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Air

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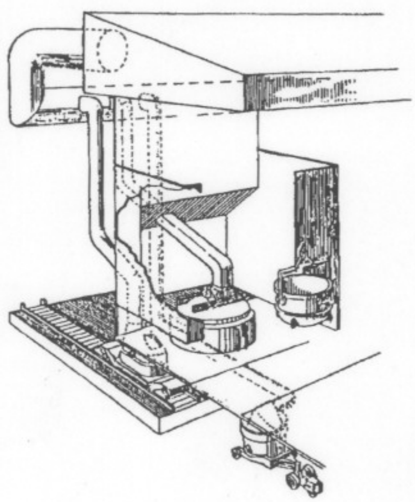
Course 345
Emission Capture and Gas Handling System Inspection

Student Manual



Office of Air Quality Planning and Standards

Clean Air

OAQPS

Air Pollution Training Institute
Course 345

Emission Capture and Gas Handling System Inspection

Student Manual

Prepared by:
Crowder Environmental Associates, Inc.
2905 Province Place
Plano, TX 75075

Entropy Environmentalist, Inc.
P.O. Box 12291
Research Triangle Park, NC 27709

EPA Project Officer
Lourdes L. Morales
and
Kirk Foster

Revised December 1995

U.S. Environmental Protection Agency
Office of Air Quality Planning and Standards
Education and Outreach Group
Research Triangle Park, NC

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Chapter 1

General Principles of Ventilation

Level 2 and 3 inspections of ventilation systems require interpretation of instrument readings and may, at times, require measurement and calculation of performance parameters. To be able to conduct these inspections effectively, it is important that you have a firm understanding of the basic information that affects the behavior of air streams. The purpose of this chapter, divided into two sections, is to give you that information. The first section defines the various parameters that are important in ventilation system evaluation and indicates techniques and information sources that may be used in their determination. The second section presents the fundamentals of fluid flow and includes a discussion of the implications of continuity and momentum relationships. Considerable emphasis is placed on Bernoulli's equation, and it is used to develop relationships for the pressures that exist in a flowing system and for determining the velocity of an air stream.

Properties of Air and Air-Water Vapor Mixtures

Standard air

Standard air is defined as air with a density of 0.075 lb_m/ft^3 and an absolute viscosity of 1.225×10^{-5} lb_m/ft-sec. This is equivalent to dry air at a temperature of 70°F and a pressure of 29.92 in. Hg.

Molecular weight

Atmospheric air is a mixture of dry air, water vapor and various impurities. Dry air itself is also a mixture of gases. Because of this, neither atmospheric air nor dry air have a true molecular weight. However, they do have an apparent molecular weight that can be calculated from their composition. Assuming dry air consists, by volume, of 78.09% nitrogen, 20.95% oxygen, 0.93% argon and 0.03% CO_2, its apparent molecular weight may be calculated as shown in Table 1-1. The apparent molecular weight of dry air with this composition is then 28.966 lb/lb-mole.

Table 1-1. Calculation of molecular weight from volume fraction					
Component	Volume fraction		Molecular weight		$\frac{lb}{lb-mole}$
N_2	0.7809	x	28.016	=	21.878
O_2	0.2095	x	32.000	=	6.704
Ar	0.0093	x	39.944	=	0.371
CO_2	0.0003	x	44.010	=	0.013
	1.0000				28.966

Suppose the compositional information was available on a weight rather than a volume basis. If dry air consisted, by weight, of 75.52% nitrogen, 23.15% oxygen, 1.28% argon and 0.04% CO_2, its apparent molecular weight would be determined as shown in Table 1-2. The apparent molecular weight of dry air with this composition is then 1/0.03452 = 28.969 lb/lb-mole. When composition information is not available, dry air is typically taken to have an apparent molecular weight of 28.95 lb/lb-mole and sometimes approximated as 29 lb/lb-mole.

Table 1-2. Calculation of molecular weight from weight fraction					
Component	Weight fraction		Molecular weight		$\frac{lb-mole}{lb}$
N_2	0.7552	÷	28.016	=	0.02696
O_2	0.2315	÷	32.000	=	0.00723
Ar	0.0128	÷	39.944	=	0.00032
CO_2	0.0005	÷	44.010	=	0.00001
	1.0000				0.03452

For wet air, the apparent molecular weight may be calculated from the composition as shown above, or by combining the molecular weights of the dry air and the water vapor on the basis or their respective volume fraction or mole fraction:

$$MW_{wet\ air} = (1 - \chi_{water})(MW_{dry\ air}) + (\chi_{water})(MW_{water}) \quad (1\text{-}1)$$

Equation of state

Equations of state relate the pressure, volume and temperature properties of a pure substance or mixture by semi-theoretical or empirical relationships. Over the range of temperature and pressure usually encountered in ventilation systems, these values may be related by the ideal or perfect gas law:

$$PV = nRT \tag{1-2}$$

where
P = absolute pressure (lb_f/ft^2)
V = gas volume (ft^3)
n = number of moles (lb-moles)
R = constant (1544.58 ft-lb_f/lb-mole-°R)
T = absolute temperature (°R)

Here, R is referred to as the universal gas constant, and its value depends on the units of the other terms in the equation. Other values of R include:

10.73 psia-ft^3/lb-mole-°R
0.73 atm-ft^3/lb-mole-°R
82.06 cm^3-atm/g-mole-°K
8.31 x 10^3 kPa-m^3/kg-mole-°K

A more useful form of the ideal gas law may be developed by noting that PV/T = nR, and that, for a given number of moles of a gas, nR is a constant. Thus, at two different conditions for the same gas, we may write:

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \tag{1-3}$$

or

$$V_1 = V_2 \left(\frac{P_2}{P_1}\right)\left(\frac{T_1}{T_2}\right) \tag{1-4}$$

Equation 1-4 allows volumes (or volume rates) to be corrected from one set of temperature and pressure conditions to another.

Another useful form of the ideal gas law may be used to calculate the molar volume, V/n. For an ideal gas at 70°F and 29.92 in. Hg (14.7 psia), the molar volume is given by:

$$\frac{V}{n} = \frac{RT}{P} = \frac{\left(10.73 \frac{\text{psia} - \text{ft}^3}{\text{lb} - \text{mole}\ ^\circ\text{R}}\right)(530^\circ \text{R})}{(14.7\ \text{psia})} = 387 \frac{\text{ft}^3}{\text{lb} - \text{mole}}$$

Density and specific volume

Density is the ratio of mass to the volume occupied, e.g., lb/ft^3 or g/cm^3. Specific volume is the volume occupied per mass and is equal to the inverse of density. Both of these quantities depend on the temperature and pressure of the system. Using the ideal gas law, and recognizing that the number of moles is given by mass divided by molecular weight, density (ρ) may be calculated from:

$$\rho = \frac{m}{V} = \frac{P \cdot MW}{RT} \tag{1-5}$$

Density can also be determined from molecular weight and molar volume:

$$\rho = \left(\frac{MW}{387}\right)\left(\frac{530}{T}\right)\left(\frac{P}{29.92}\right) \tag{1-6}$$

where MW = molecular weight
T = absolute temperature (°R)
P = absolute pressure (in. Hg)

Values for the density and other properties of air over a limited range of temperature are provided in Table 1-3.

Specific gravity

Specific gravity is the ratio of the density of a material to the density of some reference substance. For gases that reference substance is frequently dry air, while for liquids and solids it is usually water. Referring to Equation 1-5, it can be seen that for an ideal gas specific gravity is also given by the ratio of the molecular weight of the gas to the molecular weight of dry air.

Relative and absolute humidity

The state of an air-water vapor mixture is completely defined by specifying the pressure, temperature and humidity. The Gibbs-Dalton rule of partial pressures states that individual components in a mixture exert a pressure that would be the same as that exerted if the same mass of the component were present alone in the same total volume and at the same temperature. Thus, for an air-water vapor mixture:

$$p_{air} + p_{water} = P_{total} \tag{1-7}$$

Table 1-3. Properties of air (Danielson, 1973)			
Temperature °F	Specific Heat C_p	Viscosity μ	Density ρ
0	0.240	0.040	0.0863
20	0.240	0.041	0.0827
40	0.240	0.042	0.0794
60	0.240	0.043	0.0763
80	0.240	0.045	0.0734
100	0.240	0.047	0.0708
120	0.240	0.047	0.0684
140	0.240	0.048	0.0662
160	0.240	0.050	0.0639
180	0.240	0.051	0.0619
200	0.240	0.055	0.0601
250	0.241	0.055	0.0558
300	0.241	0.058	0.0521
350	0.241	0.060	0.0489
400	0.241	0.063	0.0460
450	0.242	0.065	0.0435
500	0.242	0.067	0.0412
600	0.242	0.072	0.0373
700	0.243	0.076	0.0341
800	0.244	0.080	0.0314
900	0.245	0.085	0.0295
1,000	0.246	0.089	0.0275
1,200	0.248	0.097	0.0238
1,400	0.251	0.105	0.0212
1,600	0.254	0.112	0.0192
1,800	0.257	0.120	0.0175
2,000	0.260	0.127	0.0161

Relative saturation is then defined as the ratio of the partial pressure of water vapor present to that which would be present if the air were saturated:

$$\text{Relative saturation} = \frac{p_{\text{water}}}{p_{\text{water at saturation}}} \tag{1-8}$$

It should be noted that relative saturation is also equal to the ratio of the corresponding mole fractions. Relative humidity is simply relative saturation multiplied by 100 to express it in percent.

Absolute or specific humidity is the weight of water vapor per weight of dry air, usually expressed as pounds of water per pound of dry air.

Dry-bulb, wet-bulb and dew-point temperatures

Temperature that is measured with a standard thermometer, or an equivalent device, is termed dry-bulb temperature. If you take that same standard thermometer and place a porous wick over the sensing bulb, you will have created a wet-bulb thermometer. As you move this thermometer through the air, or place it in a moving air stream, water from the wick will evaporate. When this happens, the wick cools down and continues to cool until the rate of energy transferred to the wick from the air equals the rate of energy loss caused by the evaporating water. The temperature of the bulb when the wet wick is at equilibrium is termed wet-bulb temperature. Since the rate of evaporation will depend on the moisture content of the air, wet-bulb temperature provides an indication of the humidity of the air.

It can also be shown that, for water only, the wet-bulb temperature is essentially the same as the adiabatic saturation temperature. Adiabatic processes are simply those processes which occur without exchanging heat with the surroundings. For example, cooling of a gas stream by evaporating water is a process that can usually be considered adiabatic. As this process proceeds, the amount of moisture in the gas stream and the gas stream temperature will always give approximately the same wet-bulb temperature. We will use this property later to estimate cooling water requirements for evaporators.

Dew-point temperature is the temperature at which condensation begins when moist air is gradually cooled. More precisely, it is the saturation temperature corresponding to the absolute humidity.

Enthalpy

Enthalpy is a measure of the thermal energy of a substance. Common units for ventilation work are Btu/lb or calories/gram. The Btu, or British thermal unit, is defined as the amount of heat necessary to raise one pound of water from 59°F to 60°F at a pressure of one atmosphere. The calorie is the amount of heat required to raise one gram of water at one atmosphere from 14.5°C to 15.5°C.

The enthalpy of a substance at a given temperature has no practical value except in relation to the enthalpy at another temperature or condition. Since enthalpy differences are proportional to temperature differences, arbitrary datum temperatures may be chosen to define enthalpy at any other temperature:

$$h = C_p(t - t_{ref}) \qquad (1\text{-}9)$$

where h = enthalpy (Btu/lb)
C_p = heat capacity at constant pressure (Btu/lb-°F)
t = temperature of substance (°F)
t_{ref} = reference temperature (°F)

For gases the reference temperature is typically 0°F or 60°F, although this is highly variable. For water it is usually 32°F.

The heat capacity, C_p, is a function of temperature and is determined from tabulations in reference texts. Values for air and water vapor over a limited range of temperature are provided in Table 1-4.

Table 1-4. Enthalpies of various gases (Danielson, 1973)

Temperature °F	Water vapor* Btu/lb	Air Btu/lb
100	17.8	9.6
150	40.3	21.6
200	62.7	33.6
250	85.5	45.7
300	108.2	57.8
350	131.3	70.0
400	154.3	82.1
450	177.7	94.4
500	201.0	106.7
600	248.7	131.6
700	297.1	156.7
800	346.4	182.2
900	396.7	211.4
1,000	447.7	234.1
1,200	552.9	287.2
1,400	661.3	341.5
1,600	774.2	396.8
1,800	889.8	452.9
2,000	1,003.1	509.5
2,500	1,318.1	654.3
3,000	1,640.2	802.3

*Enthalpies do *not* include the latent heat of vaporization. The latent heat of vaporization at 60°F (1,059.9 Btu/lb) should be used where necessary.

The enthalpy of water vapor is equal to the enthalpy of the water plus the latent heat of vaporization, λ_v:

$$h_{water\ vapor} = h_{water} + \lambda_v \tag{1-10}$$

Like C_p, the latent heat of vaporization is also a function of temperature and can be found in reference texts. The enthalpy of an air-water vapor mixture is given by:

$$h_{mixture}(Btu/lb_{dry\ air}) = h_{dry\ air} + \phi(h_{water\ vapor}) \tag{1-11}$$

where ϕ = absolute humidity ($lb_{water}/lb_{dry\ air}$)

Usually we are interested in the enthalpy difference, ΔH, between two temperature conditions, since this represents the amount of heat that must be added or removed in order to cause the change:

$$\Delta H = h_2 - h_1 = (C_p)_2(t_2 - t_{ref}) - (C_p)_1(t_1 - t_{ref}) \tag{1-12}$$

For exact calculations, the heat capacity corresponding to each temperature condition must be used. Approximate results can be obtained, and the calculations considerable simplified, by using a heat capacity averaged over the range of temperature change. For air, this typically results in only a small error. Assuming t_{ref} is the same for both enthalpies, our relationship for enthalpy change then becomes:

$$\Delta H = (C_p)_{avg}(t_2 - t_1) \tag{1-13}$$

An even simpler determination of enthalpy change can be made by using tabulated values of enthalpy as a function of temperature. One such tabulation has been included as Table 1-2. Thus, if one were interested in the amount of energy that must be removed in order to cool an air stream from 500°F to 100°F, one need only subtract the corresponding enthalpy values:

$$\Delta H = h_{500} - h_{100}$$

$$= 106.7\ Btu/lb - 9.6\ Btu/lb$$

$$= 97.1\ Btu/lb$$

If enthalpy values from different information sources are used in this manner, it must be remembered that the choice of a reference temperature is arbitrary and may not be the same for all tabulations. Before subtracting two enthalpy values, you must confirm that they were determined for the same reference temperature.

Psychrometric chart

Many of the values for air-water vapor mixtures that are used in ventilation calculations are available in a graphical representation known as a psychrometric chart. Three of these charts, covering a range of temperatures, have been included in Appendix B. As can be seen from perusing these figures, the information contained on a given chart varies. Figure 1-1 is a chart schematic that shows the location of possible information, and an explanation of each item follows.

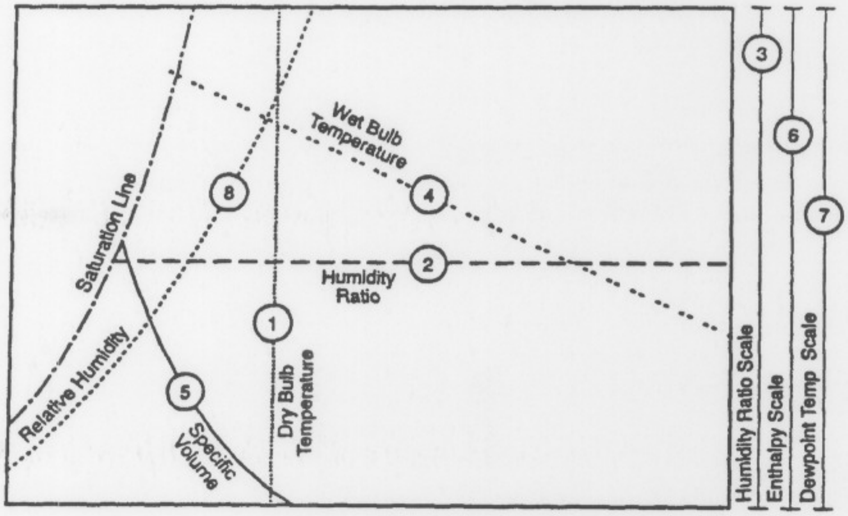


Figure 1-1. Psychrometric chart (Morse, 1965)

1. *Dry-bulb temperature:* The temperature of air read on a standard thermometer is shown on the chart by straight vertical lines. The scale is at the bottom of the chart.

2. *Absolute humidity:* The weight of water vapor per weight of dry air. On the chart these lines are horizontal and at right angles to the dry-bulb temperature lines.

3. *Absolute humidity scale:* The absolute humidity at any point on the chart is read on this scale.

4. *Wet-bulb temperature:* The temperature indicated by a thermometer whose bulb is covered by a wet porous wick and then exposed to a stream of air. The lines are straight and slope from upper left to lower right, relative to the dry-bulb temperature lines. The scale is on the curved line at the left edge of the chart.

5. *Specific volume:* The volume of mixture per weight of dry air. The lines are straight and slope from upper left to lower right, at a sharper angle than the wet-bulb temperature lines. The value is located along each line.

6. *Enthalpy:* The heat energy contained in a weight of dry air. The scale is located beyond the left edge or the right and bottom edges of the chart and is read along extensions of the wet-bulb temperature lines. On some charts this scale represents the enthalpy at saturation only. Corrections for non-saturated conditions are provided along nearly vertical lines within the chart.

7. *Dew-point temperature:* The temperature at which moisture begins to condense. The value is read on the wet-bulb temperature scale along a horizontal line of constant absolute humidity.

8. *Relative humidity:* The ratio of the partial pressure of water vapor in the air to the partial pressure at saturation. The lines are curved and extend from lower left to upper right, relative to the dry-bulb temperature lines. The value is located along each line.

9. *Vapor pressure:* The pressure exerted by the water vapor in the air. The scale is on the far right of the chart and is read along a horizontal line of constant absolute humidity.

10. *Sensible heat ratio:* The ratio of the sensible heat to the total heat of a process. These values are typically used for calculations related to conditioned air supply and are not employed in this course.

In order to use the psychrometric chart, one must first locate the position on the chart that corresponds to the conditions of the air stream. This is done by knowing any two of the above quantities and locating the point of intersection along their corresponding lines. Once this point is determined, values for the other quantities may be read from the appropriate scales. In some cases the beginning and ending points of a process may be located, and the changes in values determined by subtracting the quantities corresponding to each condition.

One value that is of interest but cannot be read directly from the psychrometric chart is the density of the air-water vapor mixture. However, this can be determined from values obtained from the chart, as follows:

$$\rho_{\text{mixture}} = \frac{(1+\phi)}{\nu} \tag{1-14}$$

where ϕ = absolute humidity ($\text{lb/lb}_{\text{dry air}}$)

ν = specific volume ($ft^3/lb_{dry\ air}$)

Principles of Fluid Flow

Continuity

Consider the flow of fluid through a tube as shown in Figure 1-2:

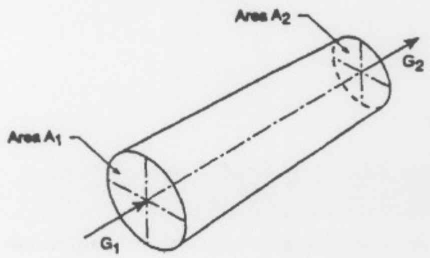


Figure 1-2. Flow diagram

The mass rate of flow through the tube (e.g., lb/min) is given by $G = \rho VA$ and the volume rate of flow (e.g., ft^3/min) is given by $Q = G/\rho = VA$. Here, ρ is the fluid density, V is the fluid velocity and A is the tube cross-sectional area. If there is no accumulation or removal of material between points 1 and 2, then we may write:

$$G_1 = G_2 \tag{1-15}$$

or

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 \tag{1-16}$$

If the fluid is incompressible or, as is usually the case in ventilation systems, the pressure is low enough that the fluid may be considered incompressible, then $\rho_1 = \rho_2$ and:

$$V_1 A_1 = V_2 A_2 \tag{1-17}$$

This relationship allows for the determination of velocity change as a gas stream flows through ducts of different diameter.

Momentum

As a fluid flows through a duct, its momentum, pressure and elevation may change. The magnitude of the change may be determined by applying the relationship, force equals rate of change of momentum, to a fluid element and then integrating over the cross-section of the duct. If frictional forces and compressibility effects are neglected, the relationship that is obtained is referred to a Bernoulli's equation:

$$\frac{V^2}{2} + \frac{P}{\rho} + gz = \text{constant} \tag{1-18}$$

where
V = fluid velocity
P = fluid pressure
ρ = fluid density
g = gravitational acceleration
z = height above a reference datum

Although this equation strictly applies only to incompressible, inviscid fluids, it is of significant importance because it relates the pressure at a point in a fluid to its position and velocity and does so in a rather simple way.

Since the conditions assumed in the development of Bernoulli's equation are approximated in most ventilation systems, it will be used to develop several useful relationships. Rearranging gives:

$$\frac{V^2}{2g} + \frac{P}{\rho g} + z = \text{constant} \tag{1-19}$$

In this form, each term represents energy per unit weight of fluid and has dimensions of length. Thus, each term may be regarded as representing a contribution to the total fluid head:

$$\frac{V^2}{2g} = \text{velocity head}$$

$$\frac{P}{\rho g} = \text{pressure head} \tag{1-20}$$

$$z = \text{potential head}$$

Consider the situation shown in Figure 1-3, in which an open tube has been inserted into a flowing fluid. The pressure of the flowing fluid causes the fluid to rise to a level, h, in the open tube.

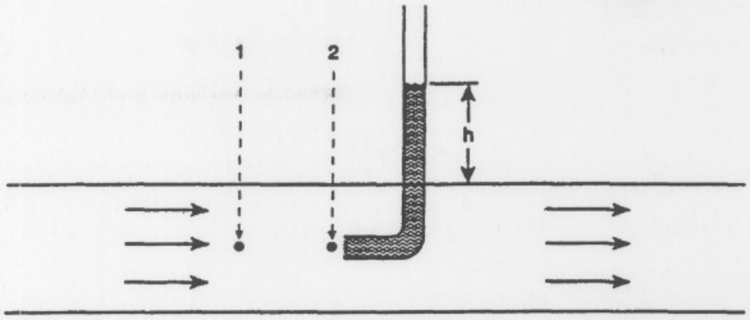


Figure 1-3. Open tube in flowing fluid

Since the terms in Bernoulli's equation sum to a constant, we may write:

$$\frac{(V_1)^2}{2g} + \frac{P_1}{\rho g} + z_1 = \frac{(V_2)^2}{2g} + \frac{P_2}{\rho g} + z_2 \tag{1-21}$$

The position of points 1 and 2 are on the same level, so $z_1 = z_2$. Also, the fluid at point 2, just at the entrance to the tube, is balanced by the fluid in the tube, so the velocity at this point is zero. Substituting gives:

$$\frac{(V_1)^2}{2g} + \frac{P_1}{\rho g} = \frac{P_2}{\rho g} \tag{1-22}$$

or

$$\frac{(V_1)^2}{2g} = \text{velocity head}$$

$$\frac{P_1}{\rho g} = \text{pressure head}$$

$$\frac{P_2}{\rho g} = \text{total head}$$

Expressing these energy heads as pressures, we may write this relationship in its more common form:

$$VP + SP = TP \tag{1-23}$$

where VP = velocity pressure
SP = static pressure
TP = total pressure

Thus, at any point in a flowing fluid, the total pressure is the sum of the velocity pressure and the static pressure. This relationship is illustrated in Figure 1-4 for an air stream on either side of a fan. Here, the manometers with one leg connected to ports that are prependicular to the flow streamlines and the other leg open to the atmosphere measure static pressure. The manometers with one leg connected to the tubes facing into the flow and the other leg open to the atmosphere measure total pressure, which is the sum of the static and velocity pressures. The manometers with one leg connected to the perpendicular ports and the other leg connected to the tubes measure the difference between total and static pressure, which is velocity pressure. It should be noted that, while static and total pressures are usually negative upstream of a fan and positive downstream of a fan, velocity pressure is always positive.

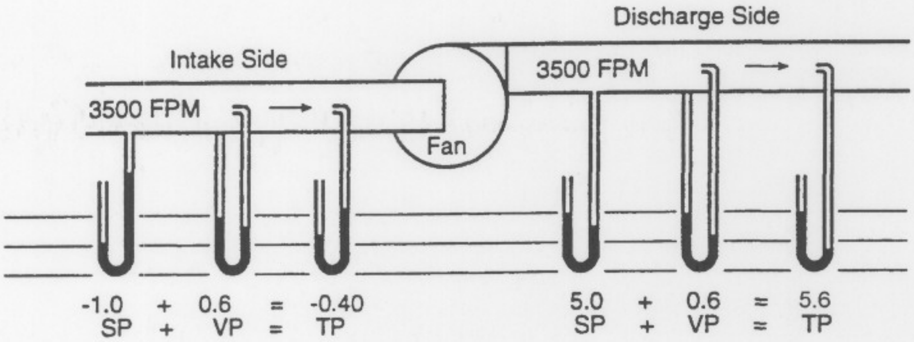


Figure 1-4. Pressure relationships

In the above development the velocity pressure, VP, was given by the term, $V^2/2g$, in units of length of fluid. In measuring velocity pressure, we typically use the pressure of the flowing fluid to displace fluid in a manometer, which we read in inches of water column. Converting the units of the velocity pressure term so that it has unit of inches of water gives:

$$\mathrm{VP} = \frac{\left(\frac{V}{60}\right)^2}{2g}\left(\frac{\rho_a}{\rho_w}\right)12 \tag{1-24}$$

where V = air velocity (ft/min)
ρ_a = air density ($\mathrm{lb/ft^3}$)
ρ_w = water density ($\mathrm{lb/ft^3}$)
g = acceleration of gravity ($\mathrm{ft/sec^2}$)

Substituting a water density at 70°F of 62.302 $\mathrm{lb/ft^3}$ and a gravity acceleration at sea level of 32.174 $\mathrm{ft/sec^2}$ gives:

$$VP = \rho_a \left(\frac{V}{1096.7} \right)^2 \qquad (1\text{-}25)$$

Since we usually measure the velocity pressure and use that to calculate the air velocity, the more useful form is:

$$V = 1096.7 \sqrt{\frac{VP}{\rho_a}} \qquad (1\text{-}26)$$

For standard air, $\rho_a = 0.075$ lb/ft^3 and our relationship becomes:

$$V = 4005\sqrt{VP} \qquad (1\text{-}27)$$

References

ACGIH, *Industrial Ventilation*, Twentieth Edition, Cincinnati, 1988.

Coulson, J.M., and J.F. Richardson, *Chemical Engineering*, Volume One, Second Edition, MacMillan, New York, 1964.

Danielson, J.A., ed., *Air Pollution Engineering Manual*, Second Edition, EPA AP-40, May 1973.

Himmelblau, D.M., *Basic Principles and Calculations in Chemical Engineering*, Fourth Edition, Prentice-Hall, Englewood Cliffs, 1982.

Jorgensen, Robert, ed., *Fan Engineering*, Seventh Edition, Buffalo Forge Company, Buffalo, 1970.

Morse, F.B., ed., *Trane Air Conditioning Manual*, The Trane Company, La Crosse, 1965.

Chapter 2

Hood Systems

The hood constitutes one of the most important components of the industrial ventilation system. When properly designed, it is instrumental in containing or capturing contaminants released by industrial processes. If the process is located outside, it is directly responsible for preventing the release of emissions to the atmosphere. When the process is located inside a building, it serves to prevent the release of contaminants to the workspace and to prevent fugitive emissions from building openings.

The goal of good hood design is high capture efficiency. The importance of this in relation to the total system is illustrated by the following equation:

$$Pt_{total} = Pt_{hood} + (1 - Pt_{hood})Pt_{collector} \tag{2-1}$$

Here, Pt is the penetration, which is one minus the fractional efficiency. Thus, even if the collector were 100 percent efficient (Pt = 0), the overall penetration of the system can be significant if the contaminants are not effectively captured by the hood.

To be able to conduct effective inspections of hood systems, it is important to understand the concepts behind good hood design and to know how to evaluate hood performance. In this chapter, we will discuss the various types of hoods, the principles that govern their design and the factors that affect their performance. We will also discuss the pressure losses associated with flow into a hood, and how knowledge of those losses can be used to estimate air volume.

Hood types

All hoods can be classified as belonging to one of four types: (1) enclosure, (2) receiver, (3) exterior and (4) push-pull. Enclosure hoods, as the name implies, envelop the process to the maximum extent possible. Typically, the designer begins by envisioning a total enclosure around the process and then removes portions of the hood only as much as is need to provide material and worker access. Because of the nature of this type of hood, it serves not to capture the contaminants but rather to contain them and remove them from within the enclosure. As a result, the quantity of air flow required for a given process is usually the least of the three hood types.

Figure 2-1, a bucket elevator, is an example of an enclosure-type hood. Buckets mounted on a rotating vertical belt are used to transfer material from one elevation to another. To reduce emissions from this process, a housing is provided that completely encloses the operation, and suction is provided to contain and remove the contaminants. The only openings are those necessary for receiving and transferring the material.

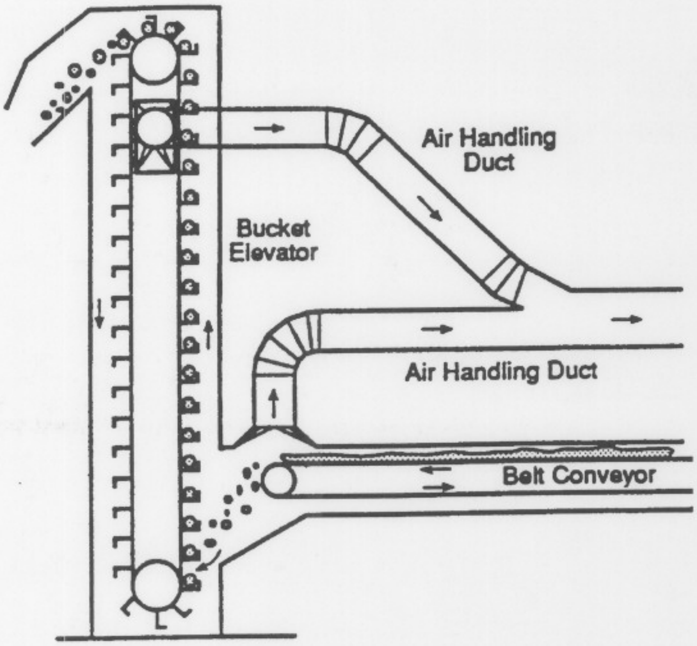


Figure 2-1. Bucket elevator enclosure

The ladle hood shown in Figure 2-2 is also an enclosure-type hood. Molten metal is transferred to a hot metal ladle from a rotating-dump torpedo car. The ladle hood contains and removes the emissions as they evolve from the transfer operation. Once the emissions have subsided, the ladle hood is tilted out of the way so that the ladle may be moved to other operations.

A receiving hood, sometimes thought of as one type of exterior hood, is located adjacent to the point of contaminant release and in an orientation that allows it to receive the emissions as they are ejected from the process. Since the hood is located along the direction of normal contaminant travel, the amount of capture capability that must be provided by the air stream is reduced.

The grinding wheel hood shown in Figure 2-3 is a receiving-type hood. Material removed by the wheel has a normal travel direction down and to the rear. The hood is mounted in this location to take advantage of this and reduce the necessary capture flow. The hood also extends around the top and sides of the wheel to provide for enclosure of any contaminants that follow the wheel motion.

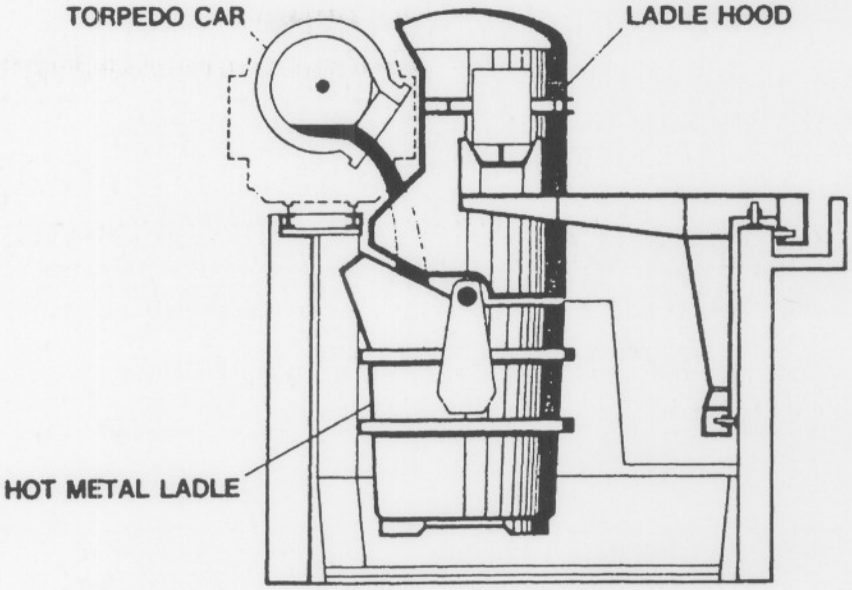


Figure 2-2. Ladle hood (Kemner et al, 1984)

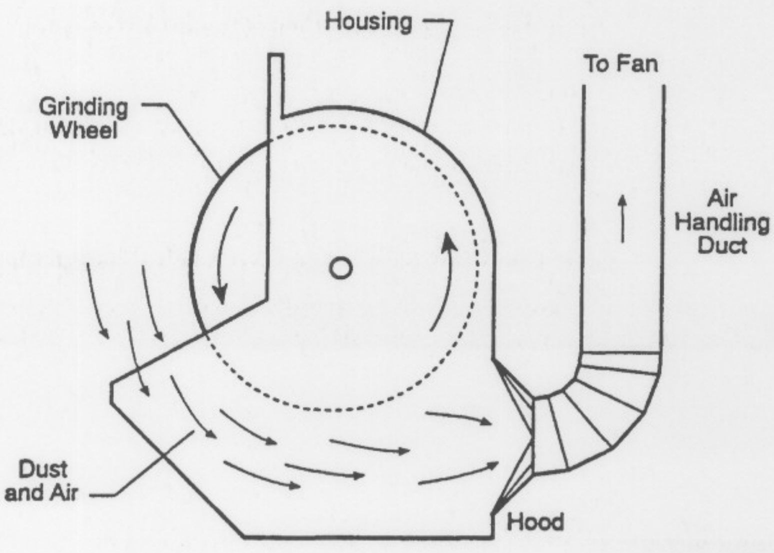


Figure 2-3. Grinding wheel hood

The bag filling process shown in Figure 2-4 also utilizes a receiving-type hood. To avoid interferences with the weighing scale, the hood is mounted above the bag opening to take advantage of the normal vertical travel of emissions and somewhat reduce the capture flow that might be required with another orientation.

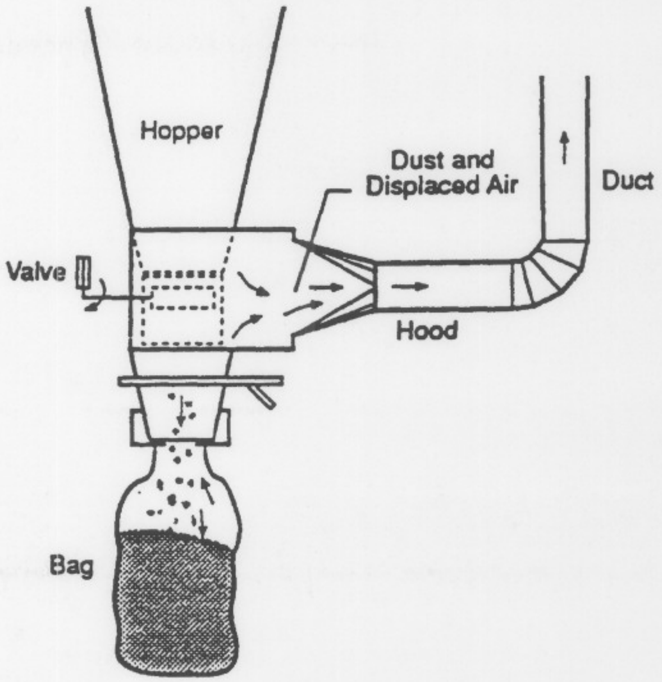


Figure 2-4. Bag filling hood

A hood that is mounted an extended distance away from the contaminant source is referred to as an exterior hood. The principal example of this type hood is the overhead canopy typically employed with hot sources. Because of the limited capability of hoods to capture and draw-in contaminants from a long distance away, this type of hood relies almost totally on the normal movement of the buoyant plume to carry them into the hood. Because the distance of plume travel may sometimes be 20-40 feet, this type hood is particularly subject to losses due to plume meander or cross-drafts that carry the plume out from under the hood. It should also be noted that as a buoyant plume rises it expands because of entrainment of outside air, making exhaust volume requirements the largest of the three hood types.

One important use of overhead canopy hoods is in the control of emissions from electric-arc furnaces used for steel production. A typical system is shown in Figure 2-5, where a canopy hood is used in combination with an enclosure-type hood at the furnace. During the melting cycle, emissions are controlled from the hood mounted on the furnace, with only a small amount of flow drawn from the canopy to remove any contaminants that might escape the furnace. During charging operations, the roof swings off the furnace to provide access, and all air flow is directed to the canopy to collect the significant plume that usually results. In the tapping cycle, the furnace tilts to pour the molten steel into a ladle, disconnecting the furnace hood from the duct through a break-flange arrangement. In the system shown, air flow is then directed to an enclosure hood at the ladle. In other systems, the air flow is again directed to the canopy hood for contaminant capture and removal.

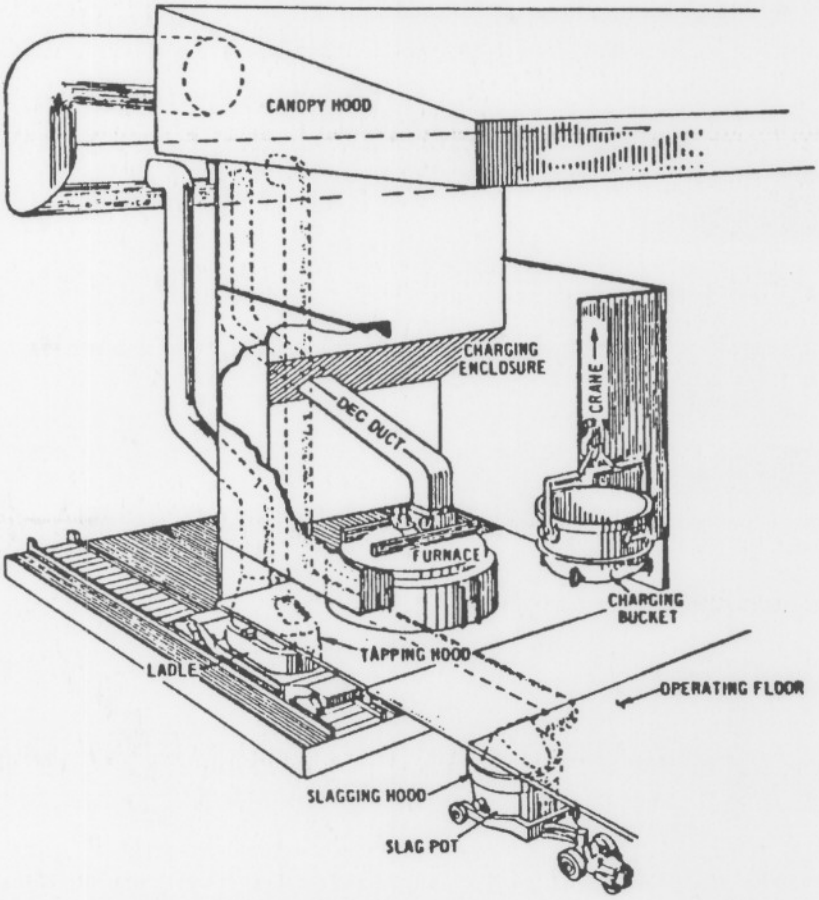


Figure 2-5. Electric arc furnace system (Kemner et al, 1984)

A variation of the exterior hood is the push-pull system shown in Figures 2-6 and 2-7. Here, a controlled jet of air is directed across the contaminant source and in the direction of the exterior hood. The exterior hood primarily serves to receive the air jet and the contaminants carried with it. The advantage of this arrangement is that velocities can be maintained at a higher level with distance from the jet source than can be maintained with distance from the exhaust source, considerably reducing the air volume requirements. However, the open area and air flow of the hood must be careful designed and the system well controlled to avoid dispersing contaminants into the workspace. Also, disturbance of the jet can occur if obstructions are placed in its path, thereby reducing the system's effectiveness.

The various hood designs presented above serve not only to illustrate the four design types but also to indicate some of the variety of designs that are employed. Unfortunately, an all-encompassing presentation of hood designs is not possible, for the variety of hood designs is as large, if not larger, than the variety of industrial processes they control. One important reference that provides information on the design, air flow requirements and resistance characteristics of a large number of industrial hoods is *Industrial Ventilation*, A Manual of Recommended Practice, published by the American Conference of Governmental Industrial Hygienists.

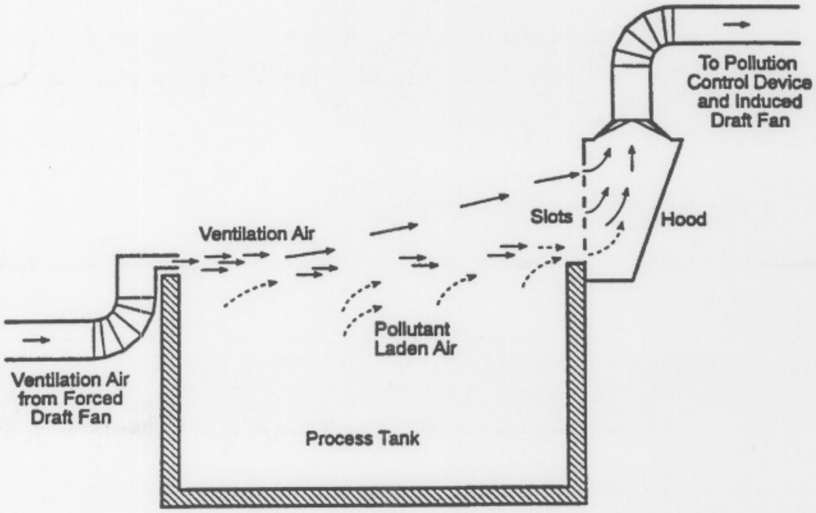


Figure 2-6. Push-pull hood for open tank

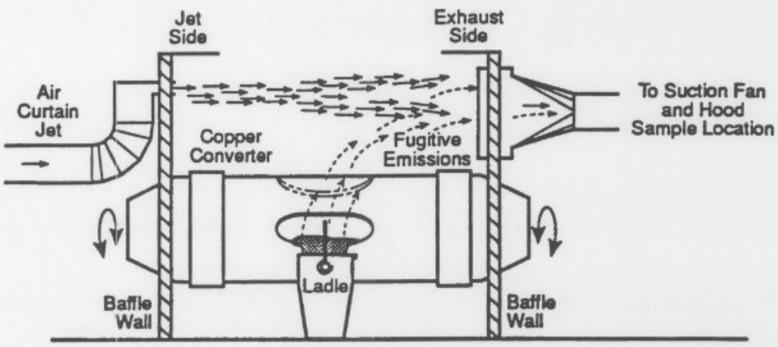


Figure 2-7. Push-pull hood for copper converter

Hood design principles

Although the design of hoods can be a complex process that at times leans more toward art than science, the basic principles that govern that design are surprisingly simple and straight-forward:

- Whenever possible, an enclosure hood should be employed.

- If an enclosure hood cannot be used, the hood should be placed as close to the source as possible and aligned with normal contaminant flow.

- To improve hood performance, duct take-offs should also be placed in-line with normal contaminant flow.

Adherence to these basic principles will result in a hood system that gives high capture efficiency while utilizing the minimum air flow necessary.

Factors affecting hood performance

Effective capture of contaminants by a hood system relies on air flow toward the hood face. This air flow must be sufficient to maintain control of the contaminants until they reach the hood. Of particular concern is external air motion that may disturb this flow and cause loss of the contaminant or require higher than normal air velocities to maintain control. Sources of air motion that must be considered when designing and placing hoods include:

- *Room air currents:* Air currents associated with the workspace ventilation system can become quite large when windows and doors are opened. Currents of as little as 50 feet/min may be enough to affect the performance of some hoods.

- *Thermal air currents:* Air currents from heat generating equipment and processes may affect hood performance. Even low heat releases, such as those from an electric motor, may be enough to cause disturbances.

- *Machinery motion:* Rotating or reciprocating machinery can be a source of significant air currents.

- *Material motion:* Downward motion of material, for example, will create a downward air current that will make the upward motion of contaminants more difficult to achieve.

- *Operator movements:* Rapid movements of an operator can create air currents of 50-100 feet/min.

Capture velocity

Capture velocity is defined as that air velocity at a point in front of a hood or at the hood face that is necessary to overcome existing air currents and cause the contaminated air to move into the hood. The needed capture velocity will depend on both the direction and velocity of the contaminants at the desired point of capture, as well as the level of disturbing air currents that must be overcome. An overhead canopy that relies primarily on plume buoyancy to convey the contaminants to the hood will require little capture velocity, generally just enough to match the plume velocity at the hood face. Contaminants generated by a high energy process that results in rapid and random contaminant motion will require quite high capture rates.

A general guide for appropriate capture velocities in several situations is provided in Table 2-1. Values at the low end of each range would be appropriate when disturbing air currents are low, the toxicity of the contaminants is low, or the hood is large, resulting in a large air mass in motion. The higher end of each range would be more appropriate when air currents are high, the toxicity of the contaminants is high, or the hood is small.

Table 2-1. Range of capture velocities (ft/min)

Type of material release	Capture velocity
With no velocity into quiet air	50-100
At low velocity into moderately still air	100-200
Active generation into zone of rapid air motion	200-500
With high velocity into zone of very rapid air motion	500-2000

Cold flow into hoods

For successful performance, most hood systems rely totally or in part on the ability to provide enough energy to capture a contaminant and draw it into the hood, i.e., to develop the necessary capture velocity. As previously indicated, the capability of an exhaust flow to maintain velocity beyond the hood face is considerably limited. This is illustrated in Figure 2-8, which shows lines of constant velocity as a function of distance from the hood face for both flanged and unflanged hoods. In general, we see that the capture velocity one hood diameter away from the hood face is less that 10 percent of the velocity at the hood face. Although the situation is improved with the addition of a flange, the improvement is only about 25 percent. In contrast, 10 percent of the face velocity of a blowing jet would be found about thirty diameters away. Thus, if one seeks to provide a large capture velocity a significant distance from an exhaust hood, very large and possibly impractical face velocities may be required. In other words, expecting a hood to provide high capture velocity several diameters away from the hood face may be expecting too much.

Figure 2-9 gives relationships for determining air volume requirements to provide a desired capture velocity a given distance from the hood face for several hood configurations. For an existing hood, these same equations could be used to estimate capture velocity once the hood flow rate is known. A method for estimating hood flow rate will be presented later in this chapter, and a method for measuring flow rate will be discussed in Chapter 6.

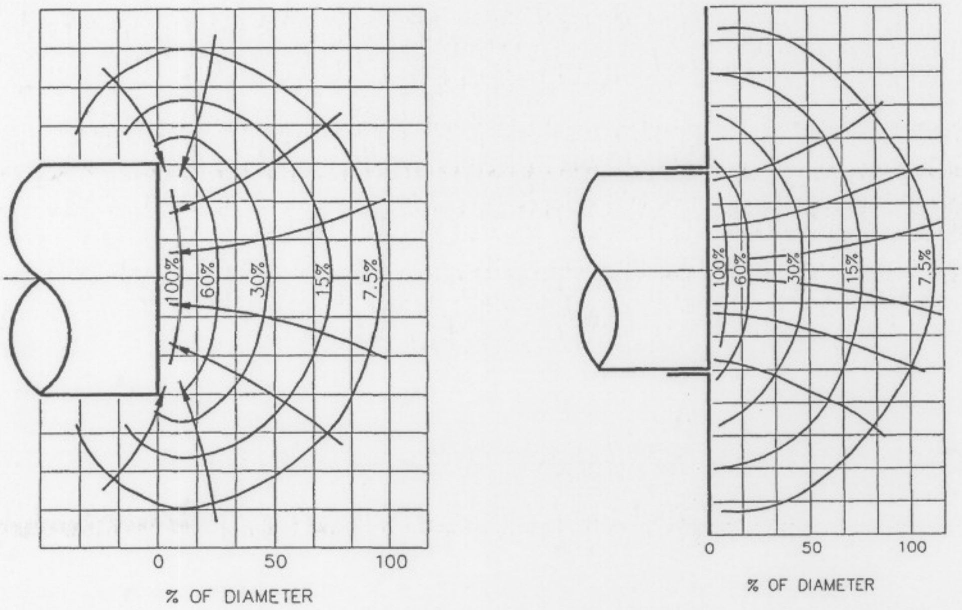


Figure 2-8. Velocity contours for flow into unflanged and flanged hoods (from *Industrial Ventilation*, 20th Edition, 1988, with permission of ACGIH)

Hot flow into hoods

Many hot sources utilize canopy hoods mounted over and well above the source for capture and removal of contaminants. As indicated previously, these hoods rely more on the buoyancy of the hot plume to carry the contaminants into the hood than on the ability to generate a capture velocity. In general, the velocity at the hood face need only be about the velocity of the plume at that point. As a hot plume rises it expands and cools by entraining outside air and its velocity decreases. Thus, as the distance between the source and the hood increases, the air volume required to capture and remove it increases. Also, the slower moving upper portion of the plume becomes more susceptible to being disturbed by air currents, a particular problem with high canopy hoods.

Hood pressure losses

To cause air to move into a hood it is necessary to provide the energy needed to accelerate the air from essentially zero velocity up to the velocity in the duct connected to the hood and to overcome the entry resistance of the hood itself. Consider the hood system shown in Figure 2-10:

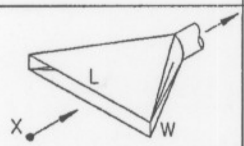
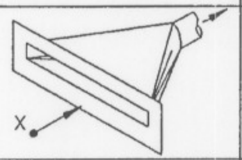
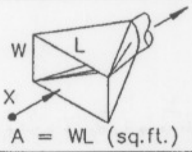
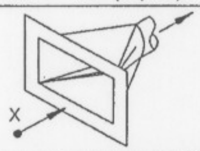
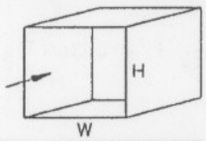
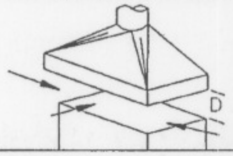
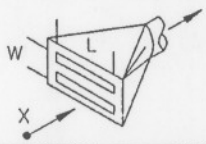
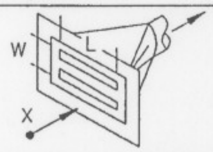
HOOD TYPE	DESCRIPTION	ASPECT RATIO, W/L	AIR FLOW
	SLOT	0.2 OR LESS	$Q = 3.7\ LVX$
	FLANGED SLOT	0.2 OR LESS	$Q = 2.6\ LVX$
	PLAIN OPENING	0.2 OR GREATER AND ROUND	$Q = V(10X^2+A)$
	FLANGED OPENING	0.2 OR GREATER AND ROUND	$Q = 0.75V(10X^2+A)$
	BOOTH	TO SUIT WORK	$Q = VA = VWH$
	CANOPY	TO SUIT WORK	$Q = 1.4\ PVD$ SEE VS-903 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 2-9. Air flow relationships for various hood types (from *Industrial Ventilation*, 20th Edition, 1988, with permission of ACGIH)

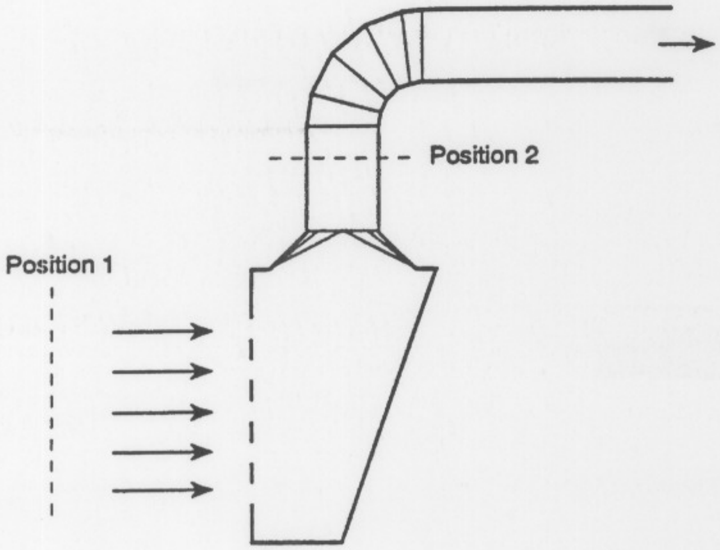


Figure 2-10. Hood system

Applying the Bernoulli equation from Chapter 1 to this situation would indicate that the total pressure at point 1 would equal the total pressure at point 2, or:

$$SP_1 + VP_1 = SP_2 + VP_2 \qquad (2\text{-}2)$$

But, as noted, there is essentially no air motion at point 1; therefore:

$$0 = SP_2 + VP_2 \qquad (2\text{-}3)$$

or

$$SP_2 = -VP_2 \qquad (2\text{-}4)$$

In reality, as air enters a hood or duct a "vena contracta" forms, as illustrated in Figure 2-11. Following the continuity relationship, this reduction in effective cross-section within the vena contracta causes an increase in velocity. Then, as the contracta expands, the velocity decreases. This is not a perfect process and results in an energy loss as static pressure is converted to velocity pressure and then back to static pressure. Therefore, we define the hood static pressure loss, SP_h, as:

$$SP_h = -SP_2 = VP_2 + h_e \qquad (2\text{-}5)$$

where h_e = hood entry loss. Hood entry loss includes entry and frictional losses into the hood and entry losses from the hood into the duct.

The hood entry loss is usually expressed as some fraction of the velocity pressure in the duct attached to the hood:

$$h_e = F_h VP \qquad (2\text{-}6)$$

where F_h = hood loss factor

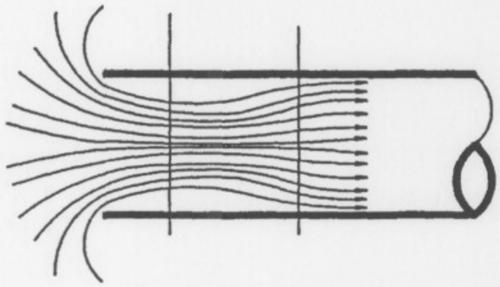


Figure 2-11. Vena contracta

Hood losses may also be described by the hood entry coefficient, C_e:

$$C_e = \sqrt{\frac{VP}{SP_h}} \tag{2-7}$$

The hood entry coefficient can be related to the hood loss factor by recalling that $SP_h = VP + h_e$. Thus:

$$C_e = \sqrt{\frac{VP}{VP + h_e}}$$

$$C_e^2 = \frac{VP}{VP + h_e} \tag{2-8}$$

$$h_e = \frac{VP}{C_e^2} - VP = \left(\frac{1 - C_e^2}{C_e^2}\right) VP$$

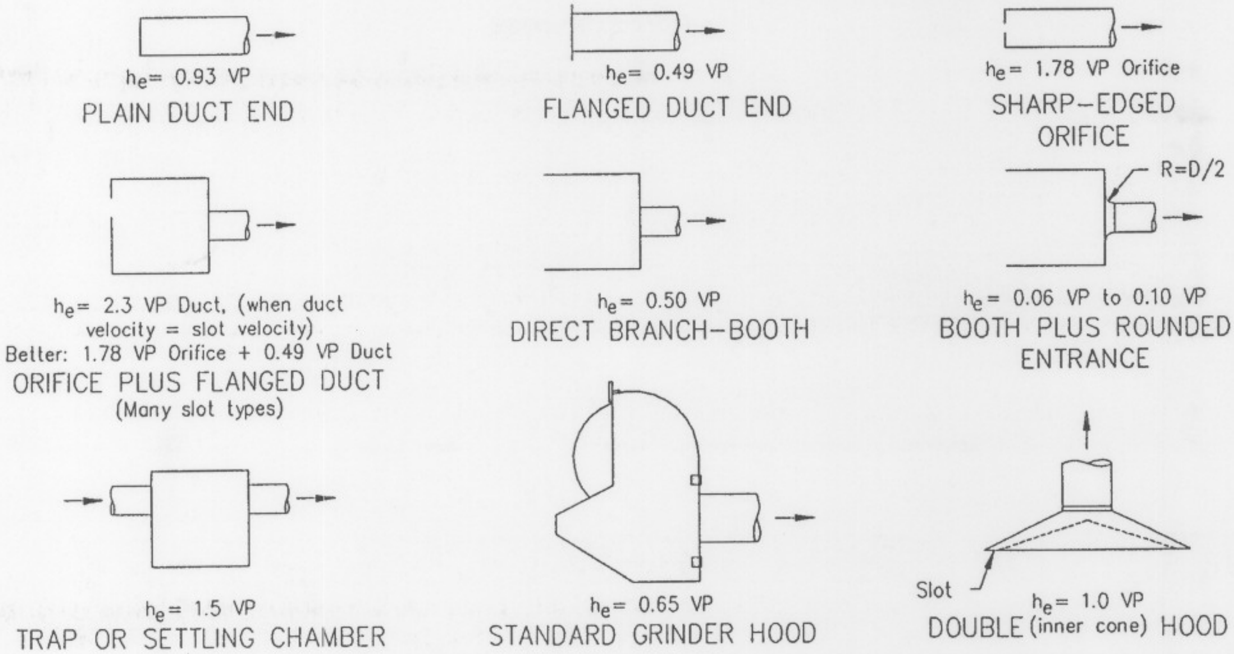
But $h_e = F_h VP$; therefore:

$$F_h = \frac{1 - C_e^2}{C_e^2} \tag{2-9}$$

and

$$C_e = \sqrt{\frac{1}{1 + F_h}} \tag{2-10}$$

Values of F_h are given in Figure 2-12 for several hood configurations.



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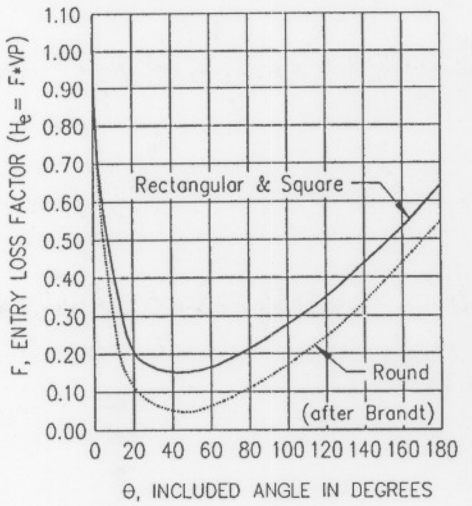
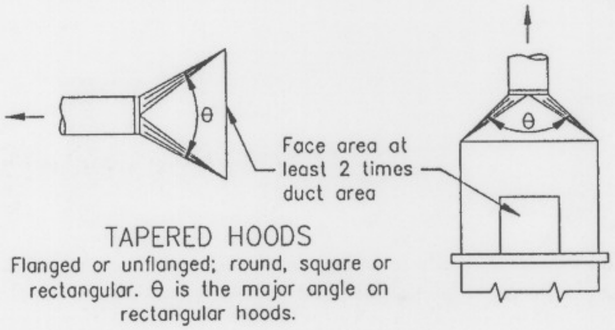
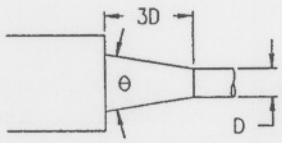




TAPERED HOODS

Flanged or unflanged; round, square or rectangular. θ is the major angle on rectangular hoods.

θ	ENTRY LOSS ROUND	ENTRY LOSS RECTANGULAR
15°	0.15 VP	0.25 VP
30°	0.08 VP	0.16 VP
45°	0.06 VP	0.15 VP
60°	0.08 VP	0.17 VP
90°	0.15 VP	0.25 VP
120°	0.26 VP	0.35 VP
150°	0.40 VP	0.48 VP



MISCELLANEOUS VALUES

HOOD	ENTRY LOSS, F
Abrasive blast chamber	1.0
Abrasive blast elevator	2.3
Abrasive separator	2.3
Elevators (enclosures)	0.69
Flanged pipe plus close elbow	0.8
Plain pipe plus close elbow	1.60

VP = Velocity Pressure in Duct
SP = Static Pressure at Throat, "wg
h_e = Entry Loss, "wg
Q = Volumetric Flowrate, cfm
A = Cross Section at Throat, ft^2

Figure 2-12. Hood entry loss factors (from *Industrial Ventilation*, 20th Edition, 1988, with permission of ACGIH)

Evaluation of hood performance

Measurements of the hood static pressure can be used to estimate the flow rate at the hood. Recalling the relationship between velocity and velocity pressure from Chapter 1 and noting that the flow rate, Q, is equal to the velocity, V, times the cross-sectional area, A, gives:

$$Q = VA = 1096.7A\sqrt{\frac{VP}{\rho}} \qquad (2\text{-}11)$$

Then, recalling that $C_e = \sqrt{\frac{VP}{SP_h}}$, substitution gives:

$$Q = 1096.7AC_e\sqrt{\frac{SP_h}{\rho}} \qquad (2\text{-}12)$$

To use this relationship to estimate hood flow rate, a static pressure measurement would be made in the duct connected to the hood and downstream of the vena contracta, usually about 1-2 duct diameters from the duct connection to the hood. Then the density of the air stream would need to be estimated directly or from a temperature measurement. Finally, the configuration of the hood would be used to determine C_e from reference information like Figure 2-11. Substituting these values, along with the duct cross-sectional area, into the above relationship would yield the flow rate estimate. It should be noted that once an acceptable hood static pressure has been determined, i.e., one that results in adequate capture velocity, subsequent inspections need only confirm this value. Techniques for measuring static pressure and temperature will be discussed in Chapter 6.

As a minimum, the performance of a hood should be visually evaluated. If dusty material is being handled by a process, the amount of fugitive loss provides an excellent indication of the effectiveness of the hood system. Refraction lines due to the escape of gaseous or vaporous contaminants may also be noticed. In addition, the physical condition of the hood should be assessed. Particular attention should be paid to any modifications that have been made to the original hood design or to any damage that it may have sustained that may affect its performance. On movable hoods, the connection between the hood system and the duct system should be assessed to determine the "fit" of the junction. Break-flanges should have a maximum gap of 1-1½ inches. Finally, the hood position should be evaluated to assess the effects of cross-drafts or other air motion on hood capture efficiency.

References

ACGIH, *Industrial Ventilation*, Twentieth Edition, Cincinnati, 1988.

Hemeon, W.C.L., *Plant and Process Ventilation*, Second Edition, Industrial Press, New York, 1963.

Kashdan, E.R., D.W. Coy, J.J. Spivey, T. Cesta and H.D. Goodfellow, *Technical Manual: Hood System Capture of Process Fugitive Particulate Emissions*, EPA-600/7-86-016, April 1986.

Kemner, W., R. Gerstle and Y. Shah, *Performance Evaluation Guide for Large Flow Ventilation Systems*, EPA-340/1-84-012, May 1984.

Chapter 3

Duct Systems

Once contaminants from a process have been captured by the hood system, it is the responsibility of the duct system to convey these contaminants to the collection device and then convey the cleaned air on to its discharge point. In designing duct systems, much of the concern is in selecting proper size ducts and in being sure the system is "balanced" so that the proper quantities of air are drawn from each hood. This involves selecting the proper transport velocity, choosing the duct sizes necessary to maintain that velocity, determining the pressure loss or resistance of each section of the duct system and then being sure that the resistance of each branch entering a junction is the same.

As inspectors, the design of a duct system is of only limited concern. However, it may be necessary to use some of the same tools as the designer in order to accomplish our goals. For example, suppose we wish to estimate the flow into a high canopy hood. From Chapter 2, we know that this can be done by measuring the static pressure in the duct just beyond the hood. In this case, however, that would require obtaining a measurement from a difficult, and perhaps dangerous, location to reach. Instead, we could measure the static pressure at some more easily reached and safer location and use the principles of duct resistance to estimate the air flow we require.

In this chapter, we will apply Bernoulli's equation to determine the relationship between pressures at different points in a duct system. This relationship will then be modified to account for losses associated with the fluid actually having viscosity, and techniques for estimating these losses will be presented. Also, transport velocity will be defined and its significance discussed. Finally, the concepts behind balancing ventilation systems will be presented.

Duct pressure loss

Consider the following duct segment shown in Figure 3-1. Applying Bernoulli's equation to points 1 and 2, we can write:

$$TP_1 = TP_2 \tag{3-1}$$

However, because of friction between the gas stream and the duct walls and because of non-ideal conversion between static pressure and velocity pressure as the gas stream accelerates or decelerates, pressure losses between the two points occur. Thus:

$$TP_1 = TP_2 + h_L \tag{3-2}$$

or

$$SP_1 + VP_1 = SP_2 + VP_2 + h_L \tag{3-3}$$

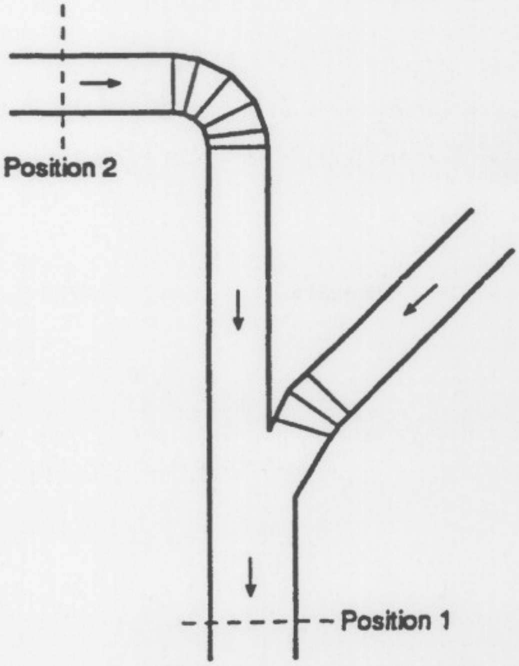


Figure 3-1. Duct segment

If we assume that the velocity between points 1 and 2 is approximately constant, then $VP_1 = VP_2$ and:

$$SP_1 = SP_2 + h_L \tag{3-4}$$

Here, h_L is the total pressure loss due to friction and non-ideal pressure conversions. For calculation purposes, we divide these losses into three categories: (1) frictional losses in straight duct, (2) fitting losses, and (3) acceleration losses. The frictional loss in straight duct is rather self-explanatory and simply involves the loss due to friction with the walls. Fitting losses occur when the gas stream flows through elbows, entries, transitions and other types of fittings. This loss results from friction with the walls of the fitting and with increased energy loss due to an increased level of turbulence. Acceleration losses (or gains) are associated with changes in the velocity of the gas stream. Accelerating a gas stream requires the input of energy, while decelerating a gas stream may result in a gain of energy. The amount of pressure loss or gain depends on the relative abruptness of the change. Smoother accelerations result in lower losses and smoother decelerations result in higher gains.

Because our interest in duct losses will usually be over relatively short distances, our determination of losses will be confined to those associated with straight ducts and fittings. There are three techniques in general use for estimating these losses: (1) the equivalent length method, (2) the velocity pressure method, and (3) the total pressure method. In the equivalent length method, fitting losses are expressed in length of equivalent straight duct. Losses are calculated by adding the equivalent length for the fittings to the actual length of straight segments and then multiplying by a factor that expresses the pressure loss per length of duct. Since the equivalent length of fittings depends on the duct size, the amount of information required to conduct an equivalent-length calculation can be large.

In the velocity pressure and total pressure methods, both straight duct and fitting losses are expressed as a factor times the velocity pressure or total pressure in the duct segment, respectively. Since the fitting loss factor does not vary with the duct size, the amount of information required for calculation is considerably reduced in comparison to the equivalent-length approach.

The technique to calculate duct system losses that is used in this course is the velocity pressure method. With this method, frictional losses in straight duct are expressed as:

$$h_{L1} = H_f L \cdot VP \tag{3-5}$$

where H_f = velocity pressure loss per foot of duct
L = length of duct (ft)

The straight-duct loss factor can be determined from the following empirical equation:

$$H_f = 0.0307\frac{V^{0.533}}{Q^{0.612}} = \frac{0.4937}{Q^{0.079}D^{1.066}} \tag{3-6}$$

where V = air velocity (ft/min)
Q = volumetric flow rate (ft^3/min)
D = duct diameter (in.)

For rectangular ducts, D is expressed as an equivalent diameter determined from:

$$D_{eq} = 1.3\frac{(A \times B)^{0.625}}{(A+B)^{0.250}} \tag{3-7}$$

where D_{eq} = equivalent diameter (in.)
A = length of one side (in.)
B = length of adjacent side (in.)

Using the equivalent diameter, an equivalent velocity is calculated. The loss factor is then calculated using Equation 3-6 and the equivalent values.

Fitting losses in the velocity pressure method are expressed as:

$$h_{L2} = F \cdot VP \tag{3-8}$$

where F is the fitting loss factor. There are a variety of fittings used in ventilation systems and factors for their resistance can be found in a number of reference texts (e.g., ACGIH, 1988; SMACNA, 1977). The fitting of most interest for short distance calculations is that for elbows. Fitting loss factors for 90° round elbows are given in Table 3-1 in terms of the ratio of the radius of turn to the duct diameter, R/D. Most round elbows have an R/D of 1.5 or 2.0, with 2.0 being the most common. For rectangular elbows, loss factors are given in Table 3-2 in terms of the duct aspect ratio, W/D, and the elbow R/D. Here, D is the length of the side parallel to and centered on the turning radius, and W is the length of the adjacent side. Resistances for other than 90° elbows are determined as a percentage of the 90° elbow resistance. Thus, the resistance of a 60° elbow is two-thirds that of a 90° elbow, while the resistance of a 45° elbow is half.

Table 3-1. Loss factors for 90° round elbows

R/D	Fraction of VP loss
1.25	0.55
1.50	0.39
1.75	0.32
2.00	0.27
2.25	0.26
2.50	0.22
2.75	0.26

Table 3-2. Loss factors for 90° rectangular elbows

	Aspect Ratio, W/D					
R/D	0.25	0.5	1.0	2.0	3.0	4.0
0.0	1.50	1.32	1.15	1.04	0.92	0.86
0.5	1.36	1.21	1.05	0.95	0.84	0.79
1.0	0.45	0.28	0.21	0.21	0.20	0.19
1.5	0.28	0.18	0.13	0.13	0.12	0.12
2.0	0.24	0.15	0.11	0.11	0.10	0.10
3.0	0.24	0.15	0.11	0.11	0.10	0.10

Velocity pressure calculation method

In general, calculating the pressure loss from one duct location to another requires determining the loss of the straight sections and the loss of all the fittings and then adding them together to get the total loss. The specific step-wise procedure for a segment, beginning at the hood, is as follows:

1. Determine the duct velocity and calculate the velocity pressure.

2. Determine the hood static pressure.

3. Multiply the straight duct length by the friction loss factor.

4. Determine the number and type of all fittings. Multiply the loss factor for each type of fitting by the number of that type and sum for all types.

5. Add the results from Steps 3 and 4 and multiply the result by the velocity pressure from Step 1.

6. Add the result from Step 5 to the hood static pressure from Step 2. If there are other losses (expressed in inches of water column), add them also.

This calculation procedure gives the total energy, expressed as static pressure, that is needed to move the gas volume through the duct segment.

Estimating hood flow rate

In the introduction, it was noted that one could measure the static pressure at some safer or more easily reached location and use the principles of duct resistance presented in this chapter to estimate the air flow at a hood. Since this application is not altogether obvious, the relationship for doing this will be developed here.

Recall from Chapter 2 the relationship for the hood entry loss coefficient, C_e:

$$C_e = \sqrt{\frac{VP}{SP_h}} \tag{3-9}$$

or

$$SP_h = \frac{VP}{C_e^2} \tag{3-10}$$

The hood static pressure could be determined by making a measurement anywhere in the duct leading to the hood and then correcting the measured value for the losses between the measurement location and the hood. Assuming losses due to both straight duct and fittings:

$$SP_h = SP_{meas} - H_f L \cdot VP - \Sigma F \cdot VP \tag{3-11}$$

Equating the two relationships for hood static pressure gives:

$$\frac{VP}{C_e^2} = SP_{meas} - H_f L \cdot VP - \Sigma F \cdot VP \tag{3-12}$$

or

$$VP = \frac{SP_{meas}}{\frac{1}{C_e^2} + H_f L + \Sigma F} \tag{3-13}$$

To obtain H_f a trial velocity will have to be assumed. The corresponding velocity pressure would be compared to the calculated value from Equation 3-13 and the calculations repeated until reasonable agreement is obtained. Once an acceptable velocity pressure has been determined, the flow rate can be calculated from:

$$Q = VA = 1096.7A\sqrt{\frac{VP}{\rho}} \tag{3-14}$$

or, if standard air is involved:

$$Q = 4005A\sqrt{VP} \tag{3-15}$$

Transport velocity

The velocity maintained within a duct segment is referred to as the transport velocity. Typical design values are given in Table 3-3. For systems conveying vapors, gases or smoke, the velocity chosen by the designer is based on a compromise between fan energy cost and duct cost. Large diameter ducts result in lower pressure losses and reduced fan energy but cost more than smaller diameter ducts. In general, this economically optimum velocity is around 1000-2000 feet/minute.

For systems handling particles, a minimum velocity is required to prevent settling. The value of this minimum velocity increases as the size or density of the particles increases. Should settling occur, the resistance of the duct will increase due to the reduction in effective flow area. If the deposited material is dry and loose, then an equilibrium condition will develop when the level of deposited material causes the velocity to increase to that needed to re-entrain. At this point the build-up will cease, but there will be a decrease in volume in that segment due to the increased resistance.

If, however, the material is sticky or tends to form solid cake, the deposition may continue until the duct is completely plugged.

Table 3-3. Range of design transport velocities	
Contaminant	Design velocity (ft/min)
Vapors, gases, smoke	Any (usually 1000-2000)
Fume	1400-2000
Very fine, light dusts	2000-2500
Dry dusts and powders	2500-3500
Average industrial dusts	3500-4000
Heavy dusts	4000-4500
Heavy or moist dusts	4500 and up

In addition to increasing duct resistance, deposition also increases the weight of a duct and may cause the duct system supports to fail. Also, hardened material deposited inside the duct may break loose as a result of vibration and travel down the duct to the fan or other equipment, causing damage.

Another concern of particle conveying systems is abrasion of the duct surface, a potential that increases with increase in transport velocity. This is a particular problem wherever there is a change in direction of the gas stream, such as at elbows or entries. The particles traveling with the gas stream possess a certain amount of inertia that will tend to carry them straight ahead as the gas stream turns. When this material strikes the duct walls, erosion can produce holes. If the duct is under negative pressure, air will enter the system through the holes, reducing the amount of air that enters at the hood, possibly causing loss of capture efficiency. If the duct is under positive pressure, fugitive emissions may result as air exits the holes. Also, segments that have reduced air flow because of holes may become susceptible to build-up problems due to the loss of transport velocity.

Somewhat related to transport velocity is the problem of duct corrosion. As the gases within a duct cool, condensation of moisture and/or acidic material can occur, depending on the gas stream composition. This cooling may be the result of infiltration of outside air or simply due to long residence times in the duct, which can be exacerbated by low transport velocities. Condensed material will tend to accumulate along the bottom of the duct and cause its first damage there. Visual inspections should concentrate on this area when corrosion damage is suspected.

The best technique for locating holes in a duct system is a visual inspection. To make more effective use of inspection time, areas on an elbow that are on the outside of the turn, areas on a straight duct opposite points of entry by other ducts, and areas along the bottom of horizontal ducts should be emphasized. On hot gas streams, a significant drop in temperature between locations along a duct could also be used to locate holes in negative pressure systems, as could an increase in oxygen concentration if the source is oxygen deficient. However, because of the time and equipment investment needed to make such measurements, they are not likely to be practical in most situations.

Unfortunately, inspecting for material build-up in a duct system cannot be done effectively without making measurements, since the accumulation is not visible from the outside. Measured static pressure differences between locations along a duct could be compared to expected values estimated using the techniques described above. Significant differences between measured and calculated values would indicate the location of a build-up. Alternatively, measurements of static pressure along a duct could be plotted as a function of equivalent length (the equivalent length of various fittings used in ventilation systems can be found in reference texts such as *Industrial Ventilation*). If there are no obstructions, the measurements will produce a straight line of constant slope, with the values decreasing (becoming more negative) in the direction of gas flow. Any unexplained deviation from this slope would indicate an area of accumulation.

Finally, it should be noted that physical damage to ducts can also increase their resistance. A duct that has been partially collapsed acts much the same as a duct that has accumulated material. The reduced cross-sectional area causes the velocity to increase as it moves through the damaged area and then decrease as it moves out of it, adding to the duct losses. Also, depending on the nature of the damage, the frictional resistance of the damaged section may be increased. Observations of physical damage should be included in the visual inspection of a duct system.

Balancing duct systems

As indicated in the introduction, balancing a branched duct system is the role of the designer and is done to insure that the correct air volumes are drawn from all hoods that are connected to a common system. As inspectors, it is important to have some understanding of how this is accomplished and the limitations of the various techniques.

The fundamental rule in balancing a duct system is that all ducts entering a junction must have equal static pressure requirements. Consider the two-hood duct system shown in Figure 3-2. At the junction, the static pressure in the longer duct that leads to Source 1 is 2.0 in. H_2O, while the static pressure in the duct leading to Source 2 is 1.5 in. H_2O. Since it is not physically possible to maintain two static pressures at one location, this condition cannot prevail. If nothing is done, the system will adjust itself by reducing the flow rate from Source 1 and increasing the flow rate from Source 2 until the same static pressure requirement results. However, the reduced flow rate at Source 1 may not be sufficient to prevent loss of contaminants. To prevent this situation, the designer must adjust the resistances of each branch so that they equal each other at the junction, while maintaining design flow rates.

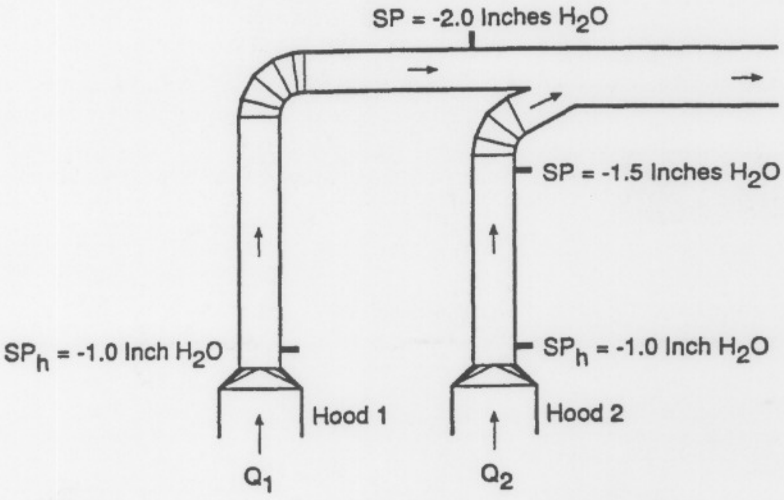


Figure 3-2. Branched duct system

Static pressure balance at a junction can be achieved through the re-design of one or both branches or by inserting a damper into the duct with the lower static pressure to raise its value up to that of the other duct. In the re-design option, changes are made in duct diameter, duct length, elbow radius, etc., or in the air volume in order to achieve matching static pressures.

Each approach to duct balancing has advantages and disadvantages. Some of the characteristics of systems balanced through design include:

1. Since the system resistance is fixed, air volumes cannot be easily changed.

2. The system has limited flexibility for future equipment changes or additions.

3. Because nothing protrudes into the gas stream, there should be no unusual erosion or accumulation problems.

4. Since balance may be achieved by making small, but acceptable, changes in hood volume, the total volume from a multiple source system may be quite different than the original design.

5. The system information must be detailed, so that branch losses may be determined accurately.

6. The system must be installed exactly as it is designed. Any deviation will change the resistance of a branch and the flow rates in the system.

Because of the need for flexibility in design and installation, most industrial ventilation systems are balanced by the use of dampers. The most popular damper is the blast gate, which is simply a flat metal blade inserted into the duct a sufficient distance to produced the desired loss. Positioning of the blades can be determined through trial-and-error measurements on the installed system or by calculation of the amount of resistance that must be added. Some of the characteristics of systems balanced with dampers include:

1. Over a limited range, the air volumes may be easily changed by changing the positions of the dampers.

2. The system is flexible for future changes or additions.

3. The damper may cause material accumulation in the duct or may be eroded.

4. Since no volume changes are required for balancing, the total volume will be the same as the original design.

5. If balance is to be achieved by trial-and-error positioning of the dampers, branches that obviously have lower resistance need not be calculated. However, determining the branch that has the greatest resistance is critical.

6. Moderate variation from the design during system installation is acceptable, particularly if the system is to be balanced by trial-and-error.

References

ACGIH, *Industrial Ventilation*, Twentieth Edition, Cincinnati, 1988.

Crowder, J.W., and K.J. Loudermilk, "Balancing of industrial ventilation systems", *JAPCA*, ***32***, 115 (1982).

Kemner, W., R. Gerstle and Y. Shah, *Performance Evaluation Guide for Large Flow Ventilation Systems*, EPA-340/1-84-012, May 1984.

SMACNA, *HVAC Duct System Design*, First Edition, Vienna (Virginia), 1977.

Chapter 4

Gas Cooling Systems

Many processes generate gas streams at temperatures that are too high for some air pollution control devices to accept. Because of this, it is necessary to employ some type of cooling device to reduce gas temperature. Since the gases must pass through the cooling device, it is an integral part of the ventilation system. This equipment will add to the resistance of the system and may change the volume and composition of the gases. If it is not functioning properly, it can affect the performance of other components in the system. Thus, the inspector should be familiar with these devices and be able to evaluate their performance in the field.

In this chapter methods of cooling industrial exhaust streams will be described, and the fundamentals governing their performance will be presented. Parameters that affect this performance will be discussed, along with procedures for conducting field inspections.

The most commonly used methods for cooling gases in industrial ventilation systems are: (1) dilution with ambient air, (2) quenching with water, and (3) natural convention and radiation from ductwork. In a limited number of cases, forced convection systems using air or water for cooling, may be encountered. The following discussion will focus on the three more common cooling methods.

Dilution with ambient air

Cooling gases by dilution with ambient air is the simplest method that can be employed. With this technique, hot gases from a process are cooled by adding ambient air in sufficient quantity to produce a mixture with the desired temperature. The fundamental relationship governing performance may be developed through a heat balance on the dilution system:

$$m_1h_1 + m_2h_2 = m_3h_3 \tag{4-1}$$

where
m_1 = mass flow rate of hot gases
h_1 = enthalpy of hot gases
m_2 = mass flow rate of dilution air
h_2 = enthalpy of dilution air
m_3 = mass flow rate of gas mixture = $m_1 + m_2$
h_3 = enthalpy of gas mixture

Recalling from Chapter 1 that enthalpy is a function of the temperature of the gas, we see that the quantity and temperature of the hot gases and the desired temperature of the mixture are the parameters that control the design. The success of the design will depend on the quantity of dilution air supplied in relation to these parameters.

Dilution cooling is used extensively when hot gases discharged from a process are collected in an exterior hood, such as a canopy. One such system is illustrated in Figure 4-1. In this case, the amount of air volume needed to insure capture and removal of the plume is generally sufficient to cool the gas stream to desirable temperatures. The technique may also be appropriate when enclosure or receiving hoods are used, if the hot gas stream is small. Here, the dilution air could be introduced through a branch into the hot duct or by a combination of capture and introduced air. However, when the hot gas stream is large, the quantity of dilution air becomes excessively large, making the cost of the downstream exhaust system and control device uneconomical.

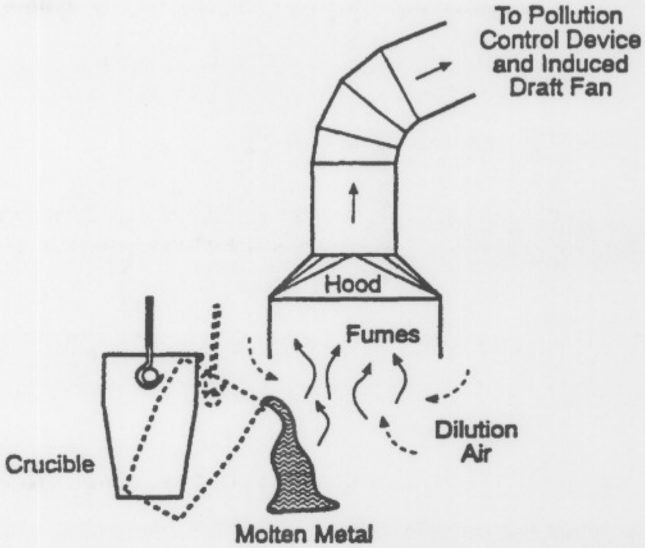


Figure 4-1. Canopy hood system

One problem with dilution cooling has already been noted, and that is that large air volumes may be required to accomplish the desired temperature reduction. Problems with temperature control may also be experienced if the quantity of dilution air is not regulated. This would only be possible if the dilution air were introduced through a branch duct where a damper could be used to modulate the flow. Here, a feed-back control system could be used to maintain some pre-set temperature. When the temperature is not controlled, corrosion problems are possible if acid or moisture dew points are reached.

The inspection of a dilution cooling system is rather straight-forward. The system should first be visually inspected to evaluate the integrity of the system and to determine if there are any indications of corrosion problems. Next, if the temperature of the mixed gas stream is being monitored, this value should be noted and evaluated in terms of compatibility with downstream equipment, particularly the control device, and with regard to moisture or acid condensation potential. If controlled introduction of the dilution air is employed, the condition and operation of the controller and damper should be evaluated and the set-point temperature compared with any monitored values. If temperature is not monitored, measurements in the mixed gas stream may be necessary to complete the evaluation.

Measurements on a mixed gas stream that employs a branch duct to introduce the cooling air could also be used to estimate the volume of air coming from the process and the volume used for dilution. To accomplish this, it would be necessary to measure the volumetric flow rate and temperature of the mixed stream and the temperature of the hot gas stream and the dilution air (usually ambient air). Next, the volumetric flow rate of the mixture would be converted to a mass flow rate using the density corresponding to the measured temperature, and the enthalpies of the three streams would be determined based on their temperature. These values would then be substituted into the heat balance shown previously to determine the mass flow rate of the hot gas and the dilution air, as follows:

$$m_{hot}h_{hot} + (m_{mix} - m_{hot})h_{dilution} = m_{mix}h_{mix} \tag{4-2}$$

or

$$m_{hot} = m_{mix}\left(\frac{h_{mix} - h_{dilution}}{h_{hot} - h_{dilution}}\right) \tag{4-3}$$

and

$$m_{dilution} = m_{mix} - m_{hot} \tag{4-4}$$

The mass flow rates would then be converted to volumetric flow rates using the densities corresponding to their respective temperatures. Although this procedure is somewhat involved, it is considerably simpler and less time consuming than measuring volumetric flow rate in all three streams, if that information is needed. The method for measuring volumetric flow rate will be discussed in Chapter 6.

Quenching with water

When the volume of hot gases is large and the amount of air needed to capture them is small, some cooling method other than dilution with ambient air is needed. Since evaporation of water requires a large amount of heat, the gas under these circumstances can be effectively cooled by simply spraying water into the hot gas stream. The fundamental relationship governing performance may again be developed through a heat balance on the evaporation system:

$$m_{gas}(h_{gas\ in} - h_{gas\ out}) = m_{water}(h_{water\ vapor} - h_{water}) \tag{4-5}$$

Again we see that the quantity and temperature of the hot gases and the desired outlet temperature are the parameters that control the design. The success of the design will depend on the quantity of water supplied in relation to these parameters and the efficiency of its evaporation.

The design of a quench chamber will, in general, depend on how critical it is that all of the water be evaporated. In systems where a wet scrubber is employed, it is only important that the temperature of the gas stream be reduced below the vaporization temperature of the scrubbing liquid. Since liquid carry-over is not a concern, the efficiency of the evaporation equipment need not be particularly high. Indeed, the evaporative cooler may be little more than a series of spray nozzles mounted in the duct leading to the scrubber.

In systems where water carry-over is a concern, a separate piece of equipment is likely to be used for evaporative cooling. Here, the gas stream velocity will be reduced to about 500-700 feet/minute, and the liquid introduced through a series of spray nozzles at pressures ranging from 50 to 150 psig in order to produce a fine spray. Where any water carry-over is undesirable, as in a fabric filter system, even lower velocities will be used and liquid pressures as high as 400 psig may be employed. In some cases, air-atomized nozzles may be used to reduce the droplet size produced and enhance evaporation. These techniques will likely be accompanied by an elaborate control system to assure that all of the spray water is evaporated before exiting the cooler. This is sometimes referred to as "dry bottom" operation.

In addition to water carry-over, another problem associated with evaporative coolers is corrosion. Since the coolers utilize water sprays, the potential for moisture and condensed acid corrosion is significant, particularly in systems where all of the water is not evaporated. The designer usually seeks to mitigate these problems through the use of appropriate corrosion resistant materials and linings.

Temperature control may also be a problem with evaporative coolers. This can occur if the rate of water addition is not controlled or if the control system is unable to respond to the rate of change of the gas stream temperature. Temperature control can also be a problem if the water atomization is not efficient. This can occur as a result of changes in nozzle performance because of erosion or plugging.

Inspection of evaporative coolers begins by first conducting a visual inspection to evaluate system integrity and indications of corrosion. If the outlet temperature is being monitored, this value should be noted and evaluated in terms of compatibility with downstream equipment. If temperature is not monitored, it may be necessary to measure it; however, this measurement may be complicated by the presence of water droplets in the gas stream. Techniques for dealing with this situation will be discussed in Chapter 6.

Next, some determination of the quantity of cooling water used should be made. The most desirable way of doing this would be to read the value indicated by a flowmeter mounted on the delivery line or in the control room. Since it is likely that such a meter will not be available or may not be functioning properly if it is available, another method to evaluate water flow rate is needed. One technique would be to observe the delivery pressure on a gauge mounted at the cooler or on the pump. The quantity of water delivered varies with the square-root of the pressure. However, to be certain that any observed changes in pressure are due only to a change in water flow rate, the condition of the nozzles must also be evaluated. If the nozzles are plugged, the pressure gauge will indicate an increase in pressure. If this condition were not known, the increase in pressure might be interpreted as an increase in water flow, when the flow rate has probably decreased. Similarly, eroded nozzles would cause a decrease in pressure, which might be interpreted as resulting from decreased water flow.

Plugged nozzles can be determined by observing the spray pattern during operation. If a viewport is not available, this will require plant personnel to remove an inspection plate, if one is installed. Eroded nozzles may also exhibit a different spray pattern, so a physical inspection of the nozzle may be needed to distinguish between the two problems. Since this should also be done by observing from outside the cooler, the results may not be very rewarding. To improve your chances of observing a damaged nozzle, you should ask plant personnel to bring you a new one from their spare-parts inventory for comparison.

Finally, if the water used in the evaporative cooler is recycled, its quality should be evaluated. Water containing large particles would be of concern because of the potential for plugging or eroding the nozzles. If the water contains small particles, there would be concerns about passing these more difficult to collect particles on to the control device after the water has been evaporated. The quality of the water can be evaluated by having plant personnel draw a sample into a clear plastic container that you provide. The sample should be well mixed by shaking and then allowed to settle. If the rate of settling is fairly rapid, then the water contains large particles. If the settling rate is very slow, the water contains fine particles.

Natural convection and radiation

When a hot gas stream flows through a duct, the duct becomes hot and heats the surrounding air. As the air near the duct becomes heated, convection currents develop that carry the heat away. This phenomenon is referred to as natural convection and may be aided by a small amount of forced convection due to wind motion. Heat may also be transferred from the duct surface by direct radiation. This behavior can be exploited to produce significant amounts of cooling by providing a section of duct that has large surface area. This is typically done by arranging the duct in a series of vertical columns, as shown in Figure 4-2. Since the velocities through the columns are usually below the necessary transport velocity for particles, the base of alternate columns are joined across a hopper for collection and removal of any settled dust.

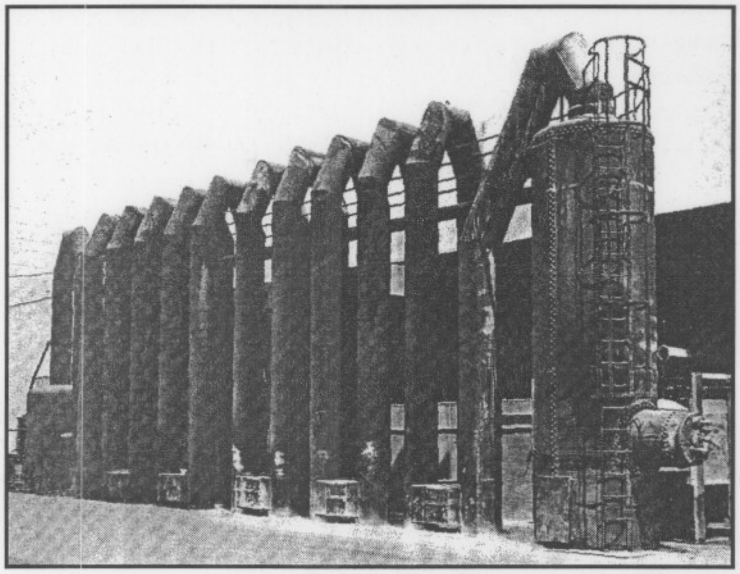


Figure 4-2. Convection and radiation cooler (Danielson, 1973)

The fundamental relationship governing the performance of the cooler may again be developed through a heat balance:

$$m_{gas}(h_{gas\ in} - h_{gas\ out}) = UA\Delta T_m \qquad (3\text{-}6)$$

where U = overall heat transfer coefficient
A = heat transfer area
ΔT_m = log-mean temperature difference

In the English system, the overall heat transfer coefficient, U, has units of Btu/hr-ft^2-°F, and represents the ability to transfer heat. It is a function of a number of parameters, including the duct diameter, the nature of the duct surfaces, the thermal conductivity of the metal, the velocity of the hot gases, the wind speed and the temperature difference between the duct and the ambient air. The log-mean temperature difference, ΔT_m, is simply a means of calculating an average difference when that difference varies along the length of the duct. Once again we see that the quantity and temperature of the hot gases and the desired outlet temperature are the parameters that control the design. The success of the design will depend on all the parameters that affect the heat transfer coefficient and on the total surface area provided.

There are a number of problems associated with convection and radiation cooling systems. First, because of the need to provide large amounts of surface area for heat transfer, the size of the cooler is usually quite large, requiring significant amounts of plant area for its installation. Also, because the velocities in the cooler are generally below the transport velocity for particles, the system must be cleaned continually to avoid build-up in the hopper sections. Finally, because there is essentially no control on the cooling process, it is not possible to control the outlet temperature. Depending on the temperature of the gases coming from the process, this could result in outlet temperatures that exceed the limitations of downstream equipment or that decrease into the range of moisture or acid condensation. This latter situation could lead to corrosion of the cooler surfaces, resulting in infiltration of outside air or the escape of fugitive emissions.

Because of the nature of cooler design and operation, the items that can be inspected on these systems is limited. As with the other devices, the convection and radiation cooler should first be visually inspected to evaluate the integrity of the system and to determine if there are any indications of corrosion problems. Particular attention should be paid to the dust removal doors, which should be checked for leakage. Next, evaluate the dust removal operation by having plant personnel open a few of the doors. If the temperature of the outlet gas stream is being monitored, this value should be noted and evaluated in terms of compatibility with downstream equipment and with regard to moisture or acid condensation potential. If temperature is not monitored, measurements in the outlet gas stream may be necessary to complete the evaluation.

References

Danielson, J.A., ed., *Air Pollution Engineering Manual*, Second Edition, EPA AP-40, May 1973.

Segal, R., and J. Richards, *Inspection Techniques for Evaluation of Air Pollution Control Equipment*, Volume II, EPA-340/1-85-022b, September 1985.

Chapter 5

Fan Systems

One of the most critical parts of an industrial ventilation system is the air mover or fan. Its function is to cause the desired amount of air to move through the system by overcoming resistances in the hoods, ducts, coolers, collection devices, stacks and any other equipment present. The fan is also one of the most complex pieces of equipment in the ventilation system. Its performance depends on the type of fan employed, the parameters of its operation, the characteristics of the system it is used in, and the properties of the gas stream it operates on. To be able to effectively inspect fan systems, the inspector must be familiar with how these various factors influence fan performance.

In this chapter the types of fans usually employed in industrial ventilation systems will be described, and the design and operating characteristics that affect their performance will be discussed. Next, the way a fan interacts with the rest of the ventilation system to determine the air volume that is moved will be described, and the manner by which changes in either the fan or the system characteristics interact to change the flow rate will be discussed. Also, installation conditions that affect fan performance will be described and techniques to evaluate their effect will be presented. Finally, the procedures used by designers in selecting a fan will be described, and techniques that can be used to evaluate the performance of an existing fan, involving variations on these procedures, will be presented.

Types of fans

A fan can be generally characterized on the basis of its location in the ventilation system with respect to the control device, as shown in Figure 5-1. Fans located before the control device are referred to as forced draft because they force or push the air through the collector. In this location, the fan acts on the dirty gas stream and may be subject to increased wear and require a higher level of maintenance, thereby increasing operating costs. The control device, however, will only have to withstand the pressure required to push the gas stream through the device and on to the stack. As a result, the collector will require less structural reinforcing and will likely be somewhat cheaper.

Fans located after the control device are referred to as induced draft because they induce or pull the flow through the collector. In this location, the fan acts on the cleaned gas stream and would be less subject to wear. This would likely require a lower level of maintenance, reducing operating costs. However, the control device will now have to be structurally reinforced to withstand essentially the entire negative pressure of the system and will probably be more expensive.

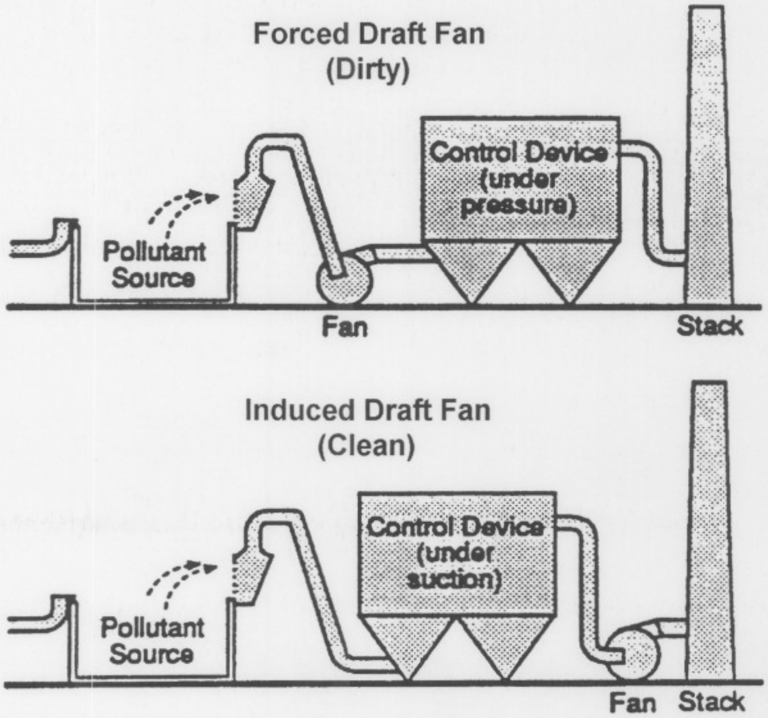


Figure 5-1. Fan Locations

Fans designs can be classified as either axial or centrifugal. A special class of fan that employs a centrifugal wheel mounted in an axial arrangement will sometimes be encountered in industrial ventilation systems. As will be discussed later, its performance is determined by the wheel design, rather than the orientation, and it generally behaves like other centrifugal fans.

Axial fans are used to move large volumes of air against low resistances. They may be used for general ventilation or in low resistance industrial ventilation systems; however, they are not used very often in air pollution control systems. Occasionally, an axial fan will be used in combination with the more common centrifugal fan to provide extra energy to overcome resistances.

There are three basic types of axial fans: (1) propeller, (2) tubeaxial and (3) vane axial. The propeller fan is used for moving air against very low static pressures, such as would be encountered in general room ventilation. Their performance is very sensitive to resistance and a small increase will cause a significant reduction in flow.

Tubeaxial and vaneaxial fans are shown in Figure 5-2. The tubeaxial or ducted fan is essentially a propeller-type fan mounted in a cylindrical housing and is capable of moving air against pressures less than about 2 in. H_2O. Since the motor is typically mounted inside the housing, it is not generally used on contaminant-containing gas streams. The vaneaxial fan is similar in construction but uses airfoil-style blades and straightening vanes on the inlet and outlet to improve efficiency, locates the motor on the outside of the casing and provides an enclosure to protect the drive system. It is capable of developing pressures up to about 8 in. H_2O.

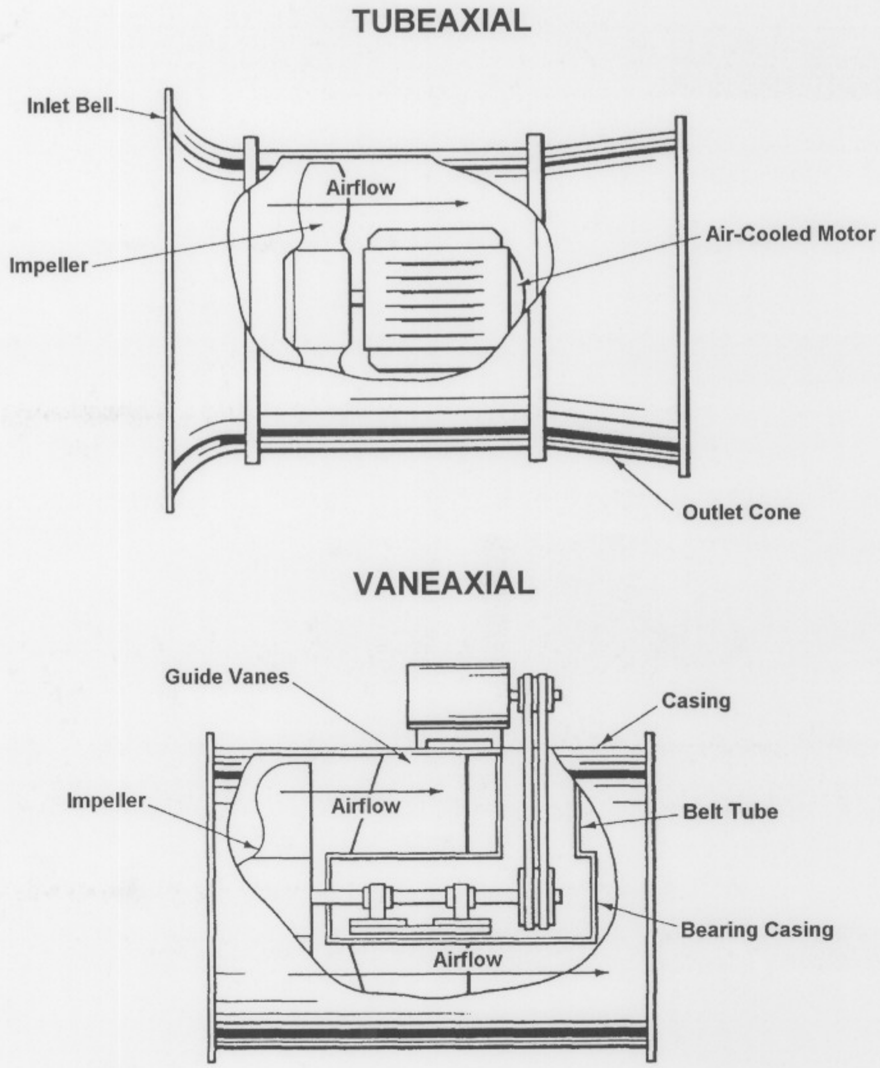


Figure 5-2. Tubeaxial and vaneaxial fans (ACGIH, 1988)

The principal fan used in air pollution control systems is the centrifugal fan. The basic design of the centrifugal fan, as illustrated in Figure 5-3, employs a fan wheel or impeller mounted inside a scroll-type housing. Air is draw into the inside of the impeller and then forced out through the housing. In general, centrifugal fans are distinguished by the design of the impeller. There are three basic impeller types: (1) forward curved, (2) radial and (3) backward inclined. The backward inclined impeller may use a standard single-thickness blade or an airfoil blade.

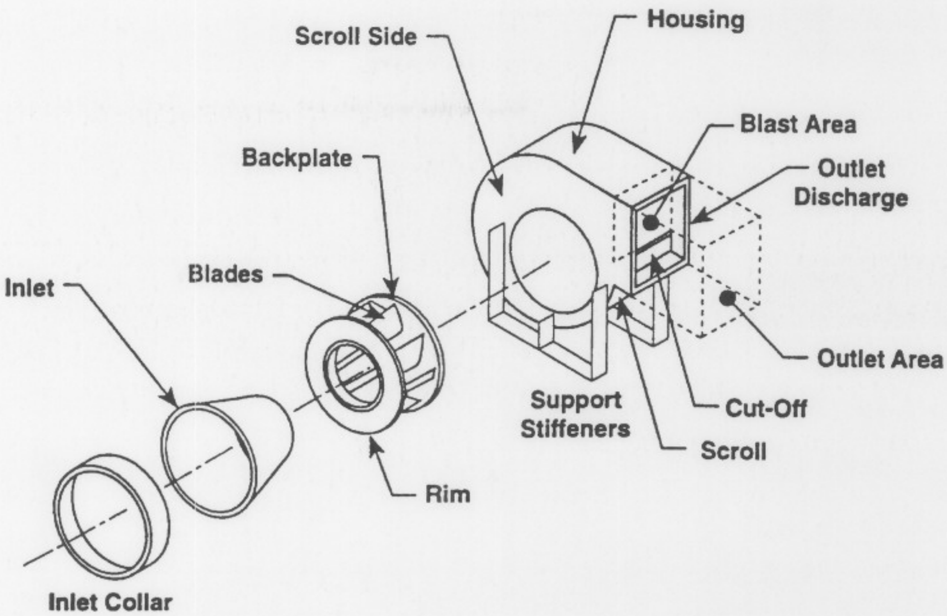


Figure 5-3. Centrifugal fan components (ACGIH, 1988)

Forward curved impellers, shown in Figure 5-4, have blades that curve into the direction of rotation. Commonly referred to as "squirrel-cage blowers", they are constructed of lightweight materials and usually have 24 to 64 closely-spaced blades. For a given duty, these impellers are the smallest of all the centrifugal types and operate at the lowest speeds. As a result, they are quiet in operation but are only able to develop low to moderate static pressures. Because of this they are not commonly used in air pollution control systems. As shown in the accompanying performance chart, the highest mechanical efficiency is developed at a point to the right of peak static pressure, at about 60 percent of the wide open volume, and the horsepower requirement rises continually toward the free delivery volume. Since particles may adhere to the closely-spaced blades and cause a balance problem, their use should be limited to clean gas streams.

Radial impellers, shown in Figure 5-5, have 6 to 10 blades that extend either straight out from the hub or along a curved radial. They are the simplest of all the centrifugal fans and the least efficient, but they are capable of developing the highest static pressures. For a given duty, they operate at moderate speeds. Highest mechanical efficiency is developed just to the left of peak static pressure, at about 30-40 percent of the wide open volume, and the horsepower requirement again rises continually toward the free delivery volume. The radial blade shape is generally resistant to material build-up and may be used in systems handling either clean or dirty air. There are a variety of impeller designs, ranging from "high efficiency, minimum material" to "heavy impact resistant". Impellers in this latter category usually have no inlet plate or backplate in order to minimize locations of potential material build-up.

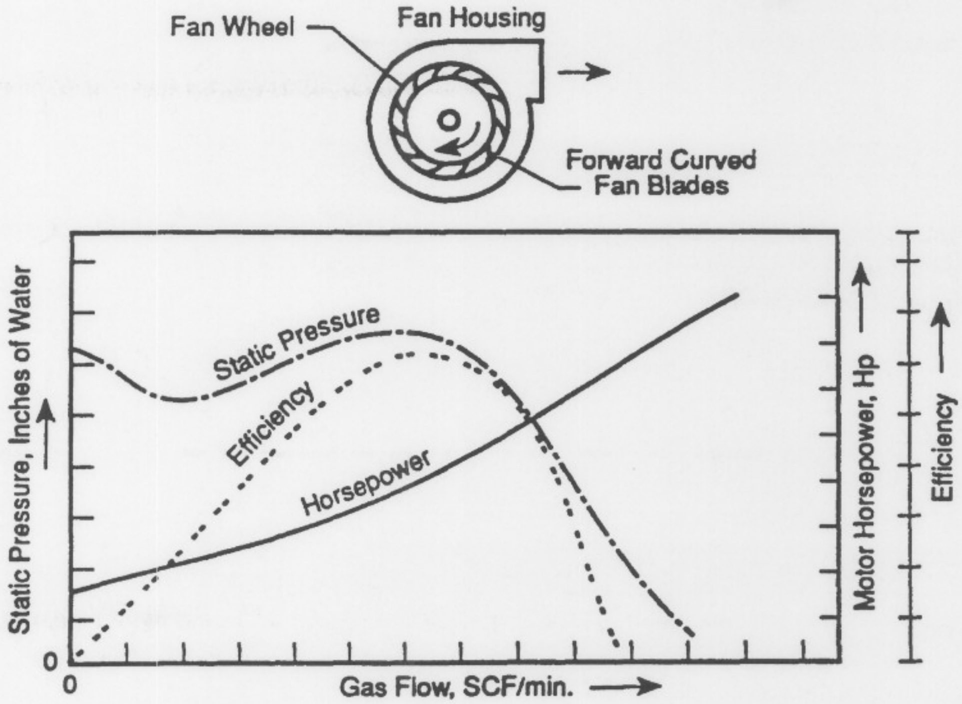


Figure 5-4. Forward curved wheel performance (ACGIH, 1988)

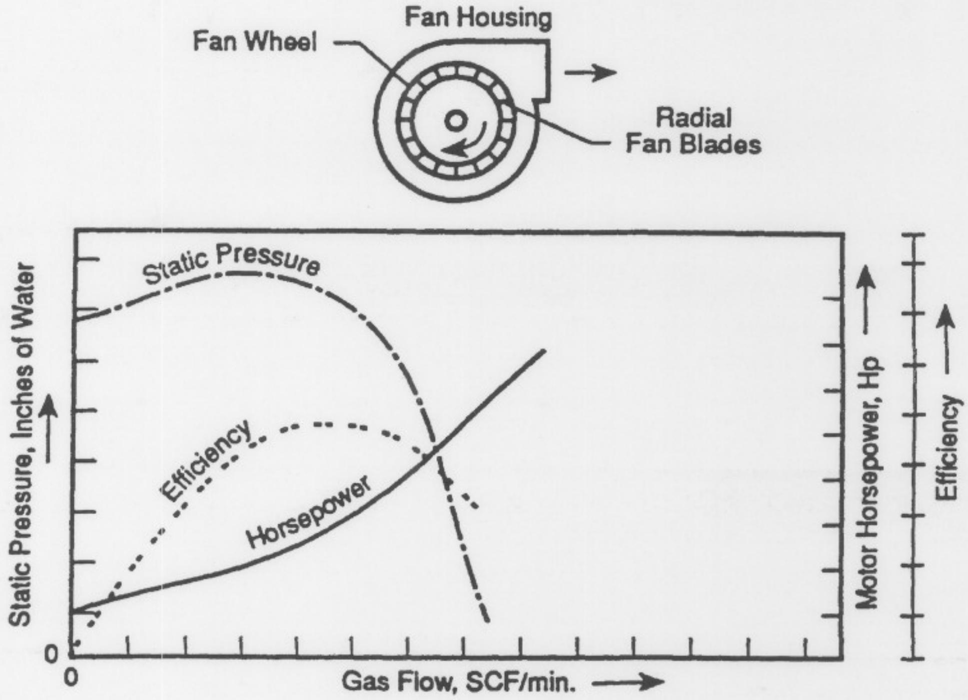


Figure 5-5. Radial wheel performance (ACGIH, 1988)

Standard backward inclined impellers, shown in Figure 5-6, have 9 to 16 single-thickness blades that incline or curve away from the direction of rotation. They have the second-highest efficiency of all the centrifugal fans and, for a given duty, will operate at the highest speed. Highest mechanical efficiency is developed to the right of peak static pressure, at about 50-60 percent of the wide open volume. A unique characteristic of the backward inclined impeller is that the horsepower requirement reaches a maximum value near the point of peak efficiency and then declines toward the free delivery volume. For this reason backward inclined fans are sometimes referred to as "non-overloading", since any variation from the optimum operating point due to a change in system resistance will result in a reduction in operating horsepower. Because the blades are single thickness, they can be used in gas streams with light dust loadings. However, they should not be used in heavy loading situations that could cause build-up on the blade surfaces.

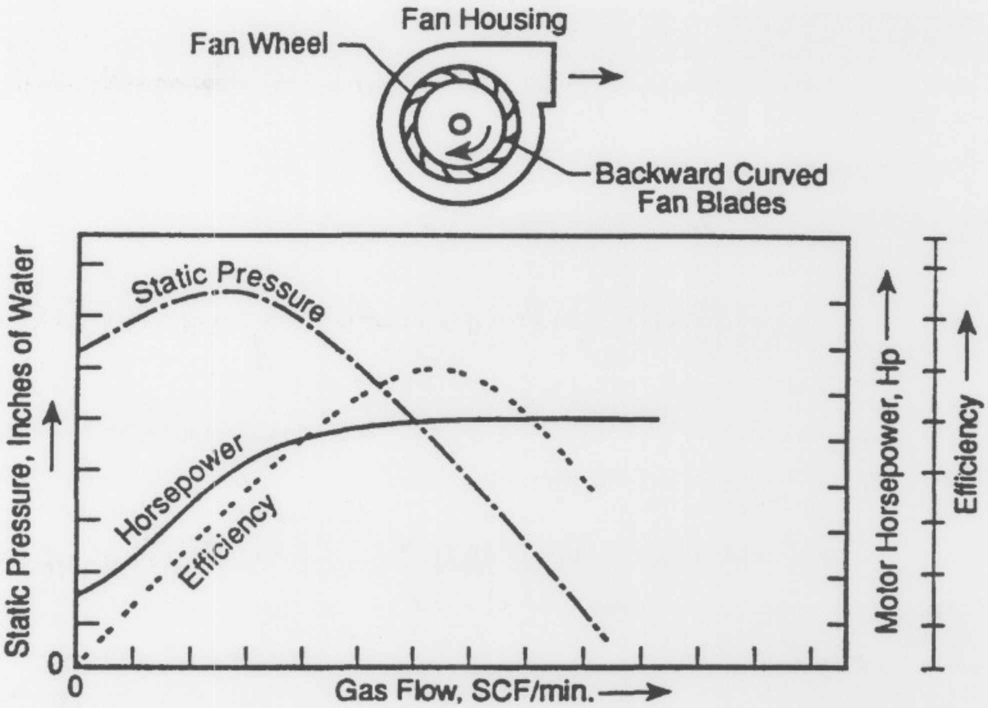


Figure 5-6. Standard backward inclined wheel performance (ACGIH, 1988)

The airfoil-type backward inclined impellers are identical in design to the standard, except they employ hollow airfoil-style blades. Peak efficiency is slightly more than the standard design but is still developed to the right of peak static pressure and at about 50-60 percent of the wide open volume. The horsepower requirement again exhibits the non-overloading characteristic. One negative feature of the airfoil design is the use of hollow blades. These blades can erode and accumulate material inside the blade, causing a balance problem. They should, therefore, be limited to clean air applications.

Figure 5-7 shows the tubular centrifugal fan. This fan employs a centrifugal wheel mounted in an axial arrangement. The impeller design is usually backward inclined but, although it has the same general performance characteristics and limitations, is somewhat less efficient. It also has lower capacity and static pressure capabilities because of the 90 degree change in direction that occurs as the air moves through the housing.

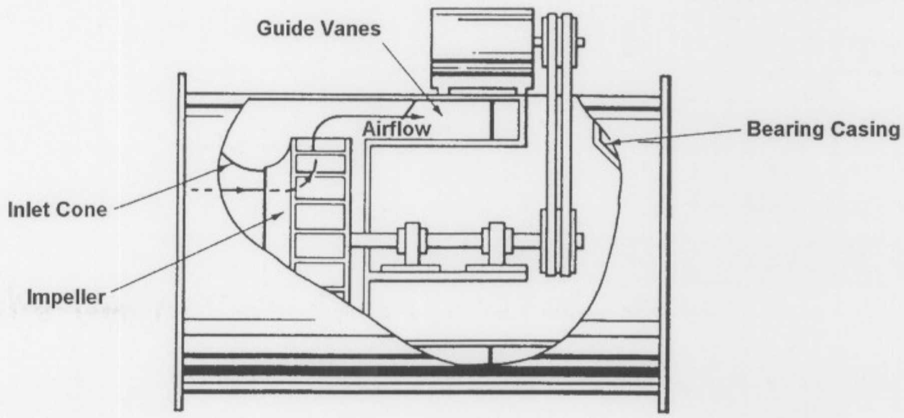


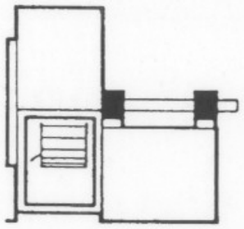
Figure 5-7. Tubular centrifugal fan

Fan arrangements

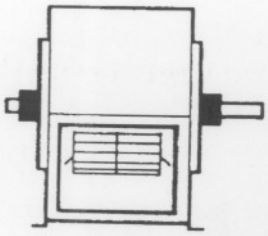
Fans are constructed with different bearing locations and motor mounting capabilities, generally referred to as "arrangements". Although these are not of any particular importance to the inspection task, knowledge of the various arrangements is desirable when it is useful to speak the language of fan systems.

There are ten basic fan arrangements and three of the most common are shown in Figure 5-8. Here, SW or DW refers to single- or double-width fans, respectively. As the name implies, double-width fans have an impeller that is twice as wide as the corresponding single-width version and a capacity that is also approximately double. Also shown are SI and DI, which refer to single- or double-inlet, respectively. The more common single-inlet fans have the air inlet on only one side, opposite the drive. Double-inlet fans have air inlets on both sides.

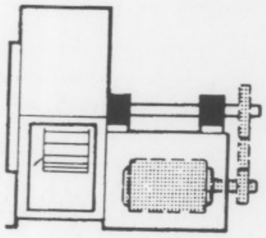
Arrangement 1 has two shaft bearings mounted on the pedestal with the impeller overhung on the end of the shaft. For a V-belt drive, the motor would be mounted on a separate base adjacent to the pedestal and at ground level, with pulleys or sheaves on both the motor shaft and the fan shaft. For direct drive, the motor would be mounted on a separate or extended base and connected directly to the shaft. Arrangement 3 also has two shaft bearings, but they are located on either side of the impeller and supported by the fan housing. The drive arrangement would be the same as Arrangement 1. Arrangement 9 is the same as Arrangement 1, but has the motor mounted on the outside of the fan base with a V-belt drive system. With the exception of Arrangement 4, which has the impeller mounted directly on the motor shaft, the other arrangements are basically variations on these three.



Arr. 1 SWSI: For belt or direct drive
Overhung impeller; bearings on base



Arr. 3 DWDI: For belt or direct drive
Bearings supported by fan housing



Arr. 9 SWSI: For belt drive
Overhung impeller; bearings on base;
motor outside base

Figure 5-8. Fan arrangements

Fan laws

The fan laws relate the performance variables for any homologous series of fans. A homologous series is simply a range of fan sizes where all of the dimensional parameters are proportional. At the same relative point of operation on any two performance curves in a homologous series, the mechanical efficiencies will be equal. Under these conditions, the following relationships apply:

$$Q_2 = Q_1\left(\frac{\text{size}_2}{\text{size}_1}\right)^3\left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)$$

$$P_2 = P_1\left(\frac{\text{size}_2}{\text{size}_1}\right)^2\left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^2\left(\frac{\rho_2}{\rho_1}\right) \tag{5-1}$$

$$\text{bhp}_2 = \text{bhp}_1\left(\frac{\text{size}_2}{\text{size}_1}\right)^5\left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^3\left(\frac{\rho_2}{\rho_1}\right)$$

The performance variables involved here are flow rate (Q), fan size (size), rotational speed (rpm), pressure (P), gas density (ρ) and horsepower (bhp). Here, the pressure may be represented by total pressure, static pressure, velocity pressure, fan total pressure or fan static pressure. These latter two terms will be defined later in this chapter.

As indicated, these expressions rely on the performance curves being homologous and apply only at the same relative point of rating. Under turbulent flow conditions, which occur in most air pollution control systems, two performance conditions will be at the same relative point of rating if the pressures and flow rates at these two conditions are related by:

$$P_2 = P_1\left(\frac{Q_2}{Q_1}\right)^2 \tag{5-2}$$

As will be seen in the next section, this is the same as saying that the two performance points must lie along the same system curve.

In actual practice, the fan laws are typically used to determine the effect of changing only one variable at a time and are most often applied to a single fan size. The most common variable of interest is fan speed. For determining the effect of changing fan speed while operating on the same gas stream ($\rho_1 = \rho_2$), the fan laws become:

$$Q_2 = Q_1\left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)$$

$$P_2 = P_1\left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^2 \tag{5-3}$$

$$\text{bhp}_2 = \text{bhp}_1\left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^3$$

Referring to the original equations, it is interesting to note that if only changes in gas density are involved, pressure capabilities and power requirements change proportionally, while flow rate is unaffected. This behavior is sometimes characterized by stating that "a fan is a constant volume machine", i.e., it moves volumes of air not masses of air.

Fan performance

Performance graphs for centrifugal fans were presented in Figures 5-4 through 5-6. One of the relationships shown on these figures was that between static pressure developed and air volume moved. This relationship is sometimes referred to as the fan curve or fan characteristic. For a particular fan turning at a given rpm, there is one and only one fan curve. It represents all of the combinations of pressure and air volume that that fan is capable of delivering when operating at that one rpm. These range from low air flow delivered against high pressure (upper left) to high air flow delivered against low pressure (lower right). What determines which condition a fan will operate at is how this curve interacts with the ventilation system characteristics, as represented by the system curve.

Normalized performance curves for three systems are shown in Figure 5-9. Plotted here is the percentage of duct system resistance as a function of the percent of duct system flow rate. This is simply a normalized version of a P verses Q plot. The system curves shown follow the general relationship characteristic of turbulent systems:

$$P_2 = P_1\left(\frac{Q_2}{Q_1}\right)^2 \tag{5-4}$$

In practice, the system curve is developed by first determining the resistance or static pressure for one flow rate through the system, using the techniques discussed in Chapters 2 and 3. Other points on the curve are then determined using the above relationship. Thus, if the design point for System A were at 100 percent volume and 100 percent resistance, increasing the flow rate to 120 percent of the design flow would increase the resistance to 144 percent of the design resistance. Likewise, decreasing the flow rate to 50 percent of the design value would decrease the resistance to 25 percent of the design resistance. Note that, on a percentage basis, the same relationships also hold for Systems B and C.

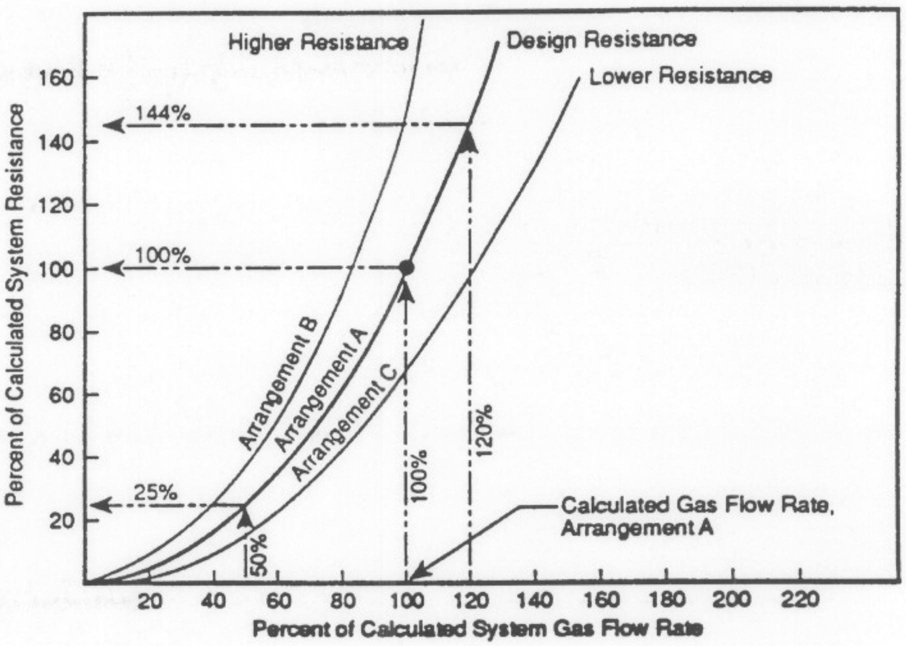


Figure 5-9. Normalized performance curves

The point of intersection of the system curve with the fan curve determines the actual fan performance. This is shown in Figure 5-10, where a normalized fan curve has been plotted with the system curves from the previous figure. Here, the 100 percent design volume of System A has been arbitrarily selected to intersect at Point 1 with the 60 percent free delivery volume of the fan. Unless actions are taken to change either the fan curve or the system curve, the performance delivered will be that indicated by the intersection point.

One way to change the flow rate would be to change the system. This could be done by closing or opening a damper, producing a system with more or less resistance and changing the system curve. For example, referring again to Figure 5-10, the flow rate could be decreased to 80 percent of the design volume by closing a damper until the more resistant System B curved is obtained, shifting the intersection to Point 2. Likewise, the flow rate could be increased to 120 percent of the design value by opening a damper and shifting the intersection to Point 3.

Changes in flow rate could also be produced by changing the fan speed, shifting the fan curve. This is illustrated in Figure 5-11, where a new fan curve representing a 10 percent increase in speed has been added. At this new speed, the point of operation shifts to Point 2. Since flow rate is proportional to fan speed, this 10 percent increase in speed produces a corresponding 10 percent increase in volume. However, following the fan laws, this 10 percent increase in speed will require a 33 percent increase in operating horsepower, which may be beyond the capabilities of the existing motor.

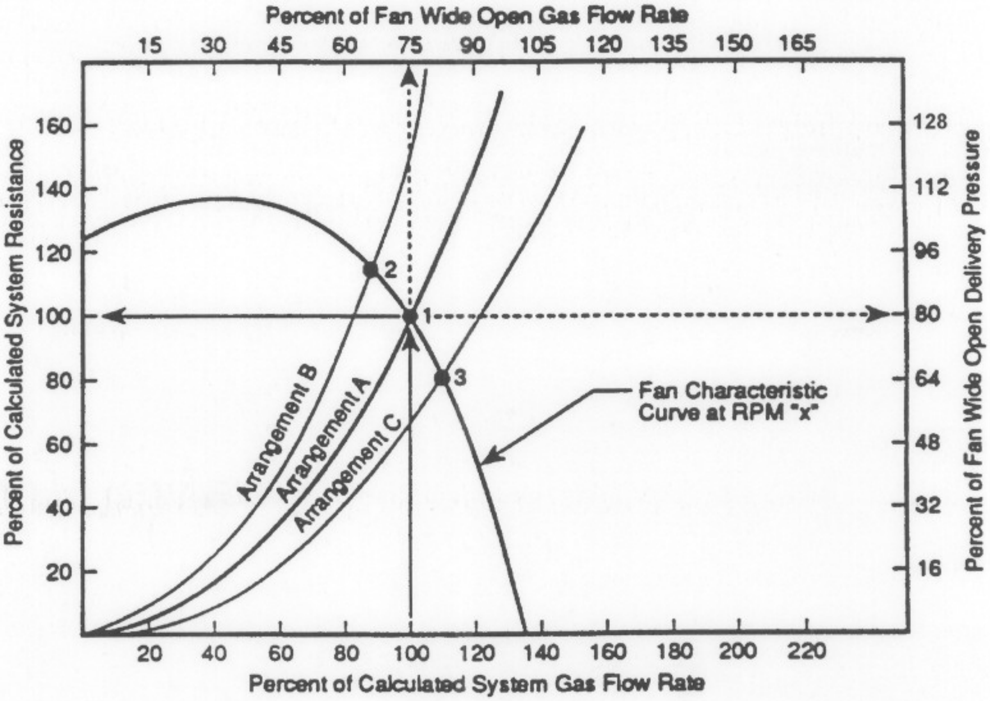


Figure 5-10. Interaction of system curves with fan curve

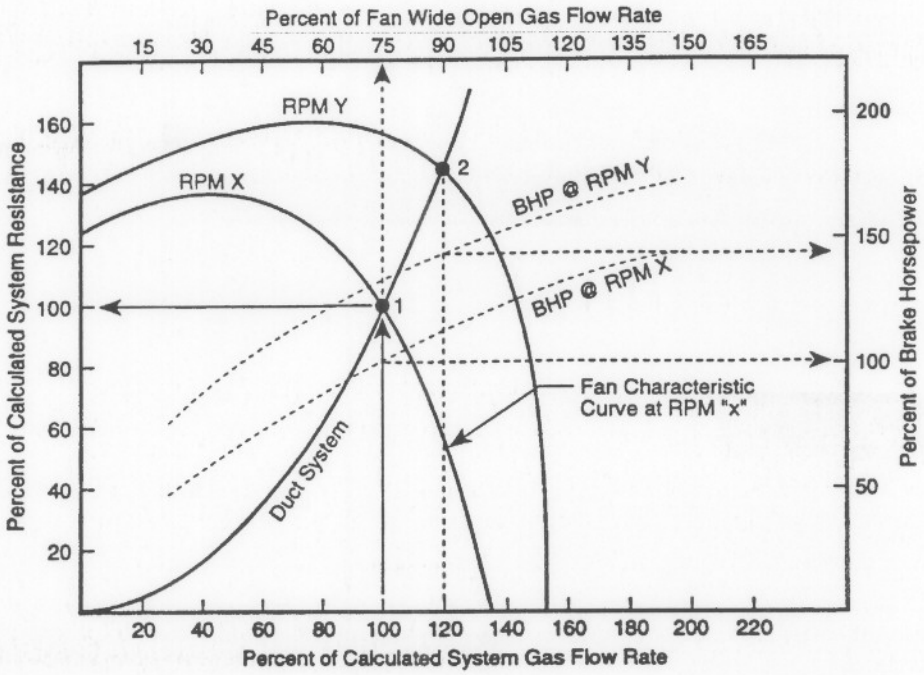


Figure 5-11. Effect of increased fan speed

According to the fan laws, changing gas density will shift the fan curve. However, since gas density also affects the system resistance, the system curve will also be shifted. This is illustrated in Figure 5-12 for a density change from 0.0375 lb/ft^3 to 0.075 lb/ft^3. As previously indicated, the new operating point will deliver the same air volume but at double the resistance and double the horsepower requirement.

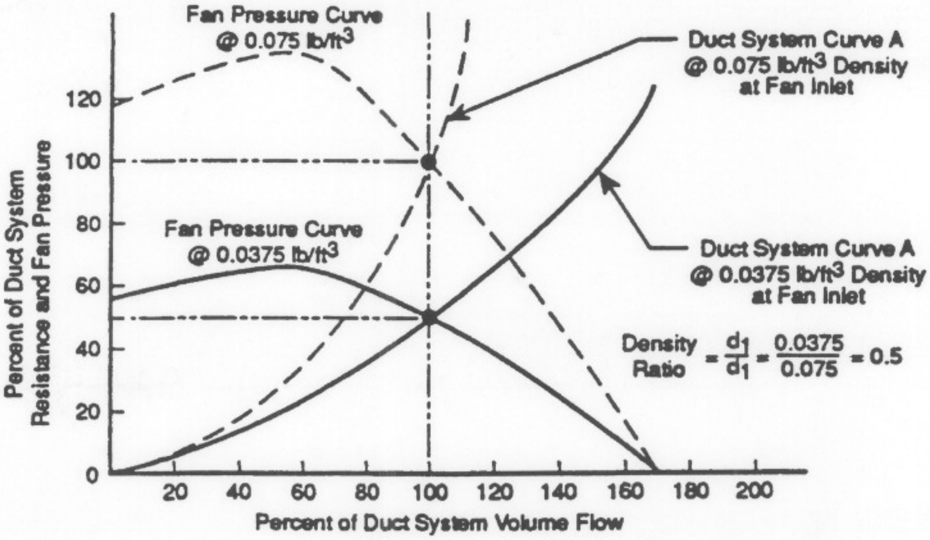


Figure 5-12. Influence of gas density

Fan performance can also be affected by the manner in which the fan is installed in the system. Most manufacturers in the United States and Canada rate the performance of their fans from tests made in accordance with the Air Movement and Control Association (AMCA) Standard 210, *Test Code for Air Moving Devices*. The test set-up prescribed by this standard is designed to produce inlet and outlet flows that are as uniform as possible. This condition insures consistency and reproducibility of test results and permits the fan to develop its maximum performance. In any installation where this uniform flow condition does not exist, fan performance will be reduced.

Installation conditions that affect fan performance are referred to as "system effects". The three most common causes of deficient performance are: (1) non-uniform inlet flow, (2) swirl at the fan inlet, and (3) improper outlet connections. These conditions alter the aerodynamic characteristics of the fan such that it does not operate at its rated performance.

The influence of system effects on fan performance is shown in Figure 5-13. Here, the solid system curve has been determined without consideration of system effects and performance corresponding to Point 1 is anticipated. However, because of system effects, performance corresponding to Point 3 is obtained. This deficient performance could be prevented by calculating the system effect loss, adding it to the system resistance and selecting a fan to operate at Point 2.

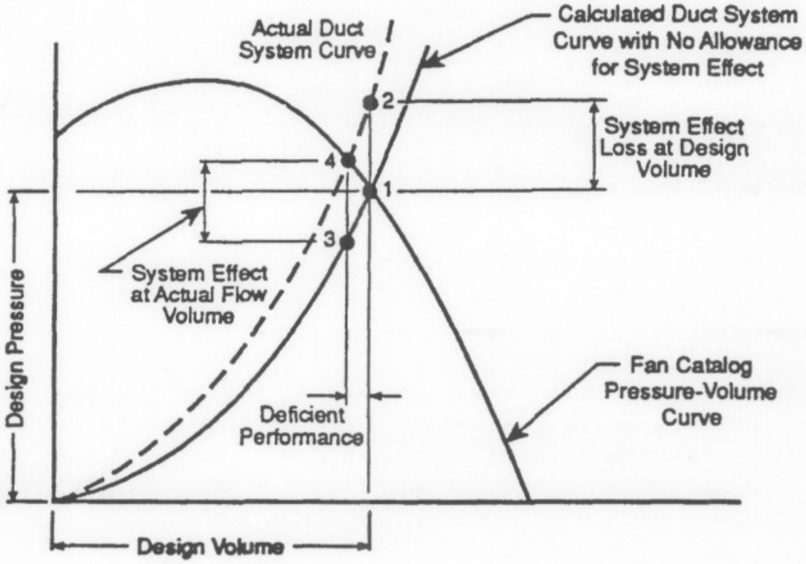


Figure 5-13. Influence of system effects

System effect loss factors for some common installation conditions are given in Tables 5-1 through 5-3. These factors are in numbers of velocity pressures lost, so the addition to the system static pressure would be equal to the loss factor times the appropriate velocity pressure. Loss factors for other situations can be found in ACGIH's *Industrial Ventilation* or in AMCA's *Fan Application Manual*, *Part 1*.

System effect losses in Tables 5-2 and 5-3 are expressed in terms of the percentage of effective duct that is present at the fan outlet and the blast area/outlet area ratio. Effective duct length is one diameter for each 1000 fpm duct velocity, with a minimum of 2.5 diameters. The ratio of blast area to outlet area can be determined from manufacturer's literature or estimated as 0.7.

Fan selection

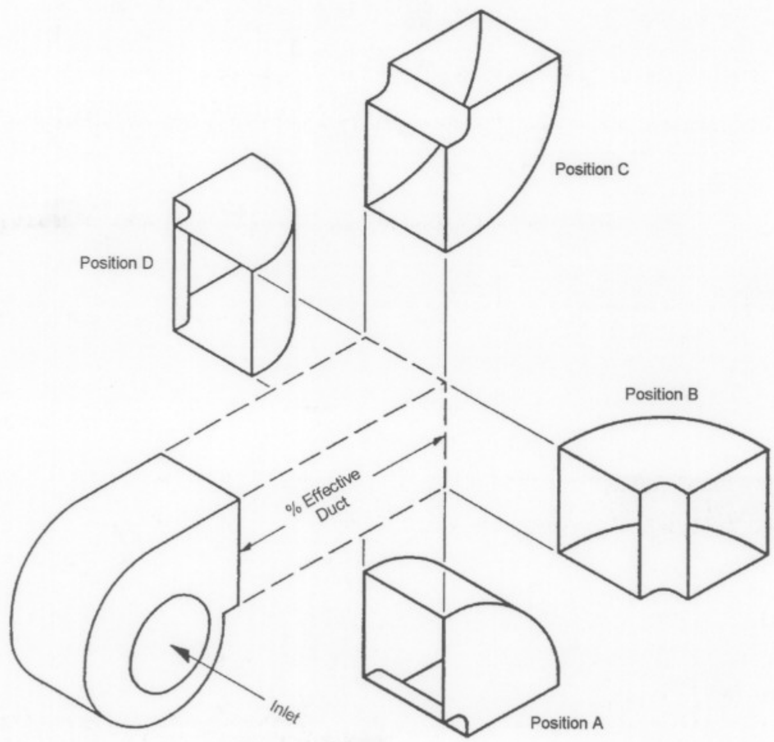
Selecting a fan is usually the responsibility of the ventilation system designer. However, since the inspector may wish to apply variations of the selection technique in evaluating the performance of an existing fan, it is important that the methods used by the designer be understood.

Fan selection is typically done using ratings tables published by manufacturers for their products. An example of a portion of one of these tables is shown in Table 5-4. In general, the ratings table is entered along the row corresponding to the design volume and down the column corresponding to the design static pressure, including system effects. The point of intersection indicates the rpm that the fan would have to turn to deliver the required performance and the horsepower that would be needed to drive it. Shaded regions are usually included on the chart to indicate areas of good mechanical efficiency.

Table 5-1. System effect factors for outlet ducts (ACGIH, 1988)

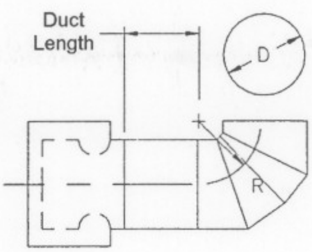
	System Effect Factors				
Blast Area / Outlet Area	No Duct	12% Effective	25% Effective	50% Effective	100% Effective
0.4	2.00	1.00	0.40	0.18	0.00
0.5	2.00	1.00	0.40	0.18	0.00
0.6	1.00	0.66	0.33	0.14	0.00
0.7	0.80	0.40	0.14	0.00	0.00
0.8	0.47	0.22	0.10	0.00	0.00
0.9	0.22	0.14	0.00	0.00	0.00
1.0	0.00	0.00	0.00	0.00	0.00

Table 5-2. System effect factors for outlet elbows (ACGIH, 1988)



Blast Area / Outlet Area	System Effect Factors					
	Elbow Position	No Duct	12% Effective	25% Effective	50% Effective	100% Effective
0.4	A	3.20	2.50	1.80	0.80	0.00
	B	4.60	3.90	2.50	1.20	0.00
	C	5.50	4.60	3.20	1.60	0.00
	D	5.50	4.60	3.20	1.60	0.00
0.5	A	2.00	1.60	1.20	0.53	0.00
	B	2.90	2.30	1.80	0.80	0.00
	C	3.90	2.90	2.30	1.00	0.00
	D	3.90	2.90	2.30	1.00	0.00
0.6	A	1.60	1.40	1.00	0.40	0.00
	B	2.00	1.60	1.20	0.53	0.00
	C	2.90	2.30	1.80	0.80	0.00
	D	2.50	2.00	1.40	0.66	0.00
0.7	A	0.66	0.53	0.40	0.18	0.00
	B	1.00	0.80	0.53	0.26	0.00
	C	1.40	1.20	0.80	0.33	0.00
	D	1.20	1.00	0.66	0.33	0.00
0.8	A	0.80	0.66	0.47	0.22	0.00
	B	1.20	1.00	0.66	0.33	0.00
	C	1.60	1.40	1.00	0.40	0.00
	D	1.40	1.20	0.80	0.33	0.00
0.9	A	0.66	0.53	0.40	0.18	0.00
	B	1.00	0.80	0.53	0.26	0.00
	C	1.20	1.00	0.66	0.33	0.00
	D	1.00	0.80	0.53	0.26	0.00
1.0	A	1.00	0.80	0.53	0.26	0.00
	B	0.66	0.53	0.40	0.18	0.00
	C	1.00	0.80	0.53	0.26	0.00
	D	1.00	0.80	0.53	0.26	0.00

Table 5-3. System effect factors for inlet elbows (ACGIH, 1988)



	System Effect Factors		
R/D	No Duct	2D Duct	5D Duct
0.50	1.80	1.00	0.53
0.75	1.40	0.80	0.40
1.00	1.20	0.66	0.33
2.00	1.00	0.53	0.33
3.00	0.66	0.40	0.20

Four or more piece unvaned mitered 90° round section elbow

Ratings tables indicate the performance of a fan when operating on air having a density of 0.075 lb/ft^3. Since a given system may be handling air of a different density, some adjustments are involved before entering the table. Remember, density does not affect fan volume, but it does influence static pressure conditions and horsepower requirements. The specific procedure involved in fan selection is as follows:

Table 5-4. Example fan ratings table (ACGIH, 1988)											
				Inlet diameter: 13" OD Outlet area: 0.93 ft² inside				Wheel diameter: 22½" Wheel circumference: 5.92 ft			
		10" SP		12" SP		14" SP		16" SP		18" SP	
CFM	OV	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
1674	1800	1863	4.86	2035	5.98	2194	7.16	2340	8.38	2479	9.67
1860	2000	1872	5.24	2040	6.39	2199	7.62	2344	8.89	2484	10.2
2046	2200	1879	5.63	2048	6.84	2206	8.13	2351	9.43	2487	10.8
2232	2400	1889	6.07	2056	7.33	2212	8.64	2357	10.0	2493	11.4
2418	2600	1900	6.53	2065	7.84	2222	9.22	2364	10.6	2501	12.1
2790	3000	1924	7.52	2088	8.96	2241	10.4	2383	12.0	2517	13.5
3162	3400	1952	8.60	2115	10.2	2265	11.8	2405	13.4	2538	15.1
3534	3800	1984	9.79	2144	11.5	2290	13.3	2428	15.0	2562	16.8
3906	4200	2018	11.0	2174	12.9	2320	14.8	2458	16.8	2587	18.7
4278	4600	2058	12.4	2209	14.5	2355	16.5	2489	18.6	2614	20.6
4650	5000	2100	13.9	2247	16.1	2390	18.3	2521	20.5	2645	22.7
5022	5400	2146	15.5	2291	17.8	2428	20.2	2558	22.6	2681	25.0

1. Determine the design air volume at actual conditions.

2. Calculate the fan static pressure at actual conditions, including system effects. Fan static pressure is defined as:

$$FSP = SP_{out} - SP_{in} - VP_{in} \tag{5-5}$$

   In calculating fan static pressure, the sign of the static pressure is important and must be included. Some manufacturers rate their fans on fan total pressure. Fan total pressure is defined as:

$$FTP = TP_{out} - TP_{in} \tag{5-6}$$

   The sign of the total pressure is again important and must be included.

3. Correct the fan static pressure to an equivalent value for standard air:

$$FSP_{equivalent} = FSP_{actual}\left(\frac{0.075}{\rho_{actual}}\right) \tag{5-7}$$

4. Enter the ratings table at the actual volume and the equivalent fan static pressure. Determine the rpm and horsepower requirements.

5. Correct the horsepower requirement to the conditions of actual operation:

$$bhp_{actual} = bhp_{equivalent}\left(\frac{\rho_{actual}}{0.075}\right) \tag{5-8}$$

Since density varies inversely with temperature, corrections for operating conditions could also be made using a ratio of absolute temperatures. Also, because the exact input parameters may not be contained in the ratings table, linear interpolation between the nearest values may be required.

In selecting the appropriate motor for a fan, the designer must give consideration to the possible air densities the fan may have to handle. For example, a system operating at elevated temperature would require the horsepower as determined above. However, when starting the system it may be necessary to handle colder higher-density air, requiring more horsepower. Should the system be located outdoors in an area that has extreme low temperatures, the horsepower requirement for start-up could be considerable.

An alternate way of dealing with this situation would be to install the horsepower required for normal operation and then use an inlet or outlet damper, together with an amperage control system, during start-up. When the fan is started on cold air, the amperage control system would sense a high current flow and close the damper to prevent or reduce air flow into the fan. The fan turning through the restricted air flow would heat it up, reducing its density and reducing the current draw. The amperage control system would sense this and open the damper a little bit to allow some air flow from the hot process. This scenario would continue until the damper was fully open and the system was operating at design conditions.

Evaluation of fan performance

The fan laws and the techniques involved in fan selection may be used by the inspector to estimate the air volume the fan is delivering. For example, if the air volume delivered by an existing fan were known, but a subsequent inspection determined that the fan speed had been changed, the new air volume could be estimated from:

$$Q_2 = Q_1\left(\frac{rpm_2}{rpm_1}\right) \tag{5-9}$$

Initial estimates of air volume could be made using measurements of the fan operating parameters, together with the appropriate ratings table. This approach would apply only to V-belt or variable-speed drive fans, for which general ratings tables are available. Direct drive fans are specially constructed to deliver the required volume when turned at the speed of the motor and do not have published performance tables. It should be noted, however, that because of our inability to make exact measurements of some of the parameters and because of a lack of precision in the ratings tables, these techniques will yield only a rough estimate of flow rate.

The most satisfactory technique for estimating fan performance from the ratings tables would be to use measurements of fan speed and fan static pressure. The specific procedure is as follows:

1. Measure or estimate fan rpm. To obtain as much accuracy as possible, measurement of the rpm is preferred. However, if this is not possible, an estimate can be used. Techniques for determining fan rpm will be discussed in Chapter 6.

2. Determine fan static pressure for operation on standard air. This would involve measuring inlet and outlet static pressures and estimating inlet velocity pressure. Because of turbulence levels near the inlet and outlet of a fan, it may be difficult to get acceptable readings. To avoid these turbulence problems, the measurements could be made some distance away from the fan, where acceptable readings can be obtained. Since static pressure losses are small in the larger ducts usually found at the fan, the error introduced by this should be minimal. If inlet or outlet dampers are present, the loss introduced by these fittings would have to be estimated and included in the determination of the respective static pressure. Likewise, any losses due to system effects should be included. Air density could be estimated from a measurement of temperature, and velocity pressure could be estimated based on the expected velocity at the inlet. The estimated fan static pressure would then be given by:

$$FSP_{estimated} = 0.075\left(\frac{SP_{out} - SP_{in} - VP_{in}}{\rho_{actual}}\right) \tag{5-9}$$

3. Enter the ratings table for the fan at the column corresponding to the estimated fan static pressure. Proceed down this column until the row containing the measured fan speed is located. Read along this row to determine the estimated flow rate. Interpolation between values in the ratings table may be necessary.

If the air volume determined from this procedure gives a significantly different velocity pressure than that assumed in Step 2, the velocity pressure should be re-estimated and the procedure repeated until a reasonable agreement is obtained.

A less satisfactory technique for estimating fan performance from the ratings tables would be to use measurements of fan static pressure and operating horsepower. Following the same general procedures outlined above, the fan static pressure would be estimated and used to enter the ratings table. The row corresponding to the estimated operating horsepower would then be located and used to determine the estimated flow rate. Techniques for determining these parameters, as well as the other parameters involved in the evaluation of fan performance, will be discussed in Chapter 6.

As a minimum, the fan inspection should include an evaluation of the condition of the fan. This would include a visual determination of the condition of the fan housing to assess any indications of corrosion, an evaluation of the inspection door seal for leakage, an assessment of the condition of isolation sleeves used to dampen vibration between the fan and the inlet and outlet ducts to determine if there are any leaks, and an evaluation of any vibration or belt squeal. Belt squeal during operation indicates that the belts are slipping on the pulleys. This can result in the loss of 200-300 rpm, with a corresponding loss in air volume. A fan that is vibrating severely represents a significant safety hazard. If this condition should be encountered, the inspection should be terminated immediately and plant personnel notified of the condition. If the fan is not operating, an inspection of the condition of the fan wheel would also be useful to identify any build-up or corrosion problems.

References

ACGIH, *Industrial Ventilation*, Twentieth Edition, Cincinnati, 1988.

AMCA, "Fan Systems", *Fan Application Manual,* Part 1, Publication 201, Arlington Heights (Illinois), 1979.

Kemner, W., R. Gerstle and Y. Shah, *Performance Evaluation Guide for Large Flow Ventilation Systems*, EPA-340/1-84-012, May 1984.

Chapter 6

Measurement of Ventilation System Parameters

In previous chapters, the use of various parameters to evaluate the performance of ventilation system components has been suggested. In this chapter, the methods available for making these measurements will be discussed and recommendations on the most appropriate techniques and procedures will be made. Where appropriate, additional techniques for estimating some parameters will be given.

Measurement ports

When a ventilation system is first inspected, it is unlikely that measurement ports will be available. If some ports are available, they are not likely to be in the locations needed or of an appropriate size. The most likely port to be found on a ventilation system is a 3 or 4 inch diameter sampling port located on or near the stack. Although a port in this location may be useful for some inspection measurements, ports of this size should, in general, be avoided. They present difficulties in sealing under both positive and negative conditions, and they may be quite difficult to open because of the large thread area.

The most useful port size for inspections is 1½-2 inch diameter, and this size is needed only if measurements of velocity pressure are anticipated. For the more routine measurements of temperature and static pressure, ports of ¼-½ inch will accommodate most measurement probes. The larger inspection ports will require the installation of a pipe stub with a threaded plug for closing. The smaller ports should simply be drilled and then covered with duct tape when not in use. Because of the potential for fire or explosion from sparks and because of possible damage to downstream equipment, the inspector should not request that ports be installed while the equipment is running. Rather, the locations and sizes needed should be marked for plant personnel, so that they may install them the next time the system is shut down.

Ports of a proper size may already be installed in some locations and used by the plant for continuous monitoring of certain parameters. In general, these ports should be avoided by the inspector. If they must be used, they should be opened only by plant personnel. Never open a port that was not placed there for your exclusive use. Plant monitoring ports may be connected to controllers that initiate equipment shut-down if the signal from them is lost.

Measurement ports are subject to the accumulation of material that may cause them to become plugged, even if they are on the clean side of the control device. Before using any port, it should be cleaned out with a non-sparking rod to assure unobstructed access to the gas stream. Also, while making measurements the port should be sealed to prevent flow in or out around the probe. Flow into or out of the port may cause an interference with the measurements being made. For inspection ports, the best sealing technique is to insert the probe through a rubber stopper and then place that stopper into or against the port. For the larger stack-sampling ports, a rubber sanding disc may be used to cover the opening. The probe, equipped with a rubber stopper, would then be inserted through the center of the sanding disc, using the stopper to complete the seal.

Finally, the inspector should not make heroic efforts to reach existing ports and should not have ports installed in locations that cannot be reached and used in safety. This should include consideration of hazards to walking and climbing, as well as the potential for exposure to inhalation, vision, hearing and fire and burn hazards.

Static pressure measurement

Static pressure measurements must be made with a square-ended probe placed at a right-angle to the flow direction. If measurements of velocity pressure are also being made, the static pressure ports on a standard commercial pitot tube that is oriented into the oncoming flow may also be used, as could one leg of the S-type pitot if it is turned at a right-angle to its normal position. These two pitot tubes will be discussed later in this chapter. The purpose of the probe orientation is to be sure that no component of velocity pressure is impacting the probe during static pressure measurements.

The area between the probe and the port opening should be sealed to avoid errors associated with flow into or out of the duct. Errors resulting from improper sealing can be as large as 10-30 percent. Flow into the duct can result in an aspiration effect at the end of the probe that can increase (make more negative) the negative pressures being measured, while flow out of the duct can add a component of velocity pressure to the measurement of positive pressures. To further mitigate this problem, the probe should be extended well into the duct while making measurements.

There are two widely used techniques for sensing the pressures measured by the probe: (1) a U-tube manometer or (2) a Magnehelic® pressure gauge. The U-tube manometer is a reference instrument that is available in a flexible or slack-tube configuration, shown in Figure 6-1, to enhance its portability. The manometer is equipped with magnets at the top and bottom to facilitate temporary mounting and is equipped with threaded connectors that are used to seal the manometer during transport.

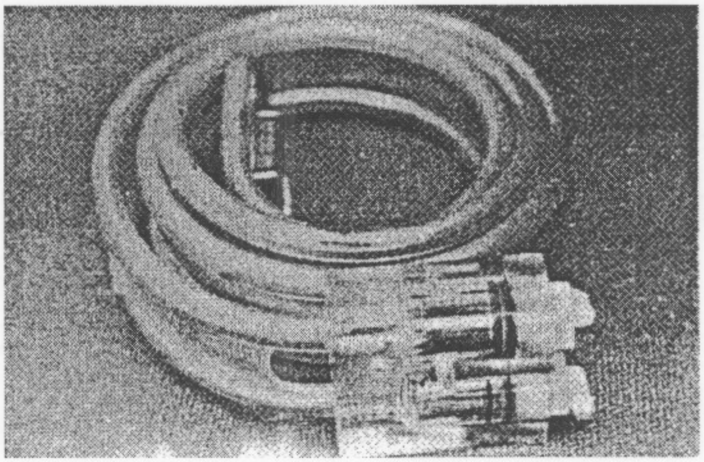


Figure 6-1. Slack tube manometer (Segal et al, 1985)

The manometer indicates the static pressure by displacing the fluid in the tube. In making static pressure measurements, one leg of the manometer is connected to the probe and the other is left open to the atmosphere. The height difference between the levels in the two columns is the pressure in height of fluid, usually expressed in inches of water. One of the principal difficulties with the U-tube manometer relates to the fluid. If the pressure in the duct exceeds the capacity of the manometer, fluid will either be drawn into the duct or blown out onto the inspector. Also, the inspector must remember to close the seals when transporting the manometer to prevent loss of fluid and to open them before making a measurement.

The Magnehelic® pressure gauge, shown in Figure 6-2, is a product of Dwyer Instruments, Inc. It senses pressure difference by deflecting a silicone rubber diaphragm and then translating that deflection to a needle indication through a magnetic linkage. Although not as accurate as the U-tube manometer, it is much more forgiving, making it easier to use in field situations. The Magnehelic® is accurate to within 2 percent of full scale and has a high resistance to shock and vibration. It is available in over 70 ranges, from 0-0.25 in. H_2O to 0-20 psig. The most useful ranges for ventilation system inspection are 0-5, 0-20 and 0-50 in. H_2O. For inspection of high pressure drop wet scrubber systems, a 0-100 in. H_2O range may be needed.

Except for the 0-0.25 and 0-0.50 in. H_2O ranges, the Magnehelic® may be used in any orientation and can accept pressures up to 15 psig without being harmed. This property allows gauges with different ranges to be combined in one instrument package, with the gauge giving the most readable indication used for recording the measurement.

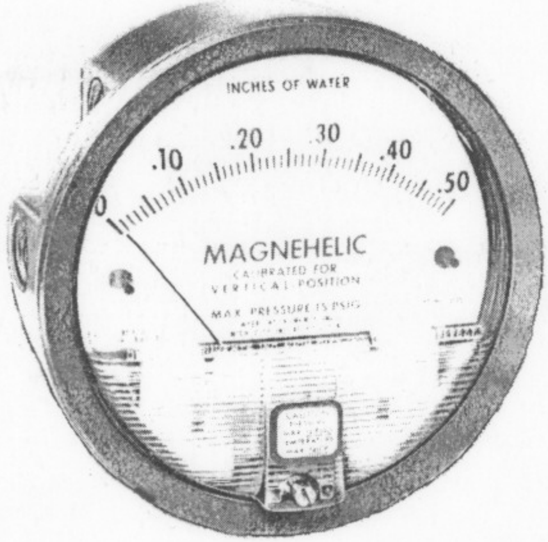


Figure 6-2. Magnehelic® pressure gauge (from *Bulletin A-30*, with permission of Dwyer Instruments, Inc.)

Because of the silicone rubber diaphragm, the ambient temperature range is limited to 20° to 140°F. This lower limitation can be accommodated when conducting inspections in cold environments by keeping the gauge in a location that is within the range and then taking it out briefly for making the measurement. For extended use under cold conditions, gauges with a lower temperature limit of -65°F are available on special order.

The Magnehelic® is not a reference instrument, so its calibration should be checked periodically. The simplest way of doing this is to check its indications against a U-tube manometer, using the set-up shown in Figure 6-3. Using a laboratory squeeze-bulb equipped with check valves, pressures from -40 to +40 in. H_2O can be easily generated. The Magnehelic® indications should be plotted against those of the manometer to check for accuracy and linearity. Gauges that give inaccurate or non-linear indications should be discarded. Also, while using the gauge its zero should be checked frequently and adjusted as needed using the set-screw on the front plate. Adjusting the zero will not affect the calibration of the gauge.

Temperature measurement

There are several techniques available for measuring temperature, including: (1) mercury thermometers, (2) dial thermometers, (3) thermistors and (4) thermocouples. Unfortunately, each of these techniques has some limitation when applied to the inspection of industrial ventilation systems. The mercury thermometer is constructed of glass and is subject to breakage, with resulting exposure to a toxic material. Also, both the mercury and dial thermometers have a limited probe extension, making the measurement of temperatures across large ducts impossible. Since locations near the wall of a hot duct will be cooler than near the center, measurements made there may not be representative of actual temperatures. The thermistor, which measures temperature through the change in resistance of a fine wire sensor, is easy to use but its response becomes non-linear over some part of its temperature range, making data interpretation difficult. Finally, the potentiometer used to measure the output of a thermocouple is not yet available in an intrinsically-safe construction and cannot be used in areas where explosive or ignitable materials may be present.

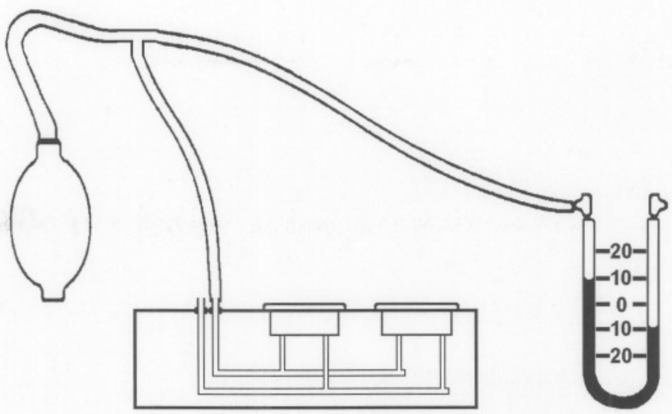


Figure 6-3. Magnehelic® calibration apparatus (Segal et al, 1985)

Despite its limitations, the thermocouple is recommended as the primary method for measuring temperatures in the inspection of industrial ventilation systems. In situations where explosive or ignitable materials may be present, use of the dial thermometer is suggested, but the inspector should be aware of the potential problems in obtaining representative measurements on large hot ducts. The thermocouple sensor is formed by joining two wires made of different metals or alloys. If the junctions at the ends of these two wires are then held at different temperatures, an electric current flows in the wire loop. This current is produced by an electromotive force whose value depends on the difference in temperature between the junctions.

The electromotive force generated by a thermocouple is measured with a potentiometer. A variety of metals and alloys are used in the construction of thermocouples, providing for measurements over different temperature ranges. The most common thermocouple, and the one recommended for use in inspections, is Type K. The Type K thermocouple has a temperature range of -400°F to +2,300°F and is constructed with a positive wire of chromel and a negative wire of alumel. Most hand-held potentiometers are calibrated for certain thermocouple types and internally convert the measured electromotive force to a temperature indication.

The thermocouple/potentiometer is not a reference instrument and must be calibrated against a National Institute of Standards and Technology (NIST) traceable thermocouple to assure highest accuracy. Since the equipment required to do this is expensive and not likely to be available to the inspector, it may be necessary to send the unit to a specialized laboratory for calibration. For most inspection situations, however, high accuracy is not required. In these cases, an acceptable evaluation of instrument accuracy may be conducted by checking its response in an ice bath and a boiling water bath. Under frequent use, this check should be done on a weekly basis. For less frequent use, it should be done prior to taking the instrument into the field.

There are several potential sources of error in making temperature measurements. One of these, use of an unrepresentative location, has already been mentioned. With the thermocouple this problem can be avoided by making measurements at several locations across the duct cross-section and averaging them. This can be done through a formal procedure, such as that to be discussed for making velocity pressure measurements, or it can be performed with random locations and mental averaging. The formal procedure will, of course, give more accurate averages. To reach locations well within the duct, the thermocouple wire will need to be supported. One of the more satisfactory techniques for doing this is to thread the wire through a small diameter copper tube, allowing the junction to protrude out the end.

Problems can also occur from the cooling of the probe due to air infiltration through the port or through leaks into the duct upstream of the measurement point. The former problem can be avoided by sealing the port in the manner described in the section on static pressure measurements. In addition, if a copper tube is used to support the thermocouple, it could be bent slightly so that it extends into the oncoming gas stream. To avoid problems from upstream leaks, the area near the measurement location should be inspected for holes in the duct or leaks in inspection hatches or expansion joints. If these are found to exist, the measurement location should be changed to an area where these leaks will have mixed into the flow. If this is not possible, the number of measurement points used to obtain an average should be increased.

Measurements downstream of evaporative coolers or wet scrubbers can be complicated due to the presence of water droplets. As these droplets impact on the sensor, the temperature will vary between the dry-bulb and wet-bulb values. However, since the degree of wetting will not be known and cannot be controlled, the exact condition of the measurement cannot be ascertained. Under these conditions, the most reasonable option is to shield the sensor from the water droplets. It should be realized, however, that doing this will likely slow the response of the sensor, requiring longer times to make the measurements.

Flow rate measurement

Measurement of gas flow rate in ducts is accomplished by first measuring the average velocity pressure and temperature of the gas stream and then calculating the average velocity. The flow rate is obtained by multiplying this average velocity by the duct cross-sectional area. Procedures for conducting this measurement are contained in 40CFR60, Appendix A, Methods 1 through 4, as part of the procedures for conducting compliance sampling. Since the level of accuracy required for inspection of industrial ventilation systems is not as high as that needed for compliance sampling, some variances to these methods will be employed to expand their application and simplify the procedures and calculations, as follows:

1. Method 1 limits the technique to ducts larger than 12 inches diameter. For inspection purposes, the procedures will be applied to ducts of all sizes. To minimize errors, a pitot tube smaller than 5/16 inch O.D. should be used in ducts smaller than 12 inches diameter.

2. Method 1 prohibits the location of measurement points within 1 inch of the wall for ducts larger than 24 inches diameter and within 0.5 inch of the wall for ducts smaller than 24 inches diameter. For inspection purposes, measurement points will be at the locations prescribed by the location procedures, with no adjustments made.

3. Method 2 requires the determination of the apparent dry molecular weight using Method 3 and the moisture content using Method 4 in order to calculate the gas velocity and flow rate. For inspection purposes, an apparent dry molecular weight of 28.95 and a moisture content of zero will be assumed.

4. Method 2 requires the measurement of the absolute stack pressure in order to calculate the gas velocity and flow rate. For inspection purposes, an absolute stack pressure of 29.92 in. Hg will be assumed.

With these changes, the procedures for determining flow rate become comparable to those recommended by ACGIH.

Measurement of velocity pressure can be made with either a standard pitot tube, shown in Figure 6-4, or an S-type (Stausscheid) pitot tube, shown in Figure 6-5. The S-type pitot is preferred when there are particles in the gas stream that could plug the static pressure holes of a standard pitot. If the construction of a standard pitot conforms to Section 2.7 of Method 2, it does not have to be calibrated and may be assigned a pitot coefficient, C_p, of 0.99. An S-type pitot may be assigned a pitot coefficient of 0.84 if its construction conforms to Section 4.1 of Method 2. Since most all of the commercially available pitot tubes conform to these requirements, they will not be repeated here. Procedures for calibrating an S-type pitot may be found in Section 4 of Method 2.

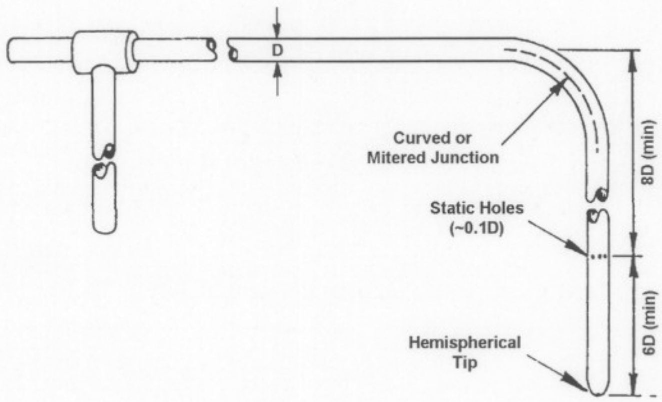


Figure 6-4. Standard pitot tube

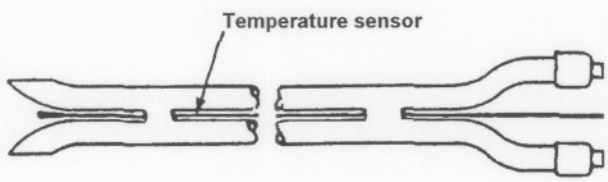


Figure 6-5. Type S pitot tube

For highest accuracy, pressures measured with the pitot tube should be read using an inclined manometer. The inclined manometer is similar to the U-tube manometer previously discussed, except that the first inch is inclined to improve the ability to read low pressures accurately. Since standard air flowing at 4005 ft/min has a velocity pressure of 1 in. H_2O, most velocity pressure readings will be made in this inclined area. If less accuracy is acceptable, a 0-2 in. H_2O Magnehelic® pressure gauge could be substituted for the manometer.

Average velocity pressure is determined by averaging the *square roots* of velocity pressures measured at prescribed locations in the duct and then squaring the result. Temperature is measured at the same time by attaching a thermocouple to the pitot tube, and its average is calculated arithmetically. The number of locations that are used to compute the averages depends on the degree of accuracy desired. Figure 6-6 provides guidance in determining the minimum number of locations, based on the upstream and downstream distances to flow obstructions. In general, any condition other than straight duct constitutes an obstruction. In using this chart, the number of locations is first determined for the distance downstream from an obstruction by reading vertically upward from the lower x-axis and then for the distance upstream from an obstruction by reading vertically downward from the upper x-axis. The larger of these two numbers is the minimum number of locations or traverse points. The number of locations for rectangular ducts is based on equivalent diameter calculated from:

$$D_{eq} = \frac{2LW}{L + W} \tag{6-1}$$

where L and W are the lengths of adjacent sides of the duct.

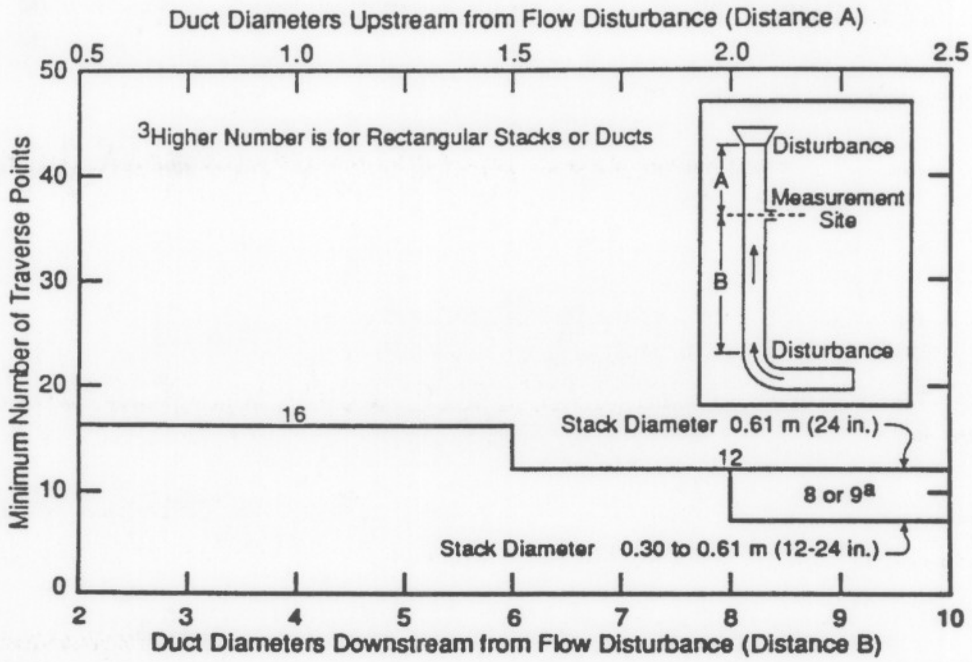


Figure 6-6. Minimum number of traverse points (40CFR60, Appendix A)

For circular ducts, the number of traverse points determined from Figure 6-6 is divided by two to determine the number of measurement points on each diameter. The location of each traverse point is given in Table 6-1 as a percentage of duct diameter from the inside wall to the point location. With rectangular ducts, the cross-section is divided into a grid of equal rectangular areas and measurements are made in the center of each grid element. The grid configuration should be either 3 x 3, 4 x 3 or 4 x 4, depending on the number of measurement locations needed.

In most ducts the direction of the gas flow is essentially parallel to the duct walls. However, downstream of such devices as cyclones, inertial demisters or ducts with tangential entry, a swirling or cyclonic motion may be encountered. When high accuracy is desired, it should be determined whether the degree of cyclonic flow is enough to cause significant error in the measurements. The procedure for accomplishing this is as follows:

1. Level and zero the manometer.

2. Connect an S-type pitot to the manometer.

3. Place the pitot tube at each traverse point so that the openings are perpendicular to the duct cross-sectional plane. In this position, each tube should be reading static pressure and the indication of the manometer should be zero.

4. If the differential pressure is not zero, rotate the pitot tube until a zero reading is obtained and record the resulting angle.

5. Calculate the average of the absolute values of the angles, including those angles that were zero (no rotation required). If the average is greater than 20 degrees, the flow conditions at that location are not acceptable.

Table 6-1. Location of traverse points in circular ducts
(Percent of duct diameter from inside wall to traverse point)

Point Number	Number of Points on a Diameter		
	4	6	8
1	6.7	4.4	3.2
2	25.0	14.6	10.5
3	75.0	29.6	19.4
4	93.3	70.4	32.3
5		85.4	67.7
6		95.6	80.6
7			89.5
8			96.8

Prior to making any measurements of velocity pressure, a leak check should be conducted, as follows:

1. Blow through the pitot impact opening until 3 in. H_2O velocity pressure registers on the pressure gauge, then close-off the opening. The pressure should remain stable for at least 15 seconds.

2. Repeat the above step for the static opening, except using suction to obtain -3 in. H_2O.

Once an acceptable location has been identified, the velocity pressure and temperature measurements are performed and the averages are calculated. The average gas velocity is then determined from:

$$V = 2.9C_p(\sqrt{VP})_{avg}\sqrt{T_{avg}} \tag{6-2}$$

where V = average gas velocity (ft/sec)
C_p = pitot coefficient (dimensionless)
VP = velocity pressure (in. H_2O)
T = gas temperature (°R)

The average gas flow rate is then determined from:

$$Q = 60VA \tag{6-3}$$

where Q = average gas flow rate (ft^3/min)
A = duct cross-sectional area (ft^2)

Other methods are available for determining gas velocity and these are typically applied to measurements at the hood face. To determine flow rate with these devices, it would be necessary to make several velocity measurements across the face of the hood, determine an average and then multiply it by the area of the hood opening. The accuracy of this technique will depend primarily on the number of measurement locations used.

One of the more common velocity measuring instruments is the rotating vane instrument shown in Figure 6-7. This anemometer has a small lightweight propeller that rotates as air flows through the instrument. The instrument is calibrated in feet and has to be used with a timing device to determine the velocity.

A second type of measuring instrument is the swinging vane anemometer shown in Figure 6-8. Inside the instrument case is an aluminum vane which deflects the pointer on the scale in proportion to the velocity. Air flows through the probe and connecting tube into the case and then through the channel which contains the vane.

A third type of anemometer is the "hot wire" shown in Figure 6-9. The probe of this instrument is provided with a wire element that is heated with current from batteries in the instrument case. As air flows over the element, its temperature changes from what it was in still air and the accompanying resistance change is translated into velocity on the indicating scale of the instrument.

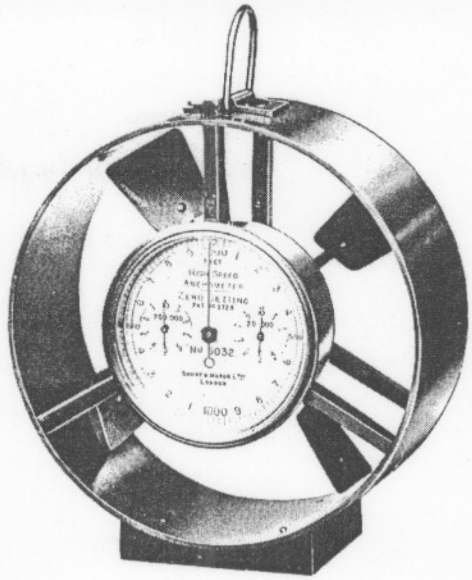


Figure 6-7. Rotating vane anemometer (from *Manual for the Balancing and Adjustment of Air Distribution Systems*, with permission of SMACNA)

Figure 6-8. Swinging vane anemometer (from *Manual for the Balancing and Adjustment of Air Distribution Systems*, with permission of SMACNA)

Fan speed measurement

Techniques available for the measurement of fan speed include: (1) standard tachometers, (2) strobetachometers and (3) phototachometers. Strobetachometers and phototachometers are expensive instruments that are not likely to be available to the inspector. Also, phototachometers require reflective tape to be placed on the drive shaft and this can only be done when the shaft is not moving.

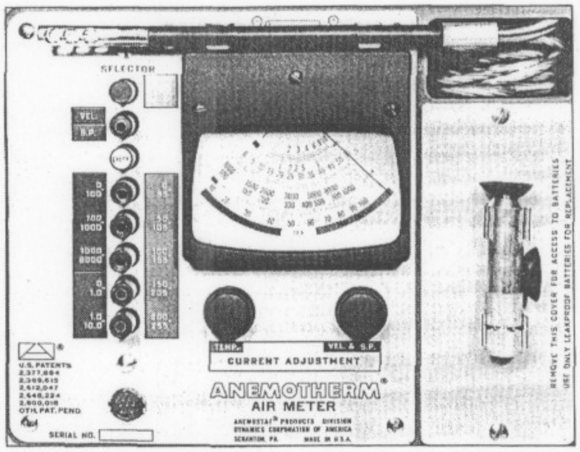


Figure 6-9. Hot wire anemometer (from *Manual for the Balancing and Adjustment of Air Distribution Systems*, with permission of SMACNA)

The recommended technique for measuring fan speed is the standard tachometer, and the easiest location for making the measurement is on the end of the shaft, through the access hole in the belt guard. If no access hole is provided, the inspector should request the assistance of plant personnel. Under no circumstances should the inspector attempt to obtain access to the shaft end by enlarging the mesh covering on the belt guard. An alternative measurement location is on the shaft, using the roller attachment supplied with the tachometer. However, using this method requires knowledge of the shaft diameter in order to calculate the rotational speed from the tachometer reading.

An estimate of the fan speed can be obtained by measuring the diameter of the fan and motor sheaves and using their ratio, as follows:

$$\text{Fan rpm} = \text{MS}\left(\frac{\text{MD}}{\text{FD}}\right) \qquad (6\text{-}4)$$

where MD = motor sheave diameter
FD = fan sheave diameter
MS = motor speed (rpm)

Motors are generally available in the nominal speeds of 600, 1200, 1800, 2400 and 3600 rpm. The actual speed is somewhat less than the nominal value and is stamped on the motor nameplate.

Horsepower measurement

Determination of operating horsepower is an involved process that is not likely to be performed very often. Also, because of the procedures and measurements required, it should be performed only by plant personnel and never by the inspector. The procedure plant personnel should use is as follows:

1. Prepare a graph like that shown in Figure 6-10, with horsepower on the x-axis and amperage on the y-axis.

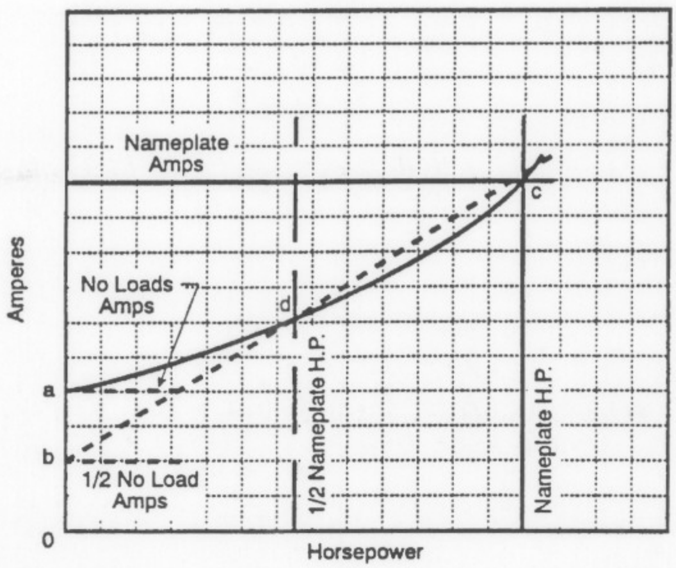


Figure 6-10. Determination of operating horsepower (SMACNA, 1967)

2. Disconnect the motor from the fan and measure the amperage when running at no load. Mark this reading as point "a" on the y-axis. Divide the no-load amperage by two and mark this value as point "b" on the y-axis.

3. Read the full load amperage from the motor nameplate and draw a horizontal line across the graph at this value. Read the rated horsepower from the nameplate and draw a vertical line at this value until it intersects the full load amperage line. Call this intersection point "c". Draw a straight line from "b" to "c".

4. Divide the rated horsepower by two and draw a vertical line at this value until it intersects the line from "b" to "c". Call this intersection point "d". Draw a smooth curve through points "a", "d" and "c".

5. Connect the motor to the fan and measure the amperage when running at load. Read the corresponding horsepower from the curve "a-d-c".

Once the relationship between amperage and horsepower is determined, it could be used in subsequent inspections, provided the motor has not been changed.

Estimates of operating horsepower can be made in two ways. Perhaps the simplest is to have plant personnel measure the amperage while running at load. This value would then be divided by the full load amperage from the nameplate and then multiplied by the rated horsepower to obtain the estimated operating horsepower. Another method would require plant personnel to measure both voltage and amperage while running at load. For three phase motors, these values would then be substituted into:

$$\text{bhp} = \sqrt{3}\left(\frac{\text{VAfe}}{746}\right) \tag{6-5}$$

where V = voltage
A = amperage
f = power factor
e = motor efficiency

For single phase motors, the square root of 3 is replaced by one. If the power factor were also measured and the motor efficiency determined from the manufacturer, then a good value of operating horsepower could be determined with this relationship. However, because of the time involved, we can usually only estimate these parameters and thereby obtain an estimate of operating horsepower. In the absence of other information, a combined factor of 0.80-0.85 should be used for the product of power factor times motor efficiency.

Use of grounding cables

When working with portable instruments in areas where potentially explosive or ignitable materials are present, all metal probes should be grounded to the duct to avoid a static discharge. The most satisfactory technique is to use a stranded cable with a pipe clamp attached to one end and a spring-loaded jaw clamp on the other, as shown in Figure 6-11. The pipe clamp (1) is firmly attached to the probe and the jaw clamp (2) is attached to the duct, usually at a flange or support. Care should be taken to assure a good connection at the duct and that all paint and rust has been penetrated. One way to check the connection would be to measure the resistance between the probe and the duct using an explosion-proof ohmmeter. If the resistance is less than 3 ohms, the connection is good. Guidance on when to use grounding cables is provided by the following list:

1. When the moisture content of the gas stream is low.
2. When the gas stream velocity across the probe is high.
3. When the gas stream contains a relatively high mass concentration of small-sized particles.
4. When there is the possibility of dust deposits in the bottom of the duct.
5. When there is any question about the need for a grounding cable.

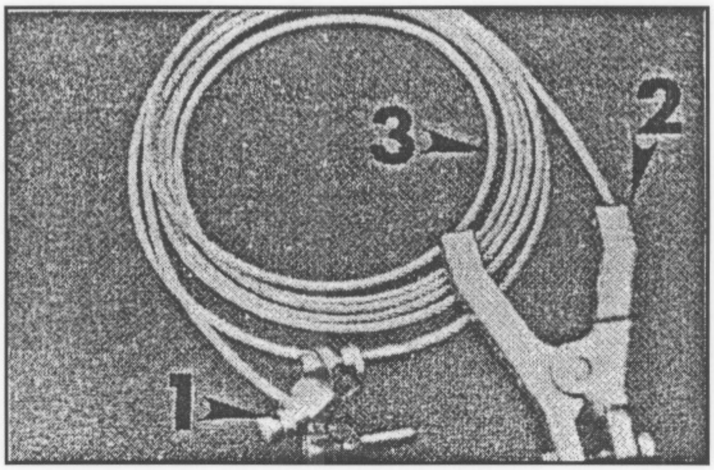


Figure 6-11. Typical grounding cable (Segal et al, 1985)

References

ACGIH, *Industrial Ventilation*, Twentieth Edition, Cincinnati, 1988.

Richards, J., *Air Pollution Source Field Inspection Notebook*, Revision 2, USEPA, APTI, June 1988.

Segal, R., and J. Richards, *Inspection Techniques for Evaluation of Air Pollution Control Equipmen*", Volume II, EPA-340/1-85-022b, September 1985.

SMACNA, *Manual for the Balancing and Adjustment of Air Distribution Systems*, First Edition, Vienna (Virginia), 1967.

USEPA, *Determination of Stack Gas Velocity and Volumetric Flow Rate*, 40CFR60, Appendix A, Method 2.

USEPA, *Sample and Velocity Traverses for Stationary Sources*, 40CFR60, Appendix A, Method 1.

Chapter 7

Ventilation System Inspection

The efforts involved in inspecting air pollution sources are generally categorized according to a system of "levels", as follows:

Level 1	Visual evaluation of stack opacity and fugitive emissions from off the plant site.
Level 2	On site evaluation of the control system relying on plant instruments for the values of any inspection parameters.
Level 3	Similar to Level 2, but relying on measurements by the inspector to determine missing or inaccurate inspection parameters.
Level 4	Similar to Level 3, but including the development of a process flowchart, determination of measurement port locations and evaluation of safety hazards and protective equipment needs. If the process or control equipment do not change, this level of inspection would only be conducted once.

The inspection level that is actually utilized is dictated by the individual situation and based on the judgment of the inspector. For example, if a Level 1 inspection indicates no problems, the inspector may elect to terminate the inspection and proceed to another facility. Or, if in the course of a Level 2 inspection, critical information is needed to complete the evaluation, the inspector may elect to proceed to Level 3, making on-site measurements to obtain the data.

These same four levels can also be applied to the inspection of industrial ventilation systems. Level 1 would be limited to an off-site evaluation of fugitive emissions from hoods and ducts, and the additional items in a Level 4 would focus on port locations and safety issues. Most inspections will be conducted at Level 2, with the occasional need for a Level 3 evaluation, and would follow the same general approach used with control devices.

In previous lessons, inspection points and procedures have been discussed that can be used in both Level 2 and Level 3 inspections. In this lesson, those items will be organized according to level and, in some cases, expanded. The purpose of this is to provide both a review of the inspection points and a "check-list" for conducting field evaluations. Under the Level 3 category, only those additional items will be listed. Level 3 inspections would also include the Level 2 items.

Level 2 inspections

Hoods

1. *Capture efficiency:* visual evaluation of fugitive losses as indicated by escaping dust or refraction lines.

2. *Physical condition:* hood modifications or damage that could affect performance; evidence of corrosion.

3. *Fit of "swing-away" joints:* evaluation of gap distance between hood system and duct system on movable hoods.

4. *Hood position/cross-drafts:* location of hood relative to point of contaminant generation; effect of air currents on contaminant capture.

Ducts

1. *Physical condition:* indications of corrosion, erosion or physical damage; presence of fugitive emissions.

2. *Position of emergency dampers:* emergency by-pass dampers should be closed and not leaking.

3. *Position of balancing dampers:* a change in damper positions will change flow rates; mark dampers with felt pen to document position for later inspections.

4. *Condition of balancing dampers:* damper blades can erode, changing system balance; have plant personnel remove a few dampers to check their condition.

Coolers

1. *Physical condition:* indications of corrosion, erosion or physical damage; presence of fugitive emissions.

2. *Outlet temperature:* observe plant instruments to determine cooler effectiveness; if controller is used, compare to set-point value.

3. *Spray pattern/nozzle condition:* indications of effective atomization on evaporative coolers.

4. *Water flow rate:* observe plant flow meters or pressure gauges to evaluate changes in water flow rate on evaporative coolers.

Fans

1. *Physical condition:* indications of corrosion.

2. *Vibration:* indications of balance problems due to material build-up or wheel erosion or corrosion; severely vibrating fans are a safety hazard.

3. *Belt squeal:* squealing belts under normal operation indicate a loss of air volume.

4. *Fan wheel build-up/corrosion:* internal inspection of non-operating fans.

5. *Condition of isolation sleeves:* check vibration isolation sleeves for holes.

6. *Rotation direction:* check rotation direction with direction marked on fan housing.

Level 3 inspections

Hoods

1. *Estimated volume:* estimate flow rate using SP_h, temperature and hood configuration.

2. *Actual volume:* determine flow rate by measuring VP and temperature.

Ducts

1. *Change in gas temperature:* measure temperature change across a section of duct to evaluate air infiltration.

2. *Change in static pressure:* measure static pressure change across a duct section to evaluate duct deposits; compare measurements to calculations for clean duct.

3. *Actual volume:* determine flow rate by measuring VP and temperature.

Coolers

1. *Inlet and outlet temperatures:* measure inlet and outlet temperature to evaluate cooling effectiveness.

2. *Water requirement:* estimate water requirement for evaporative coolers using inlet and outlet temperatures and enthalpy relationships; compare to actual use information supplied by plant personnel or indicated by flow meter.

3. *Water turbidity:* perform settling test on water sample gathered by plant personnel to evaluate particle size of solids.

4. *Air volume:* estimate air volume in dilution cooling systems using measured temperatures and enthalpy relationships; could also be determined by measuring VP and temperature.

Fans

1. *Volume changes:* estimate new flow rate using known performance (Q, rpm and q) and new rpm.

2. *Estimated volume:* estimate flow rate using rpm, FSP, temperature and ratings table; could also be done using bhp, FSP, temperature and ratings table.

Use of flowcharts

One of the first steps in solving essentially any technical problem is to draw a picture. This is especially true in the inspection of air pollution control equipment. Problems which result in excessive emissions are rarely due to simple failures of a single component, but are instead usually due to combinations of problems affecting the entire system. The inspection flowchart is a valuable tool in sorting out the usually complex and sometimes conflicting data. Additional advantages include:

- Improves inspector's ability to communicate the results of an inspection to plant personnel and to inspection supervisors.

- Organizes inspection data making anomalous trends easier to identify.

- Reduces inspection report preparation time.

Because they can serve many purposes, there are many levels of sophistication in flowchart preparation. Flowcharts for air pollution control equipment inspections should be relatively simple. One example is shown in Figure 7-1. Here, major equipment items are shown as simple blocks, while items such as the fan, pumps and the stack are represented with standard equipment symbols. The gas flow path and important material and utility streams are included and labeled for easy reference. Parameter monitoring locations are shown using standard instrument symbols.

Once the flowchart has been prepared, it can be duplicated and used to record data on individual inspections. Recorded data should be examined to be sure that static pressures decrease from the inlet of the system to the fan and from the fan to the stack. Likewise, in hot systems temperature should decrease toward the fan and away from the fan. A slight temperature increase across the fan may occur because of compression of the gas stream. Data that do not follow expected trends should be re-evaluated.

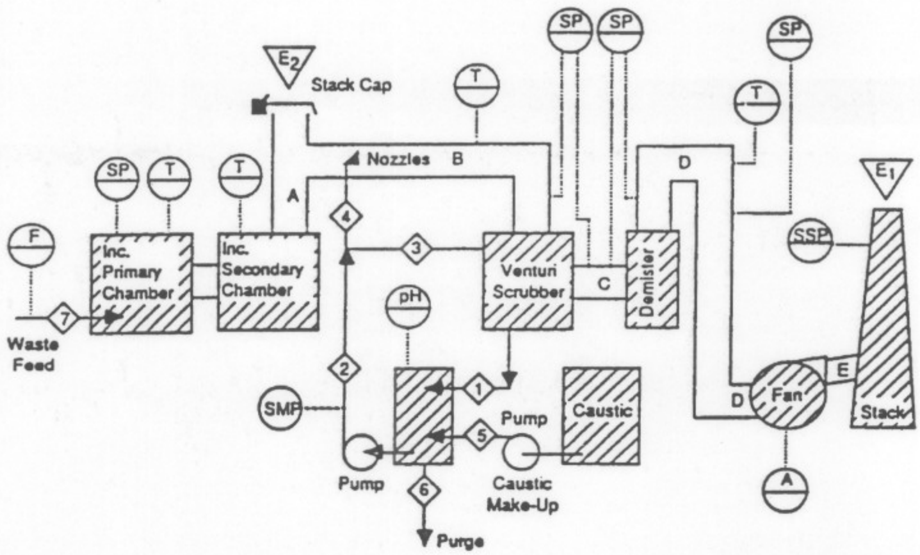


Figure 7-1. Example flowchart

References

Richards Engineering, “Flowchart preparation for air pollution source inspection", USEPA, SSCD, September 1989.

Appendix A

Conversion Factors

To Convert	Into	Multiply By
atmospheres	inches Hg	29.92
	inches H_2O	407
	pounds/square inch	14.7
	bars	1.0133
	millimeters Hg	760
Btu/minute	horsepower	0.02356
	kilowatt	0.01757
cubic feet	cubic inches	1728
	cubic meters	0.02832
	gallons	7.4805
	liters	28.32
cubic feet/minute	liters/second	0.4720
	cubic meters/second	4.72×10^{-4}
cubic meters	cubic feet	35.31
	cubic inches	61,023
	gallons	264.2
	liters	1000
feet	centimeters	30.48
	meters	0.3048
feet/minute	meters/second	0.00508
horsepower	kilowatt	0.7457
inches	feet	0.0833
	meters	0.0254
	centimeters	2.54
inches H_2O	atmospheres	2.458×10^{-3}
	inches Hg	0.07355
	pounds/square inch	0.03613
liters	cubic feet	0.03531
	cubic inches	61.02
	gallons	0.2642
meters	feet	3.281
	inches	39.37
meters/second	feet/minute	196.8
pounds	grains	7000

To Convert	Into	Multiply By
	grams	453.6
	ounces	16
pounds/cubic foot	kilograms/cubic meter	16.02
pounds/square inch	atmospheres	0.068
	inches Hg	2.036
	inches H_2O	27.68
square feet	square inches	144
	square meters	0.093

Appendix B

Additional Readings

ACGIH, *Industrial Ventilation Manual*, Twentieth Edition, Cincinnati, 1988.

Alden, J.L., *Design of Industrial Exhaust Systems*, Industrial Press, New York, 1939.

Aldina, G.J., and J.A. Janke, *APTI Course 450: Source Sampling for Particulate Pollutants--Instructor's Guide*, EPA-450/2-80-003, February 1980.

AMCA, "Fan Systems", *Fan Application Manual*, Part 1, Publication 201, Arlington Heights (Illinois), 1979.

AMCA, "Field Performance Measurements", *Fan Application Manual*, Part 3, Publication 203, Arlington Heights (Illinois), 1976.

AMCA, "Troubleshooting", *Fan Application Manual*, Part 2, Publication 202, Arlington Heights (Illinois), 1976.

ANSI, *Fundamentals Governing the Design and Operation of Local Exhaust Systems*, American National Standard Z 9.2-1971, 1972.

ANSI, *Installation of Blower and Exhaust Systems for Dust, Stock and Vapor Removal or Conveying*, American National Standard Z 33.1, 1961.

ANSI, *Safety Code for the Use, Care and Protection of Abrasive Wheels*, American National Standard B 7.1-1970, 1970.

ASHRAE, *Equipment Guide and Data Book*, New York, 1972.

ASHRAE, *Handbook of Fundamentals*, Atlanta, 1985.

ASHRAE, *HVAC Systems and Applications*, Atlanta, 1987.

Astleford, W., *Engineering Control of Welding Fumes*, NIOSH Publication 75-115, 1974.

Baliff, J., L. Greenburg and A.C. Stern, "Transport velocities for industrial dusts--an experimental study", *Ind. Hyg. Quar.*, ***9***, 85 (1948).

Barr, H.S., R.H. Hocutt and J.B. Smith, *Cotton Dust Controls in Yarn Manufacturing*, NIOSH Publication No. 74-114, 1974.

Barritt, S.L., "Duct design by programmable calculator", *Heating, Piping & Air Conditioning*, December 1978, p.28.

Bastress, E., J. Niedzwecki and A. Nugent, *Ventilation Requirements for Grinding, Buffing and Polishing Operations*, NIOSH Publication 75-107, 1974.

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

Brandt, A.D., "Should air be recirculated from industrial exhaust systems?", *Heating, Piping & Air Conditioning*, ***19***, 69 (1947).

Brandt, A.D., *Industrial Health Engineering*, Wiley, New York, 1947.

Brown, E.J., "How to select multiple-leaf dampers for proper air flow control", *Heating, Piping & Air Conditioning*, April 1960, p.167.

Brown, E.J., and H.E. Staub, "How to estimate system total pressure requirements", *Heating, Piping & Air Conditioning*, January 1971, p. 12.

Burgess, W.A., and J. Murrow, "Evaluation of hoods for low volume-high velocity exhaust systems", *AIHA Journal*, ***37***, 546 (1976).

Burgess, W.A., M.J. Ellenbecker and R.D. Treitman, *Ventilation for Control of the Work Environment*, Wiley-Interscience, New York, 1989.

Burton, D.J., *Industrial Ventilation Workbook*, DJBA, Salt Lake City, 1989.

Caplan, K.J., "Balance with blast gates--a precarious balance", *Heating, Piping & Air Conditioning*, February 1983, p. 47.

Chamberlin, R., "The control of beryllium machining operations", *Arch. Ind. Health*, ***19****(2)*, 231 (1959).

Chen, S.Y.S., "Design procedure for duct calculations", *Heating, Piping & Air Conditioning*, January 1981, p.41.

Cheremisinoff, P.N., and R.A. Young, "Fans and blowers", *Pollution Engineering*, July 1974, p. 54.

Chicago Blower Corporation, *A Course in Fan Arrangements and Classes*, CBC-300, 1981.

Chicago Blower Corporation, *Air Handling Equipment*, 1982.

Chicago Blower Corporation, *Basic Course in Fan Density Correction*, CBC-200, 1981.

Chicago Blower Corporation, *Basic Course in Fan Selection*, CBC-100, 1981.

Churchill, S.W., "Friction factor equation span all fluid flow regimes", *Chemical Engineering*, ***84***, 56 (1977).

Clapp, D.E., D.S. Groh and J.D. Nenandic, "Ventilation design by microcomputer", *AIHA Journal*, ***43***, 212 (1982).

Conroy, L., M.J. Ellenbecker and M. Flynn, "Prediction and measurement of velocity into flanged slot hoods", *AIHA Journal*, ***49***, 226 (1988).

Constance, J.A., "Estimating air friction in triangular ducts", *Air Conditioning, Heating and Ventilating*, ***60****(6)*, 85 (1963).

Crowder, J.W., and K.J. Loudermilk, "Balancing of industrial ventilation systems", *JAPCA*, ***32***, 115 (1982).

Dalla Valle, J.M., "Velocity characteristics of hoods under suction", *ASHVE Transactions*, ***38***, 387 (1932).

Dalla Valle, J.M., and T. Hatch, "Studies in the design of local exhaust hoods", *ASME Transactions*, ***54***, 31 (1932).

Dalla Valle, J.M., *Exhaust Hoods*, Industrial Press, New York, 1952.

Danielson, J.A., ed., *Air Pollution Engineering Manual*, Second Edition, EPA AP-40, May 1973.

Djamgowz, O.T., and S.A.A. Ghoneim, "Determining the pick-up air velocity of mineral dusts", *Can. Mining J.*, July 1974, p. 25.

Durr, D., N. Esman, C. Stanley and D. Weyel, "Pressure drop in elbows", *Appl. Ind. Hyg.*, **2**, 57 (1986).

Ellenbecker, M.J., R. Gempel and W.A. Burgess, "Capture efficiency of local exhaust ventilation systems", *AIHA Journal*, ***44***, 752 (1983).

Esman, N., D. Weyel and F. McGuigan, "Aerodynamic properties of exhaust hoods", *AIHA Journal*, ***47***, 448 (1986).

First, M.W., and L. Silverman, "Airfoil pitometer", *I & EC*, ***42***, 301 (1950).

Flanigan, L.J., S.G. Talbert, D.E. Semones and B.C. Kim, *Development of Design Criteria for Exhaust Systems for Open Surface Tanks*, NIOSH Publication 75-108, 1975.

Fletcher, B., "Centerline velocity characteristics of local exhaust ventilation hoods", *AIHA Journal*, ***43***, 626 (1982).

Fletcher, B., "Centerline velocity characteristics of rectangular hoods and slots under suction", *Ann. Occup. Hyg.*, ***20***, 141 (1977).

Fletcher, B., "Effect of flanges on the velocity in front of exhaust ventilation hoods", *Ann. Occup. Hyg.*, ***21***, 265 (1978).

Fletcher, B., and A. Johnson, "Velocity profiles around hoods and slots and the effects of an adjacent plane", *Ann. Occup. Hyg.*, ***25***, 365 (1986).

Flynn, M., and M.J. Ellenbecker, "Capture efficiency of flanged circular local exhaust hoods", *Ann. Occup. Hyg.*, ***30***, 497 (1986).

Flynn, M., and M.J. Ellenbecker, "Empirical validation of theoretical velocity fields into flanged circular hoods", *AIHA Journal*, ***48***, 380 (1987).

Flynn, M., and M.J. Ellenbecker, "The potential flow solution for air flow into a flanged circular hood", *AIHA Journal*, ***46***, 318 (1985).

Garrison, R., "Centerline velocity gradients for plain and flanged local exhaust inlets", *AIHA Journal*, ***42***, 739 (1981).

Garrison, R., "Velocity calculation for local exhaust inlets--graphical design", *AIHA Journal*, ***44***, 941 (1983).

Garrison, R., "Velocity calculation for local exhaust inlets--empirical design equations", *AIHA Journal*, ***44***, 937 (1983).

Garrison, R., and D. Byers, "Static pressure and velocity characteristics of circular nozzles for high velocity/low volume exhaust ventilation", *AIHA Journal*, ***41***, 803 (1980).

Garrison, R., and D. Byers, "Static pressure, velocity and noise characteristics of rectangular nozzles for high velocity/low volume exhaust ventilation", *AIHA Journal*, ***41***, 855 (1980).

Geissler, H., *Purchased Fan Performance*, Reprint No. 5483, Westinghouse Electric Corporation, Pittsburgh, 1959.

Goldfield, J., and F.E. Brandt, "Dust control techniques in the asbestos industry", *AIHA Journal*, ***35***, 799 (1974).

Goodfellow, H., *Advanced Design of Ventilation Systems for Contaminant Control*, Elsevier, New York, 1985.

Goodier, J.L., E. Boudreau, G. Coletta and R. Lucas, *Industrial Health and Safety Criteria for Abrasive Blast Cleaning Operations*, NIOSH Publication 75-112, 1975.

Guffey, S.E., and J. Hickey, "Equations for redesign of existing ventilation systems", *AIHA Journal*, ***44***, 819 (1983).

Guffey, S.G., "An easier calculation system for ventilation design", *AIHA Journal*, ***44***, 627 (1983).

Hama, G.M., "A calibrating wind tunnel for air measuring instruments", *Air Engineering*, ***41***, 18 (1967).

Hama, G.M., "Calibration of Alnor velometers", *AIHA Journal*, ***19***, 477 (1958).

Hama, G.M., and L.S. Curley, "Instrumentation for the measurement of low velocities with a pitot tube", *AIHA Journal*, ***28***, 204 (1967).

Hama, G.M., "Ventilation control of dust from bagging operations", *Heating and Ventilating*, April 1948, p.91.

Hama, G.M., and K.J. Bonkowski, "Ventilation requirements for airless spray painting", *Heating, Piping & Air Conditioning*, October 1970, p. 80.

Hampl, V., and O.E. Johnson, "Control of wood dust from horizontal belt sanding", *AIHA Journal*, ***46***, 567 (1985).

Hampl, V., O.E. Johnson and D.M. Murdock, "Application of an air curtain exhaust system at a milling process", *AIHA Journal*, ***49***, 167 (1988).

Hatch, T., "Design of exhaust hoods for dust control systems", *J. Ind. Hyg. Toxicol.*, ***18***, 595 (1936).

Healy, J.H., M.N. Patterson and E.J. Brown, "Pressure losses through fittings used in return air duct systems", *ASHRAE Transactions*, ***68***, 281 (1962).

Hemeon, W.C.L., *Plant and Process Ventilation*, Second Edition, Industrial Press, New York, 1963.

Hogan, E.E., "Design tips for better O&M", *Pollution Engineering*, March 1983, p. 16.

Huebener, D.J., and R.T. Hughes, "Development of push-pull ventilation", *AIHA Journal*, ***46***, 262 (1985).

Huebscher, R.G., "Friction equivalents for round, square and rectangular ducts", *Heating, Piping & Air Conditioning*, ***19***, 127 (1947).

Hughes, R.T., and A.A. Amendola, "Recirculating exhaust air: guides, design parameters and mathematical modeling", *Plant Engineering*, March 18, 1982, p.47.

Jorgensen, Robert, ed., *Fan Engineering*, Seventh Edition, Buffalo Forge Company, Buffalo, 1970.

Kane, J.M., "Design of exhaust systems", *Health and Ventilating*, ***42***, 68 (1947).

Kane, J.M., "The application of local exhaust ventilation to electric melting furnaces", *Trans. Am. Foundrymen's Assoc.*, ***52***, 1351 (1945).

Kashdan, E.R., D.W. Coy, J.J. Spivey, T. Cesta and H.D. Goodfellow, *Technical Manual: Hood System Capture of Process Fugitive Particulate Emissions*, EPA-600/7-86-016, April 1986.

Kemner, W., R. Gerstle and Y. Shah, *Performance Evaluation Guide for Large Flow Ventilation Systems*, EPA-340/1-84-012, May 1984.

Koshland, C.P., and M.G. Yost, "Use of a spreadsheet in the design of an industrial ventilation system", *Appl. Ind. Hyg.*, **2**, 204 (1987).

Kurz, J.L., "Comparison between electronic and pitot-tube multi-point air velocity sensors used in air flow measuring stations", *Pollution Equipment News*, December 1984, p. 58.

Loeffler, J.J., "Simplified equations for HVAC duct friction", *ASHRAE J.*, January 1980, p. 76.

Lynch, J.R., "Computer design of industrial exhaust systems", *Heating, Piping & Air Conditioning*, September 1968, p. 44.

Lynch, J.R., "Industrial ventilation: a new look at an old problem", *Michigan's Occupational Health*, ***19(3)***, 4 (1974).

Madison, R.D., and W.R. Elliot, "Friction charts for gases including correction for temperature, viscosity and pipe roughness", *Heating, Piping & Air Conditioning*, ***18***, 107 (1946).

Malchaire, J.B., "Design of industrial exhaust systems using a programmable calculator", *Ann. Occup. Hyg.*, ***24***, 217 (1981).

Markert, J.W., "Use of total pressure in air system design, fan selection", *Heating, Piping & Air Conditioning*, October 1969, p. 70.

McDermott, H.J., *Handbook of Ventilation for Contaminant Control*, Ann Arbor Science, Ann Arbor, 1977.

McInnes, R.G., B.R. Hobbs and S.V. Capone, *Guide for Inspecting Capture Systems and Control Devices at Surface Coating Operations*, EPA Contract 68-01-6316, Task 33, May 1982.

Midwest Research Institute, *Design Review Guide for Materials Handling Operations: Chapter 4 -- Capture/Control Systems*, EPA Contract 68-01-6314, Task 6, February 1983.

Morse, F.B., ed., *Trane Air Conditioning Manual*, The Trane Company, La Crosse, 1965.

NIOSH, *Industrial Ventilation 588: Student Manual*, August 1986.

Obler, H., "Ventilation modifications", *Heating, Piping & Air Conditioning*, February 1980, p. 65.

Ower, E., and R.C. Pankhurst, *The Measurement of Air Flow*, Pergamon Press, Oxford, 1977.

Pring, R.T., J.F. Knudsen and R. Dennis, "Design of exhaust ventilation for solid material handling", *I & EC*, ***41***, 2442 (1949).

Rajhans, G.S., and R.W. Thompkins, "Critical velocities of mineral dusts", *Can. Mining J.*, October 1967, p. 85.

Reason, J., "Fans", *Power*, September 1983.

Rennix, C.P., "Computer assisted ventilation design and evaluation", *Appl. Ind. Hyg.*, **2**, 32 (1987).

Richards, J., *Air Pollution Source Field Inspection Notebook*, Revision 2, USEPA, APTI, June 1988.

Rogers, A.N., *Evaluation in Fan Selection*, Reprint No. 5037, Westinghouse Electric Corporation, Pittsburgh, 1954.

Rogers, A.N., *Selection of Fan Types*, Reprint No. 5312, Westinghouse Electric Corporation, Pittsburgh, 1957.

Rutgers University, *Industrial Ventilation: Student Manual*, August 1989.

Schulte, H.F., E.C. Hyatt and F.S. Smith, "Exhaust ventilation for machine tools used on materials of high toxicity", *Arch. Ind. Hyg. Occup. Med.*, ***5***, 21 (1952).

Segal, R., and J. Richards, *Inspection Techniques for Evaluation of Air Pollution Control Equipment*, Volume II, EPA-340/1-85-022b, September 1985.

Shotwell, H.P., "A ventilation design program for hand-held programmable computers", *AIHA Journal*, ***45***, 749 (1984).

Silverman, L., "Centerline velocity characteristics of round openings under suction", *J. Ind. Hyg. Toxicol.*, ***24***, 259 (1942).

Silverman, L., "Fundamental factors in the design of lateral exhaust hoods for industrial tanks", *J. Ind. Hyg. Toxicol.*, ***23***, 187 (1941).

Silverman, L., "Velocity characteristics of narrow exhaust slots", *J. Ind. Hyg. Toxicol.*, ***24***, 267 (1942).

Sisson, W., "OSHA spray booth air flow requirements", *Pollution Engineering*, November 1975, p. 31.

SMACNA, *HVAC Duct System Design*, First Edition, Vienna (Virginia), 1977.

SMACNA, *HVAC Duct System Design: Tables and Charts*, First Edition, Vienna (Virginia), 1977.

SMACNA, *Manual for the Balancing and Adjustment of Air Distribution Systems*, First Edition, Vienna (Virginia), 1967.

Sorg, G.R., "Fan acoustic basics: Part 1", *Pollution Engineering*, March 1983, p. 34.

Sorg, G.R., "Fan acoustic basics: Part 2", *Pollution Engineering*, April 1983, p. 30.

Staheli, A.H., "Control methods for reducing chlorinated hydrocarbon emissions from vapor degreasers", *ASME*, Reprint No. 72-PEM-7, New York, 1972.

Thompson, J.E., and C.J. Trickler, "Fans and fan systems", *Chemical Engineering*, March 21, 1983, p. 48.

Tracy, W.E., *Fan Connections*, Reprint No. 5100, Westinghouse Electric Corporation, Pittsburgh, 1955.

Trickler, C.J., *Effect of System Design on the Fan*, Engineering Letter No. E-4, New York Blower, Chicago.

Trickler, C.J., *Field Testing of Fan Systems*, Engineering Letter No. E-3, New York Blower, Chicago.

Trickler, C.J., *Fundamental Characteristics of Centrifugal Fans*, Engineering Letter No. E-1, New York Blower, Chicago.

Trickler, C.J., *How to Get More Air from Your Fan*, Engineering Letter No. E-5, New York Blower, Chicago.

Trickler, C.J., *How to Select Centrifugal Fans for Quiet Operation*, Engineering Letter No. E-13, New York Blower, Chicago.

USEPA, *Determination of Stack Gas Velocity and Volumetric Flow Rate*, 40CFR60, Appendix A, Method 2.

USEPA, *Plant Inspection Workshop Handout: Design, Operation and Maintenance of Fans*, July 1979.

USEPA, *Sample and Velocity Traverses for Stationary Sources*, 40CFR60, Appendix A, Method 1.

Wright, D.K., "A new friction chart for round ducts", *ASHVE Transactions*, ***51***, 312 (1945).

Appendix C

Psychrometric Charts

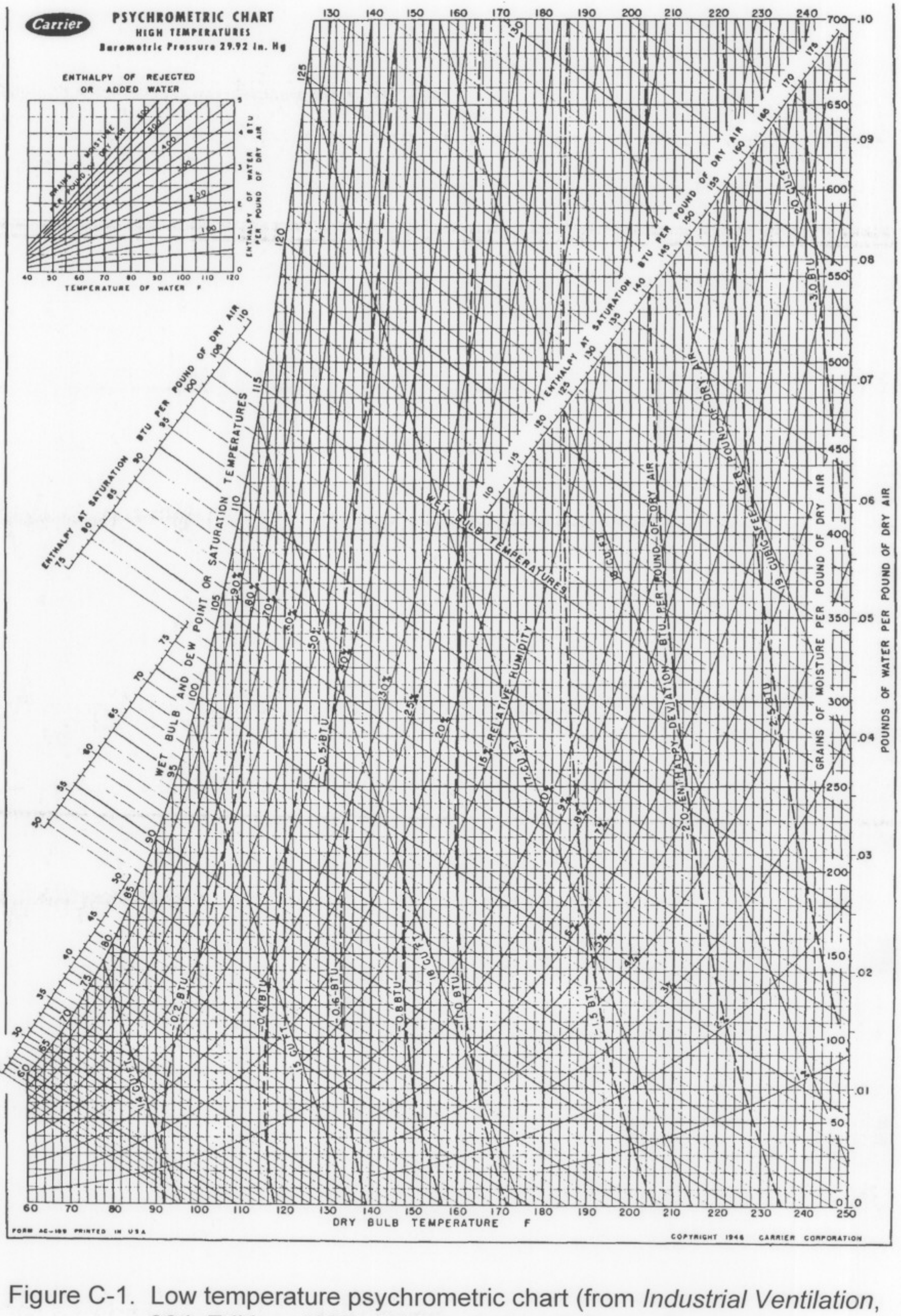


Figure C-1. Low temperature psychrometric chart (from *Industrial Ventilation*, 20th Edition, 1988, with permission of Carrier Corporation)

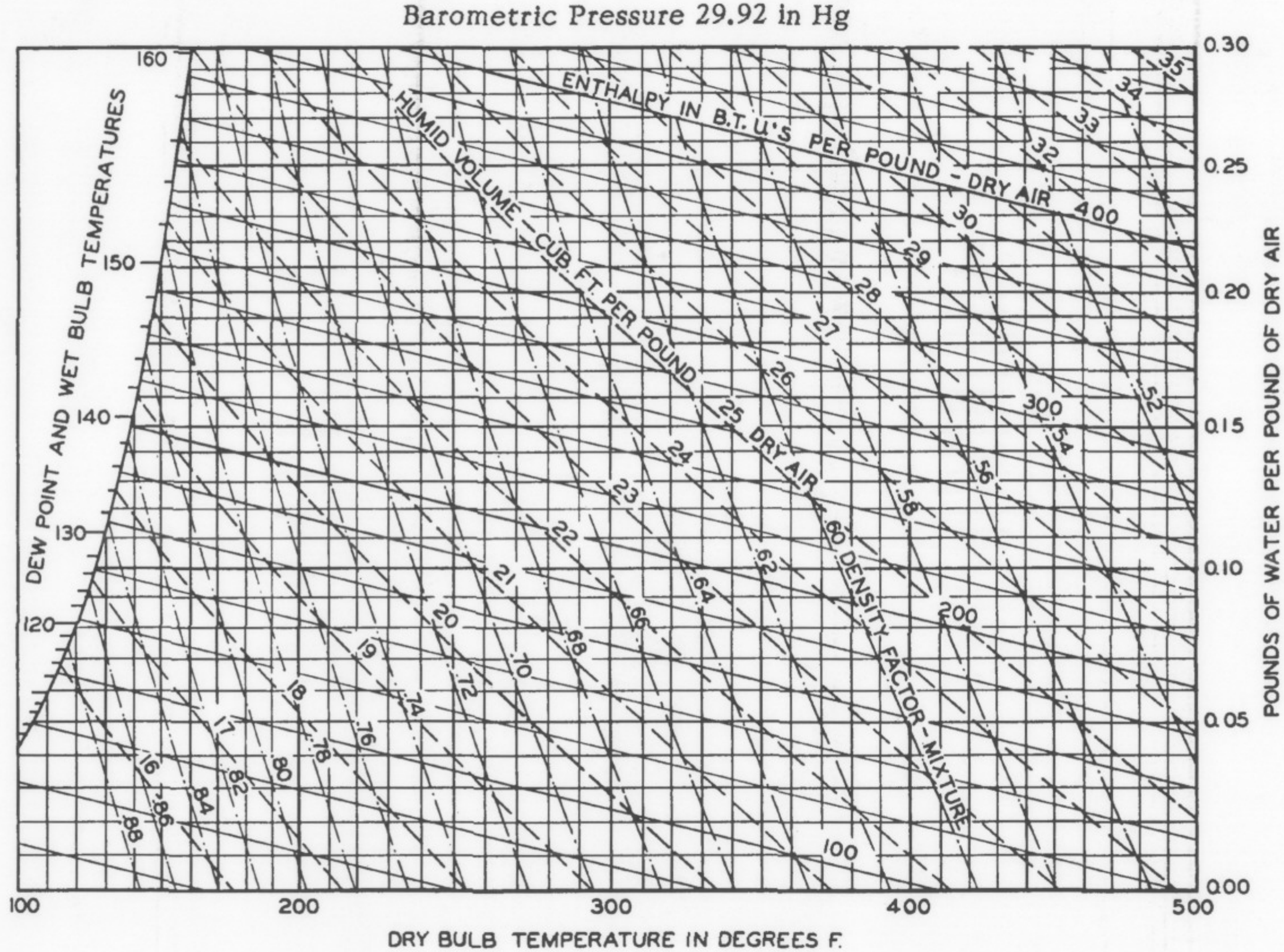


Figure C-2. Moderature temperature psychrometric chart (from *Industrial Ventilation*, 20th Edition, 1988, with permission of American Air Filter Co., Inc.)

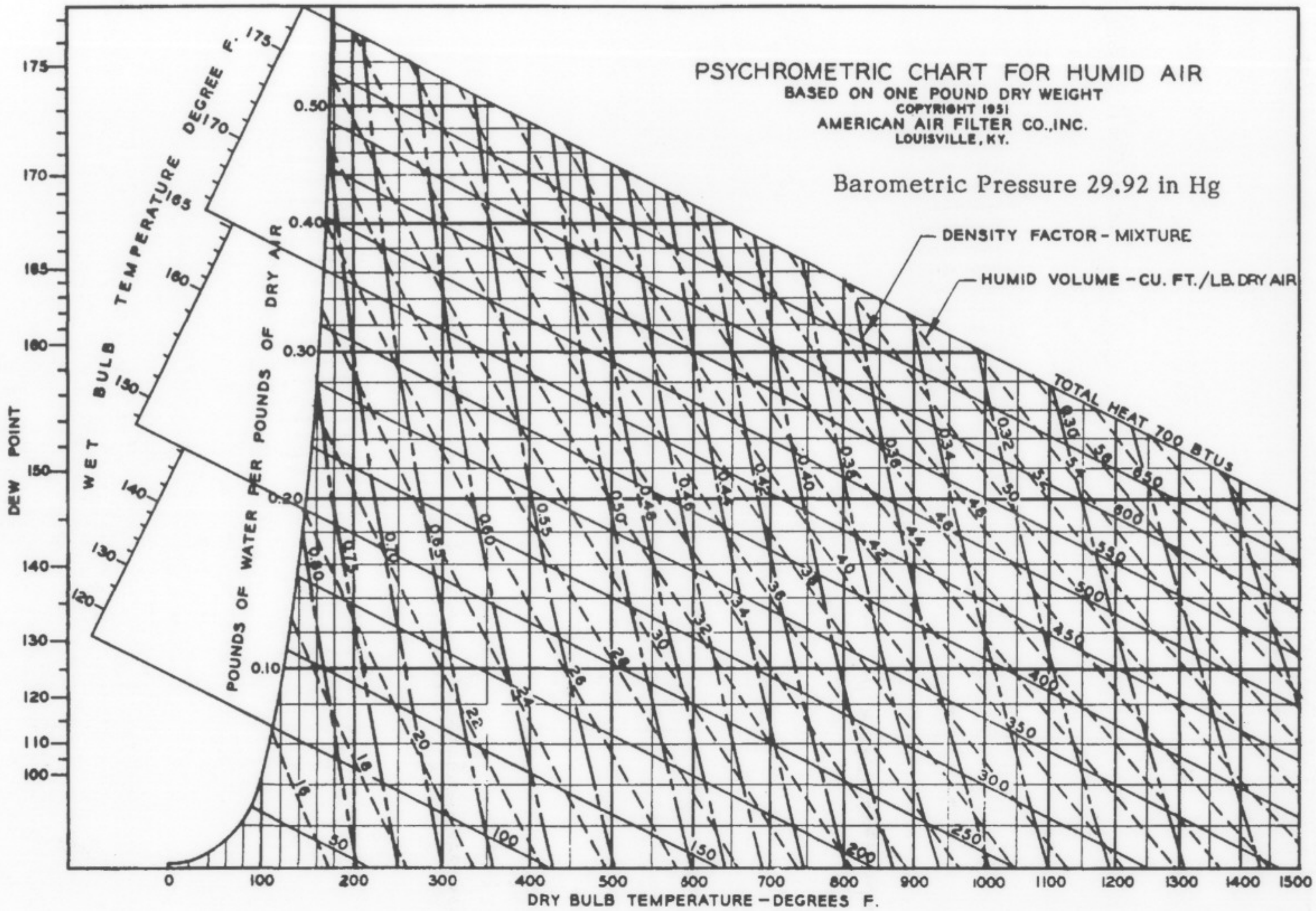


Figure C-3. High temperature psychrometric chart (from *Industrial Ventilation*, 20th Edition, 1988, with permission of American Air Filter Co., Inc.)