Ventilation and Exhaust Systems

Course No: M04-009
Credit: 4 PDH

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Contents

1 Introduction

Outdoor air that flows through a building, either unintentionally as infiltration or intentionally as ventilation, is important for two reasons:

1. Outdoor air is usually used to dilute indoor air contaminants.
2. Energy associated with heating or cooling outdoor air is a significant space-conditioning load.

Buildings use this air movement in three different modes for air exchange:

1. Forced-air ventilation
2. Natural ventilation
3. Infiltration.

These modes affect air quality, energy, and thermal comfort very differently. They also have different capabilities in maintaining a desired air exchange rate. All three modes should be included in the air exchange rate in a building at any given time.

Infiltration is the uncontrolled flow of air through unintentional openings in a building's envelope or shell. It can be driven by pressure differences across the shell (i.e., by appliance-induced pressures, temperature differences, and wind). It is an important factor in mechanically ventilated buildings.

Natural ventilation is caused by pressures from indoor-outdoor temperature differences and pressures from wind. Air flow through open windows and doors can be used to provide adequate ventilation for contaminant dilution and temperature control in some cases. In other cases, unintentional openings in the building envelope can interfere with desired natural ventilation air distribution patterns and lead to larger-than-design air flow rates.

Forced-air ventilation depends on the air flow rates through the system fans, the air flow resistance associated with the air distribution system, the air flow resistance between the zones of the building, and the air-tightness of the building envelope. Forced-air ventilation has the greatest control of air exchange rate and air distribution within a building. It is generally mandatory in larger buildings, where a minimum amount of outdoor air is required for occupant health and comfort and where a mechanical exhaust system is necessary.

2 Infiltration

Infiltration is the uncontrolled flow of air through openings in a building's envelope driven by pressure differences across the shell.

Air flow through the building shell occurs because pressure differences act on openings in the shell. Understanding infiltration requires understanding the pressures that cause the flow and the flow characteristics of the openings in the building shell.

Building pressure is determined by how much air is being introduced compared to how much is being exhausted. If more air is introduced than is exhausted, the difference should pressurize the building and leak out through cracks. Therefore, the size of the cracks, or "porosity" of the structure, is an important factor in building pressurization.

Factors Determining Building Pressure

Porosity and several other factors can affect building pressure. Some of these factors are controllable while others are not.

Building Porosity

Building porosity is composed of many variables:

- Leakage through doors
- Leakage through windows (movable and stationary)
- Leakage through elevator shafts
- Leakage through walls
- Building age—porosity will change as a building settles

- Leakage through fireplace dampers
- Leakage across the top ceiling of the heated space.

Stack Effect

Stack effect occurs when the temperature inside a building is not equal to the outdoor air temperature. Flow within the building results from the pressure differences that occur due to the differences in the air density.

The stack effect is most noticeable in multistory buildings when outdoor air temperatures are considerably less than indoor air temperatures. This results in pressure differences of some magnitude between upper and lower floors. Upper floors are of a positive pressure relative to the atmosphere while the lower floors are negative. The result is an upward air flow, generally through the elevator shafts and stair wells.

The reverse will occur during the summer when the indoor temperature is less than the outdoor temperature, but the effect will be reduced if the temperature and corresponding pressure differential between indoors and outdoors is low. During the cooling season, the temperature difference generally is not greater than 30 °F compared to a possible 80 °F temperature difference during the winter. Therefore, the infiltration of air in summer is at the upper floors and the exfiltration of air at lower floors. Resulting air flow is down through the building and is minimal.

Control in the past has been provided through isolation of elevator shafts (the most common carrier of air), building entrances, and the pressurization of first floor lobbies.

Wind Velocity and Direction

Wind velocity and direction tend to be uncontrollable factors. Air flow due to wind around or over a building will create areas in which static pressure will be different than the pressure of the undisturbed air flow (Figure D-1).

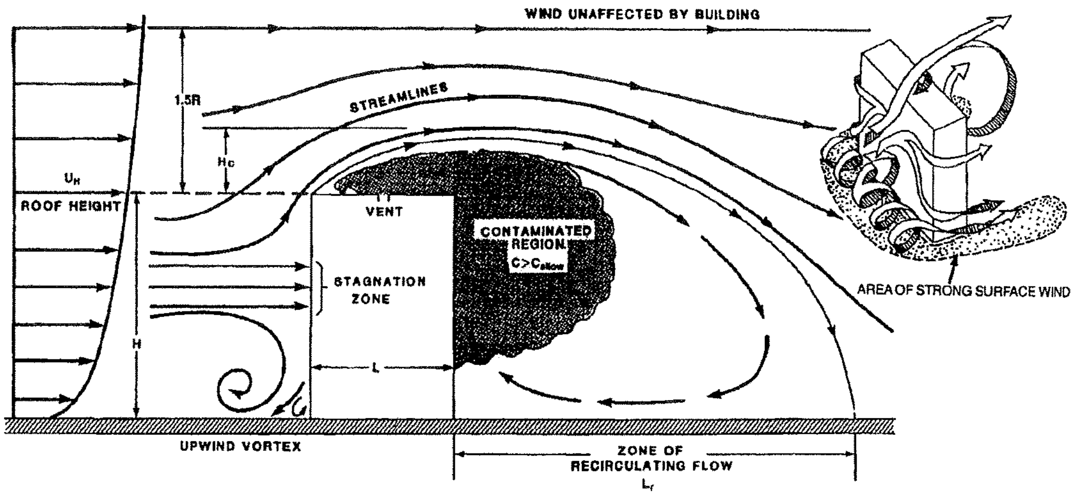

Figure D-1. Wind Velocity and Direction.

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Pressures on the windward side of the building will be positive; negative on the leeward side. On the remaining sides, static pressures will be positive or negative to lesser degrees depending upon the direction of air flow.

The terrain surrounding the building can also create wind flow changes affecting building pressures. Surface roughness of the surrounding terrain (the size and location of surrounding buildings) influences the relationship of wind velocity to building height, which will affect the pressure patterns around the building's exterior, including the roof.

Moisture and Infiltration

Buildings, like our bodies, exchange moisture and air with the environment, as well as exchanging heat. Although most of this moisture exchange occurs during the exchange of fresh air, some exchange occurs through a building's skin. This can cause problems in either hot, humid climates or very cold ones.

In hot, humid conditions, cool inside surfaces are often encountered—for example, the ceiling directly below a roof pond used for passive cooling. As hot and humid air contacts such a surface, condensation can occur. The moisture vapor in the air condenses to form visible droplets of water on the ceiling. The result can be mildly annoying water drips on the head, or serious water stains, eventually with mold growing on surfaces.

In cold climates, cold interior surfaces also occur, especially at windows. Although the air indoors may not be particularly humid (40 to 50 percent RH is common), it contains enough moisture to permit condensation on cold surfaces. Again, mild annoyance or more serious damage can result. A much less visible moisture threat occurs within walls, ceilings, or floors. Almost all common building materials, including gypsum board, concrete, clay masonry, and wood, are easily permeated by moisture. Most surface finishes are also permeable. In cold climates, the air outside contains relatively little moisture, even though the RH may be high. By contrast, inside air contains much more moisture per unit of volume, despite its probably lower RH. The result is a flow of vapor from high vapor pressure to low vapor pressure (typically warm to cold).

Such a flow occurs when the temperature within the wall (floor, etc.) drops low enough for this vapor to condense. Insulation can then become wet and thereby less effective, since water conducts heat far better than the air pockets it has filled. If wet insulation compacts, these air pockets are permanently lost. Worse yet, moisture damage can occur, such as dry rot in wood structural members. The usual remedy for such a potential problem is to install a vapor barrier within the building envelope. These barriers are commonly made of plastic film installed with as few holes as possible.

A substantial benefit of plastic films is that they reduce air flow through construction. Outdoor air is always infiltrating a building, gradually replacing the indoor air. This unintentional source of fresh air becomes a problem when temperatures outside are very different from those inside, especially when strong winds force outdoor air indoors fast enough to produce noticeably cold (or hot) drafts. Some fresh air is always desirable in buildings, but so is user control of how and where it is admitted. Therefore, the moisture-tight and infiltration-tight characteristics of plastic film vapor barriers are usually beneficial. When good vapor barriers are installed, the smaller air-change values that accompany "tight" construction may be assumed in calculations of heat flow due to infiltration and ventilation.

Methods of Calculating Infiltration

Air Change Method

The equation:

$$Q = \frac{(ACH) \times (room\ volume)}{60\ \mathrm{min}/h} = cu\ ft/\mathrm{min} \qquad \text{[Eq D-1]}$$

is used in calculating cubic feet per minute of infiltration air. In this equation, Q is the volume flow rate of air being calculated, and ACH is the number of air changes per hour expected, based on the type of construction (tight, medium, or loose) under the given conditions. Table D-1 is used in selecting values of ACH (ASHRAE 1979).

Example:

The infiltration of a room with dimensions 30 × 60 × 16 ft must be determined as part of a heat load calculation for winter time. The outside temperature is 0 °F with a 15 mi/h wind. The type of construction is medium.

Solution:

First, refer to Part B of Table D-1, and locate medium construction at 0 °F; the given value is 1.1 ACH. Inserting this value and the dimensions into the given equation provides the solution.

$$Q = \frac{(ACH)\times(roomvolume)}{60\ \text{min}/h} = \frac{(1.1)\times(30\times60\times16)}{60\ \text{min}/h} = 528\ cu\ ft/\text{min}$$

So, 528 cu ft/min of 0 °F air is entering this particular room.

Crack Method

The crack method assumes that data on wind velocities are known. It also assumes the doors and openable windows represent all the cracks by which outside air infiltrates a closed room under worst conditions. The following procedure is used to determine the infiltration.

The letter *k* represents the values for "window fit" and "door fit." These values are obtained from Part C of Table D-2 (ASHRAE 1979). *k* values are based upon tight, average, or loose fitting doors and windows.

Determine the outside average wind velocity in miles per hour and use Part A of Table D-2 to get the Velocity Head Factor (VHF). With the VHF, go to Part B of Table D-2 and use the VHF and *k*-curve to get the infiltration rate in cfm/ft.

Part A. Construction Types

Construction Type	Description
Tight	New buildings where there is close supervision of workmanship and special precautions are taken to prevent infiltration. Descriptions for tight windows and doors are given in Table 4.21.
Medium	Building is constructed using conventional construction procedures. Medium-fitting windows and doors are described in Table 4.21.
Loose	Buildings constructed with poor workmanship or older buildings where joints have separated. Loose windows and doors are described in Table 4.21.

Part B. Design Infiltration Rate (ACH) for Winter; Heating; Wind Speed = 15 mph

Type of Construction	Winter Outdoor Design Temperature (F)									
	50	40	30	20	10	0	−10	−20	−30	−40
Tight	0.4	0.5	0.6	0.6	0.7	0.8	0.8	0.9	0.9	1.0
Medium	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.2	1.3	1.4
Loose	0.8	0.9	1.0	1.2	1.3	1.4	1.5	1.6	1.8	1.9

Part C. Design Infiltration Rate (ACH) for Summer; Cooling; Wind Speed = 7.5 mph

Type of Construction	Summer Outdoor Design Temperature (F)					
	85	90	95	100	105	110
Tight	0.3	0.3	0.3	0.4	0.4	0.4
Medium	0.4	0.4	0.5	0.5	0.5	0.6
Loose	0.4	0.5	0.6	0.6	0.7	0.8

Part D. Infiltration per Square Foot of Floor Area

Ceiling Height (ft)	Air Changes per Hour																	
	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0
	cfm/ft²																	
7.5	0.04	0.05	0.06	0.08	0.09	0.10	0.11	0.13	0.14	0.15	0.16	0.18	0.19	0.20	0.21	0.23	0.24	0.25
8	0.04	0.05	0.07	0.08	0.09	0.11	0.12	0.13	0.15	0.16	0.17	0.19	0.20	0.21	0.23	0.24	0.26	0.27
8.5	0.04	0.06	0.07	0.09	0.10	0.11	0.13	0.14	0.16	0.17	0.18	0.20	0.21	0.23	0.24	0.26	0.27	0.28
9	0.05	0.06	0.08	0.09	0.11	0.12	0.14	0.15	0.17	0.18	0.20	0.21	0.23	0.24	0.26	0.27	0.29	0.30
	Btu/h ft² F																	
7.5	0.04	0.05	0.07	0.08	0.09	0.11	0.12	0.14	0.15	0.16	0.18	0.20	0.20	0.22	0.23	0.24	0.26	0.27
8	0.04	0.06	0.07	0.09	0.10	0.12	0.13	0.14	0.16	0.17	0.19	0.22	0.22	0.23	0.24	0.26	0.27	0.29
8.5	0.05	0.06	0.08	0.09	0.11	0.12	0.14	0.15	0.17	0.18	0.20	0.23	0.23	0.24	0.26	0.28	0.29	0.30
9	0.05	0.06	0.08	0.10	0.11	0.13	0.15	0.16	0.18	0.19	0.21	0.24	0.24	0.26	0.28	0.29	0.31	0.32

Table D-1. Estimated Overall Infiltration Rates for Small Buildings.

Source: *Mechanical and Electrical Equipment for Buildings,* 7th Ed., Stein, Reynolds, and McGuinness, copyright ©1986. This material is used by permission of John Wiley & Sons, Inc.

Next, determine the linear feet of "crack" (LFC). The following example illustrates how to obtain the LFC. Using the equation:

$$Q = (LFC) \times (\text{infiltration rate}) \qquad \text{[Eq D-2]}$$

determine the infiltration in cfm.

Example

Consider the same room as used in the air-change method. Two opening windows are the only exterior openings for the enclosure (sizes are given in Figure D-2).

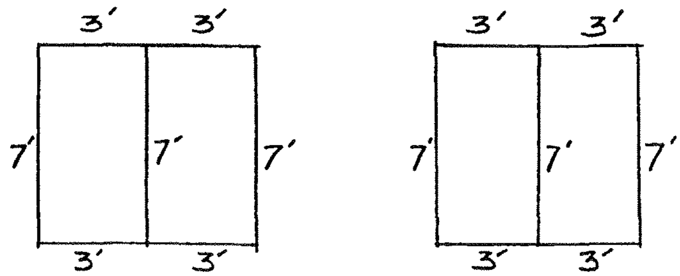


Figure D-2. Linear Feet of Crack of Windows.

From examining the windows and reading Part C of Table D-2, k = 2.0. The average wind speed for winter is found to be 15 mph.

Solution

Using Part A of Table D-2 with a wind velocity of 15 mph, the VHF is found to be 0.105.

With the obtained VHF, go to Part B of Table D-2 and obtain an approximate 0.49 (cfm/ft of crack) infiltration rate.

Next, determine the LFC of the two windows. The windows are identical in size and shape, so find the LFC for one window and multiply by 2.

LFC × 2 = (7' + 7' + 7' +3' + 3' + 3' + 3') = 66 ft of crack

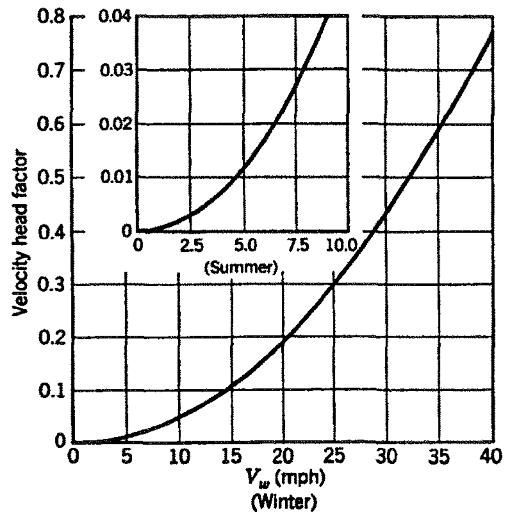
Next, substitute values into the given equation:

Q = (LFC) × (infiltration rate) = (66) × (0.49 cfm/ft) = 32.2 cu ft/min

Notice that for the same room, the air-change method estimated 528 cu ft/min was entering the room. In comparison to the crack method, there is a difference of 496 cu ft/min. It is important to select the method most appropriate for the existing circumstances.

Part A. Converting Wind Speed to Velocity Head Factor

NOTE: Typical design assumptions:
Winter wind V_w = 15 mph = velocity head factor of 0.105
Summer wind V_w = 7.5 mph = velocity head factor of 0.028



Part B. Infiltration Rates for Velocity Head Factors

NOTE: **Enter this graph with velocity head factor (from Part A) to find infiltration rate in cfm/ft of crack (using values of *k* found in Part C or D).**

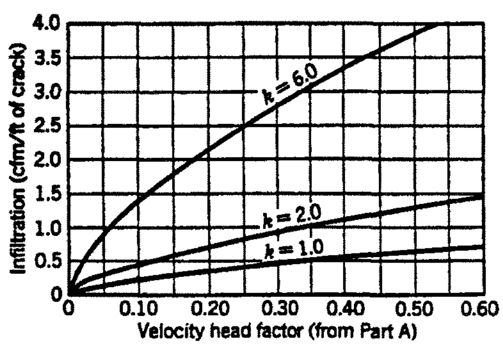


Table D-2. Approximate Infiltration Through Doors and Windows of Small Buildings.

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Part C. Classifications of Windows for Infiltration

Window Fit	*Wood Double-Hung (Locked)*	*Other Types*
Tight, k = 1.0	Weather-stripped; average gap (1/64-in. crack)	Wood casement and awning windows; weather-stripped.
		Metal casement windows; weather-stripped.
Average, k = 2.0	Nonweather-stripped; average gap (1/64-in. crack) or weather-stripped; large gap (3/32-in. crack)	All types of vertical and horizontal sliding windows; weather-stripped. If average gap (1/64-in. crack), this could be tight-fitting window.
		Metal casement windows; non-weather-stripped. If large gap (3/32-in. crack), this could be a loose-fitting window.
Loose, k = 6.0	Non-weather-stripped; large gap (3/32-in. crack)	Vertical and horizontal sliding windows; non-weather-stripped.

Part D. Classification of Residential-type Doors for Infiltration

Door Fit	*Comments*
Tight, k = 1.0	Very small perimeter gap and perfect fit weather-stripping—often characteristic of new doors
Average, k = 2.0	Small perimeter gap having stop trim fitting properly around door; weather-stripped
Loose, k = 6.0	Large perimeter gap having poor fitting stop trim; weather-stripped or Small perimeter gap; no weather-stripping

Table D-2. Approximate Infiltration Through Doors and Windows of Small Buildings (cont'd).

Curtain Wall Method

For this method, the amount of entering air is based on the wind blowing straight at an exposed wall of the room. Once again, construction classifications are used to determine values of *k*. These values of *k* are representative of tight, average, or loose fitting walls as designated in Table D-3.

Obtain the wind velocity and use Figure D-3 to determine velocity head in the form of $\Delta P_w/C_p$. ΔP_w is the change in pressure (inches of water), and C_p is the pressure coefficient for curtain wall buildings.

Leakage Coefficient	Description	Curtain Wall Construction
k = 0.22	Tight Fitting Wall	Constructed under close supervision of workmanship on wall joints. When joint seals appear inadequate they must be re-done.
k = 0.66	Average Fitting Wall	Conventional construction procedures are used.
k = 1.30	Loose Fitting Wall	Poor construction quality control or an older building having separate wall joints.

Table D-3. Curtain Wall Classification.

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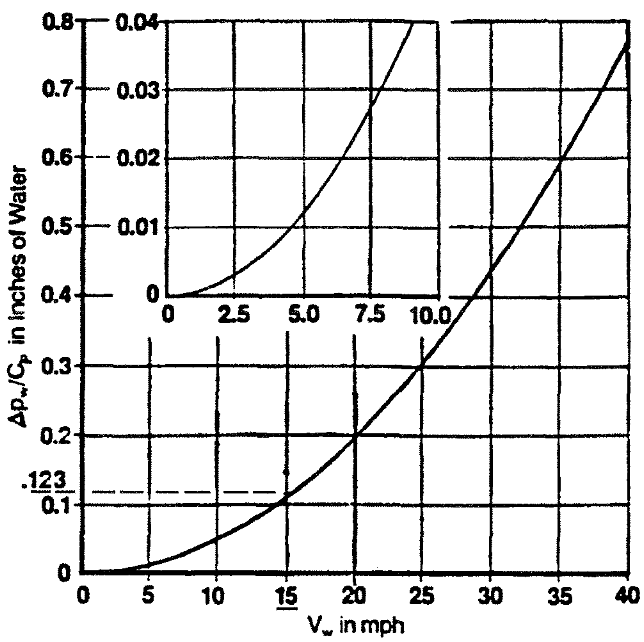

Figure D-3. Velocity Head vs. Wind Velocity.

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Then go to Table D-4 to get the C_p value. These values are determined by the direction of the wind (windward, leeward, and sides). By multiplying $\Delta P_w/C_p$ by C_p, the value of ΔP is obtained.

With ΔP, use Figure D-4 to obtain the air flow per square foot, $Q/A \; in \; cfm/sq \; ft$. The square foot area of the curtain wall under construction is then calculated and inserted into the equation Q = A × (Q/A) = cfm.

The table is for a rectangular floor-shaped building and for wind normal to windward side.	
	C_p
Windward	0.95
Leeward	-0.15
Sides	-9.40

Table D-4. Wind Pressure Coefficients for Curtain Wall Buildings.

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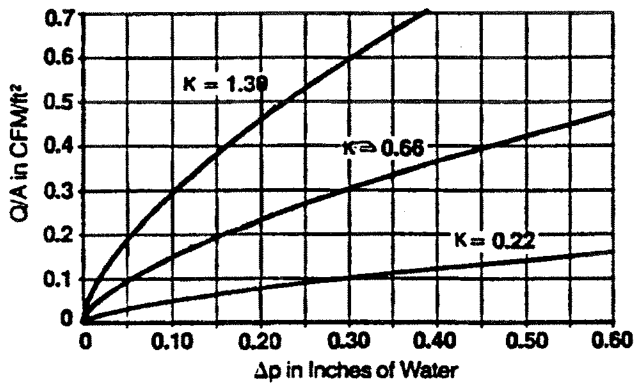

Figure D-4. Curtain Wall Infiltration for One Room or One Floor.

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Example

GIVEN: $k = 0.66$ (average)
wind velocity = 15 mph
$C_p = 0.95$ (windward)

FIND: Infiltration through the 60 x 16 ft wall of the room used in the two preceding examples.

Solution

$\Delta P_w/C_p = 0.123$ in. of water (Figure D-3)

$\Delta P = (\Delta P_w/C_p) \times C_p = (0.123) \times (0.95) = 0.1169$

Q/A = 0.175 cfm/sq ft (Figure D-4)

Area of Wall = 60 x 16 ft = 960 *sq ft*

$Q = A \times (Q/A) =$ 960 *sq ft* $\times$ (0.175 cfm/sq ft) = 168 cfm

Stack Effect Method

When there is a difference in height between inlet openings situated low in the wall (or in floors) and outlets through roofs, and when outdoor air is cooler than indoor air, natural ventilation will occur through the stack effect of warm air rising and leaving through the higher openings.

The equation:

$$Q = C \times A \times \frac{h \times (t_i - t_o)}{t_i} \qquad \text{[Eq D-3]}$$

is used in calculating infiltration due to the stack effect. In this equation:

Q = air flow (cfm)

C = constant of proportionality = 313 (This assumes a value of 65 percent of the maximum theoretical flow, due to limited effectiveness of actual openings. With less favorable conditions, due to indirect paths from openings to the stack, etc., the effectiveness drops to 50 percent, and C = 240.)

A = area of cross-section through stack or outlets ($sq\ ft$)
Note: Inlet area must be at least equal to this amount.

t_i = (higher) temperature inside (°F), within the height h

t_o = (lower) temperature outside (°F)

h = height difference between inlets and outlets.

Example

Openings to the outside are indirect to the stack of a building; therefore, C=240. The cross-sectional area through the stack was measured to be 1.5 sq ft. Outlets are measured to be 15 ft above inlets. An outside temperature of 0 °F is measured. The inside temperature is 74 °F. Determine the amount of air entering the building due to the stack effect.

Solution

$$Q = C \times A \times \frac{h \times (t_i - t_o)}{t_i}$$

$$Q = 240 \times (1.5\ sq\ ft) \times \frac{15\ ft \times (74^\circ F - 0^\circ F)}{74^\circ F}$$

$$= 1394\ cu\ ft/\text{min}$$

This is approximately three air changes per hour for the room used in the previous examples.

Natural Ventilation Guidelines

Several general guidelines should be followed when designing for natural ventilation:

- In hot, humid climates, maximize air velocities in the occupied zones for bodily cooling. In hot, arid climates, maximize air flow throughout the building for structural cooling, particularly at night when temperatures are low.
- Take advantage of topography, landscaping, and surrounding buildings to redirect airflow and give maximum exposure to breezes. Use vegetation to funnel breezes and avoid wind dams that reduce the driving pressure differential around the building. Site objects should not obstruct inlet openings.
- The stack effect requires vertical distances between openings to take advantage of the effect; the greater the vertical distance, the greater the ventilation.
- Openings with areas much larger than calculated are sometimes desirable when anticipating increased occupancy or very hot weather.
- Horizontal windows are generally better than square or vertical windows. They produce more airflow over a wider range of wind directions and are most beneficial in locations where prevailing wind patterns shift.
- Window openings should be accessible to and operable by occupants.

- Vertical air shafts or open staircases can be used to increase and take advantage of stack effects. However, enclosed staircases intended for evacuation during a fire should not be used for ventilation.

Infiltration Measurement

Fan Pressurization

The fan pressurization method, sometimes called the "Minneapolis Blower Door," is used in measuring the amount of infiltration into the building and in locating leaks.

This method measures the building leakage rate independent of weather conditions. Equipment required for a quantitative measurement includes a blower (variable speed fan), a flow meter, a pressure gauge, and (optionally) a smoke source or an infrared scanning device to locate leaks. Also, a means of sealing the fan into the doorway is required so the only air going through the doorway passes through the fan.

The fan is generally used to move a large stream of air out of the building so that even the most minute streams of air (leaks) coming in may be detected. Moving air into or out of the building causes a different air pressure inside the building relative to the outside air pressure. If air is being forced out, the inside pressure is lower and vice versa.

When the inside pressure is low, air leaks into the building through any hole it can find in the exterior envelope of the structure. Leak locations can be found by checking suspected trouble spots for drafts with a smoke stick, an infrared camera, or even a person's hand.

Some of the common leak locations are shown in Figure D-5.

Figure D-5 also shows how air flows naturally through a building. As warm air rises, it tends to escape through cracks and holes near the top of the building. This escaping air causes a slight suction, which pulls in cold air through holes near the bottom of the building. These holes throughout the interior of the building need to be sealed to reduce air movement (heat loss).

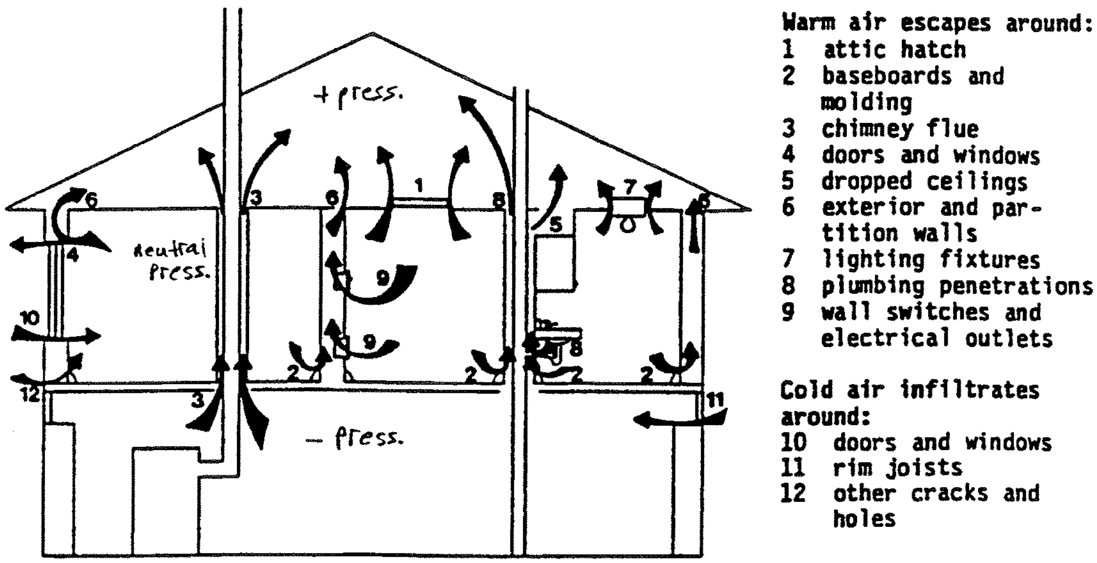

Figure D-5. Natural Air Flow Through a Building.

When calculating infiltration, it is important to select a method that facilitates the equipment needed to execute the method chosen. Any of the preceding methods may be applied in a variety of building types. If the blower door method is to be used, however, the portable air-tight door frame, variable speed fan, and pressure measuring gauges must be available.

Calculations by the air-change method have been compared to tracer gas measurements for two houses in California and two in Minnesota. The comparison showed that the air-change method can give estimates to within 20 percent of measurements for average construction under average conditions.

The air-change method should be used as a gross estimate at best. The accuracy of the crack method for design load calculations has restrictions of limited data on air leakage characteristics of components and by the difficulty of estimating pressure differences under appropriate design conditions of temperature and wind.

If a method is being selected for design purposes, some knowledge of infiltration will be necessary. In analyzing an existing condition, having the proper equipment and knowing how to use it, and knowing how to tabulate and analyze data will be required. To determine the degree of accuracy for the method chosen, the person(s) doing the analysis will need to determine their own degree of satisfaction in the selected method.

3 Ventilation

General Ventilation

General ventilation controls heat, odors, and contaminants. It may be provided by natural draft, by a combination of general supply and exhaust air fan and duct systems, by exhaust fans only (with replacement air through inlet louvers and doors), or by supply fans only (exhaust through relief louvers and doors).

It is important to provide at least a minimum amount of fresh air indoors, both for comfort and for health. Odors and a sense of staleness can be uncomfortable, and buildups of pollutants can be produced within buildings. These pollutants are easily removed with air changes through rooms.

Winter heat loss (and summer heat gain in closed, cooled buildings) occurs when fresh outdoor air enters a building to replace stale indoor air. This heat exchange must be calculated when sizing heating or cooling equipment or when estimating energy use per season.

Air exchange increases a building's thermal load in three ways. First, the incoming air must be heated or cooled from the outdoor air temperature to the indoor air temperature. Second, air exchange increases a building's moisture content, which means humid outdoor air must be dehumidified. Third, air exchange can increase a building's thermal load by decreasing the performance of the envelope insulation system. Air flowing around and through the insulation can increase heat transfer rates above design rates. Air flow within the insulation system can also decrease the system's performance due to moisture condensing in and on the insulation.

The calculation of the heat lost (or gained) by the introduction of outdoor air into spaces is:

$$q_v = (V) \times (1.08) \times (\Delta t)$$

where:

q_v = sensible heat exchange due to ventilation (Btu/h)

V = volume flow rate, in cubic feet per minute (cfm) of outdoor air introduced (see Q in Chapter 2 of this appendix and examples in infiltration section)

Δt = temperature difference between outdoor and indoor air °F

1.08 = A constant derived from the density of air at 0.075 *lb/cu ft* under average conditions, multiplied by the specific heat of air (heat required to raise 1 lb of air 1 °F), which is 0.24 Btu/lb °F, and by 60 min/h. The units of this constant are Btu min/*cu ft* °F h.

Forced Ventilation

Fans can be used to forcibly introduce the desired amount of outdoor air directly into spaces. Fan manufacturers list their capacity in cubic feet per hour (cfh) or cubic feet per minute (cfm). This outdoor air can be blown into spaces, or it can be mixed with air being recirculated so that the different temperature of outdoor air is less noticeable.

Forced ventilation offers energy conservation opportunities if a heat exchanger is used. Outgoing and incoming airstreams can be kept separate but allow heat to transfer from one stream to the other. An example of this can occur during the winter months. Incoming very cold outdoor air can be given the heat, but not the pollutants, of outgoing warm indoor air. The reverse of this would happen in the summer.

To approximate the size of a fan, the following equations are used:

$$Q = (\text{cfm outdoor air person}) \times (\text{number of people})$$

or

$$Q = (\text{cfm/sq ft floor area}) \times (\text{sq ft floor area})$$

Q is the desired flow rate. The "cfm outdoor air person" and "cfm/sq ft floor area" expressions are ASHRAE's recommended design outdoor airflow rates. These values are found in Tables D-5 through D-7 on the following pages.

Estimating Heating/Ventilating Loads

The heating/ventilating loads of a building or an area of a building can be calculated from the following data:

1. Use required weather data tables to determine the outdoor design conditions.
2. Select the indoor design conditions for each room or space to be heated using the coldest weather. Determine each temperature difference (Δt).
3. Measure or estimate the temperatures in adjacent unheated spaces. Determine each temperature difference (Δt).
4. Calculate the net areas of all walls, glass, doors, ceilings, floors, partitions, etc., from building plans or from field measurements.
5. Determine the heat transmission loss coefficients ("U" values) for each area and type of construction. "U" is the thermal transmittance and is the overall expression of the steady state rate at which heat flows through architectural skin elements (walls, roofs, floors, etc.). This term is expressed in terms of Btu/h sq ft °F, and can be found from tables and charts found in the ASHRAE *Fundamentals Handbook*.

OUTDOOR AIR REQUIREMENTS FOR VENTILATION

COMMERCIAL FACILITIES (offices, stores, shops, hotels, sports facilities)

Application	Estimated Maximum** Occupancy P/1000 ft^2 or 100 m^2	Outdoor Air Requirements cfm/ person	L/s • person	cfm/ft^2	L/s • m^2	Comments
Dry Cleaners, Laundries						Dry-cleaning processes may require more air.
Commercial laundry	10	25	13			
Commercial dry cleaner	30	30	15			
Storage, pick up	30	35	18			
Coin-operated laundries	20	15	8			
Coin-operated dry cleaner	20	15	8			
Food and Beverage Service						
Dining rooms	70	20	10			
Cafeteria, fast food	100	20	10			
Bars, cocktail lounges	100	30	15			Supplementary smoke-removal equipment may be required.
Kitchens (cooking)	20	15	8			Makeup air for hood exhaust may require more ventilating air. The sum of the outdoor air and transfer air of acceptable quality from adjacent spaces shall be sufficient to provide an exhaust rate of not less than 1.5 cfm/ft^2 (7.5 L/s•m^2).
Garages, Repair, Service Stations						
Enclosed parking garage				1.50	7.5	Distribution among people must consider worker location and concentration of running engines; stands where engines are run must incorporate systems for positive engine exhaust withdrawal. Contaminant sensors may be used to control ventilation.
Auto repair rooms				1.50	7.5	
Hotels, Motels, Resorts, Dormitories				**cfm/room**	**L/s • room**	Independent of room size.
Bedrooms				30	15	
Living rooms				30	15	
Baths				35	18	Installed capacity for intermittent use.
Lobbies	30	15	8			
Conference rooms	50	20	10			
Assembly rooms	120	15	8			
Dormitory sleeping areas	20	15	8			See also food and beverage services, merchandising, barber and beauty shops, garages.
Gambling casinos	120	30	15			Supplementary smoke-removal equipment may be required.
Offices						
Office space	7	20	10			Some office equipment may require local exhaust.
Reception areas	60	15	8			
Telecommunication centers and data entry areas	60	20	10			
Conference rooms	50	20	10			Supplementary smoke-removal equipment may be required.
Public Spaces				**cfm/ft^2**	**L/s • m^2**	
Corridors and utilities				0.05	0.25	
Public restrooms, cfm/wc or cfm/urinal		50	25			Normally supplied by transfer air. Local mechanical exhaust with no recirculation recommended.
Locker and dressing rooms				0.5	2.5	
Smoking lounge	70	60	30			
Elevators				1.00	5.0	Normally supplied by transfer air.

* Table prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen to control CO_2 and other contaminants with an adequate margin of safety and to account for health variations among people, varied activity levels, and a moderate amount of smoking.

**Net occupiable space.

Table D-5. Outdoor Air Requirements for Commercial Facilities.
Reprinted with permission from *ASHRAE Standard 62-1989.*

Application	Estimated Maximum** Occupancy P/1000 ft² or 100 m²	Outdoor Air Requirements cfm/ person	L/s • person	cfm/ft²	L/s • m²	Comments
Retail Stores, Sales Floors, and Show Room Floors						
Basement and street	30			0.30	1.50	
Upper floors	20			0.20	1.00	
Storage rooms	15			0.15	0.75	
Dressing rooms				0.20	1.00	
Malls and arcades	20			0.20	1.00	
Shipping and receiving	10			0.15	0.75	
Warehouses	5			0.05	0.25	
Smoking lounge	70	60	30			Normally supplied by transfer air, local mechanical exhaust; exhaust with no recirculation recommended.
Specialty Shops						
Barber	25	15	8			
Beauty	25	25	13			
Reducing salons	20	15	8			
Florists	8	15	8			Ventilation to optimize plant growth may dictate
Clothiers, furniture				0.30	1.50	requirements.
Hardware, drugs, fabric	8	15	8			
Supermarkets	8	15	8			
Pet shops				1.00	5.00	
Sports and Amusement						
Spectator areas	150	15	8			When internal combustion engines are operated for main-tenance of playing surfaces,
Game rooms	70	25	13			increased ventilation rates may be required.
Ice arenas (playing areas)				0.50	2.50	Incorporate systems for positive engine exhaust with-drawal.
Swimming pools (pool and deck area)				0.50	2.50	Higher values may be required for humidity control.
Playing floors (gymnasium)	30	20	10			
Ballrooms and discos	100	25	13			
Bowling alleys (seating areas)	70	25	13			
Theaters						
Ticket booths	60	20	10			Special ventilation will be needed
Lobbies	150	20	10			to eliminate special stage effects
Auditorium	150	15	8			(e.g., dry ice vapors, mists, etc.)
Stages, studios	70	15	8			
Transportation						
Waiting rooms	100	15	8			Ventilation within vehicles may require
Platforms	100	15	8			special considerations.
Vehicles	150	15	8			
Workrooms						
Meat processing	10	15	8			Spaces maintained at low temperatures
Photo studios	10	15	8			(-10°F to +50°F, or -23°C to +10°C) are not covered by
Darkrooms	10			0.50	2.50	these requirements unless the occupancy is continuous.
Pharmacy	20	15	8			Ventilation from adjoining spaces is permissible. When
Bank vaults	5	15	8			The occupancy is intermittent, infiltration will normally exceed the ventilation requirement.
Duplicating, printing				0.50	2.50	Installed equipment must incorporate positive exhaust and control (as required) or undesirable contaminants (toxic or otherwise).

INSTITUTIONAL FACILITIES

Application	Estimated Maximum** Occupancy P/1000 ft² or 100 m²	Outdoor Air Requirements cfm/ person	L/s • person	cfm/ft²	L/s • m²	Comments
Education						
Classroom	50	15	8			
Laboratories	30	20	10			Special contaminant control
Training shop	30	20	10			systems may be required for
Music rooms	50	15	8			processes or functions including
Libraries	20	15	8			laboratory animal occupancy.
Locker rooms				0.50	2.50	
Corridors				0.10	0.50	
Auditoriums	150	15	8			
Smoking lounges	70	60	30			Normally supplied by transfer air. Local mechanical exhaust with no recirculation recommended.
Hospitals, Nursing, Convalescent Homes						
Patient rooms	10	25	13			Special requirements or codes and pressure
Medical procedure	20	15	8			relationships may determine minimum
Operating rooms	20	30	15			ventilation rates and filter efficiency.
Recovery and ICU	20	15	8			Procedures generating contaminants may require higher rates.
Autopsy rooms				0.50	2.50	Air shall not be recirculated into other spaces.
Physical Therapy	20	15	8			
Correctional Facilities						
Cells	20	20	10			
Dining halls	100	15	8			
Guard stations	40	15	8			

* Table prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen to control CO_2 and other contaminants with an adequate margin of safety and to account for health variations among people, varied activity levels, and a moderate amount of smoking.

**Net occupiable space.

Table D-6. Outdoor Air Requirements for Institutional Facilities.
Reprinted with permission from *ASHRAE Standard 62-1989*.

6. Calculate the heat losses for walls, ceilings, partitions, glass, doors, and floors (above grade) to unheated areas using the equation:

$$Q = A \times U \times \Delta t \qquad \text{[Eq D-4]}$$

7. Calculate the heat losses for slab floors and basement walls below grade by using the equation:

$$Q = A \times U \times \Delta t$$

and a Dt determined by temperatures from tables and charts found in the ASHRAE *Fundamentals Handbook*.

OUTDOOR AIR REQUIREMENTS FOR VENTILATION OF RESIDENTIAL FACILITIES[a]
(Private Dwellings, Single, Multiple)

Applications	Outdoor Requirements	Comments
Living areas	0.35 air changes per hour but not less than 15 cfm (7.5 L/s) per person	For calculating the air changes per hour, the volume of the living spaces shall include all areas within the conditioned space. The ventilation is normally satisfied by infiltration and natural ventilation. Dwellings with tight enclosures may require supplemental ventilation supply for fuel-burning appliances, including fireplaces and mechanically exhausted appliances. Occupant loading shall be based on the number of bedrooms as follows: first bedroom, two persons; each additional bedroom, one person. Where higher occupant loadings are known, they shall be used.
Kitchens[b]	100 cfm (50 L/s) intermittent or 25 cfm (12 L/s) continuous or openable windows	Installed mechanical exhaust capacity[c]. Climatic conditions may affect choice of the ventilation system.
Baths, Toilets[b]	50 cfm (50 L/s) intermittent or 20 cfm (10 L/s) continuous or openable windows	Installed mechanical exhaust capacity[c].
Garages: Separate for each dwelling unit	100 cfm (50 L/s) per car	Normally satisfied by infiltration or natural ventilation.
Common for several units	1.5 cfm/ft^2 (7.5 L/s • m^2)	See "Enclosed parking garage," Table D-5

[a]In using this table, the outdoor air is assumed to be acceptable.
[b]Climatic conditions may affect choice of ventilation option chosen.
[c]Air exhausted from kitchens, bath, and toilet rooms may use air supplied through adjacent living areas to compensate for the air exhausted.

The air supplied shall meet the requirements of exhaust systems as described in 5.8 and be of sufficient quantities to meet the requirements of this table.

Table D-7. Recommended Outdoor Air Requirements for Residential Facilities.

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8. Calculate the heat losses for slab floors on grade by using the equation:

$$Q = F_2 \times P \times \Delta t \qquad \text{[Eq D-5]}$$

where P = perimeter of slab (feet), and F_2 is the "Heat Loss Coefficient of Slab Floor Construction" which can be found in the ASHRAE *Fundamentals Handbook*.

9. Calculate the infiltration of each room or area.

10. When outdoor air is introduced through a HVAC unit (makeup or ventilation air), that load must be part of the total ventilation load requirements when making the calculations. Use whichever load (ventilation or infiltration) is determined to be greater.

11. The total heating load is the sum of all the above heat transmission and infiltration/ventilation loads, which are considered to peak at the same time.

12. In buildings that have a permanent, steady internal heat source of considerable size (such as ovens, 24-h intensive lighting systems, etc.), an equivalent amount of heat could be deducted from the calculated total heating load, provided the load could not be cut off at a future date.

13. A limited amount of additional heating capacity should be added for a "pick-up load" for buildings that have night and/or weekend setback or are intermittently heated.

Design Considerations

General ventilation may be provided with either natural or mechanical supply and/or exhaust systems. Some ventilation systems must handle simultaneous exposures to hazardous substances and heat. In such cases, ventilation may consist of a combination of local, general supply, and exhaust air systems. Some factors to consider in selection and design are as follows:

- Local exhaust systems provide general ventilation for the work area.
- A balance of the supply and exhaust systems is required for either system to function as designed.
- Natural ventilation systems are most applicable when internal heat loads are high, and the building is tall enough to produce a significant stack effect.
- To provide effective general ventilation for heat relief by either natural or mechanical supply, the air must be delivered in the work zones (no more than 10 ft above the floor) with an appreciable air velocity. A sufficient exhaust volume is necessary to remove the heat liberated in the space. Local relief systems may require supplementary supply air for heat removal.
- Supply and exhaust air cannot be used interchangeably. Supply air can be delivered where it is wanted at controlled velocities, temperature, and humidity. Exhaust systems should be used to capture heat and fumes at the source.
- General building exhaust may be required in addition to local exhaust systems.
- The exhaust discharge, whether local or general, should be located where it will not be recirculated.

Ventilation Air Velocity

The level of air motion at the worker is important. At fixed work positions with light activity, the velocity should not exceed 200 fpm for continuous exposure. With high work levels and intermittent exposures, velocities of 400 to 800 fpm may be used. When high-velocity air is used, it is important to avoid the undesirable effects of hot air convection and disturbance of local exhaust ventilation systems. Table D-8 lists some acceptable air motion rates.

Exposure	Air Velocity, fpm
Continuous	
Air-conditioned space	50 to 75
Fixed workstation, general ventilation, or spot cooling	
Sitting	75 to 125
Standing	100 to 200
Intermittent, spot cooling, or relief stations	
Light heat loads and activity	1000 to 2000
Moderate heat loads and activity	2000 to 3000
High heat loads and activity	3000 to 4000

Table D-8. Acceptable Air Motion at the Worker.

Locker Room, Toilet, and Shower Space Ventilation

The ventilation of locker rooms, toilets, and shower spaces is important in removing odor and in reducing humidity. State and local regulations should be consulted when designing these facilities.

Supply air may be introduced through door or wall grilles. In some cases, plant air may be so contaminated that filtration or mechanical ventilation may be required. When control of workroom contaminants is inadequate, the total exposure to employees can be reduced by making sure that the level of contamination in the locker rooms, lunchrooms, and break rooms is minimized by pressurizing these areas with excess supply air.

When mechanical ventilation is used, the supply system should have supply fixtures such as wall grilles, ceiling diffusers, or supply plenums to distribute the air adequately throughout the area.

In locker rooms, the exhaust should be taken primarily from the toilet and shower spaces, as needed, and the remainder from the lockers and the room ceiling. Table D-9 provides a guide for ventilation of these spaces.

Description	Inch-Pound Units	SI Units
Locker Rooms		
Coat hanging or clean change room for nonlaboring shift employees with clean work clothes	1 cfm/sq ft	5 L/s•m^2
Change room for laboring employees with wet or sweaty clothes	2 cfm/sq ft; 7 cfm exhausted from each locker	10 L/s•m^2; 3 L/s exhausted from each locker
Change room for heavy laborers or workers assigned to working and cleaning where clothes will be wet or pick up odors	3 cfm/sq ft; 10 cfm exhausted from each locker	20 L/s•m^2; 5 L/s exhausted from each locker
Toilet Spaces	2 cfm/sq ft; at least 25 cfm per toilet facility; 200 cfm min.	10 L/s•m^2; at least 10 L/s per toilet facility; 90 L/s min.
Shower Spaces	2 cfm/sq ft; at least 50 cfm per shower head; 200 cfm min.	10 L/s•m^2; at least 20 L/s per shower head; 90 L/s min.

Table D-9. Ventilation for Locker Rooms, Toilets, and Shower Spaces.

4 Exhaust

Exhaust ventilation systems collect and remove airborne contaminants consisting of dusts, fumes, mists, fibers, vapors, and gases that can create an unsafe, unhealthy, or undesirable atmosphere. Exhaust systems are used to remove impurities from the air at the source, preventing them from contaminating the bulk of the air in the building. Necessary air changes are then held to a minimum.

Replacement air, which is usually conditioned, provides air to the work space to replace exhausted air, and the systems are not isolated from each other. A complete industrial ventilation program includes replacement air systems that provide a total volumetric flow rate equal to the total exhaust rate. 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.

There are two types of exhaust systems:

1. General Exhaust, in which an entire work space is exhausted without considering specific operations.
2. Local Exhaust, in which the contaminant is controlled at its source.

General Exhaust/Dilution Ventilation

The terms "general exhaust" and "dilution ventilation" are often used interchangeably. This type of exhaust refers to dilution of contaminated air with uncontaminated air in a general area, room, or building for the purpose of health hazard or nuisance control.

In general, dilution ventilation is not as satisfactory for health hazard control as is local exhaust. In some cases, dilution ventilation must be used because the operation or process prohibits local exhaust. Circumstances may be found in which dilution ventilation provides an adequate amount of control more economically than a local exhaust system. Economical considerations should not be based entirely upon the first cost of the system because dilution ventilation frequently exhausts large volumes of heat from a building and can easily be a troublesome factor.

The use of dilution ventilation has four limiting factors:

1. The quantity of contaminant generated must not be too great or the air volume necessary for dilution will be impractical.
2. Workers must be far enough away from contaminant evolution, or evolution of contaminant must be in sufficiently low concentrations so workers will not have an exposure in excess of the established Threshold Limit Values (TLVs).
3. The toxicity of the contaminant must be low.
4. The evolution of contaminants must be reasonably uniform.

Dilution ventilation is seldom applied to fumes and dusts because the high toxicities often encountered require too great a quantity of dilution air, velocity and rate of evolution are usually very high, and data on the amount of fumes and dust production are very difficult, if not impossible, to obtain.

Dilution ventilation is most often used to advantage to control the vapors from organic liquids such as the less toxic solvents. To successfully apply the principles of dilution to such a problem, factual data are needed on the rate of vapor generation or on the rate of liquid evaporation.

Basic Principles

Some basic principles to be applied to a dilution ventilation system are as follows:

- From factual data, select the amount of air required for satisfactory dilution of the contaminant.

- Locate the exhaust openings near the sources of contaminant if possible, in order to obtain the benefit of spot ventilation.

- For dilution methods to be effective, the exhaust outlet and air supply must be located so that all the air used in the ventilation passes through the zone of contamination.

- Replace exhausted air by a make-up air system. Make-up air should be heated during cold weather. Dilution ventilation systems usually handle large quantities of air by means of propeller fans. Make-up air usually must be provided if the ventilation is to be adequate and the system is to operate satisfactorily.

- The general air movements in the room caused by suction at the exhaust opening should keep the contaminated air between the operator and the exhaust opening, and not draw contaminants across the operator.

- A combined supply and exhaust system is preferred with a slight excess of exhaust if there are adjoining occupied spaces, and a slight excess of supply if there are no such spaces.

- Avoid re-entrance of the exhausted air by discharging the exhaust high above the roof line, or by assuring that no window, outside air intakes, or other such openings are near the exhaust discharge.

Local Exhaust

Local exhaust is preferable because it offers better contaminant control with minimum air volumes. This, in turn, lowers the cost of air cleaning and replacement air equipment.

Local exhaust systems can be classified as: (1) Constant Air Volume or (2) Variable Air Volume, based on the method of system operation and control. Each of these classifications can be further broken down into individual or central systems based on the arrangement of the major system components such as the fans, plenums, or duct mains and branches.

Constant Air Volume Systems

This type of system exhausts a fixed quantity of air from each safety cabinet, fume hood, or room module. Constant air volume systems will handle the same exhaust air quantity for any condition. For this reason, the capacities of the exhaust air and supply air systems will limit the total number of fume hoods and room modules to be installed. This type of system is flexible with respect to location of hoods but may incur high ownership and operating costs because of the large air volumes handled. These high costs may impose a limitation on the total number of hoods or modules that can be installed in the building.

Constant air volume systems are highly stable in operation and simple to balance. In most installations, there is no need for continuous adjustment of air balance during normal operation.

Variable Air Volume Systems

Variable air volume systems can shut down inactive fume hoods and room modules. This capability results in an economic system that reduces the air flow during periods when some of the hoods and room modules are not in use, and the exhaust air system is operated at less than full capacity. More freedom in the installation of the hoods and room modules is possible since the total number of units that may be connected does not entirely depend on the capacity of the exhaust system.

Variable air volume systems are not as stable in operation as constant air volume systems are. They are also more difficult to balance and control. Sensitive instrumentation and controls are required, which result in high initial and maintenance costs. Reliability in a corrosive atmosphere is highly questionable. For some applications, the use of balancing dampers in exhaust air ducts is prohibited by codes.

One problem associated with the variable air volume system is the regulation of the total simultaneous operating usage to match design usage factors. If the collective area of operating hood openings at any one time exceeds design opening diversity values, the proper face velocity requirements will not be achieved and personnel could be endangered. Visual and audible alarms should be equipped on hoods to warn workers of unsafe air flows.

Individual exhaust air systems. Individual exhaust air systems use a separate exhaust air fan, exhaust connection, and discharge duct for each hood or module. The exhaust for the hood or module served by the individual exhaust system does not directly affect the operation of any other area of the building, which permits selective operation of individual hoods and modules by starting or stopping the fan motor.

The recommended operation is for exhaust air fans to be on at all times and to be electrically interlocked so that, if any critical exhaust air fan is shut down, the supply air fans will shut down. Although more fans are used than for central systems, the overall space requirements are usually less for individual systems because of the small, direct duct connection. The use of more fans does increase capital and maintenance costs.

The shutdown of individual exhaust air systems will upset the proper directional air flow and may cause potentially hazardous contaminants and odors to flow into the corridor and adjacent rooms. If this type of system is used, precautions to reverse air flow (such as air locks) should be provided.

Central exhaust air systems. Central exhaust air systems consist of a common suction plenum, one fan, and branch connections to multiple exhaust terminals. This type of system generally costs less than individual exhaust air systems, costs less to maintain, permits low cost standby exhaust air fan provisions, and is applicable to remote high stack discharge requirements. Central systems are more difficult to balance and may have difficulties with parallel fan operation. The central exhaust air system is best when exhausting similar types of units such as laboratory fume hoods.

5 Fans

The fan is an air pump that causes airflow by creating a pressure difference. Fans produce pressure and/or flow by rotating blades of the impeller, imparting kinetic energy to the air by changing its velocity. By definition, the term "fan" is limited to devices producing pressure differentials of less than 28 in. w.g. at sea level. The following definitions and equations will help in the understanding of fans and their function in a system.

- *Brake Horsepower*—The actual horsepower required to drive the fan. This number is greater than a theoretical "air horsepower" because it includes loss due to turbulence and other inefficiencies in the fan, plus bearing losses. It is the power furnished by the fan motor.

- *Fan Air Volume*—The cubic feet per minute (cfm) of air handled by a fan at any air density. This is different from the cubic feet per minute of standard air (scfm), which is at 0.075 lb/ft.

- *Fan Outlet Velocity*—The theoretical velocity of the air as it leaves the fan outlet. This velocity is calculated by dividing the air volume in cfm by the fan outlet area in square feet. Since the velocity varies over the cross-section of all fan outlets, this value is only a theoretical value that could occur at a point removed from the fan. Because of this, all velocity readings, including total pressure and static pressure, should be taken farther along in a straight duct connected to the fan discharge where the flow is more uniform.

- *Fan Static Pressure (SP)*—The fan total pressure (TP) less the fan velocity pressure (VP) as shown in Figure D-6.

$$SP = TP_{(outlet)} - TP_{(inlet)} - VP_{(outlet)}$$

$$VP_{(outlet)} = TP_{(outlet)} - SP_{(outlet)}$$

$$SP = SP_{(outlet)} - TP_{(inlet)}$$

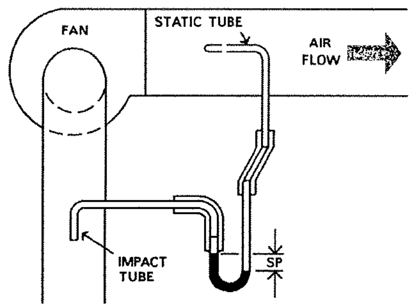

Figure D-6. Fan Static Pressure.

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- *Fan Total Pressure*—The difference between the total pressure at the fan outlet and the total pressure at the fan inlet. This value measures the total mechanical energy added to the air or gas by the fan (Figure D-7).

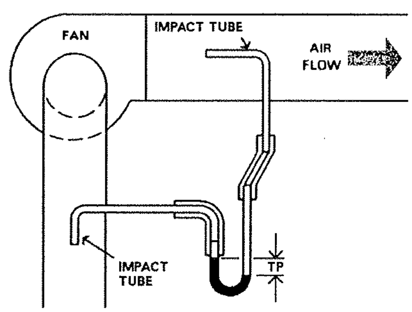

Figure D-7. Fan Total Pressure.

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- *Fan Velocity Pressure*—The pressure corresponding to the fan outlet velocity. It is the measure of kinetic energy per unit volume of flowing air (Figure D-8).

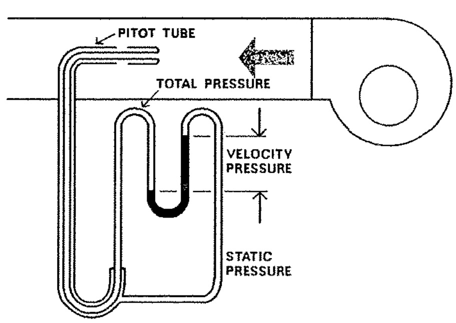

Figure D-8. Fan Velocity Pressure.

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- *Tip Speed (TS)*—The circumference of the fan wheel times the rpm of the fan, expressed in ft/min. Also known as peripheral velocity (Figure D-9).

$$TS = \frac{D \times rpm}{12}$$

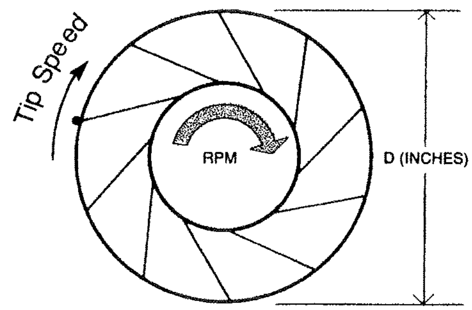

Figure D-9. Tip Speed.

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Fan Types

Fan types are generally classified by the direction of air flow through the impeller. The two main types are:

1. Centrifugal
2. Axial.

Centrifugal Fans

Centrifugal fans consist of a number of blades that are inclined in a direction opposite to the fan rotation, in a vertical position (Figure D-10). This arrangement enables the wheel to operate at a lower tip speed, giving more cfm at a lower rpm at a given static pressure. Nonoverloading characteristics are also associated with this type of fan. Centrifugal fans are used the most in comfort applications because of its wide range of quiet, efficient operation at comparatively high pressures. The centrifugal fan inlet can be readily attached to an apparatus of large cross-section, while the discharge is easily connected to relatively small ducts. Air flow can be varied to match air distribution system requirements by simple adjustments to the fan drive or control devices.

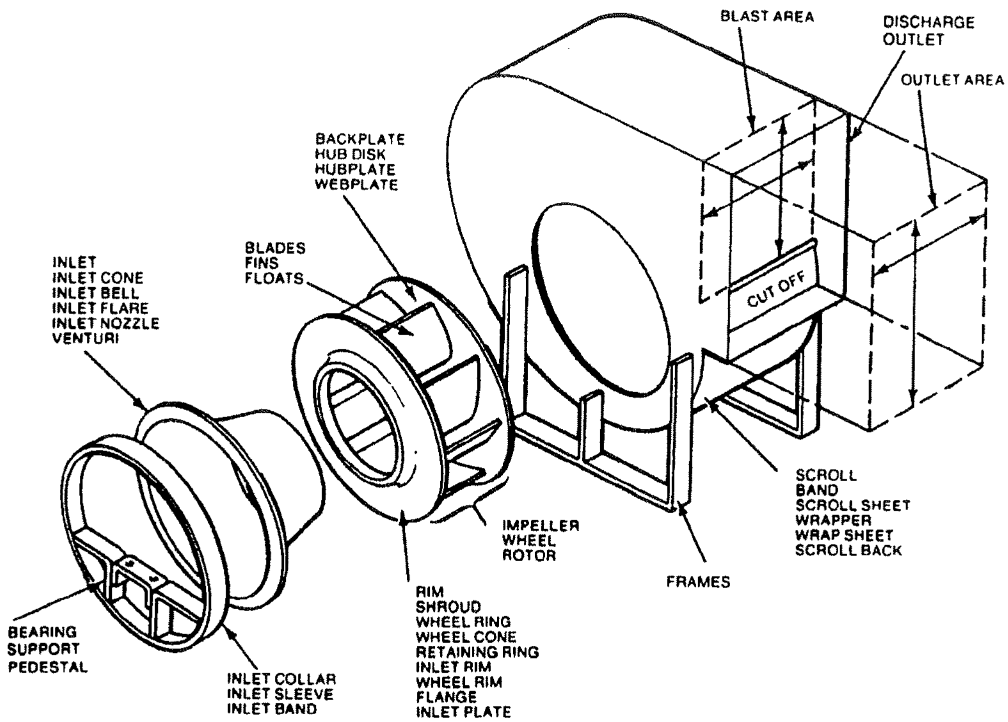

Figure D-10. Centrifugal Fan Components.

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Centrifugal fan types include:

1. Airfoil
2. Backward-curved blade
3. Radial blade
4. Forward-curved blade.

Examples of these fan types are shown in Figures D-11 through D-14. The figures are based on Air Movement and Control Association International, Inc. (AMCA) Publication 201-90.

Airfoil fans. Airfoil blades curve away from the direction of rotation, and fans consist of 10 to 16 blades. Relatively deep blades provide efficient expansion within the blade passages. When this blade is properly designed, this will be the most efficient and the highest speed of the centrifugal fan designs. The static efficiency of these fans is around 86 percent. The clearance and alignment between the wheel and inlet bell need to be very close to reach the maximum efficiency capacity. A scroll-type housing is usually used (Figure D-11).

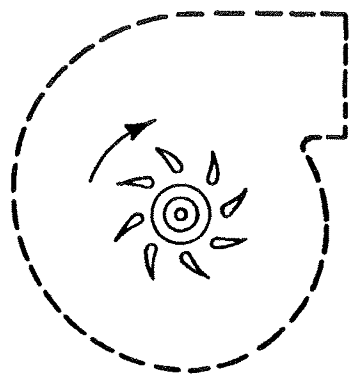

Figure D-11. Airfoil Fan.

Backward-curved fans. Fan blades are inclined in a direction opposite to the fan rotation (Figure D-12). Blades are single thickness, and fans consist of 10 to 16 blades. Fan efficiency is slightly lower than that of the airfoil fan. These fans travel at about twice the speed of the forward-curved fans. Normal selection range is usually 40 to 80 percent of wide open air flow. The static pressure proportion of the total pressure discharge is 70 percent, while the velocity pressure is 30 percent. For a given selection, the larger the fan, the more efficient it will be. Some advantages of the backward-curved fan are higher efficiency and nonoverloading characteristics. This type of blade allows material buildup and should only be used on clean air containing no condensable fumes or vapors. It is normally used for high capacity, high pressure applications where power savings may outweigh its higher first cost. Larger shaft and bearing sizes are required for higher speeds. Because of this, proper wheel balance is more important. Housing designs closely resemble those of the airfoil designs.

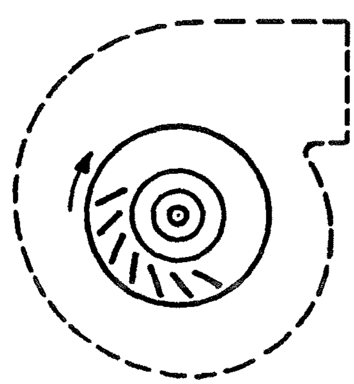

Figure D-12. Backward-Curved Fan.

Radial blade fans. Radial blade fans (Figure D-13) are used for systems handling materials likely to clog the fan wheel. These fans usually have medium tip speed and noise factor and are used for buffing exhaust, woodworking exhaust, or for applications where a heavy dust load passes through the fan. This type of blade is the simplest of all centrifugal fans and the least efficient. Horsepower rises with increasing air quantity in an almost directly proportional relation, which can lead to overloading. Fans usually include 6 to 10 blades, and the wheel is easily repaired. A scroll-type housing is usually used.

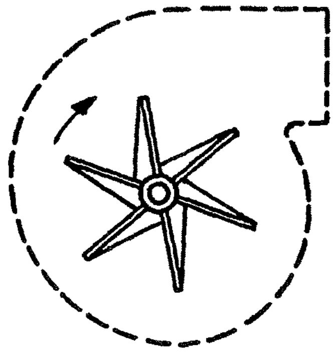

Figure D-13. Radial Blade Fan.

Forward-curved fans. The leading edges of these blades curve toward the direction of rotation. These fans usually consist of 24 to 64 shallow blades that have both the tip and heel curved forward. The efficiency of this fan is somewhat less than airfoil and backward-curved fans. Lightweight and low-cost construction, low space requirements, low tip speeds, and quiet operation are some common characteristics. Air leaves the wheel at a velocity greater than the wheel tip speed, and the primary energy is transferred to the air by use of high velocity in the wheel. The slow speed of this fan minimizes the shaft and bearing size, and it has a wide operating range, from 30 to 80 percent wide open volume. The static pressure proportion of the total pressure discharge is 20 percent, while the velocity pressure is 80 percent. Horsepower increases continuously with increasing air quantity. These fans are not recommended for fumes or dusts that would stick to the short curved blades because they would cause unbalance and would make cleaning difficult. These fans are typically used for producing high volumes at low static pressure. A scroll-type housing is usually used.

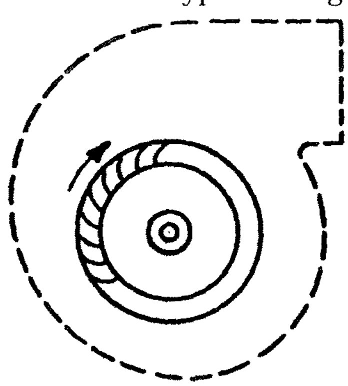

Figure D-14. Forward-Curved Fan.

Axial Fans

Axial flow fans consist of two or more blades. They are used for moving large quantities of air against a lower static pressure, and their common usage is for general ventilation. Pressure is produced from the change in velocity passing through the impeller, with none being produced by centrifugal force. Axial fan blades are divided into three types:

1. Propeller
2. Tubeaxial
3. Vaneaxial.

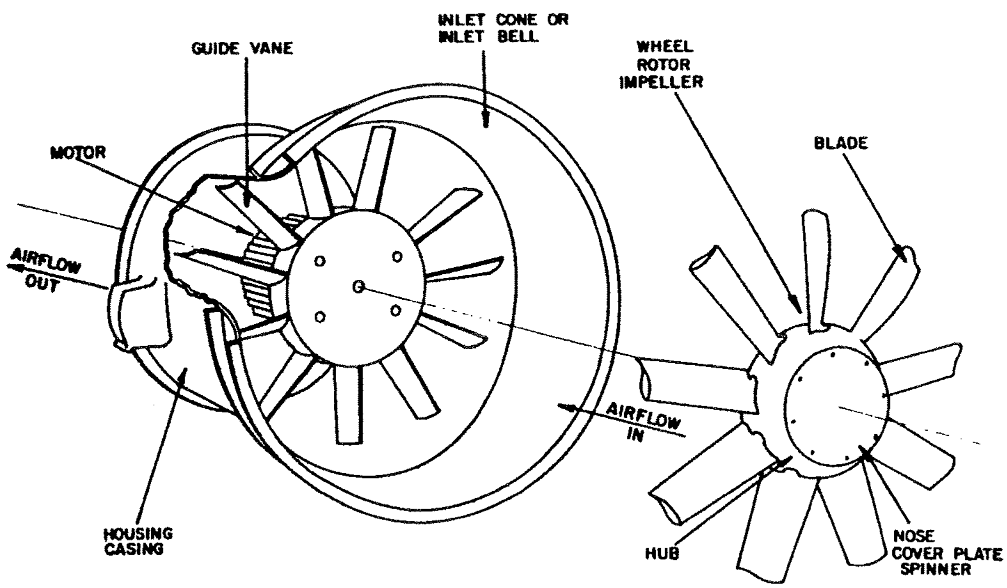

Figure D-15. Axial Fan Components.

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Propeller fans. Propeller fans are usually of inexpensive construction. Impellers are made of two or more blades, which are generally of single thickness and are attached to smaller hubs. Velocity pressure is the primary form of energy transfer. These fans work well in transferring high volumes of air at little or no static pressure differential. Propeller fans have the lowest efficiency of axial fans. Disc-propeller fans are used for moving clean air against no duct resistance. Housing generally consists of a circular ring, with the best performance coming from designs in which the housing is close to the blade tips (Figure D-16).

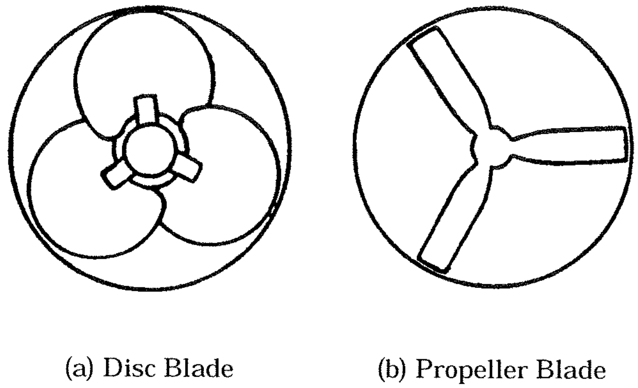


Figure D-16. Propeller Fans.

Environmental Systems Technology, W. D. Bevirt, 1984. Reprinted with permission of the National Environmental Balancing Bureau.

Tubeaxial fans. Tubeaxial fans (Figure D-17) consist of 4 to 8 blades, with the hub usually less than 50 percent of fan tip diameter. This type of fan is somewhat more efficient than the propeller fan design. It is best suited to moving air containing condensable fumes, pigments, and other materials that will collect on fan blades. Housing is a cylindrical type that has a close clearance between the tube and wheel tip.

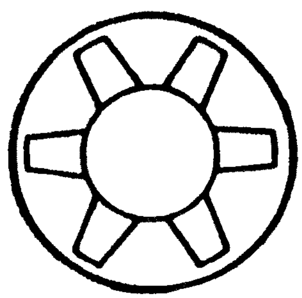


Figure D-17. Tubeaxial Fan.

Vaneaxial fans. Vaneaxial blades are adjustable, fixed, or controllable pitch types with the hub usually being greater than 50 percent of fan tip diameter. Vaneaxial fans (Figure D-18) have the highest efficiency of axial fans and can reach higher pressures. The operating range (cfm per fan) for axial fans is from 65 to 90 percent. The most efficient vaneaxial fans are those with airfoil blades, which should only be used with clean air. Vaneaxial fans are generally used for handling large volumes of air at low static pressures. Housing consists of a cylindrical tube that fits closely to the outer diameter of the blade tips.

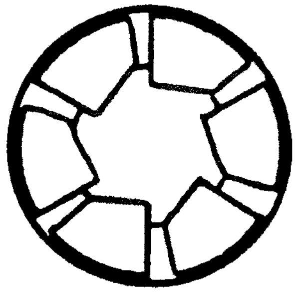


Figure D-18. Vaneaxial Fan.

Special Fan Types

Inline flow centrifugal fans. This type of fan (Figure D-19) has backward-curved blades and a special housing that permits a space-saving straight-line duct installation. The wheel is very similar to that of the airfoil. Space requirements are similar to a vaneaxial fan.

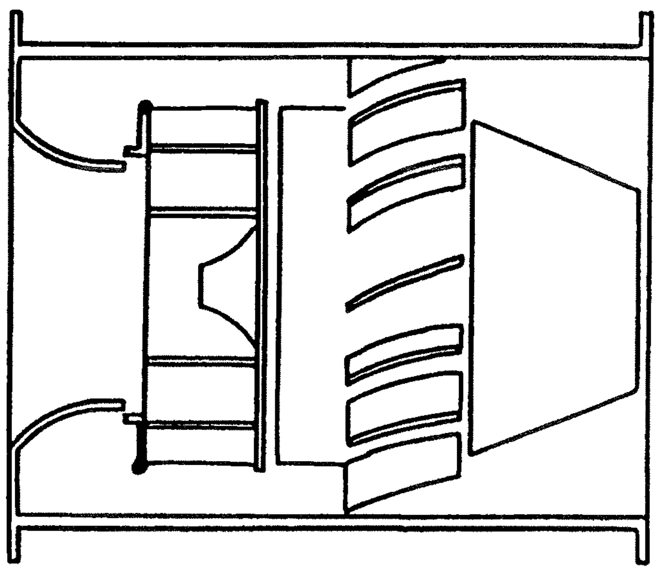
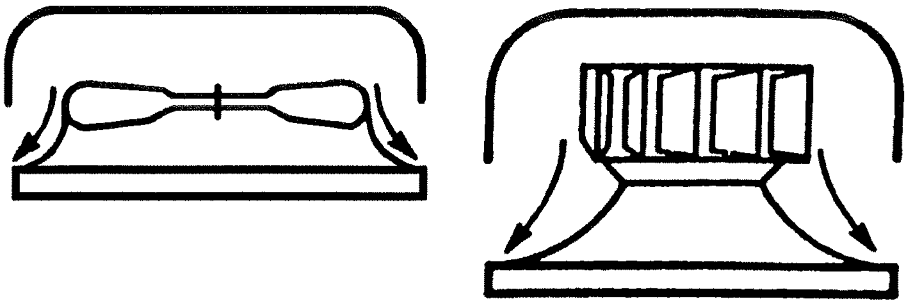


Figure D-19. Inline Flow Centrifugal Fan.

Power roof ventilators. The objective of these ventilators (Figure D-20) is to produce a high-volume flow rate at low pressure. They can be of centrifugal fan type or axial fan type.



(a) Axial Ventilator (b) Centrifugal Ventilator

Figure D-20. Power Roof Ventilators.

Adapted with permission from ASHRAE *1988 Equipment.*

Fan Classifications

Fan classifications are based on fan speeds and static pressures. "Class" refers to an AMCA standard that was developed to reflect operating conditions of the impellers, bearings, and housing of fans. Fan classifications vary with respect to impeller design type and other criteria. Figure D-21 shows an example of fan classifications.

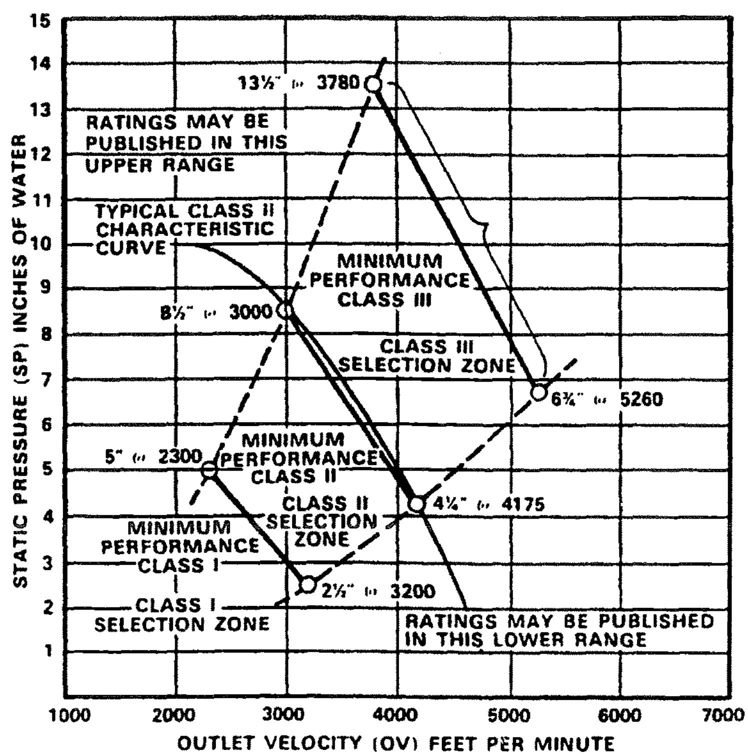


Figure D-21. Fan Class Standards.

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The published ratings of a fan should always be checked to make sure that revised operating conditions do not require a different class of fan. This type of change could also change the pressure classification of part or all of a duct system.

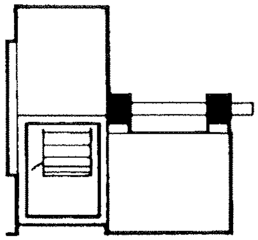
Fan Drives

Fan drive arrangements, which are standardized by AMCA, refer to the relation of the fan wheel to the bearings and the number of fan inlets. The fan drive may be belt or direct driven. First cost and space requirements are the major factors when selecting a suitable fan arrangement. A single inlet fan is about 30 percent taller than the double inlet type but only about 70 percent as wide for the same capacity. Double inlet fans are lower in cost in the larger sizes, while single inlet fans are usually less expensive in the smaller sizes. Drive arrangements designated by AMCA for centrifugal fans are shown in Figure D-22. Drive arrangements designated by AMCA for in-line fans are shown in Figure D-23.

Different motor locations for a belt-driven fan are shown in Figure D-24. This location is always determined by facing the drive side of the fan or blower and is independent of the discharge or rotation. Positions W and Z have the simplest fan base and belt guard construction.

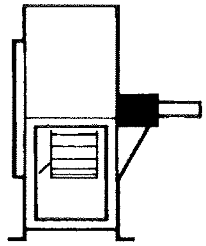
Rotation, clockwise or counter clockwise, is determined by the direction the fan wheel will be turning as viewed from the drive side of the fan (Figure D-25). The drive side of a single inlet fan is considered to be the side opposite the inlet, regardless of the actual drive location. When fans are to be inverted for ceiling suspension, the direction is determined when the fan is resting on the floor.

Most fans are driven at constant speed by constant speed motors and commonly deliver a constant air quality. Motors range from single phase and small fractional horsepower to large polyphase motors. The installed motor should be checked for sufficient starting torque to overcome the inertia of the fan wheel and drive package, and accelerate the fan to its design speed. A "V" belt is usually used to connect the motor to the driven fan. This belt also allows the synchronous speed of the motor to be converted to a lower, proper speed of the fan.



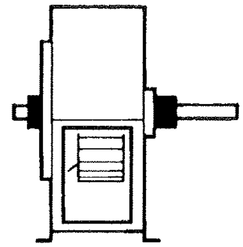
Arrangement 1 SWSI

For belt drive or direct connection, impeller overhung, two bearings on base.



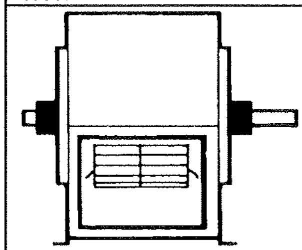
Arrangement 2 SWSI

For belt drive or direct connection, impeller overhung, bearings in bracket supported by fan housing.

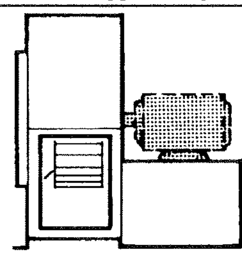


Arrangement 3 SWSI

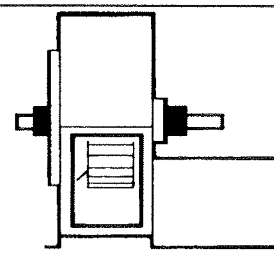
For belt drive or direct connection, one hearing on each side and supported by fan housing.



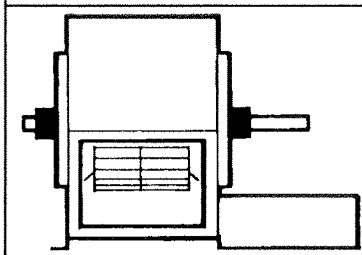
Arrangement 3 DWDI
For belt drive or direct connection, one hearing on each side and supported by fan housing.



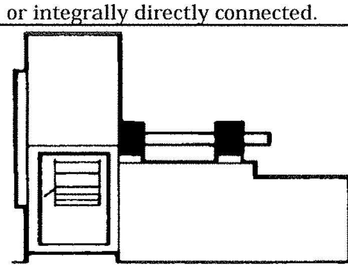
Arrangement 4 SWSI
For direct drive, impeller overhung on prime mover shaft. No hearings on fan. Prime mover base mounted or integrally directly connected.



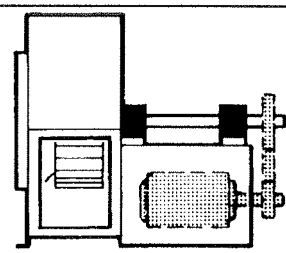
Arrangement 7 SWSI
For belt drive or direct connection. Arrangement 3 plus base for prime mover.



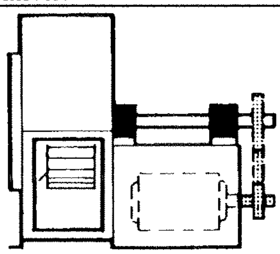
Arrangement 7 DWDI
For belt drive or direct connection, arrangement 3 plus base for prime mover.



Arrangement 8 SWSI
For belt drive or direct connection, arrangement 1 plus extended base for prime mover.



Arrangement 9 SWSI
For belt drive, impeller overhung, two bearings with prime mover outside base.



Arrangement 10 SWSI
For belt drive, impeller overhung. two bearing with prime mover inside base.

SW - Single Width DW - Double Width SI - Single Inlet DI - Double Inlet

Figure D-22. Drive Arrangements for Centrifugal Fans.

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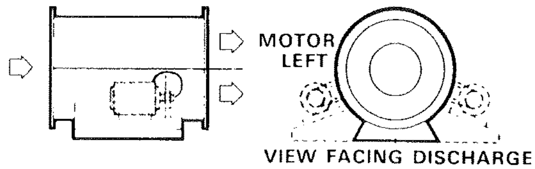
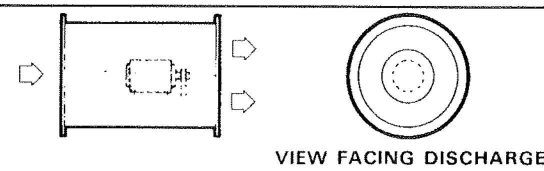


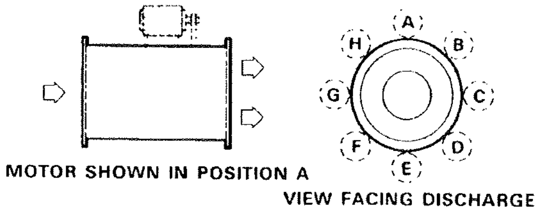
Arrangement 1

Belt drive with motor mounted independent of fan casing--typically used for motors too large for fan casing.



Arrangement 4

Direct drive with wheel overhung on motor shaft, can be used with vertical units.



Arrangement 9

Belt drive with motor located on periphery of casing in one of eight standard locations, can be used with vertical units.

Figure D-23. In-Line Fans.

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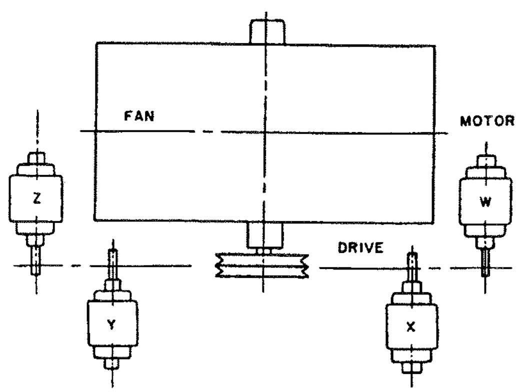

Figure D-24. Motor Positions.

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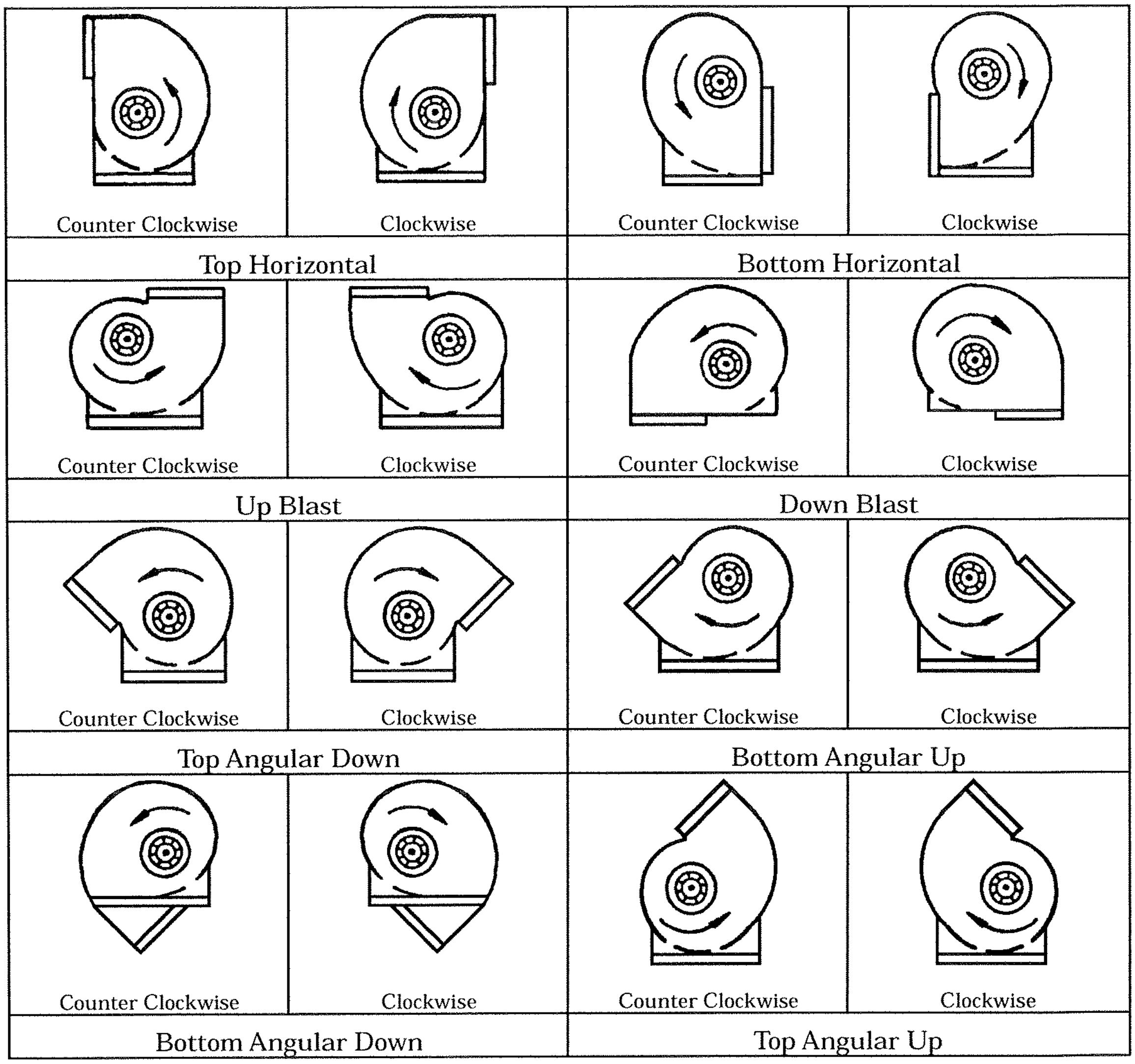

Figure D-25. Direction of Rotation and Discharge.

Reproduced with permission from the National Environmental Balancing Bureau, December 1996.

Some common conditions to consider in designing a satisfactory drive are:

1. Drives should be installed with provisions for center distance adjustment. This provision is important because all belts stretch.
2. Centers should not exceed 2-1/2 to 3 times the sum of the sheave diameters or be less than the diameter of the larger sheave.
3. The arc of contact on the smaller sheave should not be less than 120 degrees.
4. Sheave diameter ratios should not exceed 8:1.
5. Belt speed preferably should not exceed 5,000 ft/min, or be less than 1,000 ft/min—4,000 ft/min is the best practice.

6. Sheaves should be dynamically balanced for speeds in excess of 5,000 ft/min rim speed.

Some helpful points to watch for when installing drives are as follows:

1. Be sure that shafts are parallel and sheaves are in proper alignment. Check again after a few hours of operation.
2. Do not drive sheaves on or off shafts. Wipe shaft, key, and bore clean with oil. Tighten screws carefully. Recheck and retighten after a few hours of operation.
3. Belts should never be forced over sheaves.
4. In mounting belts, be sure the slack in each belt is on the same side of the drive. This side should be the slack side of the drive.
5. Belt tension should be reasonable. When in operation, the tight side of the belts should be in a straight line from sheave to sheave, and with a slight bow on the slack side. All drives should be inspected periodically to be sure belts are under proper tension and are not slipping.
6. When making replacements of multiple belts on a drive, be sure to replace the entire set with a new set of matched belts.

Fan Noise

One major cause of fan noise is surge. This is the result of periodic vibrations of the fan and ducts connected to it. It is caused by unstable operation of the fan. Surge commonly occurs when the actual static pressure is high, compared to the static pressure that the fan can reach at the particular speed at which it is operating. One way to check for surge is to relieve fan static pressure.

Another cause of fan noise is resonance. This will result in one or more sections of the duct system vibrating at the same frequency as a vibration produced by the fan. This can be checked by changing the fan speed by ±10 percent and noting whether the vibration stops.

If the fan performance is not matched to the duct system, fan noise will increase. One possibility is that the fan may be handling more air than required. Reducing the fan speed would reduce noise.

Air flow at the entrance and exit of a fan should be as smooth as possible to minimize the generation of turbulence. Conditions that produce turbulent air flow usually result in greater noise generation and increase static pressure drop in the system. The air flow on the outlet side of a fan is always turbulent for at least 3 to 6 duct diameters downstream. Fittings (such as elbows or sudden transitions) placed closer to the fan than this distance may result in noise problems. Figure D-26 shows some examples of good and bad fan outlet conditions.

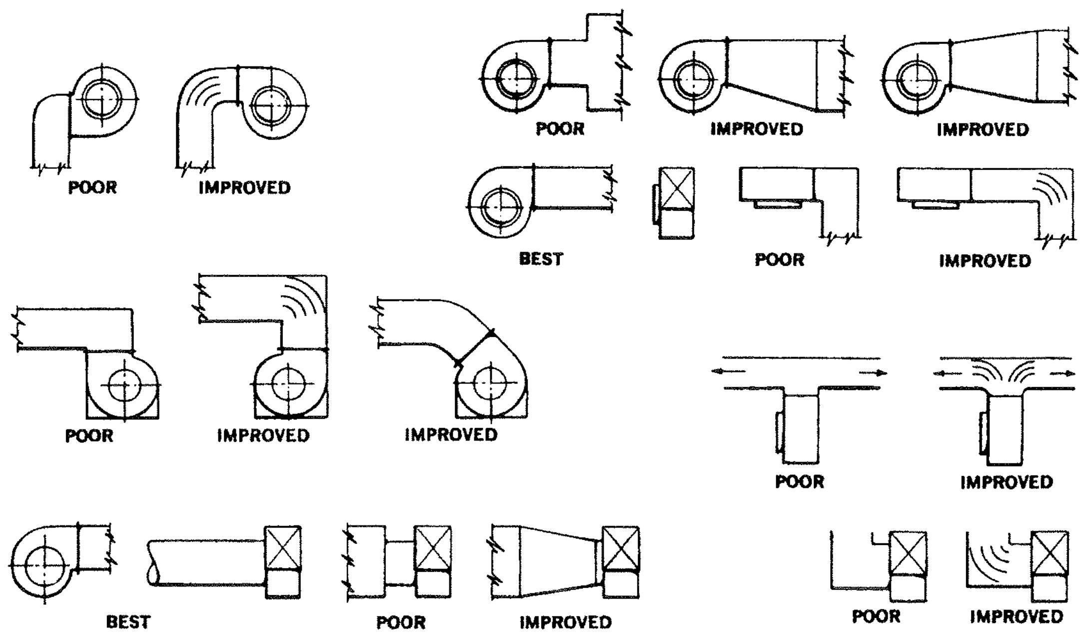

Figure D-26. Fan Outlet Conditions.

SMACNA, *HVAC Systems - Testing, Adjusting & Balance*, 2nd Ed., 1993. Used with permission.

Flexible connectors should be used on fans at each duct connection (Figure D-27). These connectors should not be pulled taut, but should be long enough to provide folds or flexibility when the fan is off.

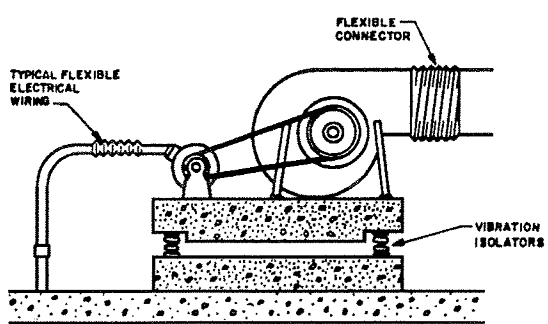

Figure D-27. Flexible Connections.

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Steel springs are usually used as noise isolators on fans to absorb their vibrations. Concrete bases are also used with fans because they can reduce the amplitude of oscillation of the equipment. This reduction, in turn, reduces the transmission of vibration through connected piping and duct.

Fan Selection

The following information is necessary before proceeding with any fan selection:

1. *Volume Required*—The first step to consider in selecting a fan or ventilator is the total cfm of the space. This can be accomplished in two different ways.

   a. *Air change method*—This method involves calculating the required number of air changes necessary to give the proper ventilation for a given space, and the total cubic feet of air space of the building. Air changes required must conform to the local health department code covering the type of installation. If no local codes are to be met, tables of air changes can be used.

$$cfm = \frac{building\ volume\ in\ cubic\ feet}{\min/air\ change} \qquad \text{[Eq D-6]}$$

   b. *Heat removal method*—The average outside temperature, desired inside temperature, and Btu per minute are required to use this method. This formula gives the amount of air to be passed through the building to maintain desired inside temperature. The cfm deals primarily with heat, which changes the temperature of the substance involved, or sensible heat. It can be applied to installations where any general ventilation of a heat problem is desired.

$$cfm = \frac{total\ Btu\ per\ \min}{0.018 \times temperature\ rise\ ^\circ F} \qquad \text{[Eq D-7]}$$

2. *Fan Static Pressure*—The fan total pressure less the fan velocity pressure.
3. *Type of Material Handled Through Fan*—These include explosive fumes, general ventilation, fibrous material (heavy dust load), removal of heat, or corrosive fumes.
4. *Direct or Belt Driven*—Direct-driven exhausts offer a more compact assembly, and assure constant fan speed. They eliminate belt slippage that occurs when belt-driven drives are not maintained. Fan speeds are limited to available motor speeds. Belt-driven drives are often preferred because quick

change in fan speed is commonly required. This capability will provide for increases in system capacity or pressure requirements due to changes in process, hood design, equipment location, or air cleaning equipment.

5. *Noise*—This is not as important in industrial exhaust situations.
6. *Operating Temperature*—Sleeve bearings are suitable to 250 °F, and ball bearings can be used up to 550 °F. Special cooling devices are required at higher temperatures.
7. *Efficiency*—Select a fan size that will handle the required volume and pressure with minimum horsepower.
8. *Space Limitations*.

6 Ducts

Air conveyed by a duct will impose two loads on the duct's structure. These loads are air pressure and velocity. A duct is a structural assembly, and its optimum construction depends on the maximum loads imposed on it. Usually, duct strength, deflection, and leakage are more functions of pressure rather than velocity.

Static pressure at specific points in an air distribution system is not necessarily the static pressure rating of the fan. Because total pressure decreases in the direction of flow, a duct construction pressure classification equal to the fan outlet pressure (or to the fan total static pressure rating) cannot economically be imposed on the entire duct system. Figure D-28 shows examples of static pressure identification. The static pressure rating changes are shown by "flags" at each point where the duct static pressure classification changes, with the number on the flag indicating the pressure class of the ductwork on each side of the dividing line.

Rectangular Duct Construction

Rectangular duct construction standards provide options for constructing ducts. These include ducts unreinforced and joined by flat type connections only, those joined by flat type joint connectors backed by a qualified reinforcement, those joined by an upright connector that meets reinforcement requirements alone or in conjunction with an incorporated reinforcement, and, in sizes over 48 in. width, those using tie rods that permit the use of smaller reinforcements. Not all options exist at all sizes and all static pressure classes.

Duct construction tables define relationships between static pressure, width, wall thickness, reinforcement spacing, and reinforcement strength so that ducts have adequate strength and acceptable deflection limits. The greater dimension of a duct determines the duct gauge for all four sides. This applies to both reinforced and unreinforced ducts.

The first step in determining construction requirements is to locate the table with the applicable static pressure. The tables that follow can be found in the Sheet Metal and Air Conditioning Contractors National Association, Inc.'s (SMACNA) HVAC Duct Construction Standards. SMACNA provides a number of tables in both U.S. and metric units for different pressure classes. This discussion will refer to the 1 in. w.g. pressure class table in the ensuing paragraphs.

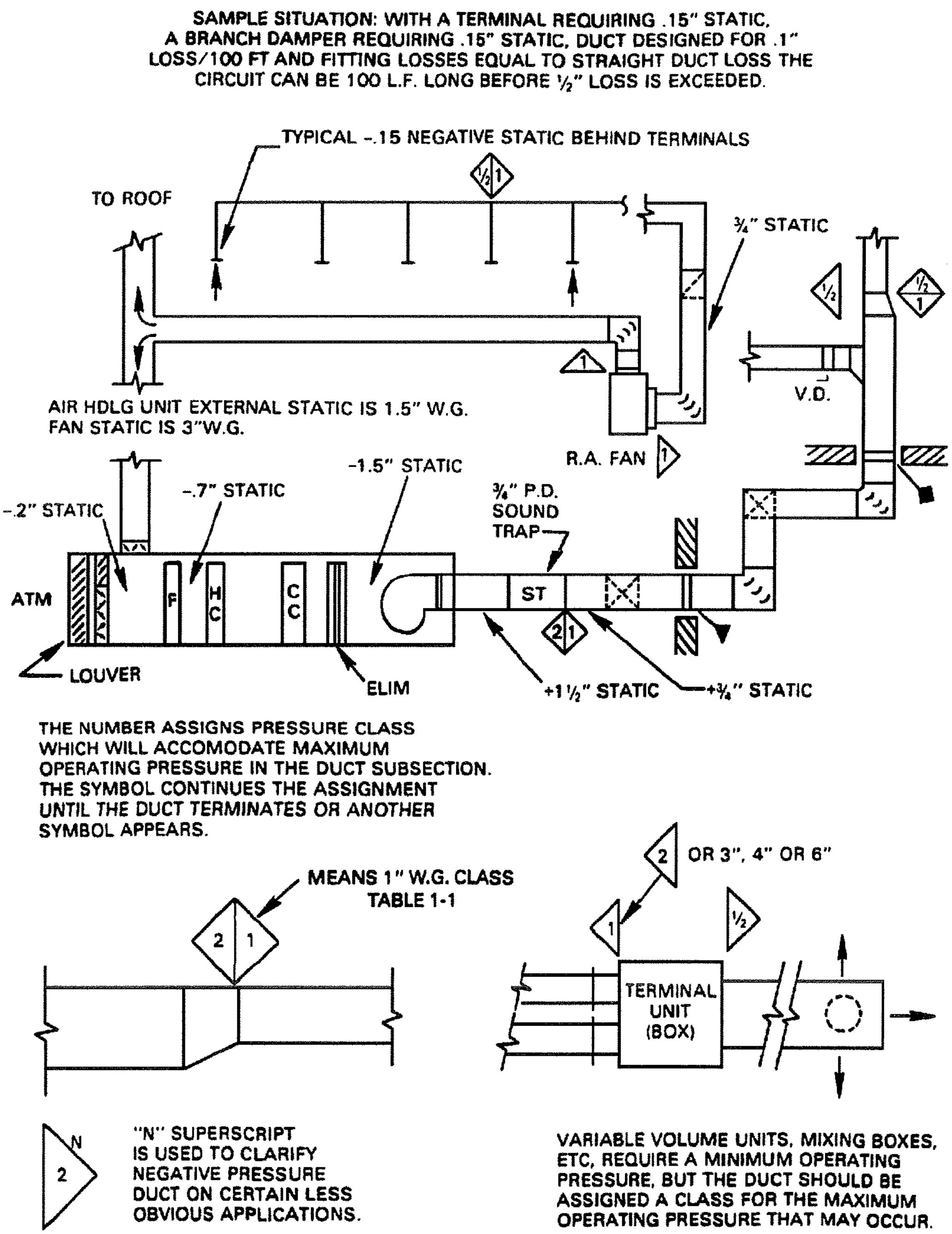

Figure D-28. Duct Pressure Class Designation.

SMACNA - *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

Duct Reinforcing

To find the minimum gauge of the duct and the minimum grade of reinforcement in Table D-10, determine the duct dimension in column 1 and go to the appropriate reinforcement spacing in columns 3 through 10. The minimum grade of reinforcement is given by the alphabet letter, and the minimum gauge of the duct is shown numerically. Each column is an alternative construction available for selection.

The duct side with the greater dimension is investigated first because this side dictates the duct gauge. Then the lesser duct dimension is found in column 1, and the duct gauge used for the wide side is located on the same line. If the duct gauge is in column 2, no reinforcement is required on that side; otherwise, the minimum reinforcement code is the alphabet letter listed under the spacing actually used. The actual duct gauge may occur in a column giving an allowable spacing greater than will be used. In such a case, the minimum reinforcement grade is that associated with the actual spacing.

Transverse Joint and Intermediate Reinforcement

The reinforcement spacing in Table D-10 denotes the distance between two joints or two intermediate reinforcements, or from a joint to an intermediate member. Any joint or reinforcement member having a corresponding letter code like those in Tables D-11 and D-12 may be used as shown in the examples below for various duct sizes (Examples 1 through 3).

The letter coding for reinforcement corresponds to a stiffness index number (EI) that is the modulus of elasticity multiplied by a moment of inertia that is appropriately based on contributing elements of the connector, the reinforcement, the duct wall, or combinations therein.

Example 1 — 18 x 12 in. duct

For a duct fabricated out of 22 ga sheet metal, column 2 of Table D-10 shows that it may be unreinforced.

If the duct is of 24 ga, the 12-in. side is unreinforced, while grade B joints are required at 10 ft minimum spacing on the 18-in. sides. Also, Table D-12 allows the T-1 drive slip to be used on the 18-in. sides. Any joint used on the 18-in. side must meet grade B regardless of joint spacing.

1" W.G. STATIC POS.OR NEG.	TABLE 1-4 RECTANGULAR DUCT REINFORCEMENT								
	NO REINFORCE-MENT REQUIRED	REINFORCEMENT CODE FOR DUCT GAGE NO.							
		REINFORCEMENT SPACING OPTIONS							
DUCT DIMENSION		10'	8'	6'	5'	4'	3'	21/2'	2'
1	2	3	4	5	6	7	8	9	10
10"dn	26 ga.	NOT REQUIRED							
11, 12"	26 ga.								
13, 14"	24 ga.	B-26	B-26	B-26	B-26	B-26	A-26	A-26	A-26
15, 16"	22 ga.	B-24	B-26	B-26	B-26	B-26	B-26	B-26	A-26
17, 18"	22 ga.	B-24	B-26	B-26	B-26	B-26	B-26	B-26	B-26
19, 20"	20 ga.	C-24	C-26	C-26	C-26	C-26	B-26	B-26	B-26
21, 22"	18 ga.	C-24	C-24	C-26	C-26	C-26	B-26	B-26	B-26
23, 24"	18 ga.	C-24	C-24	C-26	C-26	C-26	C-26	B-26	B-26
25, 26"	18 ga.	D-22	D-24	C-26	C-26	C-26	C-26	C-26	B-26
27, 28"	16 ga.	D-22	D-24	D-26	C-26	C-26	C-26	C-26	C-26
29, 30"	16 ga.	E-22	D-24	D-26	D-26	C-26	C-26	C-26	C-26
31-36"	NOT DESIGNED	E-20	E-22	E-24	D-24	D-26	C-26	C-26	C-26
37-42"		F-18	F-20	E-22	E-24	E-26	D-26	D-26	C-26
43-48"		G-16	G-18	F-20	F-22	E-24	E-26	E-26	D-26
49-54"		H-16	H-18	G-20	F-22	F-24	E-24	E-24	E-24
55-60"			H-18	G-20	G-22	F-24	F-24	E-24	E-24
61-72"				H-18G	H-18G	H-22G	F-24	F-24	F-24
73-84"				I-16G	I-18G	I-20G	H-22G	H-22G	G-22
85-96"					I-16H	I-18H	I-20G	H-20G	H-22G
97-108"						I-18G	I-18G	I-18G	I-18G
109-120"							I-18H	I-18H	I-18G

See page 1-15. Circles in the Table denotes only column numbers. For column 2, *see* Fig. 1-7. For columns 3 through 9, *see* Introduction to Schedules. The number in the box is minimum duct gage; the alphabet letter is the minimum reinforcement grade for joints and intermediates occurring at a maximum spacing interval in the column heading. A letter to the right of the gage gives a tie rodded reinforcement alternative. A "t" compels use of tie rod(s) for the reinforcement listing. For beading or crossbreaking, *see* Fig. 1-8.

Table D-10. Rectangular Duct Reinforcement.

SMACNA *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

TABLE 1-24 UNREINFORCED DUCT (WALL THICKNESS)							
DUCT DIMENSION	PRESSURE CLASS (In W.G.)						
	Positive or Negative						
	1/2"	1"	2"	3"	4"	6"	10"
8" dn	26	26	26	24	24	24	22
9, 10"	26	26	26	24	22	20	18
11, 12"	26	26	24	22	20	18	16
13, 14"	26	24	22	20	18	18	
15, 16"	26	22	20	18	18	16	
17. 18"	26	22	20	18	16		
19, 20"	24	20	18	16			
21, 22"	22	18	16	16			
23, 24"	22	18	16	16			
25, 26"	20	18					
27, 28"	18	16			Reinforcement is required.		
29, 30"	18	16					
31-36"	16						

This table gives minimum duct wall thickness (gage) for use of flat type joint systems. Plain S and hemmed S connectors are limited to 2" w.g. maximum. Slips and drives must not be less than two gages lighter then the duct wall nor below 24 gage. Double S slips must be 24 gage for ducts 30" wide or less and 22 gage for greater width.

Duct Gage	26 to 22	20	18	16
Minimum Flat Slip and Drive Gage	24	22	20	18

See Figure 1-7 for joint types.

Table D-11. Unreinforced Duct.

SMACNA *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

If the duct is of 26 gauge, the 12-in. side is unreinforced, but the 18-in. side has a maximum reinforcement spacing of 8 ft and the minimum size is grade B. Table D-12 allows the T-1 drive slip to be acceptable (up to 20 in. width and 8 ft spacing).

Example 2 — 30 x 18 in. duct

The choices for the 30-in. side are: 16 ga for unreinforced; grade E on 22 ga at 10 ft or D on 24 ga at 8 ft. For the 18-in. side, the choices are the same as outlined in Example 1 for 18-in. width.

TABLE 1-25 T-1 FLAT DRIVE ACCEPTED AS REINFORCEMENT								
DUCT WALL	**26 ga**		**24 ga**		**22 ga**		**20 ga or Heavier**	
Static Pressure	**Maximum Duct Width (W) and Maximum Reinforcement Spacing (S)**							
	W	S	W	S	W	S	W	S
1/2" w.g.	20" 18"	10' N.R.	20"	N.R.	20"	N.R.	20"	N.R.
1" w.g.	20" 14" 12"	8' 10' N.R.	20" 14"	8' N.R.	20" 18"	10' N.R.	20"	N.R.
2" w.g.	18"	5'	18" 12"	8' N.R.	18" 14"	10' N.R.	18"	N.R.
3" w.g.	12" 10"	5' 6'	18" 10"	5' N.R.	18" 12"	5' N.R.	18" 14"	6' N.R.
4" w.g.	Not Accepted		16" 8"	5' N.R.	12" 8"	6' N.R.	12"	N.R.
6" w.g.			12" 8"	5' N.R.	12" 8"	5' N.R.	12" 10"	6' N.R.
Although the flat drive slip T-1 does not satisfy the EI calculation requirements for Classes A, B or C reinforcement, tests predict its suitability for use as reinforcement within the limits of the table.								

N.R. --No reinforcement is required; however, the T-1 Joint may be used.

Table D-12. T-1 Flat Drive Accepted as Reinforcement.

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Example 3 — 54 x 30 in. duct, 5-ft joint spacing preselected

For 54-in. width, grade F at 5 ft is required if 22 ga is selected. A 24-ga duct may be used, but with 4-ft joint spacing).

For the 30-in. side, grade E is required (for 10 ft maximum spacing) on any duct gauge less than 16.

Duct Materials

A variety of materials can be used in the construction of ducts. Selection of the materials used throughout the duct system should be given careful consideration. Consideration must also be given to selection of duct components other than those materials used for the duct walls. Such items include duct liners, pressure sensitive tapes, sealants, adhesives, reinforcements, hangers, etc.

Materials and the construction of exhaust ductwork and fans depends on:

1. Nature of the hood effluents
2. Surrounding temperature
3. Lengths and arrangement of duct runs
4. Flame and smoke spread rating
5. Duct velocities and pressures.

Both present and future effluents should be evaluated when selecting duct materials and construction. These effluents may be classified as organic or inorganic chemical gases, vapors, fumes, or smokes. Exhaust system fans, ducts, and coatings can be damaged by these effluents through corrosion, dissolution, and melting.

The condensation of vapors in the exhaust system is affected by the surrounding temperature of the space in which the ductwork and fans are located. Condensation contributes to the corrosion of metals with or without the presence of chemicals.

When duct runs are short and direct, and when the air is maintained at reasonable (higher) velocities, the chance of attack by effluents is less. The longer the duct, the longer will be the period of exposure to effluents and the greater the degree of condensation. Horizontal runs provide surfaces where moisture can remain longer than it may on vertical runs. If condensation is probable, sloped ductwork and condensate drains should be provided.

Fan operation may be continuous or intermittent. Intermittent fan operation allows longer periods of wetness because of condensation.

Following is a list of duct materials and their characteristics:

- *Galvanized Steel* - Widely used as a duct material for most air handling systems; not recommended for corrosive product handling or temperatures above 400 °F. Advantages include high strength, rigidity, durability, rust resistance, availability, nonporosity, workability, and weldability. Galvanized sheets with the surface treated for painting are commonly used.

- *Carbon Steel (Black Iron)* - Applications include flues, stacks, hoods, other high temperature duct systems, kitchen exhaust systems, and ducts requiring paint or special coating. Advantages include high strength, rigidity, durability, availability, paintability, weldability, and nonporosity. Some limiting characteristics are corrosion resistance and weight.

- *Aluminum* - Aluminum can be used in duct systems for moisture laden air, louvers, special exhaust systems, ornamental duct systems, and is often substituted for galvanized steel in HVAC duct systems. Some advantages include weight, resistance to moisture corrosion, and availability. Limiting characteristics include low strength, material cost, weldability, and thermal expansion.

- *Stainless Steel* - Used in duct systems for kitchen exhaust, moisture laden air, and fume exhaust. Advantages include high resistance to corrosion from moisture and most chemicals and the ability to take a high polish. Limiting characteristics include labor and material costs, workability, and availability.

- *Copper* - Copper applications include duct systems exposed to outside elements and moisture laden air, certain chemical exhaust, and ornamental ductwork. Advantages are durability and corrosion resistance and that it accepts solder readily and is nonmagnetic. Limiting characteristics are cost, ductility, electrolysis, thermal expansion, and stains.

- *Fiberglass Reinforced Plastic (FRP)* - Applications include chemical exhaust, scrubbers, and underground duct systems. Resistance to corrosion and ease of modification are advantages of FRP. Limiting characteristics include cost, weight, range of chemical and physical properties, brittleness, fabrication (necessity of molds and expertise in mixing basic materials), and code acceptance.

- *Polyvinyl Chloride (PVC)* - Applications are exhaust systems for chemical fumes and hospitals, and underground duct systems. Advantages include resistance to corrosion, weight, weldability, and ease of modification. Limiting characteristics include cost, fabrication, code acceptance, thermal shock, and weight.

- *Polyvinyl Steel (PVS)* - Applications include underground duct systems, moisture laden air, and corrosive air systems. Some advantages are resistance to corrosion, weight, workability, fabrication, and rigidity. Some limiting characteristics include temperature limitations (250 °F maximum), weldability, code acceptance, and susceptibility to coating damage.

- *Concrete* - Concrete can be used for underground ducts and air shafts. Advantages include compressive strength and corrosion resistance. Cost, weight, porosity, and fabrication (requires forming processes) are some limiting characteristics.

- *Rigid Fibrous Glass* - Most widely used in interior HVAC low pressure duct systems. Advantages include weight, thermal insulation and vapor barrier, acoustical qualities, ease of modification, and inexpensive tooling for fabrication. Limiting characteristics include cost, susceptibility to damage, system pressure, and code acceptance.

- *Sheetrock* - Applications include ceiling plenums, corridor ducts, and air shafts. Cost and availability are advantages, while weight, code acceptance, and leakage are limiting characteristics.

7 Air Cleaners

Air cleaning devices remove contaminants from an air or gas stream. They are available in a wide range of designs to meet variations in air cleaning requirements. The degree of removal required, quantity and characteristics of the contaminant to be removed, and conditions of the air or gas stream will all have a bearing on the device selected for any given application. Air cleaning equipment is usually selected to:

1. Conform to Federal, state, or local emission standards and regulations.
2. Prevent reentrainment of contaminants to work areas where they may become a health or safety hazard.
3. Reclaim usable materials.
4. Permit cleaned air to recirculate to work spaces and/or processes.
5. Prevent physically damaging adjacent property.

For particulate contaminants, air cleaning devices are divided into two basic groups: air filters and dust collectors.

Air Filters

Air filters are designed to remove dust concentrations of the order found in outside air, and are used in ventilation, air conditioning, and heating systems where dust loading seldom exceed one grain per thousand cubic feet of air and is usually well below 0.1 grains per thousand feet of air.* All of the common types of air filters fall into three broad categories: (1) fibrous media, (2) renewable media, and (3) electronic air cleaners.

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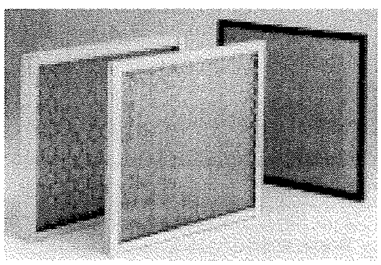
* A grain is a unit of weight measure and is equivalent to 1/7000 of a pound.

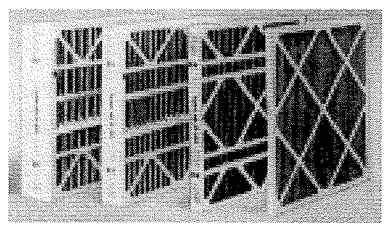
Fibrous media unit filters accumulate dust loads that cause pressure drop to increase up to some maximum permissible value. During this period of increase, efficiency also increases. At high dust loads, however, dust may adhere poorly to the filter's fibers, causing efficiency to drop. Filters in this condition should be replaced or reconditioned. This category includes viscous impingement and dry type air filters.

Another category is the renewable media filter in which fresh media is introduced into the air stream to maintain nearly constant resistance. These filters maintain nearly constant efficiency.

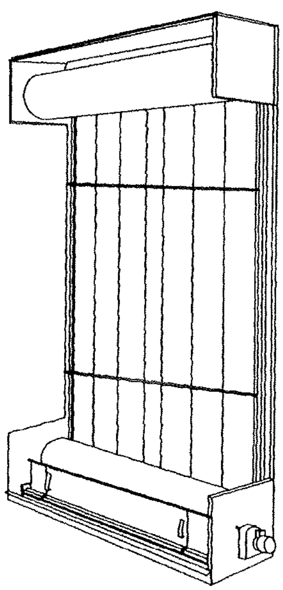
Electronic air cleaners have essentially constant pressure drop and efficiency, unless their precipitating elements become severely dust loaded. Figure D-29 shows four basic types of air filters: dry mat, pleated, roll type, and electrostatic.



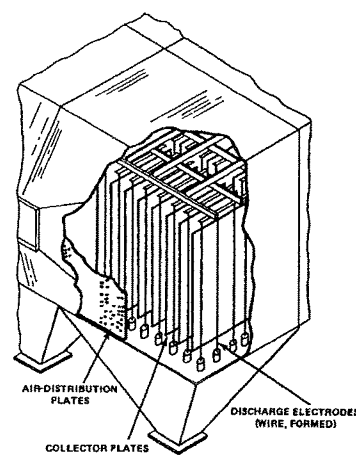
(a) Dry Mat



(b) Pleated



(c) Roll Type



(d) Electrostatic

Figure D-29. Air Filter Types.

Source for a-c: *NAFA Guide to Air Filtration,* 1993, used with permission.
Source for d: 1996 *ASHRAE Systems and Equipment Handbook.*

Manufacturers' ratings should be used for the type of filtering medium selected. Filters are usually selected for ease of maintenance and to provide the highest degree of air cleanliness feasible or required by the installation.

Dust Collectors

Dust collectors are usually designed for the much heavier loads from work shops or industrial processes where the air or gas to be cleaned originates in local systems or process stack gas effluents. Loading will vary from less than 0.1 to 20 grains or more per cubic foot. Therefore, dust collectors are and must be capable of handling concentrations some 100 to 20,000 times greater than air filters.

Dust collection equipment is available in numerous designs using a number of principles and featuring wide variation in effectiveness, first cost, operating and maintenance cost, space, arrangement, materials, and construction. Consultation with the equipment manufacturer is the recommended procedure in selecting a collector for any problem where extensive previous plant experience on the specific dust problem is not available. Factors influencing equipment selection include:

- Concentration and particle size of contaminant
- Degree of collection required
- Characteristics of air or gas stream
- Characteristics of contaminant
- Energy requirements
- Method of dust disposal.

The five basic types of dust collectors available are electrostatic precipitators, fabric filter, unit collector, wet collector, and dry centrifugal collector.

Electro-Static Precipitators

In electrostatic precipitation, a high potential electric field is established between discharge and collecting electrodes of opposite polarity. The discharge electrode is of small cross-sectional area, such as a wire or piece of flat stock, and the collection electrode is large in surface area, such as a plate.

The stream to be cleaned passes through an electrical field that develops between the electrodes. At a critical voltage, the molecules are separated into positive and negative ions. This is called "ionization" and takes place at or near the surface of the discharge electrode. Ions, having the same polarity as the discharge electrode, attach themselves to neutral particles in the gas stream as they flow through the precipitator. They are then attracted to the collecting plate, which is of opposite polarity. Upon contact with the collecting surface, dust particles lose their charge and can be easily removed by vibration, washing, or by gravity (Figure D-30).

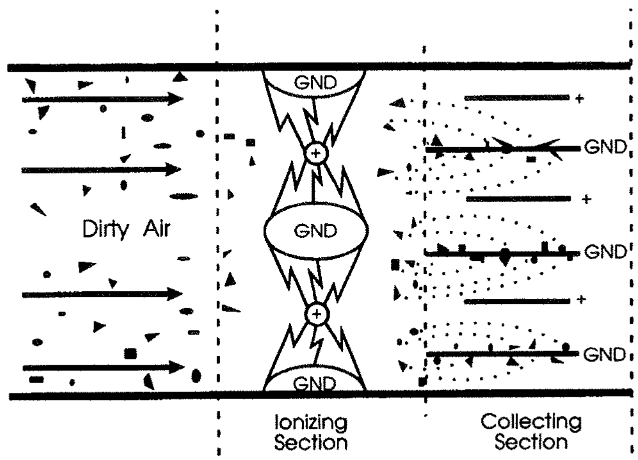


Figure D-30. Electrostatic Precipitator.

Source: *NAFA Guide to Air Filtration,* 1993. Used with permission.

The electrostatic process consists of:

1. Ionization of the gas
2. Charging the dust particles
3. Transportation of the particles to the collecting surface
4. Neutralization, or removing the charge from the dust particle
5. Removal of the particle from the collection surface.

Fabric Filter Collectors

Fabric filter collectors remove particulates from carrier gas streams by interception, impaction, and diffusion mechanisms. The fabric may be constructed of a variety of materials, and may be woven or non-woven. A heavy non-woven fabric is more efficient than a woven fabric since the void areas or pores in the felted fabric are smaller. Fabric collectors are not 100 percent efficient, but well-designed, adequately sized, and properly operated fabric collectors can be expected to operate at efficiencies in excess of 99 percent. Commercially available fabric collectors have fabric configured as tubes or stockings, envelopes, or pleated cartridges.

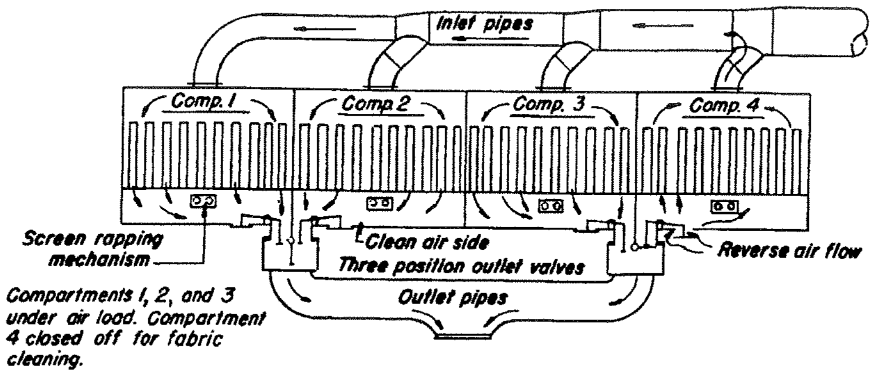


Figure D-31. Multiple-Section, Continuous-Duty, Automatic Fabric Collector.

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Unit Collectors

Unit collectors (Figure D-32) have small fabric filters and capacities in the range of 200 to 2000 cfm. They have integral air movers, small space requirements, and simplicity of installation. In most applications, cleaned air is recirculated although discharge ductwork may be used if the added resistance is within the capability of the air mover.

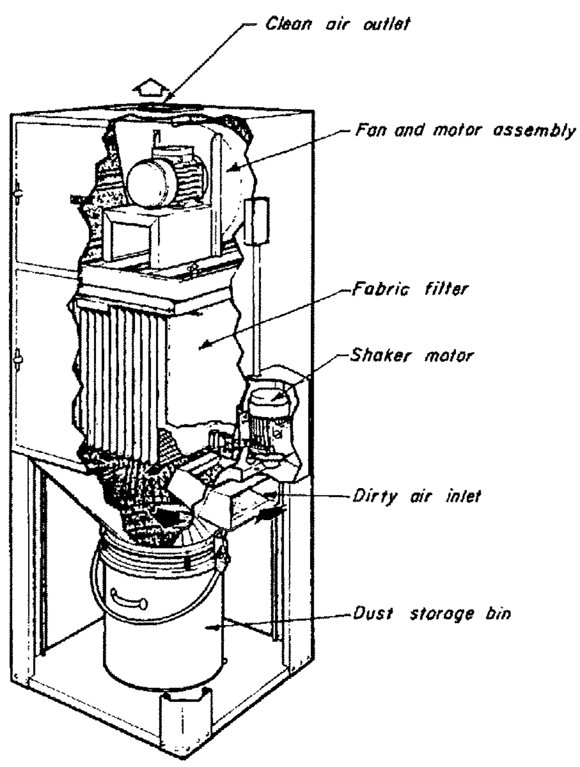

Figure D-32. Unit Collector.

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Unit collectors are often used in the metal working industry to fill the need for dust collection from isolated, portable, intermittently used, or frequently relocated dust producing operations. Typically, a single collector serves a single dust producing operation with the energy saving advantage that the collector need operate only when the dust producing machine is in operation.

Wet Collectors

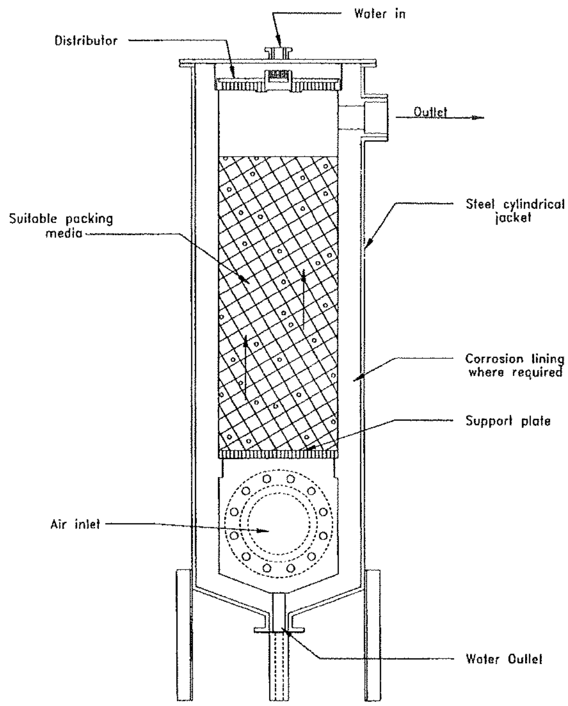
Wet collectors are commercially available in many different designs. These collectors have the ability to handle high temperature and moisture-laden gases. Wet collectors have one characteristic not found in other collectors—their ability to humidify. Humidification, the process of adding water vapor to the air stream through evaporation, may be either advantageous or disadvantageous depending on the situation.

Chamber or spray tower collectors. These consist of a round or rectangular chamber into which water is introduced via spray nozzles. The principal mechanism of these collectors is impaction of dust particles on the liquid droplets created by the nozzles. These droplets are separated from the air stream by centrifugal force or impingement on water eliminators.

Packed tower collectors. These collectors are essentially contact beds through which gases and liquid pass either concurrently, counter-currently, or in cross-flow. They are used primarily for applications involving gas, vapor, and mist removal (Figure D-33).

Wet centrifugal collectors. Wet centrifugal collectors comprise a large portion of the commercially available designs. This type uses centrifugal force to accelerate the dust particle and impinge it upon a wetted collector surface (Figure D-34).

Wet dynamic precipitators. These use water sprays within a fan housing, and obtain precipitation of the dust particles on the wetted surfaces of an impeller with a special fan blade shape (Figure D-35).



PACKED TOWER

Figure D-33. Packed Tower Collector.

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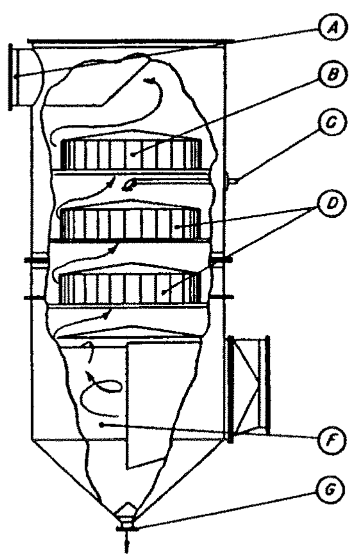

Symbols	Parts
A	*Clean air outlet.*
B	*Entrainment separator.*
C	*Water inlet.*
D	*Impingement plates.*
E	*Dirty air inlet.*
F	*Wet cyclone for collecting heavy material.*
G	*Water and sludge drain.*

Figure D-34. Wet Centrifugal Collectors.

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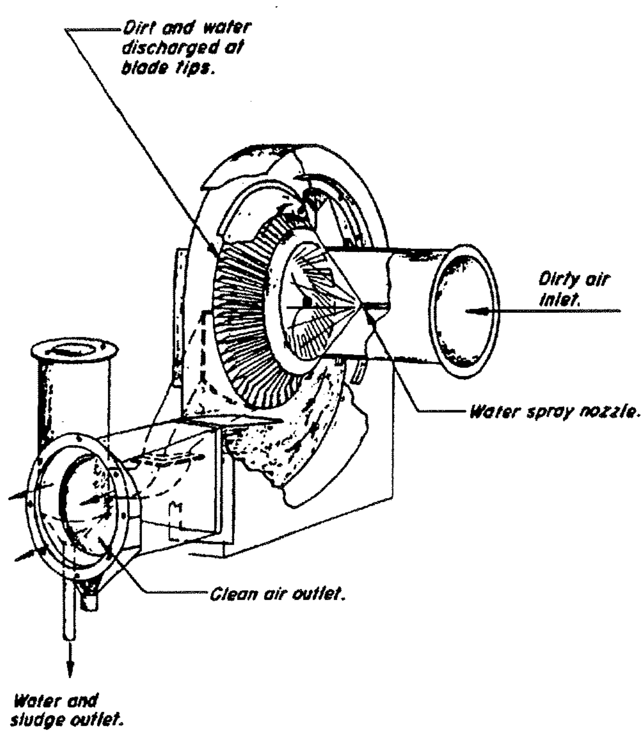

Figure D-35. Wet Dynamic Precipitator.

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Orifice type collectors. Orifice type collectors bring the air flow in contact with a sheet of water in a restricted passage. Water flow may be induced by the velocity of the air stream or maintained by pumps and weirs.

Venturi collectors. Venturi collectors use a venturi-shaped constriction to establish throat velocities considerably higher than those experienced with the orifice type. The collection mechanism of the venturi is impaction.

Dry Centrifugal Collectors

Dry centrifugal collectors can be divided into two basic groups categorized by their effectiveness in removal of smaller dust particles.

Cyclone collectors. The cyclone collector is commonly applied for the removal of coarse dusts from an air stream, as a pre-cleaner to more efficient dry or wet dust collectors, and/or as a separator in product conveying systems using an air stream to transport material. Its principal advantages are low cost, low maintenance, and low pressure drop, but it cannot be used for collection of fine particles.

High efficiency centrifugal collectors. In these collectors, higher centrifugal forces are exerted on dust particles in a gas stream. Centrifugal force is a function of peripheral velocities and angular acceleration. Improvement in dust separation efficiency has been obtained by increasing velocities through a cyclone shaped collector using a skimmer or other design feature, with a number of small diameter cyclones in parallel, and placing units in series in some unusual applications (Figure D-36).

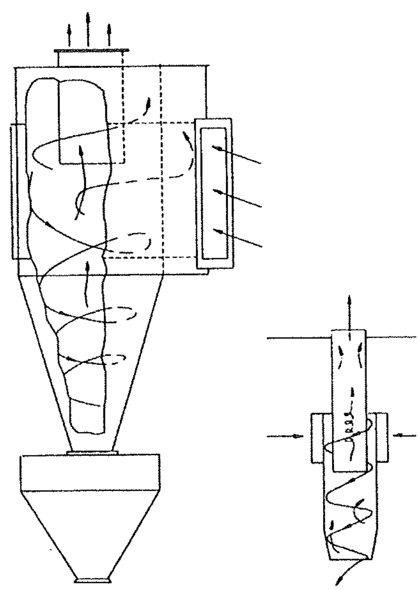

Figure D-36. High Efficiency Centrifugal Collector.

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8 Exhaust Hoods

A hood can be defined as "a ventilated enclosed work space intended to capture, contain, and exhaust fumes, vapors, and particulate matter generated inside the enclosure. It consists basically of side, back, and top enclosure panels, a work surface or counter top, an access opening called the face, a sash, and an exhaust plenum equipped with a baffle system for the regulation of air flow distribution." Listed below are some definitions that should help in the understanding of exhaust hoods (Figure D-37):

- Capture Velocity—The air velocity at any point in front of the hood or at the hood opening necessary to overcome opposing air currents, and to capture the contaminated air at that point by causing it to flow into the hood.
- Face Velocity—Air velocity at the hood opening.
- Slot Velocity—Air velocity (fpm) through the openings in a slot type hood. It is used primarily as a means of obtaining uniform air distribution across the face of the hood.
- Plenum Velocity—Air velocity (fpm) in the plenum. For good air distribution with slot type hoods, the maximum plenum velocity should be half of the slot velocity or less.
- Duct Velocity—Air velocity (fpm) through the duct cross-section. When solid material is present in the air stream, the duct velocity must be equal to the minimum design duct velocity.
- Minimum Design Duct Velocity—Minimum air velocity (fpm) required to move the particulates in the air stream.

Hoods can be classified as either enclosed or nonenclosed (Figure D-38). Enclosed hoods provide a more economical contaminant control because the effects of room air currents and the exhaust rate are small compared to those for a nonenclosed hood. When nonenclosed hoods need to be used, careful attention should be given to air flow patterns around the process and hood. Nonenclosed hoods should also be located so that the contaminant is drawn away from an operator's breathing zone. Figure D-39 illustrates the various hood types available.

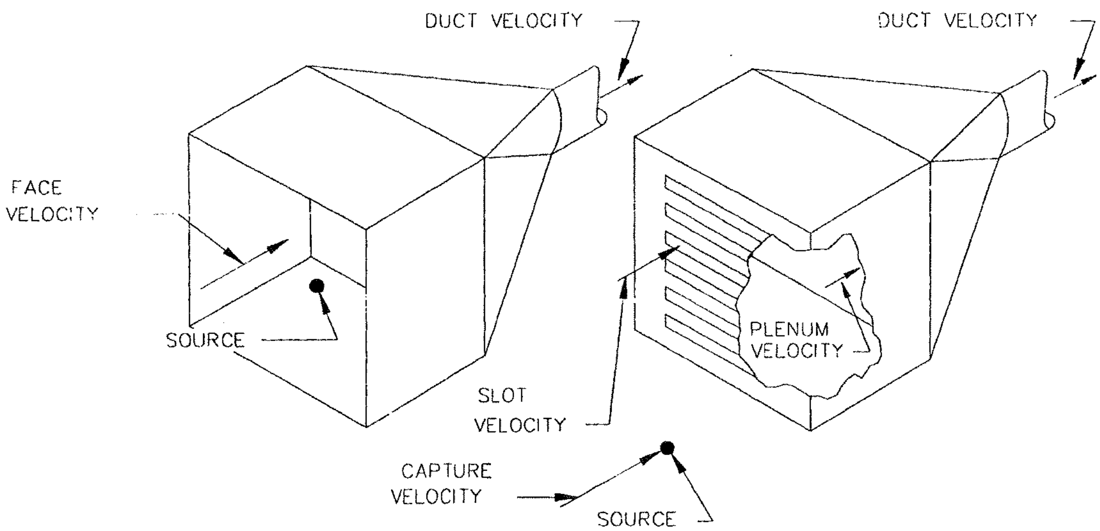

Figure D-37. Basic Exhaust Hood Terms.

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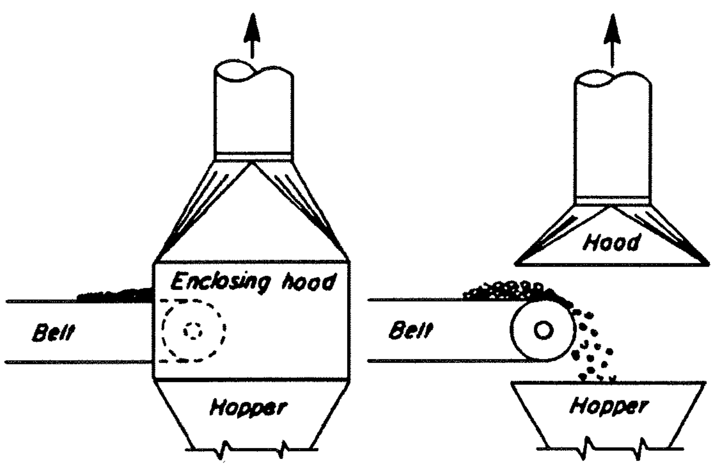

Figure D-38. Enclosed Hood (left) and Nonenclosed Hood (right).

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Hood access openings should be as small as possible. Access should be provided for inspection and maintenance. Hoods should be placed as close as possible to the source of the contaminant. The required volume varies with the square of he distance from the source.

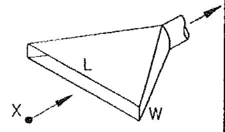
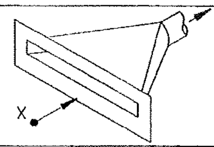
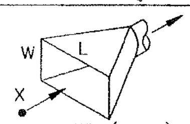
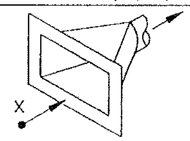
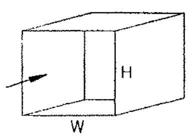
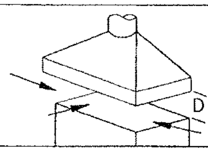
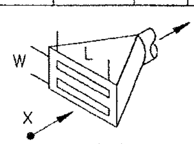
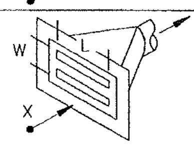
HOOD TYPE	DESCRIPTION	ASPECT RATIO,W/L	AIR FLOW
	SLOT	0.2 OR LESS	$Q = 3.7\ LVX$
	FLANGED SLOT	0.2 OR LESS	$Q = 2.6\ LVX$
 A = WL (sq.m)	PLAIN OPENING	0.2 OR GREATER AND ROUND	$Q = V(10X^2 + A)$
	FLANGED OPENING	0.2 OR GREATER AND ROUND	$Q = 0.75V(10X^2 + A)$
	BOOTH	TO SUIT WORK	$Q = VA = VWH$
	CANOPY	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 D-39. Hood Types.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

Capture Velocity

Table D-13 shows ranges and applications of capture velocities for several industrial operations. These capture velocities are based on successful experience under ideal conditions.

Condition of Dispersion of Contamination	Example	Capture Velocity (fpm)
Released with practically no velocity into quiet air.	Evaporation from tanks; degreasing, etc.	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 at very rapid air motion.	Grinding; abrasive blasting; tumbling	500-2000
In each category above, a range of capture velocity 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 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.	

Table D-13. Capture Velocities.

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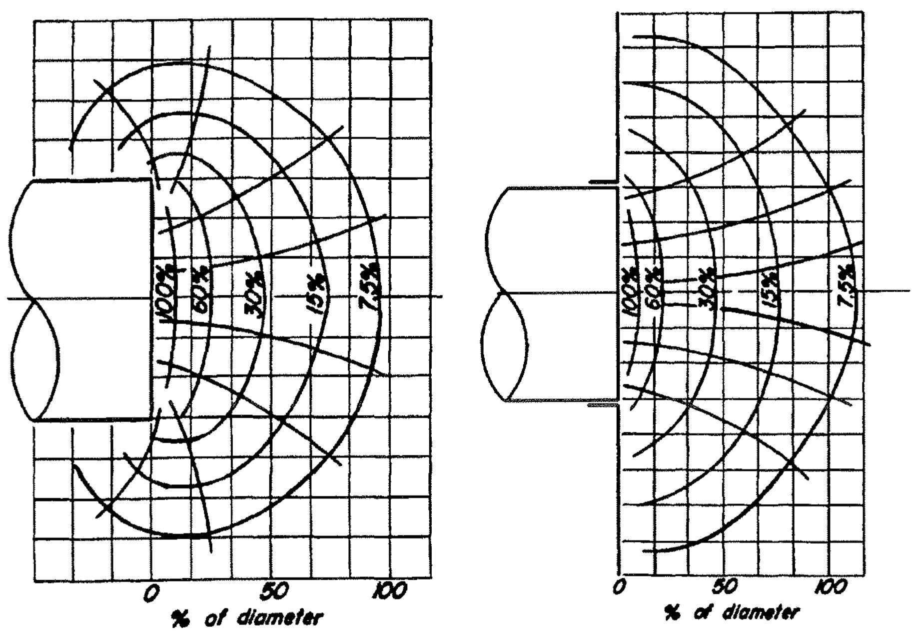
Flanging

Wherever possible, flanges (which are projected rims or collars) should be provided to eliminate air flow from ineffective zones where no contaminant exists. The hood effectiveness is increased, and air requirements can be reduced by as



(a) Circular Opening (b) Circular Opening with Flange

Figure D-40. Effect of Flanging.

From American Conference of Governmental Industrial Hygienists, *Industrial Ventilation: A Manual of Recommended Practice,* 23rd Ed., © 1998, Cincinnati, OH. Reprinted with permission.

much as 25 percent. Figure D-40 shows that the lines in front of the hood are lines of equal velocity and are called flow contours. The lines perpendicular to the flow contours are called streamlines. The tangent to a streamline at any point indicates the direction of airflow at that point.

Volumetric Flow Rate

The exhaust volumetric flow rate can be calculated after the capture velocity and hood configuration are determined.

For enclosed hoods, the exhaust volumetric flow rate can be calculated by the equation:

$$Q = V \times A \qquad \text{[Eq D-8]}$$

where:
- Q = volumetric flow rate, cfm
- V = average flow velocity, fpm
- A = flow cross-sectional area, sq ft

The inflow velocity is usually around 100 fpm.

For nonenclosed hoods, the capture velocity and the air velocity at the point of contaminant release must be equal and be directed so that the contaminant enters the hood. This results in different volumetric flow rate equations for different types of hoods. For unflanged round and rectangular openings, the required flow rate equation is:

$$Q = V \times [(10X \times 10X) + A]$$

where
- Q = flow rate, cfm
- V = capture velocity, fpm
- X = centerline distance from the hood face to the point of contaminant generation, ft
- A = hood face area, sq ft
- L = long dimension of the slot, ft

For slot hoods, the required flow rate is predicted by an equation for openings between 0.5 to 2 in. in width:

$$Q = 3.7 \times L \times V \times X$$

If a flange is installed around the hood opening, the required flow rate for plain openings is reduced to 75 percent of that for the corresponding unflanged opening. The flange size should be approximately equal to four times the area divided by the perimeter of the face hood. For flanged slots with aspect ratios less than 0.15 and flanges greater than three times the slot width, the equation is:

$$Q = 2.6 \times L \times V \times X$$

A baffle is a solid barrier that prevents airflow from unwanted areas in front of the hood. For hoods that include baffles, the DallaValle half-hood equation is used to approximate the required flow rate:

$$Q = V \times [(5X \times 5X) + A]$$

Volumetric Flow Rate Example

Design a nonenclosed hood to capture a contaminant that is released with a low velocity 2 ft in front of the face of the hood. The hood face dimensions are 1.5 × 4.0 ft. The hood rests on a bench, and a flange is placed on the sides and top of the face. The room air currents are variable in direction but less than 50 fpm, and the contaminant has low toxicity. Determine the volumetric flow rate required to capture the contaminant if the hood is used continuously.

Solution

Table D-13 shows that a capture velocity of 50 to 100 fpm is required. The capture velocity selected must be greater than the room air currents, so 80 fpm will be selected. Modifying the equation listed for baffles so reduction is included for flanges results in:

$$Q = 0.75V \times [(5X \times 5X) + A]$$

$$Q = (0.75 \times 80) \times [(5 \times 2) \times (5 \times 2) + (1.5 \times 4)] = 1{,}560 \text{ cfm}$$

Special Situations

Some operations may require exhaust flow rates different from those listed previously. Some of these reasons are:

1. The exhaust from a hot process requires special consideration because of the heated air effect near the process. Determining the flow rate for this process

requires knowing the conventional heat transfer rate and physical size of the process.

2. High-speed rotating machines such as escaping compressed air from pneumatic tools, high-speed belt material transfer systems, and falling granular materials all produce air currents. The direction and size of the airflow should be taken into consideration when designing the hood.
3. Room air currents caused by spot cooling, compensating air, or cross drafts.
4. Exhaust flow rates that are insufficient to dilute combustible vapor-air mixtures to less than about 25 percent of the lower explosive limit of the vapor.

9 Controls

Automatic control is used to modulate equipment capacity to meet load requirements, and to provide safe operation of equipment. It requires mechanical, electrical, and electronic control devices, and implies that human intervention is limited to starting and stopping equipment, and adjusting control set points.

Components of Automatic Control Systems

Control devices for HVAC systems can be grouped by their function within a complete control system. These groups are:

1. Sensing elements
2. Controllers
3. Controlled devices
4. Auxiliary devices.

Sensing Elements

A sensor is the component in the control system that measures the value of the controlled variable. The controlled variable is the variable such as temperature, humidity, or pressure that is being controlled. The change in the controlled variable produces a change in some physical or electrical property of the primary sensing element, which is then available for translation or amplification by mechanical or electrical signal. Sensors are most often used for temperature, humidity, pressure, and water or fluid flow. Sensors can also be used for flame detection, measurement of smoke density, current, CO_2, or CO.

Controllers

Controllers take the sensor effect, compare it with the desired control condition (set point), and regulate an output signal to cause action on the controlled device. The controller and sensor can be combined in a single instrument such as a room thermostat, or they can be two separate devices. There are five basic types of controllers:

1. Electric/electronic controllers
2. Indicating or recording controllers
3. Pneumatic receiver controllers
4. Direct digital controllers
5. Thermostats.

Controlled Devices

The controlled device is most frequently used to regulate or vary the flow of air within an HVAC system. Air flow control devices are called dampers. Dampers must be properly sized and selected for a particular application for the control system to control the controlled variable properly. The control system's link to the damper is called an operator or actuator. This device uses electricity, compressed air, or hydraulic fluid as a power source.

Auxiliary Devices

Some examples of auxiliary devices are: transformers, electric relays, potentiometers, manual switches, and auxiliary switches.

Outside Air Control

The total control of an air handling system can be subdivided into several elements including outside air, heating, cooling, humidity, pressure, and space conditions. The control of outside air can be accomplished in many ways.

Fixed Outside Air

This type of control system (Figure D-41) is sized to make up exhaust and exfiltration from the space. Control consists of a two-position outside air damper interlocked to open when the supply fan runs. A manual return air damper usually provides balancing. This system is not energy efficient because it does not use "free" cooling (see ***Economy Cycle***). Most individual room units use this method.

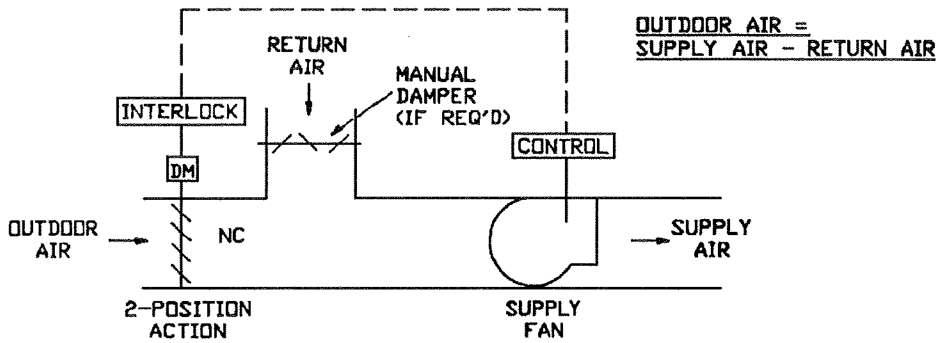


Figure D-41. Fixed Minimum Outdoor Air for Systems without Return Fans.

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100% Outside Air

This type of control (Figure D-42) can be used in buildings with large exhaust air requirements. Control consists of a two-position outside air damper interlocked to run when the supply fan runs. Interlocks are also provided between supply and exhaust fans.

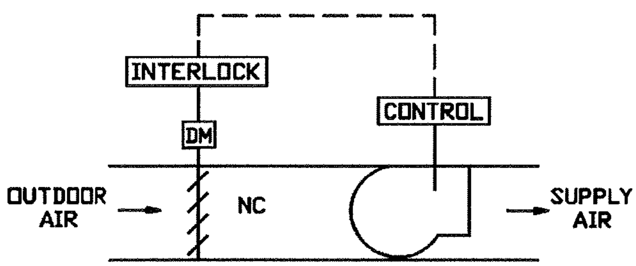

Figure D-42. 100% Outside Air Control.

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Economy Cycle

This type of control (Figure D-43) is used most often. A properly designed economy cycle is very energy efficient. When the supply fan is started, the outside air damper opens to the minimum position required for ventilation or exhaust makeup. When the outside air temperature is above a high limit, usually 70 to 75 °F, the outside air damper stays in the minimum position. When the outside air temperature is below the high limit, the outside return and relief air dampers are modulated to maintain a mixed air temperature not less than the low limit set point, usually 55 to 60 °F. In practice, a wide variety of control configurations achieve this sequence.

Thermal energy can be saved by adding reset of the low limit set point in response to cooling load. The set point is reset upward as the cooling load decreases, minimizing reheat energy usage. Many modern systems use this practice, and it is often fairly simple to retrofit older systems.

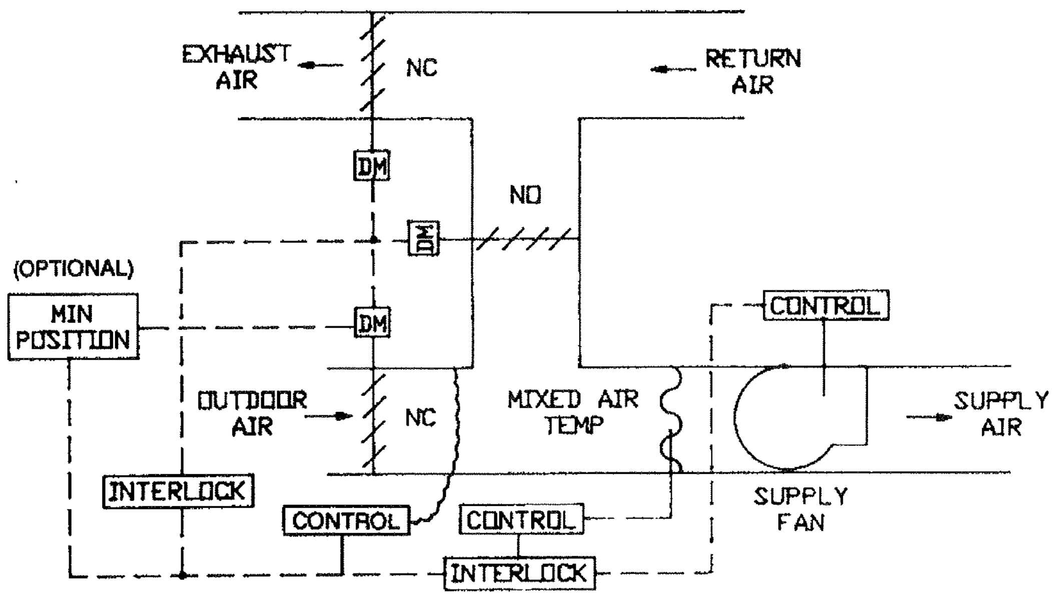

Figure D-43. Economizer Cycle Control.
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10 Acceptance Testing

Before starting an acceptance testing procedure, inspections should be made to confirm that equipment has been completely installed, proper electrical connections have been made, automatic controls are complete and operating, and the building is completely closed in with windows, doors, etc.

Prechecks

The following is a preliminary check-out procedure that should be used to confirm that equipment is ready to be tested, adjusted, and balanced:

1. Obtain all equipment data from the manufacturers, and from the design specifications.
2. Obtain and calibrate the instrumentation that will be used.
3. Make sure all measuring points are accessible.
4. Confirm the following on fans:
    a. All bearings have been lubricated.
    b. Fan wheels clear the housing.
    c. All foreign objects have been removed.
    d. Motors have been fastened securely.
    e. All drives have been correctly aligned.
    f. Belt tensions are correct.
    g. Fan rotations are correct.
    h. Duct flexible connections are properly aligned.
    i. Vibration isolators or bases have the correct springs and are in the right location, and that the springs are not collapsed.
    j. Fan flow rate.
5. Confirm the following on all duct systems:
    a. All outside air intake, return air, and exhaust air dampers are in the proper position and operable.
    b. All system volume dampers and fire dampers have been installed, are in the full open position, and are accessible.

   c. Inspect access doors and hardware for tightness and leakage, and verify that all necessary access doors have been installed.
   d. Openings have been provided in walls and plenums for proper air passage.
   e. Duct flow rates.
   f. Backdraft dampers installed and operational.
   g. Determine duct leakage.
6. Locate all start-stop, disconnect switches, electrical interlocks, and motor starters.

The following is a checklist to follow during acceptance testing.

EXHAUST SYSTEM ACCEPTANCE TESTING CHECKLIST

PROJECT:____

LOCATION:____

NAME:____

A. Prechecks	**Checked**	**Date Checked**
1. All equipment data received from the manufacturer and design specifications		
2. All instrumentation obtained and calibrated		
3. All measuring points accessible		
4. All start-stop, disconnect switches, electrical interlocks, and motor starters located		

B. Fans		**Correct** yes	no	**Date Checked**
1. All bearings lubricated				
2. Fan wheels clear housing				
3. All foreign objects removed				
4. Motors fastened securely				
5. All drives correctly aligned				
6. Belt tensions				
7. Fan rotations				
8. Duct flexible connections properly aligned				
9. Vibration isolators or bases have correct springs, are in right locations, and are not collapsed				
10. Correct flow rate (cfm)	Design	TAB	Actual	

C. Ducts		**Correct** yes	no	**Date Checked**
1. All outside air intake, return air, and exhaust air dampers are in proper position and operable				
2. All system volume dampers and fire dampers installed, in full open position and operable				
3. All necessary access doors are installed, tight, and free of leakage				
4. Openings provided in walls and plenums for proper air passage				
5. Selected duct flow rate (cfm)	Design	TAB	Actual	

After confirming that the preliminary check-out procedures have been completed, the following procedures can be reviewed. The purpose of these checks is to provide the acceptance testing team with a concise outline of what the TAB contractor was supposed to have done during TAB.

Basic to All Air Systems

1. Confirm that every item affecting the air flow of a duct system is ready. (Windows and doors are closed, ceiling tiles are in place, etc.).
2. Confirm that all automatic control devices did not affect TAB operations.
3. Establish the conditions for the maximum demand system air flow.
4. After verifying that all dampers are open or set, start all related systems (return, exhaust, etc.) and the system being balanced with each fan running at the design speed. Upon starting each fan, immediately check the fan motor amperage. If the amperage exceeds the nameplate full load amperage, stop the fan to determine the cause or make the necessary adjustments.
5. Confirm that all related system fans serving each area within the space being balanced are operating.
6. If a supply fan is connected to a return air system and an outside air intake, set all system dampers and controls so that the air returned from the individual rooms or areas supplied by the fan is returned via the related return air system. Normally this will involve opening an outside air damper to the minimum position, opening the return air damper, and closing exhaust air and relief air dampers.
7. Determine the volume of air being moved by the supply fan at design rpm by one or more of the following methods:

a. Perform a pitot tube traverse of the main duct or ducts leaving fan discharge.
b. Verify fan curves or fan performance charts. To do this, amperage, voltage, and static pressure readings need to be taken.

8. If the supply fan volume is not within ±10 percent of the design capacity at design rpm, determine the reason by reviewing all system conditions, procedures, and recorded data. Check and record the air pressure drop across filters, coils, eliminators, sound traps, etc. to see if excessive loss is occurring. Study duct and casing conditions at the fan inlet and outlet.
9. Using the methods outlined in paragraph 8, determine the volume of air being handled by a return air fan, if used; if a central exhaust fan system is used, also determine the cfm being handled by the exhaust fan. If several exhaust fans are related to a supply system, it is not generally necessary to

measure the cfm of each exhaust fan until after the supply system is balanced.

10. If the measured cfm of the supply fan, central return fan, or central exhaust fan varies more than 10 percent from design, adjust the drive of each fan to obtain the approximate required cfm. Confirm that the fan motor is not overloaded.
11. Make a preliminary survey by spot checking air circulation in various rooms. With knowledge of the supply and return or exhaust fan volumes and data from the survey, determine if the return or exhaust air system should be balanced before the supply system is balanced. The assumption is made that the supply system is not restrained by the exhaust system or the return system. However, if such a restraint exists, the exhaust system or the return system should be balanced prior to continuing with the supply system.
12. The system is considered balanced in accordance with these procedural standards when the value of the air quantity of each inlet or outlet device is measured and found to be within 10 percent of the design air quantities.

Exhaust and Return Air Systems

1. Follow procedures 1 through 7 under the previous section.
2. Determine the volume of air being moved by the exhaust fan at design rpm by fan curves, or by pitot tube traverse of the main duct or the ducts leaving the fan discharge.
3. The exhaust fan volume should be within ±10 percent of the design capacity if earlier procedures were followed. Check and record the air pressure drop across filters, coils, sound traps, etc., to see if any excessive loss is occurring. Study duct and casing conditions at the fan inlet and outlet. Record the exhaust fan suction static pressure, fan discharge static pressure, amperage, and cfm measurements. Confirm that the fan motor is not overloaded.
4. If the exhaust system is being balanced before the supply and/or return air systems, and if the measured cfm of any fan varies more than 10 percent from design, adjust the drive of each fan to obtain the approximate required cfm. Make a preliminary survey by spot checking air circulation in various areas. Then follow all procedures as outlined after the exhaust system is balanced.
5. Make pitot tube traverse on all main exhaust ducts to determine the air distribution. Investigate any branch that is very low in capacity to make sure that no blockage exists.

6. Adjust the volume dampers in the main ducts to the appropriate air flow (cfm) requirement.
7. Without adjusting any terminal device, measure and record the air flow at each terminal in the system. Study any radical conditions and correct them. Plan the sequence of branch balancing. In making the adjustments, it is preferable to adjust the branch dampers rather than the dampers at the air terminals. If the throttling process at a terminal damper involves closing the damper to a degree that generates noise, evaluate the design cfm capacity of the branch duct.
8. Working from the branch with the highest measured capacity to the branch with the lowest measured capacity, make adjustments in each branch. Beginning with the inlet device most distant from the branch and proceeding toward the branch connection, make volume adjustments at each terminal as necessary. It is important that the balancer use the proper "k" factor prescribed by the air terminal manufacturer for use in conjunction with a particular instrument. In addition, it is often necessary that the readings at grilles, registers, and diffusers be taken in a position or number of positions prescribed by the manufacturer of the air terminal device.
9. Repeat the branch balancing until the system is in balance.
10. Verify the fan capacity and operating conditions again and make a final adjustment in the fan drive if necessary.
11. Verify the action of all fan shut down controls and air flow safety controls.

Glossary

AMPLITUDE: The maximum displacement from the mean position of oscillation or vibrations.

BTU: British thermal unit; the amount of heat required to raise the temperature of 1 pound of water by 1 degree Fahrenheit.

BUILDING ENVELOPE: The imaginary shape of a building indicating its maximum volume; a transition space where the interaction between outdoor forces and indoor conditions can be watched.

CORROSION: Deterioration of metal by chemical or electrochemical reaction resulting from exposure to weathering, moisture, chemicals, or other agents in the environment in which it is placed.

ELASTIC LIMIT: The greatest stress that a material is capable of sustaining without permanent deformation upon complete release of the stress.

EXFILTRATION: The outward flow of air through a wall, joints, etc.

EXHAUST VENTILATION: The removal of foul air from a space by a mechanical means, such as a fan. Fresh air is allowed to enter through available or controlled openings.

HEAT EXCHANGER: A device designed to transfer heat between two physically separated fluids. It generally consists of a cylindrical shell with longitudinal tubes; one fluid flows on the inside, the other on the outside.

INFILTRATION: The seepage or flow of air into a room or space through cracks around windows, under doors, etc.

LATENT HEAT: The amount of heat that is absorbed in changing the state of a substance without changing its temperature.

LEEWARD: Situated away from the wind.

MODULUS OF ELASTICITY: In an elastic material that has been subject to strain below its elastic limit, the ratio of the unit stress to the corresponding unit strain.

MOMENT OF INERTIA: Of a body around an axis, the sum of the products obtained by multiplying each element of mass by the square of its distance from the axis.

NATURAL VENTILATION: Ventilation by air movement caused by natural forces, rather than by fans.

PERMEABILITY: The property of a porous material that permits the passage of water vapor through it.

PICKUP LOAD: The abnormal rate of heat consumption that takes place when a heating system is first turned on. It represents the heat dissipated in bringing the piping and radiators to their normal operating temperature.

POROSITY: A ratio, usually expressed as a percentage of the volume of voids in a material to the total volume of the material, including the voids. The voids permit gases or liquids to pass through the material.

SENSIBLE HEAT: Heat that changes the temperature of a material without a change in state, such as that which would lead to increased moisture content.

STACK EFFECT: The tendency of air in a shaft or other vertical passage to rise when heated, owing to its lower density compared with that of the surrounding air.

STATIC PRESSURE: The pressure that the fan must supply to overcome the resistance to air flow through the system ductwork and system components.

THERMAL LOAD: A load on a structure that is induced by changes in temperature.

THERMAL SHOCK: The sudden stress produced in a material as a result of a sudden temperature change.

THERMAL TRANSMITTANCE: The time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids.

VAPOR BARRIER: A moisture-impervious layer or coating (such as special paint, or a membrane on roofing felt or on building paper) that prevents the passage of moisture or vapor into a material or structure.

VAPOR PRESSURE: The component of the total pressure that is caused by the presence of a vapor, as for example, by the presence of water vapor in air.

VENTILATION: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

WINDWARD: Situated toward the direction from which the wind is blowing.

Bibliography

American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE), *Cooling and Heating Load Calculation Manual,* ASHRAE, Atlanta, GA, 1979.

ASHRAE, *Fundamentals Handbook,* 1997.

ASHRAE, *HVAC Applications Handbook,* 1995.

ASHRAE, *HVAC Systems and Equipment Handbook,* 1996.

ASHRAE, *Standard 62-1989, Ventilation for Acceptable Indoor Air Quality,* 1989.

Baturin, V. V., *Fundamentals of Industrial Ventilation*, Pergamon Press, 1972.

Bevirt, W. David, *Environmental Systems Technology,* National Environmental Balancing Bureau (NEBB), Vienna, VA, 1986.

Bradshaw, Vaughn, *Building Control Systems, 2nd Ed.*, John Wiley & Sons, New York, NY, 1993.

Committee on Industrial Ventilation, *Industrial Ventilation, 23rd Ed.*, American Conference of Governmental Industrial Hygienists, Cincinnati, OH, 1998.

Harris, Cyril, *Dictionary of Architecture and Construction*, McGraw Hill, Inc., New York, NY, 1975.

National Air Filtration Association (NAFA), *NAFA Guide to Air Filtration,* NAFA, Washington, DC, 1993.

NEBB, *Procedural Standards for Testing, Adjusting, Balancing of Environmental Systems, 4th Ed.*, 1983.

Sheet Metal and Air Conditioning Contractors National Association, Inc., (SMACNA) *HVAC Duct Construction Standards Metal and Flexible, 2nd Ed.*, SMACNA, Vienna, VA, 1995.

SMACNA, *Manual for the Balancing and Adjustment of Air Distribution Systems*, 1981.

SMACNA, *HVAC Systems Duct Design, 3rd Ed.*, 1990.

Stein, Reynolds, and McGuinness, *Mechanical and Electrical Equipment for Buildings*, 7th Ed., (John Wiley & Sons, Inc., 1986).