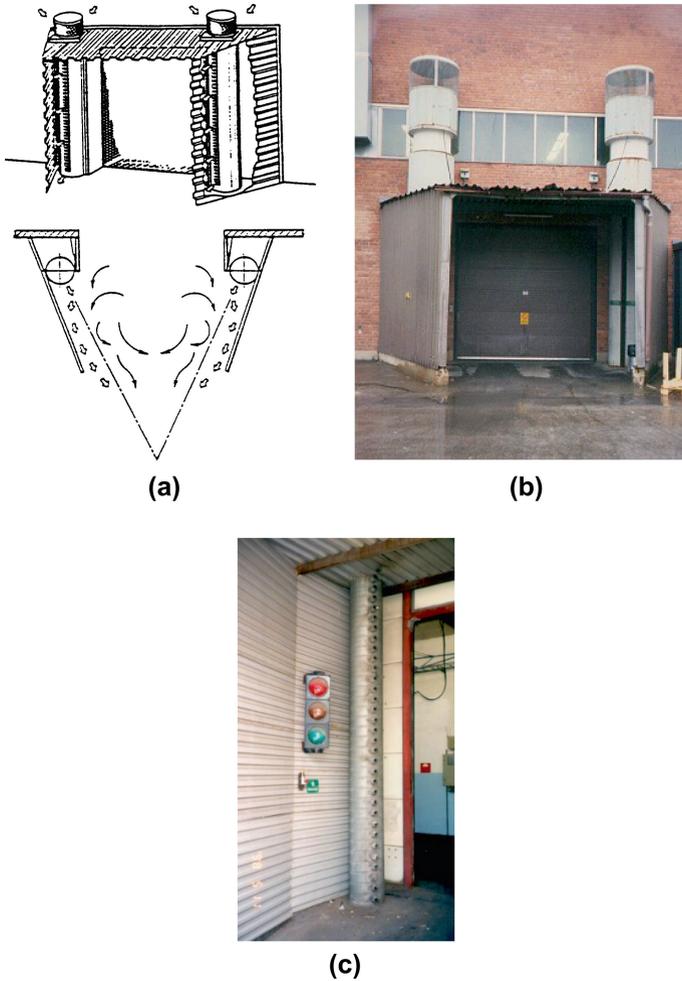
Fig. 2.16 Air curtain with a lobby and outside air supply [2]: **a** schematic, **b** general view (Scania Truck Plant in Södertälje, Sweden), **c** air distribution duct with multiple nozzles (Scania Cabin Plant in Oskarshamn, Sweden)

parts entry and exit and automated quick opening doors (Fig. 2.20). To work properly, these enclosures require a controlled flow of air through the enclosure. The enclosure is kept under negative pressure by the exhaust system connected to the enclosure. Although the enclosure alone will not control contaminants, the use of air curtains for emission prevention from non-isothermal industrial processes (ACPE) can help.

The airflow rate of the exhaust system depends on the size of the fixed apertures of the enclosure. The process equipment apertures can be equipped with ACPE to reduce the airflow rate created by the exhaust system, the size of the fan, air filter, ductwork, and energy required to operate this system [7, 8].
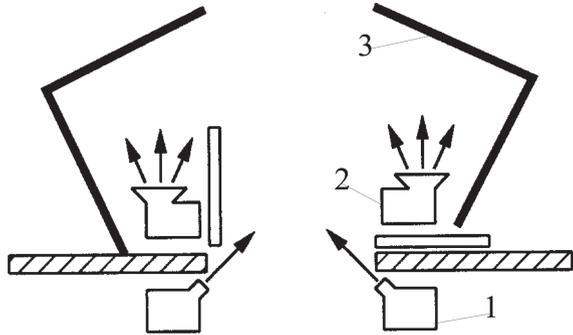
The encapsulated process can be classified as isothermal (e.g., spray-painting, mashing, welding) and non-isothermal (e.g., heat dryers). Apertures of enclosures over an isothermal process can be equipped with damper-type air curtains complemented by an exhaust system that extracts air from the enclosure. Enclosures for non-isothermal processes can be equipped with a "circulation type air curtains installed in a channel" (Fig. 2.21).

A method called "air curtains for activated exhaust hoods (ACH)" uses an air curtain incorporated into the exhaust hood design to significantly increase the effectiveness of the hood by using the "push–pull principle" of hood performance. This principle is based on creating an air curtain around the contaminating source to prevent contaminant spread throughout the room and to evacuate the contaminated air outside the room through the hood. The use of a push–pull hood reduces the amount of air to be evacuated as well as the air to be supplied in the room and thus reduces first costs and power consumption on air heating and cleaning (Fig. 2.22).
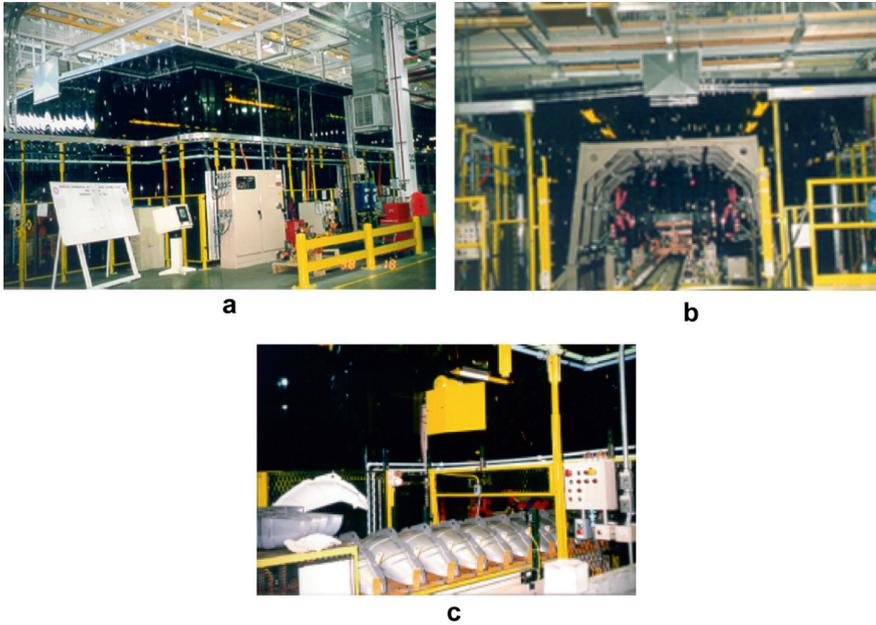
Fig. 2.19 Canopy hood with a plastic curtain enclosing part of the conveyor with a robotic spot-welding area of the body shop at the Chrysler North Jefferson Assembly Plant: **a** general view; **b** welded parts exit from the enclosure; **c** components entrance into the enclosure. *Source* A. Zhivov [2]; Reproduced with permission from the Ventilation Guide for Automotive Industry

After considering the various types of air curtains described here, subsequent selection and design of the optimal air curtain solution can be a difficult task that requires technical and economic optimization using the criteria, calculations, and methods outlined in the following chapters.
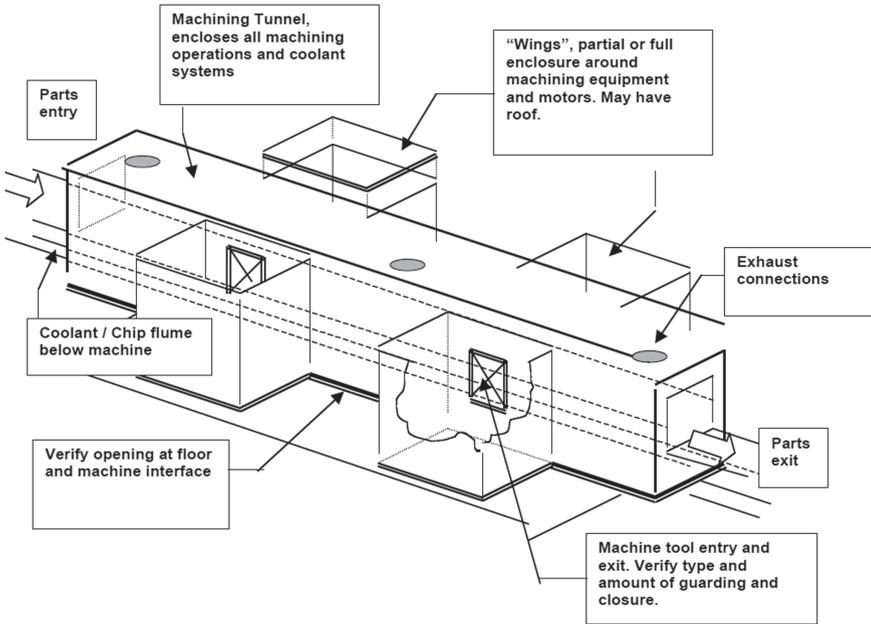
Fig. 2.20 Enclosure over the Automatic Pallet Machine, or Transfer Line [VGAI]. A series of machines and machining operations that are grouped into one continuous line, with the parts being machined moving through the operations, station by station, on a moving pallet

1 – Drying chamber   3 – Air supply duct   5 – Heater   7 – Conveyer axis
2 – Aperture   4 – Air supply slot   6 – Blower
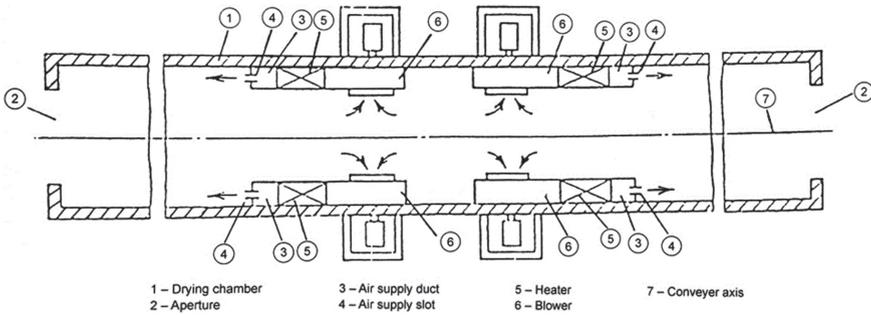
Fig. 2.21 Push–pull protection system for the drying chamber
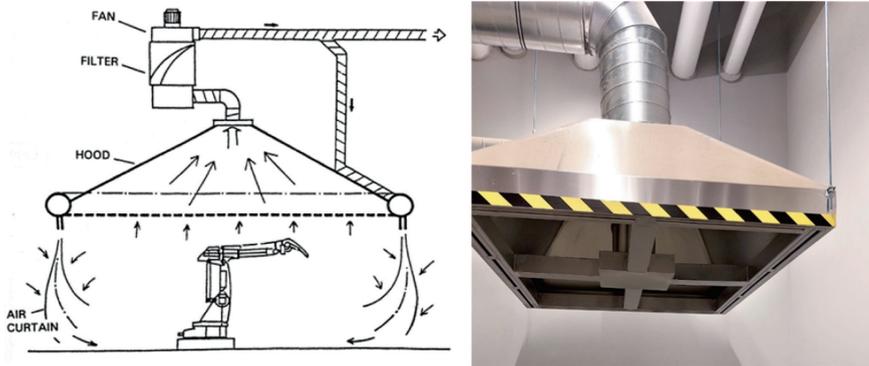
Fig. 2.22 Push–pull hood over a welding robot: schematic of application over the welding robot (left) and example of the hood (right)

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Chapter 3
Indicators/Efficiency Criteria of Air Curtains

Selection of the optimal air curtain type, even determination of parameters to guide selection, requires technical evaluation and economic analysis. This task is made more difficult when the technology will be used in atypical applications or with large-size gates since science-based and field-proven information is not readily available for these less common uses. Air curtain manufacturers' guidelines are limited; the information they contain does not support optimal air curtain selection. This absence of specific selection criteria or guidelines, when combined with a lack of case studies, can sometimes result in the selection of suboptimal or mismatched technologies that provide poor protection to the building's apertures. The following performance dimensionless criteria for air curtains support the informed selection of air curtain archetypes [1]:

- Dynamic efficiency, E.
- Energy efficiency criterion, η.
- Occupational comfort efficiency, θ.

The **dynamic efficiency** of the air curtain, E, is the ratio between the aerostatic pressure difference across the gate aperture and the initial impulse of the air curtain jets:

$$E = \Delta P (H \, x \, B)/2J_0 \qquad (3.1)$$

where

$\Delta P =$ pressure static pressure difference across the building's aperture, Pa.
$J_0 =$ initial air curtain jet momentum, N.
$H, B =$ dimensions of the aperture, m.

The **energy efficiency criterion** of the air curtain is the ratio of thermal energy use reduction because of the air curtain and the heat losses through the non-protected aperture:

$$\eta = 1 - Q_{AC}\big/Q_{wo} \tag{3.2}$$

where

Q_{AC}, Q_{WO} = heat losses through the aperture, protected by the air curtain and without air curtain protection, kW.

$\eta = \eta = 1$ indicates an ideal air curtain (that eliminates heat losses through the aperture), and $\eta = 0$ when the aperture is not equipped with the air curtain.

Note that, for a poorly selected and designed air curtain, η can be negative due to the increased heat transfer through the aperture or excessive use of thermal energy by the air curtain or by the auxiliary heating system.

The occupational comfort efficiency, θ, criteria indicate the effectiveness of the air curtain in providing acceptable (by the thermal occupancy standard) air temperature in the occupied zone in the vicinity of the aperture:

$$\theta = 1 - \frac{t_{in} - t_{mix}}{t_{in} - t_{out}} \tag{3.3}$$

where

t_{in}, t_{out}, t_{mix} represent the temperature of the internal, outdoor, and mixed air entering the building, respectively, in degrees centigrade (°C).

Thermal comfort requirements t_o the air temperature t_{mix} entering the building through the aperture are based on the type of human activity in the vicinity of the aperture, their distance to the aperture, and process requirements (when applicable).

$\theta = 1$ means the air temperature in the air mix passing through the aperture is equal to the acceptable indoor air temperature in the vicinity of the aperture ($t_{mix} = t_{in}$).

$\theta = 0$ means that, even when there is no air curtain, the temperature of the air entering the building is equal to the outside air temperature.

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Chapter 4
Calculations

The dynamic method of calculating air curtains described in this chapter, which is based on the laws of conservation of impulse and of mechanical energy within an isolated contour, is used to determine the main parameters for air curtain design. The results are in a good fit with the experimental data [1–5].

Figure 4.1 demonstrates the use of the dynamic method for a single-sided air curtain that is supplied at an angle α. The jet of the air curtain is bent by the effect of pressure difference between outside (P_{out}) and inside (P_{in}) of the building. The air curtain jet flows to the opposite side of the aperture and splits into two parts. At the point of 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. In this example, the isolating circuit has a boundary, ABVGDKMN. Surfaces AB, AN, and BV are located at a distance from the gate where the speed of air entrained by the jet is close to zero and the value of the impulse of air penetrating the surface NABV may be disregarded.

The equation of momentum for the isolated circuit in projection at the Y-axis is

$$I_0 \sin\alpha + I_0 \sin\beta = \Delta P A_0 + (P_{int} - N_y)(A_{AB} - A_0) \tag{4.1}$$

where

I_0, I_b, I_{out} are impulses of airflows, supplied by the air curtain, airflows into and out of the room (N).

A_{AB}, A_0 are the areas of the surface AB and the aperture (m^2).

N_y is the average value of reactive pressure of the surfaces VG and MN (Pa).

$$\Delta P = P_{out} - P_{in} (Pa) \tag{4.2}$$

Fig. 4.1 Theoretical model of an air curtain

Air pressure difference, ΔP, between the outside and inside air across the aperture protected by the air curtain is a very important factor in the air curtain design, and its evaluation is described in Appendix A.

The dynamic efficiency of the air curtain, E, can be defined as a ratio of aerostatic pressure forces affecting the gate aperture over the double initial impulse of the air curtain jets:

$$E = \Delta P \, A_0 / 2I_0 \tag{4.3}$$

The supply air velocity, V_0, for valuation of the initial momentum, I_0, of the air curtain jet can be determined using the following equation derived by the joint solution of Eqs. 4.1–4.3:

$$V_0 = (\Delta P A_0 / 2\beta_0 \rho A_S E)^{1/2} [m/s] \tag{4.4}$$

where

β_0 = the Boussinesq factor (1.05–1.1).
ρ = density of air supplied by the air curtain [kg/m³].
A_0 = total area of the air curtain supply nozzles/slot [m²].

It is recommended to select the value of the ratio $f = A_{\mathrm{ap}}/A_0$ based on the following considerations:

- For air curtains with heated indoor air, unheated indoor air, or CAC with indoor air: $f = 10$–40.
- For air curtains with unheated outdoor air, air curtains for cooled rooms, air curtains with long passages, or air curtains for process equipment: $f = 20$–100.

Table 4.1 Factor of dynamic efficiency, E

f	q	E	q	E	q	E
≥ 25	1.0	0.28	0.8	0.31	0.6	0.38
20	1.0	0.24	0.8	0.27	0.6	0.34
15	1.0	0.2	0.8	0.23	0.6	0.3
10	1.0	0.15	0.8	0.19	0.6	0.26

$f = A_0/A_s$, $q = G_0/G_g$, $\alpha = 25\text{--}35°$

Table 4.1 lists the average values of the parameter E.
The mass flow of air supplied by the air curtain is

$$G_O = \rho_0 V_0 A_0 [kg/s] \tag{4.5}$$

Air curtain design process includes the evaluation of the relationship between the temperature of the air supplied by the air curtain and the average temperature of the air entering the building through the aperture. Temperature distribution across the air curtain jet significantly differs from the temperature distribution in the free jet cross-section as a result of the different temperatures of air that entrains the air jet from both sides of the jet (Figs. 4.2 and 4.3).

The distribution of the air temperature difference in the air curtain compared to the outdoor air temperature ($\Omega = t_j - t_{out}$) can be calculated using the principle of conservation of heat content in the jet as follows:

$$\Omega = \Delta\theta_{o1} N \cosh^{-1} Y/CX + 0.5 \Delta t_{in}(1 - \tanh Y/CX)(1 - N \cosh^{-1} Y/CX) \tag{4.6}$$

where

N	$= (4/\pi \sqrt{3} \, Cx) \frac{\sqrt{T_\infty}}{\sqrt{T_0}} (1/\varphi)$.
Δt_{o1}	$= t_o - t_{out}$.
Δt_{in}	$= t_{in} - t_{out}$.
C	is an experimental constant ($C \approx 0.1$).
x	$= X/B_o$.
B_o	$=$ width of the air outlet slot.
$\frac{\sqrt{T_\infty}}{\sqrt{T_0}}$	$=$ correction factor for the non-isothermal jet.
T_∞	$=$ average temperature of the environment (K).
φ	$= \sqrt[4]{\zeta} =$ correction factor for the jet impulse.
ζ	$=$ factor of the local resistance of the air outlet nozzle.

Evaluation of the initial air temperature of the jet supplied by the air curtain can be done by integration of the equation describing the heat content in the cross-section at the distance X_A in the range $(-\infty; Y_A)$. The ordinate of point A can be obtained from the equation describing the principle of mass conservation in the cross-section X_A:
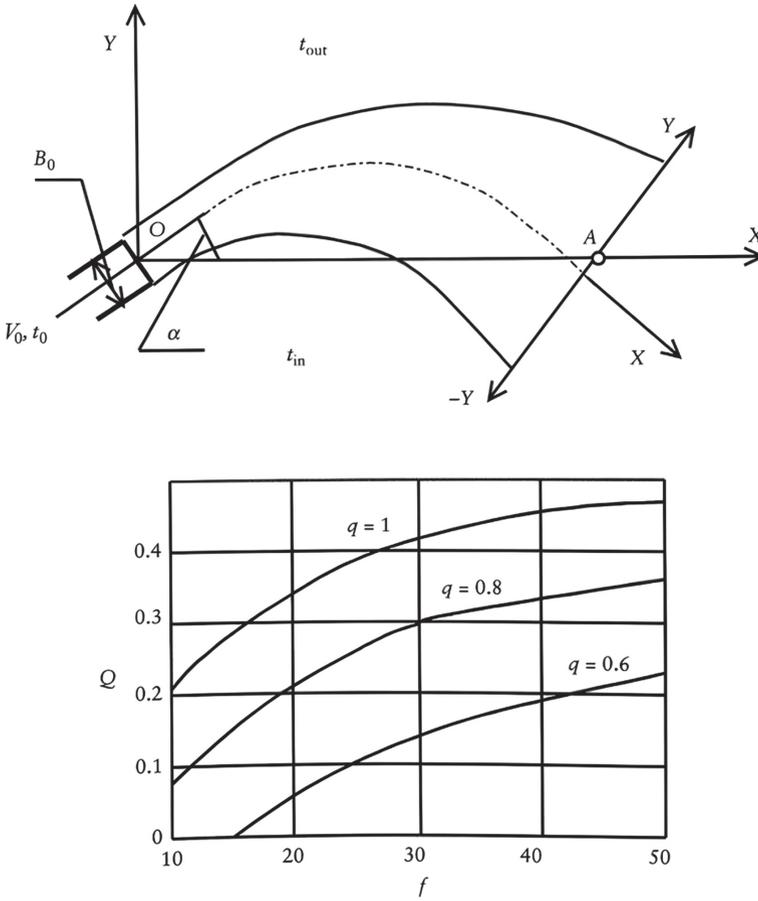
Fig. 4.2 Supporting graphs for calculation of heat losses in air curtains

$$Y_A = C \, X_A \arctan h \, D \tag{4.7}$$

where

$$D = [2 - (\rho_{\text{mix}}q/\rho_0)](1/\varphi)\left(1 \bigg/ \frac{\sqrt{t\infty}}{\sqrt{t_0}}\right) \bigg/ (\rho_{\text{mix}}q/\rho_0)(\, Cx) \tag{4.8}$$

$q = G_0/G_{\text{ap}}$ is the air curtain airflow mass ratio.
G_{ap} is the mass of airflow through the aperture.

Simplified equations can be used for practical applications. For example, in the case of an air curtain using heated indoor air. The required temperature of the supplied air can be calculated using the following simplified equation:
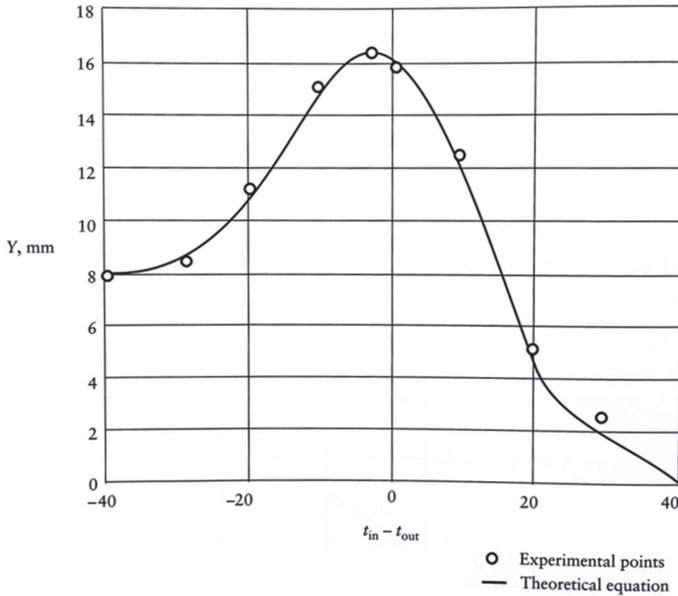
Fig. 4.3 Air temperature distributions in air curtain jet

$$t_0 = t_{out} + m_1(t_{mix} - t_{in}) + m_2(t_{in} - t_{out}) \qquad (4.9)$$

where

t_{in}, t_{out} are design temperatures of the room air and outdoor air, respectively (°C).

t_{mix} is the temperature of the air mixture coming through the aperture of the gate.

m_1 and m_2 are dimensionless factors with values listed in Table 4.2.

Usually, the temperature of air supplied by the air curtain does not exceed 70 °C. The heat generation capacity of the air curtain can be determined using the following equation:

$$Q = c_p G_0(t_0 - t_{ext})[W] \qquad (4.10)$$

where

c_p = heat capacity of air 1004 J/kg (°C).
t_{ext} = air temperature entering the air curtain unit (°C).
t_0 = air curtain supply air temperature (°C).

If the calculated value of $t_{mix} \leq t_{in}$, an additional amount of heat Q_{add} (kJ/shift) should be supplied by an additional heating system to compensate for heat losses through the aperture. This additional amount of heat, Q_{add}, can be determined using

Table 4.2 Values of the factors m_1 and m_2

f	q	m_1	m_2	q	m_1	m_2	q	m_1	m_2
10	1	2.0	–0.7	0.8	2.0	–0.6	0.6	2.0	–0.5
15	1	2.5	–1.0	0.8	2.4	–0.9	0.6	2.4	–0.8
20	1	2.9	–1.3	0.8	2.8	–1.2	0.6	2.8	–1.0
30	1	3.5	–1.8	0.8	3.4	–1.6	0.6	3.4	–1.5
40	1	4.1	–2.2	0.8	4.0	–2.0	0.6	4.0	–1.8
50	1	4.6	–2.5	0.8	4.5	–2.4	0.6	4.4	–2.2
60	1	5.1	–2.8	0.8	5.0	–2.6	0.6	4.9	–2.4
75	1	5.5	–3.0	0.8	5.4	–2.8	0.6	5.3	–2.6
90	1	5.9	–3.1	0.8	5.8	–2.9	0.6	5.7	–2.8
≥ 100	1	6.0	–3.2	0.8	5.9	–3.1	0.6	5.8	–3.0

$f = A_0/A_s, \; q = G_0/G_g$

$$Q_{add} = C_p(t_{in} - t_{mix}) \times (\rho_0 \times L_0/2) \times [(2 \times \rho_0/q \times \rho_{mix})$$
$$+ 0.6\sqrt{f} - 1)] \times n \times \tau, \, \text{kJ/shift} \tag{4.11}$$

where

L_0 = the flow of the air supplied by the air curtain, m^3/s.
n = number of episodes of the gate being open per 8 h shift.
τ = duration of the gate being open, s.

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Chapter 5
Design Methods and Examples of Calculations

5.1 Air Curtain that Supply Heated Indoor Air (ACHI)

These air curtains may be double-sided with a horizontal air supply (Fig. 5.1), or a single-sided with horizontal or vertical air supply. In all these cases, the air curtain is formed by a linear jet discharged at an angle toward the side of the aperture with a higher air pressure.

Examples of Calculation of the Parameters of ACHI

The main initial value for calculating of the air curtain is the difference in air pressure on both sides of the opening protected by the curtain described in detail in Appendix A.

Example 5.1
Initial data

Dimensions of the gate: 6×4 m (height).
Outdoor air temperature: tout $= -5$ °C.
Indoor air temperature: tin $= 20$ °C.
Wind velocity w $= 5$ m/s.
The design differential pressure can be assumed equal to the one calculated in example A1 (Appendix A): $\Delta P = 14.84$ Pa.

Calculation Procedure

Values of f can be selected from the range between 20 and 40 and q from the range between 0.6 and 0.8.

We select the following values of these parameters:

$$f = 30; q = 0.8$$

which implies an air curtain discharge aperture A_s
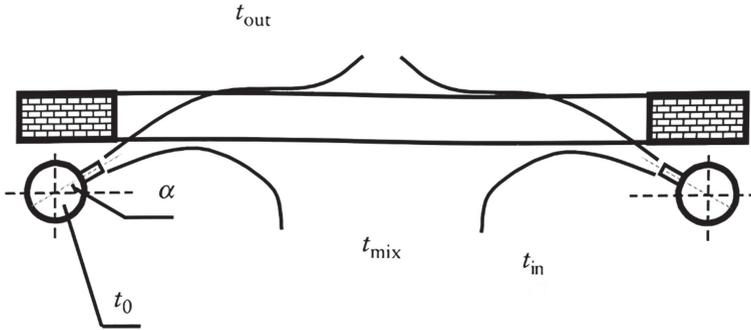
Fig. 5.1 Schematic of shutter-type air curtains

$$A_s = \frac{6 \times 4}{30} = 0.8 \text{ m}^2$$

The aperture height equals the height of the gate (4 m), which yields the width of the air supply slot.

$$b_0 = \frac{0.8}{8} = 0.1 \text{ m}$$

The discharge angle of the air curtain jet is

$$\alpha = 30° (i.e., \ \sin \alpha = 0.5) t_{mix} = t_{in} = 20 °C$$

According to data listed in Table 4.1 (p. 16), the efficiency factor E is

$$E = 0.31$$

Based on the given value f, the values of the factors m_1 and m_2 are determined from the data in Table 4.2 as

$$m_1 = 3.4; \ m_2 = -1.6$$

To determine the supply temperature of the curtain, t_0, and the thermal capacity, Q, we use:

$$t_{mix} = t_{in} = 20°C$$

The parameters of the air curtain can now be determined.
The air supply velocity is calculated according to Eq. 4.4:

$$V_o = \left(\frac{14.84 \times 30}{2 \times 1.1 \times 1.2 \times 0.31} \right)^{0.5} = \frac{23.3 \text{ m}}{\text{s}}$$

The flow of the air supplied by the air curtain is determined according to Eq. 4.5:

$$L_0 = 23.3 \times 0.8 = 18.7 \text{ m}^3/\text{s}$$

The supply air temperature of the air curtain is determined from Eq. 4.9:

$$t_0 = -5 + 3.4(20 + 5) - 1.6(20 + 5) = 40\,°\text{C}$$

The heat capacity of the air curtain is determined according to Eq. 4.10:

$$Q = 1.01 \times 1.13 \times 18.7(40 - 20) = 426.8 \text{ kW}$$

The air curtain fan is to be provided with an airflow-controlling device, e.g., variable-frequency drive or a multispeed motor, to change an airflow rate based on the inside and outside pressure difference: $Q \sim \Delta P^{0.5}$.

Example 5.2
Initial Data
   Dimensions of the gate: 3×3.4 m (height).
   Outdoor air temperature: tout $= -28\,°\text{C}$.
   Indoor air temperature: tin $= 16\,°\text{C}$.
   Wind velocity w $= 0$ m/s
   Number of episodes of the gate being open per 8 h shift: $n = 10$
   Duration of the gate being open: $\tau = 180$ s
   The design differential pressure can be assumed equal to the one calculated in the example A2 (Appendix A): $\Delta P = 3.66$ Pa.

Calculation Procedure

1. Values of f can be selected from the range between 20 to 40 and q from the range between 0.6 and 0.8 We select the following values of these parameters: $f = 30$; $q = 0.8$.
2. The efficiency factor E, according to Table 4.1 is: $E = 0.27$.
3. The initial velocity is calculated according to the formula:

$$V_0 = \left(\frac{3.66 \times 20}{2 \times 1.1 \times 1.2 \times 0.27} \right)^{0.5} = 10.1 \text{ m}^2$$

4. The flow of the air supplied by the air curtain is determined according to the formula:

$$L_0 = \frac{10.1 \times 3 \times 3.4}{20} = 5.15 \text{ m}^3/\text{s}[181.9\, cu\, ft/s]\left(18{,}544 \text{ m}^3/\text{h}\right),$$

or

$$G_0 = 1.14 \times 10.1 \times 0.51 = 5.87 \text{ kg/s}$$

The aperture height equals the height of the gate, 3.4 m. Therefore, for the two-sided air supply ($l = 2H$), the width of the air supply slot will be equal to:

$$b_0 = \frac{3 \times 3.4}{20 \times 2 \times 3.4} = 0.075 \text{ m}$$

5. The supply temperature of the air curtain is determined from Eq. 4.10:

$$t_0 = -28 + 2.8(14 + 28) - 1.2(16 + 28) = 3.8°C$$

$$L_0 = \frac{10.1 \times 3 \times 3.4}{20} = 5.15 \text{ m}^3/\text{s} \left(18{,}544 \text{ m}^3/\text{h}\right)$$

where $m_1 = 2.8$; $m_2 = -1.2$.

6. Assuming that the temperature of the mixed air entering the building through the aperture is equal to $t_{mix} = 14 \,°C$, the heat capacity of the air curtain will be equal to

$$Q = 1.01 - 5.87(36.8 - 14) = 135.2 \text{ kW}$$

7. The additional amount of heat, Q_{add}, that will be required to compensate for heat losses through the aperture will be

$$Q_{add} = 1.01(16 - 14) \times ((1.14 \times 5.15)/2) \times \left((2 \times 1.14)/(0.8 \times 1.23) + 0.6\sqrt{20} - 1\right)$$
$$\times 10 \times 180 = 42{,}759 \text{ kJ/shift}$$

Table 5.1 summarizes these two and some other examples of ACHI calculations.

Table 5.1 Examples of ACHI air curtains calculations

Gate size, m B × H	t_{out}, °C	t_{in}, °C	t_{mix}, °C	Wind, m/s	ΔP, Pa	f/q	b_0, m	V_0, m/s	L_0, m³/s	t_0, °C	Q, kW
6 × 4	−5	20	20	5	14.84	30/0.8	0.1	23.3	18.7	40.0	426.8
3 × 3.4	−28	16	14	0	3.66	20/0.8	0.075	10	5.15	36.8	135.2
3.6 × 3.6	−20	16	14	3	7.55	20/0.6	0.09	13.0	8.42	29.6	179.3
3 × 3	−20	18	14	0	2.8	20/0.6	0.075	8.0	3.6	37.2	96.1

5.2   Air Curtains that Supply Unheated Indoor Air (ACUHI)

Examples of Calculation of the Parameters of the Air Curtains with Unheated Indoor Air (ACUHI)

The methodology of calculation of characteristics of ACUHI air is like the one described in the Sect. 5.1.

Experience shows that for these types of systems, the value of f can be selected from the range between 40 and 200 and the value of q, from the range between 0.8 and 1.0.

The temperature of the airflow mixture passing through the aperture can be calculated using Eqs. 5.4–5.10.

The values of coefficients m1 and m2 can be obtained from the data listed in Table 4.2.

When the air intake for the air curtain is in the upper zone of the space, and there is no better information available, one can assume the supply air temperature to be equal to $t_0 = t_{in} + 2\,°C$.

Calculate the additional amount of heat, Q_{add}, that will be required to compensate for heat losses through the aperture.

Determine the width of the air supply slot, the cross section of the air curtain duct, and the required power of the air supply fan.

Determine the consumption of thermal energy required for heating due to outside air entering the building through the open gate per shift flowing in the with and without per shift through an open gate with and without ACUHI.

Example 5.3
Initial data

Dimensions of the gate: 4×4 m (height).
Outdoor air temperature: $t_{out} = -15\,°C$.
Indoor air temperature: $t_{in} = 20\,°C$.
Wind velocity $w = 3.5$ m/s
Number of episodes of the gate being open per 8 h shift: $n = 10$
Duration of the gate being open: $\tau = 240$ s.

The design differential pressure can be assumed equal to the one calculated in example A4 (Appendix A): $\Delta P = 4.67$ Pa.

Calculation Procedure

Select the values: $f = 40$, $q = 0.8$.

Determine the value of the dynamic efficiency coefficient E according to Fig. 4.1: E = 0.31.

The air supply velocity is calculated as

$$V_0 = [(4.67 \times 40)/(2 \times 1.1 \times 1.2 \times 0.31)]^{0.5} = 15.1 \text{ m/s}$$

The flow of the air supplied by the air curtain is calculated as

$$L_0 = (15.1 \times 4 \times 4)/40 = 6.04 \text{ m}^3/\text{s} \left[21{,}744 \text{ m}^3/\text{h}\right]$$

Select the air curtain with the air supply at the bottom and the air intake from the upper zone of the space. Assume that two air curtain units are used. For this specific case, the width of the air supply slot will be equal to

$$b_0 = (4 \times 4)/(40 \times 4) = 0.1 \text{ m}$$

According to Table 4.2: $m_1 = 4.0$ and $m_2 = 2.0$.

Since the air intake is in the upper zone of the space: $t_0 = 20 + 2 = 22$ °C.

The temperature of the air mixture flowing through the aperture with the air curtain operated is calculated as

$$t_{\text{mix}} = 22 - (1 - 4 + 2)(-15) + 2 \times 20/4.0 = 11.8 \text{ °C}$$

The additional amount of heat, Q_{add}, that will be required to compensate for heat losses through the aperture is calculated as

$$\begin{aligned} Q_{\text{add}} &= 1.01 \times (20 - 11.8) \times (6.04 \times 1.24)/2 \\ &\quad \times \left((2 \times 1.2)/(0.8 \times 1.24) + 0.6\sqrt{40} - 1\right) \times 10 \times 240 \\ &= 387{,}060 \text{ kJ/shift} = 107.5 \text{ kWh/shift} \end{aligned}$$

At design conditions, the consumption of thermal energy required for heating due to outside air entering the building through the open gate per shift without air curtain is calculated as

$$Q_o = C_P G_o(t_{\text{in}} - t_{\text{out}})n\, T, \text{ kJ/shift}$$

where

$$G_o = \frac{2}{3}\mu_0 \frac{A_0}{2} \times \left[2\rho_{\text{ext}}g \frac{(\rho_{\text{out}} - \rho_{\text{in}})H}{2}\right]^{0.5}, \text{ kg/s}$$

$$G_o = \frac{2}{3} \times 0.8 \left(\frac{4 \times 4}{2}\right) \times \left[2 \times 1.37 \times 9.8\,(1.37 - 1.2) \times \frac{4}{2}\right]^{0.5} = 12.9 \text{ kg/s}$$

$$Q_o = 1.01 \times 12.9 \times (20 + 15) \times 240 \times 10 = 1{,}094{,}436 \text{ kJ/shift}(304 \text{ kWh/shift})$$

Over the heating season, the consumption of thermal energy required for heating due to outside air entering the building through the open gate per shift without air curtain (assuming an average outside air temperature $t = -5$ °C, a heating season duration of 200 days, and building operations with two shifts (16 h/day):

$$G_0^{\text{ave}} = \frac{2}{3}0.8\frac{4 \times 4}{2} \times \left[2 \times 1.32 \times 9.8(1.32 - 1.2)\frac{4}{2}\right]^{0.5} = 10.6\,\text{kg/s}$$

$$Q_{\text{add}} = 1.01 \times (20 - 11.8) \times (6.04 \times 1.24)/2$$
$$\times \left((2 \times 1.2)/(0.8 \times 1.24) + 0.6\sqrt{40} - 1\right) \times 10 \times 240$$
$$= 387{,}060\,\text{kJ/shift} = 107.5\,\text{kWh/shift}$$

$$G_0^{\text{year}} = 1.01 \times 10.6 \times (20 + 5) \times 10 \times 240 \times 2 \times 200$$
$$= 256{,}944{,}999\,\text{kJ/year}\ (71{,}373\,\text{kWh/year})$$

The consumption of thermal energy through the aperture protected by the air curtain during the heating period is $= Q_{\text{AC}}^{\text{year}} 23{,}280$ kWh/year, which is 3.07 times less than without the air curtain.

Annual savings of thermal energy per opening:

$$\Delta Q = Q_0^{\text{year}} - Q_{\text{AC}}^{\text{year}} = 71{,}373 - 23{,}280 = 48{,}093\,\text{kWh/year}$$

The required airflow to be supplied by the air curtain can be achieved by reducing the supply slot area (width). Table 5.2 lists the results of calculations demonstrated above and the results of recalculation of the ACUHI characteristics assuming $f = 100$, $q = 1$ and $f = 200$, $q = 1$.

Table 5.2 Summary table of examples of ACUHI calculations for the 4×4 m gate, with the open gate time equal to 2400 s/shift, inside air temperature 20 °C and outside air temperature -15 °C annual heat consumption for heating due to outside air entering the building through the open gate without air curtain equal to 71,373 kWh/year

f and q characteristics	V_0, m/s	b_0, m	L_0 m³/s	Q_{woAC}^{shift} kWh/ shift	Q_{AC}^{shift} kWh/ shift	Q_{AC}^{year} kWh/year
f = 40, q = 0.8	15.1	0.1	6.04	304	109.2	23,280
f = 100, q = 1	25.1	0.04	4.02	304	122.5	26,400
f = 200, q = 1	35.5	0.02	2.84	304	119	25,720

5.3 Combined Air Curtains (CAC)

Regarding the method and recommended assumptions for CAC (Fig. 5.2) calculations,

1. It is recommended that the relative area of the supply air slot for the outside curtain, f_1, be in the range between 12 and 20; and the relative value of q be in the range between 0.8 and 1.0.
2. The ratio of the supply slot areas of the outside and insider air curtains $\kappa = A_{01}/A_{02} = f_2/f_1$ can be selected from the range of values between 2 and 5.
3. The supply air temperature t_{02} (°C) of the inside air curtain jet can be obtained using

$$t_{02} = t_{out} + \sigma_1(t_{mix} - t_{out}) - \sigma_2(t_{in} - t_{out}) \tag{5.1}$$

where the coefficients σ_1 and σ_2 can be determined using the data listed in Table 5.3.

4. If the value of calculated supply air temperature for the internal air curtain turns out to be higher than 70 °C, the calculation is repeated using the smaller value of $k = A_{01}/A_{02} = f_2/f_1$.
5. By changing parameters k and q, and the corresponding value of t_{02}, the CAC design can be further optimized based on CAC configuration and economic considerations.
6. Evaluate the value of f_κ as

$$f_k = f_1 \frac{k}{k+1} \tag{5.2}$$

7. Evaluate the average density of air, $\rho_{o\kappa}$, kg/m^3 supplied by CAC as

$$\rho_{0k} = \frac{k\rho_{01} + \rho_{02}}{k+1} \tag{5.3}$$
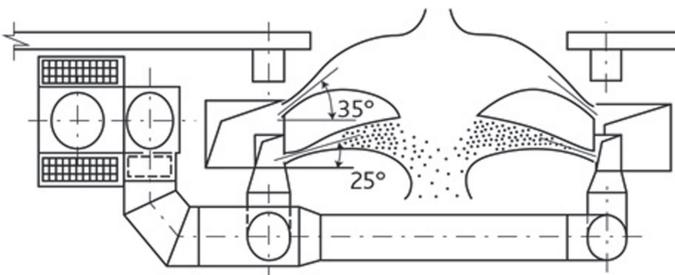
Fig. 5.2 CAC with a double-sided supply of heated and unheated air

Table 5.3 Value of the coefficients σ_1 and σ_2

f_1	q_κ	$\kappa = A_{01}/A_{02} = f_2/f_1$									
		1		2		3		4		5	
		σ_1	σ_2	σ_1	σ_2	σ_1	σ_2	σ_1	σ_2	σ_1	σ_2
10	0.8	3.3	2.0	5.6	3.7	7.7	5.4	9.8	7.1	11.8	8.7
	1.0	2.7	1.6	4.8	3.2	6.5	4.7	8.2	6.2	9.9	7.6
15	0.8	3.6	2.3	6.1	4.0	8.4	5.8	10.5	7.5	12.6	9.2
	1.0	3.1	1.7	5.2	3.4	7.1	5.0	8.9	6.6	10.7	8.0
20	0.8	4.0	2.4	6.5	4.3	8.8	6.1	11.2	7.8	13.4	9.4
	1.0	3.4	1.8	5.5	3.5	7.6	5.3	9.5	6.8	11.5	8.4
≥ 25	0.8	4.5	2.6	7.0	4.6	9.5	6.4	11.8	8.0	14.0	9.7
	1.0	3.7	1.9	6.0	3.6	8.2	5.5	10.3	7.1	12.1	8.6

Note The values of coefficients σ_1 and σ_2 for the intermediate values of f1 and q should be determined using linear interpolation

8. Using the values of f_k and q_k and the data in Table 4.1, evaluate the coefficient of dynamic efficiency E.

   Determine the internal and external CAC air curtains supply air velocity as

$$V_0 = \left(\frac{\Delta\rho \times f_x}{2\beta \times \rho_{0x} \times E} \right)^{0.5} \tag{5.4}$$

9. Determine the relative area of the supply slot of the external air curtain:

$$f_2 = \kappa f_1 \tag{5.5}$$

10. Determine airflow rates L02 and L01, m³/s, supplied by the internal and external air curtains as

$$L_{02} = \frac{V_0 A_0}{f_2} \tag{5.6}$$

$$L_{01} = \frac{V_0 A_0}{f_1} \tag{5.7}$$

11. Determine the heat capacity of the internal CAC air curtain as

$$Q = C_p \rho_{02} L_{02} (t_{02} - t_{in}) \tag{5.8}$$

Example 5.4
Initial data

This example uses the same initial data that were used in the "Examples of Calculation of the Parameters of the Air Curtains with Unheated Indoor Air (ACUHI)," in Sect. 5.2 (for ACHI).

Calculation Procedure

The following values for CAC calculations are selected: $f_1 = 15$; $q_к = 1.0$; $к = 4$.

1. From Table 5.3, $\sigma_1 = 8.9$; $\sigma_2 = 6.6$.
2. The supply air temperature for the internal air curtain is determined to be:

$$t_{02} = -28 + 8.9(14 + 28) - 6.6(16 + 28) = 55.4\,°\tilde{C}$$

Since the value is below 70 °C, it can be used for further calculations.

3. Determine $f_к$, $\rho_{0к}$ and E:

$$f_k = 15\frac{4}{4+1} = 12$$

$$\rho_{0к} = (4 \times 1.2 + 1.07)/5 = 1.17\,\text{kg/m}^3$$

$$E = 0.17$$

4. The supply air velocity for internal and external air curtains is calculated as

$$V_0 = \left(\frac{3.66 \times 12}{2 \times 1.1 \times 1.17 \times 0.17}\right)^{0.5} = 10\,\text{m/s} < 25\,\text{m/s}$$

5. Determine the relative area of the internal air curtain slot:

$$f_2 = 4 \times 15 = 60$$

6. Determine the airflow rates supplied by the internal and external air curtains:

$$l_{02} = \frac{10 \times 3.6 \times 3.6}{60} = 2.16\,\text{m}^3/\text{S}$$

$$l_{01} = \frac{10 \times 3.6 \times 3.6}{15} = 8.64\,\text{m}^3/\text{S}$$

7. Determine the heat capacity of the internal CAC air curtain:

$$Q = 1.01 \times 1.07 \times 2.16(55.4-16) = 92\,\text{kW} \qquad (5.9)$$

Note that with the same initial data, the heat capacity of the internal CAC is 2 times smaller than for the ACHI, calculated in the example in Sect. 5.2.

8. The widths of the supply air slots for the two-sided configuration of the lateral CAC are:

   for the external air curtain: $b_1 = 0.12$ m.
   for the internal air curtain: $b_2 = 0.03$ m.

5.4 Air Curtains with Recirculation of Unheated Outdoor Air (ACUHO)

Unheated outdoor air curtains can be designed to create air circulation either in the area restricted to the vicinity of the gate (Fig. 5.3) or in the larger building volume. Close circulation is preferable if the purpose is to protect the building from the air coming through the aperture; open circulation will allow heat exchange between the building air and the air of the circulation zone due to the lack of circulation zone protection by an air jet and a significant turbulent exchange between this zone and the rest of the building.

Calculation of ACUHO Parameters

1. Values of f can be selected from the range between 20 and 50.
2. Using Table 5.3, determine the critical value of the supply air jet angle of the αcr and the critical relative distance from the nozzle, $\bar{h}_{cr}$, that creates air circulation
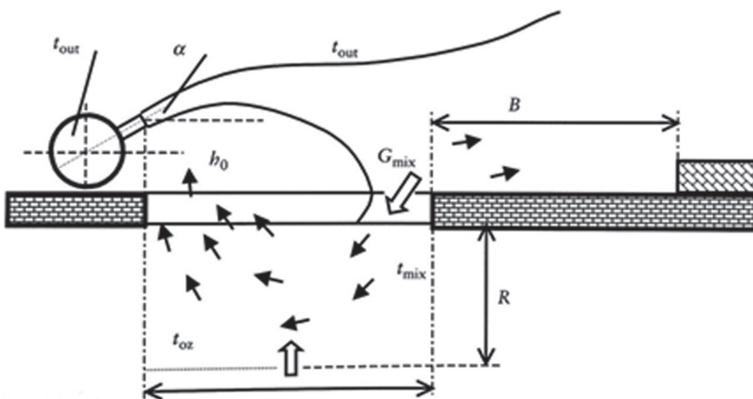
Fig. 5.3 Schematic with a horizontal cross section of air curtain using the outside unheated air (ACUHO)

in the vicinity of the gate. The optimal α and $\overline{h}_{\text{nozzle}}$ values are: $\alpha \geq \alpha\text{cr}$ and $\overline{h}_{\text{cr}}$ $\leq \overline{h}_{\text{nozzle}} \leq 0.25$.

3. Calculate the width of the supply air nozzle:

$$b_0 = A_0/(f_k) = B/f$$

4. For the selected value f and using Table 5.4, determine the values of parameters l1 and l2 and the distant from the edge of the supply air nozzle:

$$h = \overline{h}_{\text{nozzle}} \, B \tag{5.10}$$

5. Determine the dynamic efficiency of the ACUHO air curtain:

$$E = 0.5(\sin \alpha + Ry) \tag{5.11}$$

$$Ry = 0.05 + 0.3 \sin \alpha + (1 - \sin \alpha)h \tag{5.12}$$

6. Determine the temperature of air, t0, and temperature of the air supplied by the air curtain, and the air density, ρ0, assuming that outside air is heated in the air curtain ductwork by 1 °C:

$$t_0 = t_{\text{out}} + 1 \,°\text{C} \tag{5.13}$$

7. Determine the supply air velocity V_0, m/s:

$$V_0 = \left(\frac{\Delta \rho f}{2 \beta \rho_0 E} \right)^{0.5} \tag{5.14}$$

8. Determine the air flow rate L_0, m³/s supplied by the air curtain:

$$L_0 = V_0 A_0/f \tag{5.15}$$

Table 5.4 Value of the parameters for ACUHO air curtains (l_1, l_2, $\alpha_{\text{кр}}$, $\overline{h}_{HKp}$)

f	l_1	l_2	$\alpha_{\text{кр}}$, deg	$\overline{h}_{HKp}$
10	0.59	0.20	10	0
20	0.47	0.21	10	0
30	0.40	0.22	15	0.2
40	0.36	0.23	20	0.25
$\geq$50	0.32	0.24	25	0.25

9. Determine the temperature t_{mix} of the air mixture entering the circulation zone:

$$t_{mix} = t_{out} + l_1(t_0 - t_{out}) + l_2(t_{in} - t_{out}) \qquad (5.16)$$

10. Calculate the parameters r_1 and r_2:

$$r_2 = \frac{1.42 \times l_2}{0.42 + l_2} \qquad (5.17)$$

$$r_1 = \left(1 - \frac{r_2}{1.42}\right)l_1 \qquad (5.18)$$

11. Determine the heat losses Q_{loss}, kJ/shift, through the aperture protected by the ACUHO:

$$Q_{loss} = 0.42 \times C_p \times G_{mix}[t_{int} - t_{out} + (r_1(t_0 - t_{out}) - t_{mix} + t_{out})/r_2] \times n \times \tau \qquad (5.19)$$

These heat losses can be compensated by the building heating system or local heaters.

12. Calculate the size of the size of the zone R, m, in the vicinity of the aperture, within which air temperature will be between t_{mix} and t_{in}

$$R = 1.3B \qquad (5.20)$$

Example
Initial data
The pressure difference across the aperture at specified outside air conditions: $\Delta P_g = 4.25$ Pa.
Dimensions of the gate: 6×4 m (height).
Outdoor air temperature: $t_{out} = -12\,°C$.
Building air temperature: $t_{in} = 20\,°C$.
Wind velocity w $= 0$ m/s

Calculation Procedure
Select the value of $f = 40$.
According to Table 5.4, the value of the $\bar{h}_{cr}$ and αcr are: $\alpha cr = 20°$ and $\bar{h}_{cr} = 0.25$.
Based on construction limitations, we select $\alpha = 24°$ and $\bar{h}_{cr} = 0.25$.
The width of the supply air nozzle is calculated as

$$b_0 = \left(\frac{4 \times 4}{40 \times 4}\right) = 0.1\,m$$

From Table 5.4, l1 $= 0.36$ and l2 $= 0.23$.

The distance from the edge of the supply air nozzle is calculated as

$$h = 0.25 \times 4 = 1 \text{ m}$$

The coefficient of dynamic efficiency is calculated as

$$Ry = 0.05 + 0.3 \sin24 + (1 - \sin24)0.25 = 0.318$$

$$E = 0.5x(\sin24 + 0.318) = 0.363$$

The initial temperature and density of the air supplied by the air curtain are calculated as

$$t_0 = -12 + 1 = -11°C; \ \rho_0 = 1.35 \text{ kg/m}^3$$

The supply air velocity is calculated as

$$V_0 = \left(\frac{4.25 \times 40}{2 \times 1.1 \times 1.35 \times 0.363} \right)^{0.5} = 12.6 \text{ m/s [41.3 ft/s]} < 25 \text{ m/s}$$

The supply airflow rate is calculated as

$$L_0 = \frac{12.6 \times 4 \times 4}{40} = 5.04 \text{ m}^3/\text{s} \ \left(18,144 \text{ m}^3/\text{h}\right)$$

Temperature and density of the air mixture are calculated as

$$t_{\text{mix}} = -12 + 0.36(-11 + 12) + 0.23(20 + 12) = -4.3°C$$

$$\rho_{mix} = 1.31 \text{ kg/m}^3$$

The mass of the air mixture G_{mix} is calculated as

$$G_{\text{mix}} = 0.5 \times 1.31 \times 5.04 \left(0.6\sqrt{40} - 1 \right) = 9.22 \text{ kg}/\text{s}$$

Parameters r_1 and r_2 are calculated as

$$r_1 = \left(1 - \frac{0.302}{1.42} \right) 0.36 = 0.23$$

$$r_2 = \frac{1.42 \times 0.23}{0.42 + 0.23} = 0.502$$

Heat losses, Q, kJ/shift, through the aperture are calculated as:

$$Q_{\text{loss}} = 0.42 \times 1.01 \times 9.22 \times \left[20 + 12 + \frac{0.23\,(-11 + 12)a\,4.3 - 12}{0.502} \right] \times 10 \times 180$$

$$= 120{,}465 \text{ kJ/shift}$$

The size of the zone R, m, in the vicinity of the aperture, within which air temperature will be between t_{mix} and t_{in}:

$$R = 1.3 \times 4 = 5.2 \,\text{m}$$

The above calculations show that, in this example, the uncomfortable zone will have a size of 5.2 m, and the heat losses through the aperture protected by ACUHO air curtain are more than 2 times lower than heat losses without an air curtain.

5.5   Air Curtains for Cooled Spaces (ACC)

Airflow between the cold room and the adjacent part of the space is driven primarily by gravitational forces resulting from the difference in the specific gravity of the air between these spaces. The airflow pattern and the rate through the aperture in the absence of the air curtain is determined by the pressure difference across the aperture. Figure 5.4 shows the distribution of static pressure along the height of an aperture not protected by an air curtain and Fig. 5.5 shows an example of the airflow pattern for this situation. In this situation, air circulation occurs in the form of a natural convection. Cold air leaves the "cold room" at the bottom of the opening, and warm air enters at the top. Based on the mass conservation law, there is a neutral zone with no mass transfer across the aperture, located below the middle of the gate's height ($h_0 < H/2$).

When the opening is protected by an air curtain, the pressure distribution along the room height changes and can be described using the diagram shown in Fig. 5.6. In this case, the neutral line is at the top edge of the gate ($h_0 = H$). Note that the average pressure difference in the opening in the presence of ACC is greater than in its absence.

Figure 5.7 shows the airflow circulation in the presence of the air curtain. In this case, circulation occurs predominately due to forced convection. The supply jet of cold air with the initial flow rate of L_0 entrains air from both sides and divides into two parts at the floor level with one part moving back into the cold room and the rest of the air flowing into the adjacent space.

Calculation of the main parameters of the ACC can be done using the supporting schematic shown in Fig. 5.8.

The calculation method is based on the law of momentum conservation applied to the contour ABCDEF around the aperture. Equation 5.21 shows the application
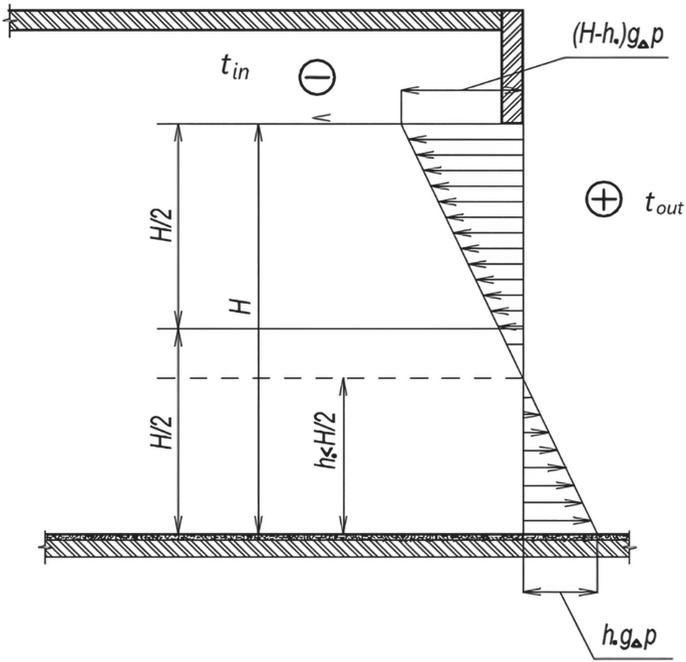
Fig. 5.4 Static pressure distribution along the space height without separation of the cold area from the rest of the space by an air curtain
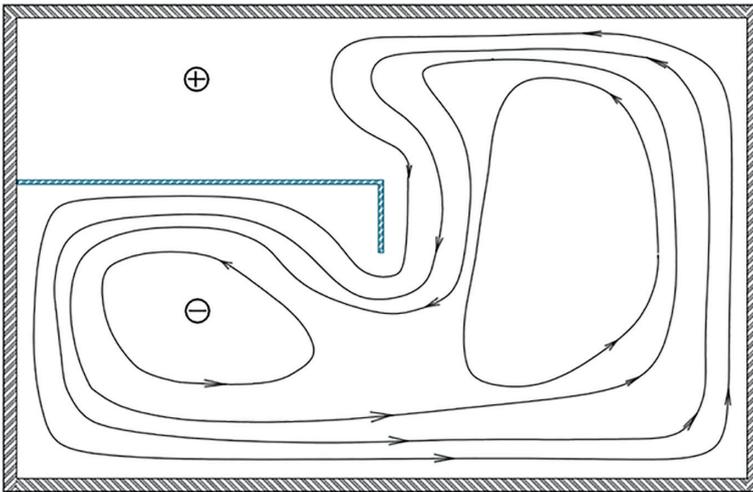
Fig. 5.5 Air circulation in the space with a cold area without air curtain
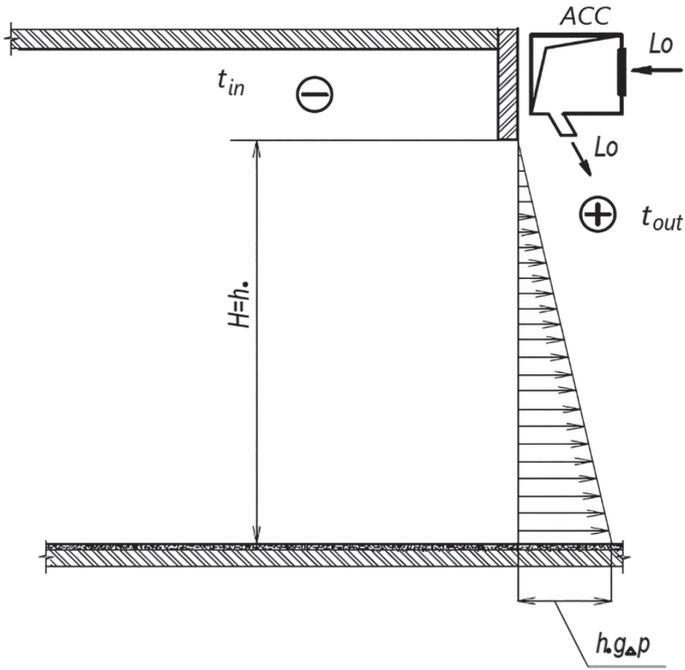
Fig. 5.6 Static pressure distribution along the space height with an air curtain separation of the cold area
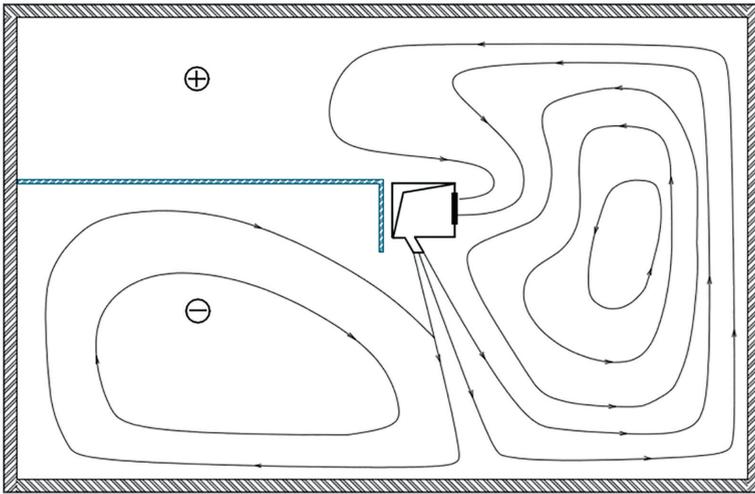
Fig. 5.7 Air circulation in the space with a cold area with an air curtain
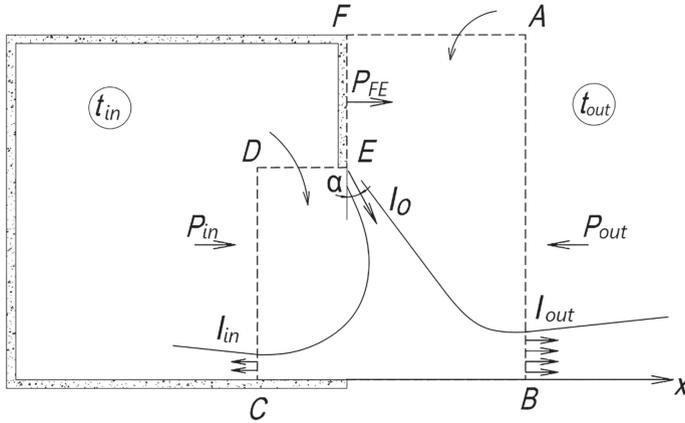
Fig. 5.8 Schematic supporting the calculation of main ACC parameters (change some indices)

of this law in projection on the X axis:

$$I_0\sin\alpha + I_{\text{out}} - I_{\text{in}} = (P_{\text{out}} - P_{\text{in}})HB + (P_{\text{out}} - P_{\text{FE}})(BH_{\text{AB}} - HB) \qquad (5.21)$$

where

I_{o}	is initial jet momentum
I_{in} and I_{out}	are momentums of the part of the jet returning inside the cold space and flowing into the external space
P_{in} and P_{out}	are static pressure inside and outside the cold room
P_{FE}	is the average value of the reactive pressure on the surface of FE
H and B	are the height and the width of the aperture.

The projection of the momentum of the entraining flows through the DE and FA planes can be neglected.

This approach allows derivation of the following main design parameters for ACC.

Pressure difference ΔP, Pa is calculated as

$$\Delta P = (P_{\text{out}} - P_{\text{in}}) = \frac{Hg(\rho_{\text{in}} - \rho_{\text{out}})}{2} \qquad (5.22)$$

where

ρ_{in}, ρ_{out}	are external and internal air density
$g = 9.8$ m	acceleration due to gravity, b_{o}

ACC supply air velocity V_0, m/s is calculated as:

$$V_0 = \left(\frac{\Delta P \times H}{2.2 b_o \rho_{\text{out}} E} \right)^{0.5} \qquad (5.23)$$

where b_o = supply air slot width, m

The coefficient of dynamic efficiency of the ACC is calculated as

$$E = 0.5(\sin \alpha + R) \tag{5.24}$$

where R is a dimensionless characteristic depending on the ratio of ACC geometric parameters.

ACC supply airflow rate, L_0, m/s^3 (for the case with downward air supply) is calculated as

$$L_0 = V_0 b_0 B \tag{5.25}$$

The energy efficiency criterion of ACC, η, where $\eta = 1$ indicates an ideal air curtain (i.e., one that eliminates heat loss/gains through the aperture) and $\eta = 0$ indicates that the aperture is not protected, i $L_0 = V_0 \times b_0 \times B$ s calculated as

$$\eta = 1 - Q_{acc}/Q_{wo} \tag{5.26}$$

where Q_{acc}, Q_{wo} = heat gains through the opening, respectively, protected and not protected by the air curtain, in kW.

Average temperature in air curtain jet near its impingement with the floor area can be calculated as

$$t_{mix} = t_{in} + C_t(t_{out} - t_{in}) \tag{5.27}$$

where C_t can be calculated using the following simplified Eq. 5.6

$$C_t = \frac{2(0.009 f + 0.05)}{0.6\sqrt{f} - 1} \tag{5.28}$$

The values of the coefficients E and C_t for the selected range H/b_0, are given in Table 5.5.

The values of the coefficient p_t, calculated using Eq. 5.23 are also listed in Table 5.5.

The following equation can be used to calculate the values of Q_{acc} and Q_{wo}:

Table 5.5 Efficiency parameters for selected ACC

H/b_0	E	C_t
20	0.15	0.27
30	0.17	0.28
40	0.19	0.29
50	0.2	0.30

Table 5.6 Summaries of calculations of ACC for several typical practical cases

B × H, m	t_{in}, C	t_{out}, C	ΔP, Pa	V_0, m/s	b_0, m	L_0, m^3/s	t_{mix},°C	η
3.6 × 3.6	−3	+16	1.6	11.2	0.09	3.6	2.5	0.67
2.4 × 2.8	−12	+18	1.9	13.3	0.056	1.79	−3.0	0.65
2.2 × 2.5	−23	+20	2.6	15.7	0.05	1.73	−10.0	0.63

$$Q_{acc} = C_p G_{mix}(t_{mix} - t_B) \tag{5.29}$$

$$Q_{wo} = C_p G_{app}(t_H - t_B) \tag{5.30}$$

where

$$G_{mix} = 0.5\rho_{CM} L_0(0.6\sqrt{f} - 1) \tag{5.31}$$

$$G_{app} = 0.7HB\,[\rho_{out}\,(\rho_{in} - \rho_{out})\,H]^{0.5} \tag{5.32}$$

The summarizes example calculations of ACC for several typical practical cases.
The examples listed in Table 5.6 show that ACC allow significant decrease of heat
transfer through the aperture:

$$Q_{acc}\big/Q_{WO} = 1 - \eta = 0.33 - 0.37$$

5.6   Mixing-Type Air Curtain (MAC)

The proposed calculation method is based on the following four steps.

1. The average hourly airflow rate, G_{ent}, kg/h, entering the building through the
    aperture/door due to the pressure difference inside and outside the building can
    be calculated using:

$$G_{ent} = \mu_{ent}(2\rho_{out}\Delta P)^{0.5}A_{app}N_T \tag{5.33}$$

where

μ_{in} is the aperture coefficient
μ_{in} $= 0.1\text{–}0.7$ (Table 5.7)
ρ_{out} $=$ outdoor air density, kg/m^3
ΔP $=$ air pressure difference across the door (see Appendix A for details), Pa
A_{app} $=$ door area, m^2

N   is the number of people entering the building through the door per hour, people/h ($N = 100$–2000 people/h)

T   is the time of the door being open when one person enters the building, s/person ($\tau = 1$–2 s) (see Table 5.7).

2. Airflow mass, G_{mac}, kg/h, to be supplied by the MAC:

$$G_{mac} = G_{ent} \frac{t_{in} - t_{out}}{t_o - t_{mix}} \qquad (5.34)$$

where

t_{mix}, t_0 are air temperatures in the curtain jet and supply air temperature, °C;
t_{in}, t_{out} = air temperatures inside and outside the building, respectively, °C.

3. Heat supplied by the MAC air curtain, kW, is calculated as:

$$Q_{mac} = G_{mac} C_p (t_{mix} - t_o)/3600 \qquad (5.35)$$

where C_p is air heat capacity, kJ/(kg°C).

4. The minimum required velocity V_0, m/s of the air supplied by the MAC, that ensures uniform mixing of the AC jet with the outside airflow entering the building through the door is calculated as:

$$V0 \geqslant K_c (\Delta P \times f/\Delta P f_{mix})^{0.5} \qquad (5.36)$$

where

K_c is a dimensionless coefficient (see Table 5.7)
F   is the relative width of the supply air slot of the MAC (Table 5.8).

Table 5.7 Protective characteristics of some representative aperture/door designs

Door type	Flow coefficient μ_{ent}	Equivalent door opening time τ, s	Typical ($\mu_{ent} \cdot \tau$) value	K_c
Single	0.7	1.7–2.0	1.3	1.0
Double	0.55–0.65	1.3–1.5	0.84	0.9
Triple	0.4–0.6	1.0–1.2	0.54	0.8
Rotating	0.1	1.3–1.5	0.14	0.5

Table 5.8 Representative f ratios (air supply slot width over the representative door size)

Air supply, f	Downward	Horizontal single-sided	Horizontal double-sided
f	H/b_0	B/b_0	$H/2b_0$
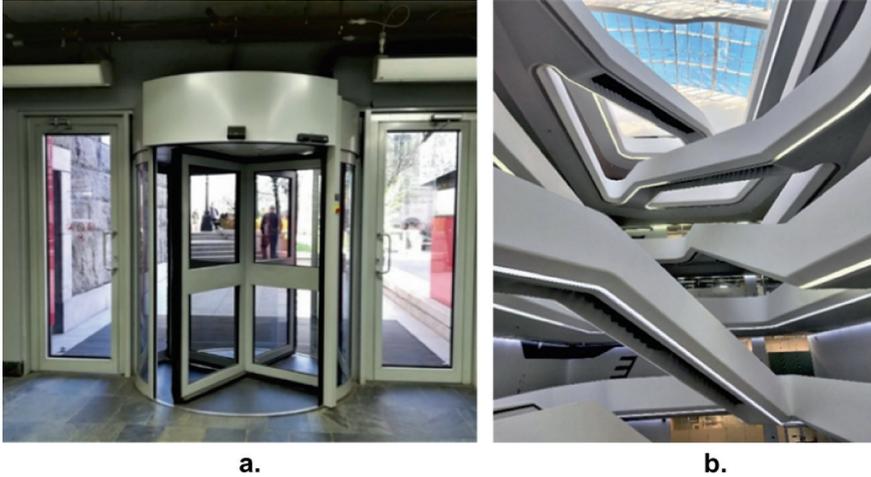
Fig. 5.9 Mixing-type air curtain in the office building with: **a** revolving and single doors with a vertical air curtain supplying air downward and **b** atrium of the office building with a height of 82.4 m

5.6.1   Example

MAC is calculated for an office center with revolving doors (Fig. 5.9).

Initial Data

- The building has an atrium with a height of 82.4 m.
- Outdoor air temperature: $t_{out} = -28$ °C; outdoor air density; $\rho_{out} = 1.44$ kg/m^3.
- Air temperature in the lobby: $t_{in} = 18$ °C; air density in the lobby: $\rho_{in} = 1.21$ kg/m^3.
- Wind speed: W = 4.9 m/s.
- Dimensions of the entrance door: 1.25×2.2 m (height).

5.6.2   Calculation Procedure

The pressure difference is calculated as

$$\Delta P = \left(1.4 \times \frac{4.9 \times 4.9 \times 1.44}{2} + 9.8 \times 0.23 \times 82.4\right)\left(1 - \frac{1}{1 + 0.1}\right) = 18.9\,Pa$$

The amount of air entering the building through the door (Eq. 5.33), assuming: $\mu_{app} = 0.1$; $N = 1500$ person/h; $\tau = 1.3$ s, is calculated as:

$$G_{ent} = 0.1 \times (2 \times 1.44 \times 18.9)^{0.5} \times 1.25 \times 2.2 \times 1500 \times 1.3 = 3956 \text{ kg/h}$$

The required air flow rate to be supplied by the MAC (Eq. 5.34) is calculated as:

$$G_{mac} = 3956 \times \frac{18 + 28}{49 - 18} = 5870 \text{kg/h}$$

The required heating energy supplied by the MAC (Eq. 5.35) is calculated as:

$$Q_{mac} = \frac{5870}{3600} \times 1.01 \times (49 - 18) = 51 \text{ kW}$$

Airflow rate supplied by the MAC: $L_{mac} = 5336 \text{ m}^3/\text{h}$.

Assume installation of a single-sided MAC with air supply along the entire height of the door with the width of the supply slot $b_0 = 0.1$ m (Fig. 5.10).

According to Table 5.8, the value of the parameter f is calculated as:

$$f = \frac{B}{b_0} = \frac{1.25}{0.1} = 12.5$$

The required minimum MAC air supply velocity is calculated as:

Fig. 5.10 Revolving door with a concave profile single-sided MAC

Table 5.9 Example calculations

B × H, m	μ_{ent}	N, person per hour	ΔP, Pa	t_{out}, °C	t_o, °C	G_0 kg/ hr	L_0 m³/h	Q, kW	f	V_0 min, s
1.25 × 2.2 rotating	0.1	1500	18.9	−28	49	5870	5336	51	12.5	7.3
1.0 × 2.0 double	0.6	500	2.8	−20	46	3193	2903	25	20	6.4

$$V_0 = 0.5 \left(18.9 \frac{1.25}{0.1} \right)^{0.5} = 12.5$$

The data listed in Table 5.9 summarize the results of MAC calculations for some representative practical examples.

5.7 Combined Air Curtain with a Lobby (CACL)

The lobby of the combined air curtain has side walls constructed with a variable angle to the axis of the lobby. The first part of the side walls is installed at an outward angle ranging between 15 and 45° to the axis of the lobby and the second part at the angle between 90 and 120° to the first part of the wall (Fig. 5.11).

The outside pair of the air curtain ducts, equipped with supply air nozzles forming incomplete radial jets, is located inside the lobby. The axis of these jets is directed perpendicular to the plane of the opening. Due to negative pressure created between the jet and the wall, the jet becomes attached to the wall. After changing direction due to the angle created between the first and the second parts of the wall, both jets shut the entrance into the lobby by impinging each other at the center of the lobby. The resulting flow turns back toward the building aperture creating two circulation zones. The internal pair of jets supply heated indoor air at an angle between 30 and 60° to the plane of the building aperture plane.

The considered flow scheme is realized under the condition of a sufficient length of the lobby, which depends on the location of the curtain ducts, the angle of adjoining

Fig. 5.11 Combined air curtain with a lobby (CACL): (1) lobby wall, (2) external duct, (3) internal duct, (4) swing gate leaf, (5) sliding gate leaf
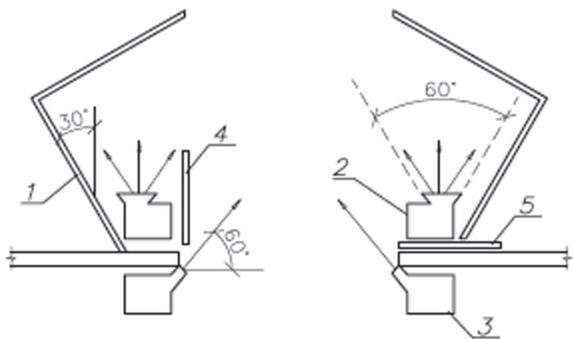
walls and the opening angle of the fan nozzle. For the design shown in Fig. 5.11 (two-sided side feed, contact angle 300, opening angle 600), the length of the lobby must be at least 1.2 times the gate width.

The ratio of airflow rates supplied by the external (L_{ext}) and internal (L_{int}) curtain characterizes the coefficient K is calculated by

$$K = \frac{L_{out}}{L_{in}} \tag{5.37}$$

The optimal value of the coefficient K is determined based on a life-cycle-cost analysis conducted for designs that will assure the same thermal environment in the vicinity of the lobby. For different climates, the optimal value of K is between 2 and 5.

5.7.1   Calculation of the CACL Parameters

1. Determine the pressure difference, ΔP, across the aperture (see Appendix A).
2. Select the value of the parameter f (the recommended range for f is between 10 and 30).

$$\frac{1}{f} = \frac{1}{f_{out}} + \frac{1}{f_{in}} \tag{5.38}$$

where f_{out}, f_{in} are, respectively, the relative area of the air supply slots of the external and internal ducts.

3. Calculate the initial velocity V_0 of air supplied by the curtain using Eq. 4.4 and assuming $E = 0.5$. The initial velocities of air supplied from the internal and external ducts of the CACL are assumed to be the same.
4. Calculate the total airflow rate L_0 supplied by the CACL using Eq. 5.39:

$$L_0 = \frac{V_0 \times B \times H}{f} \tag{5.39}$$

5. Select the value of the parameter K (Eq. 5.37) from the range $K = 2\text{--}5$.
6. Determine the air flow rates for the external and internal air supply slots.

Internal curtain (heated air) is calculated by

$$L_{in} = \frac{L_0}{K + 1} \tag{5.40}$$

outdoor curtain (unheated air) is calculated by

$$L_{out} = L_0 - L_{in} \tag{5.41}$$

7. Determine the required area of the air supply slots by:

$$A_{in} = \frac{A_0}{f_{in}} = \frac{L_{in}}{V_0} \tag{5.42}$$

$$A_{ext} = \frac{A_0}{f_{out}} = \frac{L_{out}}{V_0} \tag{5.43}$$

The temperature of the air t_0 supplied by the internal air curtain is calculated by

$$t_0 = t_{int} + \frac{\rho_{ext} \times K \times \chi \times (t_{int} - t_{ext})}{\rho_0} \tag{5.44}$$

where

t_{in}, t_{out} = respectively, the temperature of the room and outdoor air
°C; ρ_{out}, ρ_0 = the density of the external and supplied air, respectively
kg/m³: χ = coefficient characterizing the proportion of outside air entering the aperture
χ = L_{mix}/L_{out}.

It is recommended to select $\chi = 0.1$. If the calculated value of $t_0 \geq 70$ °C, select a lower value of the coefficient K and repeat the calculation according to the calculation method outlined on p. 33.

8. The thermal energy required for the CACL is calculated using equation

$$Q_{CACL} = C_p \rho_0 L_{in}(t_0 - t_{in}), \ kW \tag{5.45}$$

5.7.2   Example of CACL Parameters Calculation

Initial data
  Dimensions of the gate: 3 × 3 m (height).
  Outdoor air temperature: $t_{out} = -33$ °C.
  Indoor air temperature: $t_{in} = 16$ °C.

Calculation Procedure

1. The selected value of f is $f = 20$
2. Calculate V_0 as:

$$V_0 = \left(\frac{7.3 \times 20}{2 \times 1.1 \times 1.2 \times 0.5} \right)^{0.5} = 10.5 \, \text{m/s}$$

3. Calculate the total supplied airflow rate L_0:

$$L_0 = \left(\frac{10.5 \times 3 \times 3}{20}\right) = 4.725 \text{ m}^3\text{/s } (17{,}010 \text{ m}^3\text{/h})$$

4. Select the parameter value K equal 4
5. Calculate air flow rates supplied by the internal and external parts of the air curtain:

$$L_{in} = \frac{4.725}{4+1} = 0.945 \text{ m}^3\text{/s } (3402 \text{ m}^3\text{/h})$$

$$L_{out} = 4.725 - 0.945 = 3.78 \text{ m}^3\text{/s} (13{,}608 \text{ m}^3\text{/h})$$

6. The area and the width of the air supply slots is calculated as

$$A_{in} = \frac{0.945}{10.5} = 0.09 \text{ m}^2$$

Assuming double-sided internal air curtain

$$b_{in} = \frac{0.09}{2 \times 3} = 0.015 \text{ m}$$

$$A_{out} = \frac{3.78}{10.5} = 0.36 \text{ m}^2$$

Assuming double-sided sides external air curtain:

$$b_{out} = \frac{0.36}{2 \times 3} = 0.06 \text{ m}$$

7. Temperature of the air to supplied by the internal air curtain (Eq. 5.43) assuming that $\rho_0 = 1.1$ kg/m^3

$$t_0 = 16 + \frac{1.47 \times 4 \times 0.1(16+33)}{1.1} = 42 \text{ °C}$$

at

   $t_0 = 42$ °C
   $\rho_0 = 1.12$ кg/m^3.

(The error in calculated and assumed values of ρ_0 is less than 2%, which is satisfactory). The calculated value of t_0 does not exceed 70 °C.

8. The thermal energy required for the CACL calculated using Eq. 5.45:

$$Q = 1.01 \times 1.12 \times 0.945(42-16) = 27.8\,kW$$

5.8 Parameters of Air Curtains for Emission Prevention from Non-Isothermal Industrial Processes (ACPE)

A typical example of an enclosed, non-isothermal process is a drying tunnel, which has openings on both sides for painted pieces suspended on the monorail conveyor entering and exiting the tunnel. The drying process in the tunnel requires that the internal temperature be maintained in the range between 80 and 200 °C. Drying can be carried out using convection (blowing air across painted pieces), radiation (using radiant panels), or by a combination of these two methods.

A typical such drying tunnel solution might be configured to include a local exhaust to evacuate air from the tunnel, in combination with air curtains installed at the tunnel entrance and the exit. To ensure that harmful emissions do not enter the building, the airflow rate of exhaust ventilation is calculated based on the assumption that the negative pressure created in the tunnel is greater than gravity forces created by the temperature difference inside and outside the tunnel. This approach results in a high flow rate of exhausted air that significantly increases the cost of the drying process.

To eliminate or significantly reduce the flow of air through the entrance and exit openings, and to increase the efficiency and effectiveness of local ventilation system, a new concept was developed that employs air curtains as built-in protection installed at both tunnel openings (Fig. 5.12).
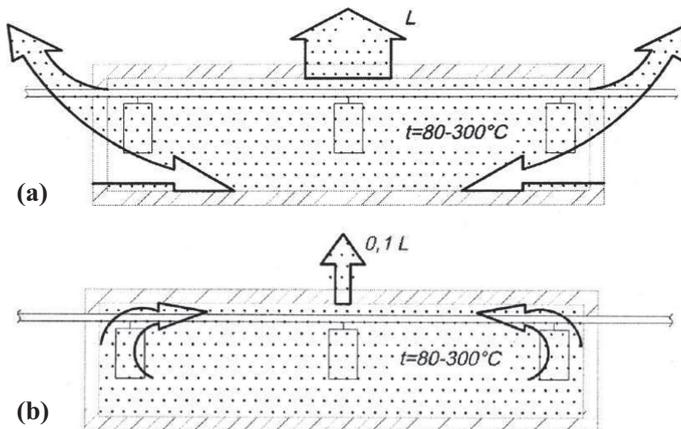
Fig. 5.12 Schematics of ACPE: **a** tradition design, **b** circulating ventilation using air curtains arranged according to the principle of "air curtain in the channel."

Fig. 5.13 Principle of operation of the "curtain in the channel": (1) fan, (2) junction box, (3) channel
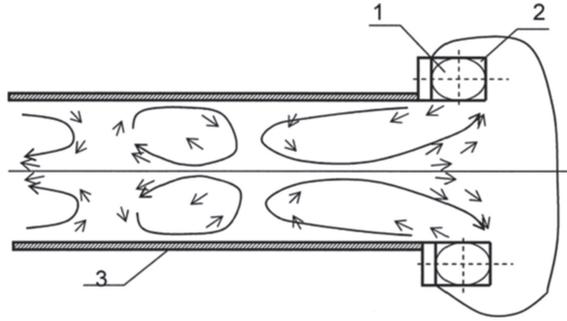
This emission control concept includes air supply from the slots located on both sides of the tunnel at a 5–10° angle to the axis of the tunnel to prevent the incoming air stream. The length of the tunnel is selected to ensure that the supply air is not forced outside. To shorten the length of the lobby, air can be supplied with an incomplete fan jet from a specially constructed outlet; the curtain jet moves along the tunnel, slows down, and turns in the opposite direction at the end of the tunnel, thus "locking" out air intake from the internal building space (Fig. 5.13).

Air curtains can be installed using vertical or horizontal air supply, and single- or double-sided ducts (Fig. 5.14). The length of the tunnel for a single-sided air curtain can be at least 5 times greater than the characteristic dimension of the tunnel $\sqrt{B \times H}$, and for the double-sided air curtain $-2.5 \sqrt{B \times H}$, where B and H are the width and the height of the tunnel. When limited space will not accommodate a tunnel of such length, it tunnel can be shortened by installing air curtains with supply air slots equipped with nozzles with an increased angle of divergence $\beta_N = 30$–$90°$ to create incomplete radial jets.

By installing such air curtains into the process enclosure, both the exhaust airflow rate and the energy required to operate air cleaning can be significantly reduced, and up to 85% of heat contained in the exhaust air and up to 85% of the heat of the purified air can be reused.

Appendix B Includes More Information About the Use of Air Curtains to Enhance the Performance of Process Equipment Enclosures. The Data in Table 5.10 summarize Examples of Calculation Results.

5.9 Air Curtains for Exhaust Hoods Enhancement (ACH)

A canopy hood with an incorporated nozzle installed along the perimeter of the hood can be used to enhance the performance of the typical canopy hood and to reduce the spread of harmful matter from the vicinity of contaminant release to the rest of the building space (e.g., from the operating zone of one or several welding robots (Fig. 5.15), where the process enclosure is not practical [1, 2]. Air supplied through the nozzle installed along the edge of the hood creates steady air curtain protection
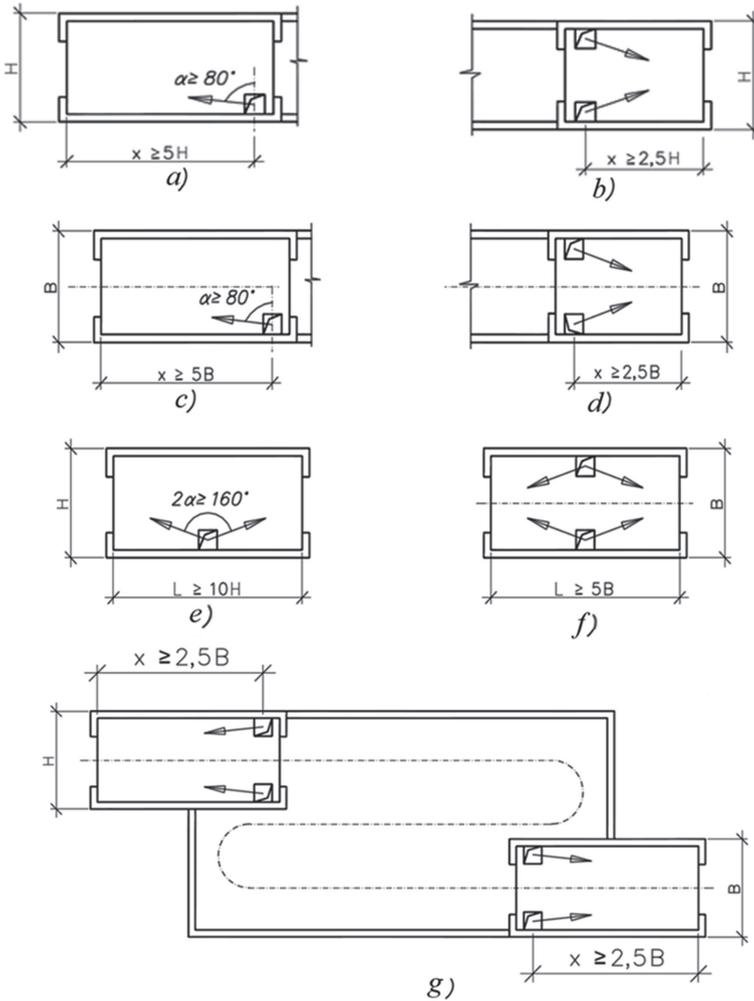
Fig. 5.14 Examples of process enclosure arrangements with air curtains installed in a tunnel: **a** horizontal single-sided, **b** horizontal double-sided, **c** vertical single-sided, **d** vertical double-sided, **e** horizontal single-sided with supply duct having two air supply slots, **f** vertical double-sided with supply duct having two air supply slots, **g** double-sided air curtains with one opening equipped with a vertical air curtain. *Note* X, B, and H are the tunnel length, width, and height, respectively

around the work area that prevents contaminant dispersion especially in the presence of cross drafts created by supply air flows, drafts from the passing vehicles, etc.

Under the negative pressure created by the hood, the air curtain jet turns at or below the level of the contaminant source toward the center (Fig. 5.16). To minimize the supply airflow rate, the nozzle can be equipped with a honeycomb insert that reduces the initial turbulence of the supply jet and thus increases the jet's thrust.

Table 5.10 Examples of air curtain calculations for enclosures over paint drying chamber

Enclosure and process description	HxB, m	L, m	F_{encl} m²	F_{ap} m²	G gr/s	C_{ign} g/m³	$\overline{F_0} = \dfrac{F_0}{F_{encl}}$	E	ΔP Pa	V_0 m/s	L_0 m³s	L_{exh} m³s
Paint drying conveyer chamber, $t_{int} =$ 130 °C, solvent: naphtha	3.24 × 1.7	16	5.5	2.2	2.5	58	1/20	0.66	5.54	9.4	2.58	0.43
Paint drying conveyer chamber, $t_{int} =$ 80 °C, solvent: ethyl alcohol	3.9 × 2.0	27.6	7.8	3.1	5	68	1/30	0.61	4.2	9.7	2.52	0.294
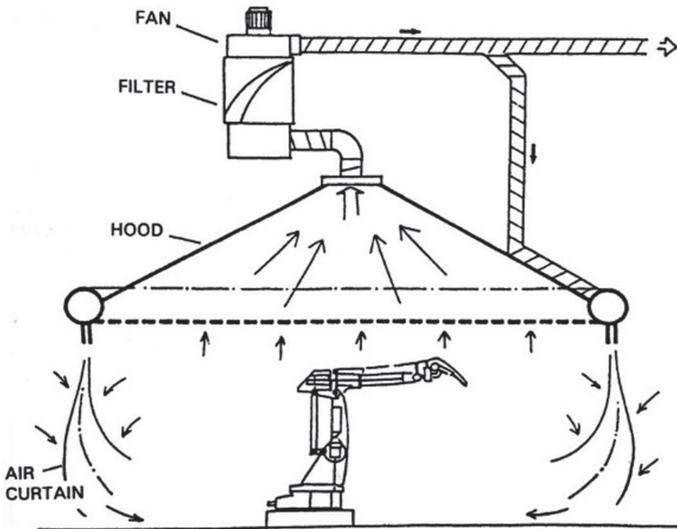
Fig. 5.15 Push–pull hood over welding robot

Figure 5.17 shows a schematic and image of the push–pull hood.
The width of the supply air slot can be calculated by:
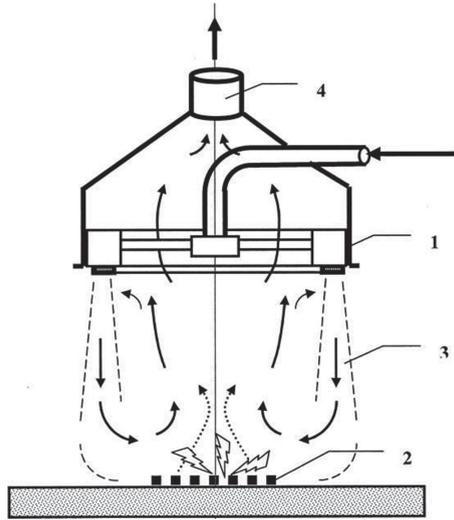
Fig. 5.16 Schematic of the push–pull hood operation of operation of the activated hood (1) supply air nozzles, (2) source of contaminant emission, (3) supply air jet, (4) exhaust duct
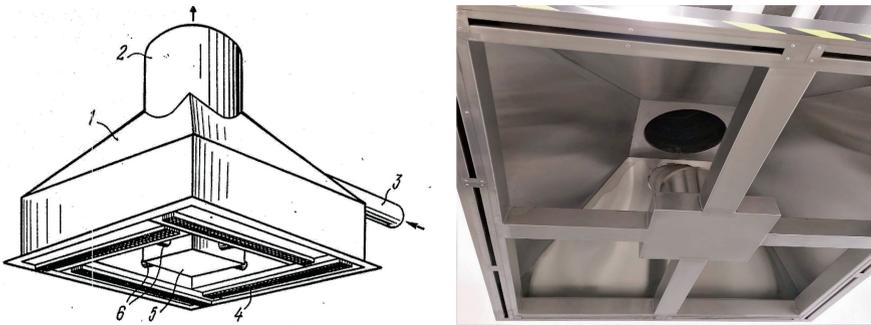
Fig. 5.17 Push–pull hood schematic (left): (1) canopy hood, (2) exhaust duct, (3) supply duct, (4) nozzles with a honeycomb, (5) distribution box, (6) connecting ducts; right: bottom view

$$b = \frac{A/P}{45\left(\frac{A}{PH}\right)^2\left(0.566\sqrt{\frac{H}{b}} - 1\right)^2 - 0.25\left(0.566\sqrt{\frac{H}{b}} + 1\right)^2}. \qquad (5.46)$$

where

b = slot width, m
A = hood cross-sectional area, m^2
P = hood perimeter, m
H = height of the hood above the contaminant source, m.

Table 5.11 Examples of push–pull hood parameters

L × B, m	H, m	b_o, m	V_o, m/s	L, m³/h		
				L_o	L_{ex}	L_{ex}/L_o
1.5 × 1.5	2.0	0.025	0.5	957	2872	3.0
2.0 × 2.0	2.5	0.5	0.5	2042	5105	2.5
0 × 3.0	3.0	0.11	0.5	5055	10,080	2.0
6.0 × 1.5	2.0	0.10	0.5	4830	8514	1.76

The amount of air supply L_0, m³/h:

$$L_0 = \beta\, F_0 V_n / \sqrt{2}, \tag{5.47}$$

where

$$F_0 = bP \tag{5.48}$$

$$B = 0.566 (H/b)^{0.5} \tag{5.49}$$

The amount of air to be exhausted L_{exh}, m³/h:

$$L_{exh} = L_0(1 + B/2) \tag{5.50}$$

The required rate of air leakage to the hood V_n, m/s:

$$V_n = \frac{A}{0.434a} lg\left(\frac{q_0}{q_a}\right) \tag{5.51}$$

A—turbulent diffusion coefficient:

$$A = 0.25\varepsilon^{1/3} 1^{4/3} \tag{5.52}$$

a—distance from the hood edge to the working place, m; q_a—contaminant concentration at the working place, mg/m^3; l—characteristic size, $l = H$, m;

$\varepsilon = AC\eta V^2/2$—amount of kinetic energy dissipated in the room, m^2/s^3; AC—air change per second, s^{-1}; V—velocity of air supply, $\eta = 1.1 - 1.2$—correction factor for the jet momentum. V_n should be assumed to be at least twice the average mobility of the indoor air.

Table 5.11 summarizes representative characteristics of push–pull hoods based on the authors' calculation.

References

1. A. Zhivov, A. Strongin, Application of the momentum conservation method to the overhead push-pull hood design, in *Proceedings of Seminar: Specially Engineered Local Exhaust and Intelligent Exhaust Systems* (Zurich, Switzerland, 2003)
2. A. Strongin, Optimization of local exhaust designs for typical industrial applications, in *Proceedings of the Industry Workshop «Industrial Process and Energy Optimization* (Gettysburg, PA, 2004)

Chapter 6
Conclusions

Unprotected open doorways/apertures of buildings and process enclosures can result in an uncomfortable and unhealthy environment and working conditions, significant heat loss, and reduced usable space near the apertures—especially in buildings located in cold climates. Air curtains are local ventilation devices used to reduce air leakage through the open apertures.

The book has summarized essential information that will facilitate a better understanding of air curtains' performance, of criteria for their selection, and of methods to calculate their technical characteristics. Chap. 2 of this book describes typical designs of air curtains (some mature but underused, and some novel) for use in large commercial and industrial buildings, warehouses, and maintenance facilities to reduce leakage of airflow through dedicated gates/apertures in building and process enclosures. Chap. 3 analyzes the advantages and disadvantages of specific technologies and provides examples of their application:

- The most used air curtains, which supply heated indoor air, are typically installed to service small, infrequently opened doors in mild climates.
- Air curtains that supply unheated internal air offer an optimal combination of low energy consumption, better indoor environment, and relatively low installed cost. However, their application is limited by the complexity of their installation and their need for extended supply air ducts, including some that must be laid in underground channels.
- Air curtains that supply unheated outside air have good airflow shuttering properties and lowest first cost. However, they cannot prevent cold areas in the vicinity of the aperture which results in a loss of usable building space.
- Use of combined air curtains thermal curtains is recommended in regions having a very cold climate.
- Air curtains supplying cold air are recommended for spaces that use artificial cooling, e.g., refrigerating chambers, air-conditioned warehouses, etc. They help

Table 6.1 Initial data for calculations (high-bay warehouse)

Aperture size, m	Design conditions				Operating conditions			
	Extreme		Average over the heating period		t_{in}. °C	Duration of the heating season	Aperture opening mode	
	t_{out}. °C	ΔP. PA	t_{out}. °C	ΔP. Pa			Number of openings per day	Duration the aperture stays open
3.6 × 3.6	−30	5.83	−5	2.33	20	250	20	240

to reduce the load placed on the refrigeration equipment and consequent energy costs and to increase the equipment service interval.

- Air curtains installed with a lobby/tunnel can effectively protect apertures both in large buildings and in enclosures of industrial processes. However, they require additional space and create construction costs to install these tunnels.
- In industrial buildings housing processes that emit harmful contaminants, air curtains can enhance the performance of local exhausts.

The use of the following indicators was proposed for the evaluation and comparison of different types of air curtains (see Chap. 3):

- Dynamic efficiency E
- Energy efficiency criterion η
- Occupational comfort efficiency θ.

To demonstrate how the values of the above criteria differ, calculations were made using Eqs. 3.1 through 4.5 and the input data listed in Table 6.1 for representative types of air curtains:

- air curtains that supply heated indoor air (ACHI)
- air curtains that supply unheated indoor air (ACUHI)
- air curtains with recirculation of unheated outdoor air (ACUHO)
- combined air curtains (CAC).

Table 6.2 summarizes the results of these calculations.

Under the conditions used in this analysis, the use of ACHI yields savings of about 50% of heat compared to the case with unprotected aperture; furthermore, ACHI assures that air temperature crossing the aperture is close to the internal air temperature in the building:

$$t_{mix} = t_{in} = 20°C; \ \theta = 1; \ E = 0.28; \ \eta = 0.49$$

ACUHI saves about 1.5 times more heat compared to ACHI, but provides lower occupational comfort efficiency:

$$t_{mix} = 11°C; \ \theta = 0.82; \ E = 0.28; \ \eta = 0.75$$

Table 6.2 Results of comparative calculations

Air curtain type	Design parameters			Efficiency				
	Initial jet velocity, m/s	The temperature of the air mixture entering the building, °C	Heat loss through the aperture, Q_{AC}. kW	E	η	θ	Annual heat loss through the opening, kWh/year	Annual heat savings kWh/year
No air curtain (unprotected aperture)	0	−30	546	0	0	0	60,000	0
ACHI	15.4	20	282	0.28	0.49	1.0	30,000	30,000
ACUHI	28	11	136	0.28	0.75	0.82	13,700	46,300
ACUHO	14.5	−18	82	0.36	0.85	0.24	9900	50,100
CAC	15	20	140	0.20	0.74	1.0	15,800	44,200

ACUHO offers the best energy savings (67% more than ACHI and 28% more than ACUHI) but the worst occupational comfort efficiency since it results in a cold air zone in the building close to the open aperture:

$$t_{\mathrm{mix}} = -18°C; \ \theta = 0.24; \ E = 0.36; \ \eta = 0.85$$

In terms of energy efficiency, CAC is comparable to ACUHI; in terms of occupational comfort efficiency, CAC is comparable to ACHI:

$$t_{\mathrm{mix}} = t_{\mathrm{in}} = 20°C; \ \theta = 1; \ E = 0.2; \ \eta = 0.74$$

Given the appropriate application and comparing air curtain alternatives using the above criteria, the ultimate selection shall be made for each specific situation based on a life-cycle cost analysis.

Appendix A
Estimating Pressure Difference Across an Aperture

The most important information required for the calculation of air curtain characteristics is the difference in air pressure, ΔP, across the building aperture to be protected:

$$\Delta P = P_{\text{out}} - P_{\text{in}} \tag{A.1}$$

where P_{out}, P_{in} are air pressure, outside and inside the building, respectively.

The value of ΔP depends on the following three factors [1–4].

- Hydrostatic pressure, due to temperature/specific gravity of the air difference between the inside and outside of the building on both sides of the aperture.
- Wind pressure differences between one side of the building and the other, depending upon both the speed and direction of the wind.
- Imbalance of supply and exhaust ventilation systems.

These pressures fluctuate over time, but most of the models developed for building ventilation [5] assume steady-state ventilation, and the input data represent values averaged over time.

This Appendix discusses the methodology to calculate the difference in air pressure, ΔP, across a building aperture to be protected, given that the values of the following parameters are known:

- Wind speed and direction.
- Internal and external air temperatures.
- Location and flow characteristics of all apertures.
- Pressure coefficients for the building envelope and the wind directions under consideration.
- Supply and exhaust ventilation airflow rates.

In practice it is difficult, if not impossible, to determine all these quantities accurately, and therefore simplified calculation methods based on empirical, simplified theoretical methods and network models for multicell/multizone buildings referred to in this section are used [6–8].

The pressure difference, ΔP, across aperture i can be calculated as:

$$\Delta P_i = \Delta P_{hi} + P_w + \left(P_{exh} - P_{sup}\right) \tag{A.2}$$

where $\left(P_{exh} - P_{sup}\right)$ is the pressure difference created by the imbalance of air supply and exhaust created by respective systems fans. For the sake of simplicity, this book omits the discussion of the effect of unbalanced ventilation.

The hydrostatic pressure difference ΔP_{hi} at the center of each aperture can be calculated using:

$$P_{h(i)} = g(\rho_{out} - \rho_{in})h_i \tag{A.3}$$

where

h_i is the vertical distance between the center of the top aperture and the center of aperture i.

ρ_{out}, ρ_{in} are the density of air outside and inside the building, kg/m^3.

e wind pressure, P_{wi} at the aperture i can be evaluated as:

$$P_{wi} = C_i \frac{\rho_{out} W^2}{2} \tag{A.4}$$

where

C_i the aerodynamic coefficient of the ith aperture.

W estimated wind speed, m/s.

Wind pressure at the aperture varies with the wind speed, wind direction, orientation of surfaces of apertures, height of the aperture, and shielding effect of surrounding buildings.

The wind speed for the calculation shall be based on the location's climate conditions.

The C_i value is usually obtained experimentally by testing models of different types of structures in wind tunnels. Values for a wide range of building shapes are cataloged in Supplement No. 3 to the 1960 edition of the National Building Code of Canada, *Handbook of Pressure Coefficients for Wind Loads* [2]. Since the wind speed and consequently the velocity pressure increase with height above the ground, a height factor is applied to the basic pressure based on a height of 10 m in the design of buildings. For example, if a building is 27 m high, the height factors that apply are 0.8 for the first 6 m, 1.0 for the next 6 m, and 1.2 for the top 15 m [2].

The mass of the airflow, G_i, through each open aperture assuming the turbulent flow, can be calculated as:

$$G_i = \mu_i \text{x} \, A_i (2\Delta P_i \times \rho_i)^{0.5}, \text{ kg/s} \tag{A.5}$$

where

μ_i discharge coefficient of airflow through the ith opening, determined by reference data (Table A.1);

A_i the area of the ith aperture area, m^2.

ρ_i air density, kg/m^3.

ΔP_i pressure difference across the ith aperture, Pa calculated using Eq. A.2.

For practical reasons, the pressure difference change along the height of the aperture can be ignored and the resulting airflow rate can be calculated based on the pressure difference at the center of the aperture.

The effect of the supply and exhaust systems imbalance can be accounted for using the following air mass balance equation for the building:

$$\sum G_i \mp \left(G_{\text{exh}} - G_{\text{sup}}\right) = 0 \tag{A.6}$$

The solution of Eq. A.6 for buildings with many apertures and cracks in the building envelope is a complex task that can be accomplished using specialized software tool, e.g., those described in [9]. For a limited number of openings, analytical calculation is possible [10]. The following examples show the implementation of the methodology described here.

A.1 Example: Air Curtain to Be Used in the Doorway of an Industrial Building to Prevent the Penetration of Cold Air into the Building

Initial data

Reference outdoor air temperature $t_{\text{out}} = -5\,°\text{C}$.

Reference wind speed $W = 5$ m/s.

Indoor air temperature: $t_{\text{in}} = 20\,°\text{C}$.

Indoor air relative humidity: $\phi = 60\%$.

The indoor air temperature and humidity are assumed to be uniform across the building space.

The wind speed value is provided for the center of the aperture, and there is no need to apply correction sheltering or height-related coefficients.
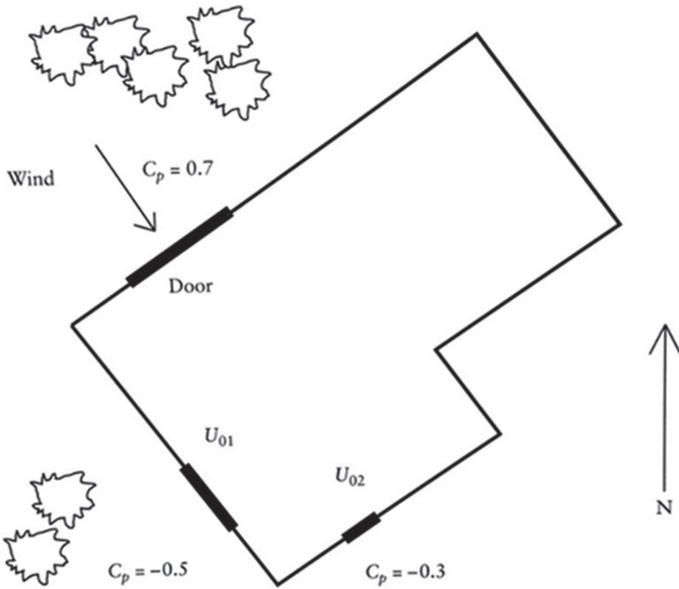
Fig. A.1 Building layout

Table A.1 Characteristics of the building apertures for Example A.1

		Doorway	Aperture 1	Aperture 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 building (Fig. A.1) is 12 m high and has three large openings. The doorway for which the air curtain is designed is used primarily for vehicular traffic, is on the ground level, and has (height × width) dimensions of 4 × 6 m. Table A.1 summarizes the characteristics of the building apertures.

Other than the doorway and two apertures described in Table A.1, the building is airtight (Table A.2).

The general supply and exhaust ventilation airflows are in balance.

The makeup air for local exhausts ($L_{exh} = 4$ m^3/s, $G_{exh} = 4.8$ kg/s) is designed.

Calculation procedure

Mass balance of the airflows through the building envelope:

$$G_{\text{doorway}} + G_{\text{aperture 1}} + G_{\text{aperture 2}} + G_{\text{local exhuats}} = 0 \tag{A.7}$$

where

G_{doorway}, $G_{\text{aperture 1}}$, $G_{\text{aperture 2}}$, and $G_{\text{local exhaust}}$ are the mass of air that passes through the doorway, apertures 1 and 2, and that is evacuated by the local exhaust, respectively

Flow through opening i:

$$G_i = \mu_i A_i [2\,\rho_{\text{out}} \Delta p_i]^{1/2} \text{ kg/s} \tag{A.8}$$

Pressure difference across opening Δp_i:

$$\Delta p_i = p_{i,\text{ out}} - p_i \tag{A.9}$$

Pressure inside building:

$$p_{i,\text{in}} = \text{const} \tag{A.10}$$

Outdoor air density: $\rho_{\text{out}} = 1.32 \text{ kg/m}^3$.
Indoor air density: $\rho_{\text{in}} = 1.2 \text{ kg/m}^3$.
Reference level 0 is in the center of the doorway, i.e., 2 m above the ground level.

Wind pressure
Pressure difference across the doorway and aperture is calculated as:

$$P_w = \frac{1.2 \times 5 \times 5}{2}$$

$$\Delta p1 = 0$$

$$\Delta p_2 = 0 + (h_1 - h_2)g(\rho_{\text{out},2} - \rho_{\text{in}}) + (c_{p2} - c_{p1})\,p_w$$
$$= 0 + (9 - 5.5) \times 9.81 \times (1.32 - 1.2)$$
$$+ \left[[-0.3 - (-0.5)]x\frac{1.2 \times 5 \times 5}{2} \right] \times 16.5$$
$$= 7.42 \text{ Pa}$$

$$\Delta P_{\text{doorway}} = 0 + (h_1 - h_g)\,g(\rho_{\text{out}, 2} - \rho_{\text{in}}) + (c_{p,g} - c_{p,1})\,p_w$$
$$= 0 + (9 - 2) \times 9.81 \times (1.32 - 1.2)$$

$$+ \left[[0.7 - (-0.5)] \times \frac{1.2 \times 5 \times 5}{2} \right] \times 16.5$$

$$= 28.04 \, \text{Pa}$$

The mass balance expressed in Eq. A.7 for this example can be presented as

$$9.86(28.04 - p_{in})^{1/2} = 9.91(p_{in} - 7.42)^{1/2} + 2.48(p_{in})^{1/2} + 4.8$$

$$P_{in} \approx 13.2 \, \text{Pa}$$

The design pressure difference across the doorway is calculated as

$$\Delta P_{doorway} = 28.04 - 13.2 = 14.84 \, \text{Pa}$$

For approximate engineering calculations, in the absence of complete initial data, simplified methods can be used [11, 13].

The pressure difference ΔP in the opening of the gate protected by an air curtain, for a conditionally "sealed" building (i.e., in which there are no other open openings) can be calculated by the formula:

$$\Delta P_{A=\pi r^2} = \frac{gH}{2}(\rho_{out} - \rho_{in}) \tag{A.11}$$

where H is the height of the aperture.

Note that wind pressure does not affect the magnitude for a "sealed" building since an increase in wind pressure leads to an equivalent increase in internal pressure.

Note that the calculation in Example A.2 uses Eq. A.11.

A.2   Example: Air Curtain to Be Used in the Doorway of an Industrial Building to Prevent the Penetration of Cold Air into the Building. The Building is Assumed to Be Airtight, and the Doorway is the Only Aperture in the Building Envelope

Initial data

Reference outdoor air temperature $t_{out} = -28 \, °C$.
Outdoor air density: $\rho_{out} = 1.44 \, \text{kg/m}^3$.
Reference wind speed $W = 5 \, \text{m/s}$.
Indoor air temperature: $t_{in} = 18 \, °C$.
Indoor air density: $\rho_{in} = 1.22 \, \text{kg/m}^3$.

Table A.2 Values of $K1$ and hr depending on the type of building

Building type	$K1$	h_r
Conditionally "sealed"	0	0.5H
The building does not have apertures used for natural ventilation. There are minor leaks, the area of which is significantly less than the gate area	0.2	0.5H
The building has apertures used for natural ventilation that are closed during the cold season, and roof fans	0.5	$h_1 + [h_2/0.25(l_{sup}/l_{exh})^2 + 1]$
The building has apertures used for natural ventilation that are open during the cold season, the area of which is comparable with the area of the gate aperture	0.8	$h_1 + [h_2/(A_{sup}/A_{exh})^2 + 1]$

Terms:
h_1 is the distance from the center of the opening equipped with a curtain to the center of the supply openings, m
h_2 is the distance between the centers of the supply and exhaust openings, m
l_{sup} and l_{exh},—the length of the supply and exhaust apertures, respectively, opened only in the warm season, m
A_{sup}, A_{exh}—is the equivalent area of supply and exhaust apertures, respectively, m^2

Indoor air relative humidity: $\phi = 60\%$.
The indoor air temperature and humidity are assumed to be uniform across the building space.

The air curtain is designed for a doorway located on the ground level with (height x width) dimensions of 3.4×3 m and is used primarily for vehicular traffic.
Calculation procedure using Eq. A.11:

$$.$$ (A.12)

It is believed that there is only one open opening in the building, protected by an AC, building, wind pressure does not affect the magnitude, since an increase in wind pressure leads to an equivalent increase in internal pressure.

In the not well-sealed building with only one aperture, the pressure difference across this aperture can be calculated using Eq. A.12 with approximations calculated in Eqs. A.3 and A.4 and summarized in Table A.2.

$$\Delta P = \frac{gh_r}{2}(\rho_{out} - \rho_{in}) + K_1 P_w$$ (A.13)

where h_r is the reference vertical distance from the center of the aperture to the centers of the supply and exhaust openings (Fig. A.2), calculated using the equations and K_1 correction factors listed in Table A.2.
Note that calculations in Examples A.3 and A.4 use Eq. A.12.

Fig. A.2 Building cross
section showing vertical
distance from the center of
the gate aperture to the
centers of the supply and
exhaust apertures: (1) gate
aperture protected by an air
curtain, (2) supply aperture,
(3) exhaust aperture
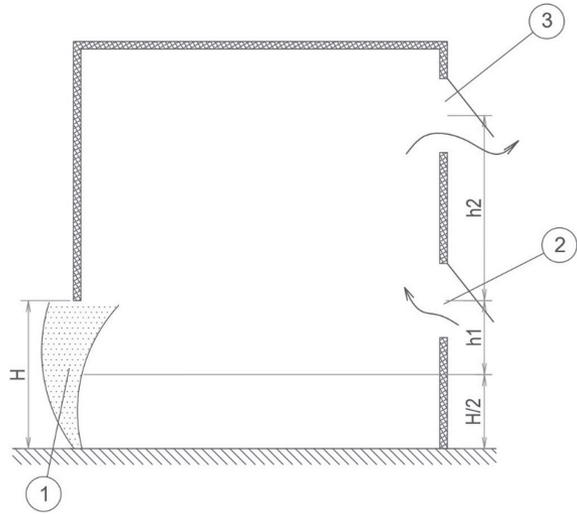
A.3   Example: Industrial Building with Large Gate
for Vehicles Entrance

Initial data

The industrial building has a large gate (3.6 × 3.6 m) used as a vehicle entrance,
provided with supply and exhaust apertures used for natural ventilation, which
are opened only during the warm season of the year.

The temperature and density of the indoor air are $t_{in} = 18\ °C$, $\rho_{in} = 1.21\ kg/m^3$.
The temperature and density of the outdoor air are $t_{out} = -20\ °C$, $\rho_{out} = 1.4\ kg/m^3$.

The length of apertures used for natural air supply is twice that of apertures used
for natural air exhaust: $l_{sup} = 2l_{ex}$, m.

The distance from the center of the gate aperture to the center of the natural air
supply aperture is $h_1 = 0.7$ m.

The distance between the centers of the supply and exhaust apertures is $h_2 = 4$ m.
Estimated wind speed $W = 3$ m/s.

Calculation procedure

Using the data from row 3 of the Table A.2 calculate the height, h_r as:

$$h_r = 0.7 + \left[4/0.25(2)^2 + 1\right] = 2.7\,m$$

Together with $K_1 = 0.5$ from Table A.2, this results in the following value of the pressure difference ΔP:

$$\Delta P = 9.8 \times 2.7 \times (1.4 - 1.21) + 0.5 \times \frac{0.8 \times 1.4 \times 9}{2} = 7.55 \, \text{Pa}$$

A.4 Example: Warehouse Building Gate

Initial data

The warehouse building has a gate with the size 4×4 m. Cracks in the building envelope are minor, and their combined area is significantly smaller than the gate area.
Indoor air temperature $t_{in} = 20\,°C+$, and its density is $\rho_{in} = 1.2 \, \text{kg/m}^3$.
Outdoor air temperature $t_{out} = -15\,°C$, and its density $\rho_{out} = 1.37 \, \text{kg/m}^3$.
Estimated wind speed $W = 3.5$ m/s.

Calculation procedure

Using the data from row 2 of the Table A.2 ($K_1 = 0.2$), use Eq. A.12 to calculate the value of the pressure difference ΔP:

$$\Delta P = 0.5 \times 4 \times 9.8 \times (1.37 - 1.2) + 0.2 \times \frac{0.8 \times 1.37 \times 12.25}{2} = 4.67 \, \text{Pa}$$

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Appendix B
Theoretical Background and Methodology for Calculating Air Curtains for Industrial Process Enclosure (ACPE)

The pressure difference, ΔP across the aperture of the industrial process enclosure in the case when the process emits both gases and heat, is defined by gravity forces created by the temperature difference inside and outside the enclosure and the mechanical exhaust from the enclosure (Fig. B.1).

$$\Delta P = (h_0 - H)g\Delta\rho + \Delta P_1 \qquad (B.1)$$

where

h_0 the height of the neutral zone, m.
H enclosure height, m.
g 9.8 m/s^2.
$\Delta\rho$ ρ_{room}.
ρ_{in} difference in air densities in the room and under the enclosure, kg/m^3.
ΔP_1 the pressure difference due to air exhaust from the enclosure, Pa.

$$\Delta P_1 = 0.25\rho_{room}\left(L_{exh}/\left[n\Sigma F_{app}\right]\right) \qquad (B.2)$$

where

L_{exh} air flow exhausted from the enclosure, m^3/s.
n the number of dedicated apertures in the enclosure.
ΣF_{app} the total area of dedicated apertures in the enclosure, m^2.

Typically, for enclosures with both gases and heat release, the value of ΔP_1 is much smaller than the value of the pressure difference resulting from gravity forces and therefore can be neglected.

The principle of momentum conservation applied to the contour $AA'\,BB'\,CC'\,DD'$ in projection on the X axis (Fig. B.2) can be expressed by Eq. B.3:

$$I_{DD'} + I_0 \sin \alpha - I_{AA'} = P_{AA'} F_1 + R_1 - P_{DD'} F_2 - R_2 + P_{\tau 1} - P_{\tau 2} \qquad \text{(B.3)}$$

where

$I_{DD'}$ and I_0 $I_{AA'}$	are airflow impulses on the surface DD', surface AA', and the initial impulse of the jet, respectively, N.
$P_{AA'}$, $P_{DD'}$	are pressure on the surfaces AA' and DD', Pa, respectively.
R_1, R_2	are Reaction Forces from the Internal Enclosure Wall Sections, N.
$P_{\tau 1}$, $P_{\tau 2}$	are the friction forces along the enclosure's lower and upper walls, N.
F_1, F_2	are the cross-sectional area of enclosure apertures 1 and 2, m².
A	is the angle of the air curtain jet supply.

The minimum amount of air to be removed from the enclosure by the mechanical exhaust systems depends upon the maximum concentration of the process emission allowed inside the enclosure (e.g., for vapors released during the drying process in the painting booth, the concentration of the solvent vapor shall not exceed 50% of the minimum value of the ignition threshold), calculated by

$$L_{\text{ex}} = 2GK/C_{\text{ign}} \qquad \text{(B.4)}$$

where

G	amount of explosive vapor emission released under the enclosure, g/s.
C_{ign}	low ignition threshold of the vapor concentration, g/m³.
K	the safety coefficient that considers the uneven distribution of the vapor under the enclose ($k = 2 \div 5$) when toxic substances are released in the installation— by creating the required suction rate in the opening from the condition of maintaining the maximum permissible concentration at the workplace, calculated according to the turbulent diffusion equation.

The airflow rate through each aperture (considering equal distribution among all apertures):

$$L_{\text{ap}} = \frac{L_{\text{exh}}}{n} \qquad \text{(B.5)}$$

The average pressure difference across the aperture due to gravity forces ($\Delta P_1 = 0$):

$$\Delta P_{\text{ave}} = 0.5gH\Delta\rho \qquad \text{(B.6)}$$

The supply air velocity of the air curtain jet:

$$V_0 = [\Delta P_{\mathrm{ave}} F_{\mathrm{encl}}/2\beta_0 \rho_0 F_0 E]^{0.5} \tag{B.7}$$

where

F_{encl}, F_0 is the cross-sectional area of the enclosure and the air curtain air supply nozzle, respectively.

β_0 is the Boussinesq coefficient ($\beta_0 = 1.05 - 1.1$).

E   is-coefficient of the air curtain dynamic efficiency.

$$E = \Delta P_{\mathrm{encl}}/2I_0 \tag{B.8}$$

Airflow rate supplied by the air curtain:

$$L_0 = V_0 F_0 \tag{B.9}$$

The dynamic efficiency coefficient can be determined using the principle of momentum conservation:

$$E = 0.5(\sin\alpha + B) \tag{B.10}$$

where B is a parameter that depends on the enclosure geometric parameters and the exhausted airflow rate. By neglecting the friction forces, Eq. B.3 will result in the following relationship for parameter B:

$$B = \left(1 + x^2\right)\frac{\rho_{\mathrm{exh}} F_0}{\rho_{\mathrm{ap}} F_{\mathrm{ap}}}\left[1 + \frac{\varsigma_{DD^1}}{2}\left(\frac{F_{\mathrm{encl}}}{F_{\mathrm{ap}}} - 1\right)\right] - x^2\frac{\rho_{\mathrm{room}} F_0}{\rho_0 F_{\mathrm{ap}}}\left[1 - \frac{\varsigma_{AA^1}}{2}\left(\frac{F_{\mathrm{encl}}}{F_{\mathrm{ap}}} - 1\right)\right] \tag{B.11}$$

where

$$\chi = \frac{L_{\mathrm{exh}}}{L_0}$$

$\varsigma_{AA'}$, $\varsigma_{DD'}$ are the coefficients of resistance for apertures AA' and DD'. Assuming, that for a typical enclosure of the painting conveyor:

$$\frac{F_{\mathrm{encl}}}{F_{\mathrm{ap1}}} = \frac{F_{\mathrm{encl}}}{F_{\mathrm{ap2}}} = 2.5; \quad \frac{p_{\mathrm{exh}}}{\rho_0} = 1; \quad \frac{p_{\mathrm{room}}}{\rho_0} = 1.5$$

Equation B.11 can be simplified as:
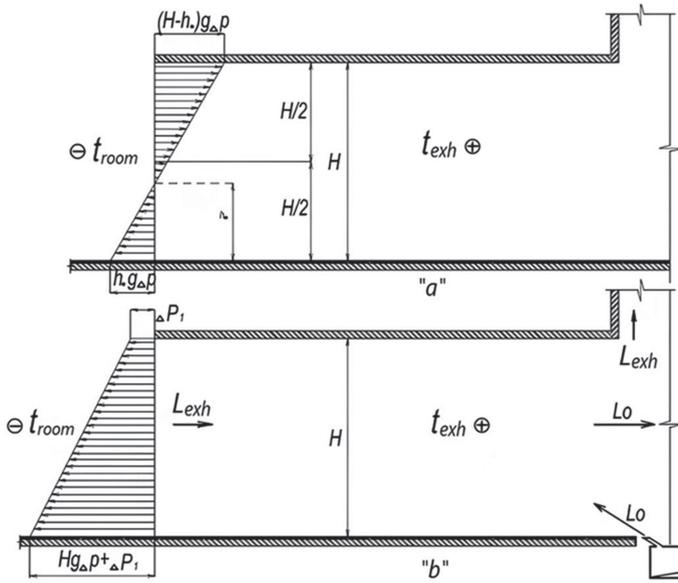
Fig. B.1 Schematic for evaluation of pressure difference across the aperture in the enclosure over the drying conveyor: **a** without ACPE air curtain and operating air exhaust from the enclosure; **b** with ACPE air curtain and operating air exhaust from the enclosure; h0 is the height of the neutral zone; $\Delta \rho = \rho_{room} - \rho_{exh}$; $\Delta P1$ pressure difference across aperture due to local exhaust operation; L_{exh}, $L0$—air flow rates exhausted by mechanical ventilation and supplied by the air curtain, respectively

$$B = 2.5 \overline{F}_0 \left[2.13(1 + \chi)^2 + \left(\frac{x}{1 + 1/x} \right)^2 - \chi^2 \right] \tag{B.12}$$

where $\overline{F}_0$ is the relative area of the air outlet slots of the air curtain:

$$\overline{F} = \frac{F_o}{F_{encl}}$$

For $\alpha > 80°$, $\sin\alpha$ value is close to 1.

The value of the dynamic efficiency coefficient E for such air curtains is typically 2–2.5 times greater than for traditional shutter-type air curtains and is in the range between 0.6 and 0.7. (Fig. B.3).

$$x = \frac{L_{exh}}{L_0}; \quad \overline{F}_0 = \frac{F_0}{F_{enc}} \tag{B.13}$$

Fig. B.2 Schematic for calculation of ACPE air curtain
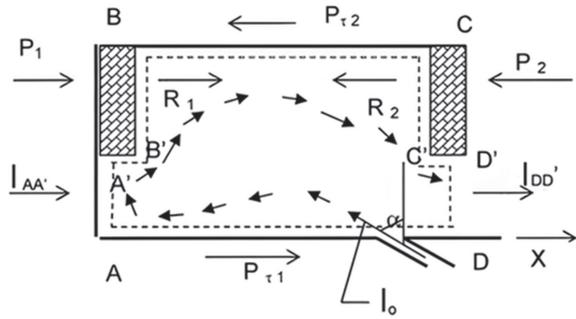
Fig. B.3 Dynamic efficiency coefficient E for ACPE air curtain (Eqs. B.9, B.10 and B.11)
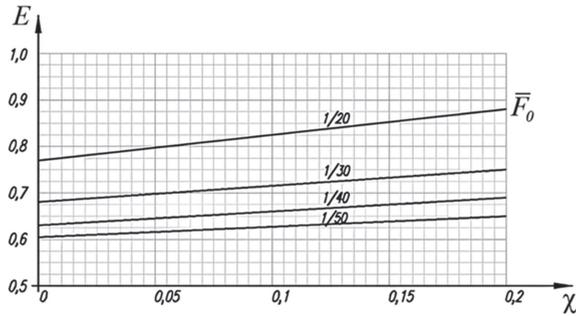
B.1 Examples of Calculation of Air Curtains for Enclosures of Industrial Process with Heat and Contaminants Emission

B.1.1 Example B1

Initial data

Enclosure of over the a single pass drying chamber has length $(L) = 16.0$ m, width $(B) = 1.7$ m, and height $(H) = 3.24$ m.

The cross-sectional area of the chamber: $F_{encl} = 5.5$ m^2.

The enclosure of the drying conveyor has two apertures.

Temperature and density of air under the enclosure: $t_{in} = 130\,°C$, $\rho_{in} = 0.87$ kg/m^3.

Temperature and density of air in the paint shop: $t_{shop} = 16\,°C$, $\rho_{shop} = 1.22$ kg/m^3.

Solvent vapors are released into the enclosure at the rate of G = 2.5 g/s.

The low ignition threshold for the vapor: $C_{ign} = 58$ g/m^3.

Calculation procedure

1. The total amount of exhaust air by mechanical ventilation from the enclosure (Eq. B.4):

$$L_{\text{exh}} = \frac{2GK}{C_{\text{ign}}} = \frac{2 \times 2.5 \times 5}{58} = 0.43\,\text{m}^3/\text{s}$$

2. The amount of air removed through each opening (Eq. B.5) is calculated as

$$L_{\text{app}} = \frac{0.43}{2} = 0.215\,\text{m}^3/\text{s}$$

3. The average pressure across the aperture (Eq. B.6) is calculated as

$$\Delta P_{\text{ave}} = 0.5gH\Delta\rho = 0.5 \times 9.8 \times 3.24(1.22 - 0.87) = 5.54\,\text{Pa}$$

4. The ratio of the enclosure dimensions: $L/H = 4.93$; $L/B = 9.4$.

   Assume, apertures are protected by a vertical two-sided air curtain ($L \geq 5B$).

5. Assume the relative area of air supply slots $\overline{F_0} = 1/20$; air is supplied at an angle $\alpha = 80°$ ($\sin\alpha \approx 1$); and the value of $\chi 1 = 0.1$.

6. The value of parameter B according to the Eq. B.11 is calculated as

$$B = \frac{2.5}{20}\left[2.13(1+0.1)^2 + \left(\frac{0.1}{1+\frac{1}{0.1}}\right)^2 - (0.1)^2\right] = 0.32$$

7. Dynamic efficiency coefficient E according to the Eq. B.10 is calculated as

$$E = 0.5(1 + 0.32) = 0.66$$

8. The supply air velocity of the air curtain jet per Eq. B.7 is calculated as

$$V_0 = \left[\frac{5.54 \times 20}{2 \times 1.1 \times 0.66 \times 0.87}\right]^{0.5} = 9.4\,\text{m/s}$$

9. The flow rate of air supplied by the air curtain at each aperture is calculated using Eq. B.9:

$$L_0 = 9.4\left(\frac{5.5}{20}\right) = 2.58\,\text{m}^3/\text{s}$$

10. Calculate the value $\chi 2$ and compare it with the value $\chi 1$ assumed in step 5:

$$\chi 2 = \frac{L_{y0}}{L_0} = \frac{0.215}{2.58} = 0.08$$

which results in a new value for $B = 0.31$ and $E = 0.655$.

The discrepancy between dynamic efficiency values E calculated in step 7 using the assumed χ value the one calculated in step 11 is 0.7%, which is satisfactory for such calculation and, therefore, the calculation does not need to be repeated.

Example B.2

Initial data

ACPE is designed for a three-way convective continuous drying chamber with the length of the working space of the camera, length $(L) = 27.6$ m, width $(B) = 2$ m and height $(H) = 3.9$ m.

Cross-sectional area: $F_{ap} = 7.8$ m^2.

Temperature and density of air under the enclosure: $t_{in} = 80\,°C$, $\rho_{in} = 1.0$ kg/m^3

Temperature and density of air in the paint shop: $t_{room} = 16\,°C$, $\rho_{room} = 1.22$ kg/m^3.

Solvent vapors (ethyl alcohol in the amount of $G = 5$ gr/s) are released in the chamber. The concentration of the lower limit of ignition: $C_{ign} = 68$ gr/m^3.

Calculation procedure

1. The total amount of exhaust air by mechanical ventilation from the enclosure (Eq. B.4) is calculated as

$$L_{exh} = 0.294\,\text{m}^3/\text{s}$$

2. The amount of air removed through each opening (Eq. B.5) is calculated as

$$L_{app} = 0.147\,\text{m}^3/\text{s}$$

3. The average pressure across the aperture (Eq. B.6) is calculated as

$$\Delta P_{ave} = 4.2\,\text{Pa}$$

4. The ratio of the enclosure dimensions:

$$L/H = 7.1;\ \ L/B = 9.4$$

5. Assume the relative area of air supply slots $\overline{F_0} = 1/30$, air is supplied at an angle $\alpha = 80°$ ($\sin \alpha \approx 1$) and the value of $\chi 1 = 0.1$.

6. According to Eq. B.11, the value of parameter B is calculated as

   $B = 0.22$.

7. According to the Eq. B.10, dynamic efficiency coefficient E is calculated as

$$E = 0.5(1 + 0.22) = 0.61$$

8. Per Eq. B.7, the supply air velocity of the air curtain jet is calculated as

$$V_0 = \left[\frac{5.54 \times 20}{2 \times 1.1 \times 0.66 \times 0.87} \right]^{0.5} = 9.4\,\text{m/s}$$

9. The flow rate of air supplied by the air curtain at each aperture is calculated using Eq. B.9:

$$L_0 = 2.52\,\text{m}^3/\text{s}$$