CHAPTER 29

INDUSTRIAL LOCAL EXHAUST SYSTEMS

INDUSTRIAL exhaust ventilation systems collect and remove airborne contaminants consisting of particulates (dust, fumes, smokes, fibers), vapors, and gases that can create an unsafe, unhealthy, or undesirable atmosphere. Exhaust systems can also salvage usable material, improve plant housekeeping, or capture and remove excessive heat or moisture.

Local Exhaust Versus General Ventilation

Local exhaust ventilation systems are normally the most cost-effective method of controlling air pollutants and excessive heat. For many manual operations, capturing pollutants at or near their source is the only way to ensure compliance with threshold limit values (TLVs) in the worker's breathing zone. Especially where recirculation is not used, local exhaust ventilation optimizes ventilation airflow, thus optimizing system costs.

In some industrial ventilation designs, the main emphasis is on filtering the air captured by local exhausts prior to evacuating it to the outdoors or returning it to the production space (Chambers 1993). As a result, these systems are evaluated by the efficiency of their filters. However, if only a small percentage of the emission is captured, the degree of separation efficiency becomes almost irrelevant.

The pollutant capturing efficiency of local ventilation systems depends on the hood design, the hood's positioning near the source of contamination, and the exhaust airflow. The selection and layout of the hood has a significant influence on the initial and operating costs of both local and general ventilation systems. In addition, poorly designed and maintained local ventilation systems can cause deterioration of building structures and equipment, negative health effects, and lower working capacity.

No local exhaust ventilation system is 100% effective in capturing pollutants and/or surplus heat. In addition, the installation of local exhaust ventilation system may not be possible under some circumstances, due to the size or mobility of the process. In such situations, **general ventilation** is needed to dilute the pollutants and/or surplus heat. Air supplied by the general ventilation system is usually conditioned. Supply air replaces air extracted by the local and general exhaust systems and improves comfort conditions in the occupied zone.

Chapter 12, Air Contaminants, of the 1997 *ASHRAE Handbook—Fundamentals* covers definitions, particle sizes, allowable concentrations, and upper and lower explosive limits of various air contaminants. Chapter 28, Ventilation of the Industrial Environment, of this volume and Chapter 1 of *Industrial Ventilation: A Manual of Recommended Practice* (ACGIH 1998) detail steps to determine the air volumes necessary to dilute the contaminant concentration using general ventilation. Refer to Chapter 28 for further information on replacement and makeup air.

If insufficient replacement air is provided, the pressure of the building will be negative relative to local atmospheric pressure. Negative pressure allows air to infiltrate through open doors, window cracks, and combustion equipment vents. As little as 12 Pa of negative pressure can cause drafts and might cause backdrafts in combustion vents, thereby creating a potential health hazard. Negative plant pressure can also cause excessive energy use. If workers near the plant perimeter complain about cold drafts, unit heaters are often installed. Heat from these units is usually drawn into the plant interior because of the velocity of the infiltration air, leading to overheating. Too often, the solution is to exhaust more air from the interior, causing increased negative pressure and more infiltration. Negative plant pressure reduces the exhaust volumetric flow rate because of increased system resistance, which could decrease local exhaust efficiency. Wind effects on building balance are discussed in Chapter 15, Airflow Around Buildings, of the 1997 *ASHRAE Handbook—Fundamentals*.

Positive-pressure plants and balanced plants (those having equal exhaust and replacement air rates) use less energy. However, if there are clean and contaminated zones in the same building, surplus pressure in the contaminated zones could cause contaminants to move into the clean zones.

The preparation of this chapter is assigned to TC 5.8, Industrial Exhaust.

LOCAL EXHAUST FUNDAMENTALS

System Components

Local exhaust ventilation systems typically consist of the following basic elements:

- Hood or entry point of the system to capture pollutants and/or excessive heat
- Duct system to transport polluted air
- Air cleaning device to remove captured pollutants from the airstream for recycling or disposal
- Air-moving device (e.g., fan or high pressure air ejector), which provides the motive power to overcome system resistance
- Exhaust stack, which discharges system air to the atmosphere
- Return duct system to return cleaned air back to the plant

System Classification

By Contaminant Source Type. Knowledge of the process or operation is essential before a hood can be designed. The type and size of the hood depends on the type and geometry of the pollution source. There are three types of pollution sources, each of which creates different contaminant movement (Posokhin 1984). **Buoyant (heat) sources** cause contaminants to move in buoyant plumes over the heated surfaces. **Nonbuoyant (diffusion) sources** create contaminant diffusion in all directions due to the concentration gradient (e.g., in the case of emission from painted surface). The emission rate is significantly affected by the intensity of the ambient air turbulence and air velocity. **Dynamic sources** create contaminant movement with an air jet (e.g., linear jet over a tank with push-pull ventilation) or due to particle flow (e.g., from a grinding wheel). In some cases, the above factors influencing contaminant distribution in the room are combined.

The geometry of the contaminant source can be **compact** or **linear**. Hoods are round, rectangular, or slotted to accommodate the geometry of the source.

By Hood Type. Hoods are either enclosing or nonenclosing (Figure 1). **Enclosing hoods** provide better and more economical contaminant control because their exhaust rates and the effects of room air currents are minimal compared to those for nonenclosing hoods. Hood access openings for inspection and maintenance should be as small as possible and out of the natural path of the contaminant. Hood performance (i.e., how well it controls the contaminant) should be checked by an industrial hygienist.

A **nonenclosing hood** can be used if access requirements make it necessary to leave all or part of the process open. Careful attention must be paid to airflow patterns around the process and hood and to the characteristics of the process in order to make nonenclosing hoods functional. Nonenclosing hoods can be classified according to their location relative to the contaminant source (Posokhin 1984) as either **updraft coaxial**, **sidedraft (lateral)**, or **downdraft** (Figure 2).

By System Mobility. Local exhaust systems with nonenclosing hoods can be **stationary** (i.e., having a fixed hood position), **moveable**, **portable**, or **built-in** (into the process equipment). Moveable (turnable) hoods are used when the process equipment must be accessed for repair, loading, and unloading (e.g., in electric ovens for melting steel). Hoods attached to flexible extraction arms (Figure 3) are used when the source of contamination is not fixed, as in arc welding (Zhivov 1993; Zhivov and Ashe 1997). Flexible extraction arms usually have a hood connected to a duct 140 to 160 mm in diameter and have higher efficiencies for lower airflow rates compared to stationary hoods. When the source of contamination is confined to a small, poorly ventilated space such as a tank, an additional flexible hose extension with a hood and a magnetic foot can be hooked on the fume extraction arm.

The portable extractor shown in Figure 4 is commonly used for the temporary extraction of fumes and solvents in confined spaces or during maintenance. It has a built-in fan and filter and a linear or round nozzle attached to a flexible hose about 45 mm in diameter. Built-in local exhausts, such as gun-mounted exhaust hoods and fume extractors built into stationary or turnover welding tables, are

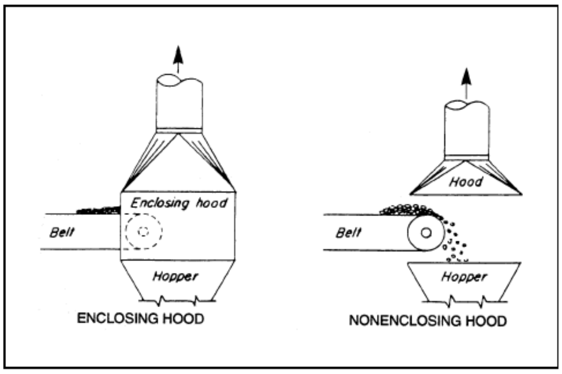


Fig. 1 Enclosing and Nonenclosing Hoods

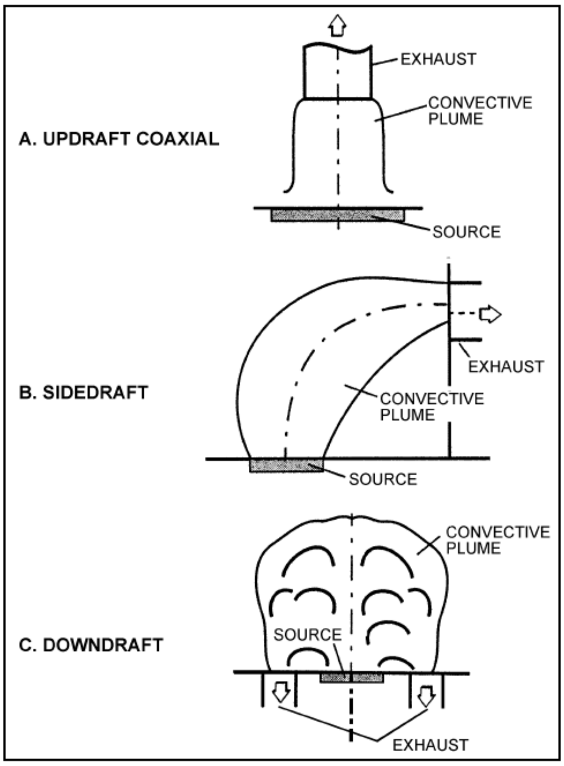


Fig. 2 Nonenclosing Hoods

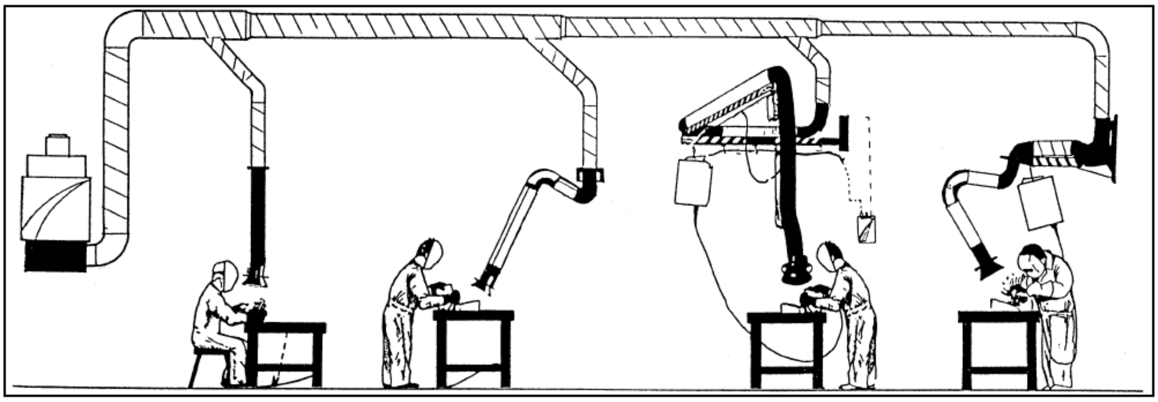


Fig. 3 Hoods Attached to Flexible Fume Extraction Arms

commonly used to evacuate welding fumes. Lateral exhaust hoods, which exhaust air through slots on the periphery of open vessels such as those used for galvanizing metals, are another example of built-in local exhausts.

Nonenclosing hoods should be located so that the contaminant is drawn away from the operator's breathing zone. **Canopy hoods** should not be used where the operator must bend over a tank or process (ACGIH 1998).

Effectiveness of Local Exhaust

The most effective hood uses the minimum exhaust airflow rate to provide maximum contaminant control. The **capturing effectiveness** should be high, but it would be difficult and costly to develop a hood that is 100% efficient. Makeup air supplied by general ventilation to replace exhausted air can dilute contaminants that are not captured by the hood (Posokhin 1984).

Capture Velocity. Capture velocity is the air velocity at the point of contaminant generation upstream of a hood. The contaminant enters the moving airstream at the point of generation and is conducted along with the air into the hood. Designers use capture velocity to select a volumetric flow rate Q to withdraw air through a hood. Table 1 shows ranges of capture velocities for several industrial operations. These figures are based on successful experience under ideal conditions. If velocities anywhere upstream of a hood are known [$V = f(Q_{x,y,z})$], the capture velocity is set equal to V_c at point (x, y, z) where contaminants are to be captured, and Q is found. The transport equations between the source and the hood must be solved to ensure that contaminants enter an inlet.

Hood Volumetric Flow Rate. After the hood configuration and capture velocity are determined, the exhaust volumetric flow rate can be calculated. For **enclosing hoods**, the **target exhaust volumetric flow rate** (the airflow rate that allows contaminant capture) is

$$Q_o^* = V_o A_o \tag{1}$$

where

Q_o^* = target exhaust volumetric flow rate, m^3/s
V_o = average air velocity in hood opening that ensures capture velocity at the point of contaminant release, m/s
A_o = hood opening area, m^2

The inflow velocity V_o is typically 0.5 m/s. However, research with laboratory hoods indicates that lower velocities can reduce the vortex downstream of the human body, thus lessening the reentrainment of contaminant into the operator's breathing zone (Caplan and Knutson 1977; Fuller and Etchells 1979). These lower face velocities require that the replacement air supply be distributed to minimize the effects of room air currents. This is one reason replacement air systems must be designed with exhaust systems in mind. Because air must enter the hood uniformly, interior baffles are sometimes necessary (Figure 5).

For **nonenclosing hoods**, the target airflow rate is proportional to some characteristic flow rate Q_o that depends on the type of contaminant source (Posokhin and Zhivov 1997):

$$Q_o^* = KQ_o \tag{2}$$

where

K = dimensionless coefficient depending on hood design
Q_o = characteristic airflow rate depending on contaminant source, m^3/s (for example, for a buoyant source, Q_o can be equal to airflow in the convective plume; for a dynamic source, Q_o can be equal to airflow rate in the jet)

For a nonenclosing hood with a nonbuoyant contaminant source, the characteristic airflow can be calculated using the following equation:

$$Q_o = V_o A_o \tag{3}$$

Table 1 Range of Capture (Control) Velocities

Condition of Contaminant Dispersion	Examples	Capture Velocity, m/s
Released with essentially no velocity into still air	Evaporation from tanks, degreasing, plating	0.25 to 0.5
Released at low velocity into moderately still air	Container filling, low-speed conveyor transfers, welding	0.5 to 1.0
Active generation into zone of rapid air motion	Barrel filling, chute loading of conveyors, crushing, cool shakeout	1.0 to 2.5
Released at high velocity into zone of very rapid air motion	Grinding, abrasive blasting, tumbling, hot shakeout	2.5 to 10

Note: In each category above, a range of capture velocities is shown. The proper choice of values depends on several factors (Alden and Kane 1982):

Lower End of Range	Upper End of Range
1. Room air currents favorable to capture	1. Distributing room air currents
2. Contaminants of low toxicity or of nuisance value only	2. Contaminants of high toxicity
3. Intermittent, low production	3. High production, heavy use
4. Large hood; large air mass in motion	4. Small hood; local control only

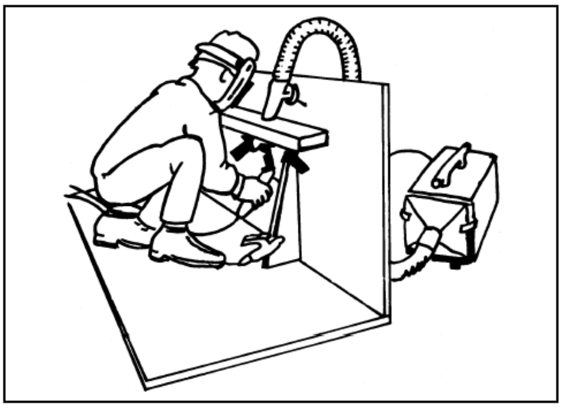


Fig. 4 Portable Fume Extractor with Built-in Fan and Filter

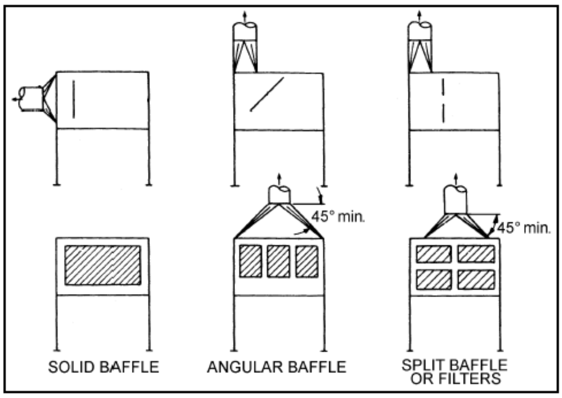


Fig. 5 Use of Interior Baffles to Ensure Good Air Distribution

An exhaust airflow rate lower than Q_o* results in reduced contaminant capturing effectiveness. An exhaust airflow rate greater than Q_o* results in excessive capturing effectiveness (Figure 6).

Airflow near the hood can be influenced by drafts from the supply air jets (spot cooling jets) or by turbulence of the ambient air caused by the jets, upward/downward convective flows, moving people, and drafts from doors and windows. Process equipment may be another source of air movement. For example, high-speed rotating machines such as pulverizers, high-speed belt material transfer systems, falling granular materials, and escaping compressed air from pneumatic tools all produce air currents.

These factors can significantly reduce the capturing effectiveness of local exhausts and should be accounted for in Equations (2) and (3) by the correction coefficient on room air movement. For example, Equation (2) will be replaced with the following:

$$Q_o^* = K_r K Q_o \tag{4}$$

where K_r = coefficient on room air movement; $K_r > 1$.

The exhausted air may contain combustible pollutant-air mixtures. In this case, the exhaust airflow rate should be increased to dilute combustible mixture to less than 25% of the lower explosive limit of the pollutant (NFPA *Standard* 86). Thus,

$$Q_o > \frac{G}{0.25 C_{exp(min)}} \tag{5}$$

where

G = amount of pollutant released by the source, mg/s
$C_{exp(min)}$ = lower explosive limit of pollutant, mg/m^3

Principles of Hood Design Optimization

Numerous studies of local exhaust and common practices have led to the development of the following list of hood design principles (Posokhin 1984):

- The hood should be located as close as possible to the source of contamination.
- The hood opening should be positioned so that it causes the contaminant to deviate the least from its natural path.
- The hood should be located so that the contaminant is drawn away from the operator's breathing zone.
- The hood must be the same size as or larger than the cross section of the flow entering the hood. If the hood is smaller than the flow, a higher volumetric flow rate will be required.
- The velocity distribution in the hood opening cross section should be nonuniform, following the velocity profile of the incoming flow. This can be achieved by incorporating vanes in the hood opening (Figure 7). In the case of a stationary hood and a contaminant source that is not fixed (e.g., welding or soldering), the air velocity along the hood must be uniform; this can be achieved using vanes or perforations.

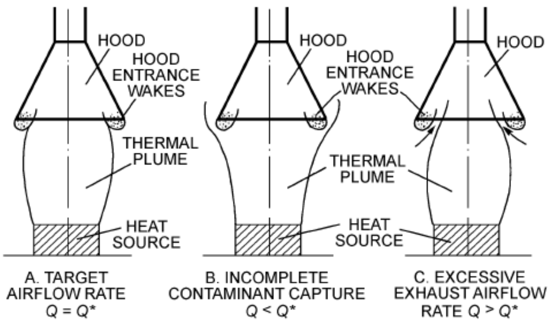


Fig. 6 Hood Performance at Different Exhaust Airflow Rates

AIR MOVEMENT IN VICINITY OF LOCAL EXHAUST

Theoretical Considerations

Airflow near the hood can be described using the incompressible, irrotational flow (i.e., potential flow) model. The total pressure p_{tot} in the area upstream of the hood remains constant and can be described with the following equation:

$$p_{tot} = p_{st} + p_d = \text{Constant} \tag{6}$$

where

p_{st} = static pressure at any point of the flow, Pa
p_d = $\rho V^2/2$ = dynamic pressure at any point of the flow, Pa
ρ = air density, kg/m^3
V = air velocity, m/s

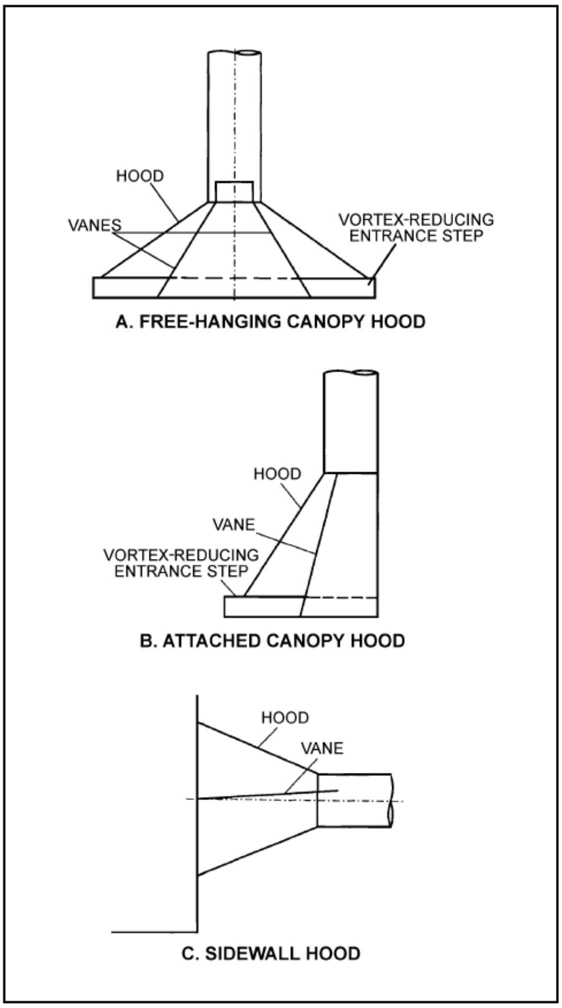


Fig. 7 Hoods with Nonuniform Velocities in Opening Cross Sections

At some distance from the hood, the total pressure p_{tot} in the airflow is equal to the ambient air pressure (i.e., $p_{tot} = 0$). Thus,

$$p_d = -p_{st} \tag{7}$$

The above discussion does not apply to wakes with vortex air movement (Figure 8).

Numerical simulation of hood performance is complex, and results depend on hood design, flow restriction by surrounding surfaces, source strength, and other boundary conditions. Thus, most currently used methods of hood design are based on analytical models and experimental studies.

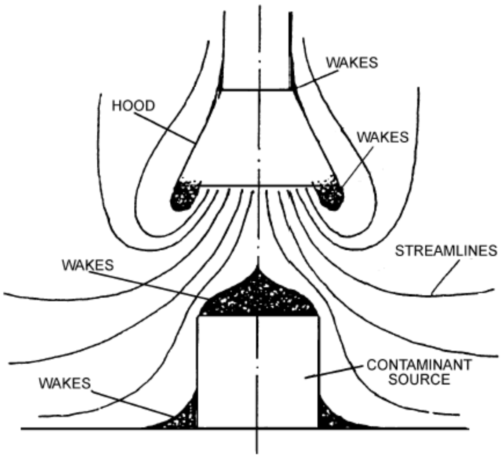


Fig. 8 Airflow in the Hood Vicinity

According to these models, the exhaust airflow rate is calculated based on a desired capture velocity at a particular location in front of the hood. It is easier to understand the design process for a sink with vanishingly small dimensions—a point or a linear source of suction. The point source can approximate airflow near a round or square/rectangular hood, and the linear source approximates the airflow near a slot hood.

A point source will draw air equally from all directions. Given the exhaust airflow, the velocity at any distance can be calculated by the following equation:

$$V_x = Q/4\pi x^2 \tag{8}$$

where

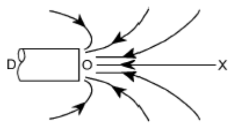
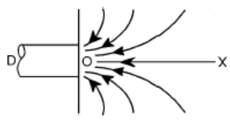
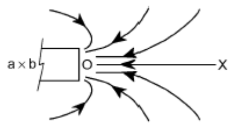
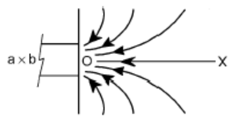
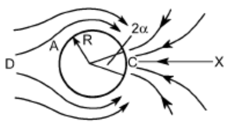
V_x = air velocity at distance x, m/s
x = distance from hood, m

A linear source will create a two-dimensional flow with the velocity V_x calculated as follows:

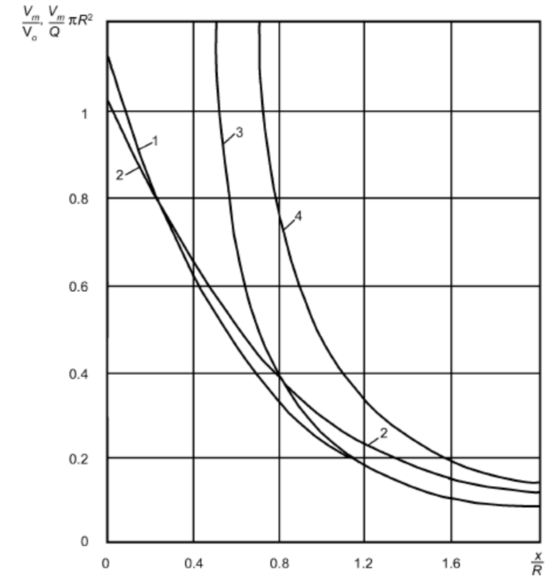
$$V_x = Q/2\pi x \tag{9}$$

Centerline velocities for different realistic hoods are presented in Table 2 (Posokhin and Zhivov 1997). Figure 9 compares relative velocity change for realistic hoods and a point source. At a distance greater than $x/R = 1$, velocities induced by realistic hoods are practically equal to those induced by a point source. This means that in

Table 2 Centerline Air Velocities Induced by Nonenclosing Hoods

Hoot Type	Schematic	Equation	Applicable Range	Reference
Round freestanding hood, unflanged		$\frac{V_x}{V_o} = (1 + 10x^2/A)^{-1}$	$x \le 1.7\sqrt{A}$; $\alpha \le 30°$	DallaValle (1952)
Round freestanding hood, flanged		$\frac{V_x}{V_o} = 1.1(0.07)^{x/D}$ $\frac{V_x}{V_o} = 0.1(x/D)^{-1.6}$	$0 \le \frac{x}{D} \le 0.5$; $C \ge D$ $0.5 \le \frac{x}{D} \le 1.5$; $C \ge D$	Garrison (1977)
Rectangular freestanding hood, unflanged		$\frac{V_x}{V_o} = (0.93 + 8.58\alpha_F^2)^{-1}$ $\alpha_F = (x/\sqrt{A})(a/b)^{\beta_F}$ $\beta_F = 0.2(x/\sqrt{A})^{-1.3}$	$1 \le \frac{a}{b} \le 16$; $0.05 \le \frac{x}{\sqrt{A}} \le 3$; $\alpha \le 30°$	Fletcher (1977)
Rectangular freestanding hood, flanged		$\left[\frac{V_x}{V_o} = 1 - \frac{2}{\pi}\operatorname{atan}\left(\frac{2x\sqrt{x^2 + a^2 + b^2}}{ab}\right)\right]$	$1 \le \frac{a}{b} \le 16$; $0 \le \frac{x}{\sqrt{A}} \le 1.6$; $\frac{C}{\sqrt{A}} \ge 1$	Tyaglo and Shepelev (1970)
Slot in the pipe wall		$\frac{V_{cx}}{V_o} = \frac{V_{AD}}{V_o} = \frac{2R}{\pi x}\operatorname{atan}\left[\frac{x+R}{x-R}\tan\left(\frac{\alpha}{2}\right)\right]$	$\|x\| \ge R$	Posokhin (1984)

C = flange width.



1 – Unrestricted circular opening with radius R
2 – Circular opening in the wall with radius R
3 – Unrestricted point source
4 – Point source in the wall
Q – Exhaust airflow rate
x – Distance from the source of suction to the point with air velocity V_m

Fig. 9 Relative Velocity Decay in Vicinity of a Point Source of Suction and of a Realistic Hood

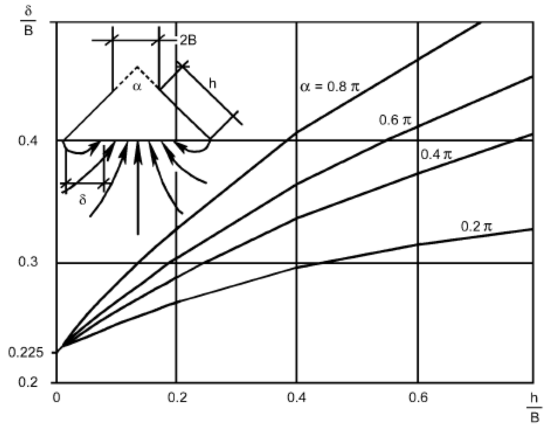


Fig. 10 Influence of Hood Configuration on Hood Entrance Wake Size δ

some cases, airflow in front of realistic hoods can be described using the simplified point source model.

Typically, velocity distribution in the hood face area is not uniform. Wakes formed close to the hood sides, or vena contracta, reduce the effective suction area of the hood. The size of these wakes and the level of velocity uniformity depend on hood design. Figure 10 shows the approximate relationship between wake size δ and the cone angle α of the hood (Posokhin 1984). Vanes, baffles, perforations, and other inserts can be used to control the size of the vena contracta and the velocity uniformity at the hood face area.

Air velocities in front of the hood suction opening depend on the exhaust airflow rate, the geometry of the hood, and the surfaces

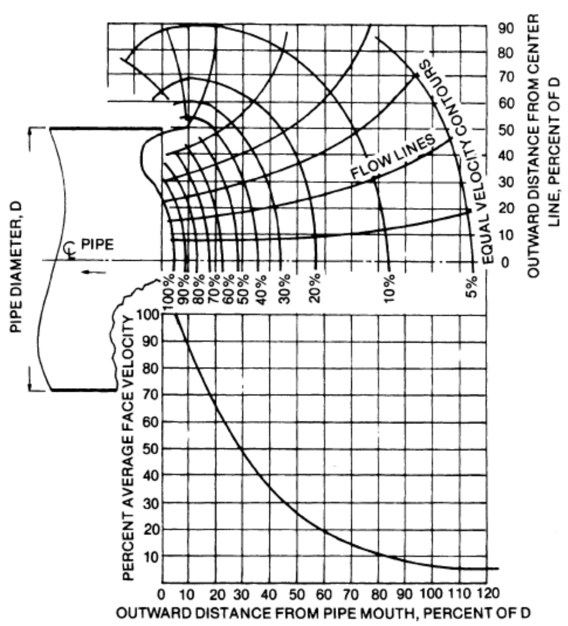


Fig. 11 Velocity Contours for Plain Round Opening

comprising the suction zone. Figure 11 shows lines of equal velocity (**velocity contours**) for a plain round opening. Studies have established the principle of similarity of velocity contours (expressed as a percentage of the hood face velocity) for zones with similar geometry (DallaValle 1952). Figure 12 (Alden and Kane 1972) shows velocity contours for a rectangular hood with an **aspect ratio** (width divided by length) of 0.333. The profiles are similar to those for the round hood but are more elongated. If the aspect ratio is lower than about 0.2 (0.15 for flanged openings), the shape of the flow pattern in front of the hood changes from approximately spherical to approximately cylindrical.

In the suction zone, the velocity decreases rapidly with distance from the hood. Velocity contours plotted in Figures 11 and 12 show that the velocity reaches 10% of the hood face velocity within the distance equal to the square root of the hood suction opening.

Air and Contaminant Distribution with Nonbuoyant Sources

Theories of hood performance with nonbuoyant pollution sources are based on the turbulent diffusion equation. The following equation allows the engineer to determine contaminant concentration decay in the uniform airflow upstream from the contaminant source:

$$C_x = C_o e^{-\frac{V}{D}x} \tag{10}$$

where

x = distance from source,m
C_o = contaminant concentration at source, mg/m³
C_x = contaminant concentration at distance x from source, mg/m³
V = air velocity in flow, m/s
D = coefficient of turbulent diffusion, m²/s
e = 2.7182818 = base for natural logarithm

The value of D depends on the air change rate in the ventilated space and the method of air supply. Studies by Posokhin (1984) show that D for locations outside supply air jets is approximately

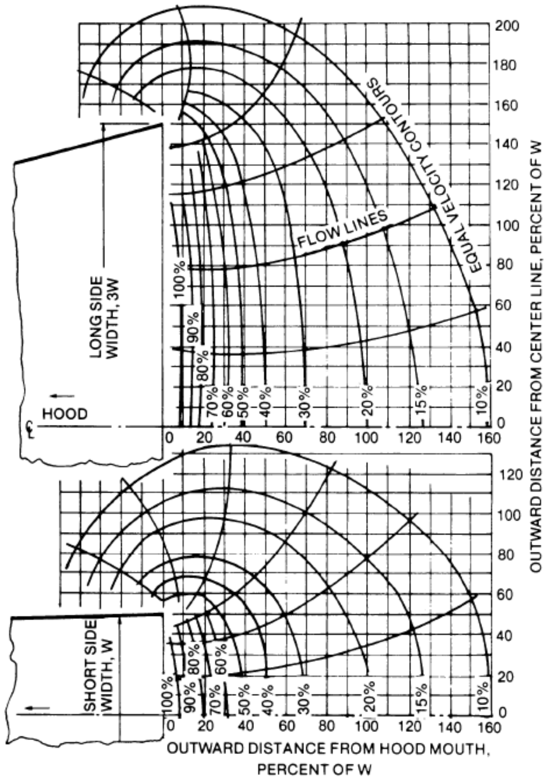


Fig. 12 Velocity Contours for Plain Rectangular Opening with Sides in a 1:3 Ratio

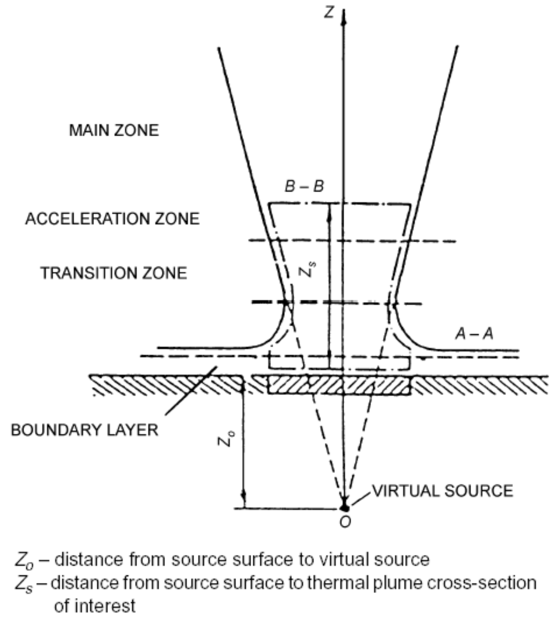


Fig. 13 Thermal Plume above Heat Source

0.025 m^2/s. Air disturbance caused by the operator or robot results in an increase of D by at least a factor of 2. Studies by Zhivov et al. (1997) showed that the value of D is affected by the velocity of cross drafts, their direction against the hood face, and the presence of an operator. For example, with a cross draft directed along the hood face with velocity $V = 0.5$ m/s, $D = 0.15$ m^2/s (with the presence of operator); an increase in cross draft velocity to $V = 1.0$ m/s results in $D = 0.3$ m^2/s.

Air and Contaminant Distribution with Buoyant Sources

The exhaust from hot processes requires special consideration because of the buoyant effect of heated air near the hot process. Determining the hood size and exhaust rate for a hot process requires an understanding of the convectional heat transfer rate (see the Chapter 3 of the 1997 *ASHRAE Handbook—Fundamentals*) and the physical size of the process. Convected heat and pollutants from the hot process are presumed to be contained in the thermal plume above the source, so the capture of the air transported with this plume will ensure the efficient capture of the contaminant (Burgess et al. 1989).

Analytical equations to calculate velocities, temperatures, airflow rates, and other parameters in thermal plumes over spot and linear heat sources with given heat loads were derived by Zeldovitch (1937), Schmidt (1941), Morton et al. (1956), and Shepelev (1961) based on the momentum and energy conservation equations and assuming Gaussian velocity and temperature difference (between plume and room air temperatures) distribution in thermal plume cross sections. These equations correspond to those received experimentally by other researchers (Popiolec 1981; Skäret 1986); for example, the equation for the airflow rate in the thermal plume is as follows:

$$Q = Cq_{conv}^{1/3}z^{5/3} \tag{11}$$

where

Q = airflow rate, m^3/s
q_{conv} = convective component of the heat source, W
z = height above the source level, m
C = experimental coefficient

Equation (11) was derived with the assumption that the heat source is very small; it does not account for the actual source dimensions.

Adjusting the point source model to realistic sources using the **virtual source method** (Figure 13) gives a reasonable estimate of the airflow rate in thermal plumes (Ivanitskaya et al. 1974; Elterman 1980; Holman 1989; Mundt 1992). The weak part of this method according to Skistad (1994) is estimating the location of the virtual point. The method of a "maximum case" and a "minimum case" (Skistad 1994) provides a tool for such estimation (Figure 14). According to the maximum case, the real source is replaced by the point source such that the border of the plume above the point source passes through the top edge of the real source (e.g., cylinder). The minimum case is when the diameter of the vena contracta of the plume is about 80% of the upper surface diameter and is located approximately 1/3 diameter above the source. For low-temperature sources, Skistad (1994) recommends the maximum case, whereas the minimum case best fits the measurements for larger, high-temperature sources.

Kofoed (1991) and Kofoed and Nielsen (1991) studied the interaction of the thermal plume with a wall and with another plume. In the case of the wall plume, the airflow rate should be decreased by a factor of 0.63; for interaction with another equal plume, it should be increased by a factor of 1.26.

Another approach to evaluating the thermal plume parameters (Nielsen 1993; Schaelin and Kofoed 1992; Davidson 1989; Aksenov and Gudzovskii 1994) is based on **computational fluid**

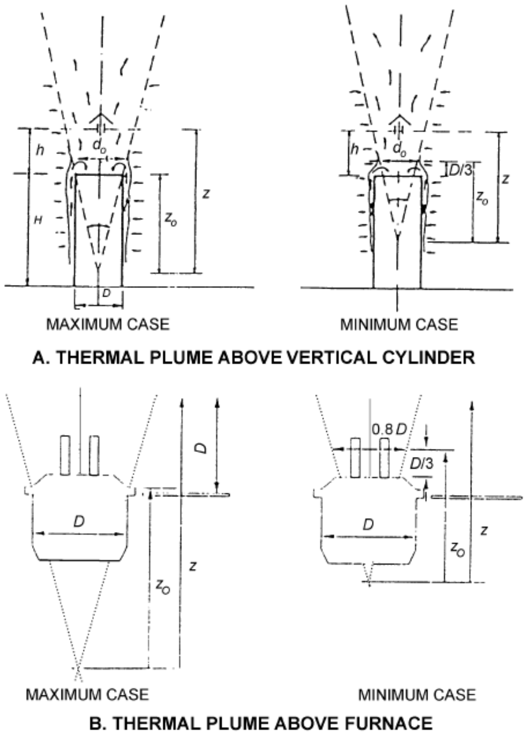


Fig. 14 "Minimum Case" and "Maximum Case" Approaches to Locating Virtual Source
[Reproduced from Skistad (1994)]

dynamics (CFD). With this approach, the airflow in the thermal plume is described by the Navier-Stokes equations and equations for energy and mass balances. Data from numerical and physical experiments on thermal plumes can be used to size overhead hoods above unusual sources or those with complex shapes.

Air Movement Created by Dynamic Sources

Push-pull systems (see the section on Jet-Assisted Hoods) supply air jets in the contaminated zone; they inject contaminated air and direct it toward the hood. Air jets in push-pull systems can be compact or linear. Table 3 presents velocity decay and airflow rates in free and attached compact and linear jets along the zone of practical interest.

To reduce the effect of room air movement on hood performance, the push air jet centerline velocity at the critical cross section (where the push air jet becomes weak, and the influence of the hood is not strong enough) should be from 1 to 2 m/s. According to Stroiizdat (1992), the centerline velocity V_x^{min} in the push jet attached to the heat source should be

$$V_x^{min} > 5.72\sqrt{\frac{l(T_s - T_r)}{T_r}} \tag{12}$$

where

l = distance between the supply nozzle and the exhaust hood, m
T_s = temperature of the heat source surface, K
T_r = room air temperature, K

Table 3 Centerline Velocity and Airflow Rate along Main (3rd) Zone of Supply Air Jet

	Air Supply	
Design Parameter	**Free Jet**	**Wall Jet**
Compact jet (supplied from round or rectangular nozzle)		
Centerline velocity, m/s	$V_x = K_1 V_o \frac{\sqrt{A_o}}{x}$	$V_x = \sqrt{2} K_1 V_o \frac{\sqrt{A_o}}{x}$
Airflow rate, m³/s	$Q_x = Q_o \frac{2}{K_1} \frac{x}{\sqrt{A_o}}$	$Q_x = Q_o \frac{\sqrt{2}}{K_1} \frac{x}{\sqrt{A_o}}$
Linear jet (supplied from slot)		
Centerline velocity, m/s	$V_x = K_1 V_o \sqrt{\frac{b_o}{x}}$	$V_x = K_1 V_o \sqrt{\frac{2b_o}{x}}$
Airflow rate, m³/s	$Q_x = Q_o \frac{1}{K_1} \sqrt{\frac{2x}{b_o}}$	$Q_x = Q_o \frac{1}{K_1} \sqrt{\frac{x}{b_o}}$

Note: K_1 = velocity decay coefficient; V_o = supply air velocity, m/s; Q_o = supply airflow rate, m³/s; A_o = effective area of air discharge, m²; x = distance along air jet from supply nozzle, m; b = slot width, m.

For the nonattached jet, the centerline velocity should be

$$V_x^{min} = 0.42(T_s - T_r)^{4/9} H^{1/3} \tag{13}$$

where H = maximum distance from the source surface to the nozzle-hood axis, m.

In the case of a push-pull hood over the tank, supply air velocity should not exceed 10 m/s to avoid waves on the liquid surface.

Grinding, polishing, and other finishing operations are another type of dynamic contaminant source. These processes produce particles and impart them with some momentum. The receiving hood used for grinding operations is positioned and sized to catch the particles, which are thrown toward the hood (Burgess et al. 1989). The airflow required for the receiving hood can be calculated based on the capture velocity. It might seem that the distance x used in Equations (8) and (9) can be reduced to $x^* = x - S$, where S is the particle stopping distance (i.e., the distance a particle ejected into still air at the initial velocity will travel while decelerating to rest due to drag forces). However, the data from Hinds (1982) show the difficulty in throwing even fairly large particles an appreciable distance in still air. Any particles in the inhalable range (i.e., $d < 10$ μm) should be considered immovable, so it should be always assumed that $x^* = x$ (Burgess et al. 1989). The hood used for grinding processes acts as a receiver for large particles ($d > 30$ μm), but not for small and intermediate particles ($3 < d < 30$ μm). Bastress et al. (1974) found that respirable particles were not captured efficiently by typical grinding wheel exhaust systems and escaped to the vicinity of the worker's breathing zone. Nonetheless, the standard hood designs recommended in ACGIH (1998), while not 100% efficient at capturing respirable particles, were sufficient to provide worker protection at or below the TLVs for total and respirable inert dust.

Pressure Losses in Local Exhausts

When air enters a hood, dynamic losses cause a loss of total pressure. This is called the **hood entry loss** and may have several components, each given by

$$\Delta p_e = C_o p_v \tag{14}$$

where

Δp_e = hood entry loss, Pa
C_o = loss factor depending on component geometry, dimensionless
p_v = appropriate velocity pressure, Pa

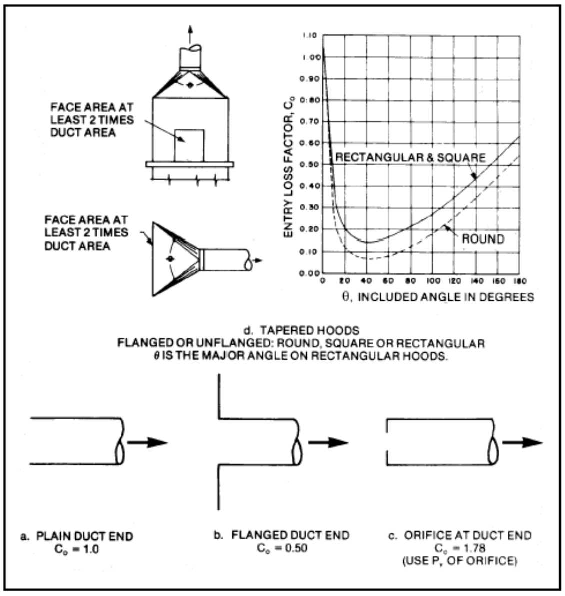


Fig. 15 Entry Losses for Typical Hoods

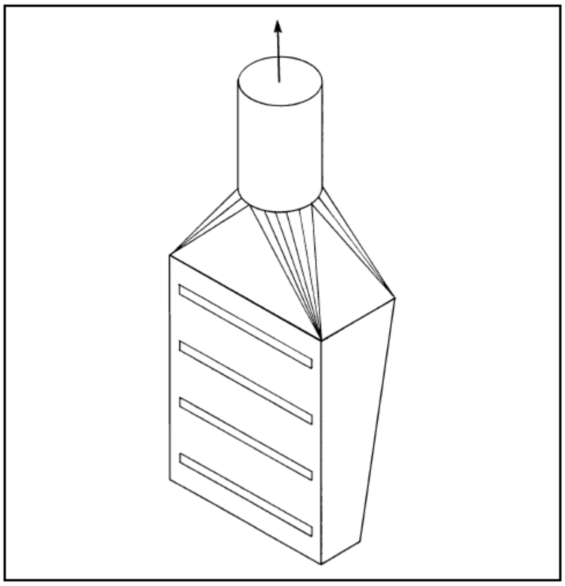


Fig. 16 Multislot Nonenclosing Hood

The following equation relates velocity to velocity pressure:

$$V = \sqrt{\frac{2p_v}{\rho}} \tag{15}$$

If the air temperature is 20°C ± 15°C, the ambient pressure is the standard 101 kPa, the duct pressure is no more than 5 kPa different from the ambient pressure, the dust loading is low (< 2 g/m^3), and moisture is not a consideration, then the density in Equation (15) is 1.2 kg/m^3, and Equation (13) simplifies to

$$V = 1.29\sqrt{p_v} \tag{16}$$

Loss factors C_o for various hood shapes are given in Figure 15. More information on loss factors can be found in Chapter 32, Duct Design, of the 1997 *ASHRAE Handbook—Fundamentals*, Idelchik et al. (1986), and ACGIH (1998). Figure 15 shows an optimum hood entry angle to minimize entry loss. However, this total included angle of 45° is impractical in many situations because of the required transition length. A 90° angle, with a corresponding loss factor of 0.25 (for rectangular openings), is standard for most tapered hoods.

The combination of several consecutive hood components may affect the values of their individual loss factors. Thus, a hood with multiple components should be treated as a single component with a pressure loss obtained in a laboratory or field test.

Total pressure is difficult to measure in a duct system because it varies from point to point across a duct, depending on the local velocity. On the other hand, static pressure remains constant across a straight duct. Therefore, measurement of static pressure in a straight duct at a single point downstream of the hood can monitor the volumetric flow rate. The absolute value of this static pressure, the **hood suction**, is given by

$$p_{st} = p_v + \Delta p_e \tag{17}$$

where p_{st} = hood suction, Pa.

Hood suction is the negative static pressure measured about three duct diameters downstream of the hood. A larger distance is required for included angles of 180° or larger.

Simple Hoods. A simple hood has only one dynamic loss. The hood suction becomes

$$p_{st} = (1 + C_o)p_v \tag{18}$$

where p_v = duct velocity pressure, Pa.

Example 1. A nonenclosing sidedraft flanged hood (Figure 20) with face dimensions of 0.45 m by 1.2 m rests on the bench. The required volumetric flow rate is 700 L/s. The duct diameter is 225 mm; this gives a duct velocity of 17.6 m/s. The hood is designed such that the largest angle of transition between the hood face and the duct is 90°. What is the suction for this hood? Assume standard air density.

Solution: The two transition angles cannot be equal. Whenever this is true, the larger angle is used to determine the loss factor from Figure 15. Because the transition piece originates from a rectangular opening, the curve marked “rectangular” must be used. This corresponds to a loss factor of 0.25. Equation (16), which assumes standard air density, can be used to determine the duct velocity pressure:

$$p_v = (17.6/1.29)^2 = 186 \text{ Pa}$$

From Equation (18),

$$p_{st} = (1 + 0.25)(186) = 232 \text{ Pa}$$

Compound Hoods. The losses for multislot hoods (see Figure 16) or single-slot hoods with a plenum (called compound hoods) must be analyzed somewhat differently. The slots distribute air over the hood face and do not influence capture efficiency. The slot velocity should be approximately 10 m/s to provide the required distribution at the minimum energy cost. Higher velocities dissipate more energy.

Losses occur when air passes through the slot and when air enters the duct. Because the velocities, and therefore the velocity pressures, can be different at the slot and at the duct entry locations, the hood suction must reflect both losses and is given by

$$p_{st} = p_v + (C_o p_v)_s + (C_o p_v)_d \quad (19)$$

where the first p_v is generally the higher of the two velocity pressures, s refers to the slot, and d refers to the duct entry location.

Example 2. A multislot hood has 3 slots, each 25 mm by 1 m. At the top of the plenum is a 90° transition into the 250 mm duct. The volumetric flow rate required for this hood is 0.78 m³/s. Determine the hood suction. Assume standard air.

Solution: The slot velocity V_s from Equation (1) is

$$V_s = 0.78/(3 \times 0.025 \times 1) = 10.4 \text{ m/s}$$

Substituting this velocity into Equation (16),

$$p_v = (10.4/1.29)^2 = 65 \text{ Pa}$$

The duct area is 0.0491 m². Therefore, the duct velocity determined from Equation (1) is

$$V_d = 0.78/0.0491 = 15.9 \text{ m/s}$$

Substituting this velocity into Equation (16),

$$p_v = (15.9/1.29)^2 = 152 \text{ Pa}$$

For a 90° transition into the duct, the loss factor is 0.25. For the slots, the loss factor is 1.78 (Figure 15). The duct velocity pressure is added to the sum of the two losses because it is larger than the slot velocity pressure. Using Equation (19),

$$p_{st} = 152 + (1.78 \times 65) + (0.25 \times 152) = 306 \text{ Pa}$$

Exhaust volume requirements, minimum duct velocities, and entry loss factors for many specific operations are given in Chapter 10 of ACGIH (1998).

Influence of Air Movement on Local Exhaust Performance

Both air movement caused directly by supply air jets and turbulence of the ambient air resulting from general ventilation system operation, convective plumes, and moving people and process equipment are at least as important as hood face velocity in controlling contaminant spillage. Caplan and Knutson (1978) recommend that air movement caused by the above factors should be less than 1/2 to 2/3 times the hood face velocity.

In studies of hoods with a vertical face area, Zhivov et al. (1997) showed that the preferred orientation of the hood relative to the most likely direction of the cross draft is 135°. This orientation achieves both the lowest contaminant concentration in the operator's breathing zone and the highest capturing effectiveness. A moderate draft from behind the operator significantly increases the contaminant concentration in the operator's breathing zone. A cross draft has minimal effect on operator exposure, but the contaminant removal by the hood is low.

To reduce the influence of cross drafts greater than 0.4 m/s on the performance of a canopy hood above a buoyant source, Stroiizdat (1992) recommends attaching one-, two-, or three-sided removable shields to the hood that drop to a height above the source of 0.8 times the equivalent diameter of the design source.

Schematics in Figure 17 show how air jets can improve hood performance.

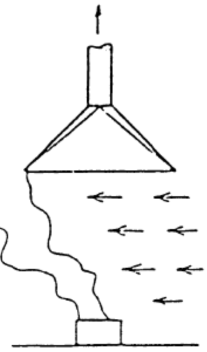
LOCAL EXHAUST FROM BUOYANT SOURCES

Overhead Hoods

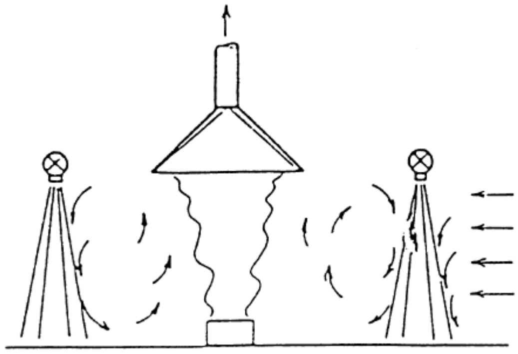
If the process cannot be completely enclosed, the canopy hood should be placed above the process so that the contaminant moves toward the hood. Canopy hoods should be applied and designed with caution to avoid drawing contaminants across the operator's breathing zone (see Figure 18). The hood's height above the process should be kept to a minimum to reduce the total exhaust airflow rate.

A **low canopy hood** is within 1 m of a process (within the thermal plume transition zone) and requires the lowest volumetric flow rate of all nonenclosing hoods. A **high canopy hood** is more than 3 m above a process and requires a higher volumetric flow rate because room air is entrained in the column of hot, contaminated air rising from the process; this situation should be avoided.

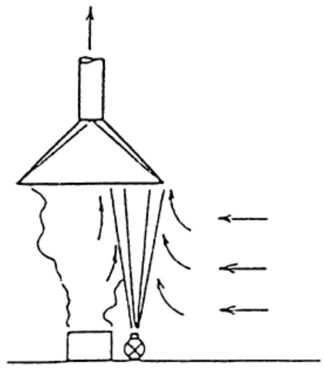
Hemeon (1963) lists Equations (20) through (24) for determining the volumetric flow rate of hot gases for low canopy hoods. Note that canopy hoods located 1 to 3 m above the process cannot be analyzed using these equations.



A. Cross draft effect on thermal plume under high hood



B. High hood protection with double-side air curtain



C. High hood protection with single-side air curtain

Fig. 17 Hood Performance Improvement with Air Jets

$$Q_o^* = \left(\frac{2gR}{pc_p} \times q_{conv} L A_p^2\right)^{1/3} \quad (20)$$

where

Q_o^* = volumetric flow rate, m^3/s
g = gravitational acceleration, 9.8 m/s^2
R = air gas constant, 287 J/(kg·K)
p = local atmospheric pressure, Pa
c_p = constant pressure specific heat for air, 1004 J/(kg·K)
q_{conv} = convection heat transfer rate, W
L = vertical height of hot object, m
A_p = cross-sectional area of airstream at upper limit of hot body, m^2

For a standard atmospheric pressure of 101.325 kPa, Equation (20) can be written as

$$Q_o^* = 0.038(q_{conv} L A_p^2)^{1/3} \quad (21)$$

For three-dimensional bodies, the area A_p in Equations (20) and (21) is approximated by the plan view area of the hot body (Figure 19A). For horizontal cylinders, A_p is the product of the length and the diameter of the rod.

For vertical surfaces, the area A_p in Equations (20) and (21) is the area of the airstream (viewed from above) as the flow leaves the vertical surface (Figure 19B). As the airstream moves upward on a vertical surface, it appears to expand at an angle of approximately 4 to 5°. Thus, A_p is given by

$$A_p = wL\tan\theta \quad (22)$$

where

w = width of vertical surface, m
L = height of vertical surface, m
θ = angle of air stream expansion, °

For horizontal heated surfaces, A_p is the surface area of the heated surface, and L is the longest length (conservative) of the horizontal surface or its diameter if it is round (Figure 19C).

If the heat transfer is caused by steam from a hot water tank,

$$q_{conv} = h_{fg} G A_p \quad (23)$$

where

q_{conv} = convective heat transfer, kW
h_{fg} = latent heat of vaporization, kJ/kg
G = steam generation rate, kg/(s·m^2)
A_p = surface area of the tank, m^2

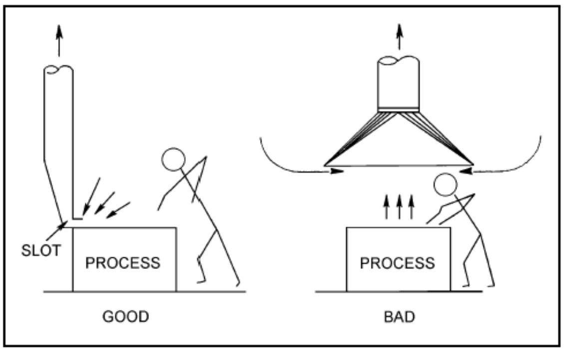


Fig. 18 Influence of Hood Location on Contamination of Air in the Operator's Breathing Zone

At 100°C, the latent heat of vaporization is 2257 kJ/kg. Using this value and Equation (23), Equation (20) simplifies to

$$Q_o^* = 5A_p(GL)^{1/3} \quad (24)$$

The exhaust volumetric flow rate determined by Equation (20) or (24) is the required exhaust flow rate when (1) a low canopy hood of the same dimensions as the hot object or surface is used and (2) side and back baffles are used to prevent room air currents from disturbing the rising air column. If side and back baffles cannot be used, the canopy hood size and the exhaust flow rate should be increased to reduce the possibility of contaminant escape around the hood. A good design provides a low canopy hood overhang equal to 40% of the distance from the hot process to the hood face on all sides (ACGIH 1998). The increased hood flow rate can be calculated using the following equation:

$$Q_t = Q_o^* + V_f(A_f - A_p) \quad (25)$$

where

Q_t = total flow rate entering hood, m^3/s
Q_o^* = flow rate determined by Equation (20) or (24), m^3/s
V_f = desired indraft velocity through the perimeter area, m/s
A_f = hood face area, m^2
A_p = plan view area of Equation (20) or (24)

A minimum indraft velocity of 0.5 m/s should be used for most design conditions. However, if room air currents are appreciable or if the contaminant discharge rate is high and the design exposure limit is low, higher values of V_f may be required.

The volumetric flow rate for a high canopy hood over a round, square, or rectangular (aspect ratio near 1) source can be predicted using Equation (11) with adjustments discussed in the section on Air and Contaminant Distribution with Buoyant Sources.

The diameter D_z of the plume at any elevation z above the virtual source can be determined by

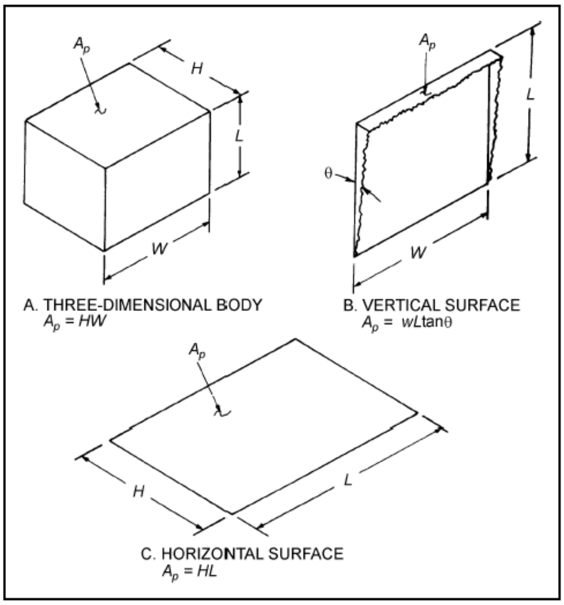


Fig. 19 A_p for Various Situations

$$D_z = 0.5z^{0.88} \tag{26}$$

High canopy hoods are extremely susceptible to room air currents. Therefore, they are typically much larger (often 100% larger) than indicated by Equation (26) and are used only if a low canopy hood cannot be used. The total flow rate exhausted from the hood can be evaluated using Equation (25) if Q_o is replaced by Q_z.

According to Posokhin (1984), the canopy hood is effective when

$$\frac{V_r(z+z_o)}{V_z b} \leq 0.35$$

where

V_r = room air velocity,m/s
z_o = distance from virtual source to upper source level, m
V_z = air velocity on thermal plume axis at hood face level, m/s
b = source width, m

Sidedraft Hoods

Sidedraft hoods are typically used when the contaminant is drawn away from the operator's breathing zone (Figure 2B). With a buoyant source, a sidedraft hood requires a higher exhaust volumetric flow rate than a low canopy hood. If a low canopy hood restricts the operation, a sidedraft hood may be more cost-effective than a high canopy hood. Examples of sidedraft hoods include multislotted "pickling" hoods near welding benches (Figure 16), flanged hoods (Figure 20), and slot hoods on tanks (Figure 21).

Sidedraft hoods should be installed with the low edge of the suction area at the level of the top of the heat source. The distance b between the hood and the source may vary depending on the width of the source (Figure 22); maximum b is equal to the width B of the source. Based on studies by Kuz'mina (1959), the following airflow rate through the sidedraft hood is recommended (Stroiizdat 1992):

$$Q_o^* = cq_{conv}^{1/3}(H+B)^{5/3} \tag{27}$$

where

c = nondimensional coefficient depending on hood design and location relative to contaminant source [see Equations (28) and (29)]
q_{conv} = convective component of the heat source, W
H = vertical distance from source top surface to hood center, m
B = source width, m

For a hood without a screen (Figure 22A),

$$c = 280\left(\frac{l}{H+B}\right)^{2/3} \tag{28}$$

For a hood with a screen (Figure 22B),

$$c = 280m\sqrt{\frac{l}{H+B}} \tag{29}$$

where $m = 1$, when $b/B = 0$; $m = 1.5$, when $b/B = 0.3$; $m = 1.8$ when $b/B = 1$, and $m = 2$ when $b/B > 1$.

For open vessels, the contaminant can be controlled by a lateral exhaust hood, which exhausts air through slots on the periphery of the vessel. The hood capturing effectiveness depends on the exhaust airflow rate and the hood design; however, it is not influenced by air velocity through the slot. Hoods are designed with air exhaust from one side of the vessel or from two sides. Air exhaust from two sides requires a lower exhaust airflow rate. In most applications, a hood with a vertical face (Figure 23A) is used when the distance h_1

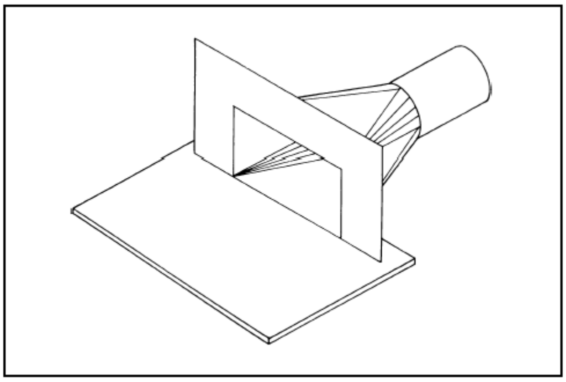


Fig. 20 Hood on Bench

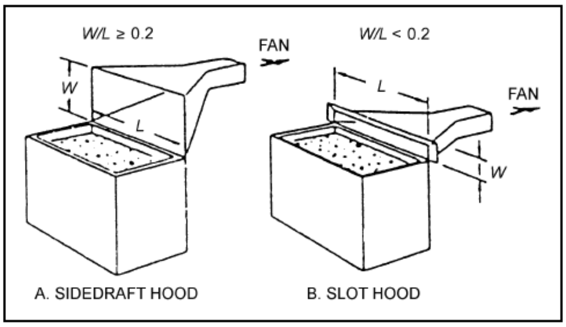


Fig. 21 Sidedraft Hood and Slot Hood on Tank

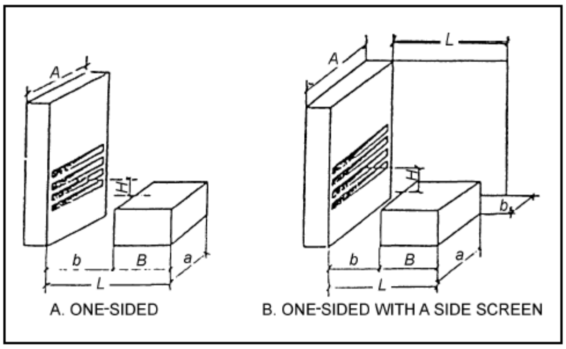


Fig. 22 Schematics of Sidedraft Hood on Work Bench

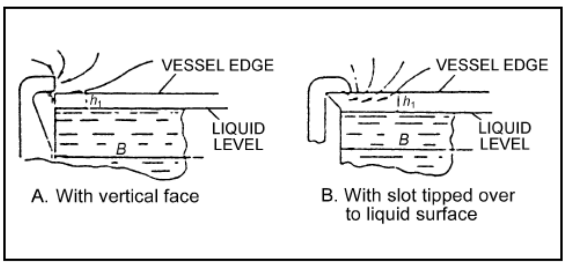


Fig. 23 Schematics of Sidedraft Slot Hood on Tank

Table 4 $K_{\Delta t}$ Coefficient Values

	Liquid-to-Air Temperature Difference, K								
	0	10	20	30	40	50	60	70	80
$K_{\Delta t}$	1	1.16	1.31	1.47	1.63	1.79	1.94	2.1	2.26

between the vessel edge and the liquid level is smaller than 100 mm (Stroiizdat 1992). When h_1 > 100 mm, hoods with the slot tipped over to the liquid surface (Figure 23B) are more effective.

Stroiizdat (1992) recommends the following exhaust airflow rate from one- and two-sided lateral slot hoods:

$$Q_o^* = 1400\left(0.53\frac{Bl}{B+l}+h\right)^{1/3} BlK_1K_{\Delta t}K_t \tag{30}$$

where

B = vessel width, m
l = vessel length, m
h = vertical distance between the liquid level and the hood face center, m
K_1 = hood design coefficient: K_1 = 1 for two-sided hood; K_1 = 1.8 for one-sided hood
$K_{\Delta t}$ = coefficient reflecting liquid temperature (see Table 4)
K_t = coefficient reflecting process toxicity (from 1 to 2; e.g., for electroplating tanks, K_t = 2)

A more cost-effective alternative to a one- or two-sided lateral hood is a **push-pull hood**, described in the section on Jet-Assisted Hoods.

Downdraft Hoods

Downdraft hoods should be considered only when overhead or sidedraft hoods are impractical. Air can be exhausted through a slotted baffle (e.g., downdraft cutting table—see Figure 24) or through a circular slot with a round source (Figure 25A) or two linear slots along the long sides of a rectangular source (Figure 25B). To achieve higher capturing effectiveness, the exhaust should be located as close to the source as possible. Capturing effectiveness decreases with an increase in source height and increases when the top of the source is located below the hood face surface. With a buoyant source, the air velocity induced by the exhaust should be equal to or greater than the air velocity in the plume above the source (Posokhin 1984).

The target airflow rate for a circular downdraft hood is

$$Q_o^* = 0.0314(q_{conv}d^5)^{1/3}\left(1-0.06\frac{q_{conv}^{vert}}{q_{conv}^{horiz}}\right)K_1K_v \tag{31}$$

For a double linear slot downdraft hood,

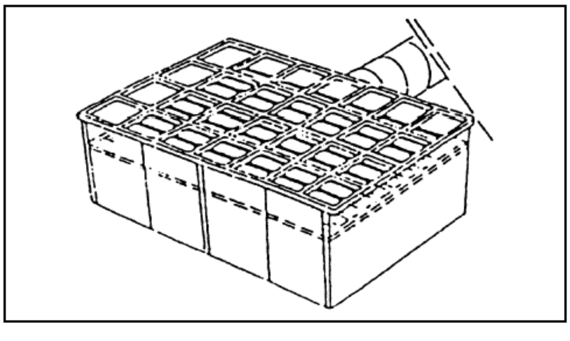


Fig. 24 Downdraft Welding Table

$$Q_o^* = 0.05q_{conv}^{1/3}lbK_1K_v \tag{32}$$

where

d = source diameter, m
l = source length, m
b = source width, m
q_{conv}^{vert} = convective heat component from the source vertical surfaces, W
q_{conv}^{horiz} = convective heat component from the source horizontal surface, W
K_1 = coefficient accounting for hood geometry that can be evaluated using graphs in Figure 25
K_v = coefficient accounting for room air movement V_r

$$= 1 + 44.7\sqrt{V_r^3\frac{d}{q_{conv}}} \quad \text{for circular downdraft hood} \tag{33}$$

$$= 1 + 44.7\sqrt{V_r^3\frac{b}{q_{conv}}} \quad \text{for double slot downdraft hood} \tag{34}$$

Example 3. A downdraft hood is to be designed to capture a contaminant from a rectangular source $l \times b \times h$ = 0.6 m × 0.5 m × 0 m. Convective heat component of the source q_{conv} = 1000 W. Room air movement V_r = 0.4 m/s. Two exhaust slots with a width b = 100 mm are located at the distance B_1 = 0.6 m and B_2 = 0.8 m. Determine the exhaust airflow rate.

Solution: Using the graph in Figure 25 for B_2/B_1 = 0.8/0.6 = 1.33, and B_1/b = 0.6/0.5 = 1.2, obtain K_1 = 5. Coefficient K_v accounting for room air movement [Equation (34)] is

$$K_v = 1 + 44.77\sqrt{0.4^3\frac{0.5}{1000}} = 1.25$$

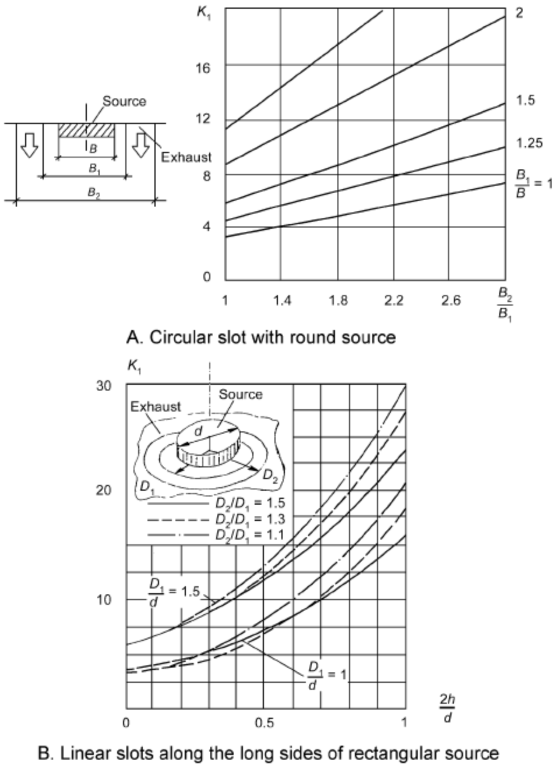


Fig. 25 K_1 Coefficient Evaluation for Downdraft Hoods

Table 4 $K_{\Delta t}$ Coefficient Values

	Liquid-to-Air Temperature Difference, K								
	0	10	20	30	40	50	60	70	80
$K_{\Delta t}$	1	1.16	1.31	1.47	1.63	1.79	1.94	2.1	2.26

between the vessel edge and the liquid level is smaller than 100 mm (Stroiizdat 1992). When h_1 > 100 mm, hoods with the slot tipped over to the liquid surface (Figure 23B) are more effective.

Stroiizdat (1992) recommends the following exhaust airflow rate from one- and two-sided lateral slot hoods:

$$Q_o^* = 1400\left(0.53\frac{Bl}{B+l} + h\right)^{1/3} BlK_1K_{\Delta t}K_t \qquad (30)$$

where

B = vessel width, m
l = vessel length, m
h = vertical distance between the liquid level and the hood face center, m
K_1 = hood design coefficient: K_1 = 1 for two-sided hood; K_1 = 1.8 for one-sided hood
$K_{\Delta t}$ = coefficient reflecting liquid temperature (see Table 4)
K_t = coefficient reflecting process toxicity (from 1 to 2; e.g., for electroplating tanks, K_t = 2)

A more cost-effective alternative to a one- or two-sided lateral hood is a **push-pull hood**, described in the section on Jet-Assisted Hoods.

Downdraft Hoods

Downdraft hoods should be considered only when overhead or sidedraft hoods are impractical. Air can be exhausted through a slotted baffle (e.g., downdraft cutting table—see Figure 24) or through a circular slot with a round source (Figure 25A) or two linear slots along the long sides of a rectangular source (Figure 25B). To achieve higher capturing effectiveness, the exhaust should be located as close to the source as possible. Capturing effectiveness decreases with an increase in source height and increases when the top of the source is located below the hood face surface. With a buoyant source, the air velocity induced by the exhaust should be equal to or greater than the air velocity in the plume above the source (Posokhin 1984).

The target airflow rate for a circular downdraft hood is

$$Q_o^* = 0.0314(q_{conv}d^5)^{1/3}\left(1 - 0.06\frac{q_{conv}^{vert}}{q_{conv}^{horiz}}\right)K_1K_v \qquad (31)$$

For a double linear slot downdraft hood,

$$Q_o^* = 0.05q_{conv}^{1/3}lbK_1K_v \qquad (32)$$

where

d = source diameter, m
l = source length, m
b = source width, m
q_{conv}^{vert} = convective heat component from the source vertical surfaces, W
q_{conv}^{horiz} = convective heat component from the source horizontal surface, W
K_1 = coefficient accounting for hood geometry that can be evaluated using graphs in Figure 25
K_v = coefficient accounting for room air movement V_r

$$= 1 + 44.7\sqrt{V_r^3\frac{d}{q_{conv}}} \quad \text{for circular downdraft hood} \qquad (33)$$

$$= 1 + 44.7\sqrt{V_r^3\frac{b}{q_{conv}}} \quad \text{for double slot downdraft hood} \qquad (34)$$

Example 3. A downdraft hood is to be designed to capture a contaminant from a rectangular source $l \times b \times h$ = 0.6 m × 0.5 m × 0 m. Convective heat component of the source q_{conv} = 1000 W. Room air movement V_r = 0.4 m/s. Two exhaust slots with a width b = 100 mm are located at the distance B_1 = 0.6 m and B_2 = 0.8 m. Determine the exhaust airflow rate.

Solution: Using the graph in Figure 25 for B_2/B_1 = 0.8/0.6 = 1.33, and B_1/b = 0.6/0.5 = 1.2, obtain K_1 = 5. Coefficient K_v accounting for room air movement [Equation (34)] is

$$K_v = 1 + 44.77\sqrt{0.4^3\frac{0.5}{1000}} = 1.25$$

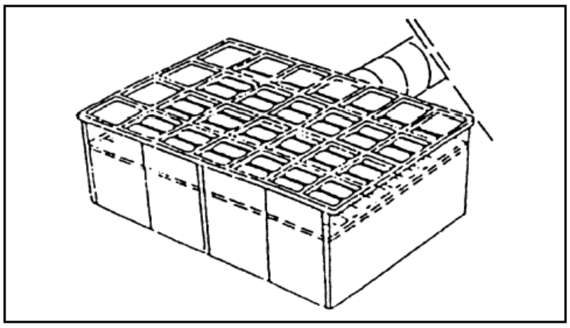
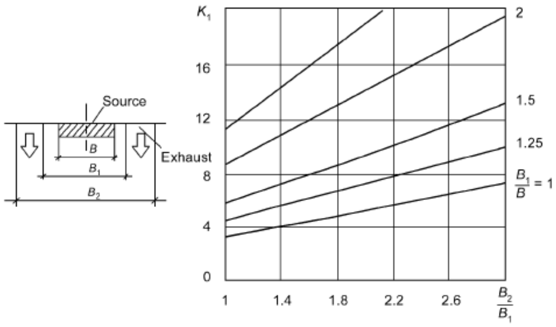
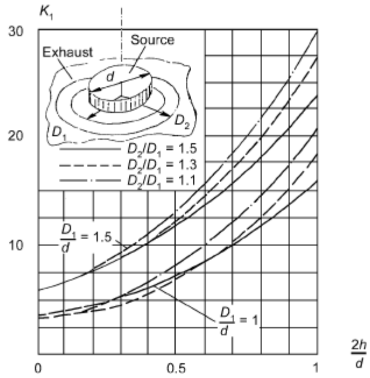


Fig. 24 Downdraft Welding Table



A. Circular slot with round source



B. Linear slots along the long sides of rectangular source

Fig. 25 K_1 Coefficient Evaluation for Downdraft Hoods

where

$$C = \frac{1}{1 + 3.74(\rho_g/\rho_{air})} \tag{36}$$

V_{min} = minimum velocity along jet, m/s
Δp = excessive pressure inside the process equipment, Pa
ρ_{air} = density of room air, kg/m^3
ρ_g = density of gas mixture releasing through the aperture in the process equipment, kg/m^3

The supply and exhaust airflow rates Q_{sup} and Q_o, m^3/s, can be determined as follows:

For a nonattached jet,

$$Q_{sup} = 0.435\frac{V_{min}}{\overline{V}_{min}}a\sqrt{bl} \tag{37}$$

$$Q_{sup} = 0.205\frac{V_{min}}{\overline{V}_{min}}alK_1K_v \tag{38}$$

For a wall jet,

$$Q_{sup} = 0.31\frac{V_{min}}{\overline{V}_{min}}a\sqrt{bl} \tag{39}$$

$$Q_{sup} = 0.103\frac{V_{min}}{\overline{V}_{min}}alK_1K_v \tag{40}$$

where

$\overline{V}_{min}$ = from graph in Figure 27
$\overline{B}$ = relative width of exhaust hood
= $B/2l$ for a nonattached jet and B/l for a wall jet
B = width of exhaust hood, m
a = length of exhaust hood, m
b = width of supply slot, m
K_1 = coefficient accounting for hood geometry can be evaluated using graphs in Figure 28
K_v = coefficient accounting for room air movement V_r

$$= 1 + \frac{V_r}{V_{min}} \tag{41}$$

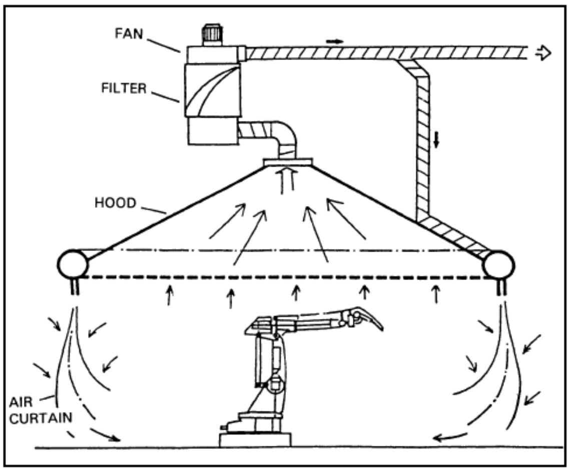


Fig. 29 Push-Pull Hood over Welding Robot

The following are some design considerations:

- Push-pull hoods are economically feasible if $l > 1$ m.
- The jet should be considered a wall jet when the distance H between the supply nozzle and the vertical surface is smaller than $0.15l$. Otherwise, the jet is nonattached.
- When flange width $h > H + B$, the hood is treated as an opening in an infinite surface; when $h \le H + B$, the hood is treated as freestanding.
- The value of the minimum velocity V_{min} along the jet should be greater than 1.5 m/s.
- The width b of the supply air slot is typically chosen to be $0.01l$. However, it should be greater than 5 mm to prevent fouling. The length a of the supply slot should be equal to the length of the aperture.
- The supply air velocity V_o should not exceed 1.5 m/s. This can be achieved by selection of the appropriate slot width b.

Example 4. A push-pull hood is to capture a contaminant from an oven aperture. The surplus pressure in the oven Δp = 2 Pa, and the temperature inside the oven t_g = 800°C (ρ_g = 0.329 kg/m^3). Canopy hood is installed at the height of l = 1.2 m from the low edge of the oven aperture. The hood projection B = 0.576 m, and the hood width is equal to the aperture width a = 1.8 m; the aperture height is 1 m. The room air velocity near the hood V_r = 0.4 m/s and the room air temperature t_{air} = 20°C (ρ_{air} = 1.2 kg/m^3). Determine the supply and exhaust airflow rates.

__Solution:__ Using the graph in Figure 27 for $\overline{B}$ = 0.576/(2 × 1.2) = 0.24, obtain $\overline{V}_{min}$ = 1.
From Equations (35) and (36) obtain parameter C and velocity V_{min}:

$$C = \frac{1}{1 + 3.74(0.329/1.2)} = 0.494$$

$$V_{min} = 9.9\sqrt{\frac{2}{0.329}}\left(\frac{\sqrt{1 + 142 \times 0.494^2} - 1}{89 \times 0.494^2}\right) = 5.59 \text{ m/s}$$

Assuming b = 0.025 m, calculate supply airflow rate [Equation (37)]:

$$Q_{sup} = 0.435 \times \frac{5.59}{1} \times 1.8\sqrt{0.025 \times 1.2} = 0.76 \text{ m}^3/\text{s}$$

Coefficient K_v accounting for room air movement [Equation (41)]:

$$K_v = 1 + \frac{0.4}{5.59} = 1.07$$

From the graph in Figure 28, K_1 = 1.
The exhaust airflow rate [Equation (38)]:

$$Q_{sup} = 0.205 \times \frac{5.59}{1} \times 1.8 \times 1.2 \times 1 \times 1.07 = 2.65 \text{ m}^3/\text{s}$$

Push-Pull Hood above Contaminated Area. A canopy hood with an incorporated slotted nozzle installed around the perimeter of the hood is used to prevent contaminant transfer from contaminated areas, for example, the operating zone of one or several welding robots (Figure 29), where enclosing hoods or other types of nonenclosing hoods are impractical (U.S. Patent). Air supplied through the nozzle creates steady air curtain protection along the contour. Due to the negative pressure created by the hood, the air curtain jet turns at or below the level of the contaminant source toward the center. To minimize the supply airflow rate, the nozzle is equipped with a honeycomb attachment that produces a low-turbulence jet. The width of the nozzle can be determined as follows:

$$b = \frac{A/P}{45\left(\frac{A}{PH}\right)^2\left(0.566\sqrt{\frac{H}{b}} - 1\right)^2 - 0.25\left(0.566\sqrt{\frac{H}{b}} + 1\right)^2} \tag{42}$$

where

b = nozzle width, m
A = hood cross-sectional area, m²
P = hood perimeter, m
H = height of hood above contaminant source, m

Push-Pull Protection System. These systems are used (Strongin et al. 1986; Strongin and Marder 1988) to prevent contaminant release from process equipment when the process requires that entering and/or exiting apertures remain open (e.g., conveyer painting chambers, cooling tunnels, etc.). The open aperture must be equipped with a tunnel and supply and exhaust air systems (Figure 30). The aperture is protected by the air jet(s) supplied through one or two slots installed along one side or two opposite sides of the tunnel and directed at angle α = 80 to 85° to the tunnel cross section. Air supplied through the slot(s) is thus directed toward the incoming room air. Moving along the tunnel, the jet(s) slow down, and their dynamic pressure is converted into static pressure, preventing room air from entering the chamber. After reaching the point with a zero centerline velocity, the jet(s) make a U-turn and redirect into the chamber. The air jet(s) can be supplied vertically (with supply air ducts installed along vertical walls) or horizontally (with supply air ducts installed along horizontal walls). The distance X (Figure 30) from the entrance of a tunnel (with cross-sectional area $B \times H$) to the supply slot location should be greater than or equal to $5B$ with a single vertical jet ($5H$ with a single horizontal jet) and $2.5B$ ($2.5H$) when air is supplied by two jets.

The air supply slot is equipped with diverging vanes (angle β between 30 to 90°) creating an air jet with an increased angle of divergence; the number n of these vanes should be greater than or equal to $\beta/10$. The increased angle of divergence of supply air jets allows a decrease in the distance X between the tunnel entrance and the slot.

Airflow rate supplied by the jet is determined as

$$Q_o^* = \sqrt{\frac{A_o b_o L_o \Delta p}{J}} \tag{43}$$

where

A_o = cross-sectional area of the tunnel, m²
b_o = supply slot width, m
L_o = supply slot length, m
J = supply jet parameter

$$= \sin\alpha + 2.5\frac{A_o}{A_c}\left[2.13(1+\psi)^2 + \left(\frac{\psi}{1+1/\psi}\right)^2 - \psi^2\right] \tag{44}$$

$$\text{for } \psi = \frac{Q_{exh}}{Q_o}$$

Δp = chamber to room pressure difference, Pa

$$= 0.5gH(\rho_{room} - \rho_c) \tag{45}$$

H = chamber height, m
g = gravitational acceleration, 9.8 m/s²
ρ_{room} = room chamber air density, kg/m³
ρ_c = chamber air density, kg/m³

The minimum airflow rate to be exhausted outside from the chamber and the corresponding amount of outdoor air to be supplied through the slot should dilute the contaminants in the chamber to the desired concentration. In the case of prevention of contaminant release from a drying chamber, the solvent vapor concentration should not exceed 25% of the lower explosive limit $C_{exp(min)}$. In this case, the exhaust airflow rate can be determined as follows:

$$Q_{exh} = \frac{GK}{0.25C_{exp(min)}} \tag{46}$$

where

G = amount of vapor release into the chamber, mg/s
K = coefficient accounting for the nonuniformity of solvent evaporation and other irregularities; typically, $2 \le K \le 5$
$C_{exp(min)}$ = lower explosive limit of pollutant, mg/m³

OTHER LOCAL EXHAUST SYSTEM COMPONENTS

Duct Design and Construction

Duct Considerations. The second component of a local exhaust ventilation system is the duct through which contaminated air is transported from the hood(s). Round ducts are preferred because they (1) offer a more uniform air velocity to resist settling of material and (2) can withstand the higher static pressures normally found in exhaust systems. When design limitations require rectangular ducts, the aspect ratio (height-to-width ratio) should be as close to unity as possible.

Minimum transport velocity is the velocity required to transport particulates without settling. Table 5 lists some generally accepted transport velocities as a function of the nature of the contaminants (ACGIH 1998). The values listed are typically higher than theoretical and experimental values to account for (1) damage to ducts, which would increase system resistance and reduce volumetric flow and duct velocity; (2) duct leakage, which tends to decrease velocity in the duct system upstream of the leak; (3) fan wheel corrosion or erosion and/or belt slippage, which could reduce fan volume; and (4) reentrainment of settled particulate caused by improper operation of the exhaust system. Design velocities can be higher than the minimum transport velocities but should never be significantly lower.

When particulate concentrations are low, the effect on fan power is negligible. Standard duct sizes and fittings should be used to cut cost and delivery time. Information on available sizes and the cost of nonstandard sizes can be obtained from the contractor(s).

Table 5 Contaminant Transport Velocities

Nature of Contaminant	Examples	Minimum Transport Velocity, m/s
Vapor, gases, smoke	All vapors, gases, smoke	Usually 5 to 10
Fumes	Welding	10 to 13
Very fine light dust	Cotton lint, wood flour, litho powder	13 to 15
Dry dusts and powders	Fine rubber dust, molding powder dust, jute lint, cotton dust, shavings (light), soap dust, leather shavings	15 to 20
Average industrial dust	Grinding dust, buffing lint (dry), wool jute dust (shaker waste), coffee beans, shoe dust, granite dust, silica flour, general material handling, brick cutting, clay dust, foundry (general), limestone dust, asbestos dust in textile industries	18 to 20
Heavy dust	Sawdust (heavy and wet), metal turnings, foundry tumbling barrels and shakeout, sand-blast dust, wood blocks, hog waste, brass turnings, cast-iron boring dust, lead dust	20 to 23
Heavy and moist dust	Lead dust with small chips, moist cement dust, asbestos chunks from transite pipe cutting machines, buffing lint (sticky), quicklime dust	23 and up

Source: Adapted from *Industrial Ventilation: A Manual of Recommended Practice* (ACGIH 1998).

where

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

Push-Pull Protection System. These systems are used (Strongin et al. 1986; Strongin and Marder 1988) to prevent contaminant release from process equipment when the process requires that entering and/or exiting apertures remain open (e.g., conveyer painting chambers, cooling tunnels, etc.). The open aperture must be equipped with a tunnel and supply and exhaust air systems (Figure 30). The aperture is protected by the air jet(s) supplied through one or two slots installed along one side or two opposite sides of the tunnel and directed at angle α = 80 to 85° to the tunnel cross section. Air supplied through the slot(s) is thus directed toward the incoming room air. Moving along the tunnel, the jet(s) slow down, and their dynamic pressure is converted into static pressure, preventing room air from entering the chamber. After reaching the point with a zero centerline velocity, the jet(s) make a U-turn and redirect into the chamber. The air jet(s) can be supplied vertically (with supply air ducts installed along vertical walls) or horizontally (with supply air ducts installed along horizontal walls). The distance X (Figure 30) from the entrance of a tunnel (with cross-sectional area $B \times H$) to the supply slot location should be greater than or equal to $5B$ with a single vertical jet ($5H$ with a single horizontal jet) and $2.5B$ ($2.5H$) when air is supplied by two jets.

The air supply slot is equipped with diverging vanes (angle β between 30 to 90°) creating an air jet with an increased angle of divergence; the number n of these vanes should be greater than or equal to $\beta/10$. The increased angle of divergence of supply air jets allows a decrease in the distance X between the tunnel entrance and the slot.

Airflow rate supplied by the jet is determined as

$$Q_o^* = \sqrt{\frac{A_o b_o L_o \Delta p}{J}} \tag{43}$$

where

A_o = cross-sectional area of the tunnel, m^2
b_o = supply slot width, m
L_o = supply slot length, m
J = supply jet parameter

$$= \sin\alpha + 2.5\frac{A_o}{A_c}\left[2.13(1+\psi)^2 + \left(\frac{\psi}{1+1/\psi}\right)^2 - \psi^2\right] \tag{44}$$

$$\text{for } \psi = \frac{Q_{exh}}{Q_o}$$

Δp = chamber to room pressure difference, Pa

$$= 0.5gH(\rho_{room} - \rho_c) \tag{45}$$

H = chamber height, m
g = gravitational acceleration, 9.8 m/s^2
ρ_{room} = room chamber air density, kg/m^3
ρ_c = chamber air density, kg/m^3

The minimum airflow rate to be exhausted outside from the chamber and the corresponding amount of outdoor air to be supplied through the slot should dilute the contaminants in the chamber to the desired concentration. In the case of prevention of contaminant release from a drying chamber, the solvent vapor concentration should not exceed 25% of the lower explosive limit $C_{exp(min)}$. In this case, the exhaust airflow rate can be determined as follows:

$$Q_{exh} = \frac{GK}{0.25C_{exp(min)}} \tag{46}$$

where

G = amount of vapor release into the chamber, mg/s
K = coefficient accounting for the nonuniformity of solvent evaporation and other irregularities; typically, $2 \le K \le 5$
$C_{exp(min)}$ = lower explosive limit of pollutant, mg/m^3

OTHER LOCAL EXHAUST SYSTEM COMPONENTS

Duct Design and Construction

Duct Considerations. The second component of a local exhaust ventilation system is the duct through which contaminated air is transported from the hood(s). Round ducts are preferred because they (1) offer a more uniform air velocity to resist settling of material and (2) can withstand the higher static pressures normally found in exhaust systems. When design limitations require rectangular ducts, the aspect ratio (height-to-width ratio) should be as close to unity as possible.

Minimum transport velocity is the velocity required to transport particulates without settling. Table 5 lists some generally accepted transport velocities as a function of the nature of the contaminants (ACGIH 1998). The values listed are typically higher than theoretical and experimental values to account for (1) damage to ducts, which would increase system resistance and reduce volumetric flow and duct velocity; (2) duct leakage, which tends to decrease velocity in the duct system upstream of the leak; (3) fan wheel corrosion or erosion and/or belt slippage, which could reduce fan volume; and (4) reentrainment of settled particulate caused by improper operation of the exhaust system. Design velocities can be higher than the minimum transport velocities but should never be significantly lower.

When particulate concentrations are low, the effect on fan power is negligible. Standard duct sizes and fittings should be used to cut cost and delivery time. Information on available sizes and the cost of nonstandard sizes can be obtained from the contractor(s).

Table 5 Contaminant Transport Velocities

Nature of Contaminant	Examples	Minimum Transport Velocity, m/s
Vapor, gases, smoke	All vapors, gases, smoke	Usually 5 to 10
Fumes	Welding	10 to 13
Very fine light dust	Cotton lint, wood flour, litho powder	13 to 15
Dry dusts and powders	Fine rubber dust, molding powder dust, jute lint, cotton dust, shavings (light), soap dust, leather shavings	15 to 20
Average industrial dust	Grinding dust, buffing lint (dry), wool jute dust (shaker waste), coffee beans, shoe dust, granite dust, silica flour, general material handling, brick cutting, clay dust, foundry (general), limestone dust, asbestos dust in textile industries	18 to 20
Heavy dust	Sawdust (heavy and wet), metal turnings, foundry tumbling barrels and shakeout, sand-blast dust, wood blocks, hog waste, brass turnings, cast-iron boring dust, lead dust	20 to 23
Heavy and moist dust	Lead dust with small chips, moist cement dust, asbestos chunks from transite pipe cutting machines, buffing lint (sticky), quicklime dust	23 and up

Source: Adapted from *Industrial Ventilation: A Manual of Recommended Practice* (ACGIH 1998).

Duct Size Determination. The size of the round duct attached to the hood can be calculated using Equation (1) for the volumetric flow rate and Table 5 for the minimum transport velocity.

Example 5. Suppose the contaminant captured by the hood in Example 1 requires a minimum transport velocity of 15 m/s. What diameter round duct should be specified?

Solution: From Equation (1), the duct area required is

$$A = 0.702/15 = 0.047\ \text{m}^2$$

Generally, the area calculated will not correspond to a standard duct size. The area of the standard size chosen should be less than that calculated. For this example, a 225 mm diameter duct with an area of 0.0398 m^2 should be chosen. The actual duct velocity is then

$$V = 0.702/0.0398 = 17.6\ \text{m/s}$$

Duct Losses. Chapter 32 of the 1997 *ASHRAE Handbook—Fundamentals* covers the basics of duct design and the design of metalworking exhaust systems. The design method presented there is based on total pressure loss, including the fitting coefficients; ACGIH (1998) calculates static pressure loss. Loss coefficients can be found in Chapter 32 of the 1997 *ASHRAE Handbook—Fundamentals* and in the ASHRAE *Duct Fitting Database* (ASHRAE 1994), which runs on a personal computer.

For systems conveying particulates, elbows with a centerline radius-to-diameter ratio (r/D) greater than 1.5 are the most suitable. If $r/D \le 1.5$, abrasion in dust-handling systems can reduce the life of elbows. Elbows, especially those with large diameters, are often made of seven or more gores. For converging flow fittings, a 30° entry angle is recommended to minimize energy losses and abrasion in dust-handling systems (Fitting ED5-1 in Chapter 32 of the 1997 *ASHRAE Handbook—Fundamentals*).

Where exhaust systems handling particulates must allow for a substantial increase in future capacity, required transport velocities can be maintained by providing open-end stub branches in the main duct. Air is admitted through these stub branches at the proper pressure and volumetric flow rate until the future connection is installed. Figure 31 shows such an air bleed-in. The use of outside air minimizes replacement air requirements. The size of the opening can be calculated by determining the pressure drop required across the orifice from the duct calculations. Then the orifice velocity pressure can be determined from one of the following equations:

$$p_{v,o} = \frac{\Delta p_{t,o}}{C_o} \qquad (47)$$

or

$$p_{v,o} = \frac{\Delta p_{s,o}}{C_o + 1} \qquad (48)$$

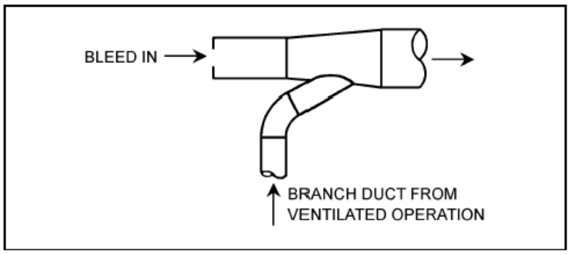


Fig. 31 Air Bleed-In

where

$p_{v,o}$ = orifice velocity pressure, Pa
$\Delta p_{t,o}$ = total pressure to be dissipated across orifice, Pa
$\Delta p_{s,o}$ = static pressure to be dissipated across orifice, Pa
C_o = orifice loss coefficient referenced to the velocity at the orifice cross-sectional area, dimensionless (Figure 15)

Equation (47) should be used if total pressure through the system is calculated; Equation (48) should be used if static pressure through the system is calculated. Once the velocity pressure is known, Equation (15) or (16) can be used to determine the orifice velocity. Equation (1) can then be used to determine the orifice size.

Integrating Duct Segments. Most systems have more than one hood. If the pressures are not designed to be the same for merging parallel airstreams, the system adjusts to equalize pressure at the common point; however, the flow rates of the two merging airstreams will not necessarily be the same as designed. As a result, the hoods can fail to control the contaminant adequately, exposing workers to potentially hazardous contaminant concentrations. Two design methods ensure that the two pressures will be equal. The preferred design self-balances without external aids. This procedure is described in the section on Industrial Exhaust System Duct Design in Chapter 32 of the 1997 *ASHRAE Handbook—Fundamentals*. The second design, which uses adjustable balance devices such as blast gates or dampers, is not recommended, especially when abrasive material is conveyed.

Duct Construction. Elbows and converging flow fittings should be made of thicker material than the straight duct, especially if abrasives are conveyed. In some cases, elbows must be constructed with a special wear strip in the heel. When corrosive material is present, alternatives such as special coatings or different duct materials (fibrous glass or stainless steel) can be used. Industrial duct construction is described in Chapter 16 of the 2000 *ASHRAE Handbook—Systems and Equipment*. Refer to SMACNA (1990) for industrial duct construction standards.

Air Cleaners

Air-cleaning equipment is usually selected to (1) conform to federal, state, or local emissions standards and regulations; (2) prevent reentrainment of contaminants to work areas; (3) reclaim usable materials; (4) permit cleaned air to recirculate to work spaces and/or processes; (5) prevent physical damage to adjacent properties; and (6) protect neighbors from contaminants.

Factors to consider when selecting air-cleaning equipment include the type of contaminant (number of components, particulate versus gaseous, and concentration), the contaminant removal efficiency required, the disposal method, and the air or gas stream characteristics. See Chapters 24 and 25 of the 2000 *ASHRAE Handbook—Systems and Equipment* for information on equipment for removing airborne contaminants. A qualified applications engineer should be consulted when selecting equipment.

The cleaner's pressure loss must be added to overall system pressure calculations. In some cleaners, specifically some fabric filters, the loss increases as operation time increases. The system design should incorporate the maximum pressure drop of the cleaner, or hood flow rates will be lower than designed during most of the duty cycle. Also, fabric collector losses are usually given only for a clean air plenum. A reacceleration to the duct velocity, with the associated entry losses, must be calculated in the design phase. Most other cleaners are rated flange-to-flange with reacceleration included in the loss.

Air-Moving Devices

The type of air-moving device used depends on the type and concentration of contaminant, the pressure rise required, and the allowable noise levels. Fans are usually selected. Chapter 18 of the 2000 *ASHRAE Handbook—Systems and Equipment* describes available fans and refers the reader to Air Movement and Control Association (AMCA) *Publication* 201, Fans and Systems, for proper connection of the fan(s) to the system. The fan should be located downstream of the air cleaner whenever possible to (1) reduce possible abrasion of the fan wheel blades and (2) create negative pressure in the air cleaner so that air leaks into it and positive control of the contaminant is maintained.

In some instances, however, the fan is located upstream from the cleaner to help remove dust. This is especially true with cyclone collectors, for example, which are used in the woodworking industry. If explosive, corrosive, flammable, or sticky materials are handled, an injector can transport the material to the air-cleaning equipment. Injectors create a shear layer that induces airflow into the duct. Injectors should be the last choice because their efficiency seldom exceeds 10%.

Energy Recovery

The transfer of energy from exhausted air to replacement air may be economically feasible, depending on (1) the location of the exhaust and replacement air ducts, (2) the temperature of the exhausted gas, and (3) the nature of the contaminants being exhausted. The efficiency of heat transfer depends on the type of heat recovery system used. Rotary air-to-air exchangers have the best efficiency, 70-80%. Cross flow fixed-surface plate exchangers and energy recovery loops with liquid coupled coils have efficiencies of 50 and 60% (Aro and Kovula 1992).

If exhausted air contains particulate matter (e.g., dust, lint) or oil mist, the exhausted air should be filtered to prevent fouling the heat exchanger. If the exhausted air contains gaseous and vaporous contaminants such as hydrocarbons and water-soluble chemicals, their effect on the heat recovery device should be investigated (Aro and Kovula 1992).

Exhaust Stacks

The exhaust stack must be designed and located to prevent the reentrainment of discharged air into supply system inlets. The building's shape and surroundings determine the atmospheric airflow over it. Chapter 15 of the 1997 *ASHRAE Handbook—Fundamentals* and Chapter 43 of this volume cover exhaust stack design.

If rain protection is important, stackhead design is preferable to weathercaps. Weathercaps, which are not recommended, have three disadvantages:

1. They deflect air downward, increasing the chance that contaminants will recirculate into air inlets.
2. They have high friction losses.
3. They provide less rain protection than a properly designed stackhead.

Figure 32 contrasts the flow patterns of weathercaps and stackheads. Loss data for weathercaps and stackheads are presented in the ASHRAE *Duct Fitting Database* (ASHRAE 1994). Losses in the straight duct form of stackheads are balanced by the pressure regain at the expansion to the larger-diameter stackhead.

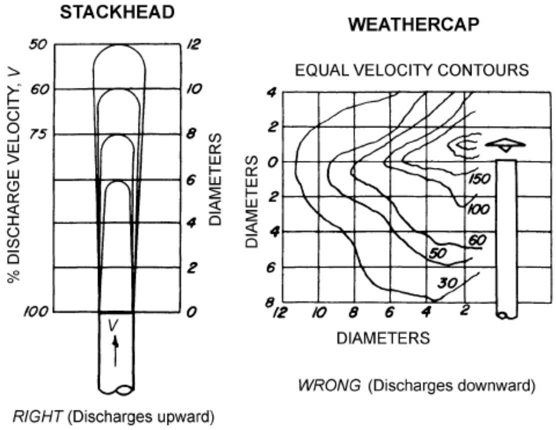


Fig. 32 Comparison of Flow Pattern for Stackheads and Weathercaps

OPERATION

System Testing

After installation, an exhaust system should be tested to ensure that it operates properly with the required flow rates through each hood. If the actual installed flow rates are different from the design values, they should be corrected before the system is used. Testing is also necessary to obtain baseline data to determine (1) compliance with federal, state, and local codes; (2) by periodic inspections, whether maintenance on the system is needed to ensure design operation; (3) whether a system has sufficient capacity for additional airflow; and (4) whether system leakage is acceptable. AMCA *Publication* 203 and Chapter 9 of ACGIH (1998) contain detailed information on the preferred methods for testing systems.

Operation and Maintenance

Periodic inspection and maintenance are required for the proper operation of exhaust systems. Systems are often changed or damaged after installation, resulting in low duct velocities and/or incorrect volumetric flow rates. Low duct velocities can cause the contaminant to settle and plug the duct, reducing flow rates at the affected hoods. Adding hoods to an existing system can change volumetric flow at the original hoods. In both cases, changed hood volumes can increase worker exposure and health risks. The maintenance program should include (1) inspecting ductwork for particulate accumulation and damage by erosion or physical abuse, (2) checking exhaust hoods for proper volumetric flow rates and physical condition, (3) checking fan drives, and (4) maintaining air-cleaning equipment according to manufacturers' guidelines.

REFERENCES

ACGIH. 1998. *Industrial ventilation: A manual of recommended practice*, 23rd edition. Committee on Industrial Ventilation, American Conference of Governmental Industrial Hygienists, Cincinnati, OH.

Aksenov, A.A. and A.V. Gudzovskii. 1994. Numerical simulation of turbulent thermal plumes in the stratified space. Proceedings of the First National Conference on Heat Transfer. Part 2—Free Convection. 21-25 November. Moscow (in Russian).

Alden, J.L. and J.M. Kane. 1982. Design of Industrial Ventilation Systems, 5th ed. Industrial Press, New York.

AMCA. 1995. Fans and systems. *Publication* 201-95. Air Movement and Control Association International, Arlington Heights, IL.

AMCA. 1995. Field performance measurement of fan systems. *Publication* 203-95.

Anichkhin, A.G. and G.N. Anichkhina. 1984. Ventilation of laboratories in Research Institutions. In "Energy efficiency improvement of mechanical systems". Nauka, Moscow.

Aro, T. and K. Kovula. 1992. Learning from experiences with Industrial Ventilation. Center for the Analysis and Dissemination of Demonstrated Energy Technologies. AIR-IX Consulting Engineers, Finland.

ASHRAE. 1994. *Duct fitting database.*

Bastress, E., J. Niedzwecki, and A. Nugent. 1974. Ventilation required for grinding, buffing, and polishing operations. U.S. Department of Health, Education, and Welfare. NIOSH. *Publication* No. 75-107. Washington, D.C.

Boshnyakov, E.N. 1975. Local exhaust with air curtains. *Water Supply and Sanitary Techniques*, #3. Moscow (in Russian).

Burgess, W.A., M.J. Ellenbecker, and R.D. Treitman. 1989. *Ventilation for control of the work environment*. John Wiley and Sons, New York.

Caplan, K.J. and G.W. Knutson. 1977. The effect of room air challenge on the efficiency of laboratory fume hoods. *ASHRAE Transactions* 83(1): 141-156.

Caplan, K.J. and G.W. Knutson. 1978. Laboratory fume hoods: Influence of room air supply. *ASHRAE Transactions* 82(1):522-37.

Cesta, T. 1988. Capture of pollutants from a buoyant point source using a lateral exhaust hood with and without assistance from air curtains. Proceedings of the 2nd International Symposium on Ventilation for Contamination Control, Ventilation '88. Pergamon Press, UK.

Chambers, D.T. 1993. *Local exhaust ventilation: A philosophical review of the current state-of-the-art with particular emphasis on improved worker protection*. DCE, Leicester, UK.

Davidson, L. 1989. Numerical simulation of turbulent flow in ventilated rooms. Ph.D. thesis, Chalmers University of Technology, Sweden.

DallaValle, J.M. 1952. *Exhaust hoods*, 2nd ed. Industrial Press, New York.

Elinskii, I.I. 1989. Ventilation and heating of galvanic shops of machine-building plants. Mashinostroyeniye, Moscow (in Russian).

Elterman, V.M. 1980. Ventilation of chemical plants. Moscow: KHIMIA (in Russian).

Fletcher, B. 1977. Center line velocity characteristics of rectangular unflanged hoods and slots under suction. *Ann. Occup. Hyg.* 20:141-46.

Fuller, F.H. and A.W. Etchells. 1979. The rating of laboratory hood performance. *ASHRAE Journal* 21(10):49-53.

Garrison, R.P. 1977. Nozzle performance and design for high-velocity/low-volume exhaust ventilation. Ph.D. thesis. University of Michigan, Ann Arbor, MI.

Hagopian, J.H. and E.K. Bastress. 1976. Recommended industrial ventilation guidelines. U.S. Department of Health, Education, and Welfare, NIOSH. *Publication* No. 76-162. Washington, D.C.

Heinsohn, R.J. 1991. *Industrial ventilation: Engineering principles*. John Wiley and Sons, New York.

Hemeon, W.C.L. 1963. *Plant and process ventilation*, p. 77. Industrial Press, New York.

Hinds, W. 1982. Aerosol technology: Properties, behavior, and measurement of airborne particles. John Wiley and Sons, New York.

Holman, J.P. 1989. *Heat transfer*. McGraw Hill, Singapore.

Idelchik, I.E., G.R. Malyavskaya, O.G. Martynenko, and E. Fried. 1986. *Handbook of hydraulic resistance*, 2nd ed. Hemisphere Publishing Corporation, subsidiary of Harper and Row, New York.

Ivanitskaya, M.Yu., and V.I. Kunitsa. 1974. Experimental studies of thermal plumes above a round heat source. Proceedings of TsNIIPromzdanii. V.37. TsNIIPromzdanii, Moscow (in Russian).

Kofoed, P. 1991. Thermal plumes in ventilated rooms. Ph.D. thesis, Aalborg University, Denmark.

Kofoed, P. and P.V. Nielsen. 1991. Thermal plumes in ventilated rooms—Vertical volume flux influenced by enclosing walls. 12th AIVC Conference, Ottawa.

Kuz'mina, L.V. 1959. Sidedraft and cornerdraft hoods. Transactions of the Institutes for Labor Protection of the VTsSPS (All-Union Central Council of Trade Unions). No. 2. Moscow: PROFIZDAT, pp. 25-34 (in Russian).

Ljungqvist, B. and C. Waering. 1988. Some observations on "modern" design of fume cupboards. Proceedings of the 2nd International Symposium on Ventilation for Contaminant Control, Ventilation '88. Pergamon Press, UK.

Morton, B.R., G. Taylor, and J.S. Turner. 1956. Turbulent gravitational convection from maintained and instantaneous sources. Proceedings of Royal Society. Vol. 234A, p. 1.

Mundt, E. 1992. Convection flows in rooms with temperature gradients—Theory and measurements. RoomVent '92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 3. Aalborg.

Nielsen, P.V. 1993. Displacement ventilation—Theory and design. Department of Building Technology and Structural Engineering. Aalborg University, Denmark.

NFPA. 1995. Standard for ovens and furnaces. ANSI/NFPA *Standard* 86-95. National Fire Protection Association, Quincy, MA.

Popiolec, Z. 1981. Problems of testing and mathematical modeling of plumes above human body and other extensive heat sources. A4-seria. No. 54. KTH, Stockholm.

Posokhin, V.N. 1984. Design of local ventilation systems for process equipment with heat and gas release. Mashinostroyeniye, Moscow (in Russian).

Posokhin, V.N. and V.A. Broida. 1980. Local exhausts incorporated with air curtains. Hydromechanics and heat transfer in sanitary technique equipment. KHTI, Kazan (in Russian).

Posokhin, V.N. and A.M. Zhivov. 1997. Principles of local exhaust design. Proceedings of the 5th International Symposium on Ventilation for Contaminant Control. Vol.1. The Canadian Environment Industry Association (CEIA), Ottawa.

Romeyko, N.F., N.E. Siromyatnikova, and E.V. Schibraev. 1976. Design of air curtains near an oven opening supplied with a hood. Heating and Ventilation. Proceedings of the A.I. Mikoyan Institute of Civil Engineers (in Russian).

Schaelin, A. and P. Kofoed. 1992. Numerical simulation of thermal plumes in rooms. RoomVent '92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 1. Aalborg, Denmark.

Schmidt, W. 1941. Turbulente Ausbreitung eines Stromes erhitzter Luft. ZAMM. Bd. 21 # 5 (in German).

Sciola, V. 1993. The practical application of reduced flow push-pull plating tank exhaust systems. 3rd International Symposium on Ventilation for Contaminant Control, Ventilation '91 (Cincinnati, OH).

Shepelev, I.A. 1961. Turbulent convective stream above a heat source. Proceedings of Acad. Sci. USSR. Mechanics and Machinery Construction #4 (in Russian).

Skäret, E. 1986. Ventilasjonsteknikk. Textbook. Institute of Heating, Ventilation and Sanitary Techniques, NTH. Trondheim (in Norwegian).

Skistad, H. 1994. *Displacement ventilation*. Research Studies Press, John Wiley and Sons, West Sussex. UK.

SMACNA. 1977. Round industrial duct standards. Sheet Metal and Air Conditioning Contractors' National Association, Vienna, VA.

SMACNA. 1980. Rectangular industrial duct construction standards.

Stoler, V.D. and Yu. L. Savelyev. 1977. Push-pull systems design for etching tanks. *Heating, Ventilation, Water Supply, and Sewage Systems Design* 8(124). TsINIS, Moscow (in Russian).

Stroiizdat. 1992. *Designer's guide*. Ventilation and air conditioning, 4th ed. Part 3(1). Stroiizdat, Moscow (in Russian).

Strongin, A.S. and M.L. Marder. 1988. Complex solution of painting shops ventilation. Proceedings of the conference "Utilization of Natural Resources and New Ventilation and Dust Transportation Systems Design". Penza (in Russian).

Strongin, A.S., M.Yu. Ivanitskaya, and E.A. Visotskaya. 1986. Studies of the application of air curtains in tunnels for local ventilation. *Heating and Ventilation. Transactions* of TsNIIpromzdanii (in Russian).

Tyaglo, I.G. and I.A. Shepelev. 1970. Air flow near an exhaust opening. *Vodosnabzheniye i Sanitarnaya Tekhnika* #5, pp. 24-25 (in Russian).

U.S. Patent. Device for removal of deleterious impurities from room atmosphere. U.S. Patent # 5,716,268. February 1998.

Zeldovitch, Y.B. 1937. Fundamental principles for free convective plumes. *Journal of Experimental and Technical Physics* 7(12). Moscow (in Russian).

Zhivov, A.M. 1993. Principles of source capturing and general ventilation design for welding premises. *ASHRAE Transactions* 99(1):979-86.

Zhivov, A.M. and J.T. Ashe. 1997. Principles of welding fume control. Proceedings of the 5th International Symposium on Ventilation for Contaminant Control. Vol. 1. The Canadian Environment Industry Association (CEIA), Ottawa.

Zhivov, A.M., L.L. Christianson, and G.L. Riskowski. 1997. Influence of space air movement on hood performance. ASHRAE *Research Project* RP-744.

BIBLIOGRAPHY

AICVF. 1991. Principles of airflow applied to the HVAC field. Collection des guides de l'AICVF. Association d'Ingenierie de Chauffage, Ventilation et Froid, Paris (in French).

Balchin, N.C. (Ed.) 1991. Health and safety in welding and allied processes, 4th ed. Abington Publishing, Cambridge, UK.

Baturin, V.V. 1972. *Fundamentals of industrial ventilation*, 3rd English ed. Pergamon Press, New York.

Braconnier, R. 1988. Bibliographic review of velocity field in the vicinity of local exhaust hood openings. *American Industrial Hygiene Association Journal* 49(4):185-98.

Brandt, A.D., R.J. Steffy, and R.G. Huebscher. 1947. Nature of air flow at suction openings. *ASHVE Transactions* 53:55-76.

British Occupational Hygiene Society (BOHS). 1987. Controlling airborne contaminants in the workplace. *Technical Guide* No. 7. Science Review Ltd. and H&H Sci. Consult., Leeds, UK.

Burgess, W.A., M.J. Ellenbecker, and R.D. Treitman. 1989. *Ventilation for control of the work environment*. John Wiley and Sons, New York.

Caplan, K.J. and G.W. Knutson. 1977. The effect of room air challenge on the efficiency of laboratory fume hoods. *ASHRAE Transactions* 83(1): 141-156.

Caplan, K.J. and G.W. Knutson. 1978. Laboratory fume hoods: Influence of room air supply. *ASHRAE Transactions* 82(1):522-37.

Cesta, T. 1988. Capture of pollutants from a buoyant point source using a lateral exhaust hood with and without assistance from air curtains. Proceedings of the 2nd International Symposium on Ventilation for Contamination Control, Ventilation '88. Pergamon Press, UK.

Chambers, D.T. 1993. *Local exhaust ventilation: A philosophical review of the current state-of-the-art with particular emphasis on improved worker protection*. DCE, Leicester, UK.

Davidson, L. 1989. Numerical simulation of turbulent flow in ventilated rooms. Ph.D. thesis, Chalmers University of Technology, Sweden.

DallaValle, J.M. 1952. *Exhaust hoods*, 2nd ed. Industrial Press, New York.

Elinskii, I.I. 1989. Ventilation and heating of galvanic shops of machine-building plants. Mashinostroyeniye, Moscow (in Russian).

Elterman, V.M. 1980. Ventilation of chemical plants. Moscow: KHIMIA (in Russian).

Fletcher, B. 1977. Center line velocity characteristics of rectangular unflanged hoods and slots under suction. *Ann. Occup. Hyg.* 20:141-46.

Fuller, F.H. and A.W. Etchells. 1979. The rating of laboratory hood performance. *ASHRAE Journal* 21(10):49-53.

Garrison, R.P. 1977. Nozzle performance and design for high-velocity/low-volume exhaust ventilation. Ph.D. thesis. University of Michigan, Ann Arbor, MI.

Hagopian, J.H. and E.K. Bastress. 1976. Recommended industrial ventilation guidelines. U.S. Department of Health, Education, and Welfare, NIOSH. *Publication* No. 76-162. Washington, D.C.

Heinsohn, R.J. 1991. *Industrial ventilation: Engineering principles*. John Wiley and Sons, New York.

Hemeon, W.C.L. 1963. *Plant and process ventilation*, p. 77. Industrial Press, New York.

Hinds, W. 1982. Aerosol technology: Properties, behavior, and measurement of airborne particles. John Wiley and Sons, New York.

Holman, J.P. 1989. *Heat transfer*. McGraw Hill, Singapore.

Idelchik, I.E., G.R. Malyavskaya, O.G. Martynenko, and E. Fried. 1986. *Handbook of hydraulic resistance*, 2nd ed. Hemisphere Publishing Corporation, subsidiary of Harper and Row, New York.

Ivanitskaya, M.Yu., and V.I. Kunitsa. 1974. Experimental studies of thermal plumes above a round heat source. Proceedings of TsNIIPromzdanii. V.37. TsNIIPromzdanii, Moscow (in Russian).

Kofoed, P. 1991. Thermal plumes in ventilated rooms. Ph.D. thesis, Aalborg University, Denmark.

Kofoed, P. and P.V. Nielsen. 1991. Thermal plumes in ventilated rooms—Vertical volume flux influenced by enclosing walls. 12th AIVC Conference, Ottawa.

Kuz'mina, L.V. 1959. Sidedraft and cornerdraft hoods. Transactions of the Institutes for Labor Protection of the VTsSPS (All-Union Central Council of Trade Unions). No. 2. Moscow: PROFIZDAT, pp. 25-34 (in Russian).

Ljungqvist, B. and C. Waering. 1988. Some observations on "modern" design of fume cupboards. Proceedings of the 2nd International Symposium on Ventilation for Contaminant Control, Ventilation '88. Pergamon Press, UK.

Morton, B.R., G. Taylor, and J.S. Turner. 1956. Turbulent gravitational convection from maintained and instantaneous sources. Proceedings of Royal Society. Vol. 234A, p. 1.

Mundt, E. 1992. Convection flows in rooms with temperature gradients—Theory and measurements. RoomVent '92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 3. Aalborg.

Nielsen, P.V. 1993. Displacement ventilation—Theory and design. Department of Building Technology and Structural Engineering. Aalborg University, Denmark.

NFPA. 1995. Standard for ovens and furnaces. ANSI/NFPA *Standard* 86-95. National Fire Protection Association, Quincy, MA.

Popiolec, Z. 1981. Problems of testing and mathematical modeling of plumes above human body and other extensive heat sources. A4-seria. No. 54. KTH, Stockholm.

Posokhin, V.N. 1984. Design of local ventilation systems for process equipment with heat and gas release. Mashinostroyeniye, Moscow (in Russian).

Posokhin, V.N. and V.A. Broida. 1980. Local exhausts incorporated with air curtains. Hydromechanics and heat transfer in sanitary technique equipment. KHTI, Kazan (in Russian).

Posokhin, V.N. and A.M. Zhivov. 1997. Principles of local exhaust design. Proceedings of the 5th International Symposium on Ventilation for Contaminant Control. Vol.1. The Canadian Environment Industry Association (CEIA), Ottawa.

Romeyko, N.F., N.E. Siromyatnikova, and E.V. Schibraev. 1976. Design of air curtains near an oven opening supplied with a hood. Heating and Ventilation. Proceedings of the A.I. Mikoyan Institute of Civil Engineers (in Russian).

Schaelin, A. and P. Kofoed. 1992. Numerical simulation of thermal plumes in rooms. RoomVent '92. Proceedings of the Third International Conference on Air Distribution in Rooms. Vol. 1. Aalborg, Denmark.

Schmidt, W. 1941. Turbulente Ausbreitung eines Stromes erhitzter Luft. ZAMM. Bd. 21 # 5 (in German).

Sciola, V. 1993. The practical application of reduced flow push-pull plating tank exhaust systems. 3rd International Symposium on Ventilation for Contaminant Control, Ventilation '91 (Cincinnati, OH).

Shepelev, I.A. 1961. Turbulent convective stream above a heat source. Proceedings of Acad. Sci. USSR. Mechanics and Machinery Construction #4 (in Russian).

Skäret, E. 1986. Ventilasjonsteknikk. Textbook. Institute of Heating, Ventilation and Sanitary Techniques, NTH. Trondheim (in Norwegian).

Skistad, H. 1994. *Displacement ventilation*. Research Studies Press, John Wiley and Sons, West Sussex. UK.

SMACNA. 1977. Round industrial duct standards. Sheet Metal and Air Conditioning Contractors' National Association, Vienna, VA.

SMACNA. 1980. Rectangular industrial duct construction standards.

Stoler, V.D. and Yu. L. Savelyev. 1977. Push-pull systems design for etching tanks. *Heating, Ventilation, Water Supply, and Sewage Systems Design* 8(124). TsINIS, Moscow (in Russian).

Stroiizdat. 1992. *Designer's guide*. Ventilation and air conditioning, 4th ed. Part 3(1). Stroiizdat, Moscow (in Russian).

Strongin, A.S. and M.L. Marder. 1988. Complex solution of painting shops ventilation. Proceedings of the conference "Utilization of Natural Resources and New Ventilation and Dust Transportation Systems Design". Penza (in Russian).

Strongin, A.S., M.Yu. Ivanitskaya, and E.A. Visotskaya. 1986. Studies of the application of air curtains in tunnels for local ventilation. *Heating and Ventilation. Transactions* of TsNIIpromzdanii (in Russian).

Tyaglo, I.G. and I.A. Shepelev. 1970. Air flow near an exhaust opening. *Vodosnabzheniye i Sanitarnaya Tekhnika* #5, pp. 24-25 (in Russian).

U.S. Patent. Device for removal of deleterious impurities from room atmosphere. U.S. Patent # 5,716,268. February 1998.

Zeldovitch, Y.B. 1937. Fundamental principles for free convective plumes. *Journal of Experimental and Technical Physics* 7(12). Moscow (in Russian).

Zhivov, A.M. 1993. Principles of source capturing and general ventilation design for welding premises. *ASHRAE Transactions* 99(1):979-86.

Zhivov, A.M. and J.T. Ashe. 1997. Principles of welding fume control. Proceedings of the 5th International Symposium on Ventilation for Contaminant Control. Vol. 1. The Canadian Environment Industry Association (CEIA), Ottawa.

Zhivov, A.M., L.L. Christianson, and G.L. Riskowski. 1997. Influence of space air movement on hood performance. ASHRAE *Research Project* RP-744.

BIBLIOGRAPHY

AICVF. 1991. Principles of airflow applied to the HVAC field. Collection des guides de l'AICVF. Association d'Ingenierie de Chauffage, Ventilation et Froid, Paris (in French).

Balchin, N.C. (Ed.) 1991. Health and safety in welding and allied processes, 4th ed. Abington Publishing, Cambridge, UK.

Baturin, V.V. 1972. *Fundamentals of industrial ventilation*, 3rd English ed. Pergamon Press, New York.

Braconnier, R. 1988. Bibliographic review of velocity field in the vicinity of local exhaust hood openings. *American Industrial Hygiene Association Journal* 49(4):185-98.

Brandt, A.D., R.J. Steffy, and R.G. Huebscher. 1947. Nature of air flow at suction openings. *ASHVE Transactions* 53:55-76.

British Occupational Hygiene Society (BOHS). 1987. Controlling airborne contaminants in the workplace. *Technical Guide* No. 7. Science Review Ltd. and H&H Sci. Consult., Leeds, UK.