INSPECTION OF
INDUSTRIAL VENTILATION SYSTEMS
STUDENT MANUAL

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LESSON 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 firm understanding of the basic information that affects the behavior of air streams. The purpose of this lesson, divided into two sections, is to give you that information. Section I defines the various parameters that are important in ventilation system evaluation and indicates techniques and information sources that may be used in their determination. Section II 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.

I. 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} $lb_m/ft\text{-}sec$. This is equivalent to dry air at a temperature of $70°F$ and a pressure of 29.92 in. Hg.

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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:

Component	Volume Fraction		Molecular Weight		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

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The apparent molecular weight of dry air with this composition is then 28.966 lb/lb-mole.

Suppose the compositional information were 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 follows:

Component	Weight Fraction		Molecular Weight		lb/lb-mole
N_2	0.7552	x	28.016	=	0.02696
O_2	0.2315	x	32.000	=	0.00723
Ar	0.0128	x	39.944	=	0.00032
CO_2	0.0005	x	44.010	=	0.00001
	1.0000				0.03452

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.

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:

$$MW_{wet\ air} = (1 - C_{water})(MW_{dry\ air}) + (C_{water})(MW_{water}) \quad (Eqn.\ 1)$$

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   1-6
related by the ideal or perfect gas law:

$$PV = nRT \quad (Eqn.\ 2)$$

where P = absolute pressure (lbf/ft^2)
V = gas volume (ft^3)
n = number of moles (lb-moles)
R = constant (1544.58 $ft\text{-}lb_f/lb\text{-}mole\text{-}°R$)
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:

Slides

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

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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:

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \qquad \text{(Eqn. 3)}$$

or

$$V_1 = V_2 \left[\frac{P_2}{P_1}\right]\left[\frac{T_1}{T_2}\right] \qquad \text{(Eqn. 4)}$$

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:

$$V/n = RT/P = \frac{\left[\dfrac{10.73 \text{ psia-ft}^3}{\text{lb-mole} - °R}\right] (530 °R)}{(14.7 \text{ psia})}$$

$$= 387 \text{ ft}^3/\text{lb-mole}$$

Density and specific volume   1-8

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

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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:

$$\rho = \frac{m}{V} \qquad \text{(Eqn. 5)}$$

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

$$\rho = \left[\frac{MW}{387}\right]\left[\frac{530\ ^\circ R}{T}\right]\left[\frac{P}{29.92}\right] \qquad \text{(Eqn. 6)}$$

where   MW = molecular weight
      T = absolute temperature ($^\circ R$)
      P = absolute pressure (in. Hg)

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

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:

$$P_{air} + P_{water} = P_{total} \qquad \text{(Eqn. 7)}$$

Table 1-1. Properties of Air (Danielson, 1973)

Temp. (°F)	Specific Heat at Constant Pressure (C_p) Btu/lb-°F	Absolute Viscosity (u) $\frac{lb}{hr\text{-}ft}$	Thermal Conductivity (k) $\frac{Btu}{hr\text{-}ft\text{-}°F}$	Prandt Number (Cu/k) dimen--sionless	Density (ρ) lb/ft^3
0	0.240	0.040	0.0124	0.77	0.0863
20	0.240	0.041	0.0128	0.77	0.0827
40	0.240	0.042	0.0132	0.77	0.0794
60	0.240	0.043	0.0136	0.76	0.0763
80	0.240	0.045	0.0140	0.76	0.0734
100	0.240	0.047	0.0145	0.76	0.0708
120	0.240	0.047	0.0149	0.76	0.0684
140	0.240	0.048	0.0153	0.76	0.0662
160	0.240	0.050	0.0158	0.76	0.0639
180	0.240	0.051	0.0162	0.76	0.0619
200	0.240	0.052	0.0166	0.76	0.0601
250	0.241	0.055	0.0174	0.76	0.0538
300	0.241	0.058	0.0182	0.76	0.0521
350	0.241	0.060	0.0191	0.76	0.0489
400	0.241	0.063	0.0200	0.76	0.0460
450	0.242	0.065	0.0207	0.76	0.0435
500	0.242	0.067	0.0214	0.76	0.412
700	0.243	0.076	0.0243	0.76	0.0341
1,000	0.246	0.089	0.0283	0.77	0.0275
1,400	0.251	0.105	0.0328	0.80	0.0212
1,800	0.257	0.120	0.0360	0.85	0.0175

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:

$$\text{Relative saturation} = p_{water}/p_{water\ at\ saturation} \quad \text{(Eqn. 8)}$$

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 as 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:

$$h = C_p(t-t_{ref})$$ (Eqn. 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, although this is highly variable, while 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 over a limited range of temperature are provided in Table 1-1.

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

$$h_{water\ vapor} = h_{water} + \lambda_v$$ (Eqn. 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:

$$h_{mixture} = h_{dry\ air} + \phi(h_{water\ vapor}) \qquad \text{(Eqn. 11)}$$

where ϕ = absolute humidity ($lb_{water}/lb_{dry\ air}$)
$h_{mixture}$ = enthalpy of mixture ($Btu/lb_{dry\ air}$)
$h_{dry\ air}$ = enthalpy of dry air ($Btu/lb_{dry\ air}$)

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:

$$\Delta H = h_2 - h_1$$

$$\Delta H = (C_p)_2(t_2 - t_{ref}) - (C_p)_1(t_1 - t_{ref}) \qquad \text{(Eqn. 12)}$$

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:

$$\Delta H = (C_p)_{avg}(t_2 - t_1) \qquad \text{(Eqn. 13)}$$

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

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Table 1-2. Enthalpies of Various Gases in Btu/Lb.
(Danielson, 1973)

Temp. (°F)	CO_2	N^2	H_2O	O_2	Air
100	5.8	6.4	17.8	8.8	9.6
150	17.6	20.6	40.3	19.8	21.6
200	29.3	34.8	62.7	30.9	33.6
250	40.3	47.7	85.5	42.1	45.7
300	51.3	59.8	108.2	53.4	57.8
350	63.1	73.3	131.3	64.8	70.0
400	74.9	84.9	154.3	76.2	82.1
450	87.0	97.5	177.7	87.2	94.4
500	99.1	110.1	201.0	99.5	106.7
600	124.5	135.6	248.7	123.2	131.6
800	176.8	187.4	346.4	171.7	182.2
1,000	231.9	240.5	447.7	221.6	234.1
1,200	289.0	294.9	552.9	272.7	287.2
1,400	347.6	350.5	661.3	324.6	341.5
1,600	407.8	407.3	774.2	377.3	396.8
1,800	469.1	464.8	889.8	430.4	452.9
2,000	531.4	523.0	1,003.1	484.5	509.5
2,200	594.3	582.0	1,130.3	538.6	567.1
2,400	658.2	642.3	1,256.8	593.5	625.0
2,500	690.2	672.3	1,318.1	621.0	654.3
3,000	852.3	823.8	1,640.2	760.1	802.3

[a] Note: The enthalpies tabulated for H_2O represent a gaseous system, and the enthalpies do <u>not</u> include the latent heat of vaporization. It is recommended that the latent heat of vaporization at 60 °F (1,059.9 Btu/lb) be used where necessary.

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.

Psychometric chart
     Many of the values for air-water vapor mixtures
that are used in ventilation calculations are avail-
able in a graphical representation known as a psycho-
metric chart. Two of these charts, covering a range of
temperatures, have been included in the Appendix of this
manual. 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. Psychometric Chart
(Based on: Morse, 1965)
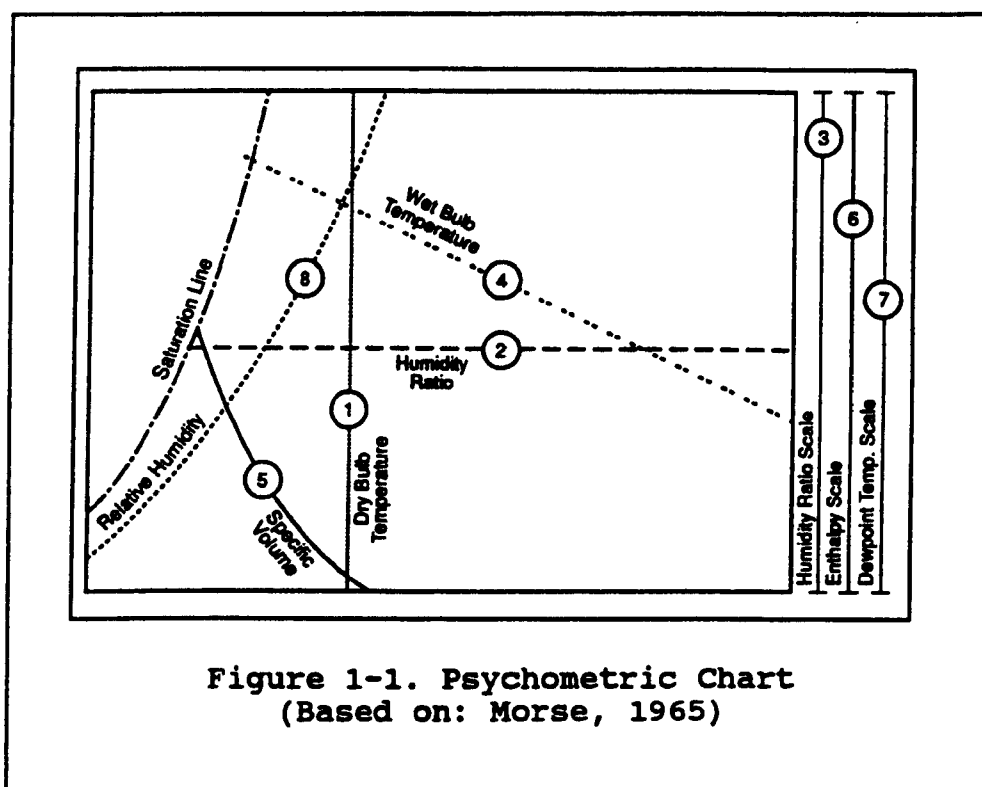
   1. *Dry-bulb temperature:* The temperature of air
      read on a standard thermometer is shown on
      the chart by straight vertical lines. The
      scale is at the bottom of the chart.

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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 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

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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 psychometric 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 psychometric chart is the density of the air-water vapor mixture. However, this can be determined from values obtained from the chart, as follows:

$$\rho_{mixture} = (1 + \phi)/U \qquad\qquad (Eqn\ 14)$$

where ϕ = absolute humidity $(lb/lb_{dry\ air})$
U = specific volume $(ft^3/lb_{dry\ air})$

II. PRINCIPLES OF FLUID FLOW

Continuity

Consider the flow of fluid through a tube as shown below:

Figure 1-2. Flow Diagram

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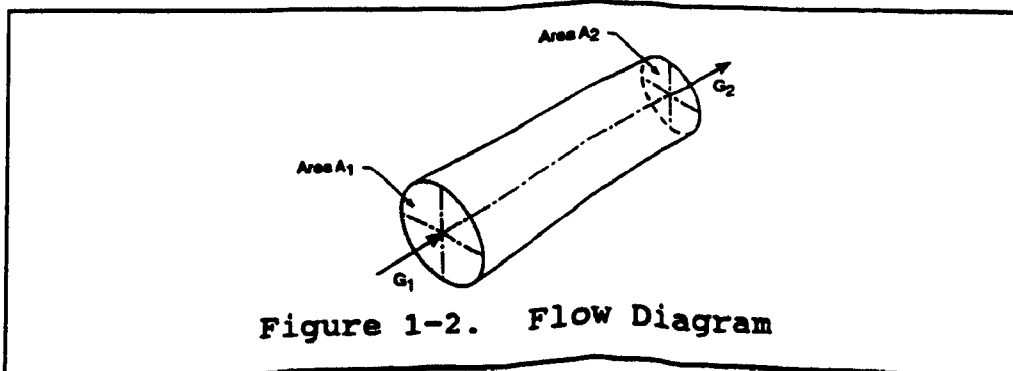
The mass rate of flow through the tube (e.g., lb/min) is given by $G = \rho VA$ and the volume rate of flow (e.g., ft^3/min) is given by $Q = G/\rho = VA$. Here, ρ is the fluid density, V is the fluid velocity and A is the tube cross-sectional area. If there is no accumulation or removal of material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

$$G_1 = G_2 \qquad \text{(Eqn. 15)}$$

or

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 \qquad \text{(Eqn. 16)}$$

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:

$$V_1 A_1 = V_2 A_2 \qquad \text{(Eqn. 17)}$$

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 and pressure 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:

$$V^2/2 + P/\rho + gz = \text{constant} \qquad \text{(Eqn 18)}$$

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

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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:

$$V^2/2g + P/\rho g + z = \text{constant} \qquad \text{(Eqn. 19)}$$

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:

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

$$P/\rho g = \text{pressure head}$$

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

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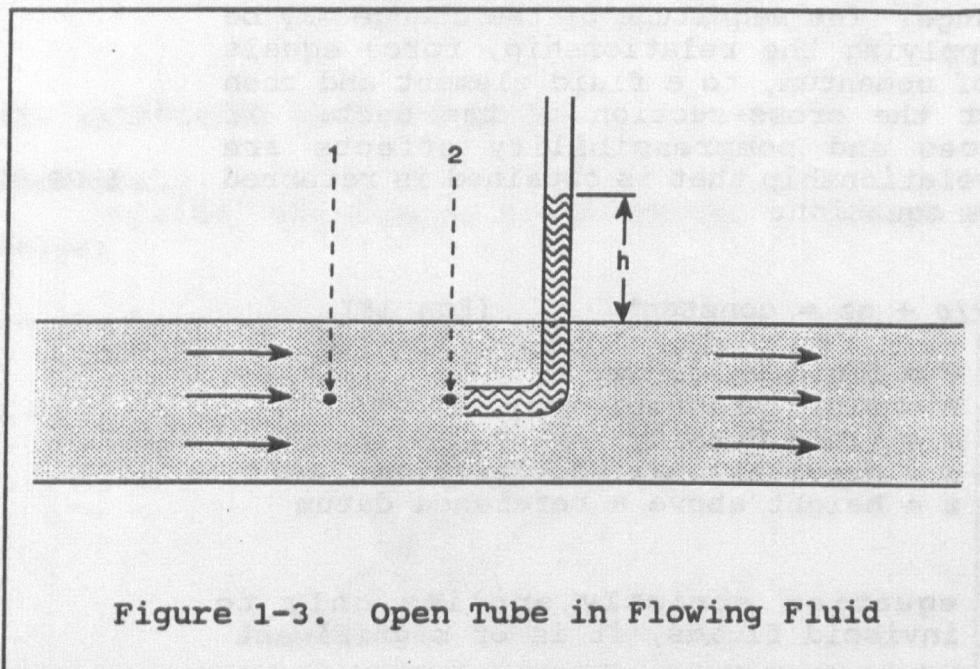
Since the terms in Bernoulli's equation sum to a constant, we may write:

$$(V_1)^2/2g + P_1/\rho g + z_1 = (V_2)^2/2g + P_2/\rho g + z_2 \quad \text{(Eqn. 20)}$$

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:

$$(V_1)^2/2g + P_1/\rho g = P_2/\rho g \quad \text{(Eqn. 21)}$$

or

$(V_1)^2/2g$ = velocity head

$P_1/\rho g$ = pressure head

$P_2/\rho g$ = total head

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

$$VP + SP = TP \quad \text{(Eqn. 22)}$$

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 perpendicular 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.

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Figure 1-4. Pressure Relationships
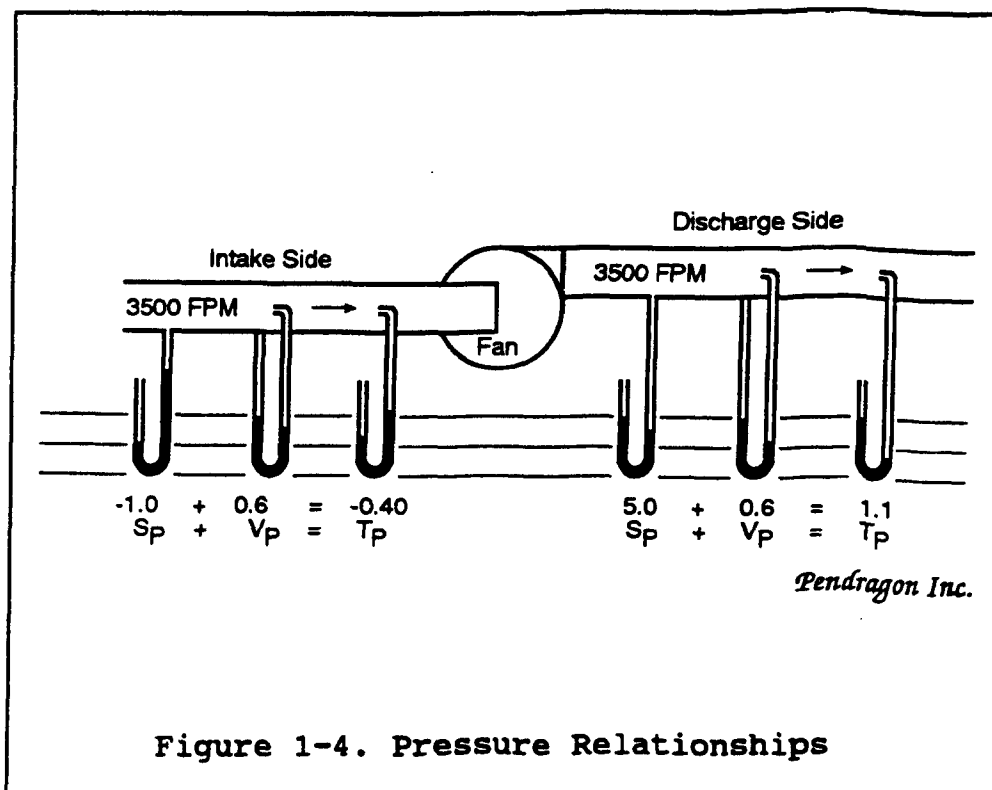
In the above development the velocity pressure, VP, was given by the term, $V^2/2g$, in units of length of fluid. In measuring velocity pressure, we typically use the pressure of the flowing fluid to displace fluid in a manometer, which we read in inches of water column. Converting the units of the velocity pressure term so that it has unit of inches of water gives:

$$VP = [(V/60)^2/2g](\rho_a/\rho_w)12 \qquad (Eqn. 23)$$

where V = air velocity (ft/min)
  ρ_a = air density (lb/ft³)
  ρ_w = water density (lb/ft³)
  g = acceleration of gravity (ft/sec²)

Substituting a water density at 70 °F of 62.302 lb/ft³ and a gravity acceleration at sea level of 32.174 ft/sec² gives: 1-27

$$VP = \rho_a(V/1096.7)^2 \qquad (Eqn. 24)$$

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

$$V = 1096.7(VP/\rho_a)^{0.5} \qquad\qquad \text{(Eqn. 25)}$$

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

$$V = 4005(VP)^{0.5} \qquad\qquad \text{(Eqn. 26)}$$

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LESSON 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:

$$Pt_{total} = Pt_{hood} + (1 - Pt_{hood})Pt_{collector} \qquad \text{(Eqn. 1)}$$

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 lesson, 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.

I. 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.

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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.
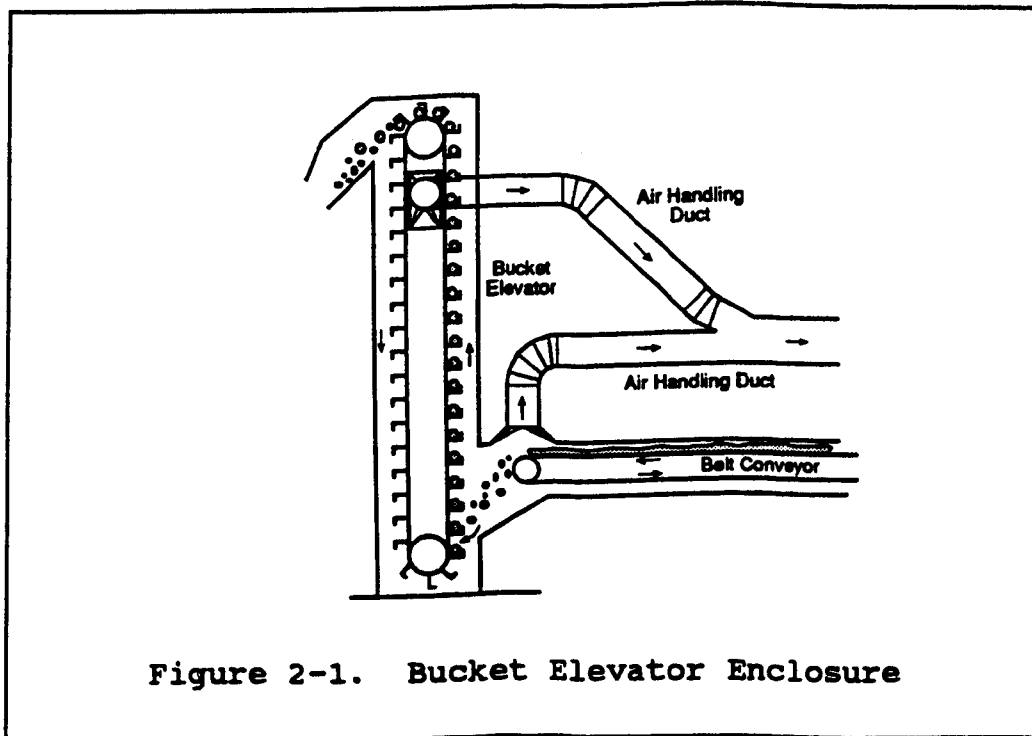
Figure 2-1. Bucket Elevator Enclosure

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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 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

Figure 2-2. Ladle Hood (Kemner et.al. 1984)
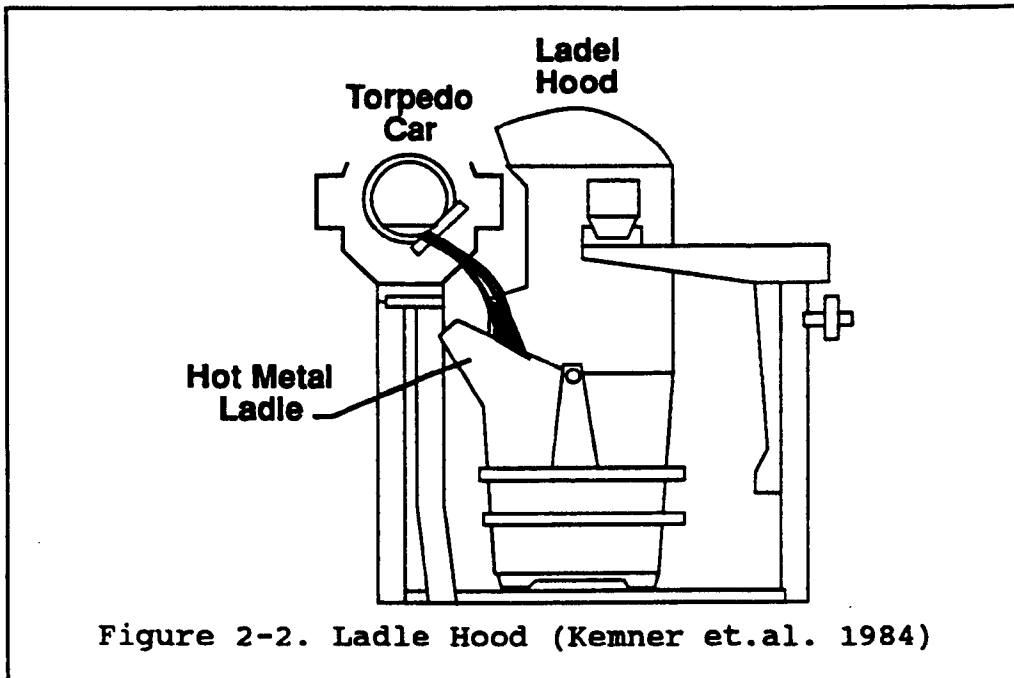
capability that must be provided by the air stream is reduced.

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.

Figure 2-3. Grinding Wheel Hood
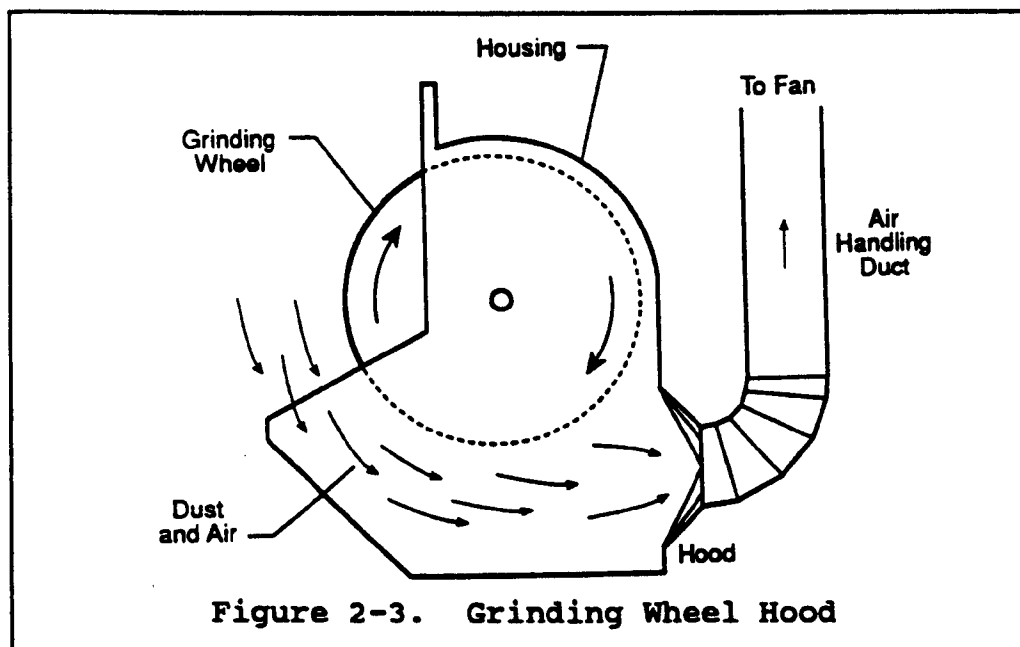
2-9

The hood also extends around the top and sides of the wheel to provide for enclosure of any contaminants that follow the wheel motion.

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
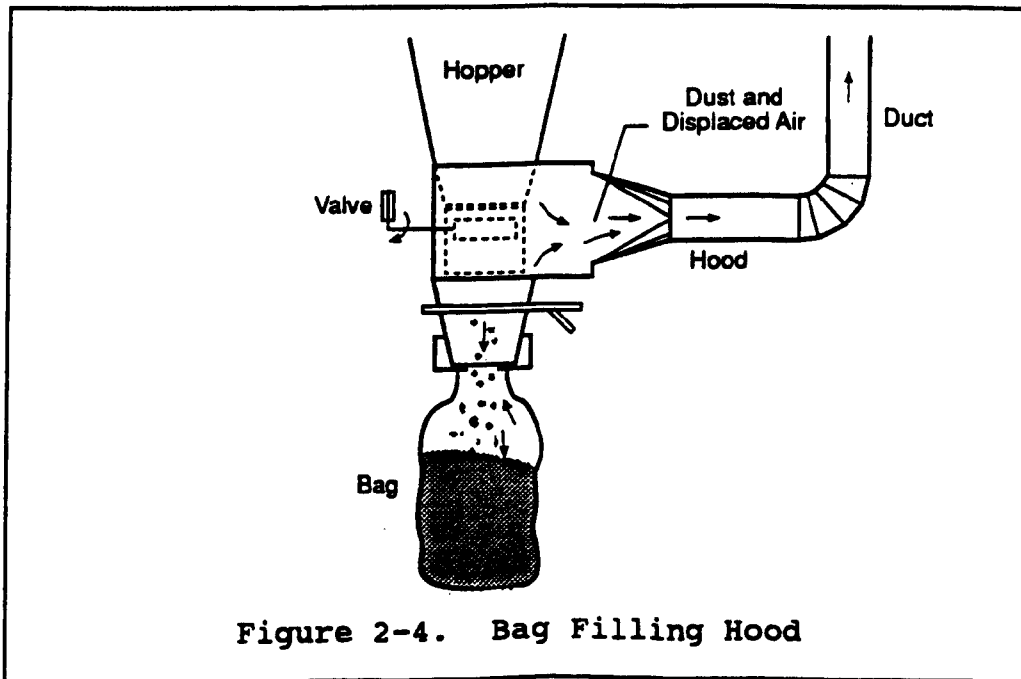
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A hood that is mounted an extended distance away from the contaminant source is referred to as an exterior hood. The principal example of this type hood is the overhead canopy typically employed with hot sources. Because of the limited capability of 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)
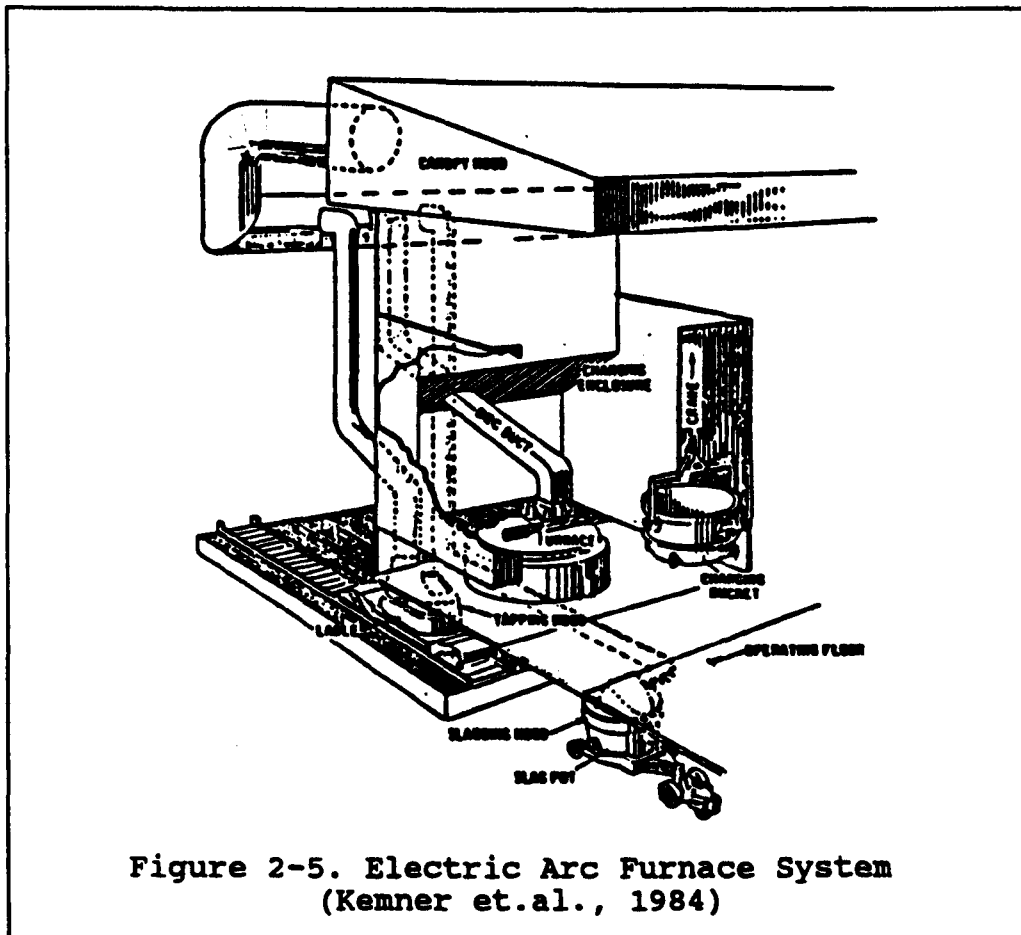
A variation of the exterior hood is the push-pull
system shown in Figures 2-6 and 2-7. Here, a control-
led jet of air is directed across the contaminant source
and in the direction of the exterior hood. The exterior
hood primarily serves to receive the air jet and the
contaminants carried with it. The 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.
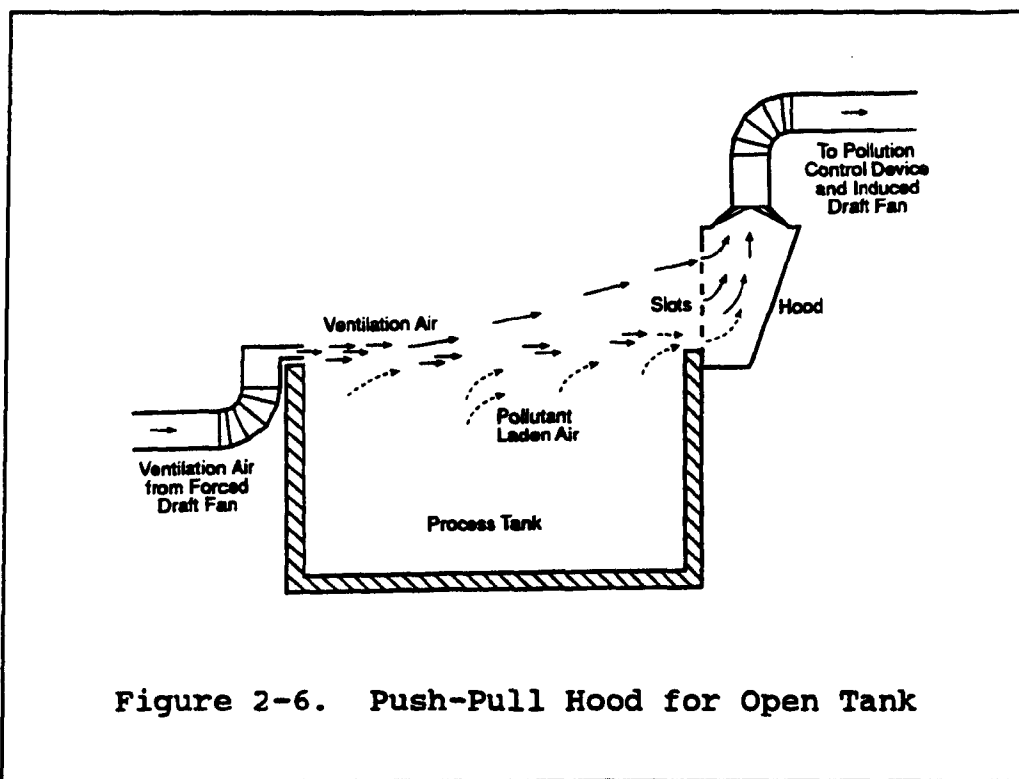
Figure 2-6. Push-Pull Hood for Open Tank

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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.

Figure 2-7. Push-Pull Hood for Copper Converter
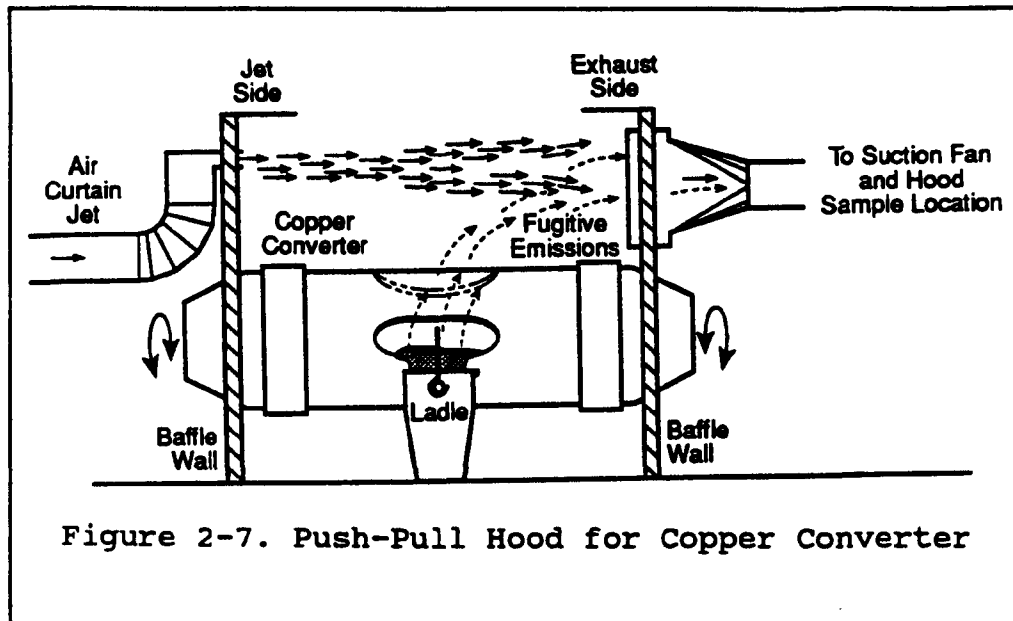
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One important reference that provides information on the design, air flow requirements and resistance characteristics of a large number of industrial hoods is, **Industrial Ventilation**, A Manual of Recommended Practice, published by the American Conference of Governmental Industrial Hygienists.

II. 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:

1. Whenever possible, use an enclosure hood.

2. 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.

3. 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 associated with the workspace ventilation system. These 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 from heat generating equipment and processes. Even low heat releases, such as those from an electric motor, may be enough to disturb hood performance.

- 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

appropriate capture velocities in several situations is provided in Table 2-1. Values at the low end of the 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 the 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

Type of Material Release	Capture Velocity (feet/Minute)
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.
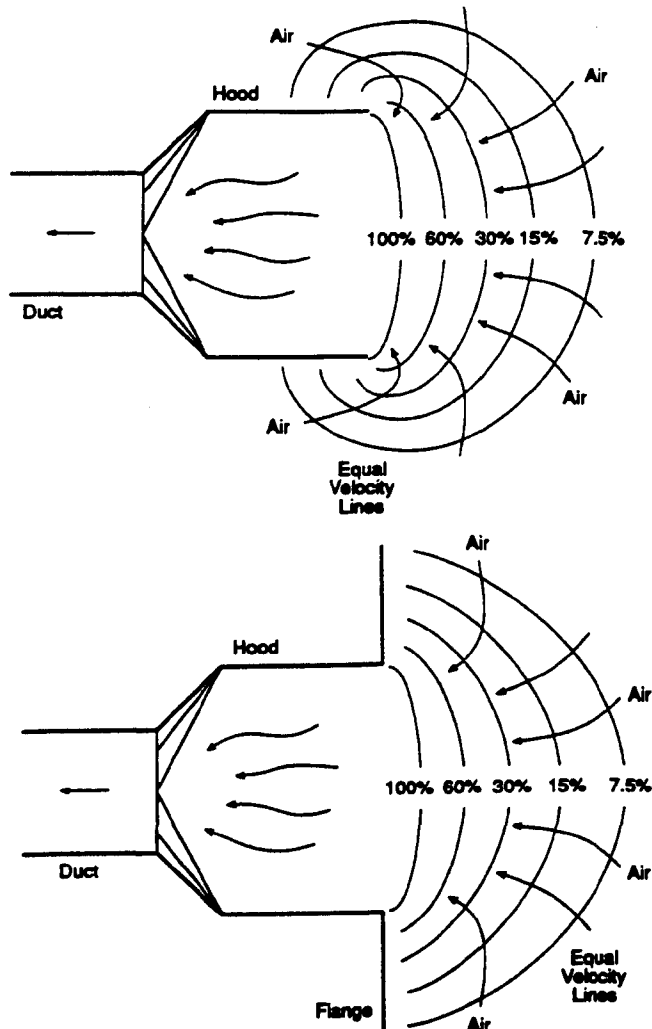
Figure 2-8.   Hood Velocity Contours
(Percent of Hood Capture Velocity)

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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
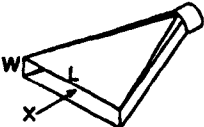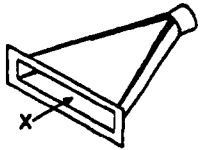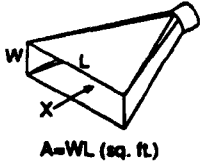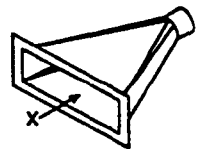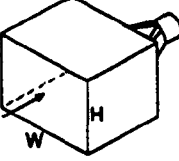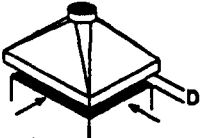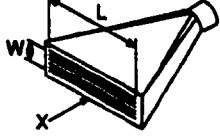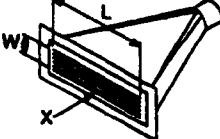
Hood Type	Description	Aspect Ratio	Air Flow
	Slot	0.2 or Less	$Q = 3.7 LVX$
	Flanged Slot	0.2 or Less	$Q = 2.6 LVX$
A=WL (sq. ft.)	Plain Opening	0.2 or Greater and Round	$Q = V (10X^2 + A)$
	Flanged Opening	0.2 or Greater and Round	$Q = 0.75V (10X^2 + A)$
	Booth	To Suit Work	$Q = VA = VWH$
	Canopy	To Suit Work	$Q = 1.4 PVD$ See VS-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 (ACGIH, 1988)

velocity once the hood flowrate is known. A method for estimating hood flowrate will be present later in this lesson, and a method for measuring flowrate will be discussed in Lesson 6.

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 following hood system:
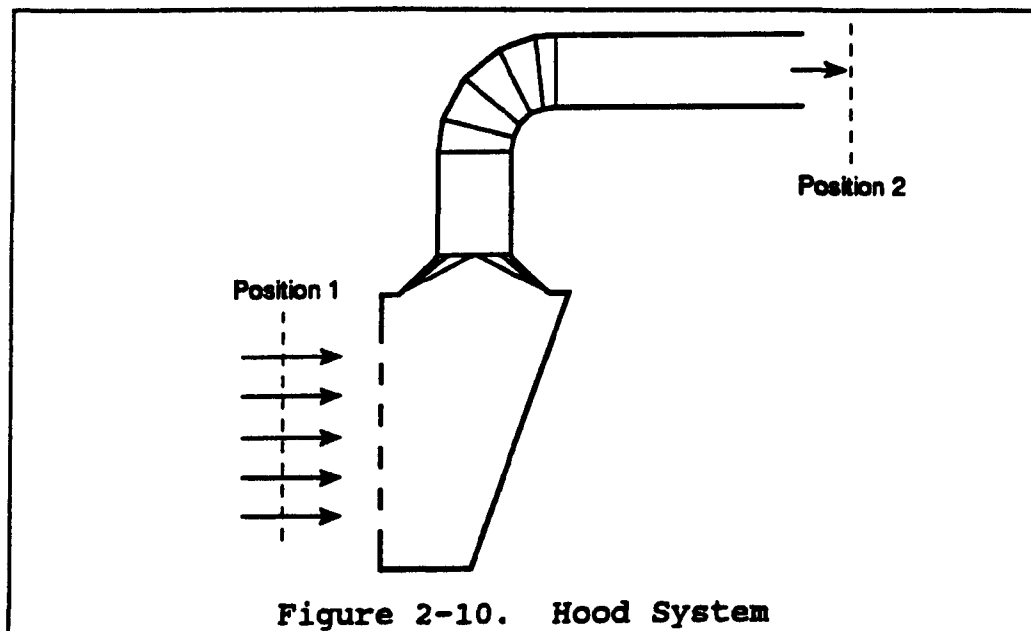
Figure 2-10. Hood System

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

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

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

$$0 = SP_2 + VP_2 \qquad \text{(Eqn. 3)}$$

or

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

In reality, as air enters a hood or duct a "vena
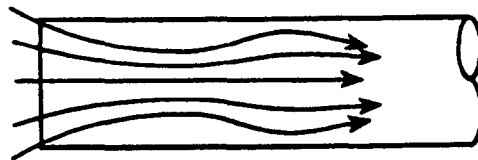contracta" forms, as illustrated in Figure 2-11.

Figure 2-11. Vena Contracta

Following the continuity relationship, this reduction
in effective cross-section within the vena contracta
causes an increase in velocity. Then, as the con-
tracta 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:

$$SP_h = -SP_2 = VP_2 + h_e \qquad \text{(Eqn. 4)}$$

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:

$$h_e = F_h VP \qquad \text{(Eqn. 5)}$$

where F_h = hood loss factor

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

$$C_e = (VP/SP_h)^{0.5} \qquad \text{(Eqn. 6)}$$

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

(Equation Set 2-7)

$$C_e = [VP/(VP + h_e)]^{0.5}$$

$$C_e^2 = VP/(VP + h_e)$$

$$h_e = VP/C_e^2 - VP = [(1 - C_e^2)/C_e^2]VP$$

But $h_e = F_h VP$; therefore:

$$F_h = (1 - C_e^2)/C_e^2 \qquad \text{(Eqn. 8)}$$

and

$$C_e = [1/(1 + F_h)]^{0.5} \qquad \text{(Eqn. 9)}$$

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

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

$$Q = VA = 1096.7A(VP/\rho)^{0.5} \qquad \text{(Eqn. 10)}$$

θ	Round	Rectangular
15°	0.15 VP	0.25VP
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
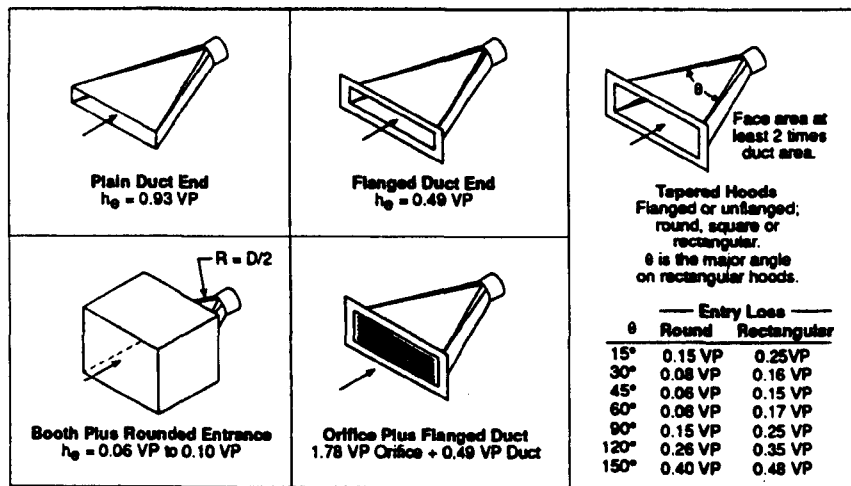
Figure 2-12. Hood Entry Loss Factors

Then, recalling that $C_e = (VP/SP_h)^{0.5}$, substitution gives:

$$Q = 1096.7AC_e(SP_h/\rho)^{0.5}$$ (Eqn. 11)

To use this relationship to estimate hood flowrate, a static pressure measurement would be made in the duct connected to the hood and downstream of the vena contracta, usually about 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 2-12. Substituting these values, along with the duct cross-sectional area, into the above relationship would yield the flowrate 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 Lesson 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

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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.

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LESSON 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 Lesson 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 lesson, 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.

I. 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:

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$$TP_1 = TP_2 \qquad \text{(Eqn. 1)}$$

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

3-3

Figure 3-1. Example Duct Segment
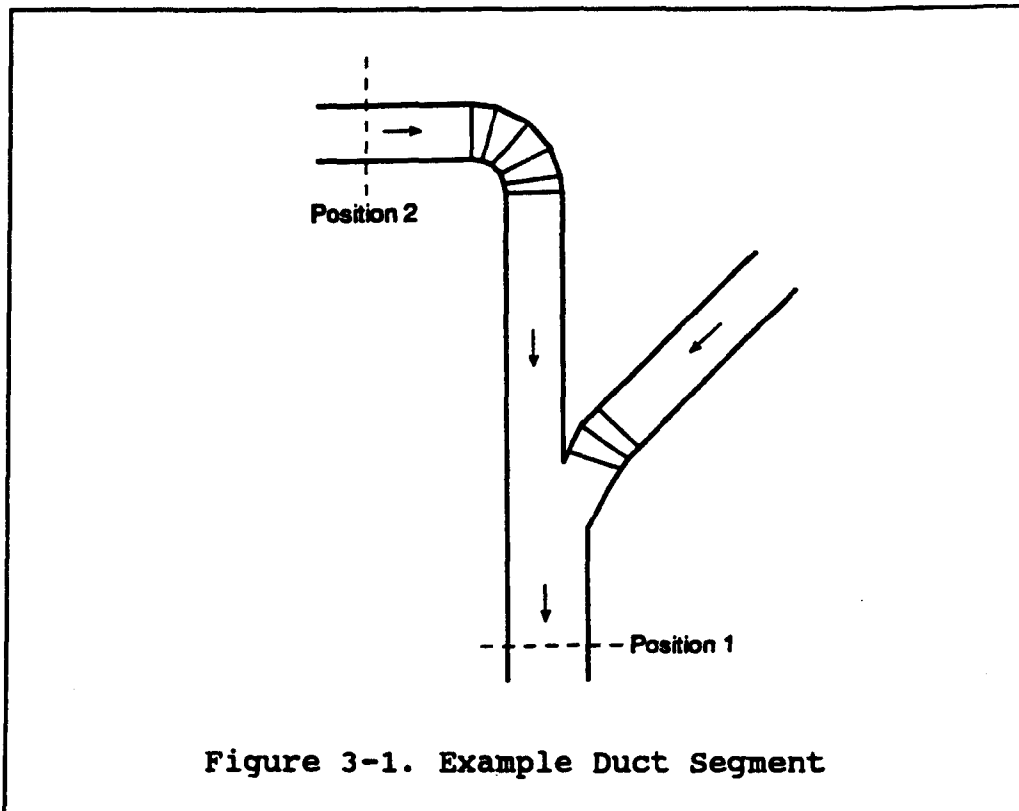
gas stream accelerates or decelerates, pressure losses between the two points occur. Thus:

$$TP_1 = TP_2 + h_L \qquad \text{(Eqn. 2)}$$

or

$$SP_1 + VP_1 = SP_2 + VP_2 + hL \qquad \text{(Eqn. 3)}$$

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

$$SP_1 = SP_2 + h_L \qquad \text{(Eqn. 4)}$$

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:

$$h_{l1} = H_f L \cdot VP \qquad \text{(Eqn. 5)}$$

where H_f = velocity pressure loss per foot of
        duct
    L = length of duct, feet

The straight-duct loss factor can be determined from the following empirical equation or from Figure 3-1.

$$H_f = 0.0307 V^{0.533} / Q^{0.612}$$

$$= 0.4937 / Q^{0.079} D^{1.066} \qquad \text{(Eqn. 6)}$$

where   V = air velocity in feet/minute
        Q = volumetric flowrate in cubic
            feet/minute
        D = duct diameter in inches

In Figure 3-2, a point is located using any two of flowrate, velocity or duct diameter and the corresponding loss factor read on the vertical scale. For rectangular ducts, an equivalent diameter is determined from:

$$D_{eq} = 1.3 (A \times B)^{0.625} / (A + B)^{0.25} \quad \text{(Eqn. 7)}$$

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

Using the equivalent diameter an equivalent velocity is calculated and the loss factor chart entered using the equivalent values.

Fitting losses in the velocity pressure method are expressed as:

$$h_{L2} = F \cdot VP \qquad \text{(Eqn. 8)}$$

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
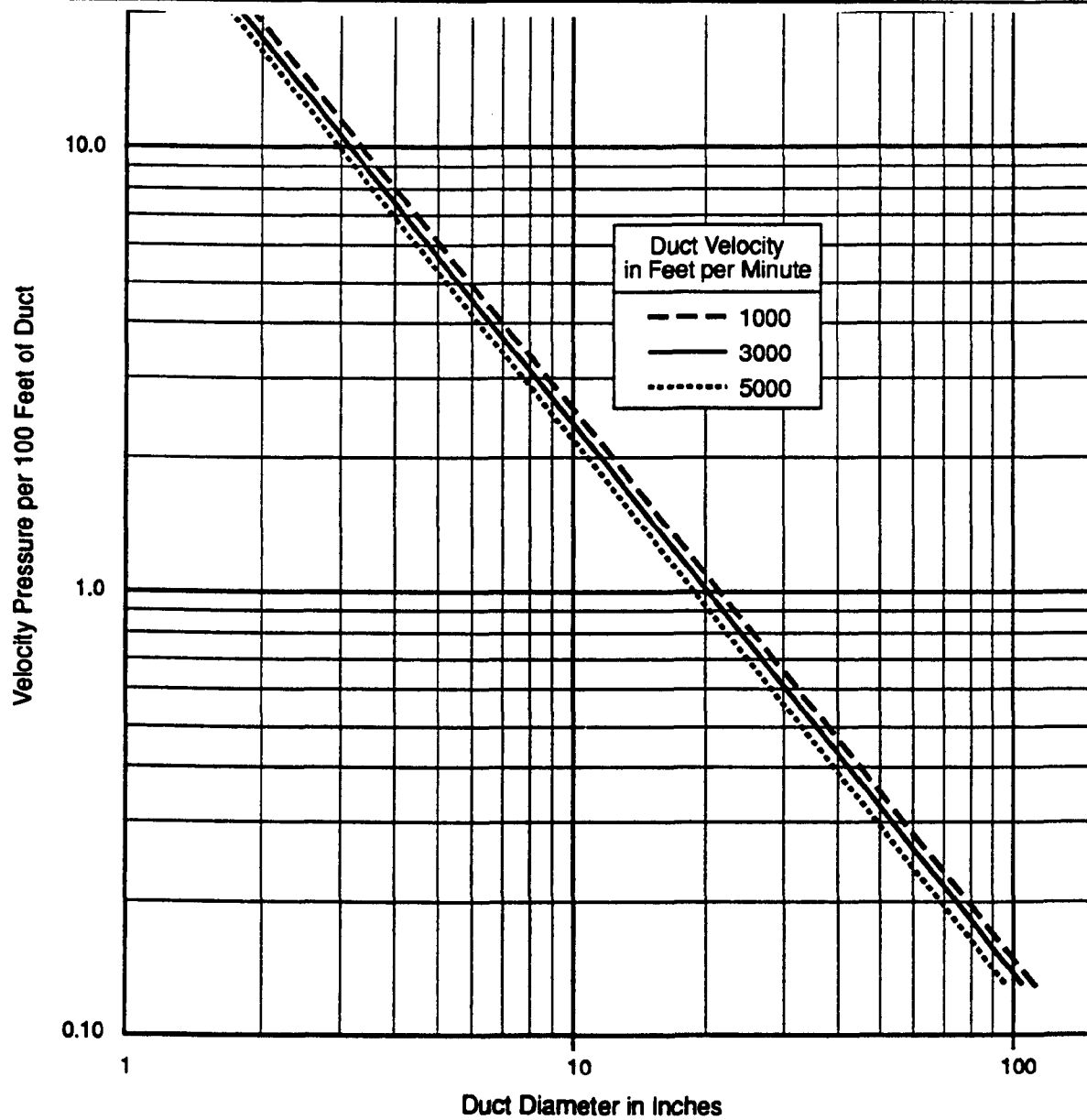
Figure 3-2. Straight Duct Frictional Losses

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 3-2. 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
at Various Aspect Ratios (W/D)

R/D	0.25	0.50	1.0	2.0	3.0	4.0
0	1.5	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 flowrate
     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 lesson 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 Lesson 1 the relationship for the hood
entry loss coefficient, C_e:

$$C_e = (VP/SP_h)^{0.5} \qquad \text{(Eqn. 9)}$$

          or

$$SP_h = VP/C_e^2 \qquad \text{(Eqn. 10)}$$

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:

$$SP_h = SP - H_f L \cdot VP - SF \cdot VP \qquad \text{(Eqn. 11)}$$

Equating the two relationships for hood static pressure gives:

$$VP/C_e^2 = SP_{meas} - H_fL \cdot VP - SF \cdot VP \qquad \text{(Eqn. 12)}$$

or

$$VP = SP_{meas}/(1/C_e^2 + H_fL + SF) \qquad \text{(Eqn. 13)}$$

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 flowrate can be calculated from:

$$Q = VA = 1096.7A(VP/\rho_a)^{0.5} \qquad \text{(Eqn. 14)}$$

or, if standard air is involved:

$$Q = 4005A(VP)^{0.5} \qquad \text{(Eqn. 15)}$$

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.

3-13

3-14

 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.

3-15

 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

Table 3-3. Range of Design Transport Velocities

Contaminant	Design Velocity (feet/Minute)
Vapors, gases, and smoke	Any (usually 1000 to 2000)
Fumes	1400 - 2000
Very fine, light dust	2000 - 2500
Dry dust and powders	2500 - 3500
Average industrial dust	3500 - 4000
Heavy dusts	4000 - 4500
Heavy or moist	4500 and up

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. 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 following,
two-hood duct system:

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

Figure 3-3. Example Duct System
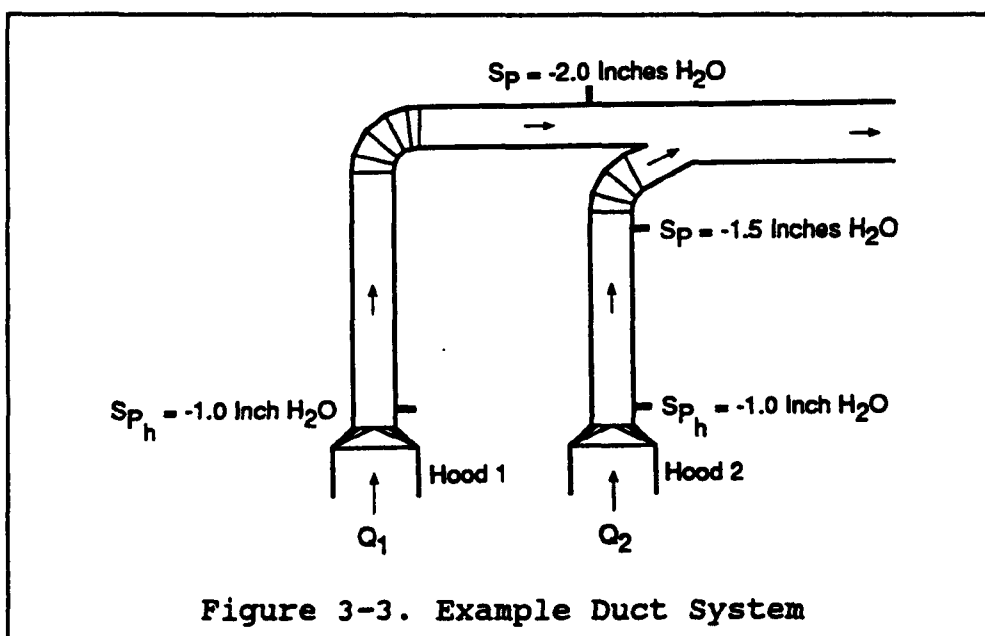
in. H_2O. Since it is not physically possible to
maintain two static pressures at one location, this
condition cannot prevail. If nothing is done, the
system will adjust itself by reducing the flowrate from
Source 1 and increasing the flowrate from Source 2 until
the same static pressure requirement results. However,
the reduced flowrate 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 flowrates.

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 flowrates 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.

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LESSON 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 lesson 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.

I. 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:

Slides

4-3

$$m_1 h_1 + m_2 h_2 = m_3 h_3 \qquad \text{(Eqn. 1)}$$

where

m_1 = mass flowrate of hot gases
h_1 = enthalpy of hot gases
m_2 = mass flowrate of dilution air
h_2 = enthalpy of dilution air
m_3 = mass flowrate of gas mixture
       $(m_1 + m_2)$
h_3 = enthalpy of gas mixture

Recalling from Lesson 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

Figure 4-1. Canopy Hood System
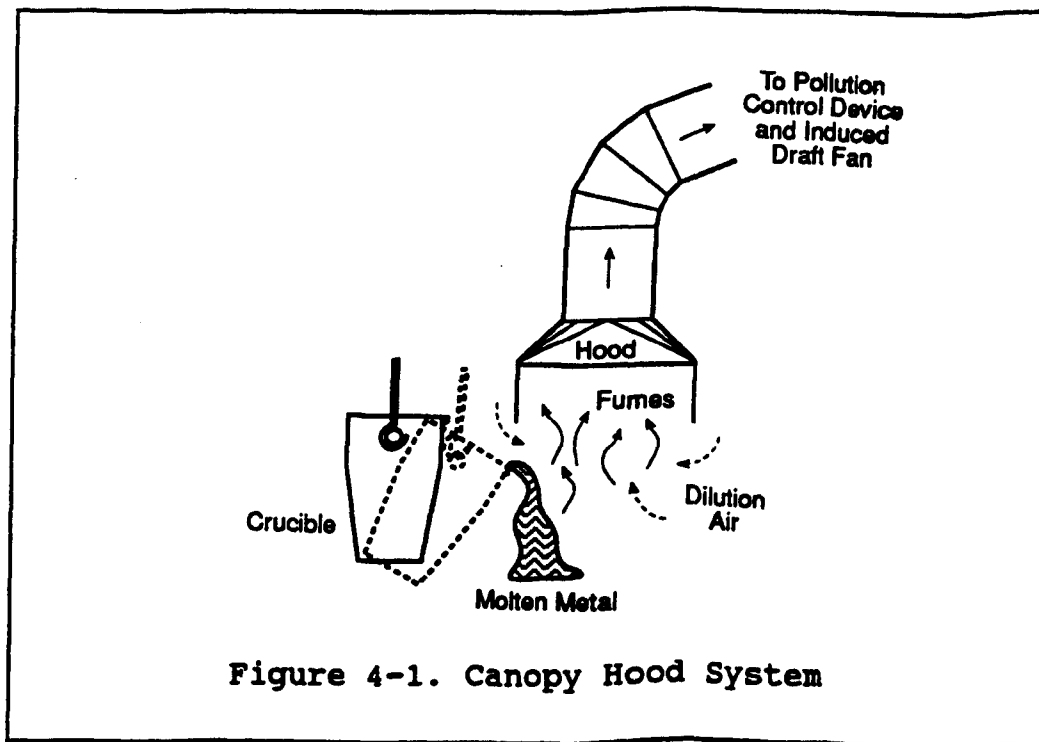
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also be appropriate when enclosure or receiving hoods are used, if the hot gas stream is small. Here, the dilution air could be introduced through a branch into the hot duct or by a combination of capture and introduced air. However, when the hot gas stream is large, the quantity of dilution air becomes excessively large, making the cost of the downstream exhaust system and control device uneconomical.

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 flowrate and temperature of the mixed stream and the temperature of the hot gas stream and the dilution air (usually ambient air). Next, the volumetric flowrate of the mixture would be converted to a mass flowrate 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 flowrate of the hot gas and the dilution air, as follows:

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

or

$$m_{hot} = m_{mix}(h_{mix} - h_{dilution})/(h_{hot} - h_{dilution}) \quad (4-3)$$

and

$$m_{dilution} = m_{mix} - m_{hot} \quad (4-4)$$

The mass flowrates would then be converted to volumetric flowrates 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 flowrate in all three streams, if that information is needed. The method for measuring volumetric flowrate will be discussed in Lesson 6.

II. 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 4-6 performance may again be developed through a heat balance on the evaporation system:

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

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 4-7 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 Lesson 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 flowrate 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 flowrate, 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 flowrate 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.

III. 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.

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

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

where U = overall heat transfer coefficient
   A = heat transfer area
   DT_m = log-mean temperature difference
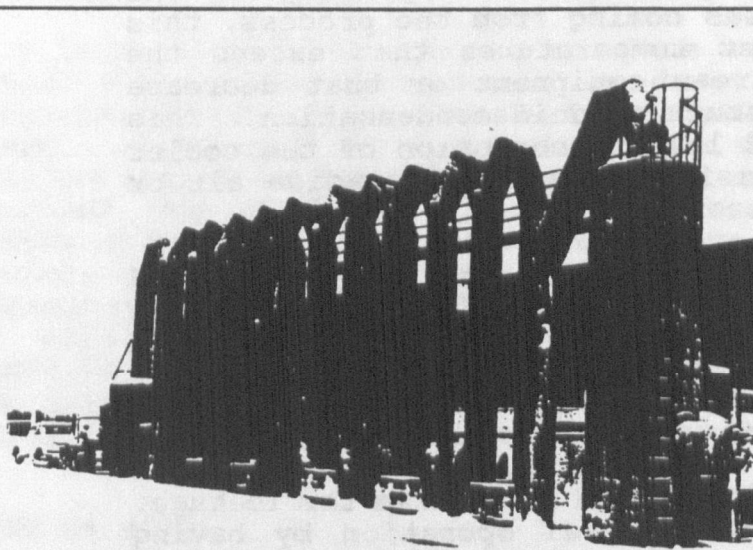
Figure 4-2. Convention and Radiation Cooler
(Danielson, 1973)

In the English system, the overall heat transfer coefficient, U, has units of $Btu/hr\text{-}ft^2\text{-}°F$, 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, DT_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.

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LESSON 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 lesson 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 flowrate 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.

I. 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.

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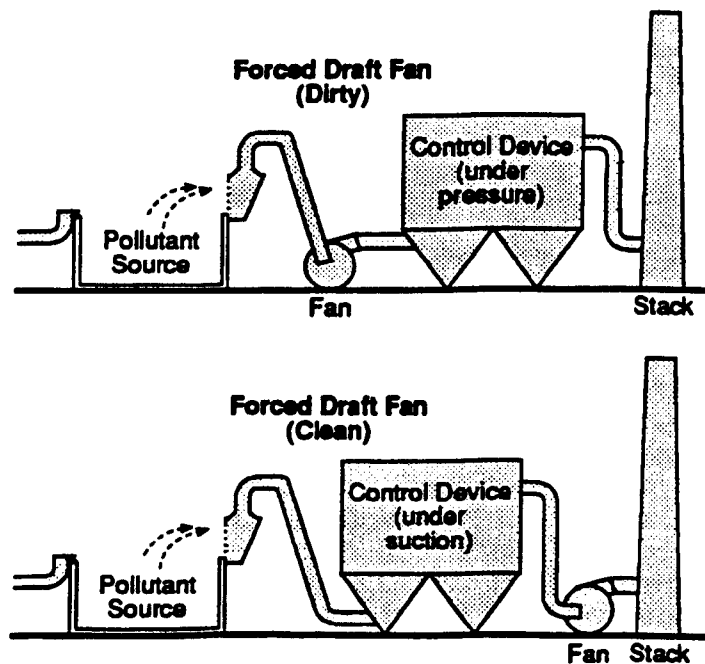
Figure 5-1. Fan Locations

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.

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.

5-3

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.
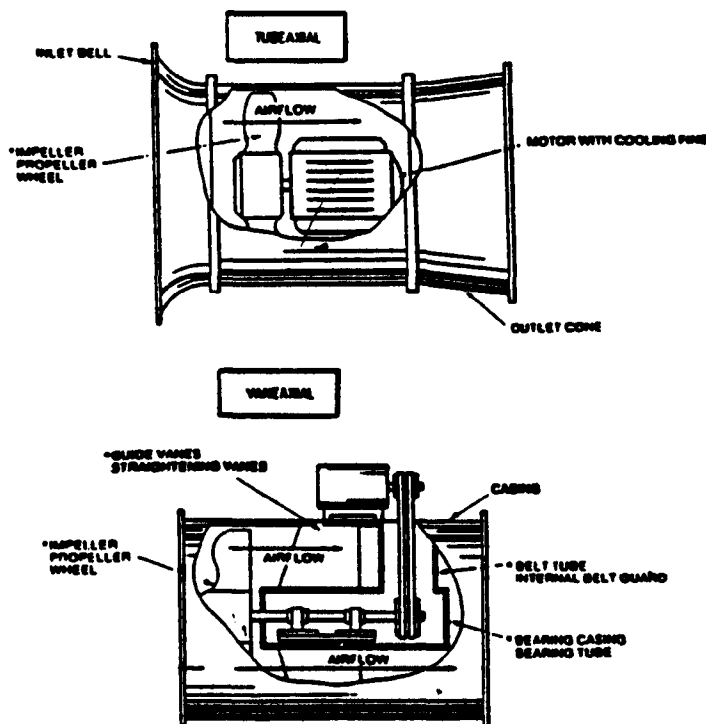
Figure 5-2. Tubeaxial and Vaneaxial Fans
(ACGIH, 1988)

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.

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)
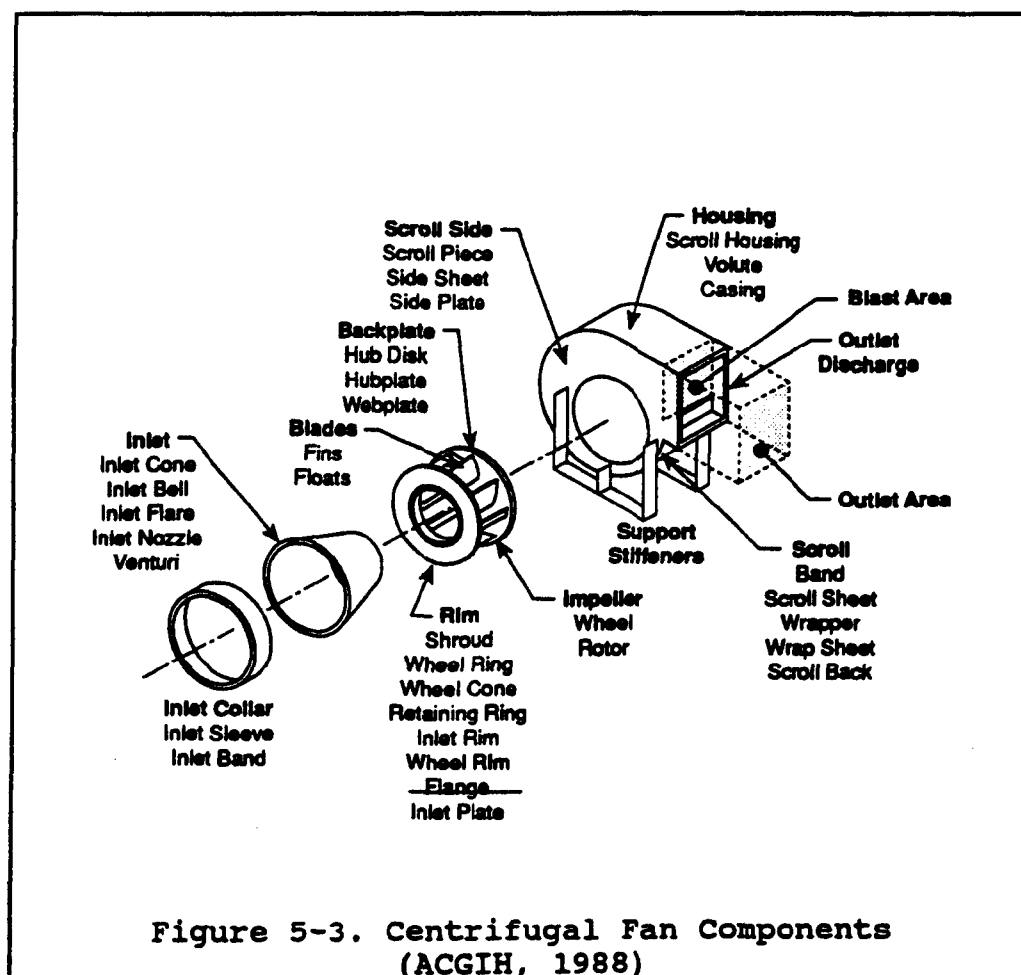
Forward curved impellers, shown in Figure 5-4, have blades that curve into the direction of rotation. Commonly referred to as "squirrel-cage blowers", they are constructed of lightweight materials and usually have 24 to 64 closely-spaced blades. For a given duty, these impellers are the smallest of all 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 (ME) is developed at a point to the right of peak static pressure (SP), at about 60 percent of the wide open volume, and the horsepower requirement (HP) 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.

Figure 5-4. Forward Curved Wheel Design
and Performance (Modifed Drawing Based
on ACGIH, 1988)
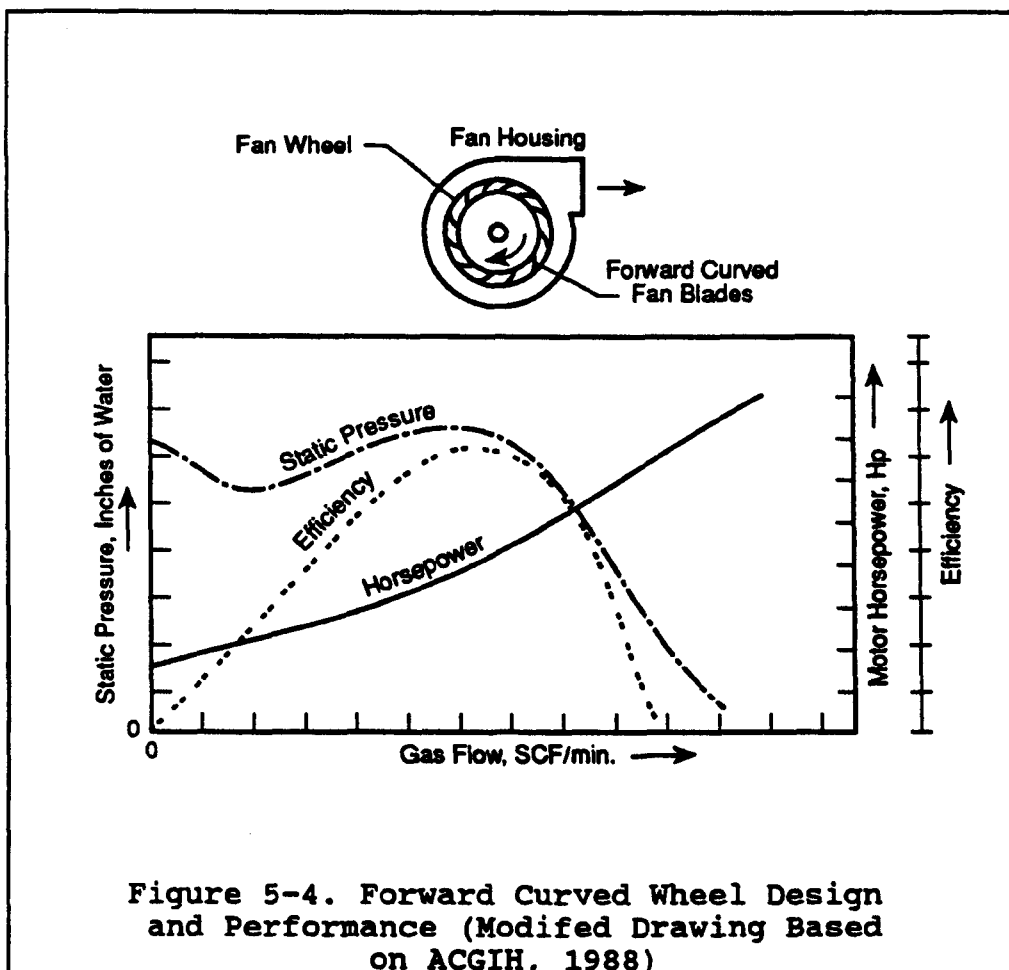
Radial impellers, shown in Figure 5-5, have 6 to 10 blades that extend either straight out from the hub (R) or in a radial direction (M). They are the simplest of all the centrifugal fans and the least efficient, but they are capable of developing the highest static pressures. For a given duty, they 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 5-5. Radial Wheel Design and Performance
(Modified Drawing Based on ACGIH, 1988)
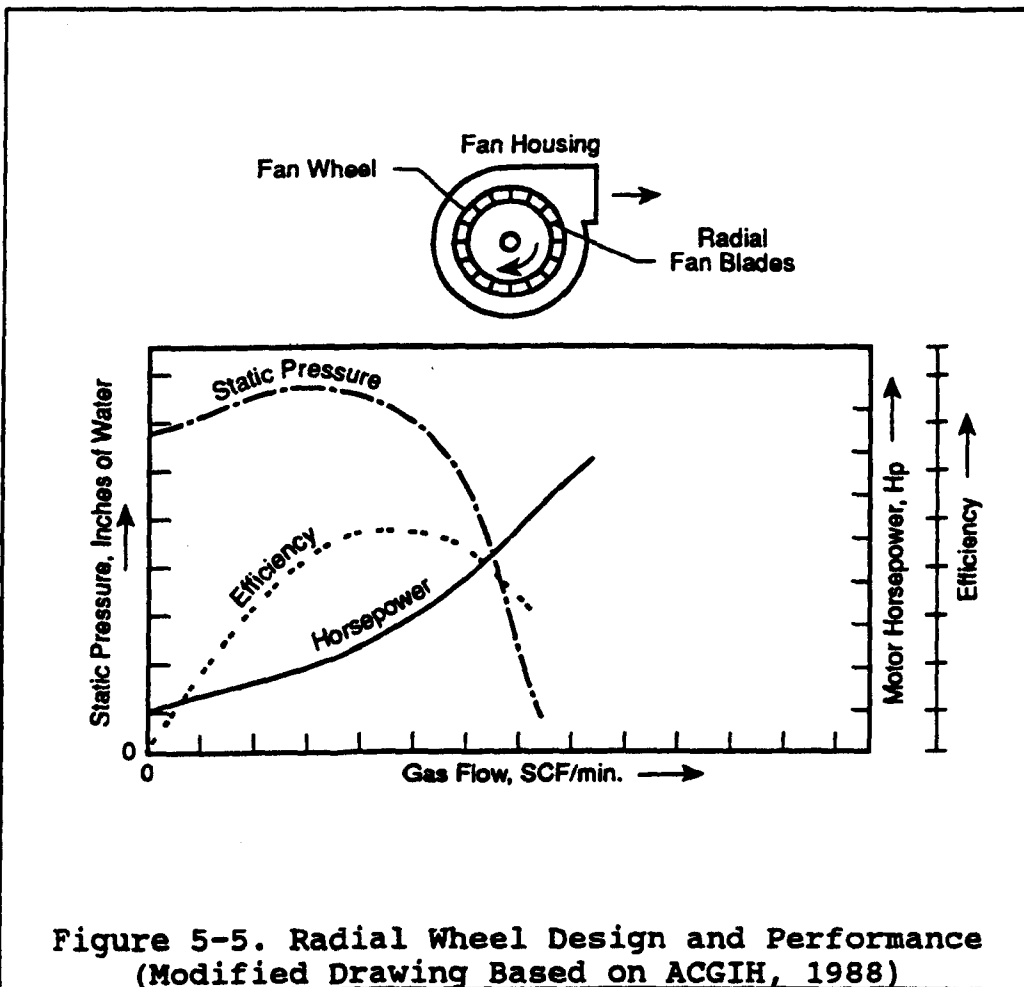
The backward inclined impellers have 9 to 16 hollow airfoil-style blades that incline or curve away from the direction of rotation. They have the highest efficiency of all the centrifugal fans and, for a given duty, will operate at the highest speed. Highest mechanical efficiency is developed to the 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. 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.

The standard backward inclined impellers, shown in Figure 5-6, are identical in design to the airfoil-type, except they employ single-thickness blades. Peak efficiency is slightly less than the airfoil 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. 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.

Fan Arrangements
Fans are constructed with different bearing locations and motor mounting capabilities, generally referred to as "arrangements". Knowledge of the various arrangements is desirable when it is useful to speak the language of fan systems.

There are ten basic fan arrangements (see AMCA, 1979) and three of the most common of these are shown in Figure 5-7. 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.

Figure 5-6. Backward Inclined Standard Wheel
Design and Performance
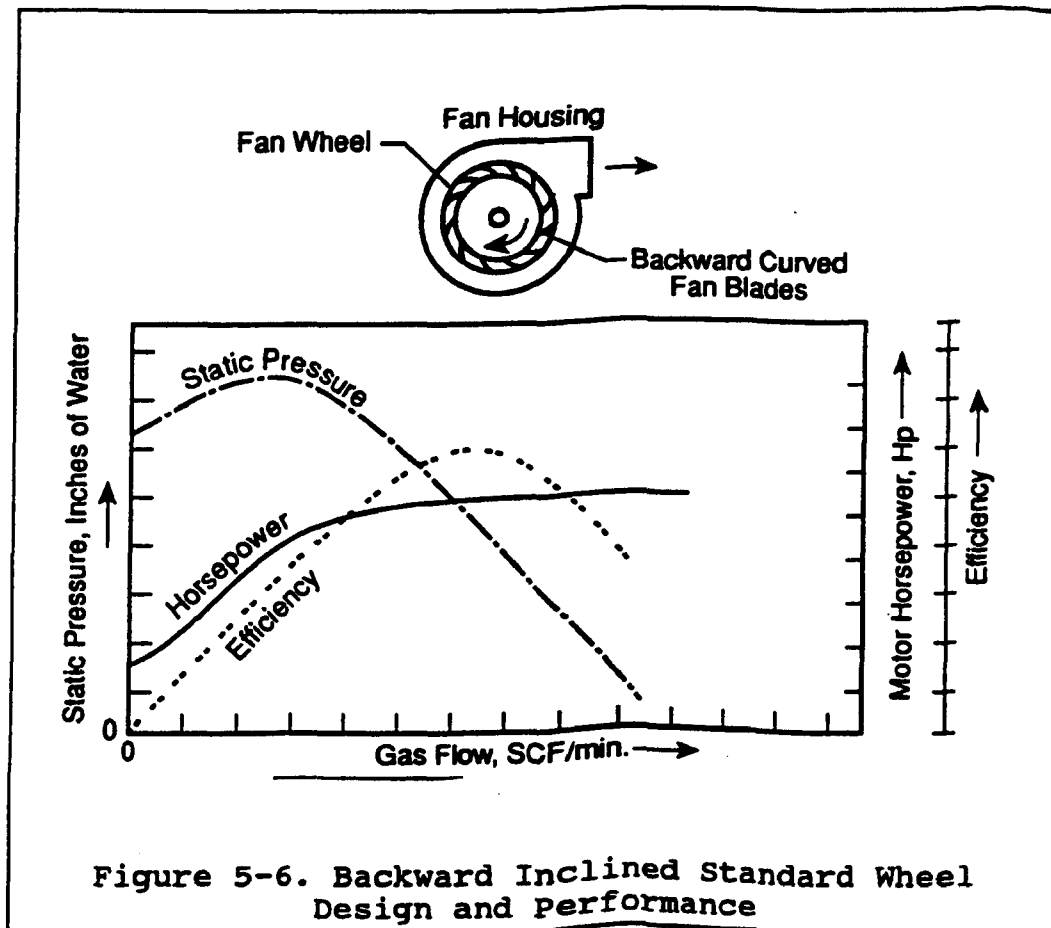
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The more common single-inlet fans have the air inlet on only one side, opposite the drive. Double-inlet fans have air inlets on both sides.

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. 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. Most of the other fan arrangements not shown in Figure 5-7, are basically variations on these three.

5-13
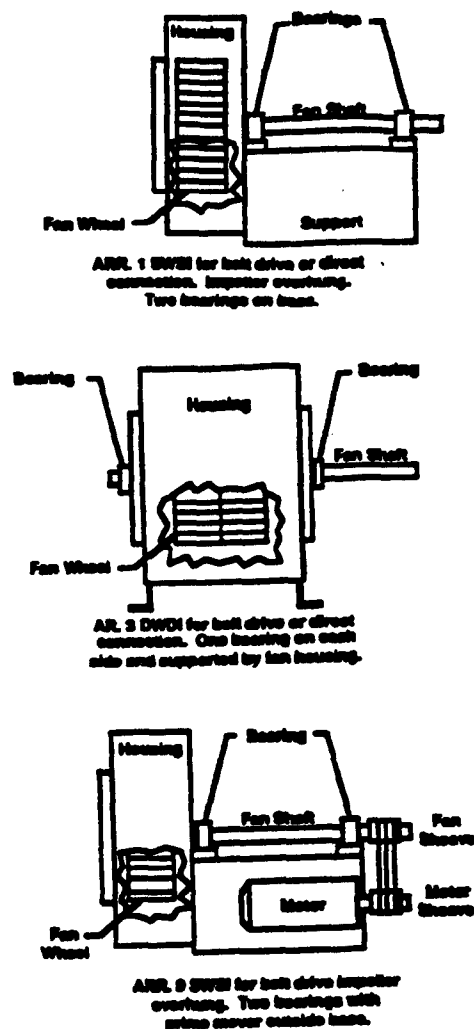
Figure 5-7. Fan Arrangements
(Modified Drawing Based on AMCA, 1979)

II. 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 dimen-
sional 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:

5-14

$$Q_2 = Q_1(size_2/size_1)^3(rpm_2/rpm_1)$$

$$P_2 = P_1(size_2/size_1)^2(rpm_2/rpm_1)^2(\rho_2/\rho_1) \qquad \text{(Eqn.1)}$$

$$bhp_2 = bhp_1(size_2/size_1)^5(rpm_2/rpm_1)^3(\rho_2/\rho_1)$$

The performance variables involved here are flowrate (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 lesson.

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 flowrates at these two conditions are related by:

$$P_2 = P_1(Q_2/Q_1)^2 \qquad \text{(Eqn. 2)}$$

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 ($r_1 = r_2$), the fan laws become:

$$Q_2 = Q_1(rpm_2/rpm_1)$$

$$P_2 = P_1(rpm_2/rpm_1)^2 \qquad \text{(Eqn. 3)}$$

$$bhp_2 = bhp_1(rpm_2/rpm_1)^3$$

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 flowrate 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.

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
pressure developed and air volume moved, labeled SP.
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 pressure/air volume
combinations that that fan is capable of when opera-
ting 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 duct system curves for three systems are   5-15
shown in Figure 5-8. Plotted here is the percentage of
duct system resistance as a function of the percent of
duct system flowrate. This is simply a normalized
version of a P verses Q plot. The system curves shown
follow the general relationship characteristic of
turbulent systems:

$$P_2 = P_1 (Q_2/Q_1)^2 \qquad\qquad (Eqn.\ 4)$$

Figure 5-8. Normalized Duct Curves
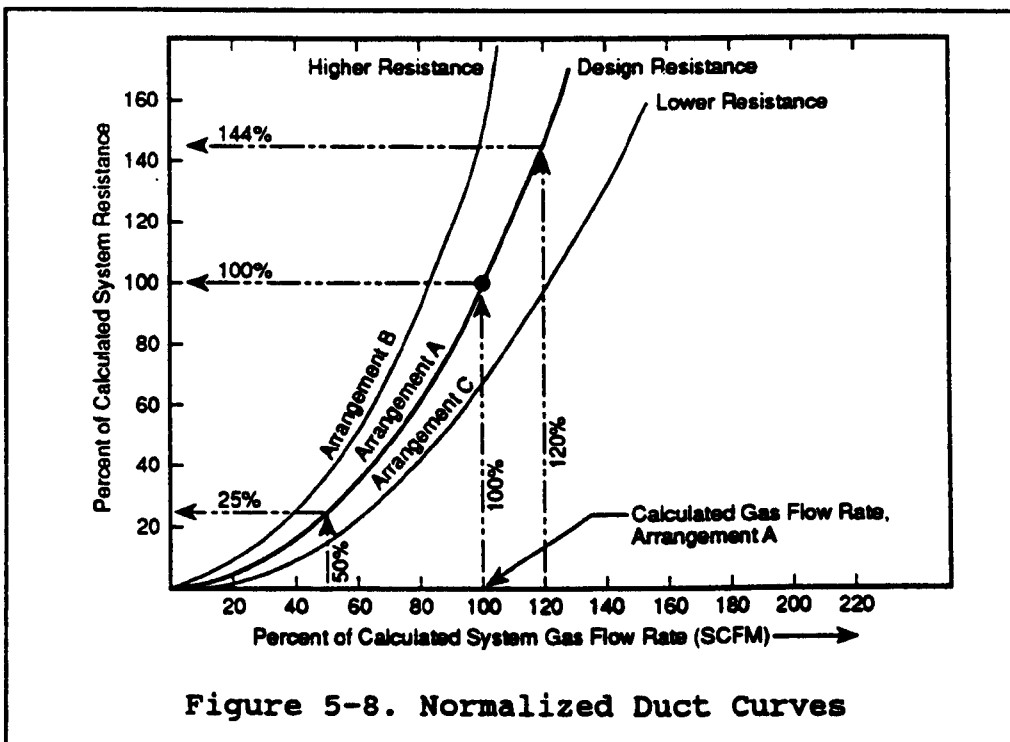
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In practice, the system curve is developed by first determining the resistance or static pressure for one flowrate through the system, using the techniques discussed in Lessons 2 and 3. Other points on the curve are then determined using the above relationship. Thus, if the design point for System A were at 100 percent volume and 100 percent resistance, increasing the flowrate to 120 percent of the design flow would increase the resistance to 144 percent of the design resistance. Likewise, decreasing the flowrate 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.

The point of intersection of the system curve with the fan curve determines the actual fan performance. This is shown in Figure 5-9, 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.
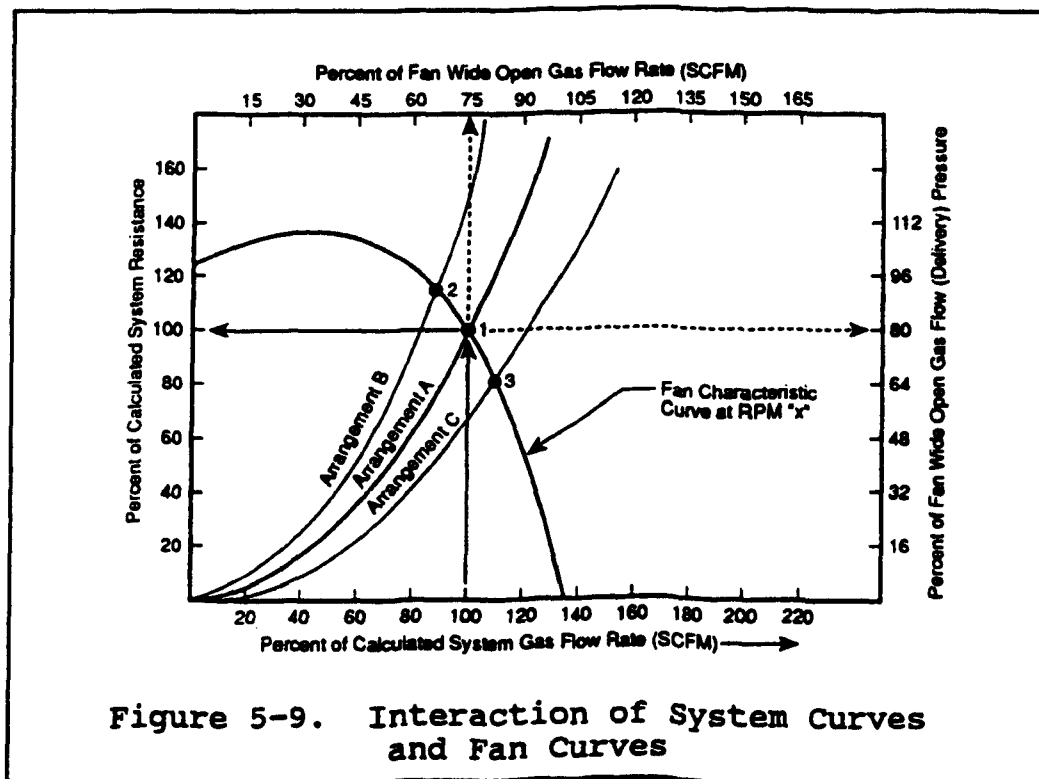
Figure 5-9. Interaction of System Curves and Fan Curves

One way to change the flowrate would be to change
the system. This could be done by closing or opening a
damper, producing a system with more or less resis-
tance and changing the system curve. For example,
referring again to Figure 5-9, the flowrate 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 flowrate could be increased to 120 percent
of the design value by opening a damper and shifting the
intersection to Point 3.

Changes in flowrate could also be produced by
changing the fan speed, shifting the fan curve. This is
illustrated in Figure 5-10, where a new fan curve
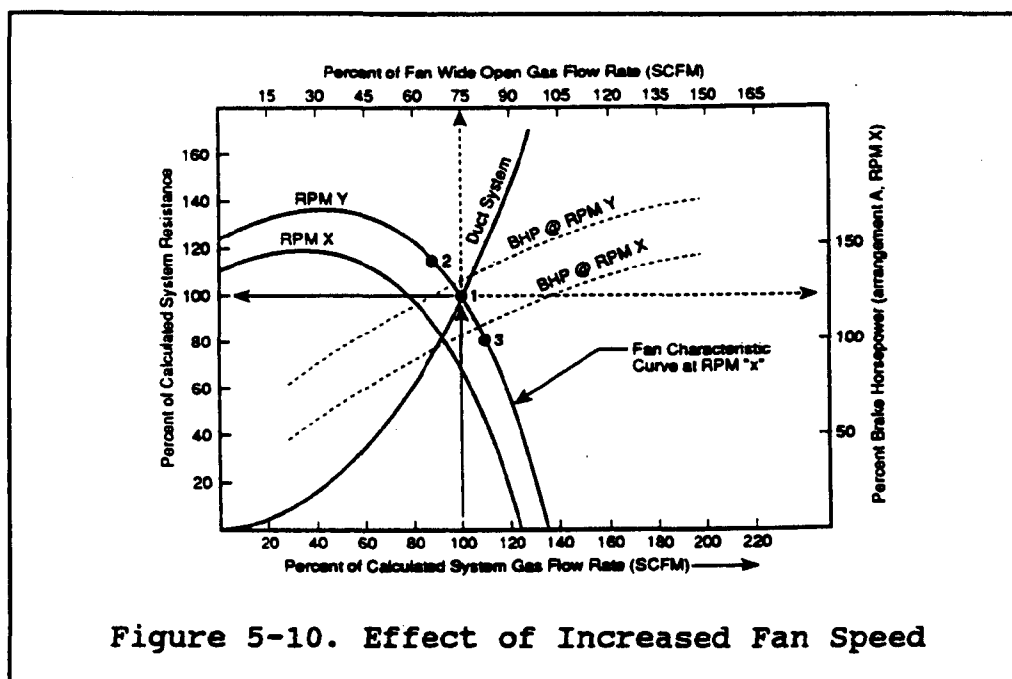
Figure 5-10. Effect of Increased Fan Speed

representing a 10 percent increase in speed has been
added. At this new speed, the point of operation shifts
to Point 2. Since flowrate 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.

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 5-11 for a density change from 0.0375 lb/ft³ to 0.075 lb/ft³. As previously indicated, the new operating point will deliver the same air volume but at double the resistance and double the horsepower requirement.

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 perfor-
mance 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.

Figure 5-11. Influence of Gas Density
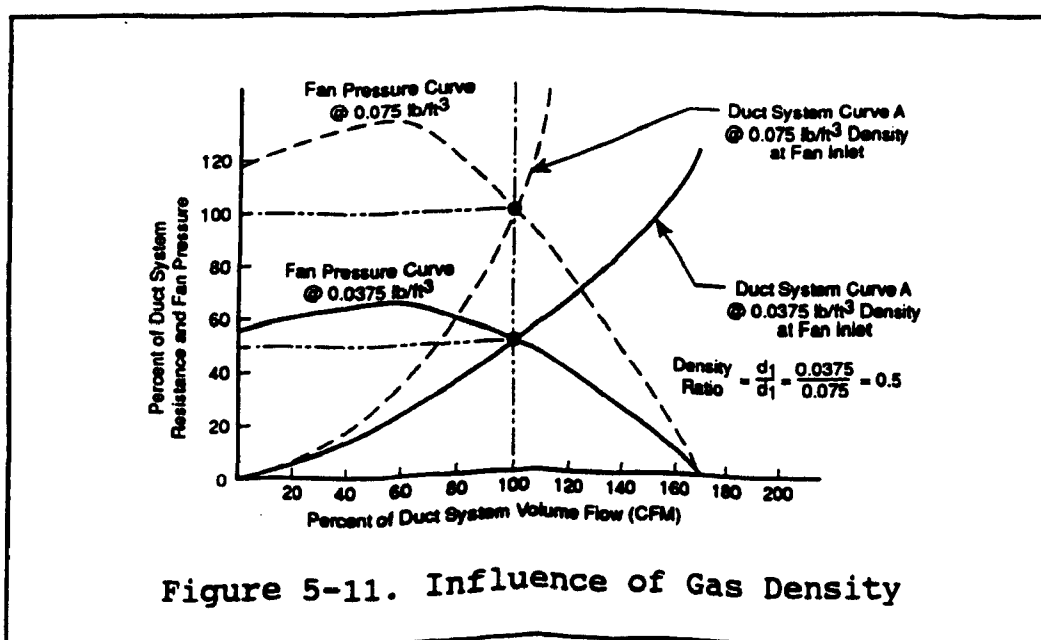
The influence of system effects on fan performance is
shown in Figure 5-12. Here, the solid system curve has
been determined without consideration of system effects
and performance corresponding to Point 1 is anticipated.
However, because of system effects the effective fan
curve is the dashed line and performance corresponding
to Point 3 is obtained.
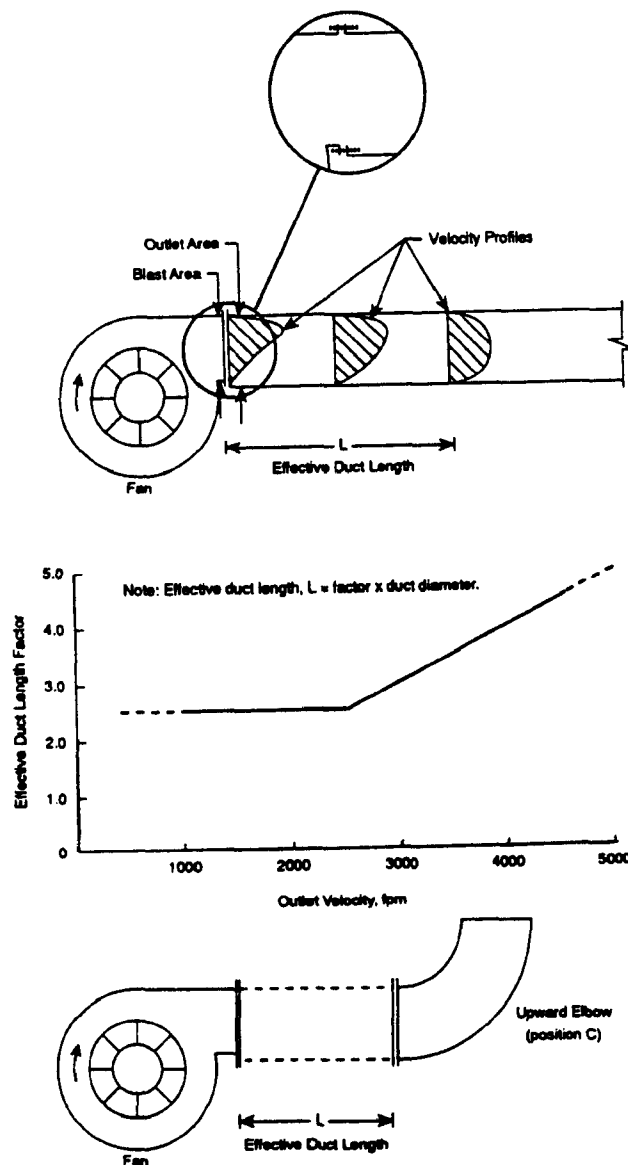
Figure 5-12. System Effects for Outlet Ducts
(Modified Drawing Based on ACGIH, 1988)

This deficient performance could be prevented by   Slides
calculating the system effect loss, adding it to the
system resistance and selecting a fan to operate at
Point 2.

System effect loss factors for some common
centrifugal fan outlet duct installation conditions are
given in Figures 5-13 and 5-14.   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.
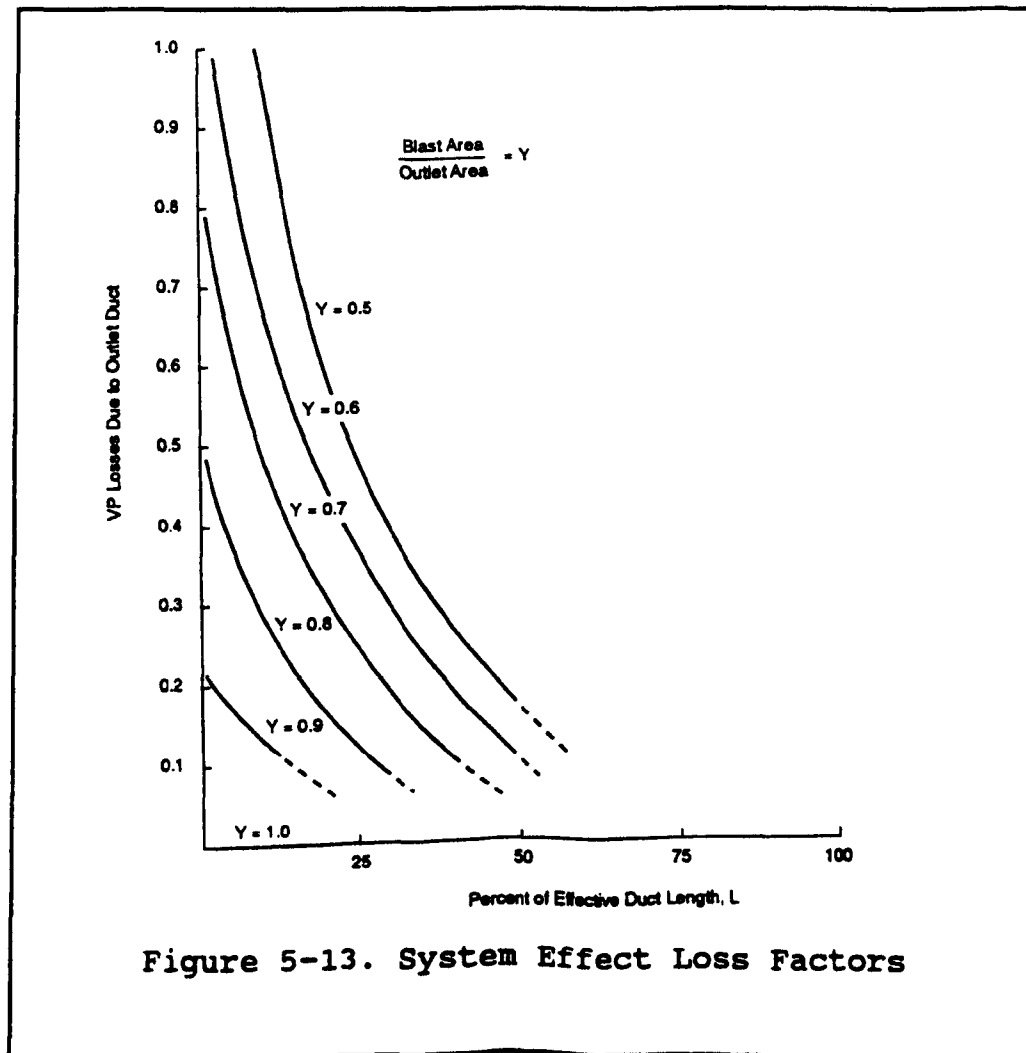
Figure 5-13. System Effect Loss Factors
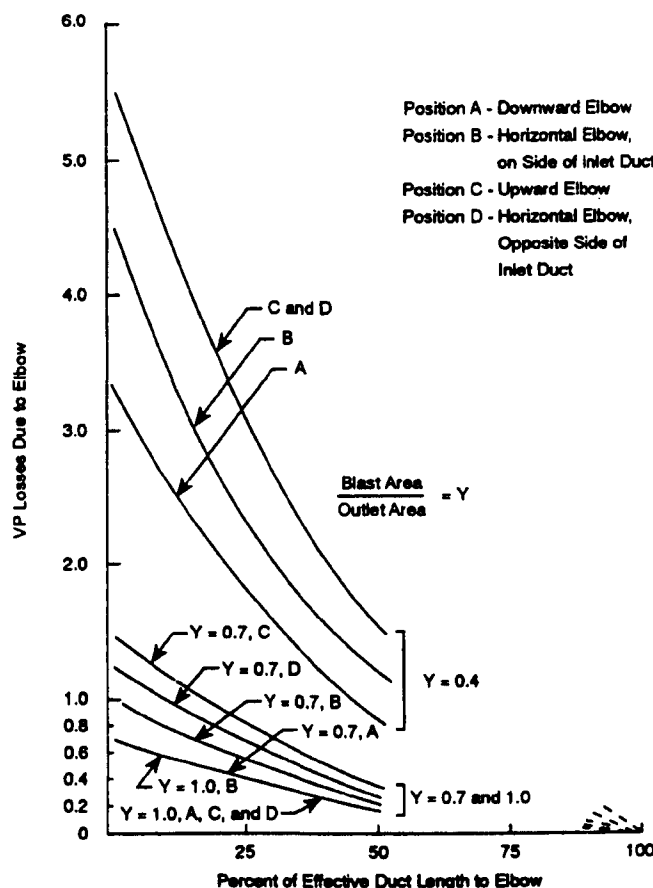
Figure 5-14. System Effect Factors for Outlet Duct Conditions Shown in Figure 5-13

System effect losses in Figures 5-13 and 5-14 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.

IV. 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 one of these tables is shown in Figure 5-18. 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.

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:

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:

$$FSP = SP_{out} - SP_{in} - VP_{in} \quad \text{(Eqn. 5)}$$

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:

$$FTP = TP_{out} - TP_{in} \quad \text{(Eqn. 6)}$$

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:

$$FSP_{equivalent} = FSP_{actual}(0.075/\rho_{actual}) \quad \text{(Eqn. 7)}$$

Figure 5-15. Example Fan Ratings Table

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:

$$bhp_{actual} = bhp_{equivalent}(\rho_{actual}/0.075) \qquad \text{(Eqn. 8)}$$

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.

V. 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:

$$Q_2 = Q_1(rpm_2/rpm_1) \qquad (Eqn. 9)$$

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 flowrate.

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 Lesson 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:

$$FSP_{estimated} = 0.075(SP_{out} - SP_{in} - VP_{in})/\rho_{actual} \quad (Eqn. 10)$$

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 flowrate. 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 flowrate. Techniques for determining these parameters, as well as the other parameters involved in the evaluation of fan performance, will be discussed in Lesson 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.

LESSON 6
MEASUREMENT OF VENTILATION SYSTEM PARAMETERS

In previous lessons, the use of various parameters to evaluate the performance of ventilation system components has been suggested. In this lesson, 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.

I. MEASUREMENT PORTS

Slides

When a ventilation system is first inspected, it is unlikely that measurement ports will be available. If   6-1
some ports are available, they are not likely to be in the locations needed or of an appropriate size. The   6-2
most likely port to be found on a ventilation system is a 3 or 4 inch diameter sampling port located on or near   6-3
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   6-4
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

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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 connecte
d to controllers that initiate equipment shutdown 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.

II. STATIC PRESSURE MEASUREMENTS

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 lesson. 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.
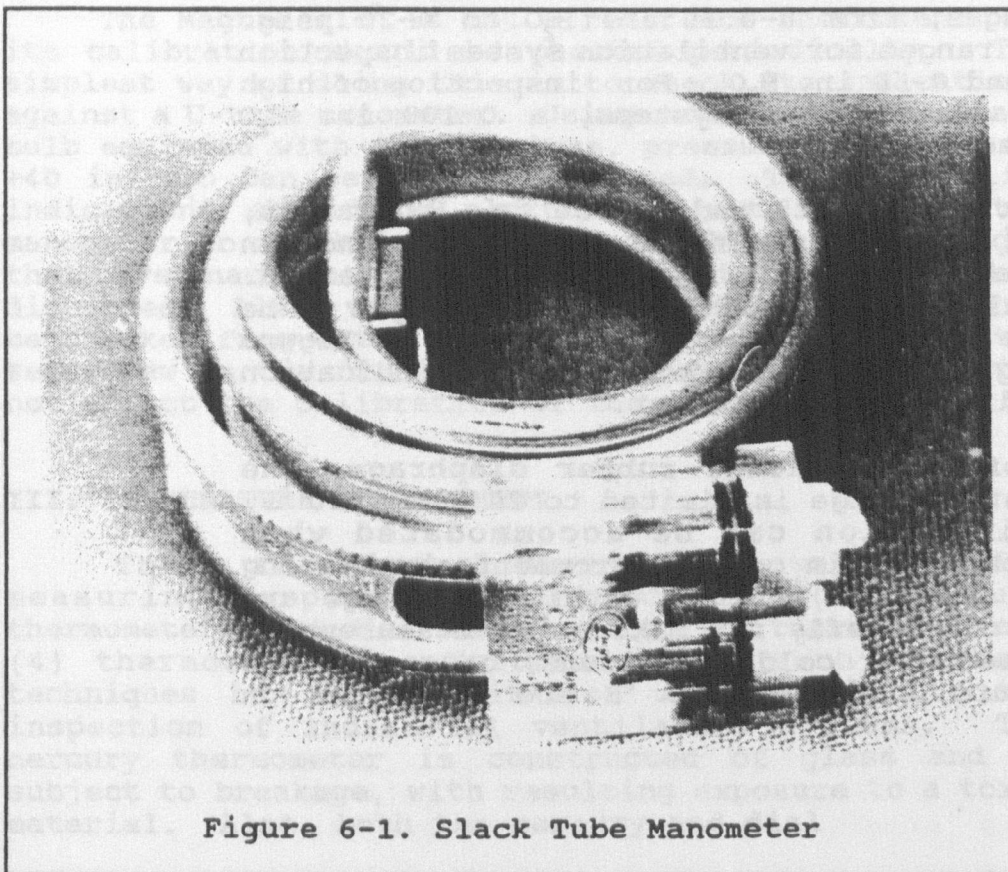
Figure 6-1. Slack Tube Manometer

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, 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 (as shown in Figure
6-2), with the gauge giving the most readable indication
used for recording the measurement.

Because of the silicone rubber diaphragm, the
ambient temperature range is limited to 20 °F 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
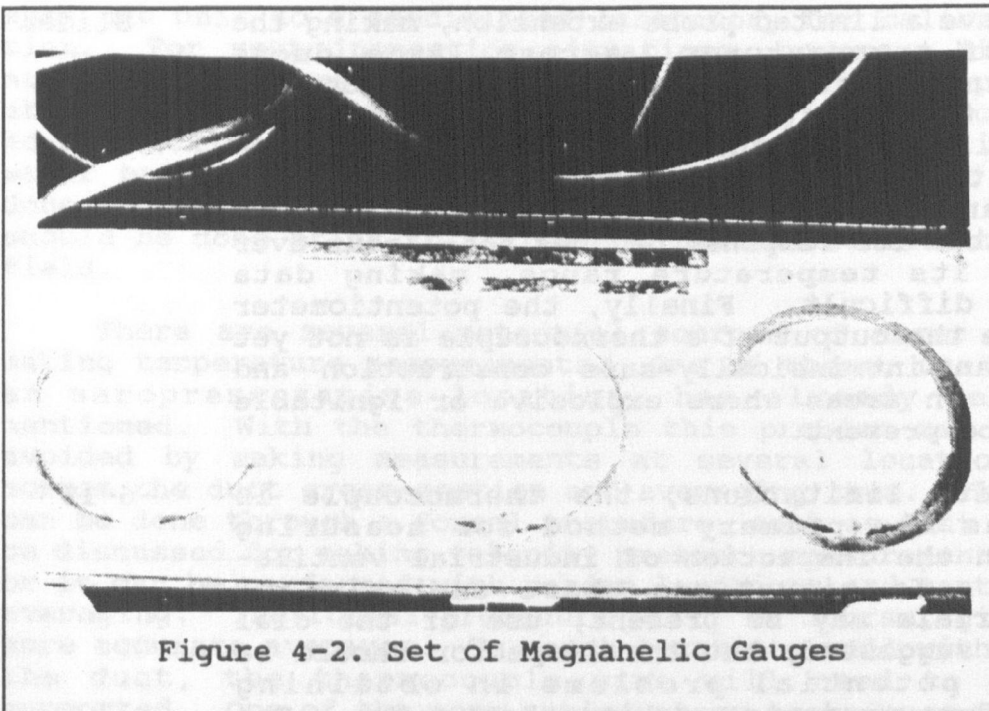lower temperature limit of -65 °F are available on
special order.

Figure 4-2. Set of Magnahelic Gauges

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 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.

III. 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 temper-
atures. 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.

Despite its limitations, the thermocouple is
recommended as the primary method for measuring
temperatures in the inspection of industrial ventila-
tion 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 thermo-
couple is measured with a potentiometer. A variety of
metals and alloys are used in the construction of
thermocouples, providing for measurements over differ-
ent 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 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.
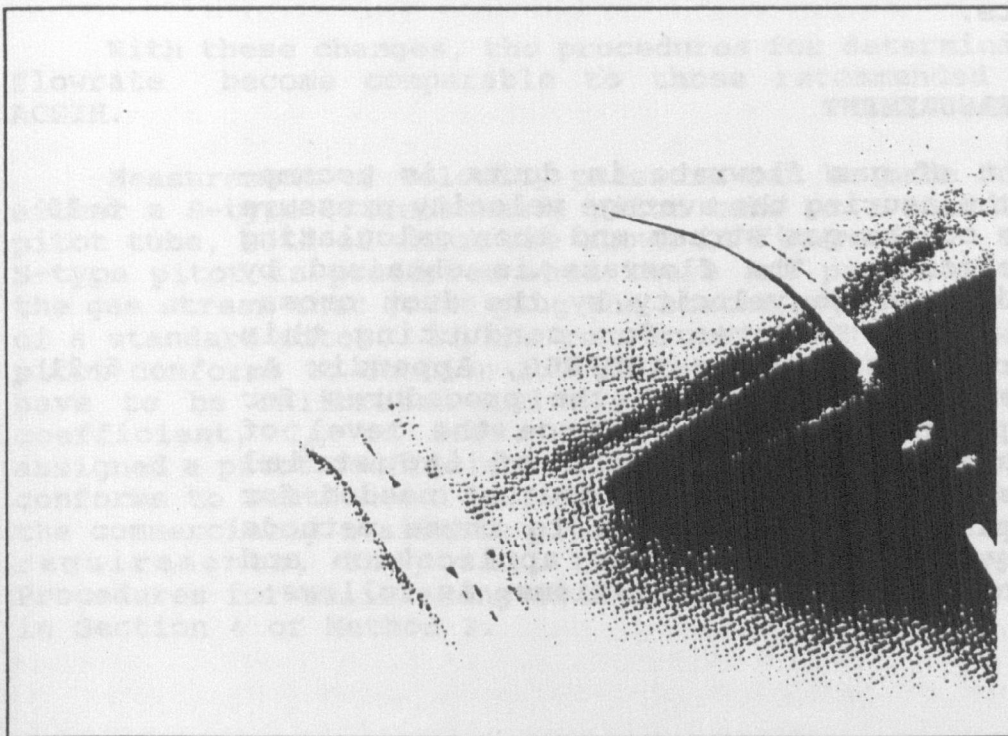
Problems can also occur from the cooling of the probe due to air infiltration through the port or through leaks into the duct upstream of the measurement point. The former problem can be avoided by sealing the port in the manner described in the section on static pressure measurements. In addition, if 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.

IV. FLOWRATE MEASUREMENT

Measurement of gas flowrate in ducts is accomplished by first measuring the average velocity pressure and temperature of the gas stream and then calculating the average velocity. The flowrate 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 flowrate. 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 flowrate. For inspection purposes,
   an absolute stack pressure of 29.92 in. Hg will be
   assumed.

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

   Measurement of velocity pressure can be made with
either a S-type (Staubscheid) pitot tube or a standard
pitot tube, both of which are shown in Figure 6-4. 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.
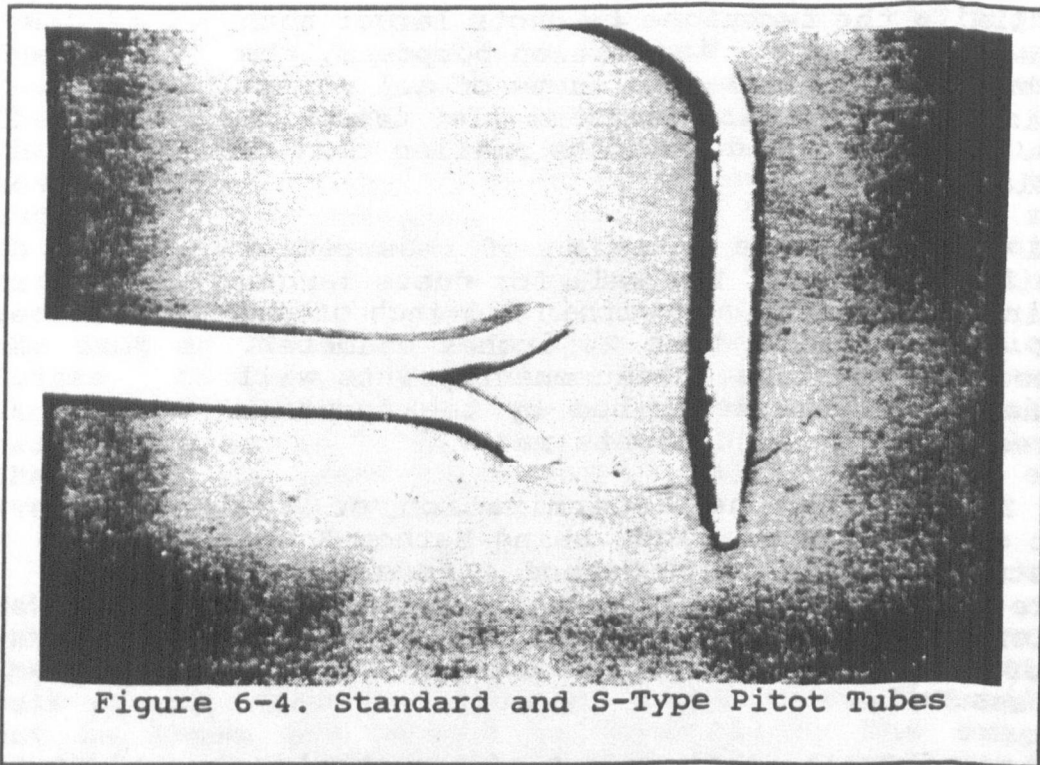
Figure 6-4. Standard and S-Type Pitot Tubes

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 feet/minute 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.
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 6-5 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

6-25
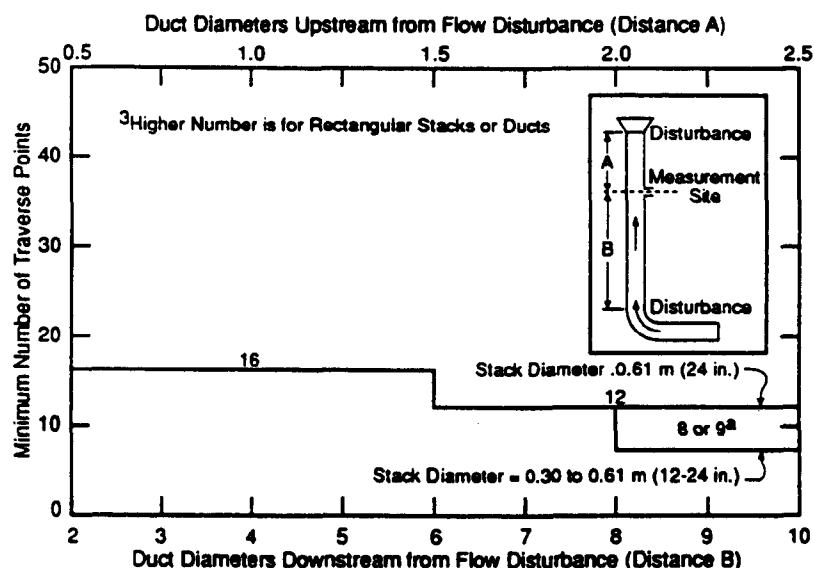
**Figure 6-5. Minimum Number of Traverse Points
(40 CFR 60, Appendix A)**

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:

$$D_{eq} = 2LW/(L + W) \quad (6-1)$$

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

For circular ducts, the number of traverse points
determined from Figure 6-5 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

6-26
and
6-27

Table 6-1. Location of Traverse Points[a]
in Circular Ducts

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	32.2
5		85.4	67.7
6		95.6	80.6
7			89.5
8			96.8

[a] Note: Values are expressed in percent of duct
diameter from inside wall to traverse point.

from the inside wall to the point location. Locations
for other numbers of points may be found in Table 1-2
of EPA Method 1. 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.

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:

$$V = 2.9C_p(\Delta P^{0.5})_{average}(T_{average})^{0.5} \quad (6-2)$$

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

The average gas flowrate is then determined from:

$$Q = 60VA \quad (6-3)$$

where Q = average gas flowrate (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 flowrate 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

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will depend primarily on the number of measurement locations used.

One of the more common velocity measuring instruments is the swinging vane anemometer. 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 second type of anemometer is the rotating vane instrument. 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 third type of anemometer is the "hot wire." 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.

V. FAN SPEED MEASUREMENT

6-29

Techniques available for the measurement of fan speed include: (1) standard tachometers, (2) strobe-tachometers and (3) phototachometers. Strobetach-ometers 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.

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

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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:

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

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.

VI. 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 6-6, with horsepower on the x-axis and amperage on the y-axis.

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".
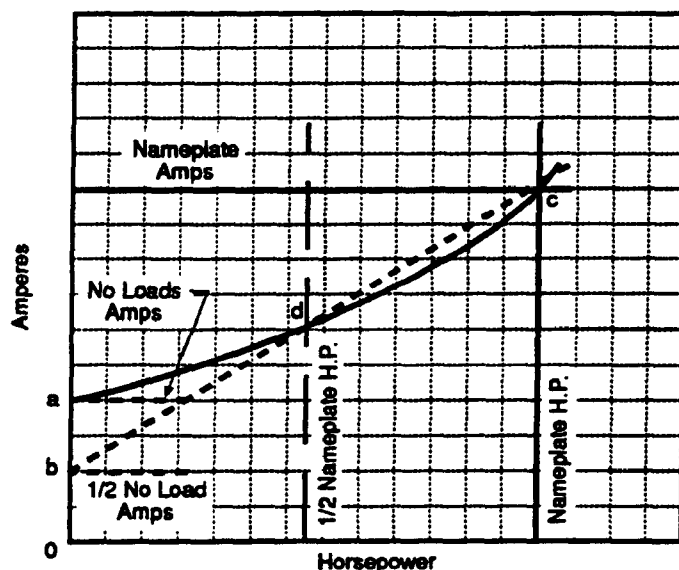
Figure 6-6. Determination of Operating Horsepower
(SMACNA, 1967)

5. Connect the motor to the fan and measure the amperage when running at load. Read the 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:

$$bhp = 3^{0.5}VAfe/746 \ (6-5)$$

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.

VI. 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 6-7. The pipe clamp is firmly attached to the probe and the jaw clamp 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.
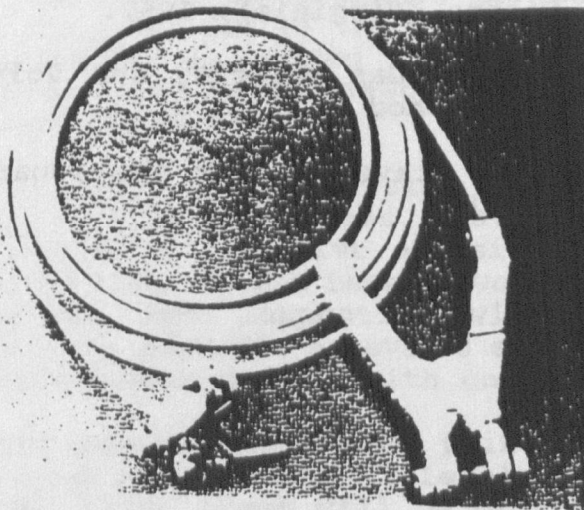
Figure 6-7. Grounding Cable

If the resistance is less that 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.

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 Equipment", 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.

LESSON 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 judgement 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.

I. 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 flowrates; 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.

<u>Coolers</u>

1. *Physical condition:* indications of corrosion,   7-5
   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 flowrate:* observe plant flow meters or
   pressure gauges to evaluate changes in water
   flowrate on evaporative coolers.

<u>Fans</u>   7-6

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.

II. LEVEL 3 INSPECTIONS   7-7

<u>Hoods</u>

1. *Estimated volume:* estimate flowrate using SP_h,   7-8
   temperature and hood configuration.

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

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1. *Change in gas temperature:* measure tempera-   7-9
   ture 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. *Air volume:* determine flowrate by measuring VP
   and temperature.

Coolers   7-10

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   7-11

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

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

III. 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.
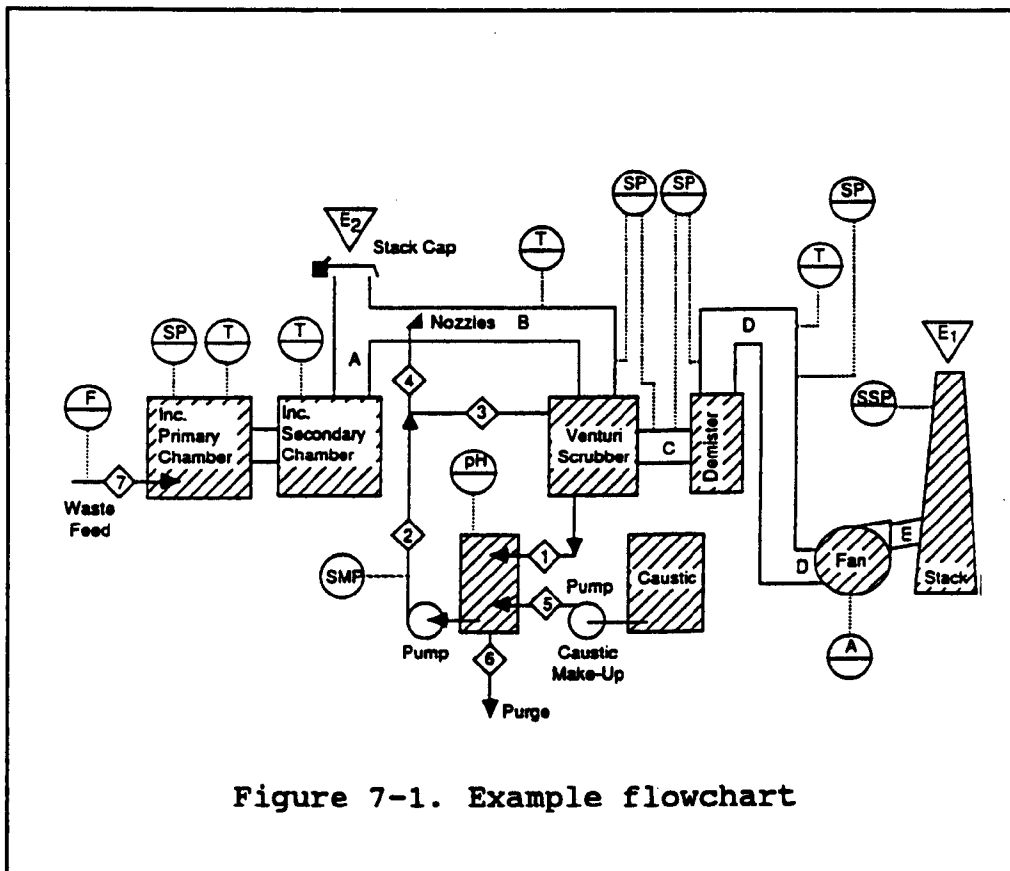
Figure 7-1. Example flowchart

Because they can serve many purposes, there are
many levels of sophistication in flowchart prepara-
tion. 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 7-12
inspections. Recorded data should be examined to be
sure that static pressures decrease from the inlet of 7-13
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.

References

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

APPENDIX A

SLIDES

VENTILATION FUNDAMENTALS

PROPERTIES OF AIR AND AIR-WATER VAPOR MIXTURES

PRINCIPLES OF FLUID FLOW

PROPERTIES OF AIR
AND AIR-WATER VAPOR
MIXTURES

STANDARD AIR

Density = 0.075 lb/ft^3

Absolute Viscosity = 1.225 x 10^{-5} lb$_m$ /ft-sec

Equivalent to dry air at 70 °F and 29.92 In. Hg

MOLECULAR WEIGHT

$$MW_{mixture} = \sum (x_{component})(MW_{component})$$

$$MW_{dry\ air} = approx.\ 28.95\ lb\ /\ lb\text{-mole}$$

$$MW_{wet\ air} = (1 - x_{water})(MW_{dry\ air}) +$$
$$(x_{water})(MW_{water})$$

IDEAL GAS LAW

P = absolute pressure (lb_f / ft_2)

V = gas volume (ft^3)

n = number of moles (lb-moles)

R = constant (1545 ft-lb_f /lb-mole-°R)

T = absolute temperature (°R)

IF MOLES DON'T CHANGE...

$$PV/T = nR = \text{constant}$$

or

$$P_1V_1/T_1 = P_2V_2/T_2$$

$$V_1 = V_2(P_2/P_1)(T_1/T_2)$$

Density =

    Weight / Unit Volume

Specific Volume =

    Volume / Unit Weight = 1 / Density

For an ideal gas:

Density (ρ) =

    (MW / 387) (530 / T) (P / 29.92)

Specific Gravity = $\rho_{gas}/ \rho_{dry\ air}$

For an ideal gas:

Specific Gravity = $MW_{gas}/MW_{dry\ air}$

GIBBS-DALTON RULE
OF PARTIAL PRESSURE

$$P_{total} = p_{dry\ air} + p_{water}$$

Relative Saturation =

$$P_{water} / P_{water\ at\ saturation}$$

Relative Humidity =
Relative Saturation x 100

Absolute Humidity =
$$Weight_{water} / Weight_{dry\ air}$$

Dry Bulb Temperature

Wet Bulb Temperature

Dew Point Temperature

ENTHALPY

The measure of the thermal energy of a substance

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

h   = enthalpy (Btu / lb)

C_p = heat capacity at constant pressure (Btu / lb-°F)

t   = temperature of substance (°F)

t_{ref} = reference temperature (°F)

t_{ref} for dry air usually equals 0 °F

t_{ref} for water usually equals 32 °F

$$h_{water\ vapor} = h_{water} + \lambda_v$$

$$\lambda_v = \text{latent heat of vaporization} \\ \text{(Btu/lb)}$$

FOR AN AIR-WATER VAPOR MIXTURE:

$$h\ (Btu/lb_{dry\ air}) = h_{dry\ air} + \phi\ (h_{water\ vapor})$$

$$\phi = \text{absolute humidity } (lb_{water}/lb_{dry\ air})$$

USUALLY INTERESTED IN ENTHALPY DIFFERENCE

$\Delta H = h_2 - h_1$

$\qquad = (C_P)_2 \, (t_2 - t_{ref}) - (C_P)_1 \, (t_1 - t_{ref})$

$\qquad = (C_P)_{avg} \, (t_2 - t_1)$

PSYCHROMETRIC CHART

PRINCIPLES OF
FLUID FLOW
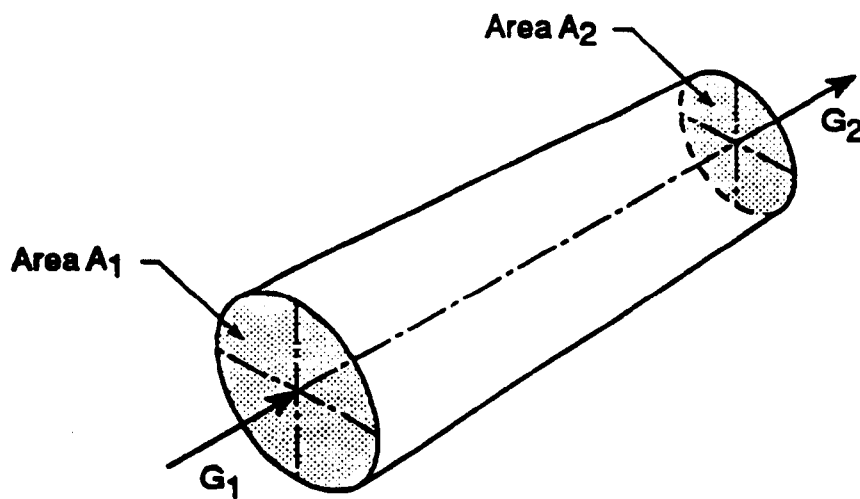
IF NO ACCUMULATION
OR REMOVAL...

$$G_1 = G_2$$

or

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_1$$

or, if $\rho 1 = \rho 2$ $$V_1 A_1 = V_2 A_2$$

BERNOULLI'S
EQUATION

For an incompressible, inviscid fluid:

$$V^2/2 + P/\rho + gZ = \text{constant}$$

V = fluid velocity

P = fluid pressure

ρ = fluid density

g = acceleration of gravity

Z = elevation of fluid

REARRANGING GIVES...

$$V^2/2g + P/\rho g + Z = \text{constant}$$

$V^2/2g$ = velocity head (ft)

$P/\rho g$ = pressure head (ft)

Z = potential head (ft)
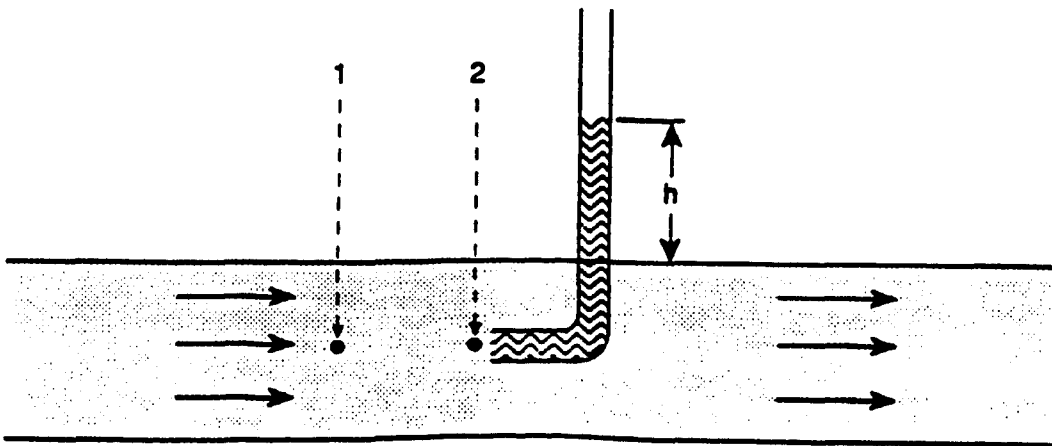
SUBSTITUTING GIVES...

$$V_1^2 / 2g + P_1 / \rho g = P_2 / \rho g$$

$V_1^2 / 2g =$ velocity pressure

$P_1 / \rho g =$ static pressure

$P_2 / \rho g =$ total pressure

Discharge Side

Intake Side

3500 FPM

3500 FPM

Fan

| -1.0 | + | 0.6 | = | -0.40 | | 5.0 | + | 0.6 | = | 1.1 |
| S_P | + | V_P | = | T_P | | S_P | + | V_P | = | T_P |
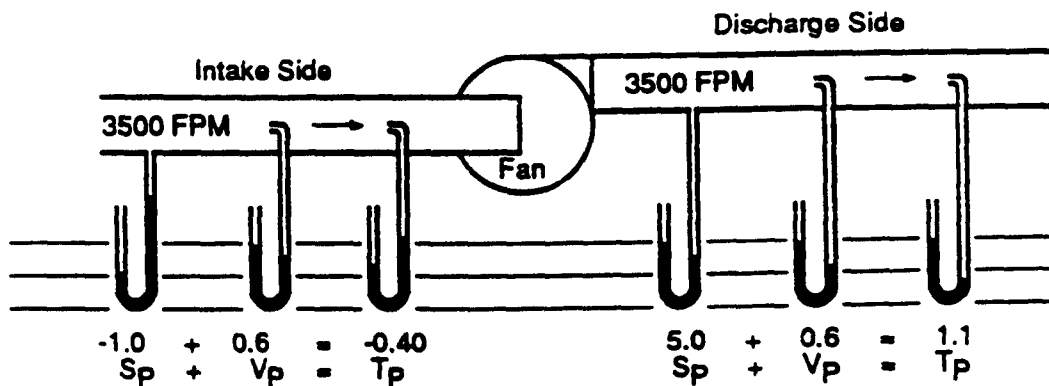
VELOCITY CAN BE DETERMINED FROM VELOCITY PRESSURE

$$V = 1096.7 \, (V_P / \rho)^{0.5}$$

V = velocity (ft/min)

V_P = velocity pressure (in. H_2O)

ρ = density (lb/ft^3)

For Standard Air (ρ = 0.075 lb/ft^3)

V = 4005 (V_P)

HOOD
SYSTEMS

**The Goal of Good Hood
Design is High Capture
Efficiency**

PENETRATION =

1 - FRACTIONAL EFFICIENCY

$$Pt_{total} =$$

$$Pt_{hood} + (1 - Pt_{hood}) \, Pt_{collector}$$

TYPES OF HOODS

- Enclosures
- Receiving
- Exterior

HOOD DESIGN
PRINCIPLES

- **Enclose whenever possible**

- **If can't enclose, place hood close to source**

- **Locate duct take-offs in line with normal contaminent flow**

Torpedo
Car

Ladel
Hood

Hot Metal
Ladle

Hopper

Dust and
Displaced Air

Duct

Valve

Hood

Bag

Slide 2-9

Housing

Grinding
Wheel

To Fan

Air
Handling
Duct

○

Dust
and Air

Hood

Slide 2-10

D_h

D_c

Hot Source

y x_f

D_s

Hypothetical
Point Source

HOOD DESIGN PRINCIPLES

• **Enclose whenever possible**

• **If can't enclose, place hood close to source**

• **Locate duct take-offs in line with normal contaminent flow**

Slide 2-12

To Pollution Control Device and Induced Draft Fan

Ventilation Air

Slots

Hood

Ventilation Air from Forced Draft Fan

Pollutant Laden Air

Process Tank

Jet Side — Exhaust Side

Air Curtain Jet

Copper Converter

Fugitive Emissions

To Suction Fan and Hood Sample Location

Ladle

Baffle Wall

Baffle Wall
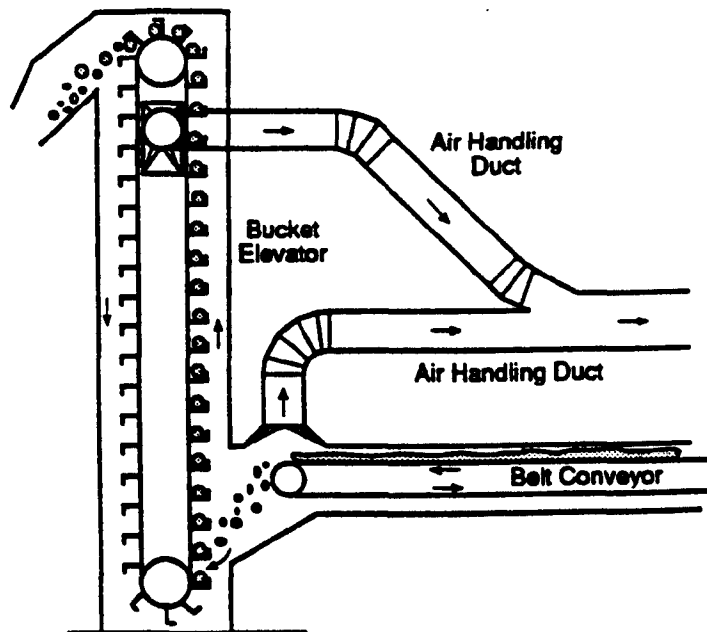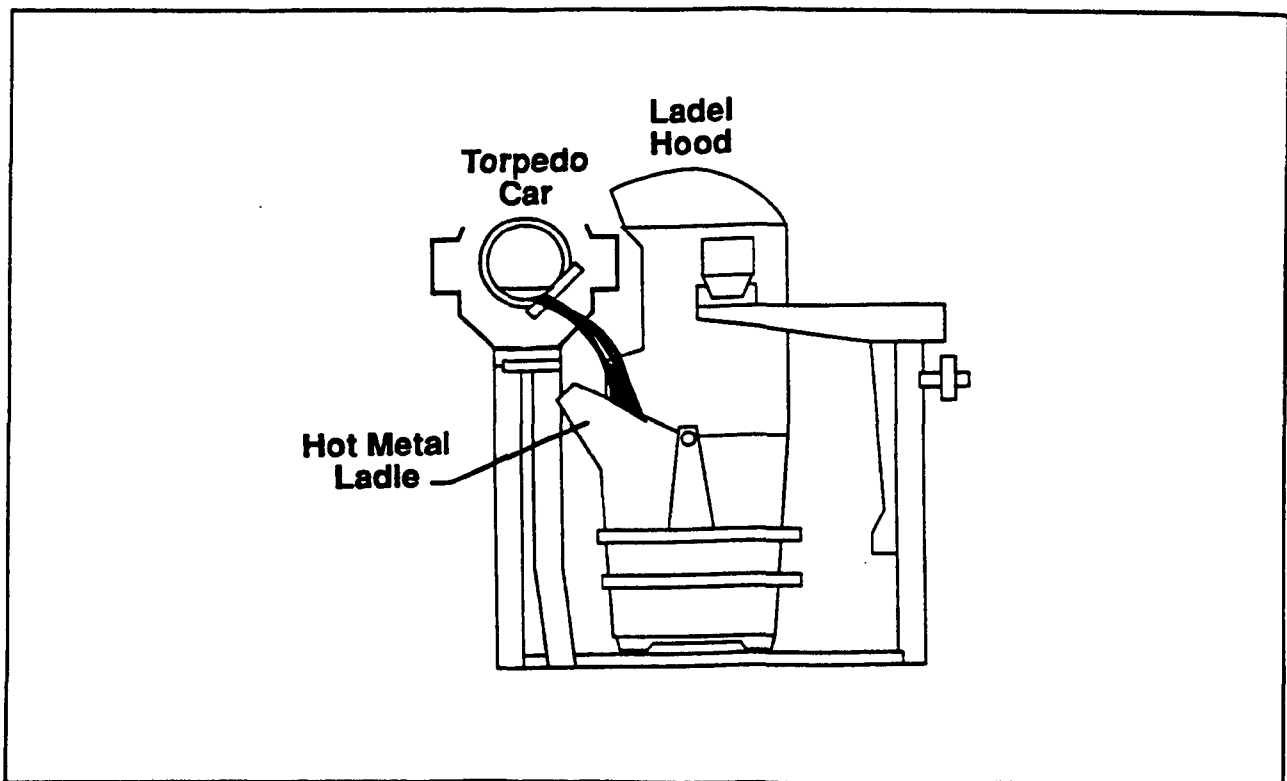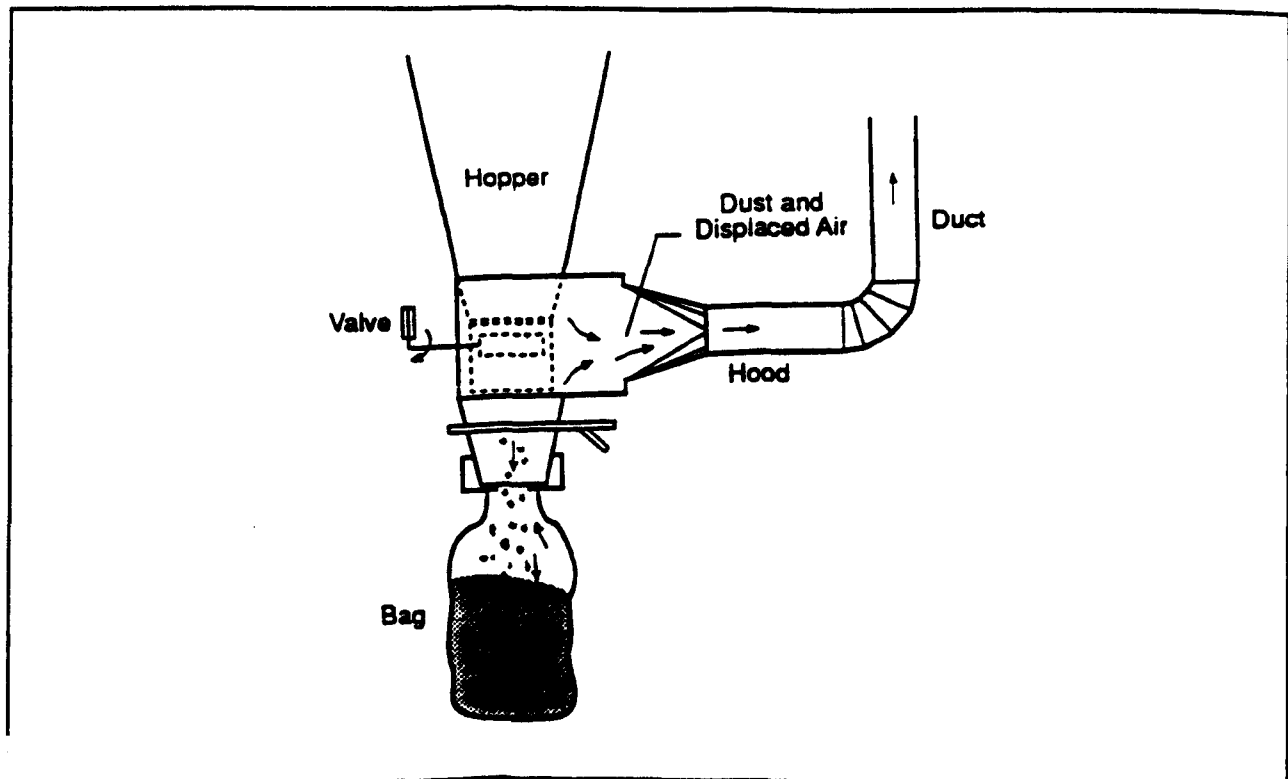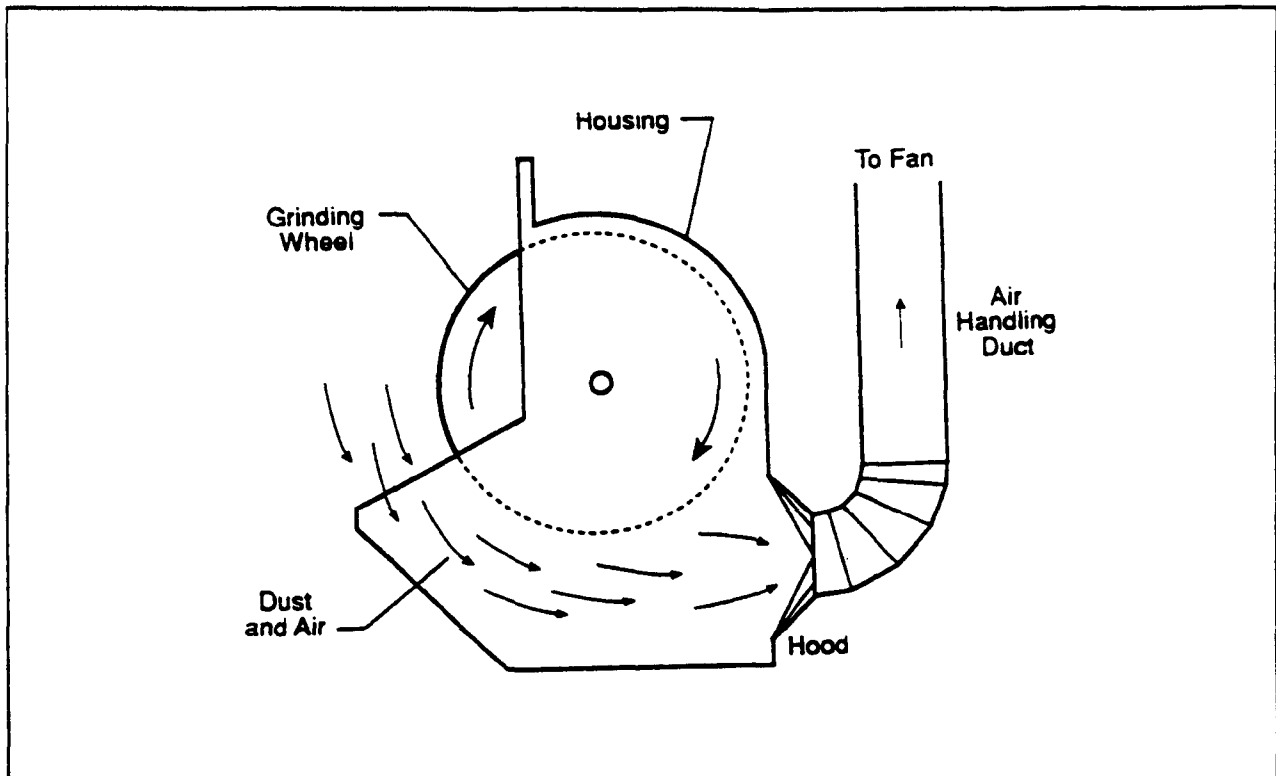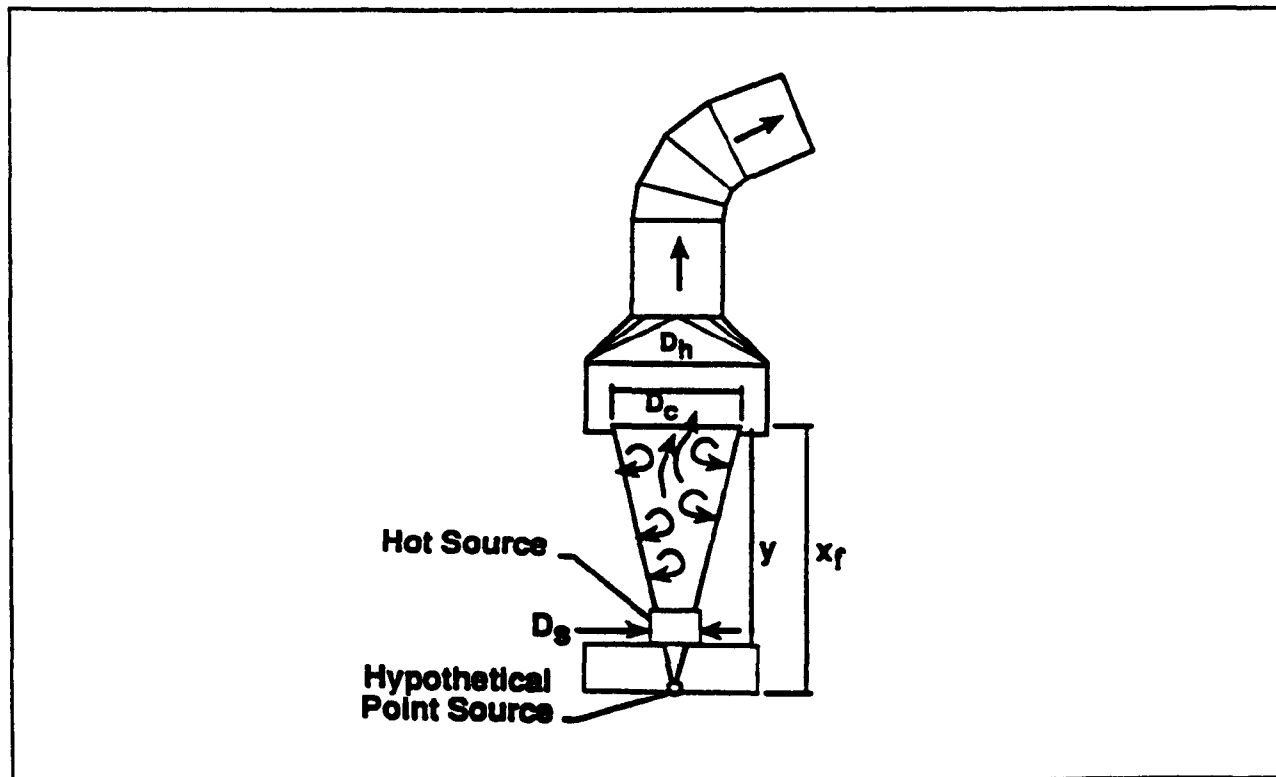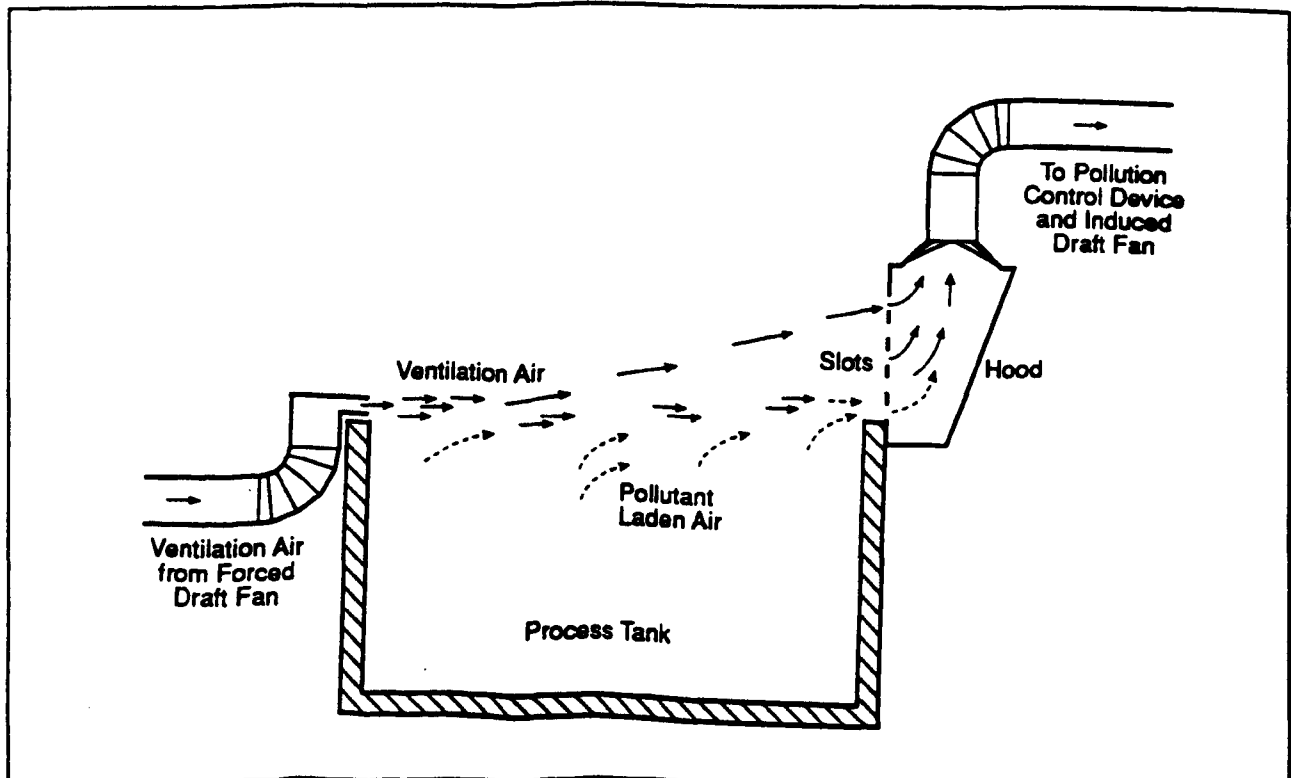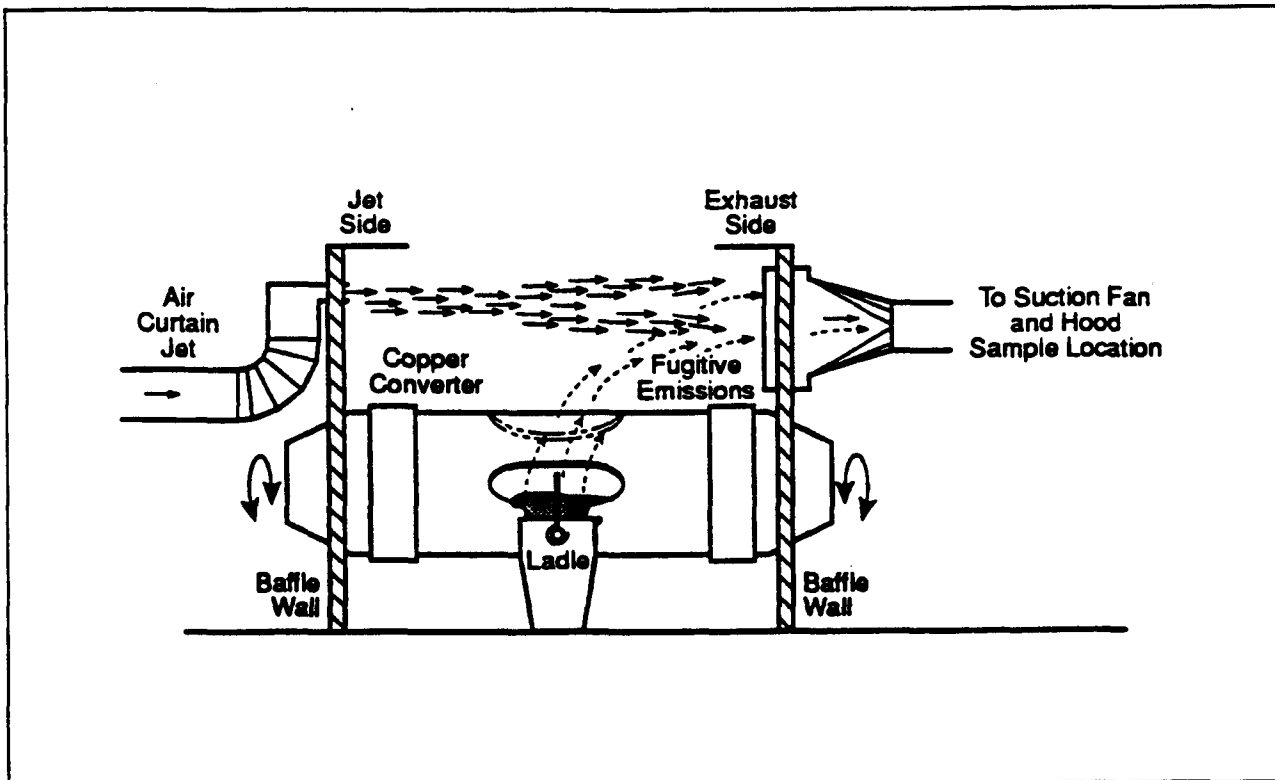
FACTORS AFFECTING HOOD PERFORMANCE

- Thermal air currents

- Motion of machinery

- Material motion

- Movement of operator

- Room air currents

CAPTURE VELOCITY

Air velocity in front of hood or at hood face necessary to overcome air currents and cause air to move into hood.

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

FOR COLD FLOW
INTO HOODS

Capture velocity decreases
with distance from hood face.

Hood

Duct

Flange

Air

Air

Air

Air

100% 60% 30% 15% 7.5%

Equal
Velocity
Lines

Slide 2 - 20a

Hood Type	Description	Aspect Ratio	Air Flow
W L X	Slot	0.2 or Less	$Q = 3.7 \, LVX$
X	Flanged Slot	0.2 or Less	$Q = 2.6 \, LVX$
W L X A=WL (sq. ft.)	Plain Opening	0.2 or Greater and Round	$Q = V (10X^2 + A)$
X	Flanged Opening	0.2 or Greater and Round	$Q = 0.75V (10X^2 + A)$

A-24

Hood Type	Description	Aspect Ratio	Air Flow
	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	$Q = 0.75V(10X^2 + A)$

FOR HOT FLOW INTO HOODS

- As plume rises it expands, cools and slows down.

- Long rise distances make plume more subject to movement by air currents.

- Because of distance between source and hood, air volumes are usually large.

Position 2

Position 1
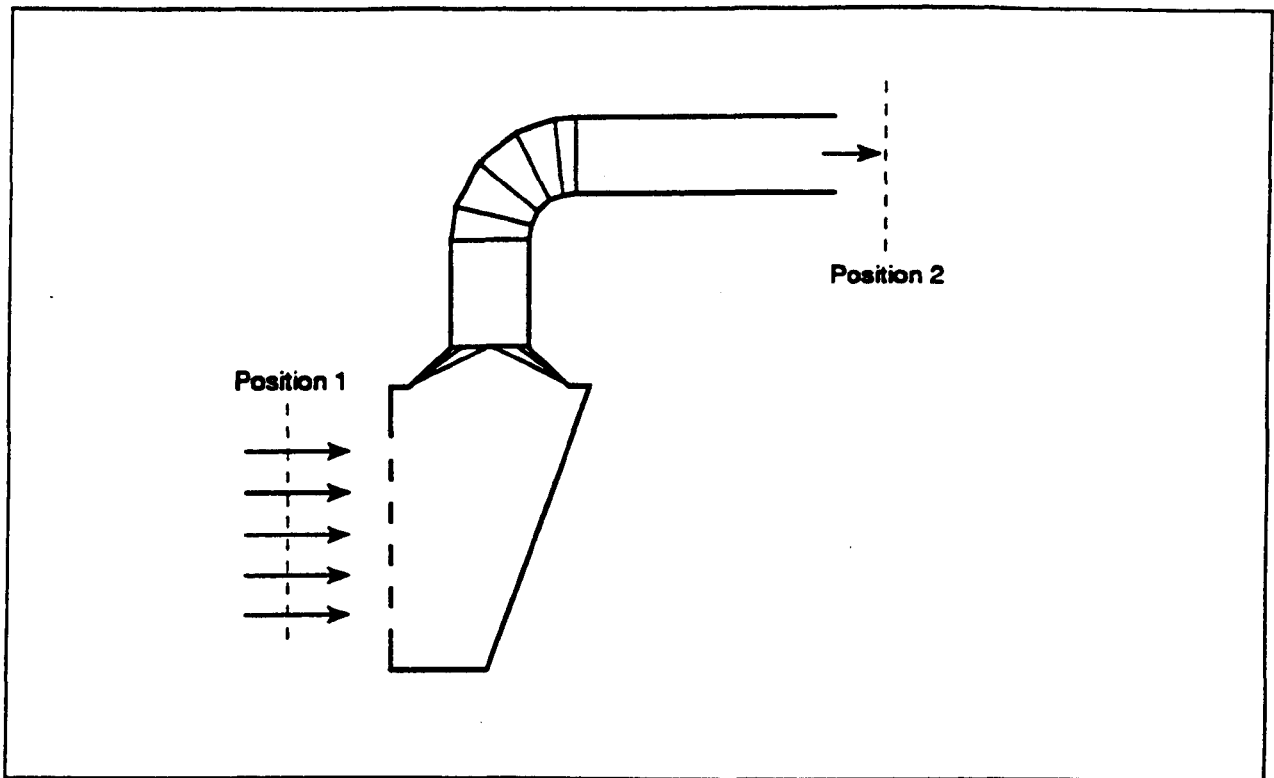
**But there is no air motion
at point 1; therefore:**

$$0 = SP_2 + VP_2$$

or

$$SP_2 = -VP_2$$

Vena Contracta
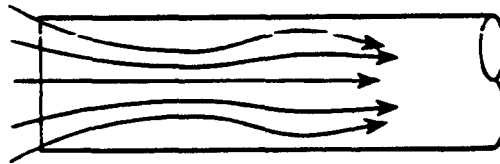
$$Sp_h = -Sp_2 = Vp_2 + h_\theta$$

$$h_\theta = \text{hood entry loss} = F_h Vp$$

$$F_h = \text{hood entry loss factor}$$

HOOD ENTRY COEFFICIENT

$$C_e = (VP/SP_h)^{0.5}$$

$$C_e \text{ AND } F_h \text{ ARE RELATED}$$

$$h_e = [(1 - C_e{}^2)/C_e{}^2] \, VP$$

therefore:

$$F_h = (1 - C_e{}^2)/C_e{}^2$$

$$\text{ESTIMATING HOOD VOLUME}$$

$$Q = VA \quad = 1096.7A \, (VP \, /\rho)$$

$$= 1096.7AC_e \, (SP_h \, /\rho)$$

Q = volumetric flow rate (ft^3 /min)

V = hood face velocity (ft^3 /min)

A = hood inlet area (ft^3)

ρ = air density (lb/ft^3)

DUCT

SYSTEMS

Position 2

Position 1
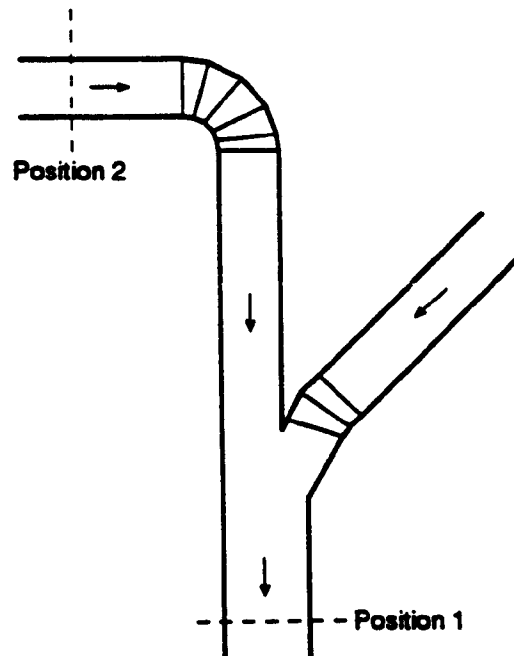
**Because of friction and
non-ideal conversion
between SP and VP...**

$$TP_1 = T P_2 + h_L$$

or

$$SP_1 + VP_1 = SP_2 + VP_2 + h_L$$

**If velocity is constant
between points 1 and 2...**

$$SP_1 = SP_2 + h_L$$

TYPES OF LOSSES

- Frictional losses

- Fitting losses

- Acceleration losses

WAYS TO
ESTIMATE LOSSES

- Equivalent length

- Velocity pressure

- Total pressure

FRICTIONAL LOSSES

$$h_{L1} = H_f L \text{ (VP)}$$

h_{L1} = frictional loss (in. H_2O)

H_f = frictional loss factor (V_p /ft)

L = duct length (ft)

VP = velocity pressure (in. H_2O)

FITTING LOSSES

$$h_{L2} = F \text{ (VP)}$$

h_{L2} = fitting loss (in. H_2O)

F = fitting loss factor (VP)

VP = velocity pressure (in. H_2O)

VELOCITY PRESSURE CALCULATION METHOD

1. Determine duct velocity and VP.

2. Determine hood static pressure.

3. Multiply straight duct length by friction loss factor.

VELOCITY PRESSURE CALCULATION METHOD (cont'd)

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

5. Add results for steps 3 and 4 and multiply by duct VP from step 1.

Slide 3-11

VELOCITY PRESSURE
CALCULATION METHOD (cont'd)

6. Add result from step 5 to the hood
static pressure.

7. Add result from step 6 to any other
losses expressed in inches H_2O.

Slide 3-12

This calculation gives the total
energy required, expressed as
static pressure, to move the gas
volume through the duct segment.

TRANSPORT VELOCITY

That duct velocity required
to prevent build-up

RANGE OF DESIGN VELOCITIES

Contaminent	Design Velocity (ft/min)
Vapors, gases	Usually 1000 - 2000
Fumes	1400 - 2000
Very fine, light dust	2000 - 2500
Dry dust and powders	2500 - 3500
Average industrial dust	3500 - 4000
Heavy dusts	4000 - 4500
Heavy or moist	4500 and up

BUILD-UP CONCERNS

- Increase duct resistance

- Duct pluggage

- Vibration breakup

- Increased weight

Slide 3-16

DISTRIBUTION OF FLOW IN BRANCHED DUCTWORK

All ducts entering a junction must have equal static pressure requirements.

Sp = -2.0 Inches H_2O

Sp = -1.5 Inches H_2O

Sp_h = -1.0 Inch H_2O

Sp_h = -1.0 Inch H_2O

Hood 1

Hood 2

Q_1

Q_2
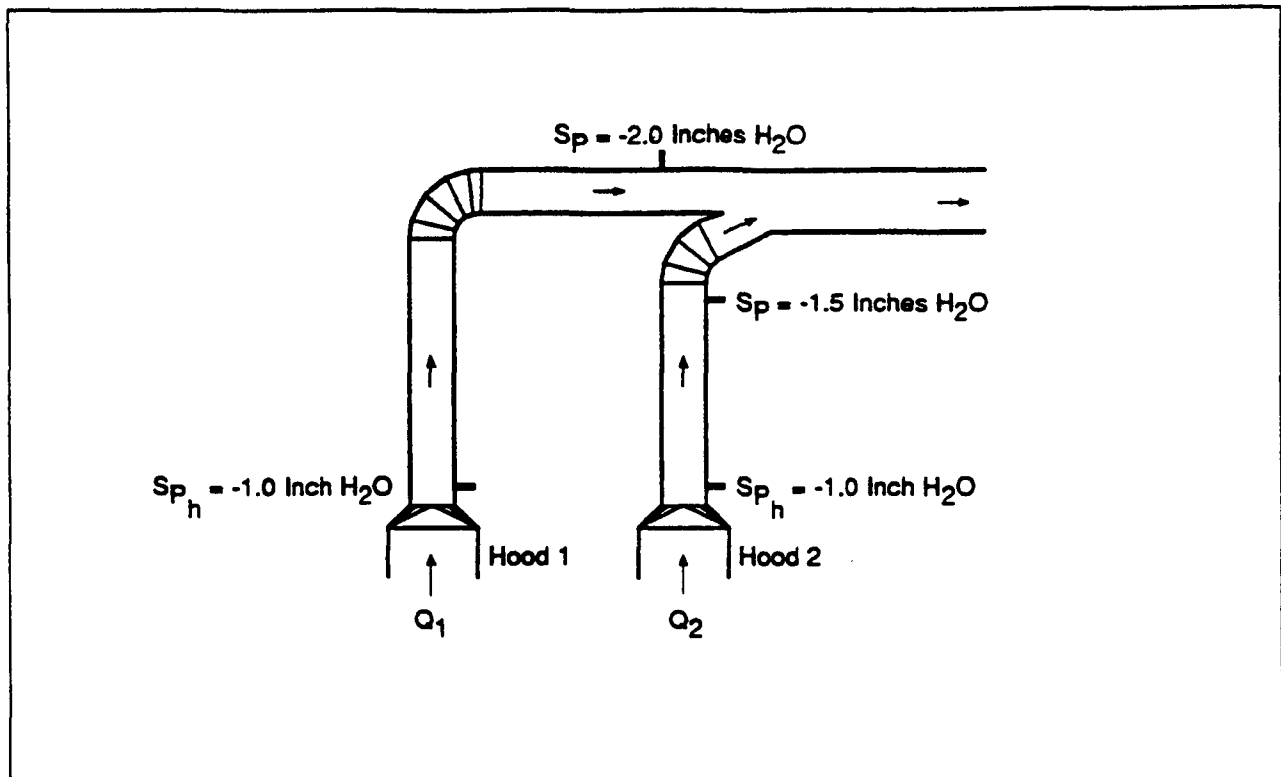
TECHNIQUES TO OBTAIN BALANCE

• **Through design**

• **With dampers**

CHARACTERISTICS OF
BALANCE THROUGH DESIGN

•Volume not easily changed

•Little flexibility

•No unusual erosion or build-up problems

•Final volume may be greater than design

•All branch losses identified

•Must be installed exactly as designed

CHARACTERISTICS OF
BALANCE WITH DAMPERS

•Volumes easily changed

•Great flexibility

•Dampers erode or cause build-up

•Balance achieved with design volume

•May miss branch of greatest resistance

•Moderate variation in installation allowed

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GAS
COOLING
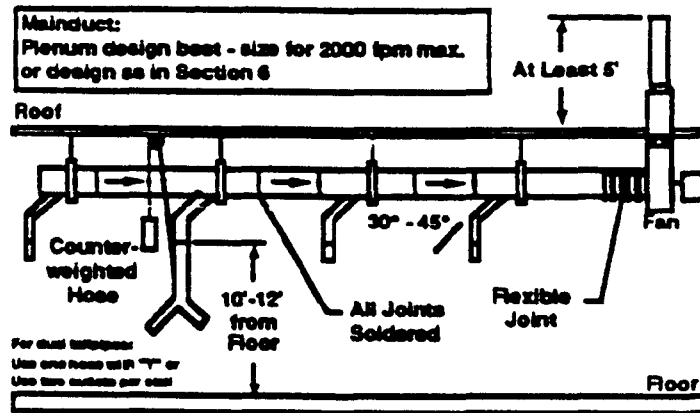
METHOD FOR
COOLING GASES

• Dilution with cooler gases

• Quenching with water

• Natural convection and radiation

• Forced convection

Dilution with Cooler Gases

m_2

T_2 = ambient

Damper

m_3

T_1 = hot

m_3

T_3 = mix
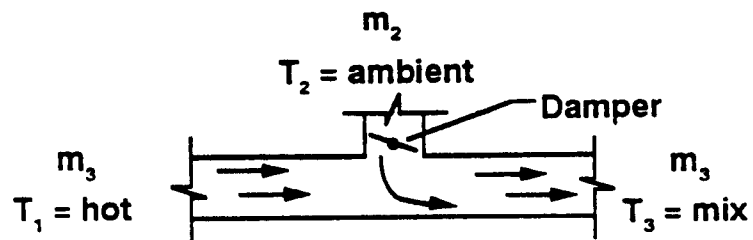
From Heat Balance:

$$m_1 h_{T1} + m_2 h_{T2} = m_3 h_{T3}$$

$$mh = (lb/min)(Btu/lb) = Btu/min$$

$$m_3 = m_1 + m_2$$

PROBLEMS WITH
DILUTION

• **Large air volume**

• **Lack of temperature control**

• **Corrosion potential**
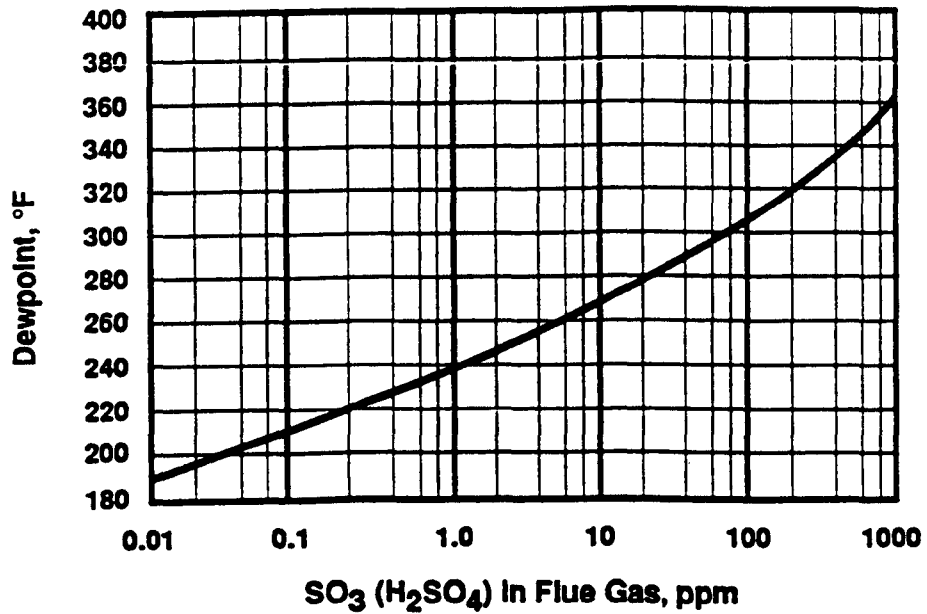
QUENCHING WITH WATER

**Heat loss from gas stream is taken
up by evaporating water...**

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

PROBLEMS WITH QUENCHING

- Water carryover
- Temperature control
- Corrosion

NATURAL CONVECTION AND RADIATION

Heat lost from gas stream is taken up by atmosphere

$$m_{gas} (h_{gas\ in} - h_{gas\ out}) = U A \Delta T$$
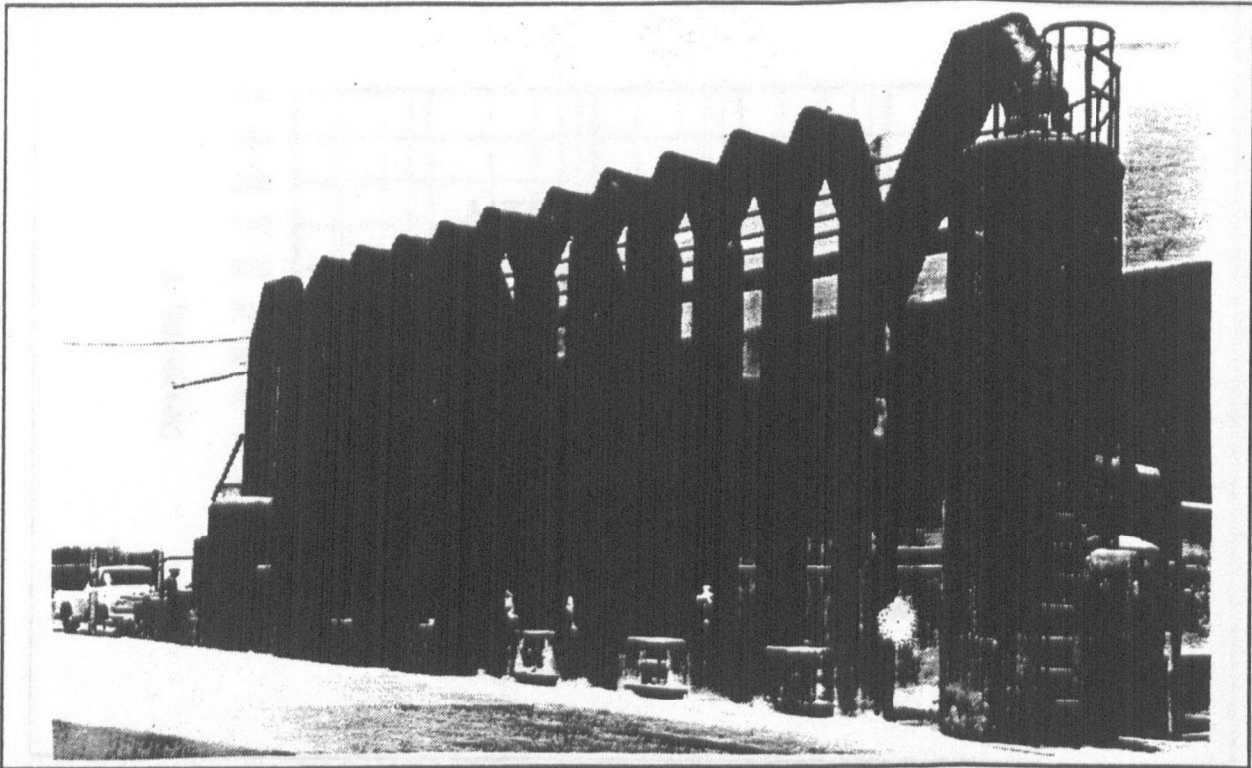
PROBLEMS WITH
CONVECTION AND
RADIATION COOLING

- Large size

- Dust settling

- Lack of temperature control

- Corrosion potential

FAN
SYSTEMS
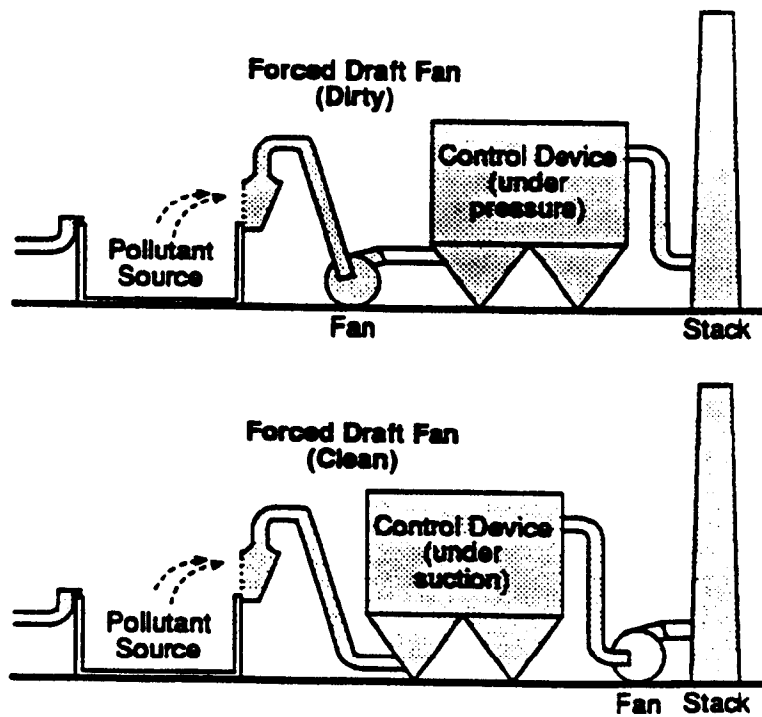
TYPES OF FANS

- Axial

- Centrifugal

- Special

AXIAL FANS

- Propeller

- Tubeaxial

- Vaneaxial
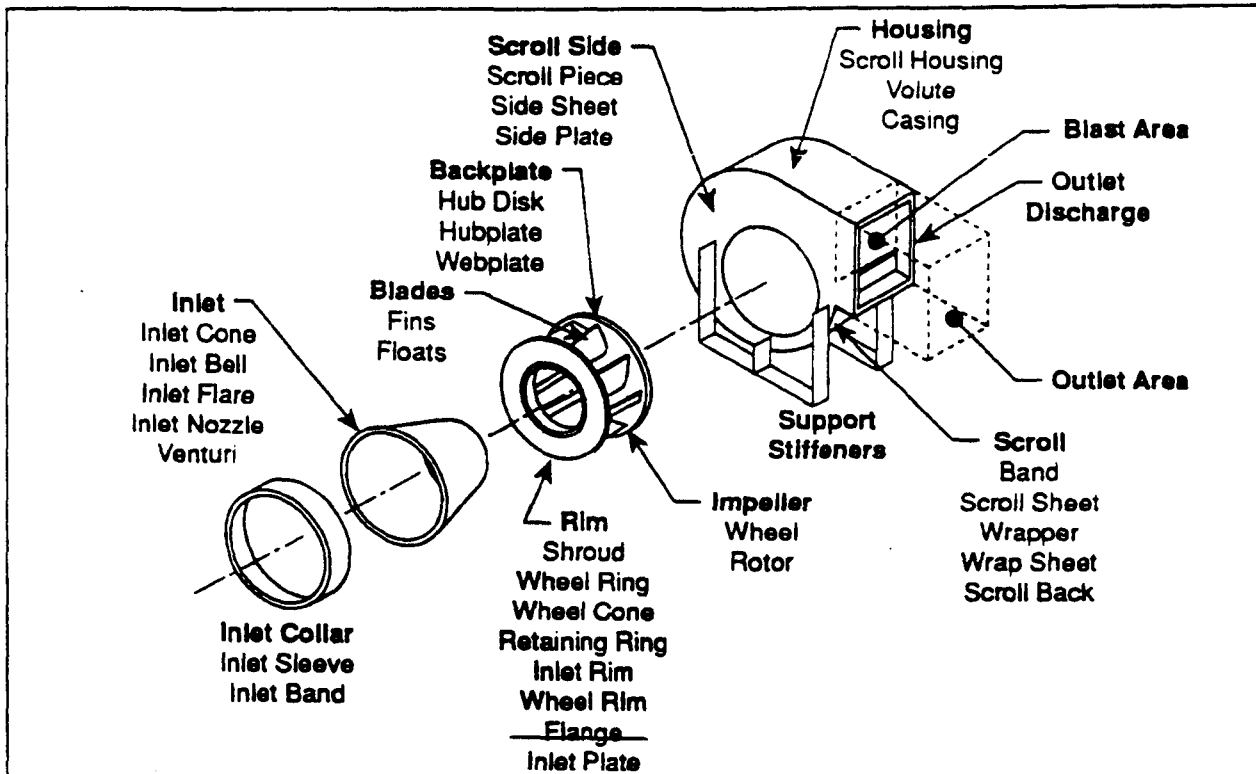
CENTRIFUGAL

- **Forward curved**

- **Radial**

- **Backward inclined**
  - **standard blade**
  - **airfoil blade**

Forward Curved
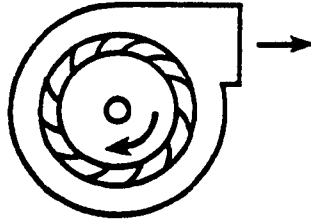
- **Has 24-64 shallow blades**
- **Efficiency less than backward inclined**
- **Smallest of all centrifugal fans**
- **Operates at lowest speed**

Fan Wheel — Fan Housing

Forward Curved Fan Blades

Static Pressure

Efficiency

Horsepower

Static Pressure, Inches of Water

Motor Horsepower, Hp

Efficiency

Gas Flow, SCF/min.
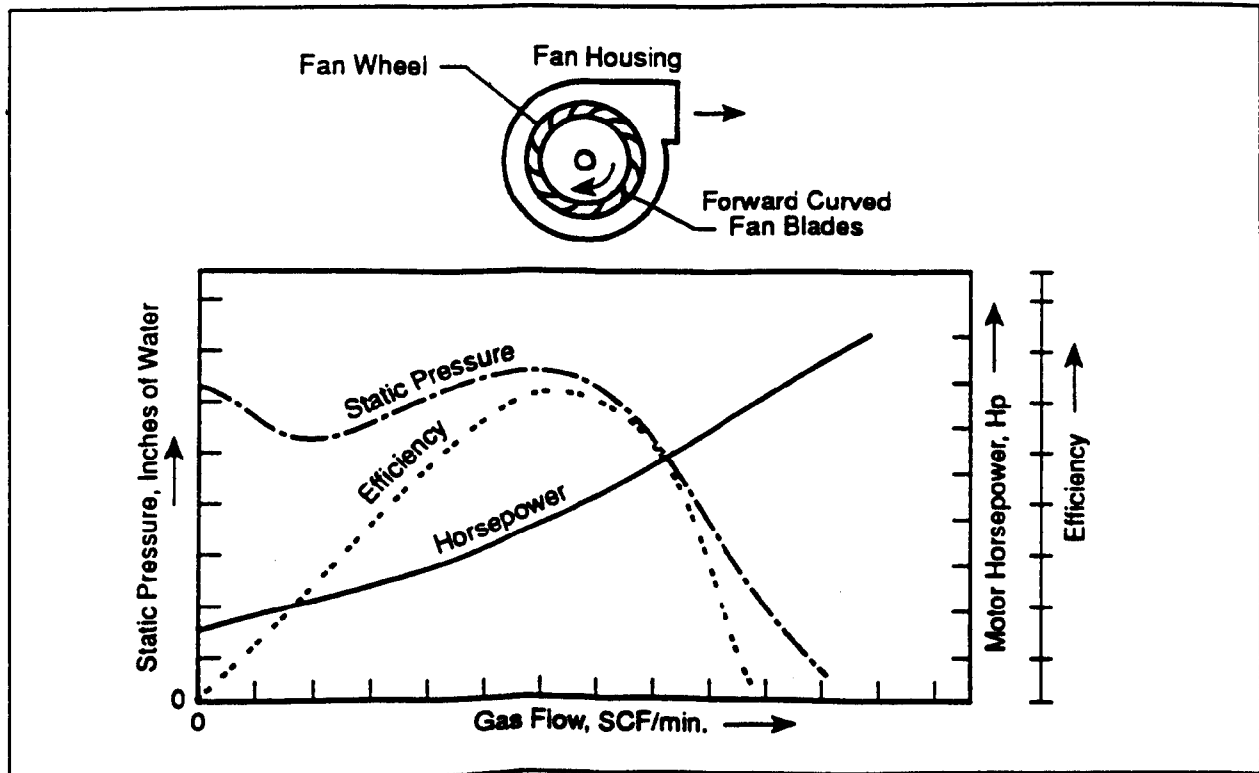
Radial
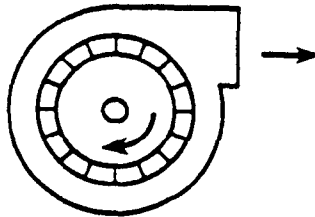
- Has 6-10 blades
- Least efficient
- Narrowest of all centrifugal fans
- Operates at medium speed
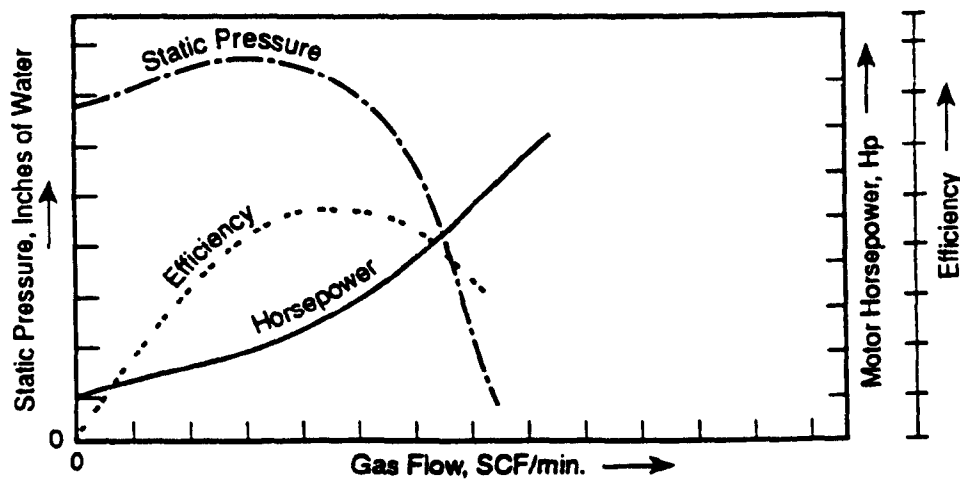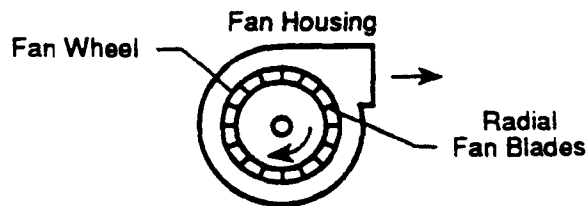
Backward Inclined Airfoil Blade
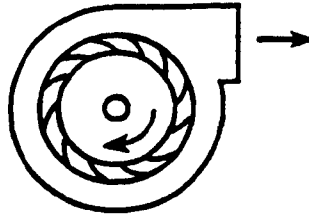
- **Has 9-16 blades**
- **Most efficient**
- **Operates at highest speed**

Fan Housing

Fan Wheel

Backward Curved Fan Blades

Static Pressure

Horsepower

Efficiency

Static Pressure, Inches of Water

Motor Horsepower, Hp

Efficiency

Gas Flow. SCF/min.
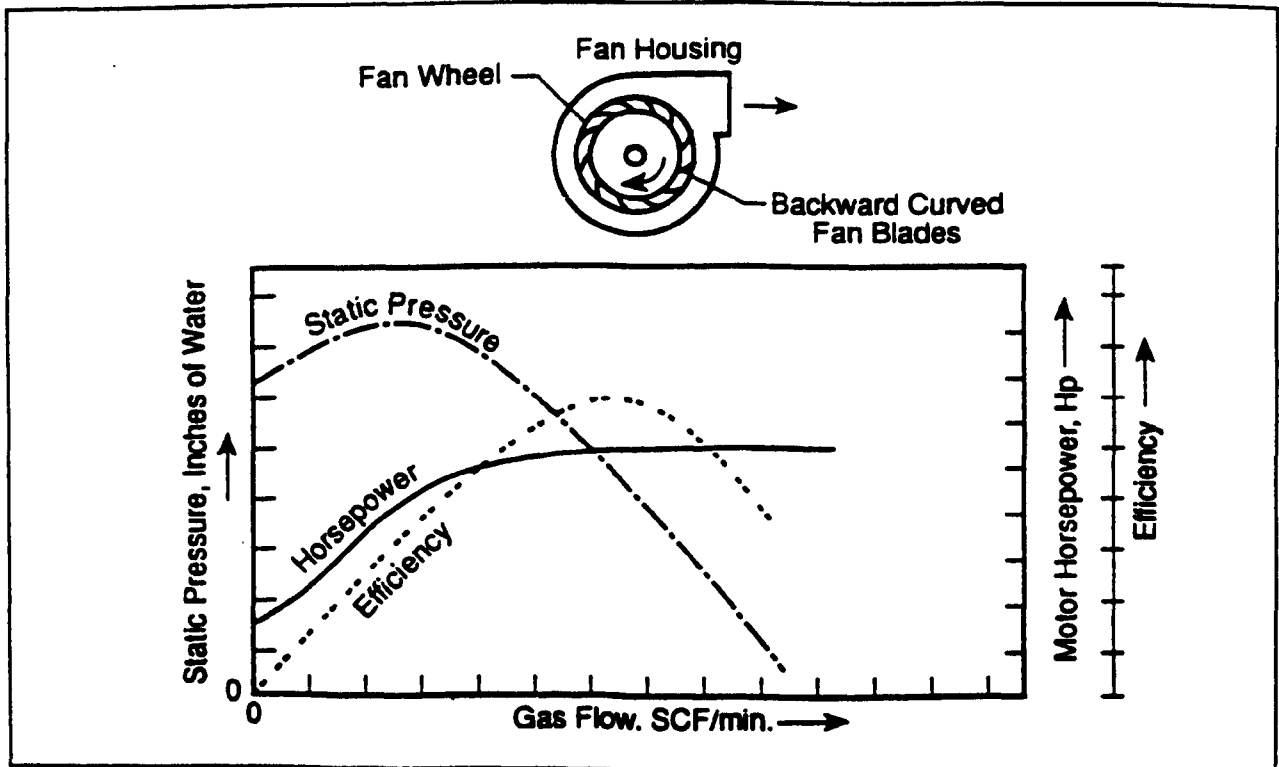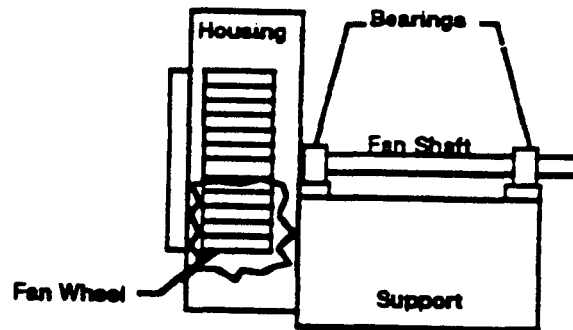
Slide 5-13

ARR. 1 SWSI for belt drive or direct
connection. Impeller overhung.
Two bearings on base.

Slide 5-14

FAN LAWS

$$Q_1 = Q_2 \, (size_2 / size_1)^3 \, (rpm_2 / rpm_1)$$

$$P_2 = P_1 \, (size_2 / size_1)^2 \, (rpm_2 / rpm_1)^2 \, (\rho_2 \rho_1)$$

$$bhp_2 = bhp_1 \, (size_2 / size_1)^5 \, (rpm^2 / rpm^1)^3 \, (\rho_2 / \rho_1)$$

Slide 5-15
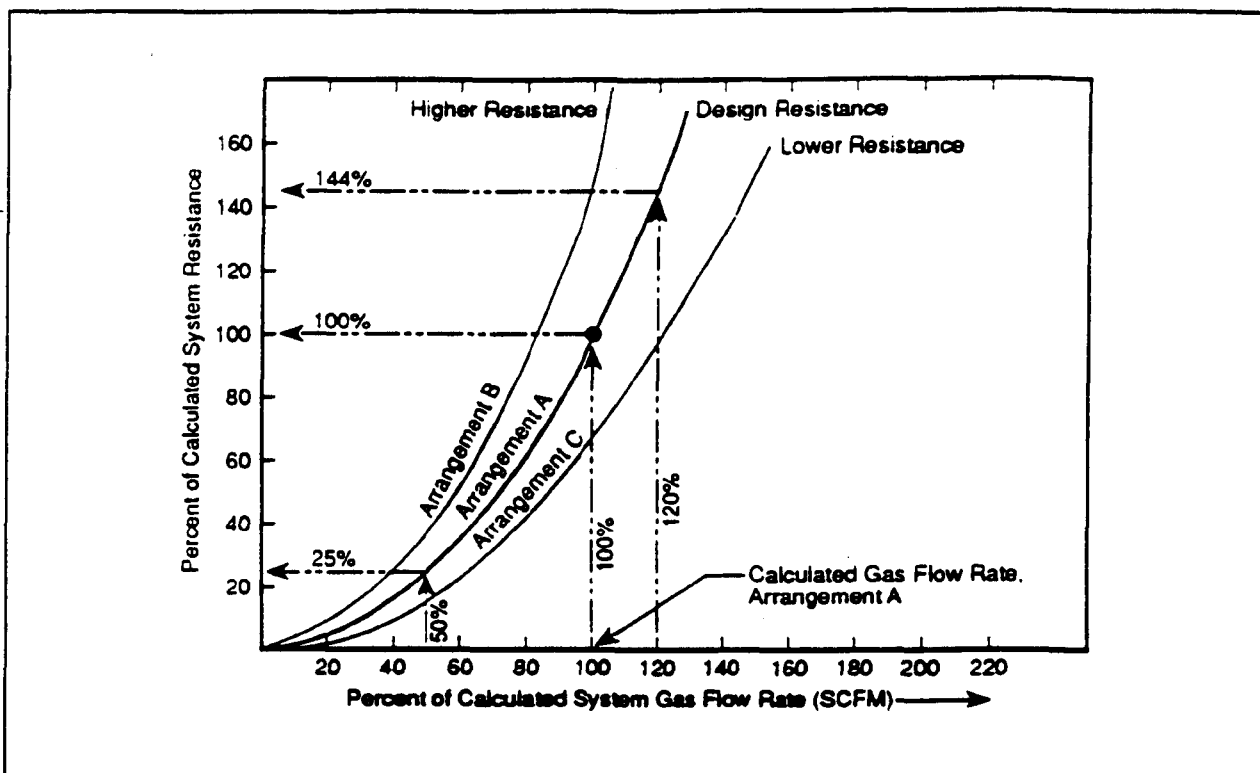
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Slide 5-16

A-52
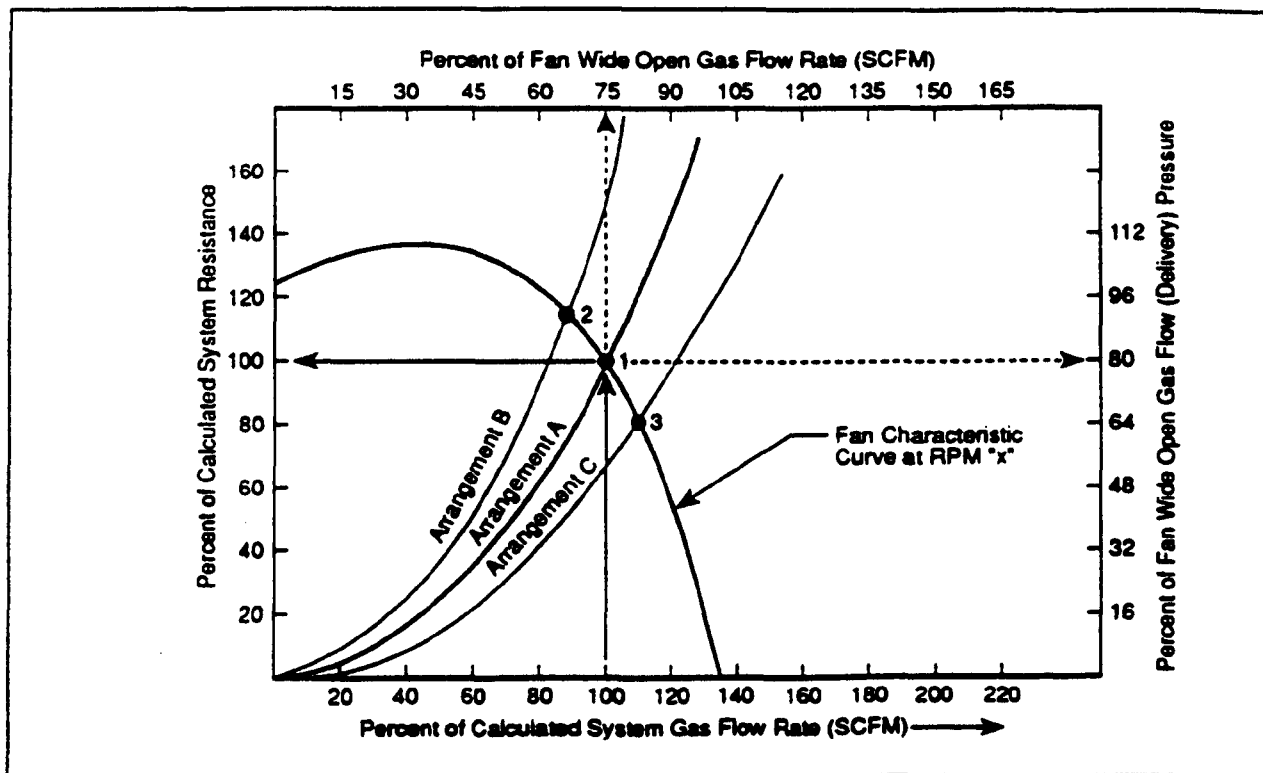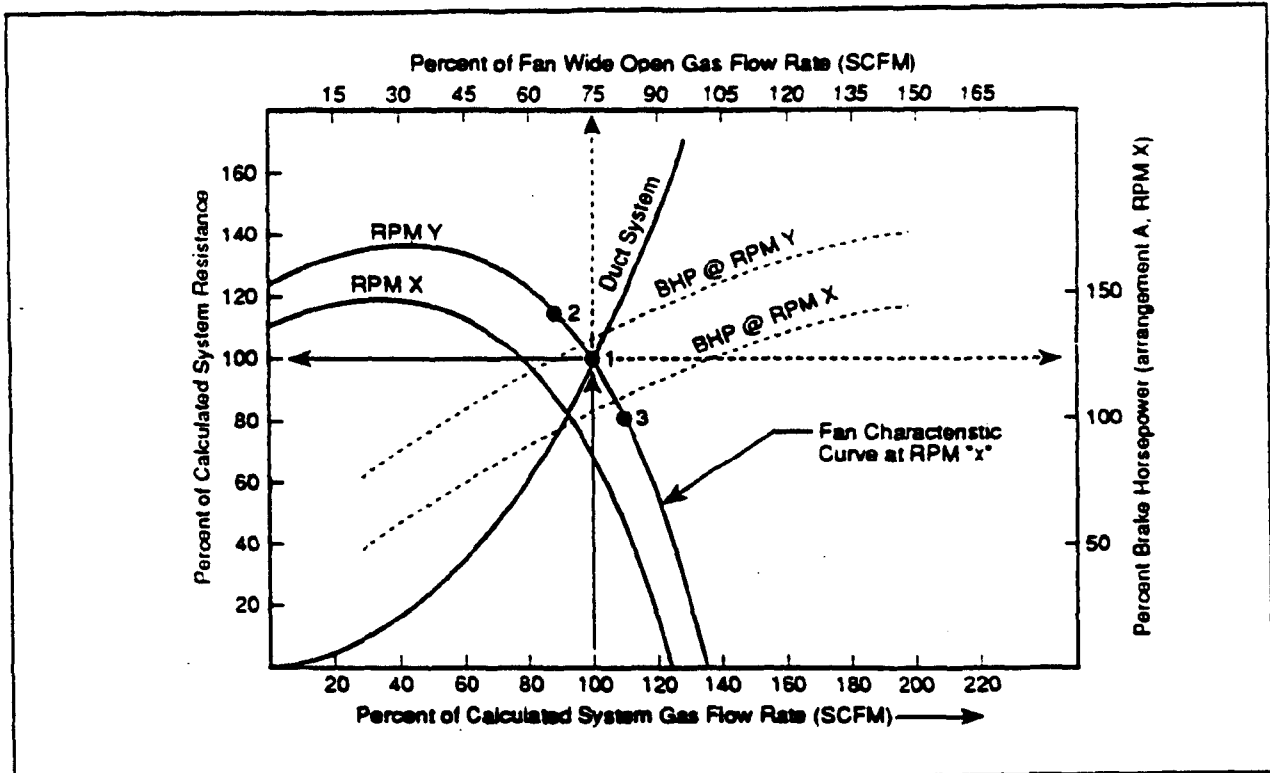
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Slide 5-18

A-53
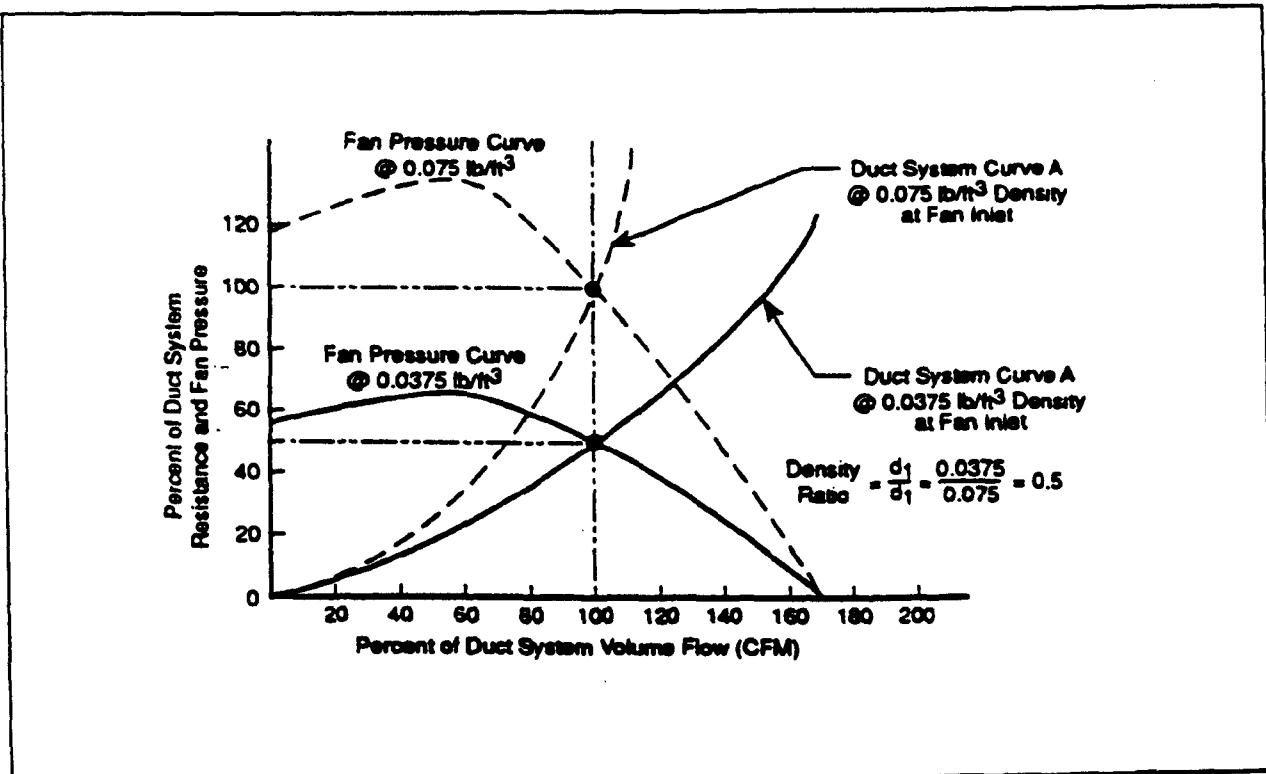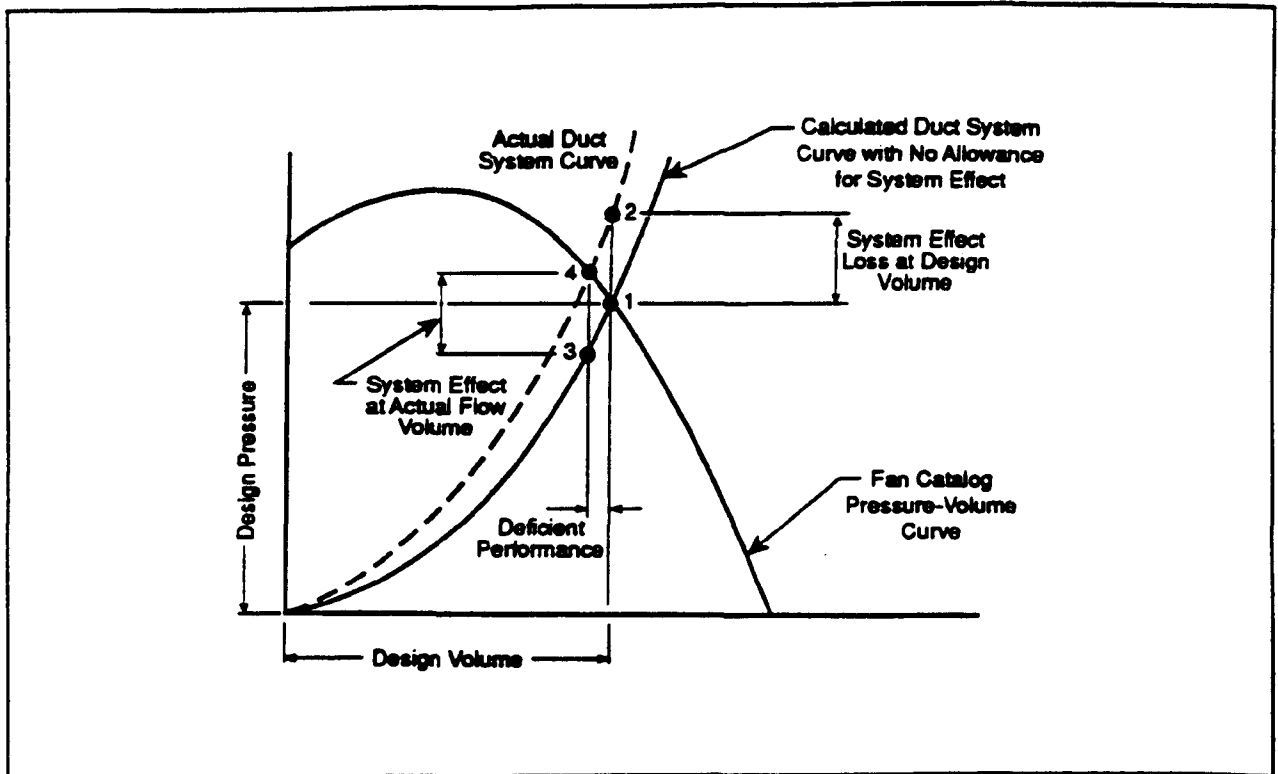
FAN SELECTION

1. Determine volume required at actual conditions

2. Calculate fan static pressure (FSP) at actual conditions— $FSP = SP_{out} - SP_{in} - VP_{in}$

3. Correct FSP to equivalent value for standard air—

$$FSP_e = FSP_A (0.075/\rho_a)$$

FAN SELECTION

4. **Enter ratings table at actual volume and equivalent FSP**

5. **Determine rpm and bhp**

6. **Correct bhp to actual conditions**

    $bhp_a = bhp_e(\rho_a/0.075)$

MEASUREMENT OF
VENTILATION SYSTEM
PARAMETERS

MEASUREMENT PORTS
FOR PORTABLE
INSPECTION EQUIPMENT

Slide 6 - 3
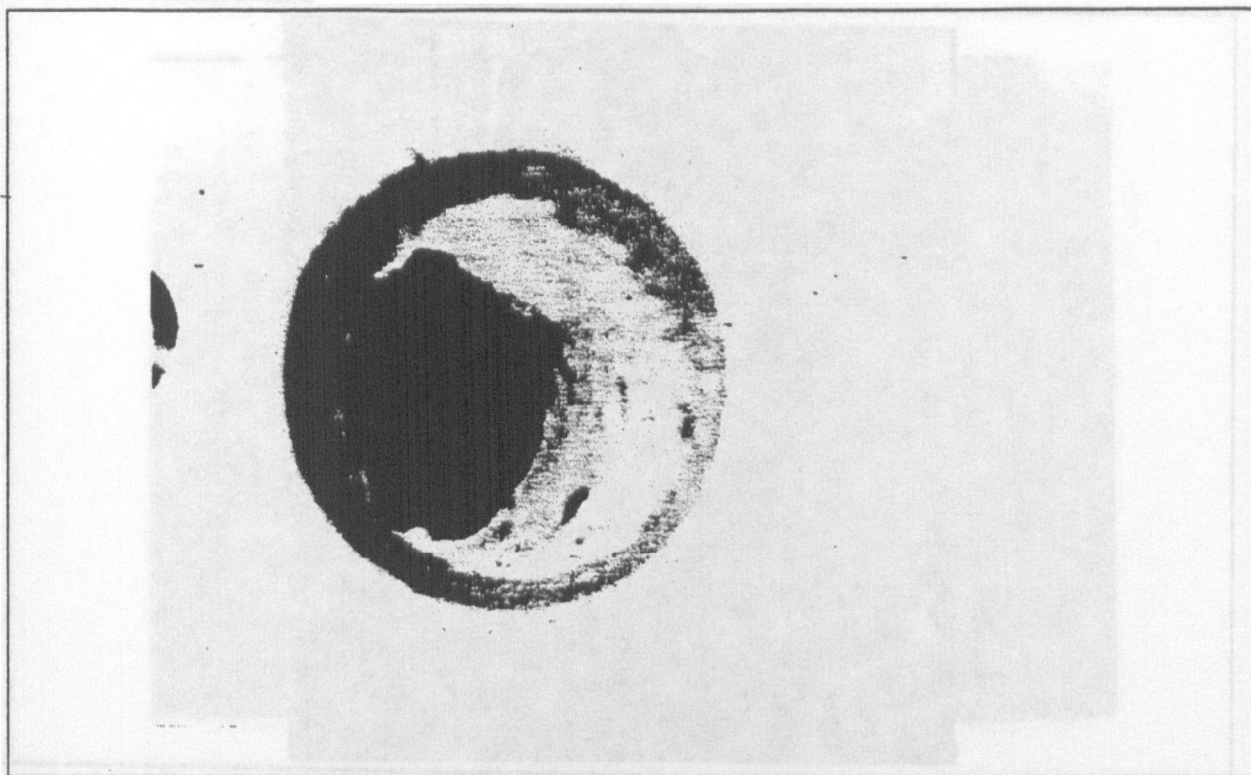
Slide 6 - 4
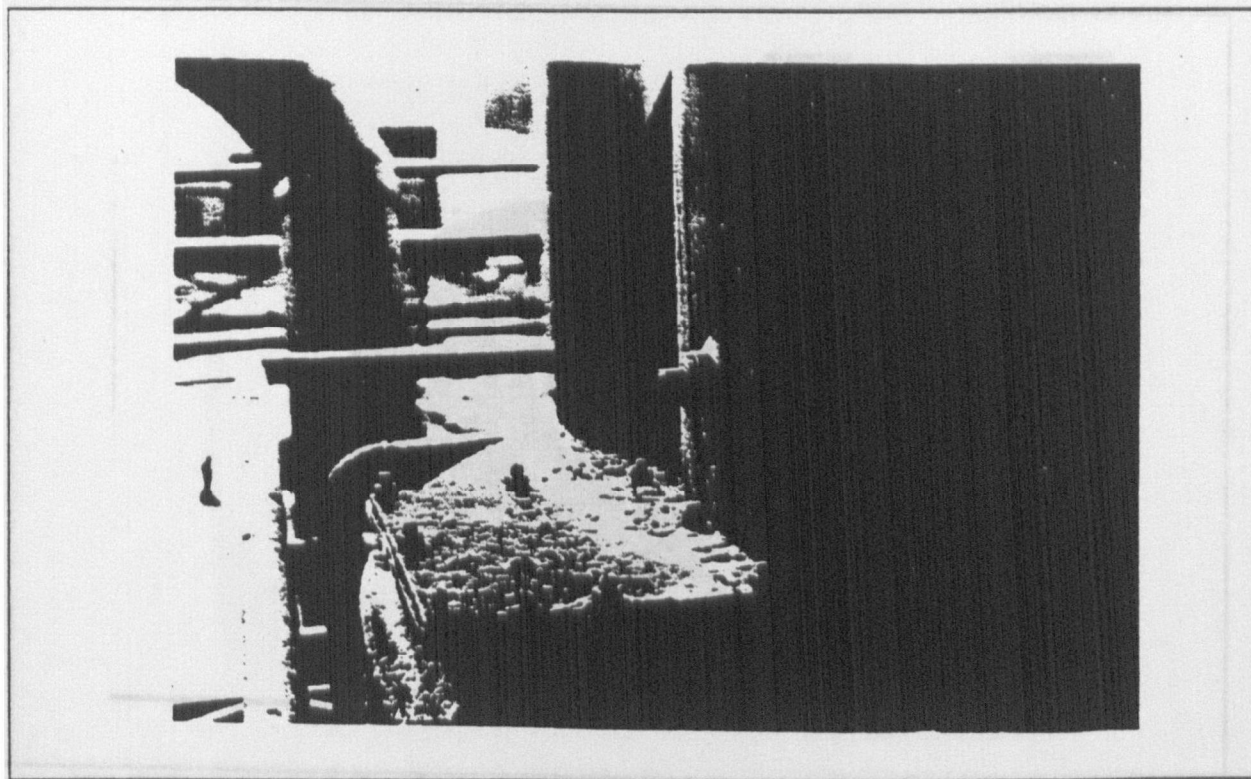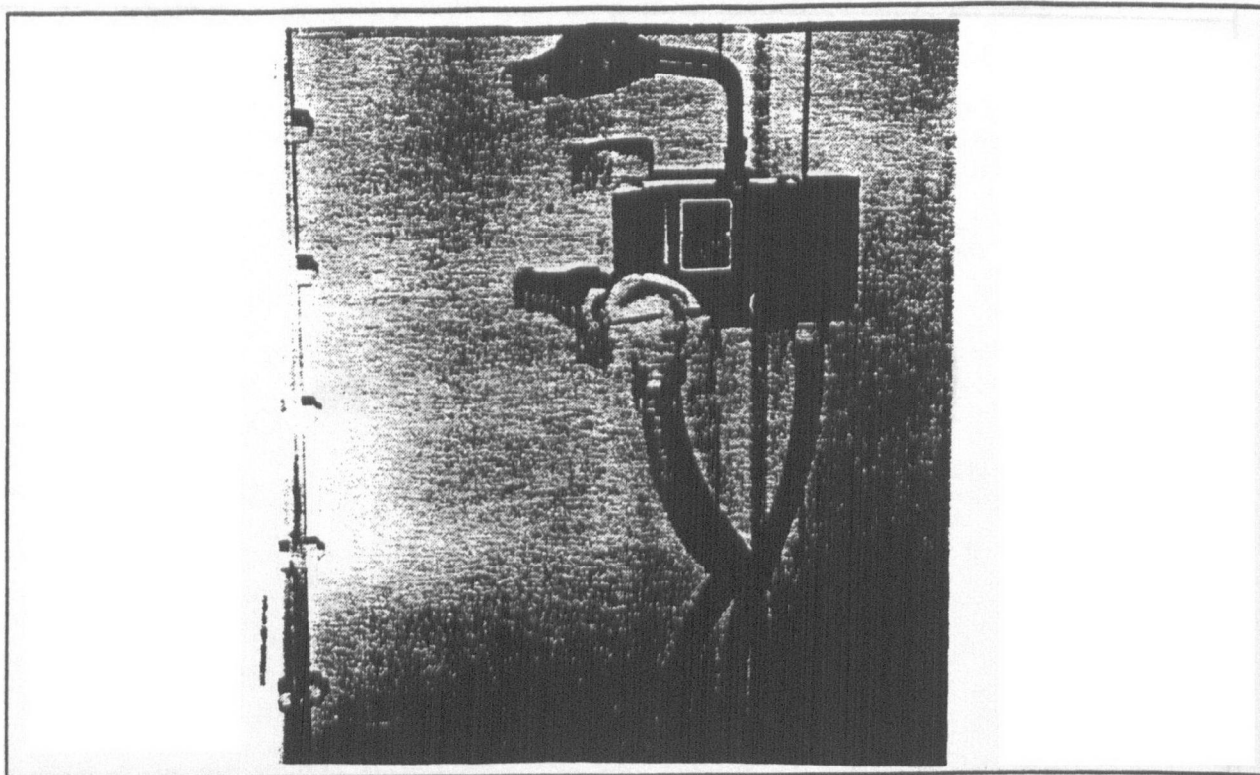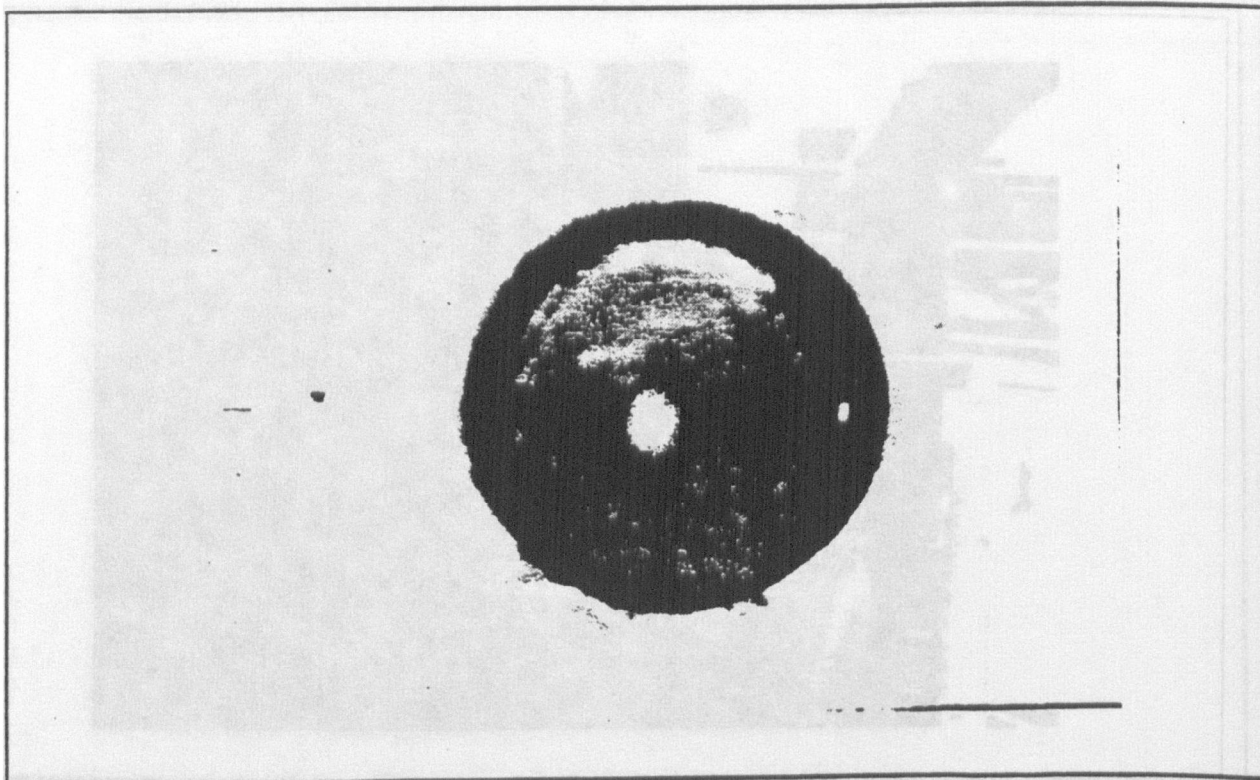
Slide 6 - 5

Slide 6 - 6

Slide 6 - 7
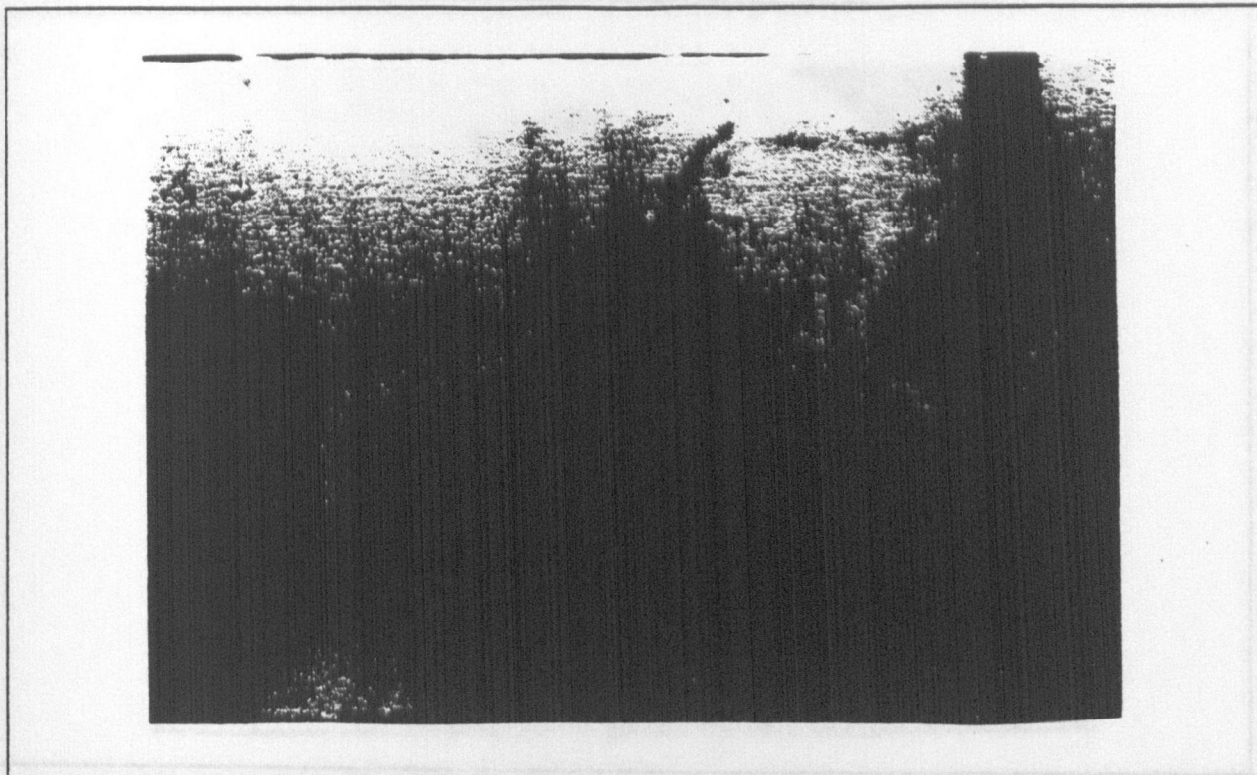
Slide 6 - 8
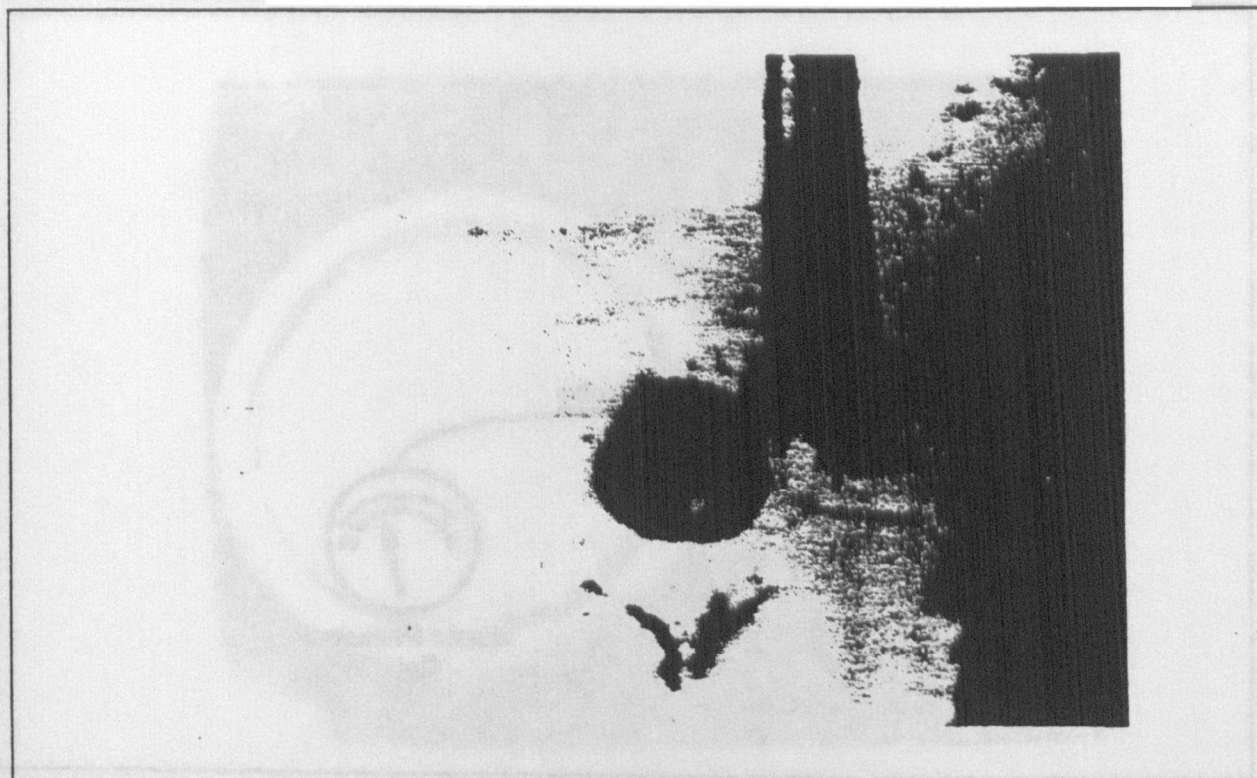
MEASUREMENT OF PRESSURE

- **Manometer**

- **Differential pressure gauge**
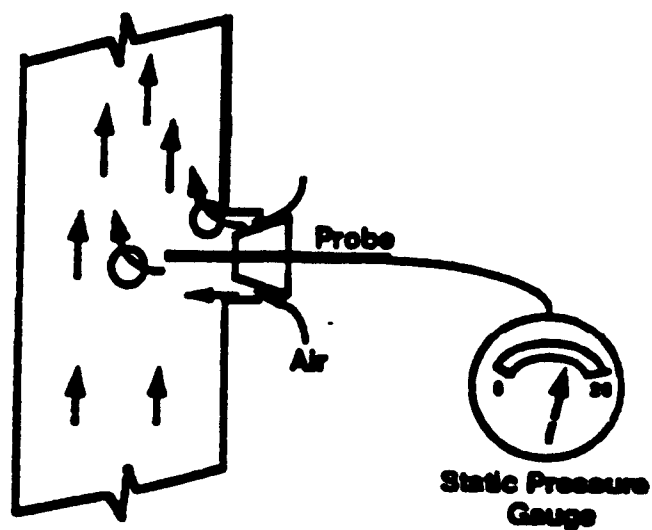
Slide 6 - 11
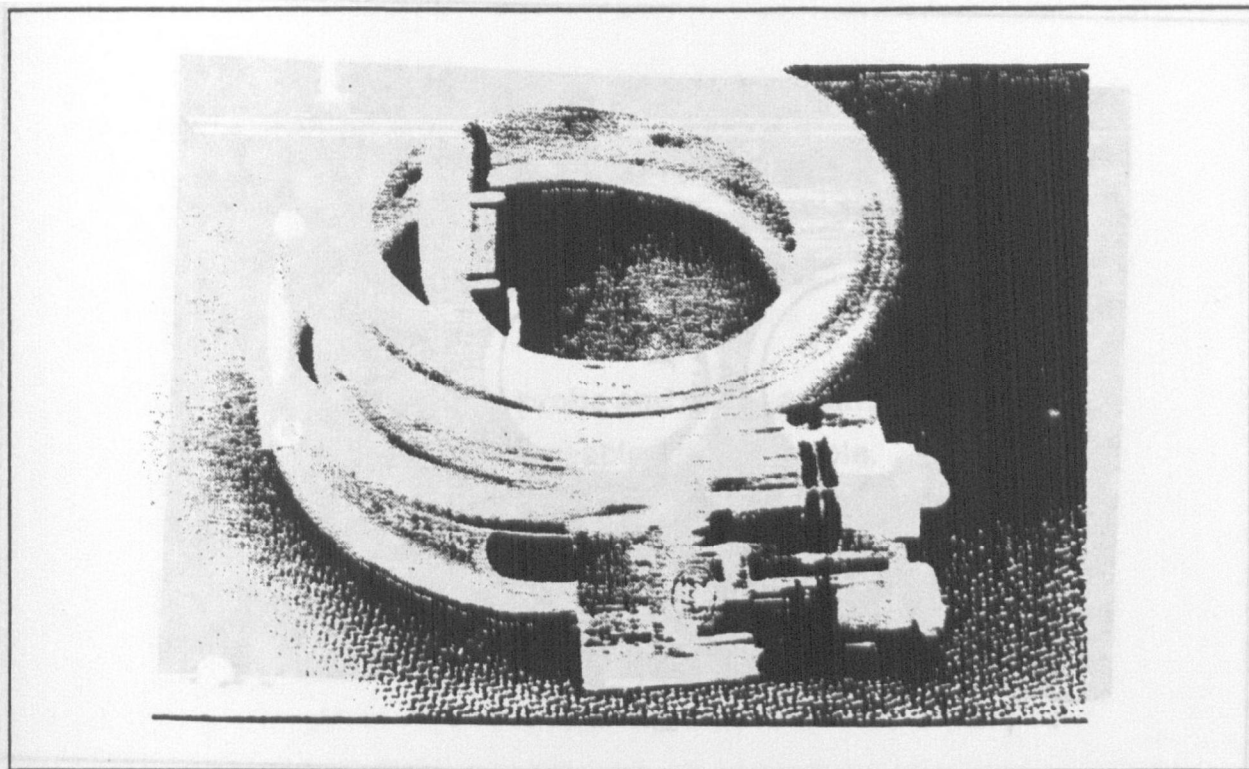
Slide 6 - 12
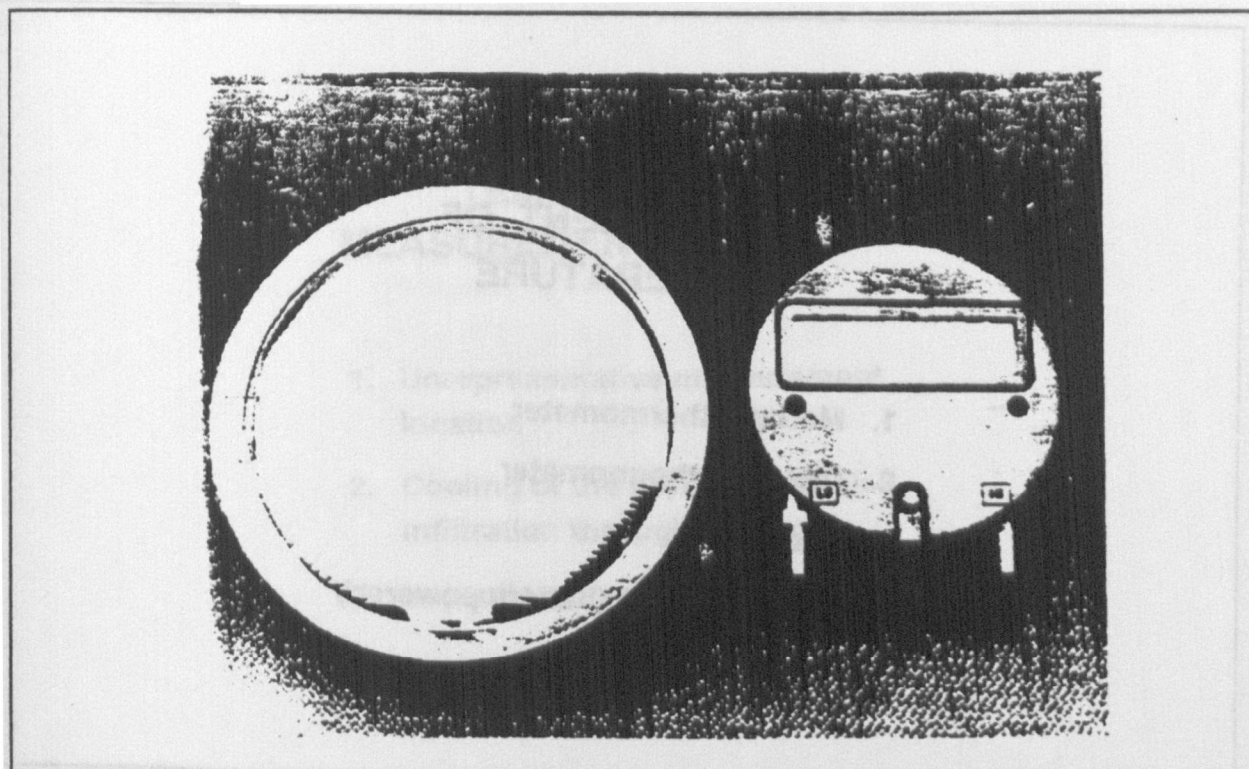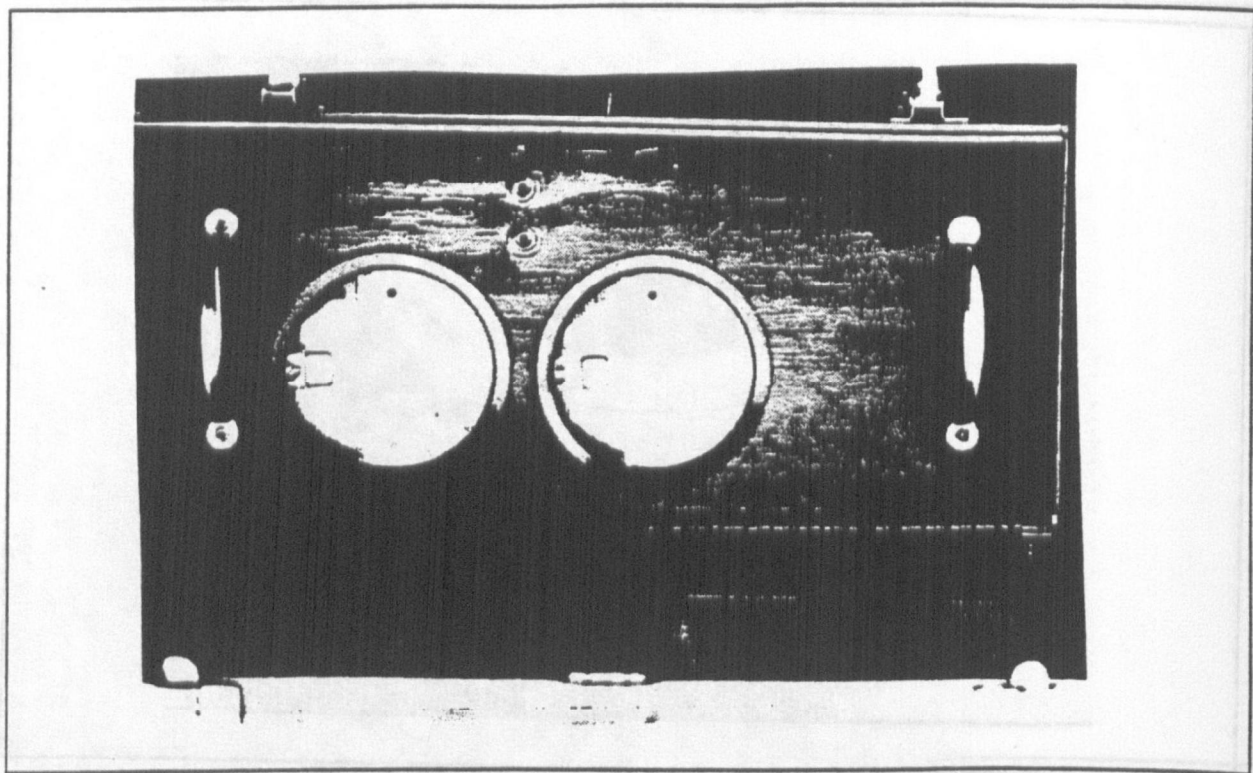
TECHNIQUES FOR THE MEASUREMENT OF TEMPERATURE

1. Mercury thermometer

2. Dial-type thermometer

3. Thermister

4. Thermocouple (battery powered)

THERMOCOUPLES

1. Calibrate probe and meter against an NIST traceable thermocouple.

2. Check ice point and boiling point values prior to each day.

TEMPERATURE MEASUREMENT ERRORS

1. Unrepresentative measurement location

2. Cooling of the probe due to air infiltration through the port

3. Impaction of water droplets

Sandling Disc
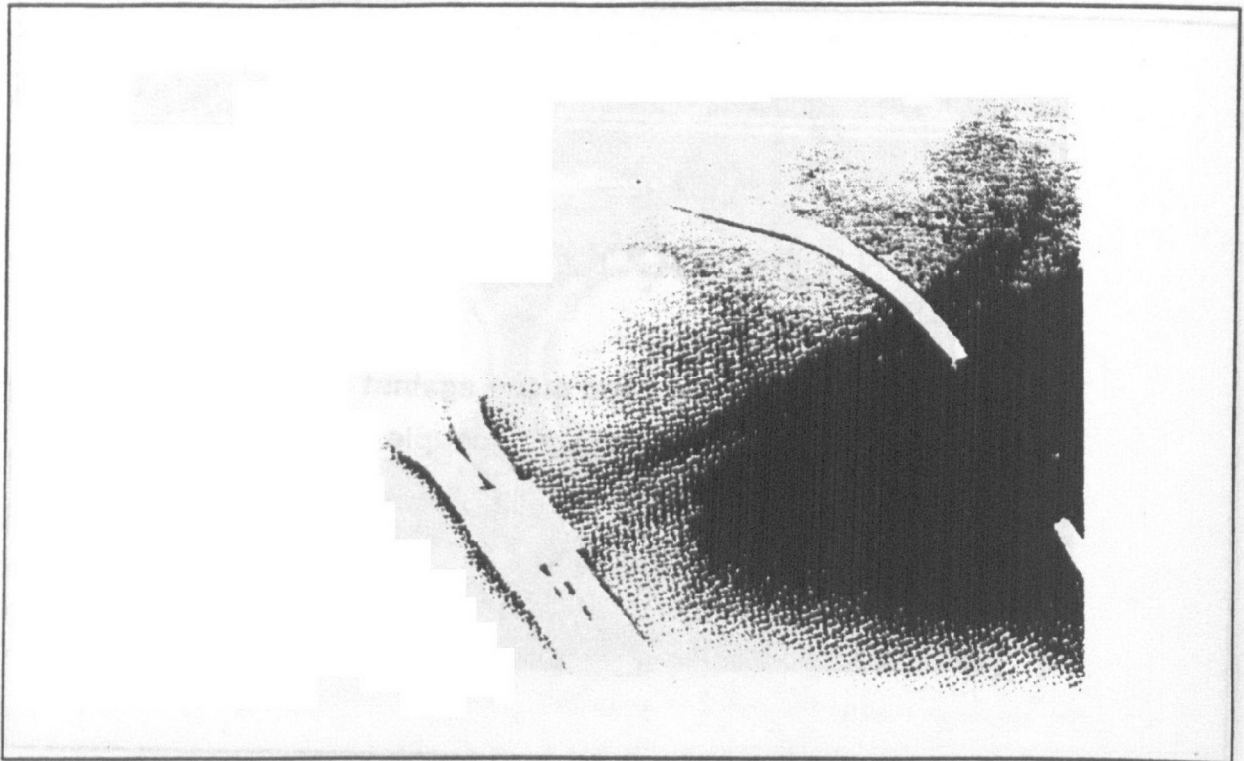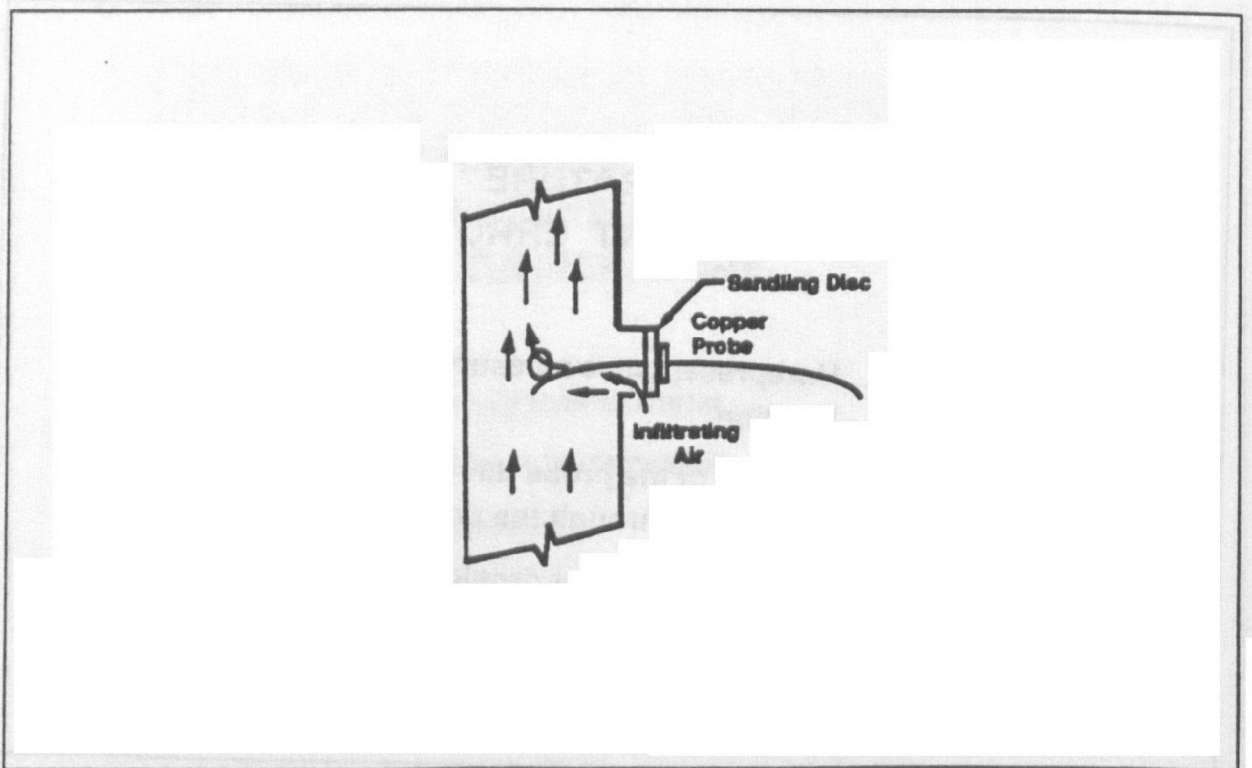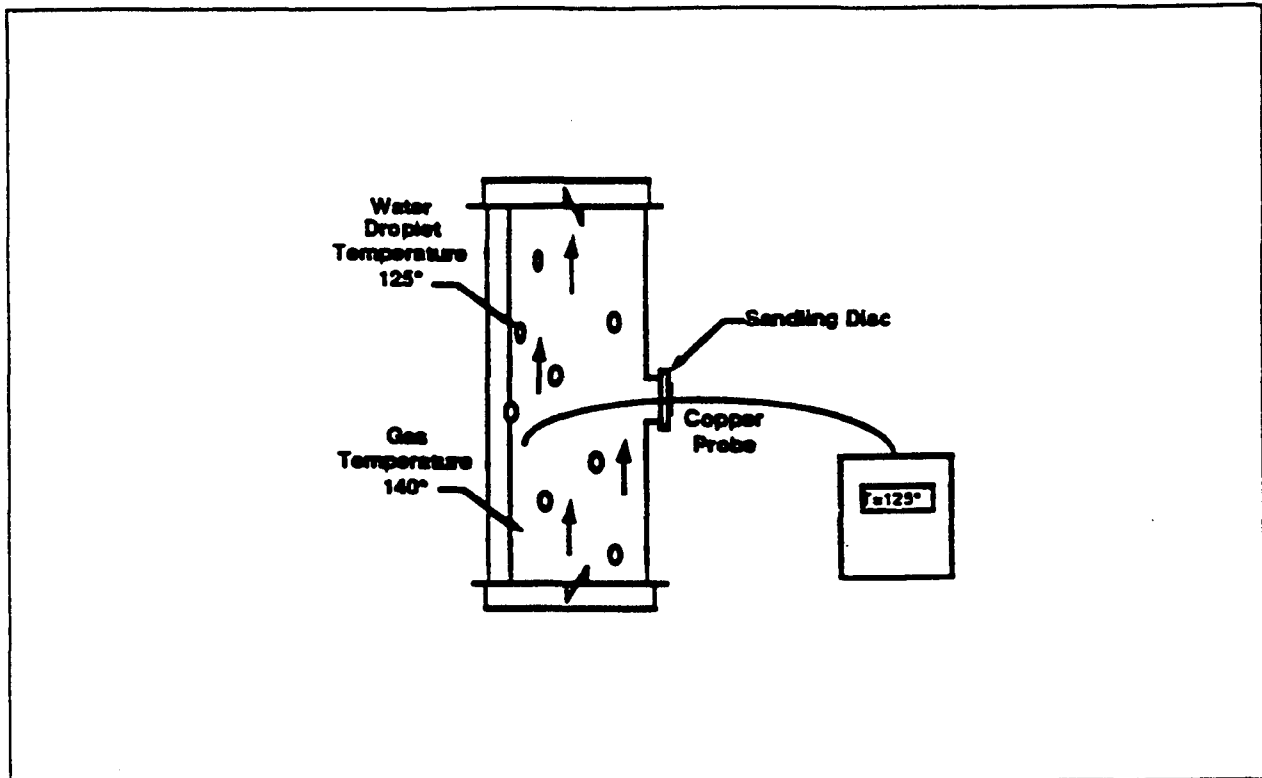
Copper
Probe

Infiltrating
Air

MEASUREMENT OF
GAS FLOW RATE

40CFR60 APPENDIX A
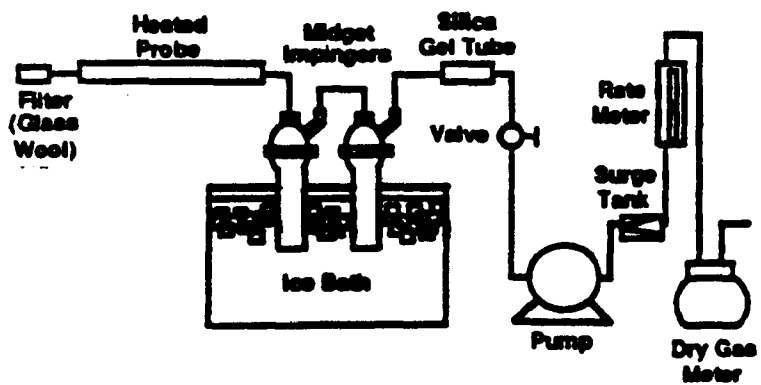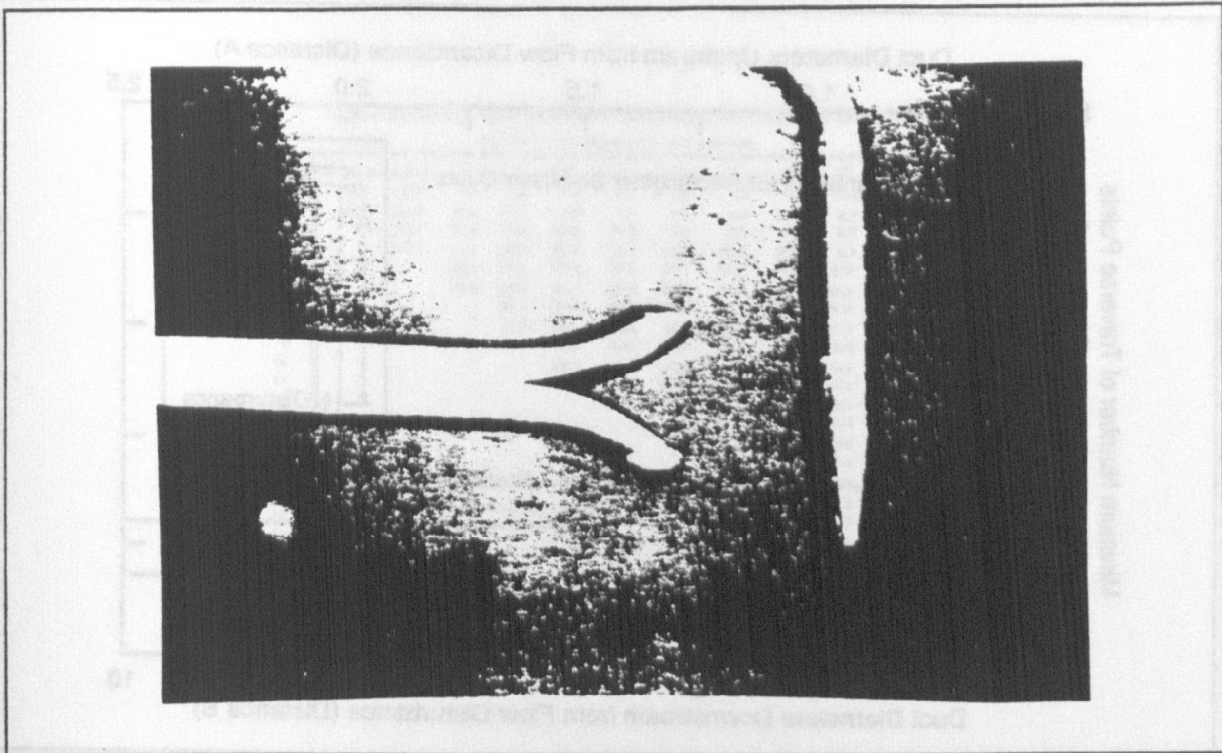
METHODS 1 - 4
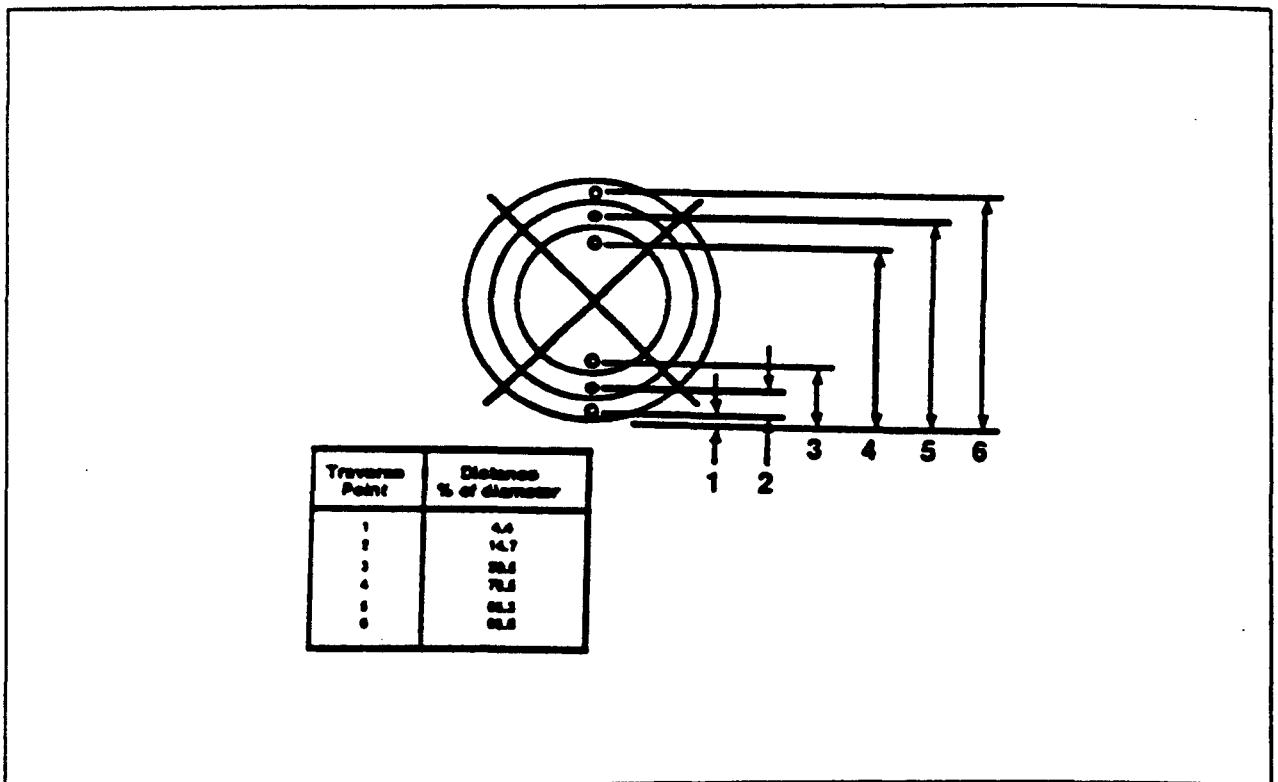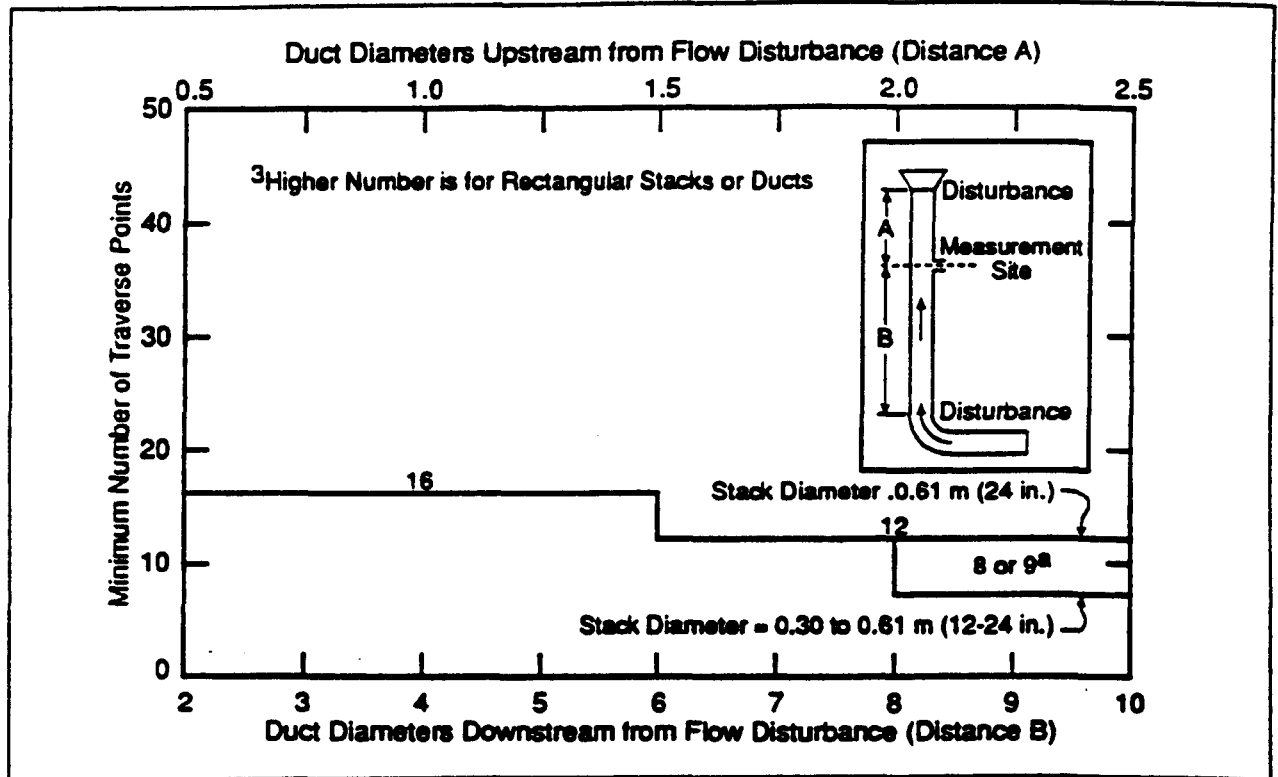
$$(C_p)_s = (C_p)_{std}(\Delta P_{std}/\Delta P_s)^{0.5}$$

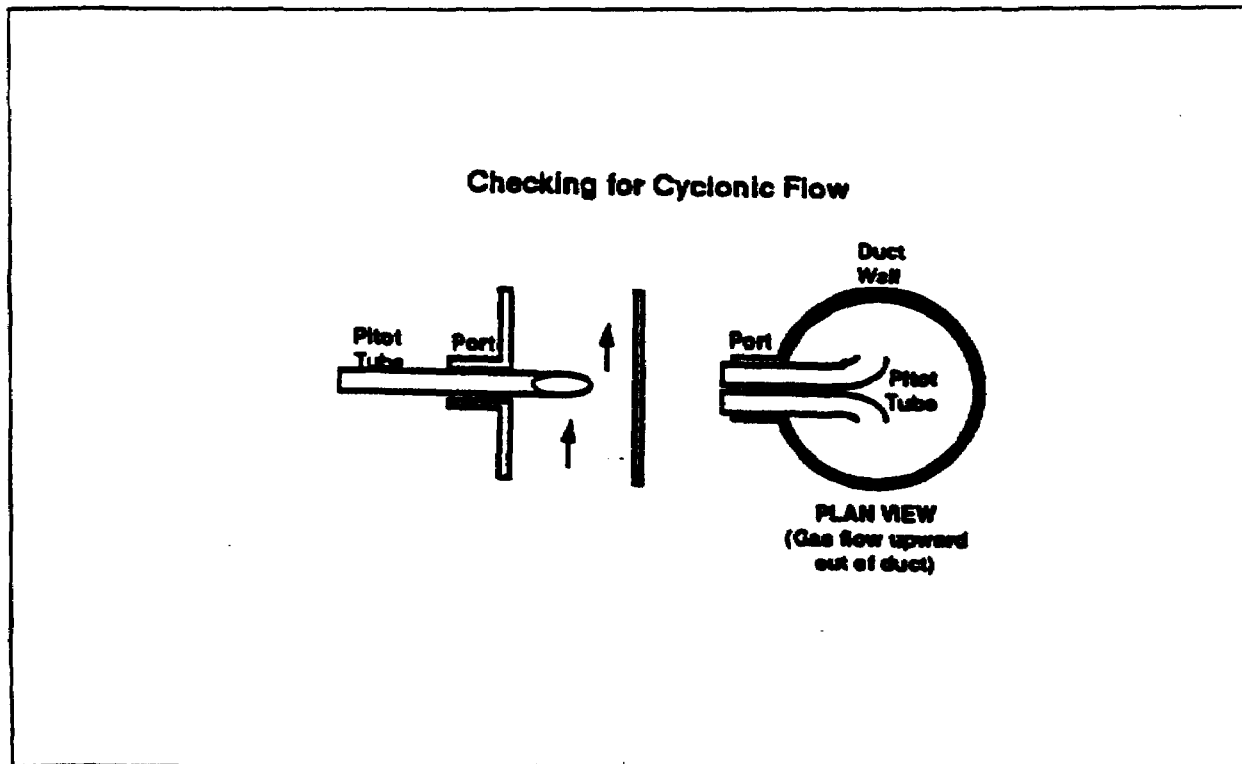$$(A \text{ or } B) = E \left| (C_p)_s - (C_p)_{avg}(A \text{ or } B) \right|$$

$\leftarrow$ must be ≤ 0.01

$$\left| (C_p)_{avg}(A) - (C_p)_{avg}(B) \right| \leftarrow \text{must be} \leq 0.01$$

Duct Diameters Upstream from Flow Disturbance (Distance A)

Minimum Number of Traverse Points

[3]Higher Number is for Rectangular Stacks or Ducts

16

Stack Diameter .0.61 m (24 in.)

12

8 or 9[a]

Stack Diameter = 0.30 to 0.61 m (12-24 in.)

Duct Diameters Downstream from Flow Disturbance (Distance B)

LOCATION OF TRAVERSE POINTS IN CIRCULAR STACKS

Checking for Cyclonic Flow

MEASUREMENT OF
FAN SPEED

1. **Manual tachometer**

2. **Phototachometer**

3. **Strobetachometer**

4. **Sheeve ratio calculation**
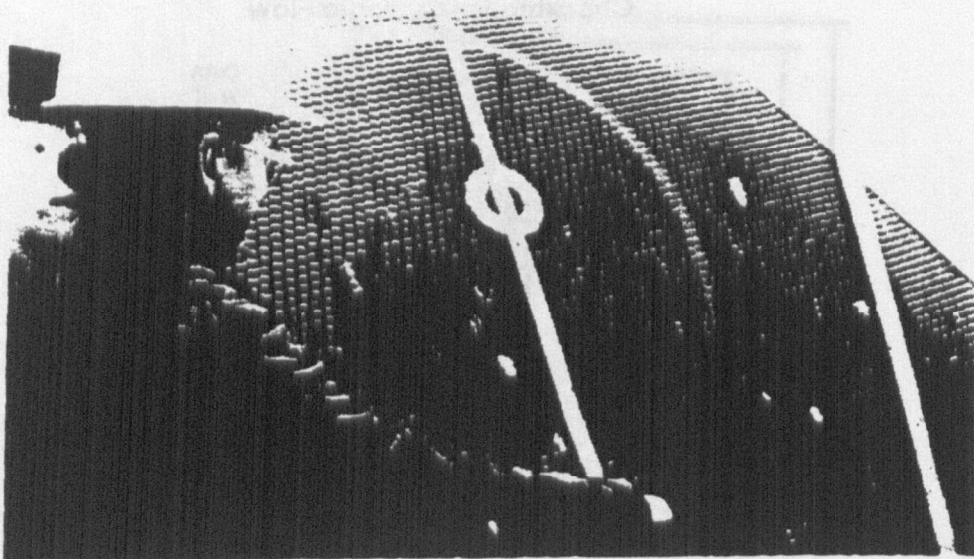
RPM = MS(MD/FD)
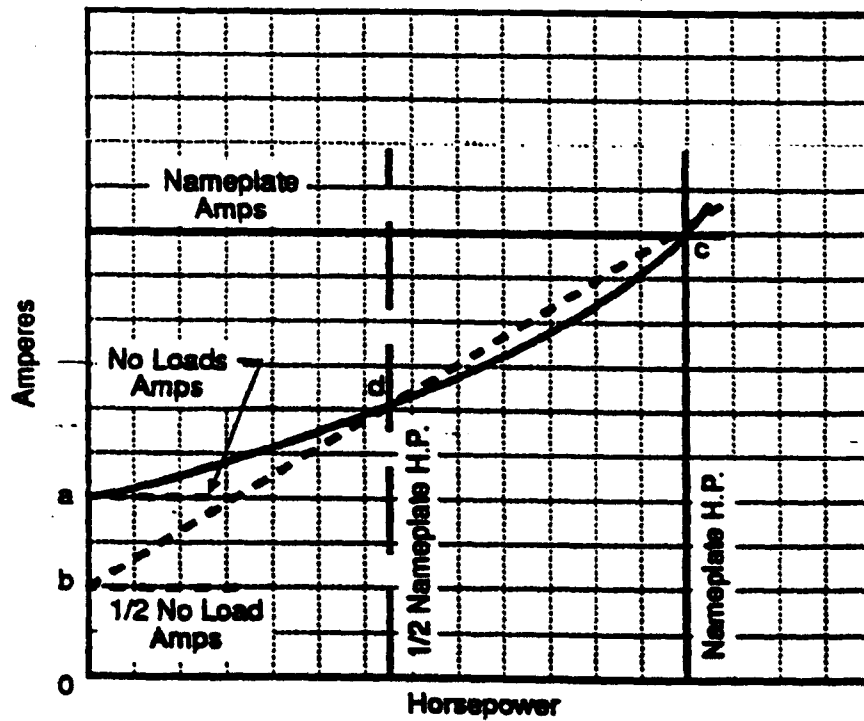
$$bhp = 3^{0.5} (Volts) (Amps) (Power Factor) (Efficiency) / 746$$

Three - Phase   Three - Wire Amp.
Volt Meter Connections
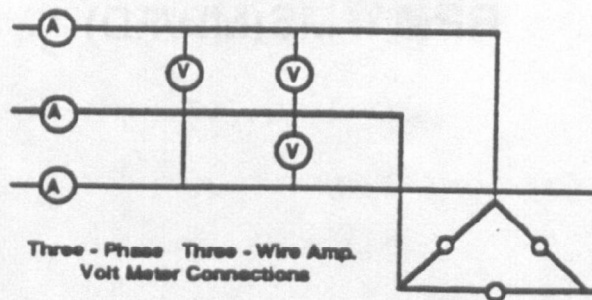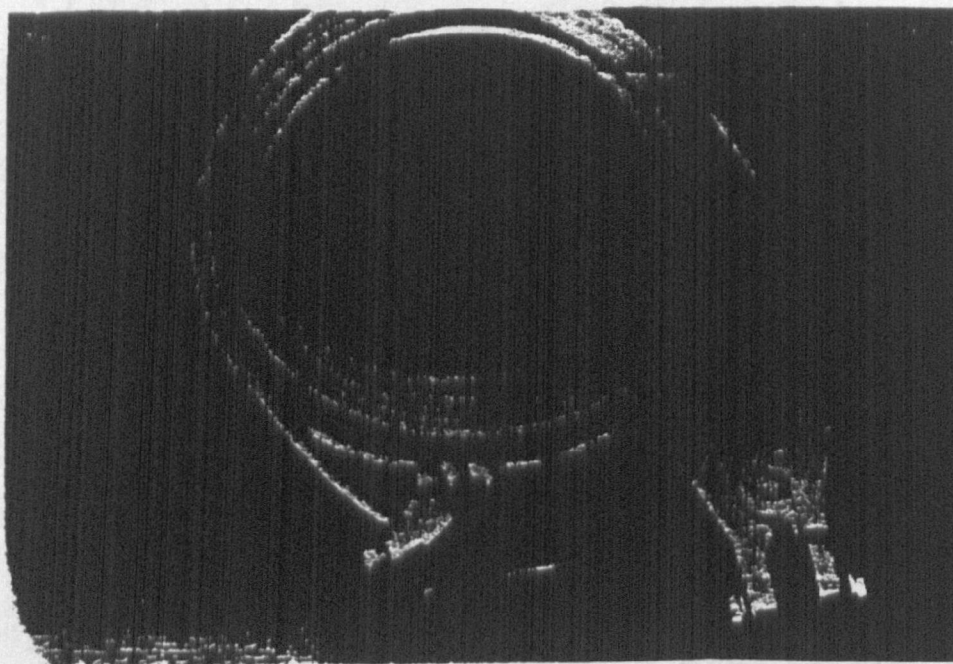
VENTILATION SYSTEM
INSPECTION

LEVEL 2

HOODS

- Capture efficiency

- Condition

- Fit of "swing-away" joints

- Hood position/cross drafts

DUCTS

- Condition

- Position of emergency dampers

- Position of balancing dampers

- Conduction of balancing dampers

COOLERS

- Condition

- Outlet temperature

- Spray pattern / nozzle condition

- Water flowrate

FANS

- Condition

- Vibration

- Belt squeal

- Fan wheel build-up/corrosion

- Condition of isolation sleeves

- Rotation direction

LEVEL 3

HOODS

- **Estimated volume using SP$_h$, ρ and configuration**

- **Estimated volume using VP$_{avg}$ and ρ**

DUCTS

- Change in gas temperature

- Change in static pressure

- Actual volume using VP_{avg} and ρ

COOLERS

- Inlet and outlet temperatures

- Estimated water requirement using temperature and enthalpy relationships

- Estimated air flow using temperatures and enthalpy relationships

- Water turbidity

FANS

- Estimated volume using known performance (Q, rpm and ρ) and new rpm

- Estimated volume using rpm, FSP, temperature and ratings tables

- Estimated volume using FSP, bhp, temperature and ratings tables
  - Estimate bhp from voltage and amperage measurements
  - Estimate bhp from amperage measurement and name plate full-load ratings

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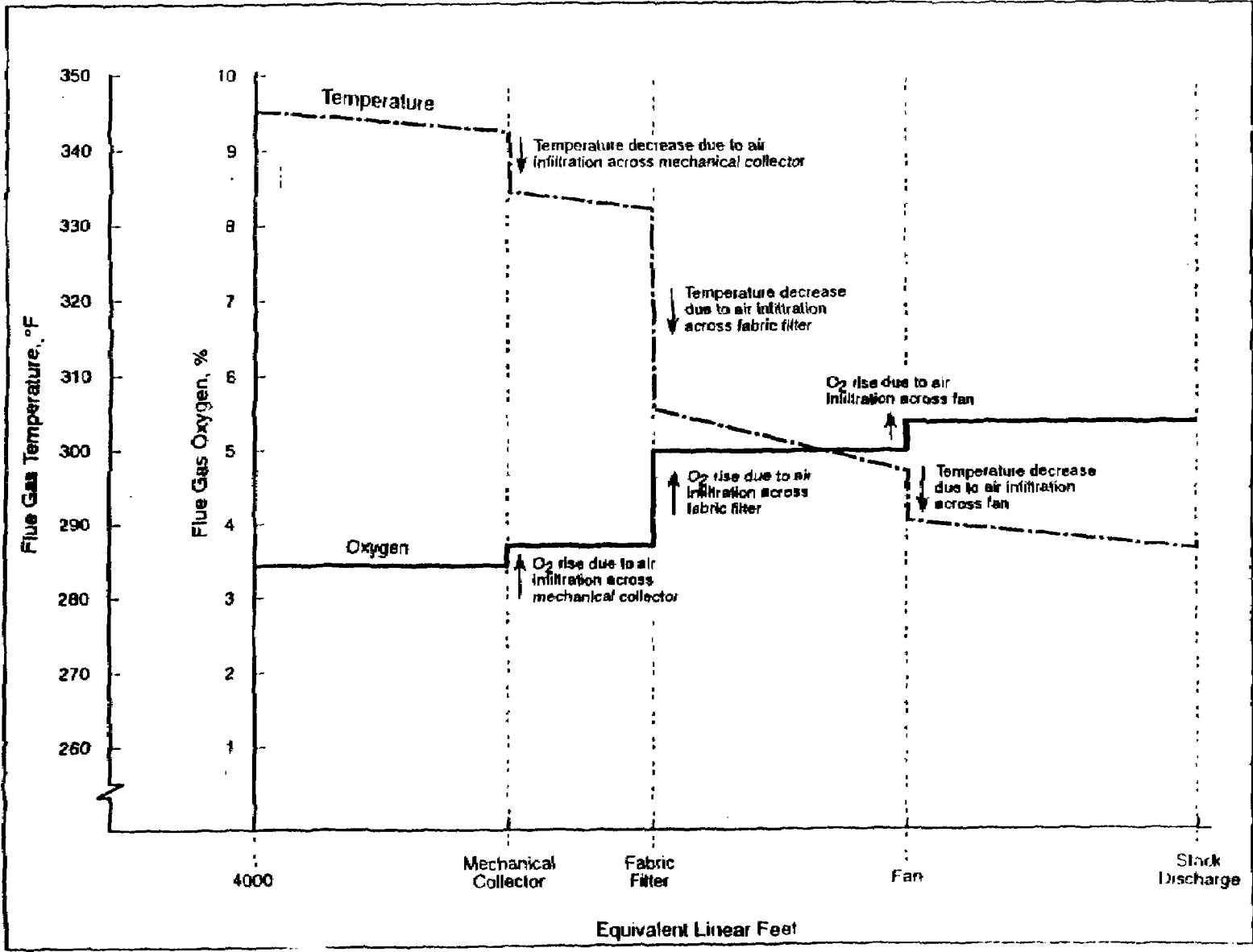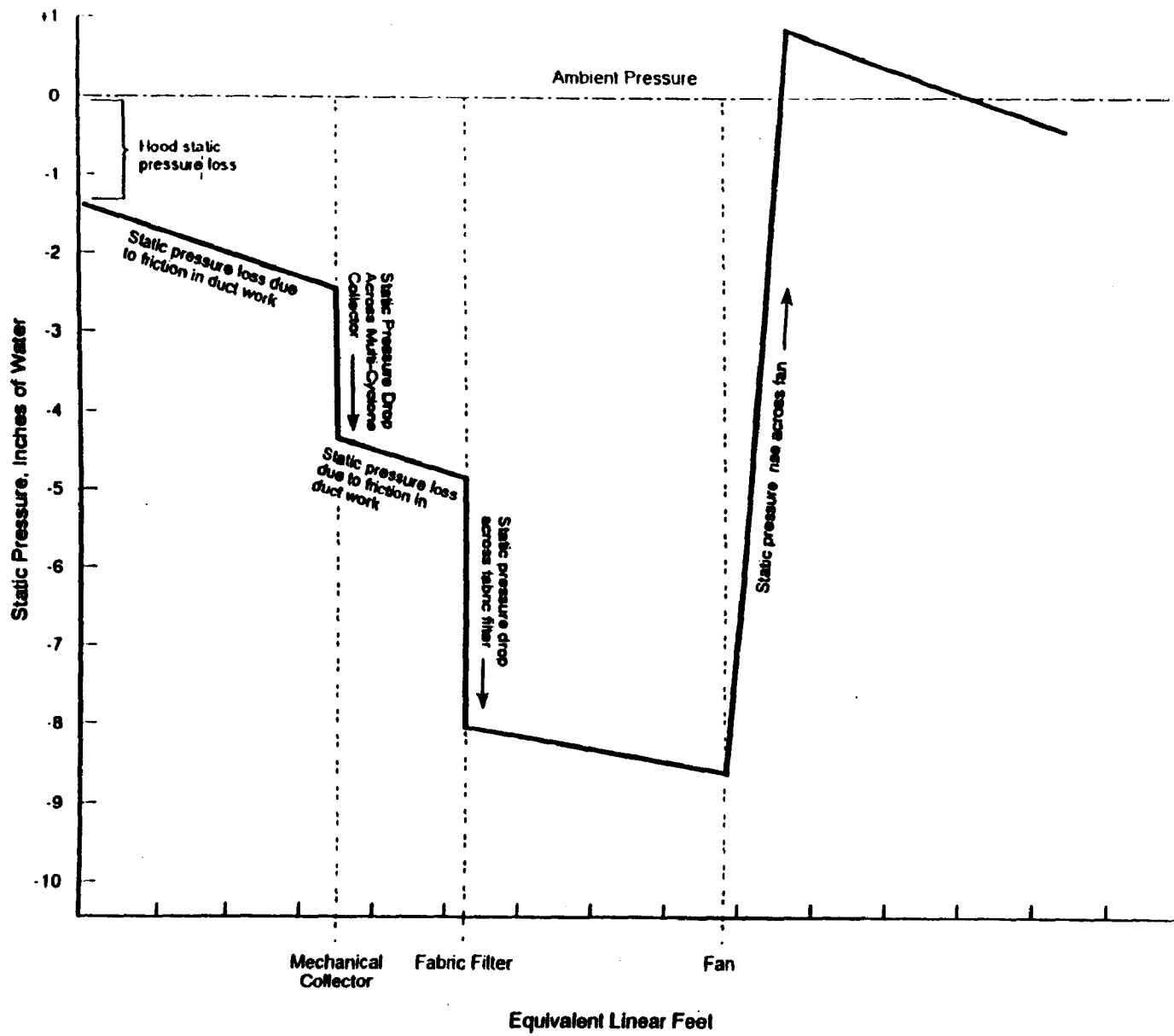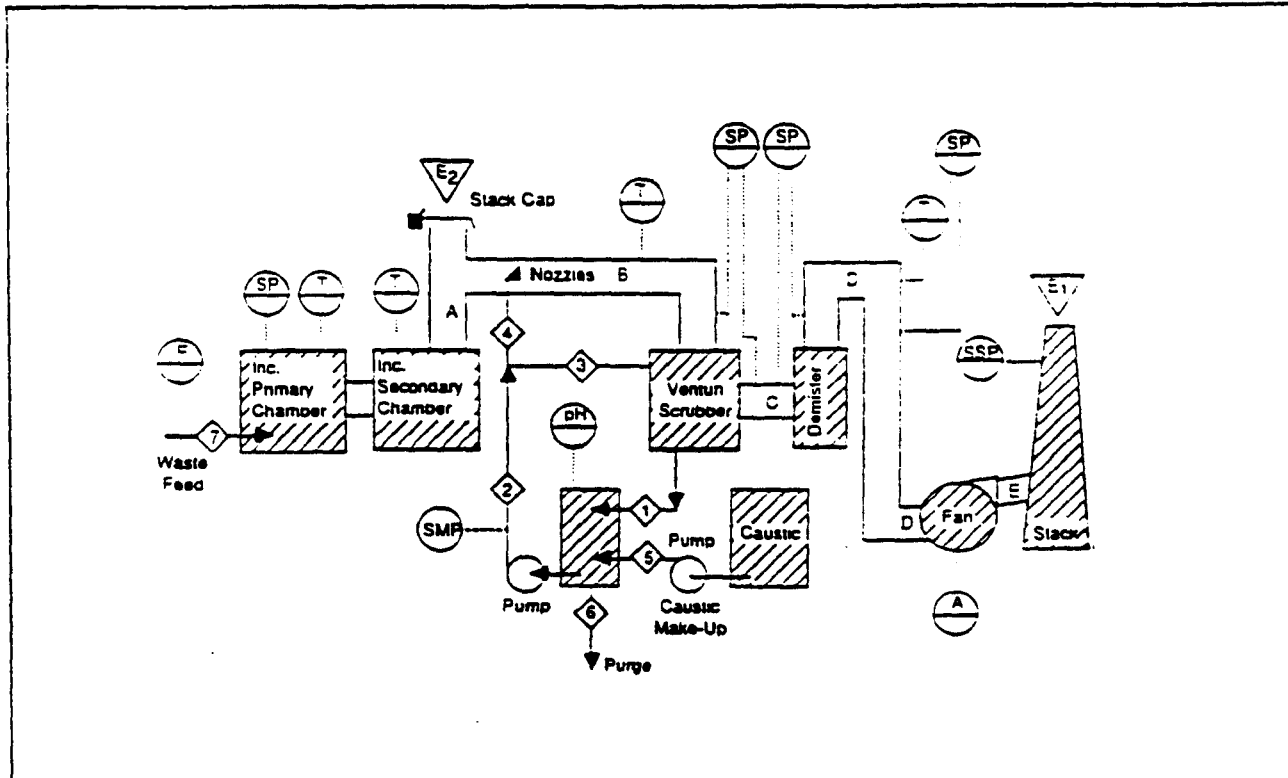
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APPENDIX C

PSYCHROMETRIC CHARTS

Psychrometric Chart for Humid Air

Barometric Pressure 29.92 in. Hg

Density Factor - Mixture

Total Heat
700 BTUs / Lb. Dry Air

Barometric Pressure 29 92 In Hg

Dry Bulb Temperature In Degrees F

Dew Point and Wet Bulb Temperatures

Pounds of Water per Pound of Dry Air

C-3