

McGill AirFlow's Duct System Design Guide

Design Advisory

Design Advisory #6: CAS-DA6-2003

Adaptable Exhaust Systems Can Offer Cost Savings

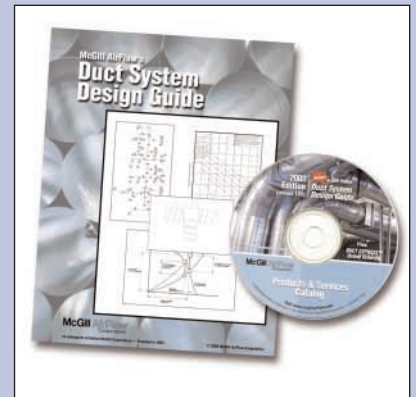
“Chapter 5: Designing Exhaust Duct Systems” of McGill AirFlow’s Duct System Design Guide, assists the designer in selecting the appropriate capture, face, slot, plenum, and duct velocities needed to transport a contaminate from its source to the collection equipment or exhaust fan. However, properly sized branch ductwork, hood configurations, and fan selection do not necessarily guarantee the best working system. Exhaust system designs can get complex when all the exhaust points are brought together in the trunk duct, especially those systems where branches are active at different times due to variances in equipment usage.

Designers often try to optimize exhaust systems by sizing trunk ductwork based on the minimum airflow and velocity needs of the maximum number of active duct drops. Equipment is then selected based on pressures generated by operating the duct system at these maximum exhaust airflow rates. Because of cost, few exhaust systems incorporate automated control devices to maintain system-operating efficiencies. Manually operated devices are the norm. Changes in equipment layout affect duct system layouts and further complicate balancing and system operation. Designing adaptable exhaust systems can significantly decrease the downtime involved with changeovers, with little increase in first time costs, and allows maximum flexibility in achieving optimum performance.

For maximum flexibility, McGill AirFlow offers hybrid exhaust systems that combine our quick-connect ductwork (UNI-QUICK™ Moveable Duct Systems) with our standard spiral duct. Centrally locating 20-foot standard length trunks of round spiral duct with assembled 45° lateral taps installed every 10 to 20 feet over the entire manufacturing area offers optimum coverage. Using UNI-QUICK duct for the drops with a cut-off positioned six feet above the floor allows for quick and easy layout adjustments. Flexible hose is recommended for making connection to equipment not accessible with rigid ductwork. Unused 45° lateral taps should be closed off with quick-connect caps.

Using spiral round duct allows the trunk duct system to be one to two even gauges lighter than equivalent longitudinal seam duct and have up to three fewer joints per 20 feet. This results in significantly lower material and installation costs and fewer locations for blockage. Multiple 45° lateral tap locations allow for inexpensive relocation of ductwork to equipment during a layout change. Incorporating UNI-QUICK ductwork in drops, although slightly more expensive, greatly reduces labor costs associated with relocating conventional ductwork and also offers labor savings when having to clean the ductwork.

UNI-QUICK™ is a trademark of United McGill Corporation.



Duct System Design Guide

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CHAPTER 5: Designing Exhaust Duct Systems

5.1 Defining the Application and Determining Parameters

In this section the process of designing an exhaust duct system is discussed. Before starting the process, a number of parameters and definitions must be identified. These items are necessary so that a design incorporates all the components that have an impact on the duct system. The fan that will pull the air through the duct system is to be sized based on parameters that will be determined during the design process.

The designer needs to have information about the processes that are going to be encountered and the type and temperature of the materials that will be exhausted. Each hood will have various parameters that are discussed in the following section. The hood locations and connection points should be determined before the design stage. A drawing should be made of the layout and hood details. Duct and fittings that connect the various hoods should be located on the drawing and the location of the fan and any miscellaneous components shown. Duct runs should be direct between the fan and pickup points. In addition, the designer should acquire a drawing of the building and location of the machinery, structure, lighting, piping, and other ductwork.

5.2 Determining Capture Velocities and Air Volume Requirements

One of the basic components of an exhaust duct system design is the hood, which serves as the main inlet device. Hoods are devices that permit the capture and removal of contaminated air or gas from the work environment. A designer can design the hood or select it from pre-designed hoods. Generally, there are two different types of hoods: enclosed and exterior. In the enclosed type, the process is totally or partially enclosed, and the exterior hood is one that is next to or above the process. Various hood designs can be found in **Appendix A.9.5**.

Before examining the design process, the following basic definitions of common hood terminology need to be understood.

Capture Velocity: Air velocity required to capture (entrain) the particulate or fumes being collected.

Face Velocity: Air velocity at the opening of the hood.

Slot Velocity: Air velocity through a slot in the hood.

Plenum Velocity: Air velocity in the plenum.

Duct Velocity: Air velocity in the duct connected to the hood.

An example of the various terms is shown in **Figure 5.1**.

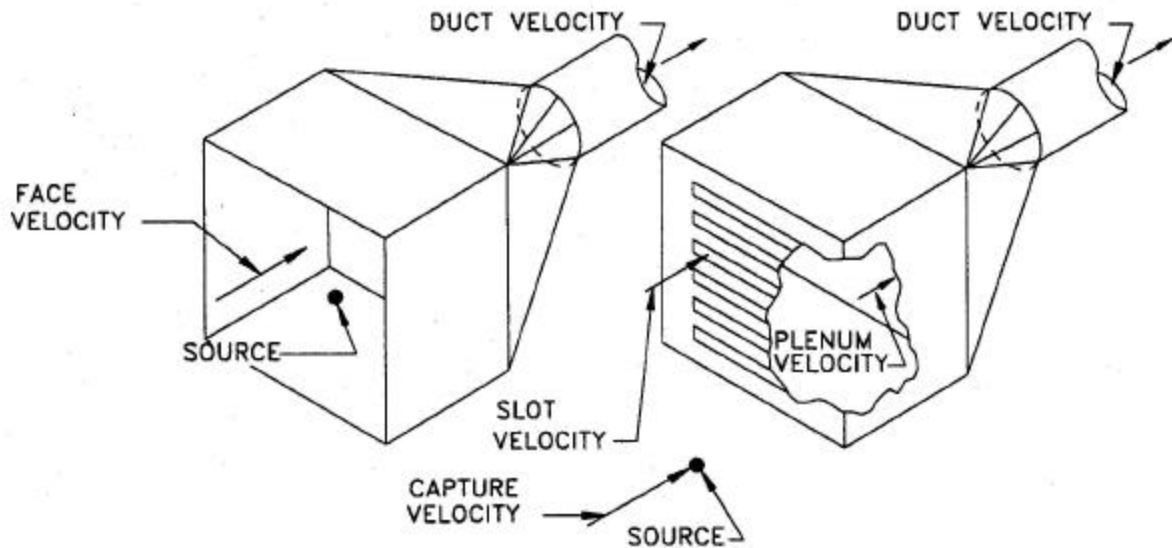


Figure 5.1
Hood Terminology

(Reprinted from Figure 3-1 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

The exhaust air volume flow rate is determined by knowing the following:

1. Required velocity to capture or entrain the material or fumes (capture velocity).
2. Size and type of the hood entrance (dimensions).
3. Distance from the hood entrance to the process, which is producing the material or fumes to be exhausted.
4. Miscellaneous factors such as crosscurrents, temperature, obstructions, and adjacent equipment.

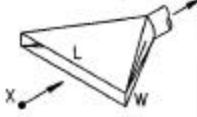

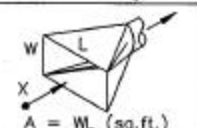

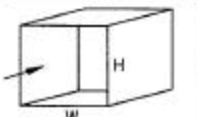
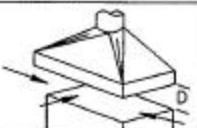
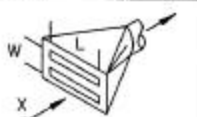
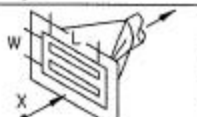
In general, the volume flow rate required at each inlet of a particulate/contaminant collection system is equal to the product of the capture velocity and the area of the intake opening ($Q = AV$). Capture velocity is the necessary velocity in the area affected by the hood opening to overcome opposing air currents and capture the particulate or contaminant. Examples of typical capture velocities are shown in **Table 5.1**. Note the different types of conditions and applications.

Table 5.1
Typical Capture Velocities

(Reprinted from Table 3-1 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

Conditions of Dispersion of Contaminant	Examples	Capture Velocity (fpm)		
Released with practically no velocity into quiet air.	Evaporation from tanks, degreasing, etc.	50-100		
Released at low velocity into moderately still air.	Spray booths, intermittent container filling, low speed conveyor transfers, welding, plating, and pickling.	100-200		
Active generation into zone of rapid air motion.	Spray painting in shallow booths, barrel filling, conveyor loading, and crushers.	200-500		
Released at high initial velocity into zone of very rapid air motion.	Grinding, abrasive blasting, and tumbling.	500-2,000		
In each category above, a range of capture velocities is shown and the proper choice of values depends on several factors:				
<p>NOTES:</p> <table border="0" style="width: 100%;"> <tr> <td style="width: 50%; vertical-align: top;"> <p><u>Lower End of Range</u></p> <ol style="list-style-type: none"> 1. Room air current minimal favorable to capture. 2. Contaminants of low toxicity or of nuisance value only. 3. Intermittent, low production. 4. Large hood: large air mass in motion. </td> <td style="width: 50%; vertical-align: top;"> <p><u>Upper End of Range</u></p> <ol style="list-style-type: none"> 1. Disturbing room air currents. 2. Contaminants of high toxicity. 3. High production, heavy use. 4. Small hood: local control only. </td> </tr> </table>			<p><u>Lower End of Range</u></p> <ol style="list-style-type: none"> 1. Room air current minimal favorable to capture. 2. Contaminants of low toxicity or of nuisance value only. 3. Intermittent, low production. 4. Large hood: large air mass in motion. 	<p><u>Upper End of Range</u></p> <ol style="list-style-type: none"> 1. Disturbing room air currents. 2. Contaminants of high toxicity. 3. High production, heavy use. 4. Small hood: local control only.
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Typical intake configurations are shown in **Figure 5.2** with various hood types, descriptions, aspect ratios, and air volume flow rate equations.

HOOD TYPE	DESCRIPTION	ASPECT RATIO, W/L	AIR FLOW
	SLOT	0.2 OR LESS	$Q = 3.7 LVX$
	FLANGED SLOT	0.2 OR LESS	$Q = 2.6 LVX$
	PLAIN OPENING	0.2 OR GREATER AND ROUND	$Q = V(10X^2 + A)$
	FLANGED OPENING	0.2 OR GREATER AND ROUND	$Q = 0.75V(10X^2 + A)$
	BOOTH	TO SUIT WORK	$Q = VA = VWH$
	CANOPY	TO SUIT WORK	$Q = 1.4 PVD$ SEE VS-99-03 P = PERIMETER D = HEIGHT ABOVE WORK
	PLAIN MULTIPLE SLOT OPENING 2 OR MORE SLOTS	0.2 OR GREATER	$Q = V(10X^2 + A)$
	FLANGED MULTIPLE SLOT OPENING 2 OR MORE SLOTS	0.2 OR GREATER	$Q = 0.75V(10X^2 + A)$

AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	<i>HOOD TYPES</i>
DATE <i>1-88</i>	FIGURE <i>3-11</i>

Figure 5.2
Intake Configurations

(Reprinted from Figure 3-11 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

Sample Problem 5-1

What is the volume flow rate for the open face hood shown in **Figure 5.1** if the application is a spray booth with intermittent, low production? Assume the length is 3 feet and width is 2 feet.

Answer: From **Equation 1.1**, the volume flow rate is equal to the hood face area times the capture velocity. From **Table 5.1**, a capture velocity of 100 *fpm* is selected based on the given information.

$$\begin{aligned}
 Q &= V \times A \\
 &= 100 \text{ fpm} \times (3 \text{ ft} \times 2 \text{ ft}) \\
 &= \underline{600 \text{ cfm}}
 \end{aligned}$$

The design of a hood used in an exhaust duct system should provide the capabilities to contain or capture an air contaminant. A hood design must provide enough air to control the contamination with minimum airflow and energy consumption. By minimizing the airflow, the designer can use a smaller fan and minimize the energy needed to move the air. Generally speaking, the cost of operation of the system on an annual basis is directly proportional to the volume flow rate required to exhaust the contaminant and system total pressure. Since the system total pressure affects the operating cost directly, it is in the interest of the designer to minimize its impact.

A proper hood design should enclose the operation completely whenever possible. It should also provide access, as required, to workers for cleaning, maintenance, and operation. Openings should be kept away from the natural path of the contaminant. Flanges can be added to the hood to keep uncontaminated airflow from entering. Another device that can be added to the hood to restrict airflow are baffles. Serious consideration should be given to the use of flanges and baffles on hoods; refer to **Figure 5.2** for illustrations of intake configurations. A canopy hood is effective for controlling hot processes but should not be used when someone must work over the process. This type of a hood will be illustrated later in the section.

One last point is that the exhaust volume flow rate is calculated after the hood is designed or selected for the specific application. The reason for this is that all design considerations should be evaluated in the hood design process before the airflow requirements are determined. When the initial airflow volume requirements are calculated, they should be for *standard air* conditions. If nonstandard air conditions are encountered, then the equivalent mass of the air must be maintained to perform the capture of contaminants. Therefore, the actual volume flow rate will be more or less than the calculated value, depending on the actual temperature and pressure.

Calculate the volume flow rate (Q_{std}) requirements at standard conditions using **Appendix A.9.5, Chapter 3, "Local Exhaust Hoods"** or **Appendix A.9.5, Chapter 10, "Specific Operations"**. After determining the volume flow rate, adjust it by the density ratio to determine the actual volume flow rate (Q_{act}).

Correction of the volume flow rate for nonstandard conditions is calculated from the following equation.

The mass flow rate is:

$$\dot{m} = \rho_{std} Q_{std} = \rho_{act} Q_{act} \qquad \text{Equation 5.1a}$$

which can be rewritten as:

$$Q_{act} = \frac{\rho_{std}}{\rho_{act}} Q_{std} \quad \text{Equation 5.1b}$$

where:

- \dot{m} = Mass flow rate (lb_m/min)
- ρ_{std} = Density at standard conditions (lb_m/ft^3)
- ρ_{act} = Density at actual conditions (lb_m/ft^3)
- Q_{std} = Volume flow rate at standard conditions (cfm)
- Q_{act} = Volume flow rate at actual conditions (cfm)

If humidity can be neglected, **Equation 1.20** solved for the density ratio and substituted into **Equation 5.1b** to calculate the actual flow rate from:

$$Q_{act} = Q_{std} \frac{T_a + 460}{530} \frac{29.921}{b} \quad \text{Equation 5.2}$$

If humidity cannot be neglected, then the actual density (ρ_{act}) should be calculated from **Equation 5.3** and used in **Equation 5.1b** to calculate the actual volume flow rate (Q_{act}).

$$\rho_{act} = \frac{b - p_w}{53.352} + \frac{p_w}{85.778} \frac{70.53}{T_a + 460} \quad \text{Equation 5.3}$$

where:

- 53.352** = Gas constant for air ($ft-lbf/lb_m-^{\circ}R$)
- 85.778** = Gas constant for water vapor ($ft-lbf/lb_m-^{\circ}R$)
- 70.53** = Conversion factor ($inches\ Hg\ to\ psf$)
- T_a = Actual temperature ($^{\circ}F$)
- b = Actual barometric pressure ($inches\ Hg$)
- p_w = Is defined in **Equation 5.4** ($inches\ Hg$)

$$p_w = \frac{bW}{(0.62198 + W)} \quad \text{Equation 5.4}$$

where:

- p_w = Water vapor partial pressure (inches Hg)
 W = Humidity ratio (lb_m water/lb_m dry air)

Sample Problem 5-2

What is the capture velocity, width and length of the hood, and volume flow rate at standard and actual conditions for the hood shown in **Figure 5.3**? Note that (1) the fume is nontoxic at 400 °F with no moisture evaporating from a tank, (2) there are some disturbing room currents, (3) at least 1 foot clearance is needed from the process to the hood for maintenance access on one side, (4) three sides of the hood can be enclosed (use a canopy hood configuration), and (5) the process tank is 4 H 4 feet and the atmospheric pressure is 29.921 inches Hg.

Answer: Select a capture velocity of 100 fpm from **Table 5.1** or **Figure 5.3**. The width is 4 feet since the tank is 4 feet wide, and the hood length is 4.4 feet [4 + (0.4 H 1)]. From **Figure 5.3**, the volume flow rate is determined from the equation:

$$Q = LHV$$

where:

- L = Length (ft)
 H = Height (ft)
 V = Capture velocity (fpm)

$$Q_{std} = 4.4 \times 1 \times 100 = 440 \text{ scfm (standard cfm)}$$

The volume flow rate required at standard conditions is 440 cfm and the actual volume flow rate is determined from **Equation 5.1b** or **5.2**. Before using these equations the density at actual conditions needs to be determined using **Equation 1.20**.

$$\rho_{act} = \rho_{std} \frac{530}{T_a + 460} \frac{29.921}{29.92} = 0.075 \frac{530}{400 + 460} \frac{29.921}{29.92}$$

$$\rho_{act} = 0.046 \text{ lb}_m/\text{ft}^3$$

The volume flow rate at actual conditions is determined using **Equation 5.1b**:

$$Q_{act} = Q_{std} \frac{\rho_{std}}{\rho_{act}} = 440 \frac{0.075}{0.046} = 717 \text{ acfm (actual cfm)}$$

Notice how the actual volume flow rate is much higher than the 440 cfm rate at standard conditions.

To maintain the required mass flow rate (equal to ρAV), the velocity must be increased to compensate for a decrease in density. Thus the size of the duct connected to this tank will be used on 717 *cfm*.

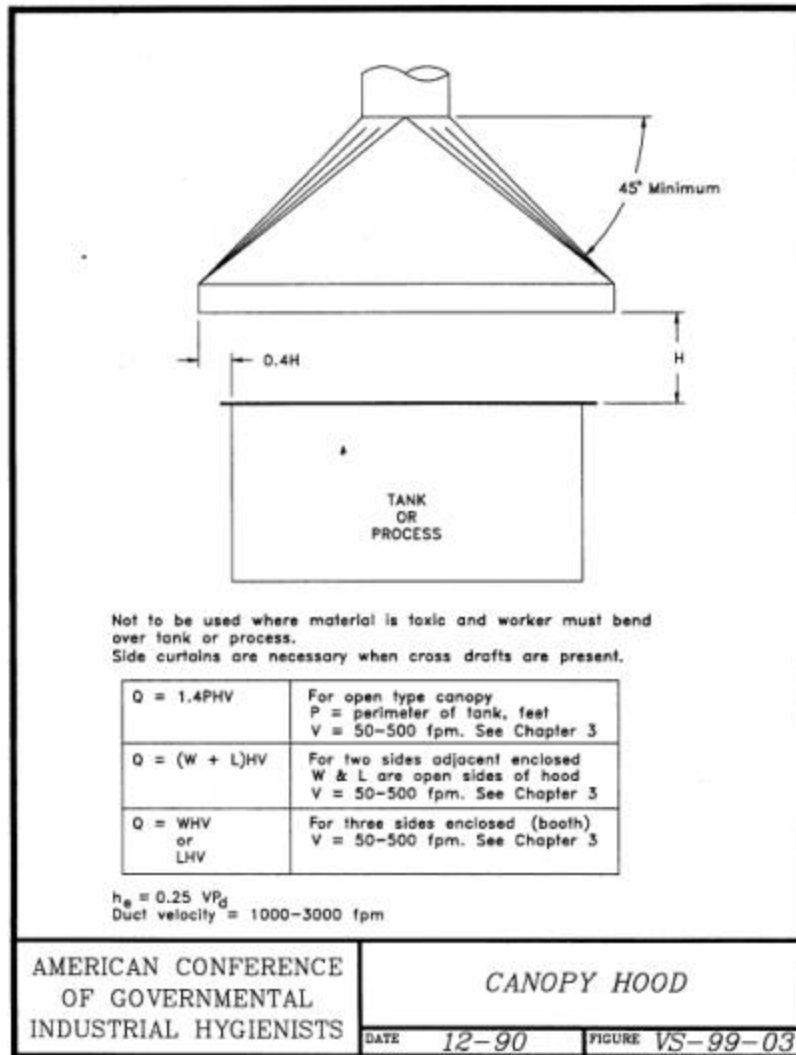


Figure 5.3
Canopy Hood

(Reprinted from Figure VS-99-03 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995.
Published by ACGIH®, Cincinnati, OH)

5.3 Locating Duct Runs

Once the hoods are selected or designed and the inlets are located, they are connected together with duct and fittings. In routing the ductwork, try to keep the duct runs as simple as possible. Avoid using many fittings since they increase pressure loss and add to the system energy requirements. **Section 5.6** will show that fittings typically contribute more to the pressure loss than the duct. Selection of the type of fitting can also affect system pressure loss, which is discussed in **Section 6.3**. A drawing showing hood locations, as mentioned in **Section 5.1**, is recommended so that the duct location can be made. The drawing should also reflect the machinery, structure, lighting, piping, and other ductwork. Once the duct run is located, the designer must determine the duct sizes.

5.4 Determining Duct Sizes Based on Velocity Constraints

Before a duct system can be sized, the volume flow rate (*cfm*) must be determined as was shown in **Section 5.2**, for each of the hoods. Once the designer knows the volume flow rate and the duct runs are located, then the ducts can be sized. Ducts are sized by dividing the volume flow rate by the velocity in the duct ($A = Q/V$); however, the designer must select the correct velocity. Duct velocity depends on the nature of the contaminant in the duct.

As the contaminant leaves the hood, it enters the duct system and moves toward the fan. Exhaust systems carrying particulate must maintain a minimum velocity to avoid material fallout and possible plugging. This minimum acceptable velocity is called the *carrying velocity* and its value depends on the type of particulate being conveyed. **Table 5.2**, "Duct Carrying Velocities" shows recommended velocities for various contaminants. These velocities are greater than the theoretical velocity to provide a factor of safety for practical considerations such as duct leakage, fan performance, duct damage, branch area reduction due to clogging, etc.

Table 5.2
Duct Carrying Velocities

(Reprinted from Table 3-2 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

Nature of Contaminant	Examples	Design Velocity
Vapors, gases, smoke	All vapors, gases, and smoke	Any desired economical velocity
Fumes	Welding	2,000-2,500 <i>fpm</i>
Very fine light dust	Cotton lint, wood flour, litho powder	2,500-3,000 <i>fpm</i>
Dry dust and powder	Fine rubber dust, Bakelite molding powder dust, jute lint, cotton dust, shavings (light), soap dust, leather shavings	3,000-4,000 <i>fpm</i>
Average industrial dust	Grinding dust, buffing lint (dry), wool jute dust (shaker waste), coffee beans, shoe dust, granite dust, silica flour, general material handling, brick cutting, clay dust, foundry (general), limestone dust, packaging and weighing asbestos dust in textile industries	3,500-4,000 <i>fpm</i>
Heavy dust	Sawdust (heavy and wet), metal turnings, foundry tumbling barrels and shake-out, sand blast dust, wood blocks, hog waste, brass turnings, cast iron boring dust, lead dust	4,000-4,500 <i>fpm</i>
Heavy or moist	Lead dusts with small chips, moist cement dust, asbestos chunks from tansite pipe cutting machines, buffing lint (sticky), quicklime dust	4,500 <i>fpm</i> or greater

Sample Problem 5-3 illustrates the determination of duct size and the use of the carrying velocity. Sometimes the carrying velocity is called transport velocity.

Sample Problem 5-3

Determine the size of the duct that is connected to the hood mentioned in **Sample Problem 5-2**.

Answer: Determine the required minimum design carrying velocity from **Table 5.2**. From this table, a velocity of 1,800 fpm is selected. Use **Equation 1.1** to determine the duct area:

$$Q = V \times A \text{ or } A = Q_{act}/V$$

$$Q_{act} = 717 \text{ cfm (from Sample Problem 5-2)}$$

$$A = 717/1,800 = \underline{0.398 \text{ ft}^2}$$

Multiply this value by 144 to obtain the area in *square inches*:

$$A = 0.398 \times 144 = \underline{57.3 \text{ square inches}}$$

Determine the maximum diameter of the duct from:

$$A = pD^2/4 = 57.3 \text{ square inches}$$

Calculating for the diameter:

$$D_{max} = \sqrt{4 A/p} = \underline{8.54 \text{ inches}}$$

Round off the diameter to the next smallest available size to maintain the carrying velocity:

$$D = \underline{8.5 \text{ inches}}$$

Use this size; along with the actual volume flow rate and density, to calculate the velocity pressure, friction loss, hood loss, etc.

5.5 Mixing of Two Air Streams

Many exhaust duct systems involve mixing two air streams. One process of contaminants may be at one temperature and the other process at another. At the junction point of two air streams, as in a lateral fitting, the upstream and branch meet at a point to form the downstream side of the fitting, as shown in **Figure 5.4**.

The process of mixing two airstreams at different temperatures is somewhat involved. It entails using the mass flow rates and *enthalpies* to calculate the *mixed air temperature* ($T_{d \text{ inlet}}$).

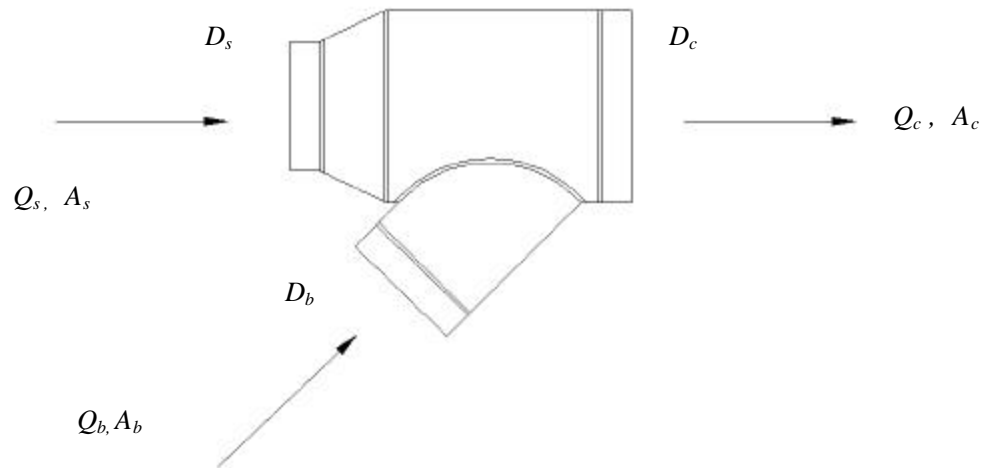


Figure 5.4
Mixing at Junction

The enthalpy of air can be expressed as:

$$h = C_p T + W(1,061 + 0.444T) \quad \text{Equation 5.5}$$

where:

h = Enthalpy of the air exiting or entering the section (*Btu/lb_m dry air*)

W = Humidity ratio (*lb_m water vapor/lb_m dry air*)

C_p = Specific heat of air (*0.24 Btu/lb_m - °F*)

T = Temperature of the air (*°F*)

The enthalpy at the straight-through (upstream) or branch of a converging-flow fitting can be calculated using **Equations 5.6** and **5.7**:

$$h_{s \text{ exit}} = 0.24T_{s \text{ exit}} + W_s(1061 + 0.444T_{s \text{ exit}}) \quad \text{Equation 5.6}$$

$$h_{b \text{ exit}} = 0.24T_{b \text{ exit}} + W_b(1061 + 0.444T_{b \text{ exit}}) \quad \text{Equation 5.7}$$

If water vapor is present in the branch or straight-through (upstream) section, then the humidity ratio of the common (downstream) is calculated using **Equation 5.8**:

$$W_c = \frac{\dot{m}_{sa} W_s + \dot{m}_{ba} W_b}{\dot{m}_{sa} + \dot{m}_{ba}} \quad \text{Equation 5.8}$$

The mass of dry air can be calculated from **Equation 5.9**, which is a ratio of total mass flow rate and humidity ratio:

$$\dot{m}_a = \frac{\dot{m}}{(1 + W)} \quad \text{Equation 5.9}$$

where:

\dot{m}_a = Mass flow rate of the dry air (lb_m/min)

\dot{m} = Mass flow rate from **Equation 5.1a** (lb_m/min)

W = Humidity ratio (lb_m water vapor/ lb_m dry air)

From an energy balance the sum of the energy entering the fitting must be equal to the sum of the energy exiting the fitting. To express the enthalpy of the mixture, use **Equation 5.10**.

$$h_{c\ inlet} = \frac{\dot{m}_{sa} h_{s\ exit} + \dot{m}_{ba} h_{b\ exit}}{\dot{m}_{sa} + \dot{m}_{ba}} \quad \text{Equation 5.10}$$

where:

\dot{m}_{sa} = Straight-through (upstream) dry air mass flow rate calculated from **Equation 5.9** (lb_m/min)

\dot{m}_{ba} = Branch dry air mass flow rate calculated from **Equation 5.9** (lb_m/min)

h = Enthalpy as defined in **Equation 5.5** (Btu/lb_m dry air)

Solving **Equation 5.5** for $T_{c\ inlet}$ and substituting the specific heat of air for C_p , results in **Equation 5.11**:

$$T_{c\ inlet} = \frac{h_{c\ inlet} - 1.061W_c}{0.24 + 0.444W_c} \quad \text{Equation 5.11}$$

where:

$h_{c\ inlet}$ = Enthalpy calculated from **Equation 5.10** (Btu/lb_m dry air)

W_c = Humidity ratio calculated from **Equation 5.8** (lb_m water vapor/ lb_m dry air)

When neglecting moisture, **Equation 5.11** simplifies to the following:

$$T_{c\ inlet} = \frac{\dot{m}_s T_{s\ exit} + \dot{m}_b T_{b\ exit}}{\dot{m}_s + \dot{m}_b} \quad \text{Equation 5.12}$$

To calculate the common (downstream) volume flow rate, first use **Equation 5.13** to calculate the common (downstream) mixed air, mass flow rate:

$$\dot{m}_c = \dot{m}_s + \dot{m}_b \quad \text{Equation 5.13}$$

where:

- \dot{m}_c = Common mass flow rate equal to the sum of the straight-through (upstream) and branch sections (lb_m/min)
- \dot{m}_s = Mass flow rate in the straight-through (upstream) section (lb_m/min)
- \dot{m}_b = Mass flow rate in the branch sections (lb_m/min)

Using **Equation 5.1a** and solving for the common (downstream) volume flow rate (Q_d):

$$Q_c = \frac{\dot{m}_c}{\rho_c} \quad \text{Equation 5.14}$$

where:

- ρ_c = Density of air in the common (downstream) section, calculated using **Equation 1.20** or **Equation 5.3** (lb_m/ft^3)

To determine mixed temperatures, energies and densities, the temperature at the end of a duct run, entering the branch, needs to be determined. If a duct is running through an area that has an ambient temperature different from that of the air entering the duct system, then the exit temperature will be different from the inlet temperature of that duct section, due to heat transfer. **Equation 5.15** is used for calculating this exit temperature.

$$T_{exit} = (T_{inlet} - T_{amb}) e^{-\frac{UPL}{60\dot{m}c_p}} + T_{amb} \quad \text{Equation 5.15}$$

where:

- U = Overall heat transfer coefficient ($Btu/hr-ft^2-^{\circ}F$)
- P = Perimeter of the duct (ft)
- L = Length of the duct (ft)
- \dot{m} = Mass flow rate **Equation 5.1a** (lb_m/min)
- C_p = Specific heat of air ($0.24 Btu/lb_m-^{\circ}F$)
- 60 = Conversion for lb_m/min to lb_m/hr
- T_{exit} = Temperature exiting the duct ($^{\circ}F$)
- T_{inlet} = Temperature entering the duct ($^{\circ}F$)

T_{amb} = Temperature of the air around the outside of the duct ($^{\circ}F$)

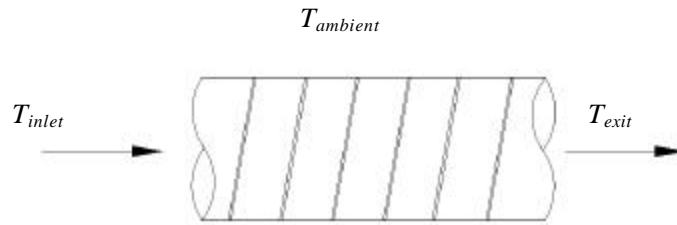


Figure 5.5
Duct Exit Temperature

Temperature reductions or increases will affect the density of the air if the changes are significant. Overall heat transfer coefficients can be obtained from **Appendix A.9.2** for various materials such as bare sheet metal, insulation of various thicknesses, and nonmetallic materials. If a duct system is being designed to carry away vapors that could condense, then the temperature at the exit is important. The vapor saturation temperature can be compared to the exit temperature to determine if condensation will occur. Condensing vapors inside the duct could pose a problem if the duct is not constructed for liquid tightness or if the vapors are corrosive.

5.6 Determining System Pressure Requirements

Constant volume and constant mass concepts must be understood before calculating system pressure requirements. Recall the basic relationship from the continuity equation in **Appendix A.3.1**:

$$Q = A \times V \text{ and } Q = \frac{\dot{m}}{\rho} \quad \text{Equation A.3}$$

Since nonstandard conditions result in a density change, the constant volume and constant mass situations. In the case of a constant volume ($Q = \text{constant}$) situation, the mass flow rate must increase or decrease in proportion to the density change. Fans are constant volume devices. They will move the same volume regardless of air density; however, the pressure (and horsepower) will vary in proportion to density changes. When an air system operates at altitudes above 2,000 feet, temperatures below $30^{\circ}F$, or temperatures above $120^{\circ}F$, the system pressure must be corrected for the change in air density.

To correct for constant volume operation at nonstandard temperature/pressure, take the following steps:

1. Calculate the system pressure loss using the actual air volume (acfm), assuming standard conditions.
2. Multiply the total pressure by the appropriate density correction factor from **Appendix A.1.5**. This will be the new system pressure requirement for operation at the nonstandard conditions.
3. Recalculate horsepower requirements. Fan operation specifications should be based

on the corrected pressure and horsepower.

The other situation is the constant mass flow rate. In this situation, referring to **Equation 5.16**, the volume flow rate must increase or decrease inversely to the density change. Exhaust systems which capture particulate must maintain a constant mass flow rate. When exhaust systems are required to maintain a constant mass flow rate and are operated at nonstandard conditions, the system volume must be corrected for the change in air density.

To correct for constant mass operation at nonstandard temperature/pressure, take the following steps:

1. Calculate the volume requirements assuming standard conditions (*scfm*).
2. Multiply the standard volume by the inverse of the appropriate density factor from **Appendix A.1.5**. This will be the actual volume (*acfm*) requirement for operation at the nonstandard conditions.
3. All design and pressure calculations should be based on corrected volumes.

Additional information will be discussed in **Section 6.3** on fans. **Sample Problem 5.4** illustrates the various aspects of exhaust design in determining system pressure requirements.

Sample Problem 5-4

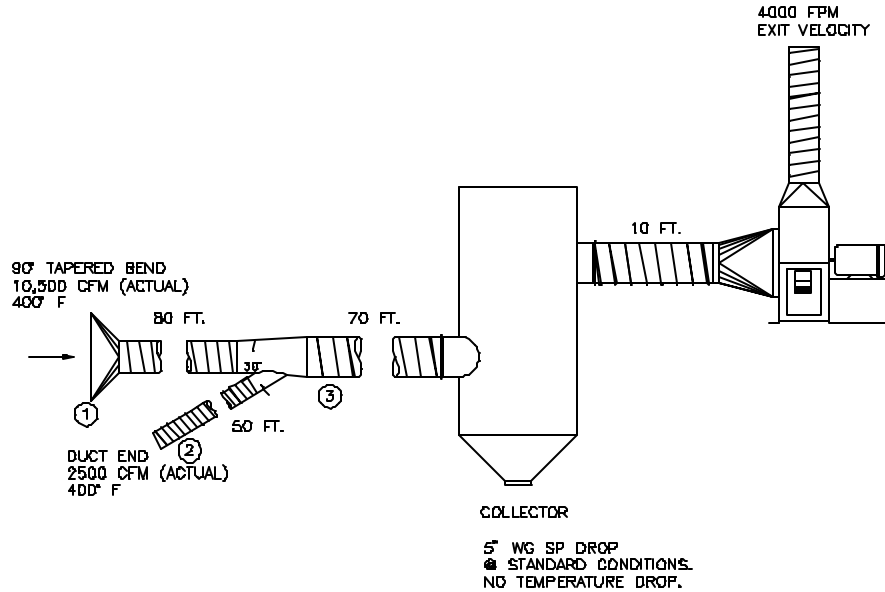


Figure 5.6
Exhaust System

The system shown in **Figure 5.6** has a barometric pressure of 24.9 inches Hg, humidity ratio of 0 lb_m water vapor per lb_m dry air, with an elevation of 5,000 feet above sea level, and an ambient temperature of 70°F (duct is insulated to minimize temperature drop). In duct sections 1 and 2 the required carrying velocity is 4,000 fpm. What are the duct sizes, section velocity/velocity pressure, section total/static pressure drop, path excess pressures, and fan total/static pressure? Assume temperature is constant throughout system.

Answer: The calculation requires three separate parts consisting of initial sizing, pressure losses, and pressure determination. Since the inlet flows are at the same temperature, all losses will be calculated at standard air, then the fan pressure requirement will be adjusted for the decrease density at 400 °F.

Part 1: Determine initial duct sizes, volume flow rates, and actual velocities for each duct section.

$$\text{Section 1: } A_{duct} = \frac{Q_{act}}{V} = \frac{10,500 \text{ cfm}}{4,000 \text{ fpm}} = \underline{2.63 \text{ ft}^2}$$

$$= 378 \text{ in}^2$$

$$D_{max} = (4A/p)^{0.5} = [4(378 \text{ in}^2)/p]^{0.5} = \underline{21.94 \text{ inches}}$$

Therefore, a 21-inch diameter duct should be used to maintain a minimum carrying velocity of 4,000 fpm. Now calculate the actual velocity.

$$V = \frac{Q}{A} = \frac{(10,500 \text{ cfm})}{(p(21)^2/4)(144 \text{ in}^2/\text{ft}^2)}$$

$$= \underline{4.365 \text{ fpm}}$$

$$VP_s = (4,365 \text{ fpm}/4,005)^2 = \underline{1.19 \text{ inches wg}}$$

Section 2: $A_{duct} = Q_{act}/V = 2,500 \text{ cfm}/4,000 \text{ fpm} = \underline{0.63 \text{ ft}^2}$
 $= 0.625 \text{ ft}^2 \times (144 \text{ in}^2/\text{ft}^2) = \underline{90.72 \text{ in}^2}$

$$D_{max} = [4(90 \text{ in}^2)/\rho]^{0.5} = \underline{10.75 \text{ inches}}$$

Therefore, since spiral duct is available in 0.5-inch increments through 15 inches diameter, a 10.5-inch diameter duct will be used.

$$V = Q/A = (2,500 \text{ cfm})/(\rho(10.5 \text{ in}^2/4)/(144 \text{ in}^2/\text{ft}^2)) = \underline{4,158 \text{ fpm}}$$

$$VP_b = (4,158 \text{ fpm}/4,005)^2 = \underline{1.08 \text{ inches wg}}$$

Section 3: $Q_{act} = 10,500 + 2,500 = \underline{13,000 \text{ cfm}}$

$$A_{duct} = 13,000 \text{ cfm}/4,000 \text{ fpm} = \underline{3.25 \text{ ft}^2}$$

$$= 3.25 \text{ ft}^2 \times (144 \text{ in}^2/\text{ft}^2) = \underline{468 \text{ in}^2}$$

$$D_{max} = [4(468 \text{ in}^2)/\rho]^{0.5} = \underline{24.41 \text{ inches}}$$

Therefore, a 24-inch diameter duct will be used.

$$V = (13,000 \text{ cfm})/(\rho(24 \text{ in}^2/4)/(144 \text{ in}^2/\text{ft}^2)) = \underline{4,138 \text{ fpm}}$$

$$VP_c = (4,138 \text{ fpm}/4,005)^2 = \underline{1.07 \text{ inches wg}}$$

The preceding calculations are summarized in **Table 5.3**.

Table 5.3
Initial Sizing Information

Section	Downstream Section	Length (ft)	Inlet Temp. °F	Actual Air Density (lb _m /ft ³)	Vol Flow Rate (cfm)	Min. Vel. (fpm)	Duct Dia. (inch)	Actual Vel (fpm)	VP [*] (inches wg)	Exit Temp. °F
1	3	80	400	0.038	10,500	4,000	21	4,365	1.19	400
2	3	50	400	0.038	2,500	4,000	10.5	4,158	1.08	400
3	0	80	400	0.038	13,000	4,000	24	4,138	1.07	400

*Based on standard conditions

Part 2: Determine the pressure losses for the branches and sections.

Calculate the area and area ratios of the straight-through (upstream), common (downstream), and branch:

$$A_s = 2.41 \text{ ft}^2; \quad A_s/A_c = 0.77; \quad Q_b/Q_c = 0.19 \quad C_s = -0.06$$

$$A_b = 0.60 \text{ ft}^2; \quad \underline{A_b/A_c = 0.19}; \quad \underline{Q_s/Q_c = 0.81} \quad \underline{C_b = 0.11}$$

$$A_c = 3.14 \text{ ft}^2;$$

The C_s and C_b values were obtained from **ED5-1** of the ASHRAE Duct Fitting Database Program (reference 9.1.12) for a 30° lateral with $A_s/A_c = 0.8$.

Using **Equations 4.3** and **4.6**, determine the total pressure:

$$\begin{aligned} DTP_{s-c} &= C_s \times VP_s &= -0.06(1.19) &= \underline{-0.07 \text{ inches wg}} \\ DTP_{b-c} &= C_b \times VP_b &= 0.11(1.08) &= \underline{0.12 \text{ inches wg}} \end{aligned}$$

Using **Equations 4.4** and **4.7**, determine the static pressure:

$$\begin{aligned} DSP_{s-c} &= (C_s - 1)VP_s + VP_c = (-0.06 - 1)(1.19) + 1.07 = \underline{-0.19 \text{ inches wg}} \\ DSP_{b-c} &= (C_b - 1)VP_b + VP_c = (0.11 - 1)(1.08) + 1.0 = \underline{0.11 \text{ inches wg}} \end{aligned}$$

Determine the section pressure losses:

Section 1:

Hood (entrance) Loss

From **Equation 4.8** $DTP_{hl} = C_{hl} \times VP_s$, where $C_{hl} = 0.25$ for a 90° tapered hood from an entrance loss chart (see **Appendix A.9.5**).

$$DTP_{hl} = 0.25(1.19) = \underline{0.30 \text{ inches wg}}$$

From **Equation 4.10:**

$$DSP_{hl} = (C_{hl} + 1)VP_s = (0.25 + 1) \times (1.19) = \underline{1.49 \text{ inches wg}}$$

Duct Friction Loss

The duct pressure drop per 100 ft. is calculated from **Equation 1.6:**

$$\begin{aligned} DP_{duct 1/100 \text{ ft}} &= 2.56(1/21)^{1.18} (4,365/1,000)^{1.8} \\ &= \underline{1.00 \text{ inches wg/100 ft}} \end{aligned}$$

Therefore:

$$DTP_{duct 1} = (1.00 \text{ inch wg/100 ft}) \times 80 \text{ ft} = \underline{0.80 \text{ inches wg}}$$

The duct static pressure is determined from:

$$DSP_{duct 1} = DTP_{duct 1} = \underline{0.80 \text{ inches wg}}$$

Losses for Entire Section

The straight-through (upstream) branch losses from previous calculations are:

$$\begin{aligned} DTP_{s-c} &= -0.07 \text{ inches wg} \\ DSP_{s-c} &= -0.19 \text{ inches wg} \end{aligned}$$

The total pressure loss for Section 1 is:

$$\begin{aligned} DTP_{Section\ 1} &= DTP_{h1} + DTP_{duct\ 1} + DTP_{s-c} = 0.30 + 0.80 + (-0.07) \\ &= \underline{1.03 \text{ inches wg}} \end{aligned}$$

The static pressure loss for Section 1 is:

$$\begin{aligned} DSP_{Section\ 1} &= DSP_{h1} + DSP_{duct\ 1} + DSP_{s-c} = 1.49 + 0.80 + (-0.19) \\ &= \underline{2.10 \text{ inches wg}} \end{aligned}$$

Section 2:

Plain Duct End (entrance) Loss

$$DTP_{h2} = C_{h2} \times VP_b$$

where $C_{h2} = 0.93$ for a plain duct entrance loss (see **Appendix A.9.5**)

$$= 0.93(1.08) = 1.00 \text{ inches wg}$$

$$DSP_{h2} = (C_{h2} + 1)VP_b = (0.93 + 1) \times (1.08) = \underline{2.08 \text{ inches wg}}$$

Duct Friction Loss

The duct pressure drop per 100 ft. for section 2 is:

$$\begin{aligned} DP_{duct\ 2}/100\ ft &= 2.56(1/10.5)^{1.18} (4,158/1,000)^{1.8} \\ &= \underline{2.08 \text{ inches wg}/100\ ft} \end{aligned}$$

Therefore:

$$DTP_{duct\ 2} = (2.08 \text{ inch wg}/100\ ft) \times 50\ ft = \underline{1.04 \text{ inches wg}}$$

The duct static pressure is determined from:

$$DSP_{duct\ 2} = DTP_{duct\ 2} = \underline{1.04 \text{ inches wg}}$$

Losses for Entire Section

The branch losses from previous calculations are:

$$\begin{aligned} DTP_{b-c} &= 0.12 \text{ inches wg} \\ DSP_{b-c} &= 0.11 \text{ inches wg} \end{aligned}$$

The total pressure loss for Section 2 is:

$$\begin{aligned} DTP_{Section\ 2} &= DTP_{h2} + DTP_{duct\ 2} + DTP_{b-c} = 1.00 + 1.04 + 0.12 \\ &= \underline{2.16\ inches\ wg} \end{aligned}$$

The static pressure loss for Section 2 is:

$$\begin{aligned} DSP_{Section\ 2} &= DSP_{h2} + DSP_{duct\ 2} + DSP_{b-c} = 2.08 + 1.04 + 0.11 \\ &= \underline{3.23\ inches\ wg} \end{aligned}$$

Section 3:

Duct Friction Loss

The duct pressure drop per 100 ft. for section 3 is:

$$\begin{aligned} DP_{duct\ 3/100\ ft} &= 2.56(1/24)^{1.18} (4,138/1,000)^{1.8} \\ &= \underline{0.78\ inches\ wg/100\ ft} \end{aligned}$$

Therefore:

$$DTP_{duct\ 3} = (0.78\ inch\ wg/100\ ft) \times 80\ ft = \underline{0.62\ inches\ wg}$$

The duct static pressure is determined from:

$$DSP_{duct\ 3} = DTP_{duct\ 3} = \underline{0.62\ inches\ wg}$$

Collector Loss

$$\begin{aligned} DTP_{collector} &= 5.00\ inches\ wg \\ DSP_{collector} &= 5.00\ inches\ wg \end{aligned}$$

Losses for Entire Section

The total pressure loss for Section 3 is:

$$DTP_{Section\ 3} = DTP_{duct\ 3} + DTP_{collector} = 0.62 + 5.00 = \underline{5.62\ inches\ wg}$$

The static pressure loss for Section 3 is:

$$DSP_{Section\ 3} = DSP_{duct\ 3} + DSP_{collector} = 0.62 + 5.00 = \underline{5.62\ inches\ wg}$$

The preceding pressure loss calculations are summarized in **Table 5.4**.

Table 5.4
Pressure Losses

Section	Inlet			Duct	Fitting	Branch			Other		Section Total	
	Loss Coeff.	<i>DTP</i> (in wg)	<i>DSP</i> (in wg)	<i>DP</i> (in wg)	<i>DP</i> (in wg)	Loss Coeff.	<i>DTP</i> (in wg)	<i>DSP</i> (in wg)	Loss Coeff.	<i>DP</i> (in wg)	<i>DTP</i> (in wg)	<i>DSP</i> (in wg)
1	0.25	0.30	1.49	0.80		-0.06	-0.07	-0.19			1.03	2.11
2	0.93	1.00	2.08	1.04		0.11	0.12	0.11			2.16	3.23
3				0.62						5.0	5.62	5.62

Part 3: Determine the path excess pressures and fan total/static pressure.

Determine the design leg of the exhaust system (leg with the most resistance):

$$\begin{aligned} \text{Path (1-3): } \quad DTP &= 1.03 + 5.62 = \underline{6.65 \text{ inch wg}} \\ DSP &= 2.11 + 5.62 = \underline{7.73 \text{ inches wg}} \end{aligned}$$

$$\begin{aligned} \text{Path (2-3): } \quad DTP &= 2.16 + 5.62 = \underline{7.78 \text{ inches wg}} \\ DSP &= 3.23 + 5.62 = \underline{8.85 \text{ inches wg}} \end{aligned}$$

The pressure requirement is greatest in path (2-3), so this is the design leg of the exhaust system.

Calculate the excess pressure (the difference in pressure between the design leg and any other leg of the system):

Excess total pressure:

$$\text{Path (1-3): } 7.78 - 6.65 = \underline{1.13 \text{ inches wg}}$$

Section 6.1 we see how to balance a system so that the excess pressure goes to zero in non-design legs of the system.

Calculate the Total and Static Pressure of the Fan:

Recall that the fan discharge velocity is 4,000 fpm. Since the only energy left on the discharge side is velocity pressure, the exit total pressure is equal to the velocity pressure.

$$\begin{aligned} TP_{exit} &= VP_{exit} \\ &= \left(\frac{4,000}{4,005} \right)^2 = 1.07 \text{ inches wg} \end{aligned}$$

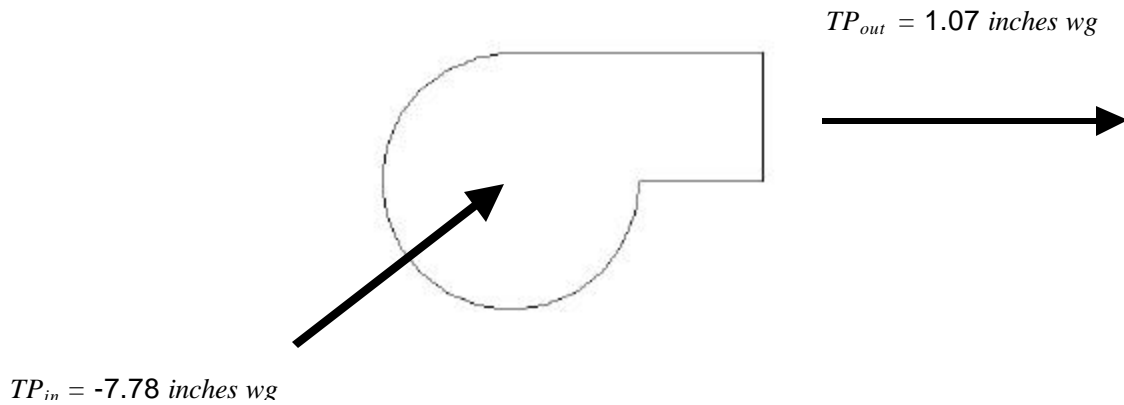


Figure 5.7
Fan

$$Fan \ TP = TP_{out} - TP_{in} = 1.07 - (-7.78) = \underline{8.85 \text{ inches wg}}$$

$$Fan \ SP = SP_{out} - SP_{in} - VP_{in} = 0 - (-8.85) - 1.07 = \underline{7.78 \text{ inches wg}}$$

From a fan manufacturer's data at standard conditions, the fan needs to be sized for 8.85 inches wg total pressure at 13,000 cfm or sized for 7.78 inches wg static pressure at 13,000 cfm.

Since the volume of flow is actually at 400°F and the elevation is at 5000 ft., we need to correct the Fan Total and Static Pressure for the lighter density. These corrections for nonstandard conditions can be expressed by the following:

$$Fan \ TP_{actual} = Fan \ TP \times (\text{density correction factor})$$

$$Fan \ SP_{actual} = Fan \ SP \times (\text{density correction factor})$$

Using the nonstandard temperature of 400°F, barometric pressure of 24.90 inches Hg, and an elevation of 5,000 feet above sea level, the density correction factor from **Appendix A.1.5** in **Table A.4** is 0.51.

$$Fan \ TP_{actual} = 8.85(0.51) = \underline{4.51 \text{ inches wg}}$$

$$Fan \ SP_{actual} = 7.78(0.51) = \underline{3.97 \text{ inches wg}}$$