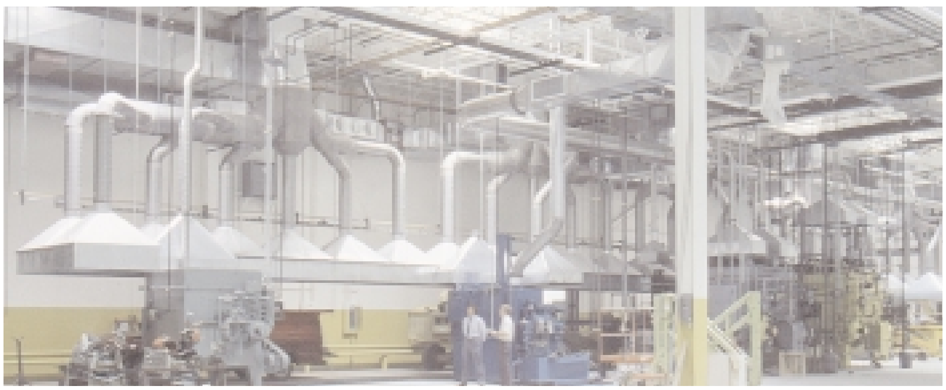
Design Advisory #5: CAS-DA5-2003

Exhaust and Return Duct Systems Are Not Equal!



This large exhaust duct system being used for fume removal, required heavier gauge and smaller diameter ductwork along with high-pressure blowers.

The laws of conservation of mass and energy apply equally to both exhaust and return air systems, but this is where the similarity ends. Exhaust duct systems are generally designed to convey contaminated air from industrial process and work spaces to the outdoors, while return air systems are generally designed to circulate "spent" air through a conditioning unit for reintroduction back into the conditioned space.

"Chapter 4: Airflow Fundamentals for Exhaust Duct Systems" of McGill AirFlow's *Duct System Design Guide* is the first of three chapters that address design approaches for negative pressure duct system design. It is important to keep in mind while reading these chapters: **Exhaust and Return Duct Systems Are Not Equal!**

Exhaust systems use capture and carrying airflow velocities to properly convey contaminants. Duct sizes in exhaust systems are generally small and of heavier gauge material in order to sustain the pressures created by airflow velocities exceeding 3000 fpm, erosion associated with particulates, and corrosion associated with chemicals in the air stream. High-pressure blowers are needed to "pull" the air through the small diameter ductwork, especially when the air is being processed through cleaning equipment. Long radius elbows and 30-to-45 degree converging flow fittings are incorporated to minimize pressure loss and to improve particulate flow.

Return systems are generally based on low-pressure design considerations to minimize operating costs. Return system duct diameters are much larger than exhaust system duct diameters since airflow velocities in return systems are below 2000 fpm. Return system fittings often incorporate short radius elbows and 90-degree converging fittings due to space constraints in commercial buildings. However, less efficient construction of return systems does not necessarily lessen system performance.

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CHAPTER 4: Airflow Fundamentals for Exhaust Duct Systems

4.1 Overview

This chapter presents basic airflow principles and equations that may be applied to both return and exhaust duct systems. Students or novice designers should read and study this material thoroughly before proceeding with **Chapter 5** or **Chapter 6**. Experienced designers may find a review of these principles helpful. Those who are comfortable with their knowledge of airflow fundamentals may proceed to **Chapter 5**. Whatever the level of experience, the reader should find the material about derivations in **Appendix A.3** interesting and informative.

The two fundamental concepts, which govern the flow of air in ducts, the laws of conservation of mass and conservation of energy, are discussed in **Chapter 1**. From these principles, the continuity and pressure equations for supply systems were derived. These same basic equations can be used to derive the continuity and pressure equations for return and exhaust systems.

4.2 Conservation of Mass

4.2.1 Continuity Equation

The volume flow rate, velocity and area are related as shown in the continuity **Equation 1.1**. Knowing any two of these properties, the equation can be solved to yield the value of the third. **Sample Problems 1-1 through 1-3** illustrate use of the continuity equation.

4.2.2 Converging Flows

According to the law of conservation of mass, the volume flow rate before a flow convergence is equal to the sum of the volume flows before the convergence if the density is constant. **Figure 4.1** and **Equation 4.1** illustrate this point.

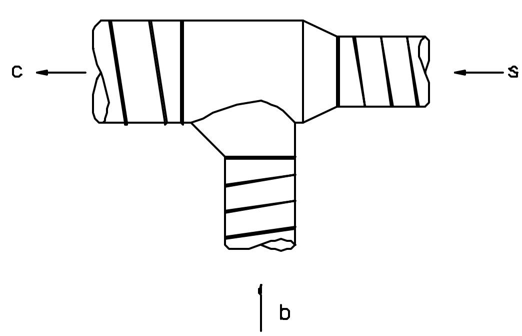


Figure 4.1
Converging Flow

$$Q_c = Q_b + Q_s$$ **Equation 4.1**

where:

Q_c = Common (downstream) volume flow rate (*cfm*)

Q_s = Straight-through (upstream) volume flow rate (*cfm*)

Q_b = Branch volume flow rate (*cfm*)

Sample Problem 4-1

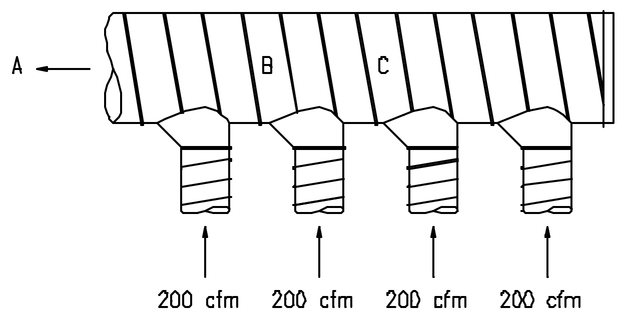


Figure 4.2
Multiple Converging Flow

*The system segment shown in **Figure 4.2** has four inlets, each delivering 200 cfm. What is the volume flow rate at points A, B and C?*

Answer: A = 800 *cfm*; B = 600 *cfm*; C = 400 *cfm*

The total volume flow rate at any point is simply the sum of all the upstream volume flow rates. The volume flow rate of all branches and/or trunks of any system can be determined in this way and combined to obtain the total volume flow rate of the system.

4.3 Conservation of Energy

The law of conservation of energy is discussed in **Section 1.3.** It states that the total energy per unit volume of air flowing in a duct system is equal to the sum of the static energy, kinetic energy and potential energy. **Equations 1.3** and **1.4** describe these relationships in terms of pressure and pressure losses. A thorough discussion of total, static and velocity pressure is given in **Sections 1.3.1, 1.3.2** and **1.3.3**.

Sign Convention

When a **total or static pressure** measurement is expressed as a positive number, it means that pressure is greater than the local atmospheric pressure. More typical for return and exhaust systems, the total or static pressure is negative which indicates the that pressure is less than the local atmospheric pressure.

By convention, positive changes in total or static pressure represent losses, and negative changes represent regains or increases. This is true for negative pressure return and exhaust systems as well. For example, if the total or static pressure change as air flows from point A to point B in a system is a positive number, then there is a total or static pressure loss between A and B, and the total or static pressure at A will be greater than the static pressure at B. Conversely, if the total or static pressure change as air flows between these points is negative, the total or static pressure at B will be greater than the static pressure at A, even though it may still be negative. Remember that a lower negative number numerically represents a higher pressure. For example, as the air flows from point A to point B, if the static pressure at A is - 5 inches wg and the static pressure at B is - 3 inches wg, then the static pressure at B is higher than the static pressure at A but the pressure loss is negative (ΔSP_{A-B} = - 5 minus -3 which equals -2 inches wg). This can happen if velocity pressure is converted to static pressure, thus increasing the static pressure at B. Normally, the pressure drop will be positive and the pressure B downstream of A will be lower.

Velocity pressure on the other hand is a vector quantity (requires direction) and is always a positive number even though the sign convention for changes in velocity pressure is the same as that described for total or static pressure. Velocity (and thus velocity pressure) can increase or decrease as the air flows toward the fan. Velocity must increase if the duct diameter (area) is reduced without a corresponding reduction in air flow volume. Similarly, the velocity must decrease if the air flow volume is reduced without a corresponding reduction in duct diameter. Thus, the velocity and the velocity pressure in a duct system are constantly changing. Use **Equation 1.5** to calculate the velocity pressure.

4.4 Pressure Losses

Total pressure represents the energy of the air flowing in a duct system. The total pressure continually increases as the air moves from the inlets, through the duct system, reaching its maximum value at the fan. Total pressure losses represent the irreversible conversion of static and kinetic energy to internal energy in the form of heat or flow separation. These losses are classified as either friction losses or dynamic losses.

4.4.1 Pressure Loss in Duct (Friction Loss)

Friction losses are produced whenever moving air flows in contact with a fixed boundary. These are discussed in **Section 1.4**. Dynamic losses are the result of turbulence or change in size, shape, direction, or volume flow rate in a duct system.

4.4.2 Pressure Loss in Return or Exhaust Fittings (Dynamic Losses)

Dynamic losses will result whenever the direction or volume of air flowing in a duct is altered or when the size or shape of the duct carrying the air is altered. Fittings of any type will produce dynamic losses. The dynamic loss of a fitting is generally proportional to the severity of the airflow disturbance. A smooth, large radius elbow, for example, will have a much lower dynamic loss than a mitered (two-piece) sharp-bend elbow. Similarly, a 45E branch fitting will usually have lower dynamic losses than a straight 90E tee branch. The fittings are characterized by a dimensionless parameter know as the loss coefficient which is discussed in **Section 1.5.1.**

Elbows

Table 4.1 shows typical loss coefficients for 8-inch diameter elbows of various constructions. Mitered elbows with or without vanes should be avoided in exhaust designs. For a complete discussion of elbows and their associated pressure losses, see **Section 1.5.2**.

Table 4.1
Loss Coefficient Comparisons for Abrupt-Turn Fittings

90E Elbows, 8-inch Diameter	
Fitting	**Loss Coefficient**[1]
Flat-back	0.07
Die-Stamped/Pressed, 1.5 Centerline Radius	0.11
Seven-Gored, 2.5 Centerline Radius	0.10
Five-Gored, 1.5 Centerline Radius	0.22
Mitered with Turning Vanes	0.52
Mitered without Turning Vanes	1.24

[1]For elbows: ***DTP = DSP = C x VP***

Converging-Flow Branches

The pressure losses in converging-flow fittings are somewhat more complicated than elbows, for two reasons: (1) there are multiple flow paths, (2) there will almost always be velocity changes as air flow volumes combine and sizes change, (3) the area ratios must be considered, (4) the volume flow rate ratios must be considered, and (5) the fitting type and included angle must be considered.

First, consider the case of air flowing from the upstream to the downstream. Referring to **Figure 4.1**, this is from **s** (straight-through) to **c** (common). (Refer to **Appendix A.1.1** for clarification of upstream and downstream.)

As is the case for elbows, loss coefficients are determined experimentally for converging-flow fittings. However, it is now necessary to specify which flow paths the equation parameters refer to. By definition:

$$C_s = \frac{DTP_{s-c}}{VP_s} \qquad \textbf{Equation 4.2}$$

where:

C_s = Straight-through (upstream) loss coefficient

DTP_{s-c} = Total pressure loss, straight-through (upstream) to common (downstream) (*inches wg*)

VP_s	=	Straight-through (upstream) velocity pressure (*inches wg*)

Rewriting in terms of total pressure loss:

$$DTP_{s-c} = C_s \text{ x } V$$ **Equation 4.3**

Therefore, the total pressure loss of air flowing from the straight-through section of a converging-flow fitting is directly proportional to the straight-through loss coefficient and velocity pressure. For duct and elbows, the total pressure loss is always equal to the static pressure loss, because there is no change in velocity. However, converging-flow fittings almost always have velocity changes associated with them. If DVP is not zero, then the total and static pressure losses cannot be equal.

For converging-flow fittings, the static pressure loss of air flowing from upstream to downstream can be determined (remembering: $DTP = DSP + DVP = C \text{ x } VP$) from **Equation 4.4**.

$$DSP_{s-c} = VP_s\,(C_s - 1) + VP_c$$ **Equation 4.4**

where:

DSP_{s-c}	=	Static pressure loss, straight-through (upstream) to common (downstream) (*inches wg*)
VP_c	=	Common (downstream) velocity pressure (*inches wg*)

The derivation of this equation is shown in **Appendix A.3.7**.

Similarly, for the branch side of a converging fitting, the airflow is from branch to common (downstream). Referring to **Figure 4.1**, this is from *b* to *c*.

$$C_b = \frac{DTP_{b-c}}{VP_b}$$ **Equation 4.5**

where:

C_b	=	Branch loss coefficient
DTP_{b-c}	=	Total pressure loss, branch to common (downstream) (*inches wg*)
VP_b	=	Branch velocity pressure (*inches wg*)

Rewriting in terms of total pressure loss:

$$DTP_{b-c} = C_b \text{ x } VP_b$$ **Equation 4.6**

Therefore, the total pressure loss of air flowing from the branch of a converging-flow fitting is directly proportional to the branch loss coefficient and the branch velocity pressure. Using a similar derivation to the straight-through for the branch, the static pressure loss can be calculated from:

$$DSP_{b-c} = VP_b(C_b - 1) + VP_c \qquad \textbf{Equation 4.7}$$

where:

DSP_{b-c} = Static pressure loss, branch to common (downstream) (*inches wg*)

VP_b = Branch velocity pressure (*inches wg*)

The derivation of this equation is also shown in **Appendix A.3.7**.

Before looking at an example, look at the parameters for a typical fitting, as shown in **Figure 4.3**. Note that each of the straight-through (upstream), common (downstream), and branch junctions of the fitting consist of a number of variables such as density, diameter, mass flow rate, area, velocity, volume flow rate, and angle of entrance of the branch into the fitting body.

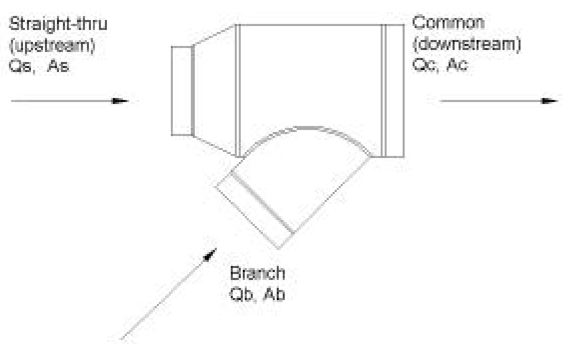


Figure 4.3
Definition of Parameters for a Typical Fitting

When analyzing fittings in an exhaust system, note that the following factors affect the converging fitting loss coefficient: fitting type and included angle, area ratio (A_s / A_c), area ratio (A_b / A_c), volume ratio (Q_b / Q_c) where:

A_s / A_c = Straight-through to common (upstream-to-downstream) area ratio

A_b / A_c = Branch-to-common (downstream) area ratio

Q_b / Q_c = Branch-to-common (downstream) volume flow rate ratio

Included angle = Angle formed by the centerlines of the straight-through (upstream) and branch

Each converging-flow fitting has two loss coefficients (C_b and C_s). Although curves of the loss coefficients are available with the ordinate of graph being C_b or C_s, and the abscissa of the graphs being Q_b / Q_c, it is much easier to use the table format given in the ASHRAE Duct Fitting Database Program (reference 9.1.12). The available fittings are 30E and 45E laterals, 90E tees, Y-branches, bullhead tees with and without turning vanes, heel-tapped elbows, and capped fittings.

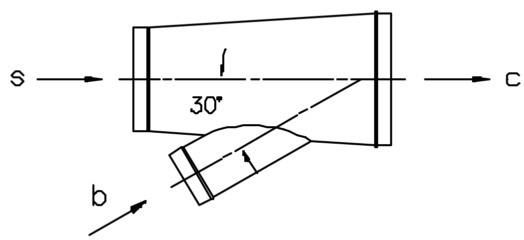
Since they are symmetrical, the loss coefficients for Y-branches, bullhead tees with vanes, and bullhead tees without vanes are calculated with the following rules:

1. C_s is always for the side with the larger dimension when the sides are not equal. C_b is for the smaller side when the sides are not equal ($A_s > A_b$).

2. C_b is for the smaller volume flow rate when the sides are equal ($A_s = A_b$). C_s is for the side with the larger volume flow rate when the sides are equal.

Sample Problem 4-2

What are the static and total pressure losses for flow from s to c and b to c in the 30E lateral shown below?



Q_s = 10,500 *cfm* $\quad$ Q_b = 2,500 *cfm* $\quad$ Q_c = 13,000 *cfm*
D_s = 25 *inches* $\quad$ D_b = 12 *inches* $\quad$ D_c = 27 *inches*

Answer: From **Equation 4.1**

$Q_c = Q_s + Q_b$ = 10,500 + 2,500 = $\underline{13{,}000\ cfm}$

From:

$$A = \pi D^2/576, \textit{square feet}$$

$$A_s = \pi 25^2/576 = \underline{3.41\ \textit{square feet}}$$
$$A_b = \pi 12^2/576 = \underline{0.79\ \textit{square feet}}$$
$$A_c = \pi 27^2/576 = \underline{3.98\ \textit{square feet}}$$

Area and volume flow rate ratios:

A_s / A_c = 3.41/3.98 = $\underline{0.86}$

A_b / A_c = 0.79/3.98 = $\underline{0.20}$

Q_b / Q_c = 2,500/13,000 = $\underline{0.19}$

Q_s / Q_c = 10,500/13,000 = $\underline{0.81}$

Reference: ASHRAE Duct Fitting Database Number ED5-1

Main and branch loss coefficients:

$C_s = \underline{-0.1}$ *(Main)*

$C_b = \underline{0.12}$ *(Branch)*

Velocities in each section from:

$$V_s = Q_s / A_s = 10{,}500/3.41 = \underline{3{,}079\ fpm}$$
$$V_b = Q_b / A_b = 2{,}500/0.79 = \underline{3{,}183\ fpm}$$
$$V_c = Q_c / A_c = 13{,}000/3.98 = \underline{3{,}266\ fpm}$$

Velocity pressure in each section from **Equation 4.5**:

$$VP_s = 0.075(V_s/1{,}097)^2 = 0.075(3{,}079/1{,}097)^2 = \underline{0.59\ inch\ wg}$$
$$VP_b = 0.075(V_b/1{,}097)^2 = 0.075(3{,}183/1{,}097)^2 = \underline{0.63\ inch\ wg}$$
$$VP_c = 0.075(V_c/1{,}097)^2 = 0.075(3{,}266/1{,}097)^2 = \underline{0.66\ inch\ wg}$$

Total pressure drop, straight-through to common from **Equation 4.3**:

$$\Delta TP_{s-c} = C_s \, x \, VP_s = -0.1 \text{ x } 0.59 = \underline{-0.06\ inch\ wg}$$

Total pressure drop, branch to common from **Equation 4.6**:

$$\Delta TP_{b-c} = C_b \, x \, VP_b = 0.2 \text{ x } 0.63 = \underline{0.13\ inch\ wg}$$

Static pressure drop, straight-through from **Equation 4.4**:

$$\Delta SP_{s-c} = VP_s(C_s - 1) + VP_c = 0.59(-0.1-1) + 0.66$$
$$= \underline{0.01\ inch\ wg}$$

Static pressure drops from **Equation 4.7**:

$$\Delta SP_{b-c} = VP_b(C_b - 1) + VP_c = 0.63(0.12-1) + 0.66$$
$$= \underline{0.11\ inch\ wg}$$

Entrances

Entrances into an exhaust system, such as hoods, are available in many different sizes and shapes. Once a particular hood is selected, the designer must determine the entrance. The hood entrance loss is one component in determining the total system resistance or Fan total pressure required. The following is a single tapered hood.

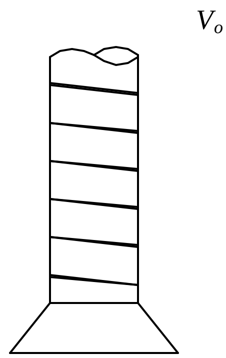


Figure 4.4
Hood

In **Figure 4.4**, a hood is shown with the exhaust stream entering from the surrounding atmosphere. Attached to the hood is a duct, which carries the air away from the hood. The basic equation for determining the hood total pressure loss is given by the following:

$$DTP_h = C_h \, x \, VP_o \qquad \textbf{Equation 4.8}$$

where:

VP_o	=	Velocity pressure in the duct (*inches wg*)
C_h	=	Hood loss coefficient

In **Equation 4.8**, the total pressure loss of the hood can be calculated knowing the hood loss coefficient and the velocity pressure in the duct connected to the hood. The static pressure relationship for this hood would be given by the following:

$$DSP_h = C_h VP_o - DVP_{0-o} \qquad \textbf{Equation 4.9}$$

The air surrounding the hood is at atmospheric pressure and the air velocity is assumed to be zero, then **Equation 4.9** reduces to the following:

$$DSP_h = (C_h + 1) VP_o \qquad \textbf{Equation 4.10}$$

With **Equations 4.8** and **4.10**, the designer can calculate the total and static pressure losses of the hood. Derivations are found in **Appendix A.3.9** for pressure loss equations for single and compound hoods.

The ASHRAE Duct Fitting Database Program (**Appendix A.8.2**) presents loss coefficients of various types of intakes. For example, a plain duct end type of hood shows a value of C_h = 2.03 for fitting ED1-7. Other hood loss coefficients can be found in the **Industrial Ventilation Manual** from the American Conference of Governmental Industrial Hygienists. From their manual an example is the standard grinder hood, which has a 0.65 loss coefficient. See **Appendix A.9.5**

Miscellaneous Fittings and Components

In addition to duct, fittings, and hoods, there are other components to consider when designing exhaust systems. These components produce a pressure loss and add to fan resistance. The following items are pressure loss components: cyclones, scrubbers, collectors, dryers, electrostatic precipitators, and filters.

Designers are directed to consult the manufacturer for total pressure losses at standard conditions or for a loss coefficient. Keep in mind that if a loss coefficient is given, the designer should be sure of what the reference velocity is (upstream, downstream, or other). The designer is directed to **Appendices A.9.2** and **A.9.5** for additional information concerning miscellaneous components.

4.4.3 *Nonstandard Conditions for Dynamic Losses*

Section 1.4.5 presented equations for correcting the calculated friction loss of a system for nonstandard conditions of temperature and or elevation. For dynamic total pressure losses a correction factor needs to be determined for changes in density. These factors are given in **Section 1.5.6**.

If the density-corrected velocity pressure is used to calculate all dynamic fitting losses, then no further corrections (except friction loss corrections) are required. Alternatively, if the pressure losses are calculated assuming standard conditions, the results can be corrected by multiplying by the ratio of actual density divided by standard density. For most applications, this ratio can be calculated as shown in **Equation 1.20**.