Chapter

19

Local Exhaust Ventilation

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*T**his chapter describes the following aspects of* local exhaust ventilation (LEV) *systems: typical components, principles of operation, methods to evaluate performance, and ways to resolve problems. It is designed to give safety and industrial hygiene practitioners sufficient knowledge to determine where LEV is needed, participate with engineers and others during system design, and evaluate and troubleshoot existing systems.*

LEV systems are covered in this chapter while dilution ventilation is described in Chapter 20. The Bibliography for both chapters appears at the end of Chapter 20, Dilution Ventilation for Industrial Workplaces.

INTRODUCTION

Ventilation is an important method for reducing employee exposures to airborne contaminants. However, ventilation is only one way to reduce exposures and may not be as economical or effective as other control techniques, such as reducing emissions into the work area by sealing equipment to prevent contaminant release or substituting less toxic or volatile chemicals.

There are two major types of industrial ventilation:

➣ *Dilution systems* reduce the concentration of contaminants released into the workroom by mixing with air flowing through the room. Either natural or mechanically induced air movement can be used to dilute contaminants.

➣ *Local exhaust ventilation (LEV) systems* capture or contain contaminants at their source before they escape into the workroom environment. The main advantage of local exhaust systems is that they remove contaminants rather than just dilute them. Even with LEV, some airborne contaminants may still be in the workroom air due to uncontrolled sources or less than 100% collection efficiency at

the hoods. A second major advantage of local exhaust is that these systems require less airflow than dilution ventilation systems in the same applications. The total airflow is especially important for plants that are heated or cooled since heating and air conditioning costs are an important operating expense.

LOCAL EXHAUST SYSTEM COMPONENTS

A typical local exhaust system consists of the following elements (Figure 19–1):

➤ *Hoods*—any point where air is drawn into the ventilation system to capture or control contaminants. Some hoods are designed to fit around existing machinery while others are located next to the contaminant source. Even a plain duct opening is called a "hood" if that is where air enters the system. Different hoods work in different ways: some reach out and capture contaminants; others catch contaminants thrown into the hood; still others contain contaminants released inside the hood and prevent them from escaping into the workroom. Some hood designs feature a long, narrow slot to distribute the airflow along the length of an open surface tank, welding bench, or laboratory hood.

➤ *Ducts*—the network of piping that connects the hoods and other system components.

➤ *Fan*—the air-moving device that provides the energy to draw air and contaminants into the exhaust system and through the ducts and other components. It functions by inducing a negative pressure or suction in the ducts leading to the hoods and positive pressure in the system after the fan. The fan converts electrical power into pressure and increased air velocity.

➤ *Air Cleaner*—a device to remove airborne materials that may be needed before the exhaust air is discharged into the community environment. Air cleaners to remove both solid (particulate) and gaseous contaminants are available.

Although not formally part of an LEV system, the arrangement for supplying makeup air to the work area that is being ventilated is also very important. An insufficient quantity of makeup air may cause poor fan operation, inefficient combustion in furnaces, drafts, and problems with slamming doors in the work area.

An LEV system is usually planned to fit existing machinery or industrial processes. A hood shape and location are chosen based upon the source of contamination. The airflow volume into each hood is then determined from reference sources such as the ACGIH *Industrial Ventilation Manual* (ACGIH, 1998). Next, the need for an air cleaner is determined, and, if needed, a type and size are selected. With this information the duct layout can be determined and the duct diameters calculated. Finally, the fan type and size needed to draw the required amount of air while overcoming friction and other resistance can be determined. After installation, the
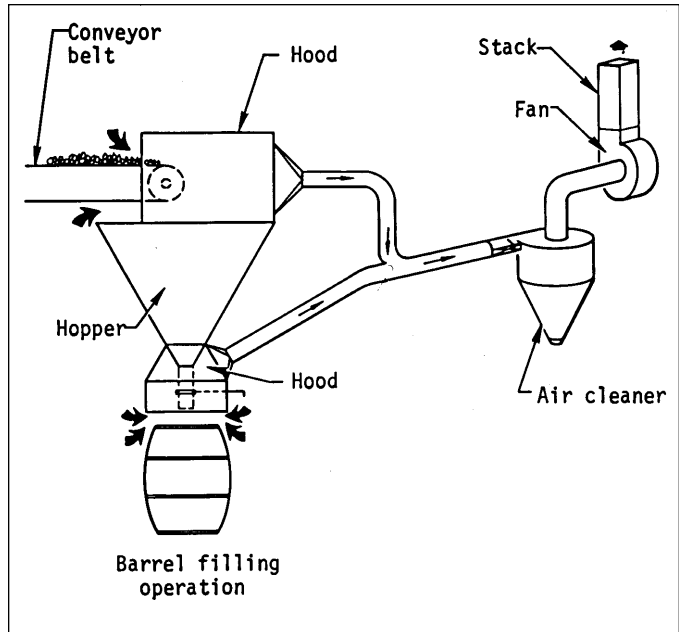

Figure 19–1. Typical Local Exhaust Ventilation system. (Source: McDermott, 2000.)

system is tested to assure that it is meeting design criteria. System design is beyond the scope of this chapter; however, the references listed at the end of Chapter 20 describe how to design LEV systems (McDermott, 2000; ACGIH, 1998).

Hoods

The hood is the most important part of an LEV system. No local exhaust system will work properly unless enough of the contaminants are retained or captured by the hoods to reduce the concentration in the workroom air below acceptable limits. Both the design and location of the hoods are critical in determining whether a system will work. A poor hood design may prevent the ventilation system from performing adequately.

Hood selection is an area where the health and safety professional can make a significant contribution since the keys to good hood selection include the following: a knowledge of hood and airflow principles, an understanding of the plant processes, and a familiarity with employee work patterns around each process. In many plants, the health and safety staff has the best overall understanding of these three areas. Fortunately, once the fundamentals of hood selection are understood, there is a ready reference source for specific hood designs. The ACGIH *Manual* contains almost 150 design plates showing layout, design parameters, and airflow recommendations for different hoods (ACGIH, 1998).

HOOD TYPES

Three different types of hoods are used in local ventilation systems: capturing hoods, enclosures, and receiving hoods. Each works according to one of the following principles to control contaminants:
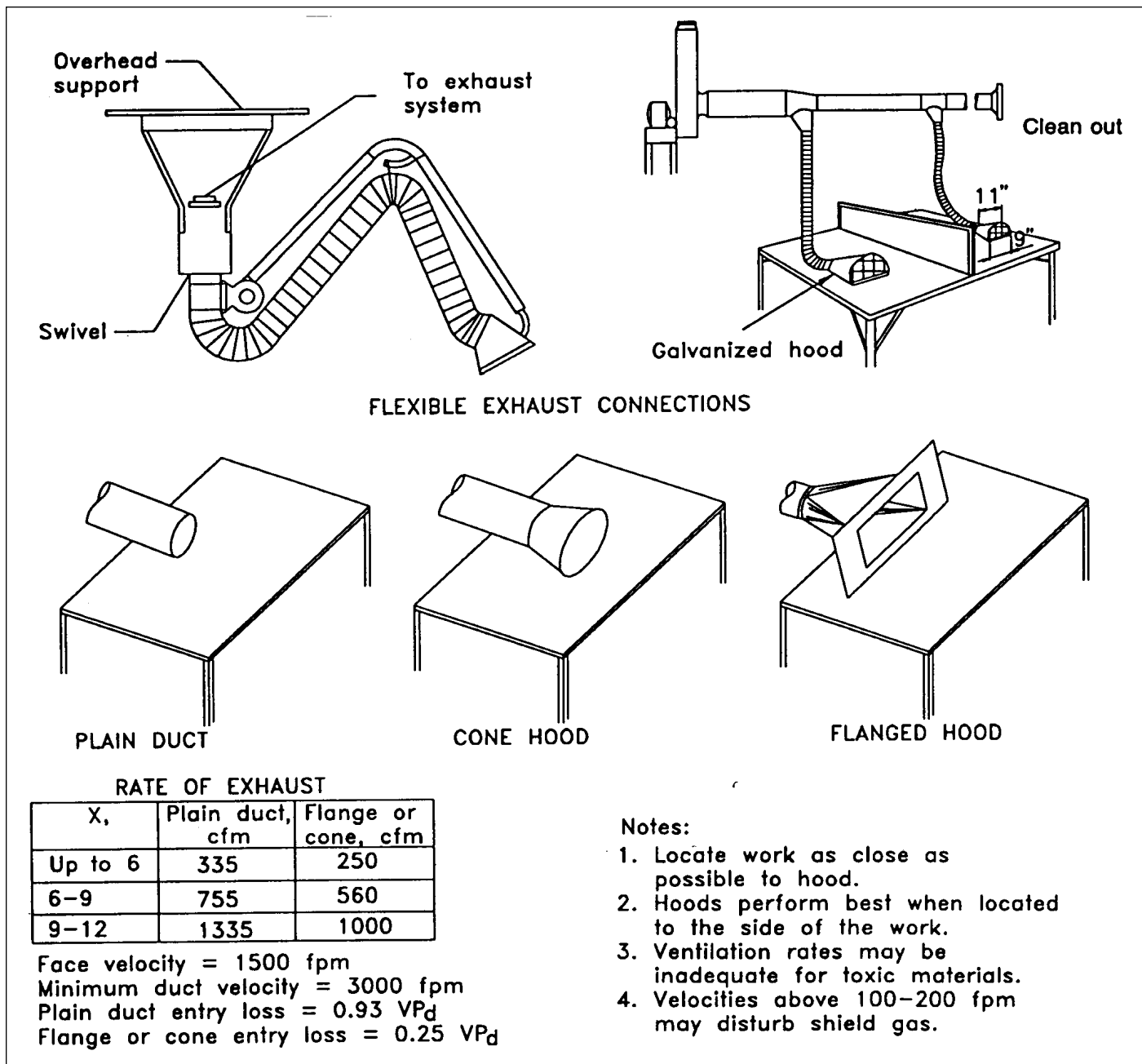
Overhead support

To exhaust system

Clean out

Swivel

11"

9"

Galvanized hood

FLEXIBLE EXHAUST CONNECTIONS

PLAIN DUCT

CONE HOOD

FLANGED HOOD

RATE OF EXHAUST

X,	Plain duct, cfm	Flange or cone, cfm
Up to 6	335	250
6–9	755	560
9–12	1335	1000

Face velocity = 1500 fpm
Minimum duct velocity = 3000 fpm
Plain duct entry loss = 0.93 VP_d
Flange or cone entry loss = 0.25 VP_d

Notes:
1. Locate work as close as possible to hood.
2. Hoods perform best when located to the side of the work.
3. Ventilation rates may be inadequate for toxic materials.
4. Velocities above 100–200 fpm may disturb shield gas.

Figure 19–2. Capturing hood for welding fumes. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice,* 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

➤ *Capturing Hoods*—Hoods that "reach out" to capture contaminants in the workroom air (Figure 19–2). Airflow into the hood is calculated to generate sufficient capture velocity in the air space in front of the hood. The needed capture velocity depends on the amount and motion of contaminants and contaminated air (Table 19–A). This type of hood is widely used since it can be placed alongside the contaminant source rather than surrounding it as with an enclosure. The primary disadvantage is that large air volumes may be needed to generate an adequate capture velocity at the contaminant source.

Other disadvantages are that crossdrafts in the workroom can severely degrade the capture efficiency of the hood, and the "reach" of most capturing hoods is limited to about 2 ft from the hood opening.

➤ *Enclosures*—Hoods that surround the contaminant source as much as possible. Contaminants are kept inside the enclosure by air flowing in through openings in the enclosure (Figure 19–3). Laboratory hoods and paint spray booths are typical examples of this hood type. The quantity of air required for contaminant control is calculated by multiplying the inward air velocity needed to

Table 19–A. *Range of Capture Velocities*

Dispersion of Contaminant	Examples	Capture Velocity, ft/min
Released with practically no velocity into quiet air.	Evaporation from tank; degreasing.	50–100
Released at low velocity into moderately still air.	Spray booths; intermittent container filling; low-speed conveyor transfers; welding; plating; pickling.	100–200
Active generation into zone of rapid air motion.	Spray painting in shallow booths; barrel filling; conveyor loading; crushers.	200–500
Released at high initial velocity into zone of very rapid air motion.	Grinding; abrasive blasting; tumbling.	500–2000

In each category above, a range of capture velocities is shown. The proper choice of values depends on several factors:

Lower End of Range	Upper End of Range
1. Room air currents minimal or favorable to capture.	1. Disturbing room air currents.
2. Contaminants 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.

(Source: From American Conference of Governmental Industrial Hygienists (ACGIH®) *Industrial Ventilation: A Manual of Recommended Practice*, 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

prevent escape by the areas of the openings into the enclosure. The more complete the enclosure, the less airflow is needed for control. Employees generally do not work inside enclosures while contaminants are being generated, although they may reach into the enclosure as long as they do not breathe contaminated air. Due to low exhaust rates, enclosures are often the most economical hoods to install if the open area of the enclosure is not large. Inward face velocities of 100–150 ft/min are typical. Good room conditions are critical for proper enclo-
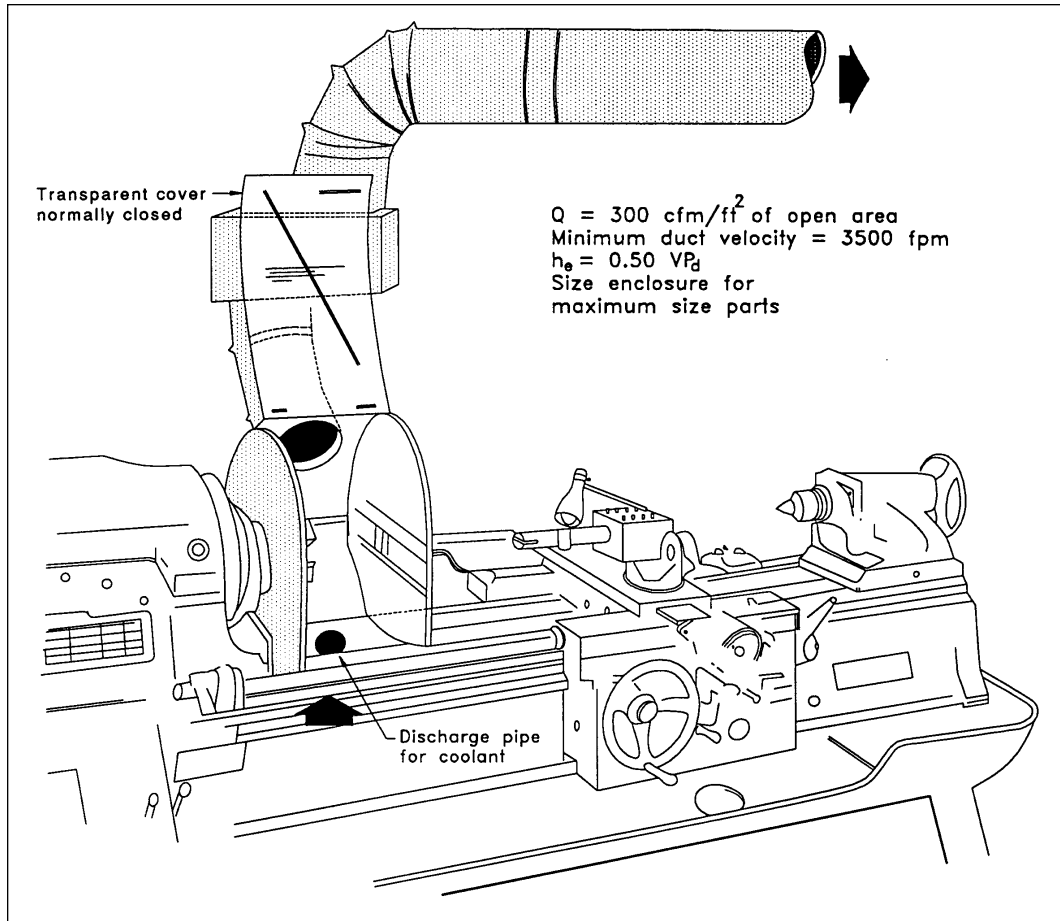

Transparent cover
normally closed

$Q = 300$ cfm/ft^2 of open area
Minimum duct velocity = 3500 fpm
$h_e = 0.50$ VP$_d$
Size enclosure for
maximum size parts

Discharge pipe
for coolant

Figure 19–3. Design details for an enclosure from the ACGIH *Manual.* (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice,* 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

sure performance, including sufficient replacement air, supply outlets located and designed to avoid drafts, and protection against disruptive air currents from open doors and foot traffic near the hoods.

➤ *Receiving Hoods*—Some processes "throw" a stream of contaminants in a specific direction. For example, a furnace may emit a hot stream of gases that rises above the unit. A grinder throws a stream of material tangentially from the point of contact between the grinding wheel and workpiece. The ideal hood for this type of process is one that is positioned so it catches the contaminants thrown at it (Figure 19–4). A major limitation to the use of receiving hoods is that gases, vapors, and the very small particles that can be inhaled and retained in the human respiratory system do not travel very far in air unless carried by moving air. This means that receiving hoods are not very useful for health protection ventilation systems unless the process emits quantities of hot air or air with sufficient velocity to carry the respirable contaminants into the hood.

In addition to these three major hood classifications, two special hood types are used in LEV systems:

➤ *Slot Hoods*—Some capturing hoods and enclosures feature a narrow "slot" to distribute the inward airflow across the entire hood. By definition, a slot is at least five times as long as it is high. A typical example is a long, open surface tank that has limited space for a hood yet has a need for good air distribution over its entire length (Figure 19–5). Similarly, a laboratory hood has slots along the back panel to develop more uniform air velocity through the hood opening. One disadvantage of a slot hood is that it creates more energy loss than a hood without a slot. This is due to the turbulence and high air velocity through the narrow slot. Extra suction is needed to move the air through the slot, which requires a larger fan than for a comparable system with no slot.

It is important to realize that a high slot velocity does not significantly increase the reach of the hood. The purpose of the slot is solely to distribute the inward velocity along the length of the slot. As a rule-of-thumb, a slot velocity of 2,000 ft/min often gives good air distribution without excessive pressure loss.

➤ *Canopy Hoods*—A canopy hood (Figure 19–6) generally can be used only as a receiving hood over hot processes to collect the gases and vapors rising into the hood. However, a canopy cannot be used when workers must lean over the tank or process because workers will breathe the contaminated air as the contaminants rise.

Canopies for *unheated* processes must be designed as capturing hoods. However, the large airflow volumes needed to develop adequate velocities below the canopy plus the two-foot limitation on capture distance often make these hoods impractical. The solution is another type of capturing hood, such as a side draft or slot hood, or an enclosure.

Ducts

Ducts carry air between the hoods, air cleaner, fan, and discharge stack. Common duct materials for LEV systems include galvanized steel, aluminum, stainless steel, plastics, and wire-wrapped fabric flexible duct. Stainless steel and specialty plastics are used where protection against corrosion is needed. Wire-wrapped flexible fabric ducts are often not recommended for LEV systems carrying particulates because of their tendency to accumulate settled material and sag unless rigidly supported, and because of the difficulty in cleaning out settled material.

Selecting duct diameters for a system is often a trade-off between initial and operating cost. Smaller diameter ducts are less expensive to fabricate and install than larger diameter ducts. However, the resulting higher duct velocities in smaller diameter ducts increase pressure losses, thus requiring a larger fan with higher power consumption. Systems carrying particulates generally need to maintain a certain minimum transport velocity to avoid material settling in the ducts. For common dusts, this velocity is often 3,000–4,000 ft/min. For more dense materials, larger particles, or sticky materials, the minimum velocity needed is higher. Although systems handling vapors and gases have no minimum duct velocity criteria, as a rule-of-thumb, duct velocities of 2,000–3,000 ft/min usually result in a good balance between initial duct construction cost and fan operating cost. Hood design diagrams in the ACGIH *Manual* usually specify duct velocity criteria for the system.

Air movement is always accompanied by friction where the air meets the duct surface. As a result, the air velocity close to the duct wall is low while at the center of the duct the velocity is higher than the overall average value. Figure 19–7 depicts a very simplified view of the duct velocity profile; any disturbances to smooth airflow such as elbows, branch duct entries, or air cleaners cause an uneven distribution that gradually returns to the typical profile illustrated in the figure.

In addition to friction, turbulence occurs in the ducts due to changes in air velocity and direction. Some loss will occur at every hood, elbow, duct enlargement, or duct junction. Since the fan must be large enough to move the required quantity of air while overcoming the friction and turbulent losses, it is important to avoid duct features that cause unnecessary pressure drop. These include narrow ducts, small radius elbows, and perpendicular junctions where two ducts join.

Another duct consideration is the duct segment just before the fan. The fan can do the greatest amount of work on incoming air only if the airflow into the fan is straight and uniform. Spinning or nonuniform flow patterns reduce the fan's air volume and/or static pressure output. Major reasons for poor flow patterns are elbows, dampers, duct junctions, or other flow disturbances near the fan. For an existing system, installing flow straighteners in the inlet duct can help to restore straight flow into the fan.
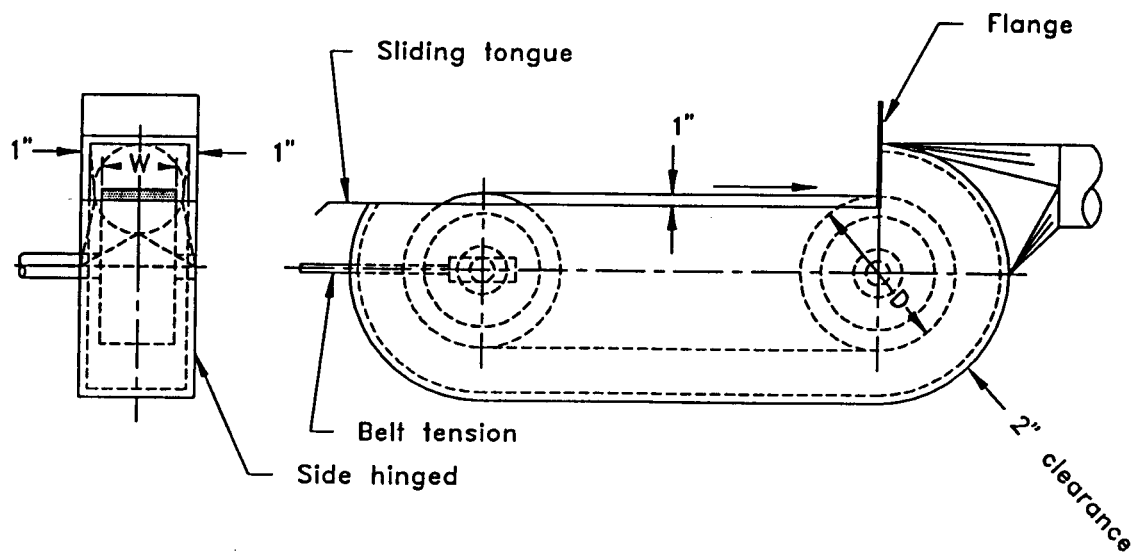
Belt width, inches	Exhaust flow rate, cfm
Up to 3	220
3 to 5	300
5 to 7	390
7 to 9	500
9 to 11	610
11 to 13	740

Minimum duct velocity = 3500 fpm, 4500 fpm if material is wet or sticky

h_e = 0.65 VP_d for straight take-off

= 0.45 VP_d for tapered take-off

Notes:

1. Consult applicable NFPA codes.
2. Caution: Do not mix ferrous and nonferrous metals in same exhaust system.

Figure 19–4. Metal polishing belt—design details for a receiving hood from the ACGIH *Manual.* (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice*, 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

The ACGIH *Manual* contains guidelines for other duct design parameters, such as pressure loss factors, provisions for clean-out ports, and duct wall thickness.

EXHAUST STACKS

Every LEV system should have at least a short, straight duct or exhaust stack attached to the fan outlet. This helps to change high, uneven velocity patterns at the fan outlet into a uniform flow and results in a phenomenon called *static pressure regain.* This permits the fan to be more efficient in moving air through the system.

The proper stack height, location, and discharge velocity are important in minimizing reentry of exhausted contaminants into the building and in avoiding problems when the
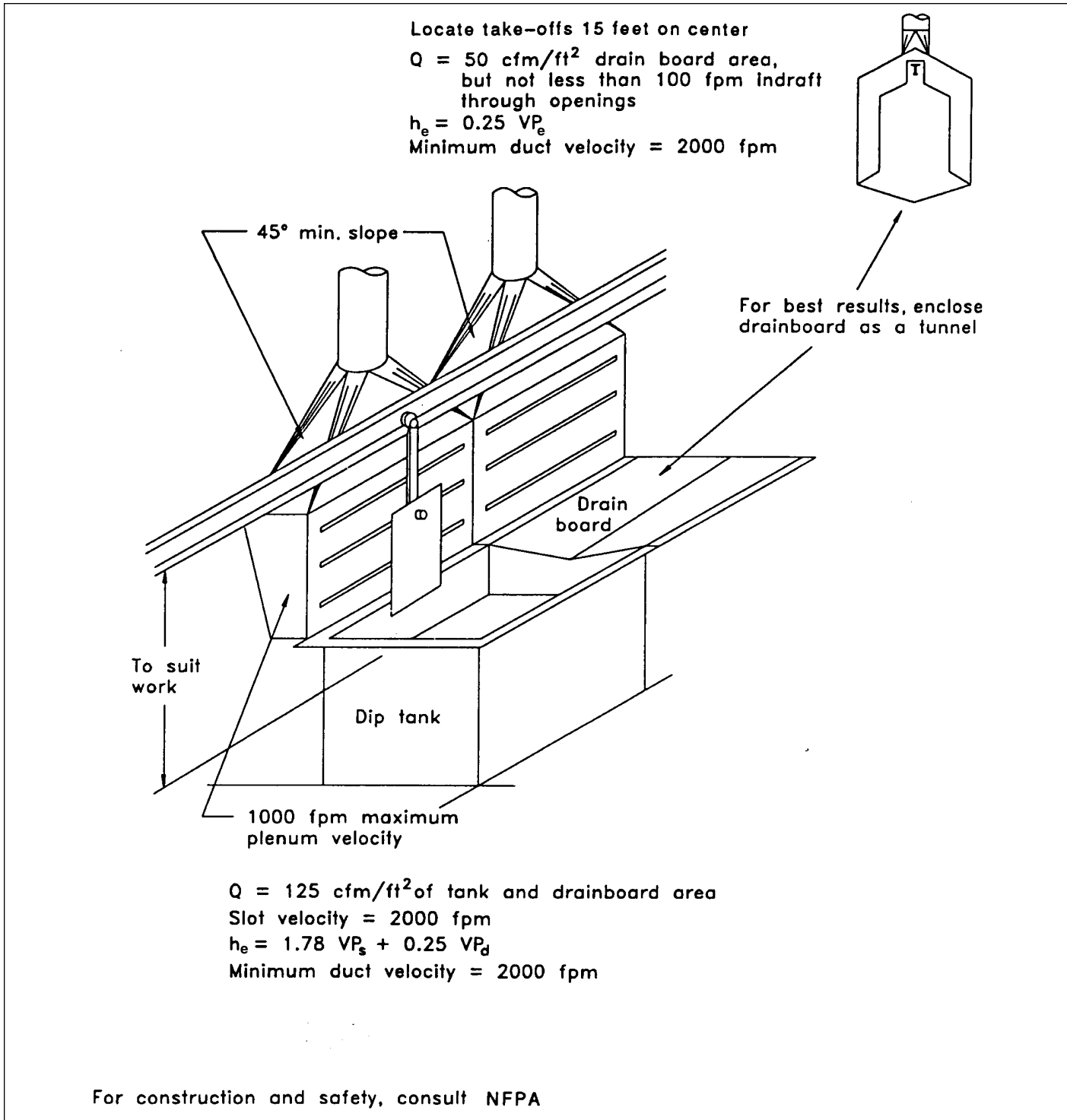
Locate take-offs 15 feet on center

$Q = 50$ cfm/ft^2 drain board area, but not less than 100 fpm indraft through openings

$h_e = 0.25$ VP$_e$

Minimum duct velocity = 2000 fpm

45° min. slope

For best results, enclose drainboard as a tunnel

Drain board

To suit work

Dip tank

1000 fpm maximum plenum velocity

$Q = 125$ cfm/ft^2 of tank and drainboard area

Slot velocity = 2000 fpm

$h_e = 1.78$ VP$_s$ + 0.25 VP$_d$

Minimum duct velocity = 2000 fpm

For construction and safety, consult NFPA

Figure 19–5. Slot hood to distribute airflow along an open surface tank. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice,* 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

contaminants reach ground level. Airflow patterns around buildings are complex, consisting of several discrete air turbulence and recirculating zones. Generally, the air layers near the roof tend to wash across the roof or circulate so they hit the downwind side of the building or the ground. The exhaust plume should be discharged above these layers so it will not contaminate intakes, and where sufficient dilution occurs before the plume reaches the ground or adjacent buildings. A high stack discharge velocity (3,000 ft/min or higher) helps to disperse contaminants since the air jet action
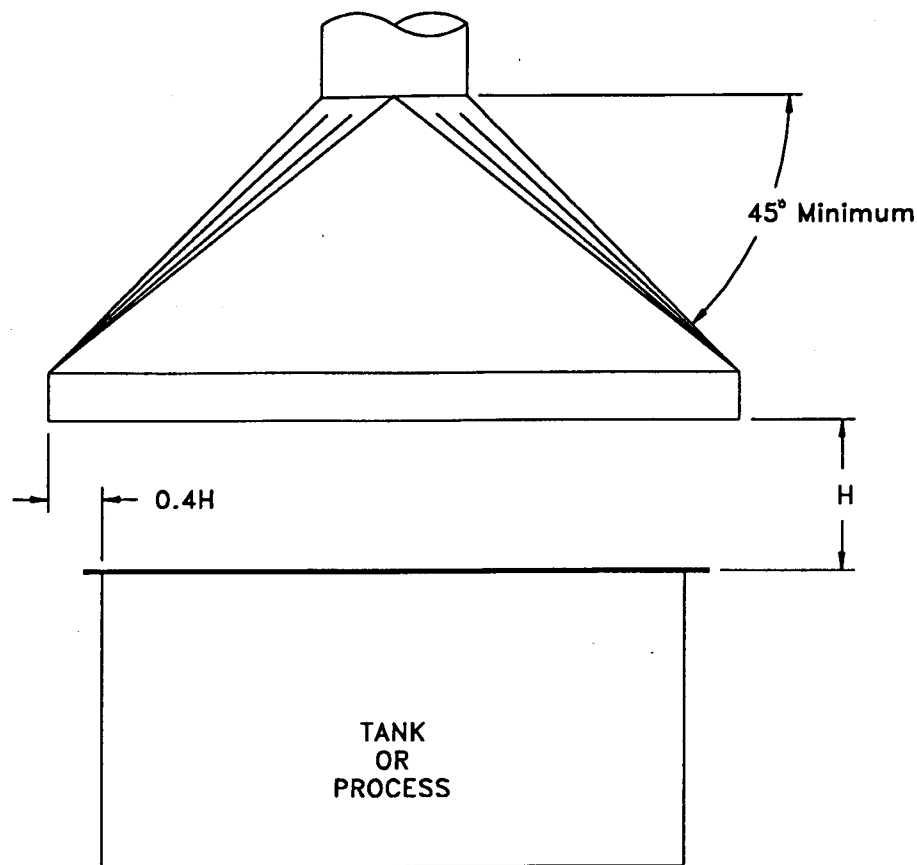
Not to be used where material is toxic and worker must bend over tank or process.
Side curtains are necessary when cross drafts are present.

$Q = 1.4PHV$	For open type canopy P = perimeter of tank, feet V = 50–500 fpm.
$Q = (W + L)HV$	For two sides adjacent enclosed W & L are open sides of hood V = 50–500 fpm.
$Q = WHV$ or LHV	For three sides enclosed (booth) V = 50–500 fpm.

$h_e = 0.25 \, VP_d$
Duct velocity = 1000–3000 fpm

Figure 19–6. Canopy hoods are usually used over hot processes since the contaminants rise into the hood. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice,* 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

can increase the effective stack height except under severe wind conditions.

Wind direction and velocity are other important factors. If there is a prevailing wind direction at the site, it should help to locate the stack on the downwind side of the roof. However, the location of the stack and air intakes should recognize that wind will often blow from other than the prevailing direction. A very low wind speed allows the plume to
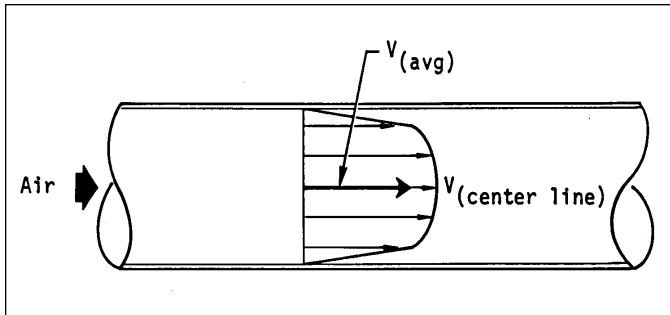
Figure 19–7. Velocity distribution in a duct. The average duct velocity is less than the centerline velocity. (Source: McDermott, 2000.)

rise due to the discharge velocity and any thermal head. As wind velocity increases, the first effect will be to decrease plume rise and the resulting dilution. Still higher winds will increase turbulence, which increases the dilution. Unfortunately, the occurrence of some reentry usually cannot be ruled out, so in very sensitive situations, air cleaners on the discharged air or relocation of air intakes may be needed to eliminate problems.

Fans

The fan generates the suction in the system that draws contaminated air into the hoods and through the ducts. A variety of different fans are available, but they all fall into one of two main classes: centrifugal fans or axial flow fans. Centrifugal fans move air by centrifugal action. Blades on a rotating fan wheel throw air outward from the center inlet at a higher velocity or pressure than air entering the fan. With axial fans, the air travels parallel to the fan shaft and leaves the fan in the same direction as it entered. A screw or propeller action produces airflow.

In LEV systems, centrifugal fans are more widely used than axial fans because they are usually quieter, less expensive to install and operate, and generate higher pressures than axial flow fans of the same airflow capacity. Centrifugal fans can be divided into three categories depending on the shape and setting of the fan wheel blades (Figure 19–8):

➤ *Radial-blade fans* (Figure 19–8a) have flat blades that extend straight out from the center hub. They are used for dust systems since their flat blades minimize the buildup of dust on the blades. These fans also have large openings between blades and are therefore less likely to clog. They can be built with thick blades to withstand erosion and impact damage from airborne solids. Their major disadvantage is that they are the least efficient fan for local exhaust systems. Their heavy construction adds to their cost.

➤ *Forward-curved blade fans* (Figure 19–8b) are useful when large volumes of air must be moved against moderate pressures with low noise levels. These fans have many cup-shaped blades that accelerate the air and discharge it at a higher velocity than the fan wheel tip is moving.

➤ *Backward-inclined/backward-curved blade fans* have blades that are inclined backward from the direction of fan wheel rotation (Figure 19–8c). The blades are of uniform thickness. If they are straight (flat), the fan is called a *backward-inclined blade fan*; if the blades are curved back, the fan is called a *backward-curved blade fan*. Since these fans are more efficient than the forward-curved fan, they are often used for handling large volumes of air containing little dust. *Airfoil* fans are a modification of the backward-curved blade fan. The blades of airfoil fans are shaped like the cross section of an airplane wing. This shape reduces noise and allows the fan to function smoothly without pulsing airflow through its entire operating range.

Fans perform at their maximum efficiency only when the airflow into the fan is smooth. Any design feature in the system that causes turbulence or spinning air motion at the fan inlet will reduce the fan's ability to move air and generate pressure. The most common cause of inlet problems is a duct elbow too close to the fan inlet. Elbows should be at least five duct diameters from the fan inlet unless turning vanes in the elbow or another method is used to straighten the airflow.

An elbow too close to the fan outlet will also decrease performance because of the high velocity, turbulent flow at the outlet. Elbows should be at least five duct diameters, and preferably 10 diameters, away from the fan outlet.

Air-Cleaning Devices

The purpose of this section is to give a broad overview of the types of air cleaners that are available. An important step in
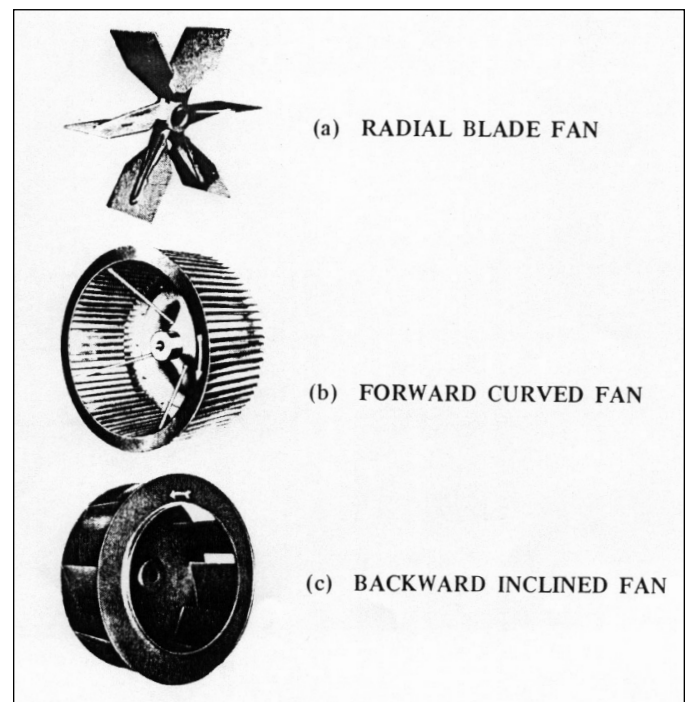

Figure 19–8. Common types of centrifugal fans. (Source: McDermott, 2000.)

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system design is the determination of whether an air cleaner is needed to reduce the amount of contaminants discharged to the environment. Local regulations usually are the major factor is this decision.

The ideal air cleaner for a specific application would have these features: low initial and operating cost, high efficiency for the contaminants, no decline in operating efficiency or any service interruptions between scheduled cleaning and maintenance cycles, and provisions for normal maintenance and cleaning without hazardous employee exposures.

The types of devices to consider depend primarily on the physical state of the contaminants (i.e., whether they are particulates or gases/vapors). For most situations, no single device is highly efficient for both small particulates and for gases/vapors. Scrubbing devices are widely used to collect some particles and gases or vapors in a single unit, but these combination units are generally not highly efficient for fine particles.

PARTICULATE REMOVAL

Typical air cleaners for particulates include the following:

➤ *Filters* trap particulates as the exhaust gas flows through a porous medium. Filters may be made of woven or felted (pressed) fabric, paper, or woven metal, depending on the application. They are available in a variety of configurations, such as mats, cartridges, bags, and envelopes. Filters have the general advantage of being able to handle varying exhaust gas flow rates and particle loadings. Filter devices fall into two major categories: disposable filters that often use inexpensive materials and are available in different configurations, and reusable filter elements in a housing that is equipped with a cleaning mechanism for periodic removal of trapped material. Selection of disposable or reusable filters is based on the expense of replacing the elements versus the added initial cost of the filter-cleaning mechanism.

➤ *Electrostatic precipitators* charge particles by means of an electric field that is strong enough to produce ions that adhere to the particles. The charged particles are then collected with a weaker electric field that causes the particles to migrate toward and adhere to the electrode with the opposite charge. Precipitators find greatest use in systems where gas volume is large and high collection efficiency for small particles is needed.

➤ *Cyclones* impart a circular motion to the exhaust gas that causes particulates to move to the outer part of the airstream where they impact the cyclone walls. Since air velocity is lower at the wall, the particulates drop down the wall into a collection hopper at the bottom. Cyclones may also be operated as wet collectors if a water spray is installed to wet the particles at the inlet. This increases the effective size of small particles, thus increasing collection efficiency.

➤ *Wet scrubbers* contact particles with water or another liquid and then collect the droplets. To collect extremely

fine particles, it is necessary to generate small droplets moving at high speed. Scrubbers can remove particles as small as 0.2 μm; however, the energy required to generate small droplets and cause adequate contact rises exponentially as the particle size decreases. Scrubbers that utilize absorption or chemical reaction as a collection mechanism are also widely used for gas and vapor removal.

The "baghouse" (Figure 19–9) is a typical example of a particulate air cleaner. It consists of tubular fabric filters arranged in a housing along with the cleaning mechanism, which can be an automatic or manual shaking device, a means of blowing air back through the bags from the clean side, or another method of dislodging the accumulated dust cake. Chunks of the cake that are dislodged during the cleaning cycle should be large enough so that they are not reentrained in the exhaust gas stream, or the section being cleaned should be isolated from the remainder of the baghouse during its cleaning cycle. Baghouses can collect practically all particles greater than 1 μm in diameter as well as a large percentage of submicron particles.

GAS AND VAPOR REMOVAL

Major removal techniques for gases and vapors are absorption, adsorption, and oxidation:
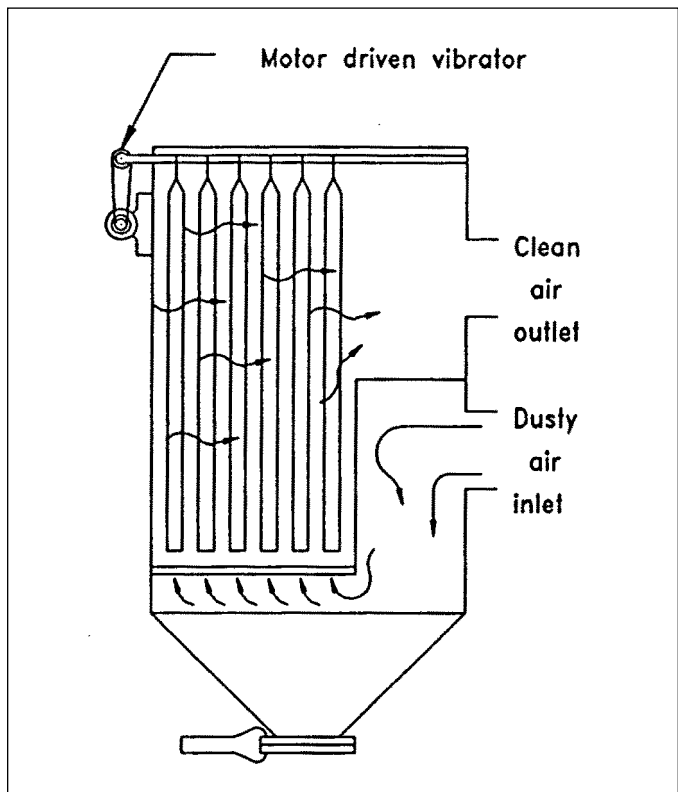

Figure 19–9. Typical baghouse air cleaner. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice,* 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)
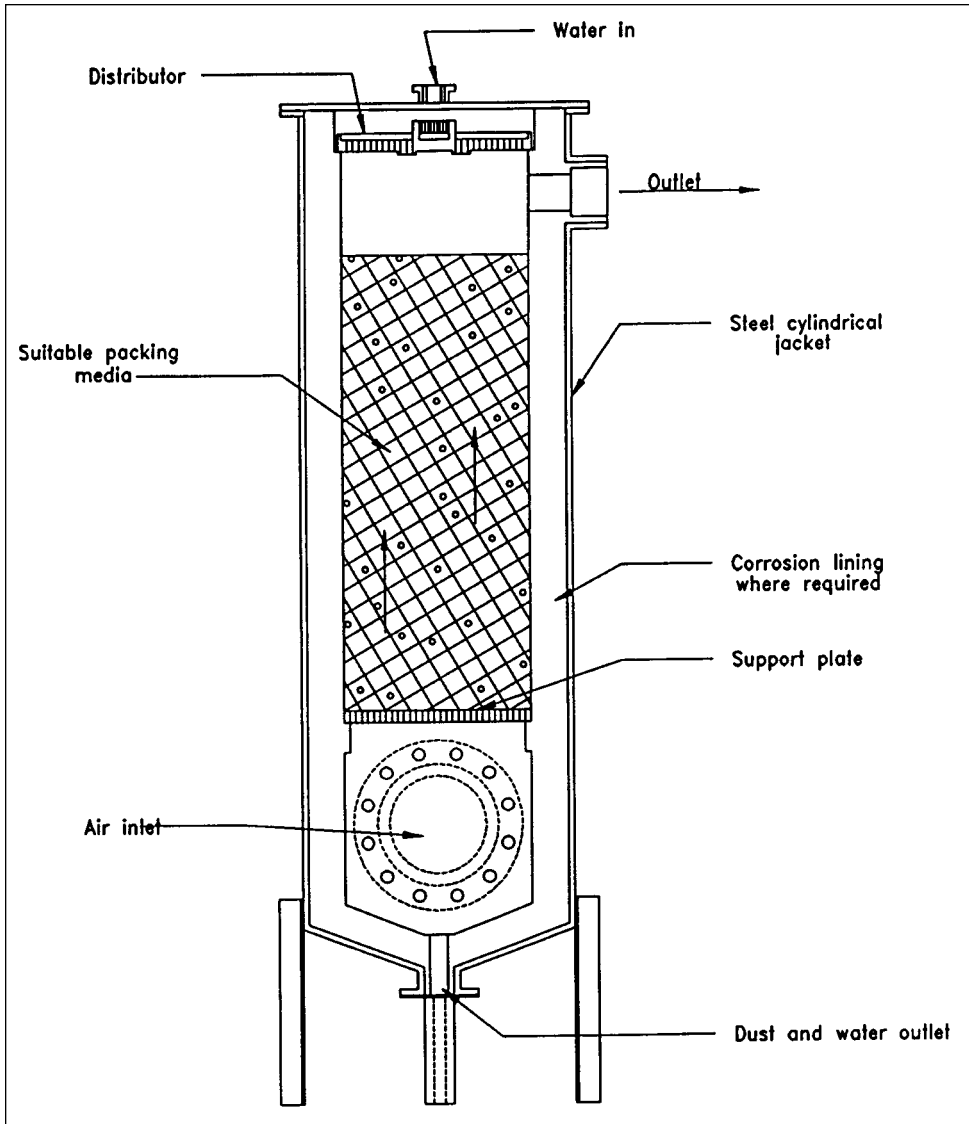
Figure 19–10. Typical packed bed air cleaner. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice,* 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

➢ *Absorption* is a diffusion process in which molecules are transferred from the exhaust gas to a liquid. The diffusion occurs because there is a gradient between the contaminant concentration in the exhaust gas and the liquid phase. This causes the contaminant to move from the higher level in the gas phase to the lower concentration in the liquid. The laws of mass transfer govern absorption. Mass transfer occurs at the interface between the gas or vapor molecule and the liquid and is enhanced by the following factors:

➢ high interfacial area between the exhaust gas and the liquid

➢ turbulent contact between the two phases

➢ high solubility of the gas or vapor in the liquid phase

➢ higher temperature, which affects solubility

For easily absorbed contaminants, a spray chamber or another simple device may work. However, for materials with low solubility or where a chemical reaction occurs between the contaminant and liquid prior to absorption, a packed bed (Figure 19–10) is often used to maximize contact. Reactive scrubbing is a special case of gas/vapor scrubbing. In reactive scrubbing, the contaminant reacts with the liquid to form a compound that is retained in the liquid.

➢ *Adsorption* is the process in which a gas or vapor adheres to the surface of a porous solid material. It occurs when the contaminant condenses into very small liquid droplets at an ambient temperature higher than its boiling point. This principle is well-known to industrial hygienists through use of activated carbon sampling devices. Since no chemical reaction is involved, adsorption is reversible. The contaminant can be recovered, if warranted, from the adsorbent by heating, steam flushing, air stripping, vacuum treating, or any other method that vaporizes the condensed material. Removing the adsorbate regenerates the adsorbent for further use. In addition to activated carbon, popular adsorbents include silica gel and molecular sieves.

➢ *Oxidation or combustion* devices can be used when the air contaminants are combustible. They oxidize (burn) the

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contaminants under a variety of operating conditions. Many are designed for gases and vapors; often these do not work well when the airstream contains particulates. They are very useful for processes that release extremely odorous organic vapors and fumes. The major expense associated with combustion systems is the auxiliary fuel needed to heat incoming exhaust gas and assure complete combustion. Some devices use a catalyst that causes the contaminant to oxidize at a lower temperature than normal in order to save fuel. Since most LEV systems exhaust room air with very low levels of contaminants, combustion is often not cost-effective. Combustion devices find more application with process vents or similar sources where the contaminant concentration is relatively high.

Makeup Air

Makeup air is air that enters the workroom to replace air exhausted through the ventilation system. A room or plant with insufficient makeup air is said to be "air bound" or "air starved." A ventilation system will not work properly if there is not enough air in the room to exhaust. This means that if the ambient static pressure within the room becomes slightly negative, the fan may not work properly against this additional resistance.

Makeup air should be supplied through a planned system rather than through random infiltration. The system should have the following features:
➤ The supply rate should exceed the exhaust rate by about 10%. This slight positive pressure in the building helps to keep out drafts and dust. The exception is a situation where no dust or airborne chemicals should travel from the workroom to adjacent offices or other areas. Then a slight negative pressure inside the workroom is preferred.
➤ The air should flow from cleaner areas of the plant through areas where contaminants may be present and finally to the exhaust system. Flow should also be from normal temperature areas to high heat process areas. The makeup air supply system can be designed to provide some cooling in the summer in hot process areas.
➤ Makeup air should be introduced into the "occupied zone" of the plant, generally 8–10 ft from the floor. This gives the workers the benefit of breathing fresh air and, if the air is tempered (heated or cooled) maximizes the comfort provided by the makeup air.
➤ The air should be heated in winter to a temperature of about 65 F.
➤ Makeup air inlets outside the building must be located so that no contaminated air from nearby exhaust stacks or chimneys is drawn into the makeup air system.

AIRFLOW AND PRESSURE PRINCIPLES

This section describes the principles that govern LEV system operation. The information is useful in understanding how the components described above function, and how to test and upgrade an installed system. The principles fall into two broad categories: *airflow* and *pressure*.

Airflow Principles

The basic airflow concept in ventilation systems is called the "equation of continuity," which expresses mass balance as air flows through different parts of the system. The equation of continuity is expressed as follows:

$$Q = V \times A \qquad (1)$$

where Q = airflow, ft^3/min
V = air velocity, ft/min
A = area of airflow, ft^2

At each point in a closed system (no additional air entering the duct):

$$Q = V_1 \times A_1 = V_2 \times A_2 \qquad (2)$$

where $_{1, 2}$ = two locations within the closed system

These equations have several important applications, including the following:
➤ The volumetric airflow (Q) through an opening (doorway, laboratory hood face) or in a duct can be readily determined using Equation (1) by measuring the average air velocity through the duct or opening and the area of the duct or opening.
➤ After Q is calculated or measured at one location in an LEV system, the velocity at other locations can be calculated from the Q and cross-sectional area at the new location, using Equation (2). This calculation is valid as long as no additional air enters the system through another hood. The calculation is useful to assure that adequate duct velocity is maintained at different parts of the system as duct diameter changes or to select the stack diameter to give a high discharge velocity for dispersion.

Another useful set of equations describes the relationship between volumetric airflow (Q) into a hood and the capture velocity generated out in front of the hood. These are shown in Figure 19–11 for different capture hood types. The equations can either be used to calculate the Q needed to generate the required capture velocity at *X* distance in front of the hood or to determine the velocity that will be generated by a given value of Q.

It important to note that these equations refer only to the centerline velocity, which is the air velocity along a line extending out from the center of the hood or duct, and do not describe the velocity distribution across the hood opening. Also, any distance *X* may be substituted into the equations to give an answer, but in practice a capturing hood can only reach out about two feet to draw in contaminants.
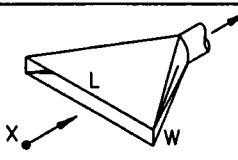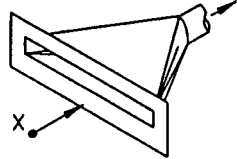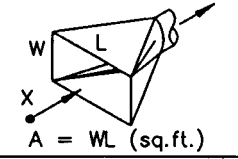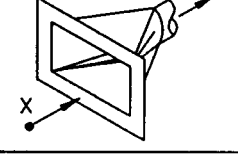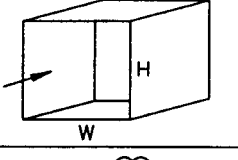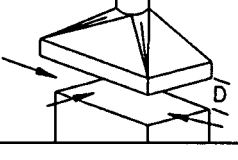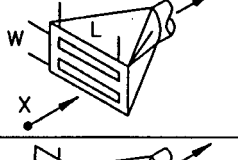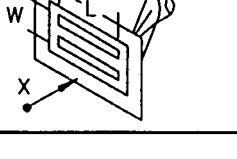
HOOD TYPE	DESCRIPTION	ASPECT RATIO,W/L	AIR FLOW
	SLOT	0.2 OR LESS	$Q = 3.7\ LVX$
	FLANGED SLOT	0.2 OR LESS	$Q = 2.6\ LVX$
A = WL (sq.ft.)	PLAIN OPENING	0.2 OR GREATER AND ROUND	$Q = V(10X^2+A)$
	FLANGED OPENING	0.2 OR GREATER AND ROUND	$Q = 0.75V(10X^2+A)$
	BOOTH	TO SUIT WORK	$Q = VA = VWH$
	CANOPY	TO SUIT WORK	$Q = 1.4\ PVD$ SEE VS– 99-03 P = PERIMETER D = HEIGHT ABOVE WORK
	PLAIN MULTIPLE SLOT OPENING 2 OR MORE SLOTS	0.2 OR GREATER	$Q = V(10X^2+A)$
	FLANGED MULTIPLE SLOT OPENING 2 OR MORE SLOTS	0.2 OR GREATER	$Q = 0.75V(10X^2+A)$

Figure 19–11. Equations for calculating air flow (Q) for common hood types. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice,* 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

For example, the centerline velocity outside a freely hanging plain hood (see Figure 19–11) is found by using the following equation:

$$Q = V_x\,(10X^2 + A) \qquad (3)$$

where V_x = air velocity at X distance, ft/min
Q = airflow into hood, ft³/min
A = area of hood face, ft²
X = distance outward from hood along hood axis (i.e., centerline), ft

Figure 19–12 (left diagram) shows the velocity contours in front of this hood type. The contours show that the velocity drops off significantly on either side of the centerline. One reason for this is that the hood draws air from behind the hood outside the contamination zone.

The plain hood can be easily improved by adding a flange or collar to reduce the air drawn from behind the hood as shown in Figure 19–12 (right diagram). This decreases the airflow requirement needed to develop the same V_x by about 25% for a flanged hood compared to a plain hood and changes the previous equation to the following:
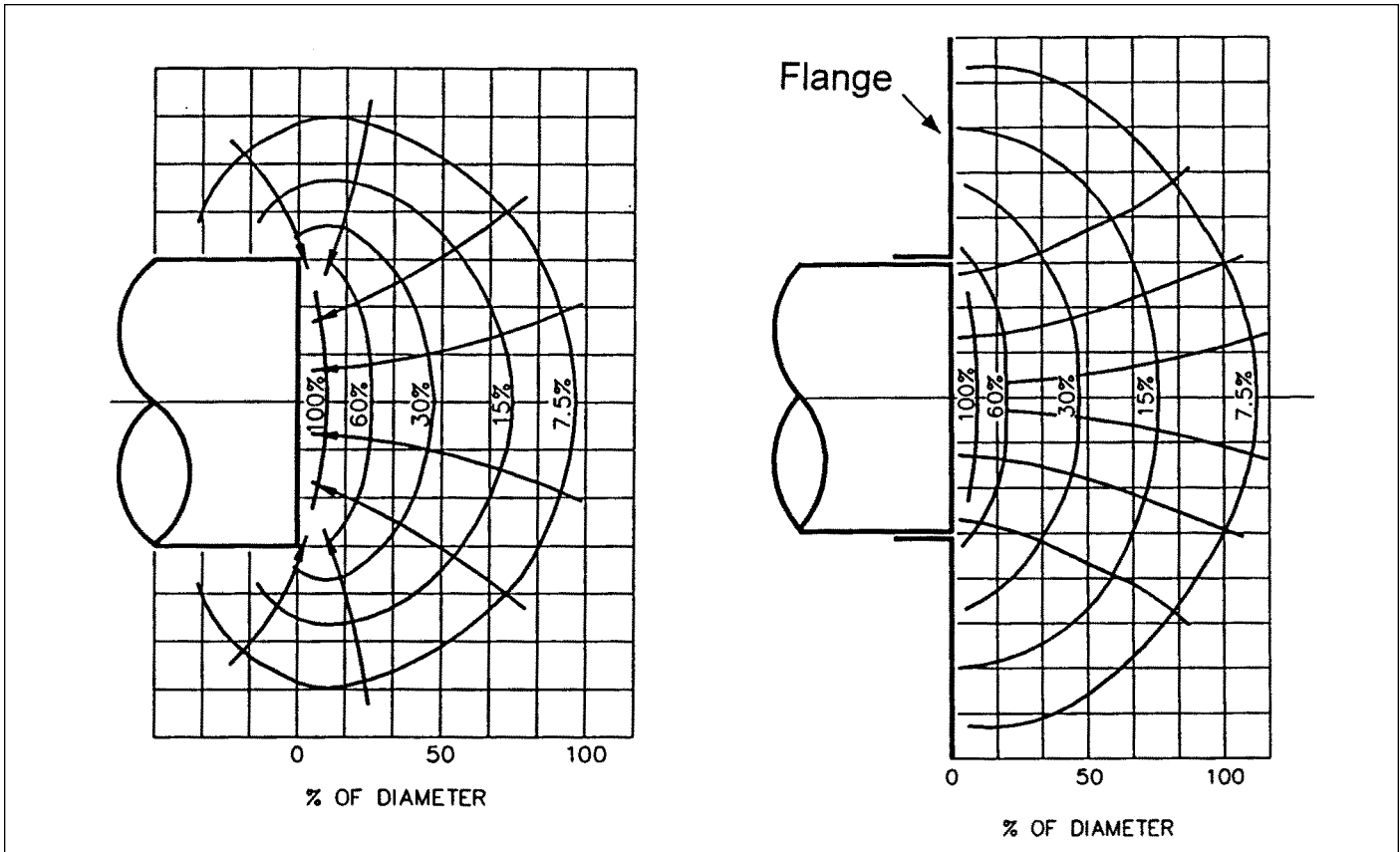
Figure 19–12. Velocity profile in front of a plain hood (left) and flanged hood (right). The flange reduces the quantity of air drawn from behind the hood so there is more inward air velocity along the centerline. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice*, 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

$$Q = 0.75\,V_x\,(10X^2 + A) \tag{4}$$

Example: A 4 in. x 8 in. (area = 32 in.^{2} or 0.22 ft^2) flanged suspended hood is drawing 500 ft^3/min of air. What is the velocity 6 in. (0.5 ft) in front of the hood?

Answer:

$$Q = 0.75\,V_x\,(10X^2 + A)$$

500 ft^3/min $= 0.75\,V_x\,[10(0.5^2) + 0.22] = 0.75\,V_x\,[10(0.25) + 0.22]$

500 ft^3/min $= 0.75\,V_x\,[2.50 + 0.22] = 0.75\,V_x\,[2.72]$

$$V_x = \frac{500 \text{ ft}^3/\text{min}}{(0.75)(2.72)} = 245 \text{ ft/min}$$

When considering airflow into hoods, there may be a tendency to assume that heavier-than-air vapors will tend to settle to the workroom floor and can be collected by a hood located there. In reality, for the small amounts of vapor in contaminated air (1,000 ppm means 1,000 parts of contaminants plus 999,000 parts of air), the resulting density of the mixture is so close to that of air that random air currents disperse the materials throughout the room. The exception to this rule is that a sizable leak of a dense gas or vapor (such as a compressed gas that chills by expansion as it escapes) will form a plume that moves along the ground. In this situation, a floor-level hood might be advantageous, but general room concentrations may still be significant.

Pressure Concepts

Air moves because there is a difference in pressure between two points. In an exhaust system, the fan develops negative pressure (suction) that extends back through the ducts to each hood. There the suction starts room air moving into the hoods and through the system. On the discharge side of the fan, positive pressure pushes the air through any remaining ducts and out the stack.

In ventilation work, pressure is expressed in units of "inches of water gauge." This represents the height of a water column (e.g., the weight of water) that the pressure will support. It is equivalent to more common pressure units, such as "pounds per square inch," but is more convenient to use since typical pressures in an LEV system are small. For example, an LEV system may have a maximum static pressure of 5 inches of water, which is equivalent to only 0.2 psi.

Pressure can be measured directly in inches of water using a U-tube manometer (Figure 19–13). The pressure value is
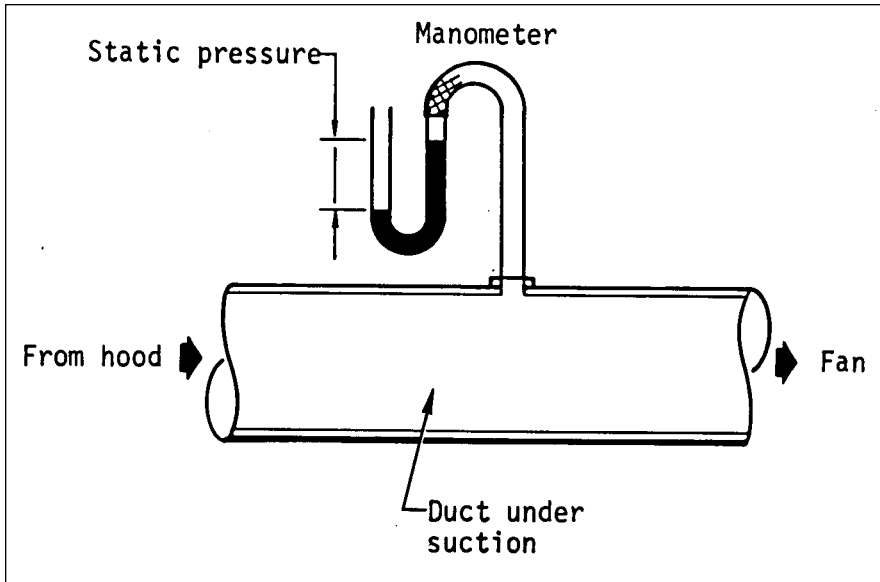
Figure 19–13. Static pressure in "inches of water" is measured with a water manometer. (Source: McDermott, 2000.)

the difference between the water level in each leg of the tube. Mechanical pressure gauges and electronic transducer units calibrated in inches of water are also commonly used.

TWO TYPES OF PRESSURE

A local exhaust system has two types of pressure, static pressure and velocity pressure.

➤ *Static pressure* (either negative or positive) pulls inward on the ducts before the fan and pushes outward on the ducts after the fan. Static pressure in a ventilation system acts to collapse the walls of the ducts on the suction side (inlet) of the fan and to burst the ducts on the discharge side. It acts equally at all locations in the duct (center as well as at the walls). The easiest way to measure it is by using a water manometer to read the bursting force on the duct walls (Figure 19–13), although the same reading will be obtained at any point across the duct at that location. Static pressure can be viewed as potential energy in the system that is available to start air moving and keep it moving by overcoming friction and turbulent losses.

➤ *Velocity pressure* is due to air moving through the system, which represents kinetic energy. Velocity pressure is exerted by air in motion and has a positive sign in the direction of airflow. Velocity pressure is determined by measuring the average velocity at the point, and using the following equation:

$$VP = \left(\frac{V}{4,005}\right)^2 \qquad (5)$$

or

$$V = 4,005 \sqrt{VP} \qquad (6)$$

where
VP = velocity pressure, inches of water
V = velocity, ft/min

The equations for velocity pressure assume standard air density, which is 0.075 lb/ft³· Density is affected by the moisture content and temperature of the air as well as altitude above sea level. Density corrections are needed to velocity pressure readings if any of the following three conditions occur: (1) moisture exceeds about 0.2 pounds of water per pound of air; (2) air temperature is outside the 40 F to 100 F range; or (3) altitude exceeds +1,000 ft relative to sea level. The density correction for temperature and atmospheric pressure can be calculated using the following equation:

$$\text{Density}_{\text{actual}} = \frac{0.075 \text{ lb}}{\text{ft}^3} \times \frac{530 \text{ F}}{(460+t) \text{ F}} \times \frac{\text{Bar. press}}{29.92} \qquad (7)$$

where
$\text{Density}_{\text{actual}}$ = actual air density, lb/ft³
t = temperature, °F
Bar. press. = Barometric pressure, inches of mercury

For density corrections due to elevated moisture, see the ACGIH *Ventilation Manual.*

With nonstandard air, the velocity pressure equation becomes:

$$VP = \left(\frac{V}{4,005}\right)^2 \left(\frac{\text{Density}_{\text{actual}}}{0.075 \text{ lb/ft}^3}\right) \qquad (8)$$

The sum of velocity pressure and static pressure at any point in the system equals the total pressure. The concept of total pressure is not very important in most LEV work, but illustrates that static pressure can be changed into velocity pressure and vice versa without an overall loss of pressure (or energy) from the system.

The concept of pressure governs most aspects of ventilation system operation. For example, room air has almost no velocity so its velocity pressure is zero. The LEV system draws that air into a hood and accelerates it up to the duct

velocity. At that point the air has a velocity pressure value corresponding to that duct velocity according to Equation (5). For this to occur, the system must give up the same amount of static pressure (i.e., the static pressure is converted into velocity pressure).

Example: What is the acceleration loss to accelerate room air up to 2,500 ft/min duct velocity?

Answer:

$$VP = \left(\frac{V}{4,005}\right)^2$$

$$VP = \left(\frac{2,500}{4,005}\right)^2 = (0.62)^2$$

$$VP = 0.38 \text{ inches of water}$$

This means that accelerating room air to 2,500 ft/min duct velocity will cause a loss of 0.38 inches of water suction from the system.

HOOD ENTRY LOSS

In addition to the acceleration loss that occurs at the hood, there is additional loss because turbulence occurs as the air enters the hood and duct. This turbulent loss is called *hood entry loss*, and is separate from acceleration loss.

The energy lost due to turbulence at the hood, expressed in units of "equivalent number of velocity pressures lost," is called the *hood entry loss coefficient (F)* and has been measured experimentally for many hood shapes (Figure 19–14). For a typical hood, the hood entry loss occurs as the air enters the duct at the hood, so the coefficient is referred to as Fd. For hoods with a long, narrow slot to distribute airflow, a loss also occurs at the slot; the coefficient for this loss is called Fs. Most slot hoods have a total hood entry loss made up of separate slot and duct entry components.

A hood shape that does not cause much turbulence has a lower entry loss coefficient than a hood with close clearances, sharp corners, or other features that produce a lot of turbulence. For example, as shown in Figure 19–14, a plain duct opening has a hood entry loss coefficient of about 93 percent of the duct velocity pressure while a smooth, bell-shaped entry reduces the turbulence so that the loss coefficient is just 4 percent of the duct velocity pressure. A narrow slot hood causes such severe turbulence that the slot entry loss coefficient is almost twice the slot velocity pressure.

To calculate the hood entry loss in units of inches of water, multiply the hood entry loss coefficient (F) by the duct or slot velocity pressure. For a hood without a slot, the equation is as follows:

$$h_e = (F_d)(VP_d) \tag{9}$$

For a slot hood, the total entry loss is the sum of the h_e value plus the additional entry loss at the slot, calculated by the following equation:

$$h_{e(slot\ hood)} = (F_d)(VP_d) + (F_s)(VP_s) \tag{10}$$

where
h_e = hood entry loss, inches of water
$h_{e(slot\ hood)}$ = total hood entry loss for slot hood, including both loss at slot and as air enters duct, inches of water
F_d = duct entry loss coefficient for hood
F_s = slot entry loss coefficient for slot hood
VP_d = duct velocity pressure, inches of water
VP_s = slot velocity pressure, inches of water

The hood design information in Figures 19–2 to 19–6 illustrates the hood entry loss associated with these hood configurations. Specific points include the following:
➢ Figure 19–2 (welding hood) shows the two advantages of using a flanged or cone versus a plain opening:
  ➢ With a flange or cone, the hood entry loss is $0.25VP_d$ compared to $0.93VP_d$ for a plain opening.
  ➢ The required airflow (Q) is lower with the flange or cone compared to the plain opening since less of the air is drawn from behind the hood.
➢ Figure 19–4 (metal polishing belt) also shows how steps to reduce turbulence will reduce hood entry loss. The term *take-off* refers to the connection between the hood and duct. With a tapered take-off (shown in the diagram), there is a transition section between the hood and duct to smooth out airflow patterns, resulting in a hood entry of $0.45VP_d$. With a straight take-off (not shown), there is no transition, resulting in more turbulence and a higher hood entry loss of $0.65VP_d$.
➢ Figure 19–5 (dip tank) shows the higher overall loss from a slotted hood. There is a loss of $1.78VP_s$ as the air moves through the slot into the plenum chamber. Then there is a loss of $0.25VP_d$ as the air enters the duct. The total loss is the sum of these two components. Note that the ACGIH *Manual* uses h_e for the total loss rather than $h_{e(slot\ hood)}$ as in Equation (10).

The hood entry loss coefficients for many different hood shapes can be found in the ACGIH *Ventilation Manual*.

HOOD STATIC PRESSURE

For a hood to operate properly, the fan must generate enough suction or static pressure in the duct near the hood to overcome both the acceleration loss and the hood entry loss while drawing the correct amount of air into the hood. This amount of suction is called the *hood static pressure* and is easily measured using a water manometer (Figure 19–15).

For a hood without a slot, hood static pressure is calculated using the following equation:
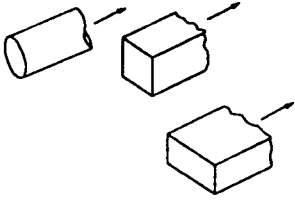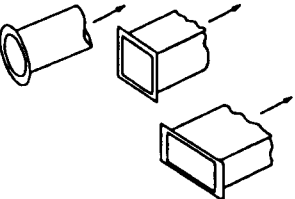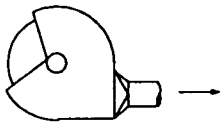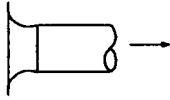
HOOD TYPE	DESCRIPTION	HOOD ENTRY LOSS (F_d)
	PLAIN OPENING	0.93
	FLANGED OPENING	0.49
	TYPICAL GRINDING HOOD	(STRAIGHT TAKEOFF) 0.65 (TAPERED TAKEOFF) 0.40
	BELL MOUTH INLET	0.04

Figure 19–14. Hood Entry Loss coefficients (F_d) for common hood types. For the grinding hood a "tapered take-off" means that there is a transition section between the hood and duct to smooth airflow, while a straight take-off (not shown) has no transition piece. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice*, 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)

SP_h = Acceleration Loss + Hood Entry Loss
   = (1.0 x duct velocity pressure) + (F_d x duct velocity pressure)
   = $1.0\ VP_d + F_d\ VP_d$

$$SP_h = (1.0 + F_d)\ VP_d \qquad (11)$$

where SP_h = hood static pressure (for a hood without a slot), inches of water
   F_d = entry loss coefficient for hood
   VP_d = duct velocity pressure, inches of water

For a slot hood, the loss caused by the slot must be added in:

$$SP_{h(slot)} = (1.0 + F_d)\ VP_d + F_s\ VP_s \qquad (12)$$

where $SP_{h(slot)}$ = hood static pressure for slot hood, inches of water
   F_s = slot entry loss coefficient
   VP_s = slot velocity pressure, inches of water

Hood static pressure is important for the following two reasons:

➤ During system design, the SP_h can be calculated and represents the suction that is needed at the hood in order for the hood to function properly. Then the fan can be selected to move the required Q while generating sufficient static pressure so the required SP_h will be available at each hood. As a corollary, in an existing system if the fan cannot generate the required SP_h at a hood, then the hood will never function properly.

➤ Once a ventilation system is installed and operating properly, the hood static pressure can be measured and recorded. Periodic readings can be compared to the original value to determine if the suction available at the hood is still adequate to draw the required amount of air for proper hood operation.

Example: The barrel-filling hood in Figure 19–1 has a hood entry loss coefficient (F_d) of 0.25. Measurements show that the hood static pressure is 1.8 inches of water in the 4-inch diameter circular duct (area = 0.087 ft²) at the hood. Estimate the airflow into this hood.
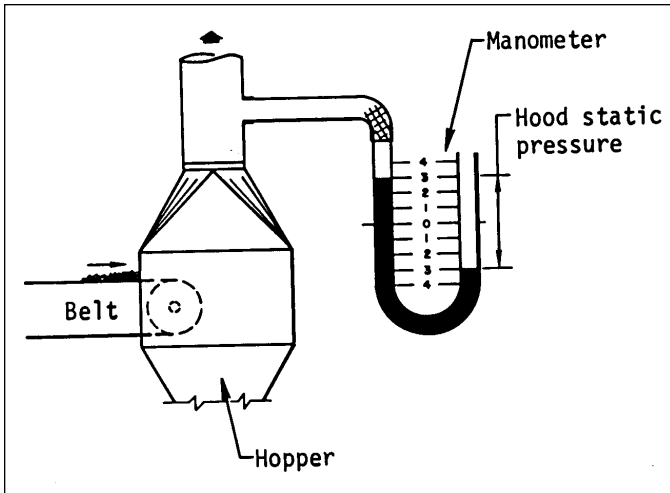
Figure 19–15. Hood Static Pressure, measured in the duct near the hood, represents the suction or potential energy available to draw air into the hood. (Source: McDermott, 2000).

Answer: A summary of the data is as follows:

$F_d = 0.25$
$SP_h = 1.8$ in. of water
Duct Diameter = 4 in. (area = 0.087 ft^2)

From Equation (11):
$SP_h = (1.0 + F_d) VP_d$
1.8 in. $H_2O = (1.0 + 0.25) VP_d$

Solving for VP_d:
$VP_d = 1.44$ in. of water

From Equation (9):
$V = 4,005 \sqrt{VP}$

$V = 4,005 \sqrt{1.44} = 4,806$ ft/min

From Equation (1):
$Q = V \times A$
$Q = 4,806$ ft/min $\times 0.087$ ft^2
$Q = 418$ ft^3/min

PRESSURE LOSS IN THE DUCTS

Air flowing through the ductwork meets resistance in the form of friction and turbulence. Straight duct lengths result in friction loss, while elbows, junctions, air cleaners, and other features cause turbulence losses. These losses can be expressed as pressure drop since they represent pressure lost from the system that the fan must generate to make the system work properly.

Because of these losses, the following static pressure profile exists in an LEV system:

➤ Before the fan, the greatest suction value occurs at the fan inlet. Moving from the fan toward the hoods, the suction decreases because it is used to overcome friction and turbulence losses until, at the hood, just enough static pressure remains to overcome the acceleration and hood entry loss.
➤ After the fan, the greatest value of positive static pressure is at the fan outlet. The positive static pressure is used to overcome friction loss in any ducts and the stack as well as turbulence losses in any elbows. At the stack discharge, any remaining static pressure is used to discharge the air at a higher velocity to aid contaminant dispersion.

FAN PRESSURE

As described throughout this chapter, the fan generates pressure that causes the air to move through the system. In order to characterize fan performance, both the volumetric airflow (Q) and the pressure that the fan generates must be specified.

For LEV systems, this fan pressure is called *fan static pressure* (FSP). It is calculated from an equation used by fan manufacturers as part of a standard test of fan performance:

$$FSP = |SP_{inlet}| + |SP_{outlet}| - VP_{inlet} \qquad (13)$$

where   FSP = fan static pressure, inches of water
SP = static pressure, inches of water
VP = velocity pressure, inches of water
inlet, outlet = fan inlet and outlet

FSP represents the net pressure that the fan adds to the system. The following points should be noted:
➤ The SP_{inlet} value is the total suction that the fan must generate to first accelerate room air up to duct velocity and then overcome all pressure losses before the fan.
➤ The SP_{outlet} value is the positive pressure required at the fan outlet to overcome the friction and other losses after the fan and finally discharge the air from the stack.
➤ The VP_{inlet} value is subtracted because it represents the energy in the moving air reaching the fan that was included in the SP_{inlet} value as the acceleration loss at the hood.

FAN LAWS DESCRIBE FAN PERFORMANCE

The following three equations, called *fan laws*, describe the relationship of volumetric airflow, fan static pressure, and brake horsepower to rotating speed for a specific fan:
➤ Changes in volumetric airflow (ft^3/min) vary directly with changes in fan speed. For example, for a given fan doubling the speed will double the volume output.

$$\frac{Q_1}{Q_2} = \frac{R_1}{R_2} \qquad (14)$$

where   R = fan rotating speed, rev/min
Q = airflow, ft^3/min

➤ Changes in static pressure vary directly with the square of changes in fan speed.

$$\frac{FSP_1}{FSP_2} = \left(\frac{R_1}{R_2}\right)^2 \qquad (15)$$

where   FSP = fan static pressure, inches of water

➤ Changes in brake horsepower vary directly with the cube of changes in fan speed. *Brake horsepower* is the energy required to operate the fan, but does not include any drive loss between the fan and motor.

$$\frac{BHP_1}{BHP_2} = \left(\frac{R_1}{R_2}\right)^3 \qquad (16)$$

where   BHP = brake horsepower

Since all three of these fan laws act together, any change in fan speed to increase volume output also increases fan static pressure and brake horsepower.

This can be important since a common method of increasing Q in an existing system is to increase the fan rotating speed. This will increase volumetric airflow, but the brake horsepower, which represents electrical power consumption, increases as the third power of increases in fan rotating speed. For example, doubling the fan speed requires eight times more electrical power to run the fan (i.e., $2^3 = 8$). This could make increasing the fan speed a poor economic decision compared to replacing the existing fan with a larger model better suited for the application.

Fan manufacturers specify a maximum safe rotating speed for each fan to prevent mechanical failure. This speed cannot be exceeded when attempting to increase airflow in an LEV system.

Example: A fan is moving 4,000 ft³/min of air. A tachometer reading shows the rotating speed to be 1,650 rev/min. Measurement of electrical consumption shows that the consumption is equivalent to 2.23 brake horsepower. Calculate the effect on rotating speed and horsepower of increasing the airflow to 5,000 ft³/min.

Answer: Since rotating speed and volume output are proportional, use Equation (14) to find the new rotating speed:

$$\frac{Q_1}{Q_2} = \frac{R_1}{R_2}$$

where   R = fan rotating speed, rev/min
           Q = airflow, ft³/min

$$\frac{4,000}{5,000} = \frac{1,650}{R_2}$$
$$R_2 = 2,063 \text{ rev/min}$$

Calculate the new brake horsepower using Equation (16):

$$\frac{BHP_1}{BHP_2} = \left(\frac{R_1}{R_2}\right)^3$$

where BHP = brake horsepower

$$\frac{2.23}{BHP_2} = \left(\frac{1,650}{2,063}\right)^3$$

$$\frac{2.23}{BHP_2} = (0.80)^3 = 0.51$$

$$BHP_2 = \frac{2.23}{0.51}$$

$$BHP_2 = 4.4 \text{ horsepower}$$

Whether or not the best solution lies in increasing the fan speed to achieve the desired performance is usually based on an economic evaluation. The cost of the added electrical power can be calculated over the projected life of the fan. For small increases in fan speed, the existing fan motor may be adequate but the fan manufacturer's literature will show whether a higher horsepower motor is needed. Also check the fan specifications to see whether the existing fan is safe at the required rotating speed.

LEV PERFORMANCE EVALUATION AND IMPROVEMENT

This section describes typical tests used to evaluate LEV system performance and, where applicable, simple steps to diagnose and resolve problems.

Smoke Tube Tests

Smoke tubes are glass tubes containing a chemical that produces a chemical fume (smoke) as room air is blown through the tube with a hand-operated bulb. They are useful for the following tests:
➤ evaluating the capture range of hoods
➤ identifying drafts and other factors that can interfere with hood performance
➤ demonstrating the capture distance of hoods to workers so they can position the hood or work item properly

Velocity Measurements

Several devices are available when a quantitative measurement of velocity is required. The main categories include:
➤ *Swinging vane velometer* contains a vane or paddle that moves according to the velocity of the air passing through the instrument. The paddle is connected mechanically to a meter that displays the velocity.
➤ *Thermo-anemometer* works on the principle that the resistance of a heated wire changes with temperature variations. Air moving over the heated wire element changes its tem-

perature depending on the air velocity. The anemometer is calibrated directly in feet per minute. Very high moisture levels may cause inaccurate readings if the moisture affects the change of resistance in the heated element.

➣ *Rotating vane anemometer* has a propeller-like velocity sensor connected either to a mechanical or electronic readout unit. This type of device comes in a variety of sizes; the smaller ones have a thin probe while larger units have a propeller that is several inches in diameter.

➣ *Pitot-tube devices* that determine the velocity pressure inside a duct and are connected to a liquid manometer or pressure sensor that displays output in either inches of water velocity pressure or directly in velocity. (A pitot tube is a special probe-like device that accurately measures static and total pressures inside a duct. The difference between total and static pressures equals velocity pressure.)

Typical applications for the first three devices are measuring capture velocity outside of capturing hoods, face velocity for enclosures, and slot velocities (Figure 19–16).

DUCT VELOCITY TRAVERSE

A traverse involves measuring the velocity at a number of points across the duct area since velocity distribution is not uniform within the duct. Typically, a pitot tube is used for the traverse, but any velometer with a narrow probe sensor may also be used. The measurement points are selected to divide the duct into enough zones of equal area according to the following guidelines:

➣ For round ducts two traverses at right angles should be made. For ducts 6 in. in diameter or smaller, make two 6-point traverses. For ducts 6 to 48 in., make at least two 10-point traverses. Above 48 in. or for smaller ducts where large velocity variations are suspected, make two 20-point traverses. The locations of the measuring points are selected to divide the duct into equal annular areas (Figure 19–17); they are *not* equidistant points along the duct diameter.

➣ For rectangular ducts, the cross section is divided into equal areas and a reading is taken at the center of each area. For accurate duct measurements, at least 16 readings should be taken, but the distance between measuring points should not exceed 6 in. Similarly, for large openings, such as laboratory hoods or paint spray booths, a series of velocity readings should be taken and averaged to estimate the average air velocity into the enclosure.

Regardless of duct shape, the best place to perform a traverse is at least 7.5 duct diameters downstream from any
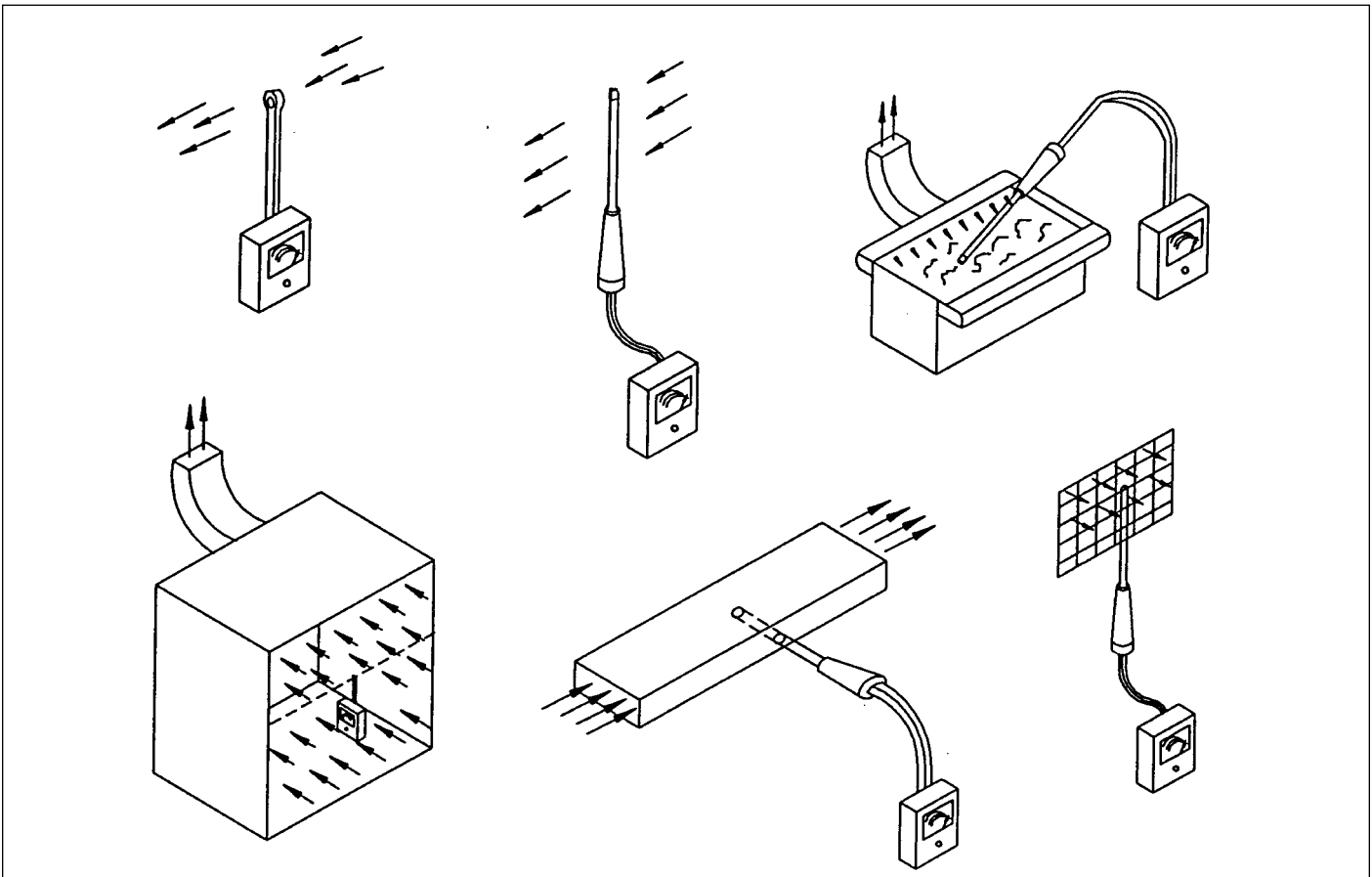

Figure 19–16. Typical applications for a velometer. (Source: From American Conference of Governmental Industrial Hygienists (ACGIH®). *Industrial Ventilation: A Manual of Recommended Practice,* 23rd ed. Copyright 1998, Cincinnati. Reprinted with permission.)
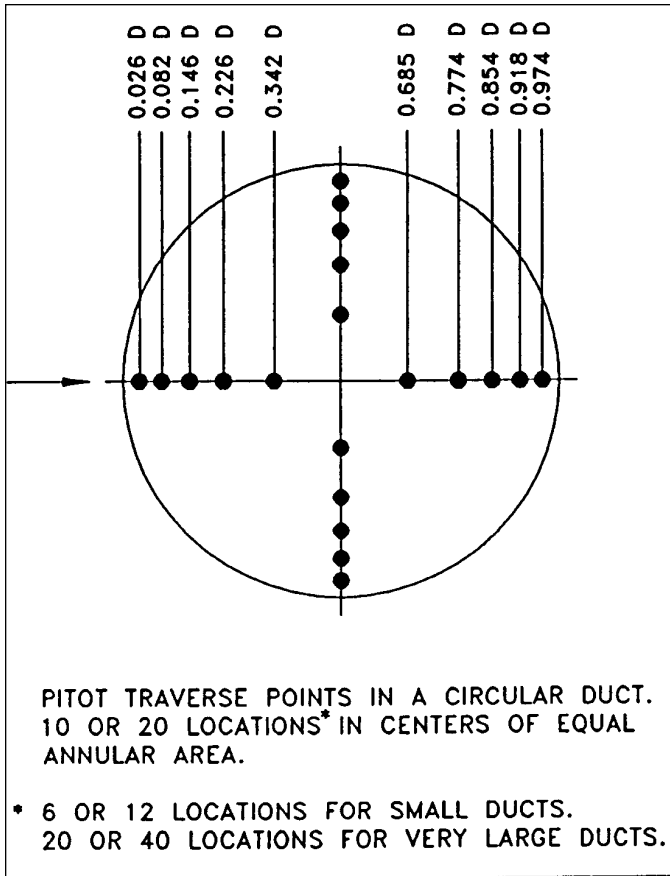
0.026 D
0.082 D
0.146 D
0.226 D
0.342 D
0.685 D
0.774 D
0.854 D
0.918 D
0.974 D

PITOT TRAVERSE POINTS IN A CIRCULAR DUCT.
10 OR 20 LOCATIONS* IN CENTERS OF EQUAL
ANNULAR AREA.

* 6 OR 12 LOCATIONS FOR SMALL DUCTS.
  20 OR 40 LOCATIONS FOR VERY LARGE DUCTS.

Figure 19–17. Measuring locations for a 10-point velocity traverse in a round duct. These points divide the duct into equal area annular rings.

Static Pressure Measurements

Periodic static pressure tests at the hoods and other locations in the LEV system are an excellent way to evaluate performance and diagnose problems. Since static pressure represents the potential energy available to move air in the system, changes in static pressure over time can indicate situations that degrade system performance. The greatest value of static pressure tests is from comparing current results to earlier readings. The LEV system should be balanced and hood velocities measured to assure adequate performance before instituting a static pressure test program.

To measure static pressure in a duct, a small diameter hole (1/16 to 1/8 in.) is drilled through the duct wall. It is important that no burrs around the drilled hole protrude into the flowing airstream; these can be removed by inserting a thin rod with a 90–degree bend into the hole and rotating it to smooth the edges. The holes should be at least 7.5 duct diameters downstream from any disturbance, such as an

elbow, damper, or branch duct entry. If this is not possible, then four holes should be drilled 90 degrees apart around the duct and the measured static pressure values averaged. The end of a rubber tube attached to a U-tube manometer or other pressure sensor is pressed against the duct over the hole; the static pressure is read as inches of water (Figure 19–13). While a short length of metal tubing can be brazed onto the duct around the hole for the static pressure tube connection, simply holding the end of the rubber tubing against the duct will give an accurate reading. The hole does not have to be capped between tests if it is small enough. Permanent installation of a manometer or other pressure gauge is used for LEV systems where continuous measurements are needed,

HOOD STATIC PRESSURE

The principle of measuring hood static pressure is illustrated in Figure 19–15. The ideal location is about one duct diameter from the throat of a hood with a tapered transition into the duct, and about three duct diameters for plain openings or flanged hoods. This location will avoid measuring hood static pressure within the turbulent zone in the duct near the hood.

In setting up a hood static pressure test program, determine the airflow into hoods using velometer readings or a velocity traverse across the ducts. Adjust airflow to each hood until it meets design criteria, and then measure and record hood static pressure along with volumetric airflow (Q). If subsequent hood static pressure readings show a decrease at a hood, the change in flow rate can be calculated using the following equation:

$$\frac{Q}{Q_o} = \frac{\sqrt{SP_h}}{\sqrt{SP_{h(o)}}} \tag{17}$$

where Q = current volumetric airflow rate, ft^3/min
Q_o = original volumetric airflow rate, ft^3/min
SP_h = current hood static pressure, inches of water
$SP_{h(o)}$ = original hood static pressure, inches of water

This technique is valid for all hood designs, including slot hoods.

Example: The hood static pressure at the conveyor belt discharge hood (i.e., the upper hood) in Figure 19–1 was 1.15 in. of water with airflow of 1,050 ft^3/min. Recent tests show the hood static pressure is now 0.87 in. of water. What is the current airflow into the hood?

Answer:

$$\frac{Q}{Q_o} = \frac{\sqrt{SP_h}}{\sqrt{SP_{h(o)}}}$$

$$Q = Q_o \frac{\sqrt{SP_h}}{\sqrt{SP_{h(o)}}}$$

major disturbance to smooth airflow, such as dampers or elbows. More specific guidelines for evaluating the validity of velocity traverse results are contained in the ACGIH *Ventilation Manual.*

$$Q = 1050 \text{ ft}^3/\text{min} \, \frac{\sqrt{0.87}}{\sqrt{1.15}} = 1050 \text{ ft}^3/\text{min} \left(\frac{0.93}{1.07}\right) = 914 \text{ ft}^3/\text{min}$$

The decline in airflow could be caused by a loose fan belt, plugged duct, or other problem.

OTHER STATIC PRESSURE TEST LOCATIONS

Static pressure tests at various other system locations are also valuable in measuring system performance and diagnosing malfunctions. Typical locations (in addition to hoods) to measure static pressure are at entries into main ducts, on each side of air cleaners, on each side of the fan, and at several points along long ducts. Initial and periodic static pressure readings should be recorded on a data sheet. Usually differences between readings at the same location that exceed 10 percent should be investigated to determine the reason for the change.

Here is an example of how static pressure tests can help diagnose ventilation system problems. A ventilation system (Figure 19–18) consists of six hoods and branch ducts, a main duct, an air cleaner, and a fan. Over several years the following problems were identified:

➤ The hood static pressures are low in all hoods (Figure 19–18a). Further tests show that static pressure on the hood side of the air cleaner is low, while it is higher than usual between the air cleaner and fan. These readings indicate that the air cleaner is causing too much resistance and should be cleaned to restore proper performance.

➤ The static pressure readings on both sides of the air cleaner are higher than usual (Figure 19–18b) while the hood static pressure readings for all hoods are low. This indicates that the main duct is partially plugged just before the air cleaner.

➤ The hood static pressure readings are low (Figure 19–18c) as is the static pressure on both sides of the air cleaner. This indicates that the fan is not working properly, the discharge stack is plugged, or there is a loose duct joint between the air cleaner and fan.

➤ All static pressure readings are normal except the value at one hood, which is too low (Figure 19–18d). This indicates that the branch duct is plugged between the test point and the main duct.

➤ All static pressure readings are near normal except one

(a) Aircleaner is clogged

(b) Main duct plugged

(c) Reduced fan performance, discharge stacked plugged or loose duct joint

(d) Branch plugged at X

(e) Branch plugged at X
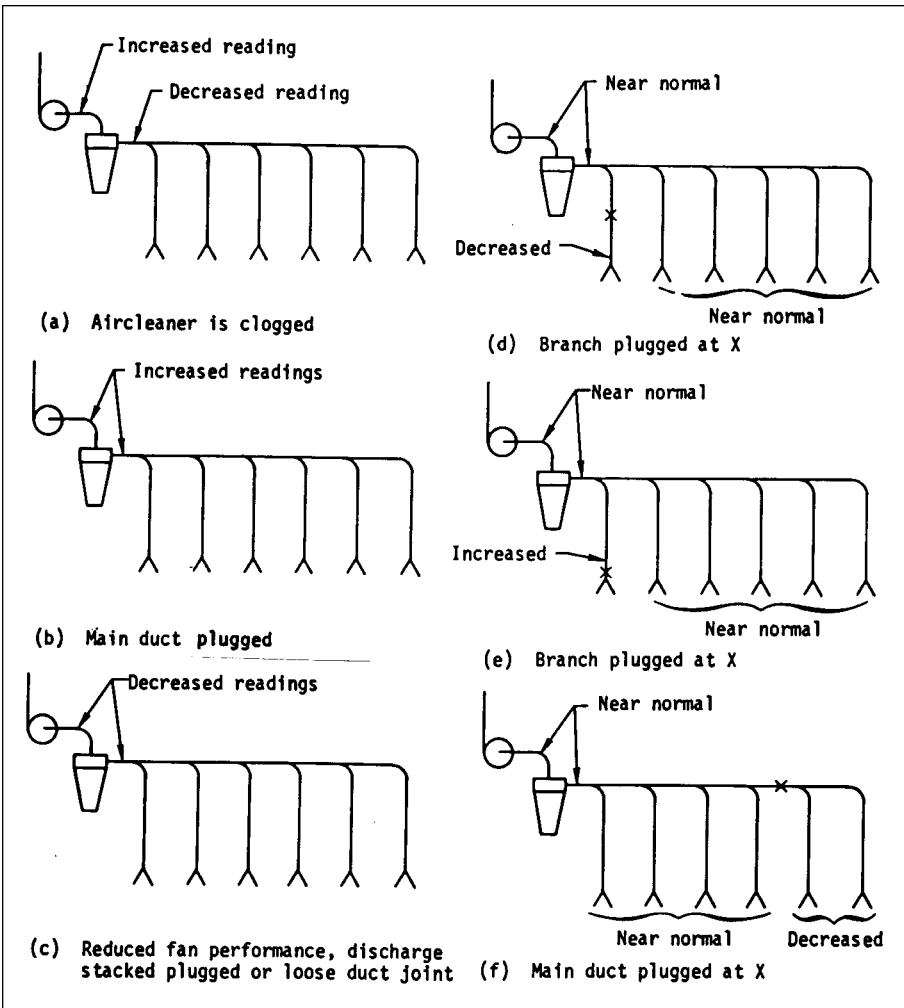
(f) Main duct plugged at X

Figure 19–18. LEV system problems diagnosed by comparing static pressure reading with earlier tests. (Source: McDermott, 2000.)

hood static pressure reading, which is increased (Figure 19–18e). This means that a blockage exists between the hood opening and the hood static pressure test point.

➤ All static pressure readings are near normal except the hood static pressure readings in two adjacent hoods, which are decreased (Figure 19–18f). This indicates that the main duct is plugged near the two hoods with lower hood static pressure values.

All of these difficulties were quickly solved once the source of trouble was recognized. Note that where a blockage caused a higher static pressure reading on the fan side of the blockage, this occurred because the blockage prevented the suction (static pressure) from being converted into velocity pressure.

Static pressure tests in a system that once operated properly can help to identify the following fan problems:

➤ loose fan belt or another drive problem
➤ material deposited on blades (especially with forward-inclined blade fan) or blade erosion/corrosion
➤ centrifugal fan motor wired incorrectly and therefore rotating backward. (Larger fan motors are wired in a three-phase configuration, and if the electrical supply wires are connected to the wrong terminals, the motor rotates in the wrong direction. The result is that the fan will be excessively noisy and will move less air than expected. However, an axial flow fan rotating backward will move air in the wrong direction.)

SUMMARY

Local exhaust ventilation (LEV) systems are an important technique for controlling employee exposures to airborne contaminants. A sound understanding of system components and the airflow and pressure principles that govern system operation will help the safety and industrial hygiene practitioner apply LEV properly.

A Bibliography for this chapter appears at the end of Chapter 20, Dilution Ventilation for Industrial Workplaces.