Design Advisory #1: CAS-DA1-2003

The Fundamentals of Duct System Design

Webster's Dictionary defines "fundamentalism" as "a movement or point of view marked by a rigid adherence to fundamental or basic principles". At McGill AirFlow we take pride in having participated since 1951 in the development and establishment of duct system design fundamentals. This has included active participation in technical and standards committees for professional organizations that serve the HVAC industry, such as ASHRAE, ASTM, AMCA, and SMACNA.

This is the first *Design Advisory* for McGill AirFlow's new ***Duct System Design Guide***, being issued in bimonthly installments to CAS subscribers. These releases address real-world applications and design topics. Submit comments and questions about these releases to our ***Ask An HVAC Expert*** service!

Chapter One of the ***Duct System Design Guide*** presents the fundamentals of duct system design — establishing a strong technical foundation that will aid in understanding and persevering over future topics.

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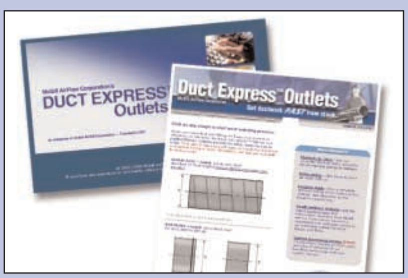

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CHAPTER 1: Airflow Fundamentals for Supply Duct Systems

1.1 Overview

This section presents basic airflow principles and equations for supply systems. Students or novice designers should read and study this material thoroughly before proceeding with the design sections. Experienced designers may find a review of these principles helpful. Those who are comfortable with their knowledge of airflow fundamentals may proceed to **Chapter 2**. 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, are the laws of conservation of mass and conservation of energy. From these principles are derived the basic continuity and pressure equations, which are the basis for duct system designs.

1.2 Conservation of Mass

The law of conservation of mass for a steady flow states that the mass flowing into a control volume must equal the mass flowing out of the control volume. For a one-dimensional flow of constant density, this mass flow is proportional to the product of the local average velocity and the cross-sectional area of the duct. **Appendix A.3.1** shows how these relationships are combined to derive the continuity equation.

1.2.1 Continuity Equation

The volume flow rate of air is the product of the cross-sectional area of the duct through which it flows and its average velocity. As an equation, this is written:

$$Q = A \times V \qquad \textbf{Equation 1.1}$$

where:

Q	=	Volume flow rate (cubic feet per minute or *cfm*)
A	=	Duct cross-sectional area (ft^2) ($=\pi D^2/576$ where D is diameter in inches)
V	=	Velocity (feet per minute or *fpm*)

The volume flow rate, velocity and area are related as shown in **Equation 1.1**. Knowing any two of these properties, the equation can be solved to yield the value of the third. The following sample problems illustrate the usefulness of the continuity equation.

Sample Problem 1-1

If the average velocity in a 20-inch diameter duct section is measured and found to be 1,700 feet per minute, what is the volume flow rate at that point?

Answer: $A = \pi \times (20^2) / 576 = 2.18\ ft^2$

$V = 1{,}700\ fpm$

$Q = A \times V = 2.18\ ft^2 \times 1{,}700\ fpm = \underline{3{,}706\ cfm}$

Sample Problem 1-2

If the volume flow rate in a section of 24-inch duct is 5,500 cfm, what will be the average velocity of the air at that point? What would be the velocity if the same volume of air were flowing through a 20-inch duct?

Answer: $A_{24} = \pi \times (24^2) / 576 = 3.14\ ft^2$

$Q = 5{,}500\ cfm$

$Q = A \times V$, therefore: $V = Q/A$

$V = (5{,}500\ cfm) / 3.14\ ft^2 = \underline{1{,}752\ fpm}$ (for 24-inch duct)

$A_{20} = \pi \times (20^2) / 576 = 2.18\ ft^2$

$V = (5{,}500\ cfm) / 2.18\ ft^2 = \underline{2{,}523\ fpm}$ (for 20-inch duct)

Sample Problem 1-3

If the volume flow rate in a section of duct is required to be 5,500 cfm, and it is desired to maintain a velocity of 2,000 fpm, what size duct will be required?

Answer: $V = 2{,}000\ fpm$

$Q = 5{,}500\ cfm$

$A = (5{,}500\ cfm) / (2{,}000\ fpm) = 2.75\ ft^2$

$D = (576 \times A/\pi)^{0.5} = (576 \times 2.75\ ft^2/\pi)^{0.5} = 22.45\ inches$

Use: $D = \underline{22\ inches}$

then:

$V_{actual} = (5{,}500\ cfm)/(\pi \times (22^2) / 576) = \underline{2083\ fpm}$

1.2.2 Diverging Flows

According to the law of conservation of mass, the volume flow rate before a flow divergence is equal to the sum of the volume flows after the divergence. **Figure 1.1** and **Equation 1.2** illustrate this point.

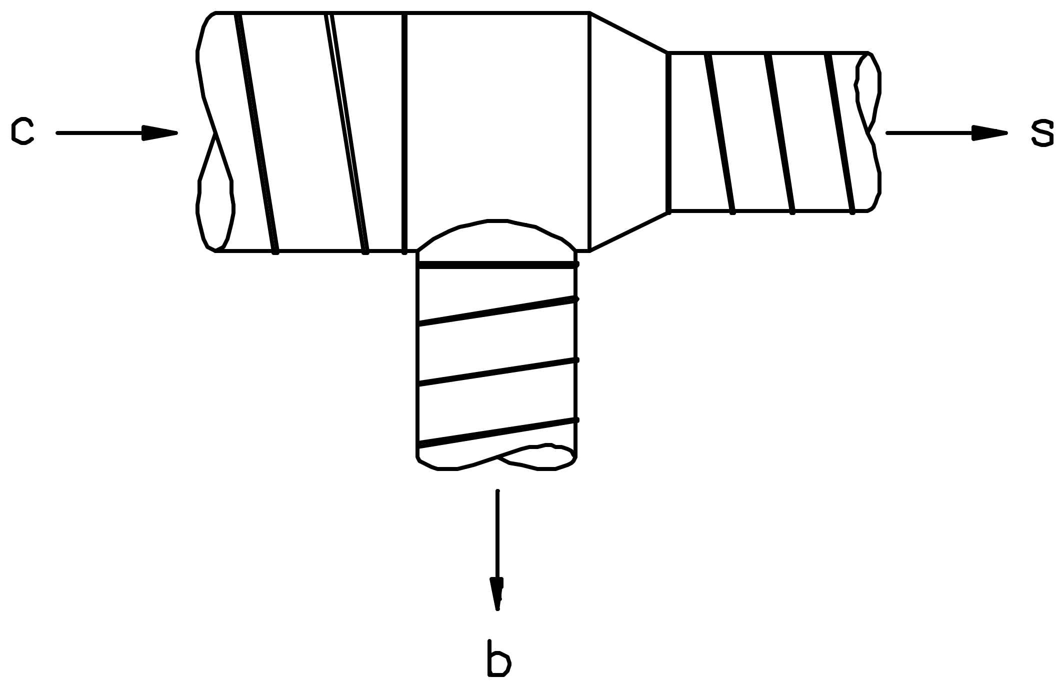

Figure 1.1
Diverging Flow

$$Q_c = Q_b + Q_s \qquad \textbf{Equation 1.2}$$

where:

Q_c	=	Common (upstream) volume flow rate (*cfm*)
Q_b	=	Branch volume flow rate (*cfm*)
Q_s	=	Straight-through (downstream) volume flow rate (*cfm*)

Sample Problem 1-4

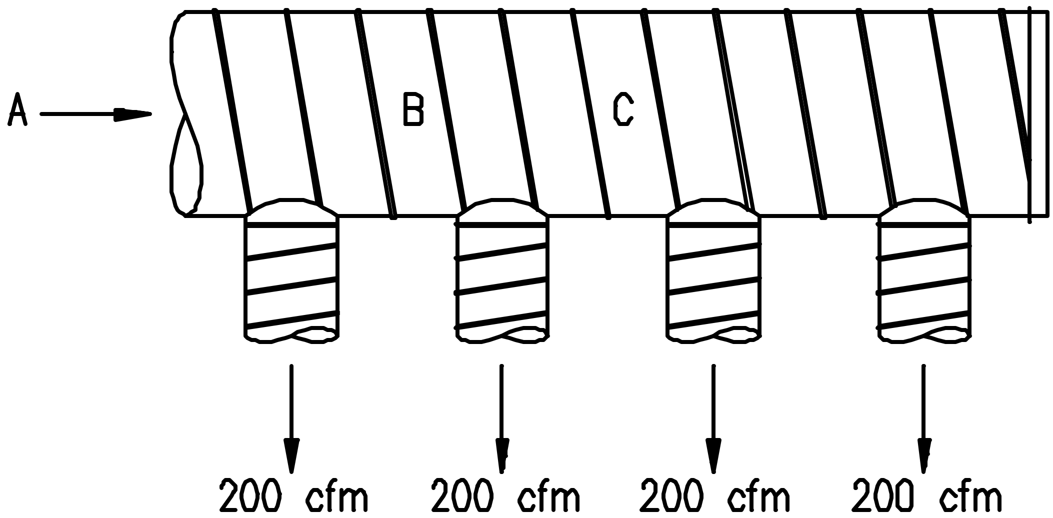

Figure 1.2
Multiple Diverging Flow

The system segment shown in Figure 1.2 has four outlets, 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 downstream 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.

1.3 Conservation of Energy

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.

When applied to airflow in ducts, the flow work or static energy is represented by the static pressure of the air, and the velocity pressure of the air represents the kinetic energy. Potential energy is due to elevation above a reference datum and is often negligible in HVAC duct design systems.

Consequently, the total pressure (or total energy) of air flowing in a duct system is generally equal to the sum of the static pressure and the velocity pressure. As an equation, this is written:

$$TP = SP + VP \qquad \textbf{Equation 1.3}$$

where:

TP = Total pressure

SP = Static pressure

VP = Velocity pressure

Furthermore, when elevation changes are negligible, from the law of conservation of energy, written for a steady, non-compressible flow for a fixed control volume, the change in total pressure between any two points of a system is equal to the sum of the change in static pressure between the points and the change in velocity pressure between the points. This relationship is represented in the following equation:

$$DTP = DSP + DVP \qquad \textbf{Equation 1.4}$$

Appendix A.3.2 shows the derivation of **Equation 1.4** from the general equation of the first law of thermodynamics.

Pressure (or pressure loss) is important to all duct designs and sizing methods. Many times, systems are sized to operate at a certain pressure or not in excess of a certain pressure. Higher pressure at the same volume flow rate means that more energy is required from the fan, and this will raise the operating cost.

The English unit most commonly used to describe pressure in a duct system is the inch of water gauge (*inch wg*). One pound per square inch (*psi*), the standard measure of atmospheric pressure, equals approximately 27.7 *inches wg*.

1.3.1 Static Pressure

Static pressure is a measure of the static energy of the air flowing in a duct system. It is static in that it can exist without a movement of the air stream. The air which fills a balloon is a good example of static pressure; it is exerted equally in all directions, and the magnitude of the pressure is reflected by the size of the balloon.

The atmospheric pressure of air is a static pressure. At sea level, this pressure is equal to approximately 14.7 pounds per square inch. For air to flow in a duct system, a pressure differential must exist. That is, energy must be imparted to the system (by a fan or air handling device) to raise the pressure above or below atmospheric pressure.

Air always flows from an area of higher pressure to an area of lower pressure. Because the static pressure is above atmospheric at a fan outlet, air will flow from the fan through any connecting ductwork until it reaches atmospheric pressure at the discharge. Because the static pressure is below atmospheric at a fan inlet, air will flow from the higher atmospheric pressure through an intake and any connecting ductwork until it reaches the area of lowest static pressure at the fan inlet. The first type of system is referred to as a *positive pressure* or *supply* air system, and the second type as a *negative pressure*, *exhaust*, or *return* air system.

Static Pressure Losses

The initial static pressure differential (from atmospheric) is produced by adding energy at the fan. This pressure differential is completely dissipated by losses as the air flows from the fan to the system discharge. Static pressure losses are caused by increases in velocity pressure as well as friction and dynamic losses.

Sign Convention

When a static pressure measurement is expressed as a positive number, it means the pressure is greater than the local atmospheric pressure. Negative static pressure measurements indicate a pressure less than local atmospheric pressure.

By convention, positive changes in static pressure represent losses, and negative changes represent regains or increases. For example, if the static pressure change as air flows from point A to point B in a system is a positive number, then there is a static pressure loss between points A and B, and the static pressure at A must be greater than the static pressure at B. Conversely, if the static pressure change as air flows between these points is negative, the static pressure at B must be greater than the static pressure at A.

1.3.2 Velocity Pressure

Velocity pressure is a measure of the kinetic energy of the air flowing in a duct system. It is directly proportional to the velocity of the air. For air at standard density (0.075 pounds per cubic foot), the relationship is:

$$VP = \rho \left(\frac{V}{1{,}097} \right)^2 = \left(\frac{V}{4{,}005} \right)^2 \qquad \textbf{Equation 1.5}$$

where:

VP = Velocity pressure (*inches wg*)

V = Air velocity (*fpm*)

ρ = Density (lb_m/ft^3)

Appendix A.3.3 provides a derivation of this relationship from the kinetic energy term. This derivation also provides an equation for determining velocity pressures at nonstandard densities. **Appendix A.1.6** provides a table of velocities and corresponding velocity pressures at standard conditions.

Velocity pressure is always a positive number, and the sign convention for changes in velocity pressure is the same as that described for static pressure.

From **Equation 1.1**, it can be seen that velocity must increase if the duct diameter (area) is reduced without a corresponding reduction in air volume. Similarly, the velocity must decrease if the air volume is reduced without a corresponding reduction in duct diameter. Thus, the velocity and the velocity pressure in a duct system are constantly changing.

1.3.3 Total Pressure

Total pressure represents the energy of the air flowing in a duct system. Because energy cannot be created or increased except by adding work or heat, there is no way to increase the total pressure once the air leaves the fan. The total pressure is at its maximum value at the fan outlet and must continually decrease as the air moves through the duct system toward the outlets. Total pressure losses represent the irreversible conversion of static and kinetic energy to internal energy in the form of heat. These losses are classified as either friction losses or dynamic losses.

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 changes in size, shape, direction, or volume flow rate in a duct system. These losses are discussed in **Section 1.5**.

Referring to **Equation 1.4**, note that if the decrease in velocity pressure between two points in a system is greater than the total pressure loss, the static pressure must increase to maintain the equality. Alternatively, an increase in velocity pressure will result in a reduction in static pressure, equal to the sum of the velocity pressure increase and the total pressure loss. When there is both a decrease in velocity and a reduction in static pressure, the total pressure will be reduced by the sum of these losses. These three concepts are illustrated in **Figure 1.3**.

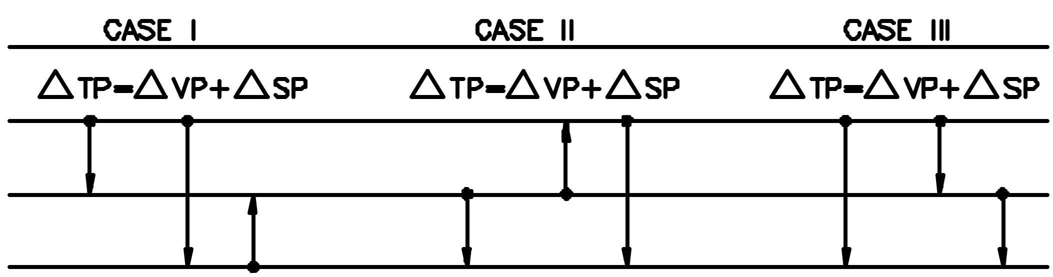

Figure 1.3
Conservation of Energy Relationship

1.4 Pressure Loss In Duct (Friction Loss)

When air flows through a duct, friction is generated between the flowing air and the stationary duct wall. Energy must be provided to overcome this friction, and any energy converted irreversibly to heat is known as a friction loss. The fan initially provides this energy in the form of pressure. The amount of pressure necessary to overcome the friction in any section of duct depends on (1) the length of the duct, (2) the diameter of the duct, (3) the velocity (or volume) of the air flowing in the duct, and (4) the friction factor of the duct.

The friction factor is a function of duct diameter, velocity, fluid viscosity, air density and surface roughness. For *nonstandard conditions*, see **Section 1.5**. The surface roughness can have a substantial impact on pressure loss, and this is discussed in **Appendix A.3.5**.

These factors are combined in the *Darcy* equation to yield the pressure loss, or the energy requirement for a particular section of duct. **Appendix A.3.4** discusses the use and application of the Darcy equation.

1.4.1 Round Duct

One of the most important and useful tools available to the designer of duct systems is a friction loss chart (see **Appendix A.4.1.1**). This chart is based on the Darcy equation, and combines duct diameter, velocity, volume flow rate and pressure loss. The chart is arranged in such a manner that, knowing any two of these properties (at standard conditions), it is possible to determine the other two.

The chart is arranged with pressure loss (per 100 feet of duct length) on the horizontal axis, volume flow rate on the vertical axis, duct diameter on diagonals sloping upward from left to right, and velocity on diagonals sloping downward from left to right.

Examination of this chart (or the Darcy equation) reveals several interesting air flow properties: (1) at a constant volume flow rate, reducing the duct diameter will increase the pressure loss; (2) to maintain a constant pressure loss in ducts of different size, larger volume flow rates require larger duct diameters; and (3) for a given duct diameter, larger volume flow rates will increase the pressure loss.

The following sample problems will give the reader a feel for these important relationships. Although there are many nomographs or duct calculators available to speed the calculation of duct friction loss problems, novice designers should use the friction loss charts to better visualize the relationships. The friction loss chart in the Appendix is approximated by **Equation 1.6.**

$$\frac{\Delta P}{100ft.} = 2.56\left(\frac{1}{D}\right)^{1.18}\left(\frac{V}{1000}\right)^{1.8} \qquad \textbf{Equation 1.6.}$$

where:

$\frac{\Delta P}{100ft.}$ = the friction loss per 100 ft of duct (*inches wg*)

D = the duct diameter (*inches*)

V = the velocity of the air flow in the duct (*fpm*)

Sample Problem 1-5

What is the friction loss of a 150-foot long section of 18-inch diameter duct, carrying 2,500 cfm? What is the air velocity in this duct?

Answer: From the friction loss chart, find the horizontal line that represents 2,500 *cfm*. Move across this line to the point where it intersects the diagonal line which represents an 18-inch diameter duct. From this point, drop down to the horizontal (pressure) axis and read the friction loss. This value is approximately 0.16 *inches wg*. This represents the pressure loss of a 100-foot section of 18-inch diameter duct carrying 2,500 *cfm*. To determine the pressure loss for a 150-foot duct section, it is necessary to multiply the 100-foot loss by a factor of 1.5. Therefore, the pressure loss is 0.24 *inches wg*.

At the intersection of the 2,500 *cfm* line and the 18-inch diameter line, locate the nearest velocity diagonal. The velocity is approximately 1,400 *fpm*.

Equation 1.6 could have also been used to solve this problem. To use **Equation 1.6** we must first calculate the velocity using **Equation 1.1**. To calculate the velocity, we have to determine the cross-sectional area of the 18-inch diameter duct. $A = \pi \times 18^2 / 576 = 1.77\ ft^2$. The calculated velocity is $Q/A = 2500/1.77 = 1412\ fpm$. The pressure loss per 100 ft is calculated as:

$$\frac{\Delta P}{100ft.} = 2.56 \left(\frac{1}{18}\right)^{1.18} \left(\frac{1412}{1000}\right)^{1.8} = \underline{0.16\ inches\ \ wg}$$

For a 150 *ft* section the pressure is 1.5 x 0.16 = 0.24 *inches wg*.

Sample Problem 1-6

Part of a system you have designed includes a 20-inch diameter, 500-foot duct run, carrying 3,000 cfm. You now discover there is only 16 inches of space in which to install this section. What will be the increase in pressure loss of the duct section if 16-inch duct is used in place of 20-inch?

Answer: From the friction loss chart or **Equation 1.6**, find the pressure loss per 100 *feet* for 3,000 *cfm* flowing through a 20-*inch* diameter duct. This value is 0.14 *inches wg*, or 0.70 *inches wg* for 500 *feet*. Similarly, the friction loss for 500 *feet* of 16-*inch* diameter duct carrying 3,000 *cfm* is found to be 1.95 *inches wg*.

Reducing this duct diameter by 4 inches results in an increase of pressure loss of 1.25 *inches wg*.

Sample Problem 1-7

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*An installed system includes a 20-inch diameter, 500-**foot** duct run carrying 3,000 cfm. Due to unanticipated conditions downstream, it has become necessary to increase the volume flow rate in this duct section to 3,600 cfm. What will be the impact on the pressure loss of the section?*

Answer: From Sample Problem 1-6, the pressure drop of this section, as installed, is 0.70 *inches wg*. To find the new pressure drop, move up the 20-*inch* diameter line until it intersects the 3,600 *cfm* volume flow rate line. At this point, read down to the friction loss axis and find that the new friction loss is 0.19 *inches wg* per 100 feet, or 0.95 *inches wg* for 500 *feet*. Therefore, this 20 percent increase in volume will result in a 36 percent increase in the pressure loss for the duct section.

Sample Problem 1-8

A system is being designed so that the pressure loss in all duct sections is equal to 0.2 inches wg per 100 feet. What size duct will be required to carry (a) 500 cfm; (b) 1,500 cfm; (c) 5,000 cfm; (d) 15,000 cfm?

Answer: From the friction loss chart, find the vertical line that represents 0.20 *inches wg* per 100 feet friction loss. Move up this line to the point where it intersects the horizontal line, which represents a volume flow rate of 500 *cfm*. At this point, locate the nearest diagonal line, representing duct diameter. It will either be 9-*inch* duct or 9.5-*inch* duct (if half-inch duct sizes are available).

Similarly, for the other volumes, find the line representing duct size, which is nearest to the intersection of the 0.20 *inches wg* per 100 feet friction loss line and the appropriate volume. At 1,500 *cfm*, 14-*inch* duct is required; at 5,000 *cfm*, 22-*inch* duct is required; and at 15,000 *cfm*, 33-*inch* duct is required. In each case, the duct will carry the specified volume with a pressure loss of approximately 0.20 *inches wg per 100 feet*.

1.4.2 Flat Oval Duct

Flat oval duct has the advantage of allowing a greater duct cross-sectional area to be accommodated in areas with reduced vertical clearances. **Figure 1.4** shows a typical cross section of flat oval duct. References in **Appendices A.1**, **A.2**, and **A.9** provide additional information about flat oval duct.

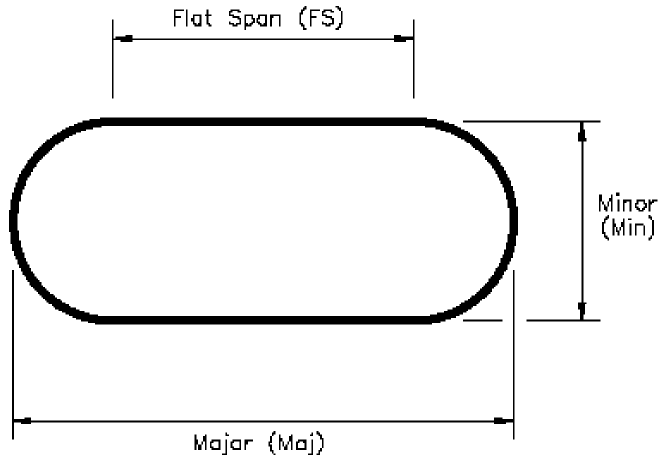


Figure 1.4
Flat Oval Dimensions

The Darcy equation is *not* applicable to flat oval duct, and there are no friction loss charts available for non-round duct shapes. To calculate the friction loss for flat oval duct, it is necessary to determine the equivalent round diameter of the flat oval size, and then determine the friction loss for the equivalent round duct.

The equivalent round diameter of flat oval duct is the diameter of round duct that has the same pressure loss per unit length, at the same volume flow rate, as the flat oval duct. **Equation 1.7** can be used to calculate the equivalent round diameter for flat oval duct with cross-sectional area A, and perimeter P. The equivalent round diameters for many standard sizes of flat oval duct are given in **Appendix A.1.3.2**.

$$D_{eq} = \frac{1.55\,(A\,)^{0.625}}{(P\,)^{0.25}} \qquad \textbf{Equation 1.7}$$

where:

D_{eq} = Equivalent round diameter (*ft*)

A = Flat oval cross-sectional area (ft^2)

P = Flat oval perimeter (*ft*)

The flat oval cross-sectional area is calculated using **Equation 1.8**.

$$A = \frac{(FS\ x\ min\,) + \frac{(\,\mathrm{p}\ x\ min^2\,)}{4}}{144} \qquad \textbf{Equation 1.8}$$

where:

A = Cross-sectional area (ft^2)

FS = Flat Span (*inches*) = maj - min

min = Minor axis (*inches*)

maj = Major axis (*inches*)

The perimeter of flat oval is calculated using **Equation 1.9**.

$$P = \frac{(\,\mathrm{p}\ x\ min\,) + (\,2\ x\ FS)}{12} \qquad \textbf{Equation 1.9}$$

where:

P = Flat oval perimeter (*ft*)

When calculating the air velocity in flat oval duct, it is necessary to use the actual cross-sectional area of the flat oval shape, not the area of the equivalent round duct. To use **Equation 1.6** however, the air velocity is calculated using the equivalent round diameter cross sectional area.

Sample Problem 1-9

A 12-inch x 45-inch flat oval duct is designed to carry 10,000 cfm. What is the pressure loss per 100 feet of this section? What is the velocity?

Answer: **Equations 1.8** and **1.9** are used to calculate the area and perimeter of the flat oval duct. For 12 x 45 duct, A = 3.54 ft^2 from: {(45 - 12) x 12 + (π x 12^2)/4}/144 and P = 8.64 *ft* from: {(π x 12) + (2 x (45-12))}/12. Substituting into **Equation 1.7**, D_{eq} = 1.99 *ft* . 24 inches. From the friction loss chart, the pressure loss of 10,000 *cfm* air flowing in a 24-inch round duct is <u>0.50 *inch wg* per 100 feet</u>.

The velocity can be calculated from **Equation 1.1**:

$$V = Q/A = 10{,}000 / 3.54 = \underline{2{,}825\ fpm}$$

Note that the velocity of the same air volume flowing in the 24-*inch* diameter round duct is 10,000 / 3.14, or 3,185 *fpm*.

Alternatively, to use **Equation 1.6**, we must first calculate the velocity assuming the cross-sectional area is determined from the equivalent round

$$A_{Deq} = \frac{(\pi \times 24^2)}{576} = \underline{3.14\ ft^2}$$

$$V_{Deq} = \frac{10000}{3.14} = \underline{3185\ fpm}$$

$$\frac{\Delta P}{100ft.} = 2.56\left(\frac{1}{24}\right)^{1.18}\left(\frac{3185}{1000}\right)^{1.8} = \underline{0.48\ inches\ wg}$$

Which is close to the 0.50 *incheswg* determined from the friction loss chart.

Sample Problem 1-10

In Sample ***Problem 1-6****, what size flat oval duct would be required in order to maintain the original (0.70 inches wg) pressure drop, and still fit within the 16-inch space allowance?*

Answer: In this situation, the available space will dictate the minor axis dimension of the flat oval duct. It is always advisable to allow at least 2 to 4 *inches* for the reinforcement, which may be required on any flat oval or rectangular duct product. Therefore, we will assume that the largest minor axis that can be accommodated is 12 *inches*.

Since we want to select a flat oval size which will have the same pressure loss as a 20-*inch* round duct, the major axis dimension can be determined by solving **Equation 1.7** with D_{eq} = 1.67 *feet* (20 *inches*), and *min* = 12 *inches*. Using **Equations 1.8** and **1.9** for determining A and P, as functions of the major axis dimension (*maj*) and the minor axis dimension (*min*). Unfortunately, this requires an iterative solution.

A simpler solution is to refer to the tables in **Appendix A.1.3.2**. These tables list the various available flat oval sizes and their respective equivalent round diameters. Since we already know that the minor axis must be 12 *inches*, we look for a flat oval size with a 12-*inch* minor axis and an equivalent round diameter of 20 *inches*. The required flat oval size is <u>12 *inches* x 31 *inches*</u>.

If it is determined that there is room for a 14-*inch* minor axis duct, the required size would be <u>14 *inches* x 27 *inches*</u> (D_{eq} = 20 *inches*).

1.4.3 Rectangular Duct

Rectangular duct is fabricated by breaking two individual sheets of sheet metal (called L-sections) that have the appropriate duct dimensions (side and side adjacent) and joining them together by one of several techniques. Rectangular duct is also used when height restrictions are employed in a duct design. **Equation 1.10** can be used to calculate the equivalent round diameter D_{eq} of rectangular duct. The equivalent round diameter D_{eq} for many standard sizes of rectangular duct are given in **Appendix 1.4.**

$$D_{eq} = \frac{1.30\,(ab)^{0.625}}{(a+b)^{0.250}} \qquad \textbf{Equation 1.10}$$

where:

D_{eq}	=	Equivalent round diameter (*inches*)
a	=	Duct side length (*inches*)
b	=	Other duct side length (*inches*)

When calculating air velocity in rectangular duct, it is necessary to use the actual cross-sectional area of the rectangular shape not the area of the equivalent round diameter.

Sample Problem 1-11

A 12-inch x 45-inch rectangular duct is designed to carry 10,000 cfm. What is the pressure loss per 100 feet of this section? What is the velocity?

Answer: Using **Equation 1.10**, D_{eq} = 24 *inches*. From the friction loss chart, the loss of 10,000 *cfm* air flowing in a 24-*inch* round duct is <u>0.50 *inch wg per 100 feet*.</u>

The velocity can be calculated from **Equation 1.1**:

$$V = \frac{Q}{A} = \frac{10000}{(12 \text{ x } 45)/144} = \underline{2{,}667 \ fpm}$$

Note that the velocity in the rectangular duct is less than in the flat oval with the same major and minor dimensions. (See **Sample Problem 1-9**)

1.4.4 Acoustically Lined and Double-wall Duct

Applying an inner liner or a perforated inner metal shell sandwiching insulation between an inner and outer wall, increases the surface roughness that air sees and thus increases the friction losses of duct. *Acoustically lined round and rectangular duct* consist of a single-wall duct with an internal insulation liner but no inner metal shell. *Double-wall duct* that is acoustically insulated consists of a solid outer shell, a thermal/acoustical insulation, and a metal inner liner (either solid or perforated). The inner dimensions of lined duct or the metal inner liner dimensions of double-wall duct, are the nominal duct size dimensions that are used to determine the cross-sectional area for airflow calculations. The single-wall dimensions of lined duct or outer shell dimensions of double-wall duct depend on the insulation thickness. For a 1-*inch* thick insulation, the dimensions are 2 *inches* larger than the inner dimensions of lined duct or metal inner liner dimensions.

Acoustically Lined Duct

Correction factors to the friction loss determined from the friction loss chart or for **Equation 1.6** have not been developed for internally insulated duct. Therefore the designer must use the Darcey equation as given in **Appendix A.3.4.** Assume an absolute roughness of e = 0.015.

Double-wall Duct

Corrections factors to the friction loss determined from the friction loss chart or **Equation 1.6** have been developed for when a perforated metal inner liner is used. **Figure 1.5** is a chart, which gives the correction factors. This information is repeated in **Appendix A.4.1.2.** Note that these corrections are a function of duct diameter and velocity. If the duct shape is flat oval or rectangular, use the equivalent round diameter based on the perforated metal inner liner dimensions. If the inner shell of the double-wall duct does not use perforated metal, use the same friction loss as a single-wall duct of the same dimensions as the metal inner shell.

Correction factor to be applied to the friction loss of single-wall duct
to calculate the friction loss of double-wall duct with a perforated metal inner liner

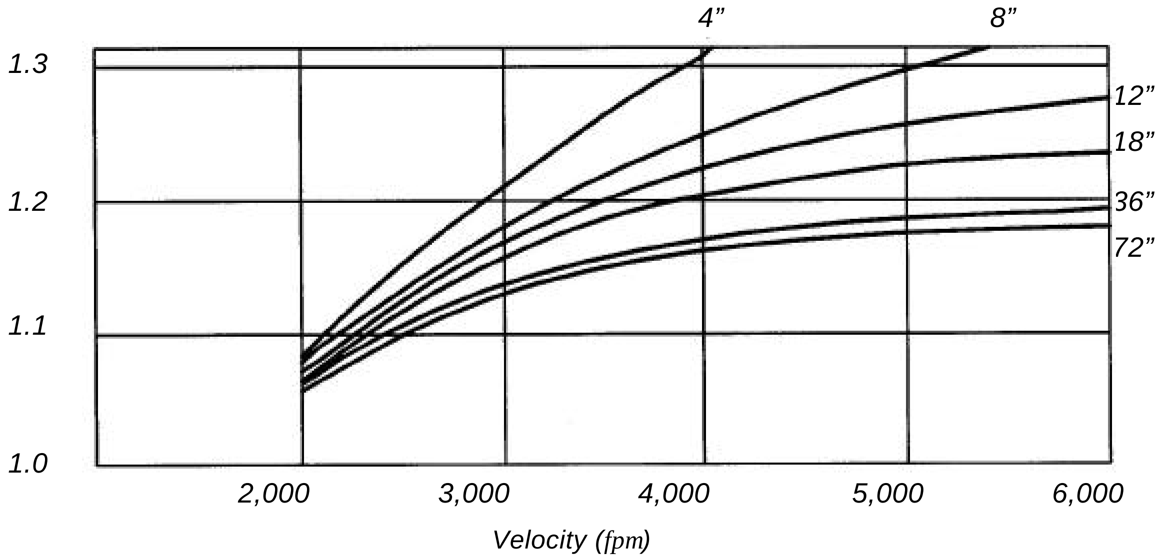


Figure 1.5
Correction Factors for Double-Wall Duct with Perforated Metal Inner Liner

When only thermal insulation is required, the metal inner liner may be specified as solid rather than perforated metal. In this case, the friction losses are identical to those for single-wall duct with a diameter equal to the metal inner liner diameter.

For acoustically insulated flat oval or rectangular duct with a perforated metal inner liner, use the correction factors of the equivalent round diameter and the actual velocity (based on the metal inner liner of the flat oval or rectangular cross section). The reference in **Appendix A.9.2** addresses friction losses for lined rectangular duct.

Sample Problem 1-12

What is the friction loss of a 100-foot section of 22-inch diameter double-wall duct (with perforated metal inner liner), carrying 8,000 cfm?

Answer: The pressure loss for 100 feet of 22-*inch*, single-wall duct, carrying 8,000 *cfm* is found from the friction loss chart to be 0.50 *inches wg*. The velocity is 3,000 *fpm*.

From **Figure 1.5**, the correction factor for 22-*inch* duct (interpolated) at 3,000 *fpm* is approximately1.16.Therefore; the pressure loss of this section of double-wall duct is 0.50 x 1.16, or 0.58 *inches wg*.

1.4.5 Nonstandard Conditions

All loss calculations thus far have been made assuming a standard air density of 0.075 *pounds per cubic foot*. When the actual design conditions vary appreciably from standard (i.e., temperature is " 30°*F* from 70°*F*, elevation above 1,500 *feet*, or moisture greater than 0.02 *pounds water per pound dry air*), the air density and viscosity will change. If the Darcy equation is used to calculate friction losses and the friction factor and velocity pressure are calculated using actual conditions, no additional corrections are necessary. If a nomograph or friction chart is used to calculate friction losses at standard conditions, correction factors should be applied.

The corrections for nonstandard conditions discussed above apply to duct friction losses only. Other corrections are applicable to the dynamic losses of fittings, as will be explained in the following section. For a more in-depth presentation of these and other correction factors, see Reference in **Appendix A.9.2**. Tables for determining correction factors are included in **Appendix A.1.5**.

A temperature correction factor, K_t , can be calculated as follows:

$$K_t = \left(\frac{530}{(T_a + 460)} \right)^{0.825} \qquad \textbf{Equation 1.11}$$

where:

K_t = Nonstandard temperature correction factor

T_a = Actual temperature of air in the duct (°F)

An elevation correction factor, K_e , can be calculated as follows:

$$K_e = [1 - (6.8754 \times 10^{-6})(Z)]^{4.73} \qquad \textbf{Equation 1.12a}$$

Equation 1.12a can also be written as follows:

$$K_e = \left(\frac{b}{29.921} \right)^{0.9} \qquad \textbf{Equation 1.12b}$$

where:

K_e = Nonstandard elevation correction factor

Z = Elevation above sea level (*feet*)

b = Actual barometric pressure (*inches Hg*)

When both a nonstandard temperature and a nonstandard elevation are present, the correction factors are multiplicative. As an equation:

$$K_f = K_t \times K_e \qquad \textbf{Equation 1.13}$$

where:

K_f = Total friction loss correction factor

The calculated duct friction pressure loss should be multiplied by the appropriate correction factor, K_t , K_e , or K_f , to obtain the actual pressure loss at the nonstandard conditions.

Sample Problem 1-13

The friction loss for a certain segment of a duct system is calculated to be 2.5 inches wg at standard conditions. What is the corrected friction loss if (a) the design temperature is 30°F; (b) the design temperature is 110°F; (c) the design elevation is 5,000 feet above sea level; (d) both (b) and (c).

Answer:

1. Substituting into **Equation 1.11**: $K_t = [530/(30 + 460)]^{0.825} = 1.07$; Corrected friction loss = 2.5 x 1.07 = <u>2.68 *inches wg.*</u>
2. $K_t = [530/(110 + 460)]^{0.825} = 0.94$; Corrected friction loss = <u>2.35 *inches wg.*</u>
3. Substituting into **Equation 1.12a**: $K_e = [1 - (6.8754 \times 10^{-6})(5{,}000)]^{4.73} = 0.85$; Corrected friction loss = <u>2.13 *inches wg.*</u>
4. Substituting into **Equation 1.13**: $K_f = 0.94 \times 0.85 = 0.80$; Corrected friction loss = <u>2.00 *inches wg.*</u>

If moisture in the airstream is a concern, a humidity correction factor, K_h can be calculated as follows:

$$K_h = \left(1 - \frac{(0.378 P_{ws})}{b}\right)^{0.9} \qquad \textbf{Equation 1.14}$$

where:

P_{ws} = Saturation pressure of water vapor at the dew point temperature, (*inches Hg*)

b = Actual barometric pressure, (*inches Hg*)

The total friction loss correction factor, K_f ,is expressed as:

$$K_f = K_t \times K_e \times K_h \qquad \textbf{Equation 1.15}$$

1.5 Pressure Loss in Supply Fittings

As mentioned in **Section 1.3**, pressure losses can be the result of either friction losses or dynamic losses. **Section 1.4** discussed friction losses produced by air flowing over a fixed boundary. This section will address dynamic losses. Friction losses are primarily associated with duct sections, while dynamic losses are exclusively attributable to fittings or obstructions.

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 45° branch fitting will usually have lower dynamic losses than a straight 90° tee branch.

1.5.1 Loss Coefficients

In order to quantify fitting losses, a dimensionless parameter known as a loss coefficient has been developed. Every fitting has associated loss coefficients, which can be determined experimentally by measuring the total pressure loss through the fitting for varying flow conditions. **Equation 1.16a** is the general equation for the loss coefficient of a fitting.

$$C = \frac{\Delta TP}{VP}$$ **Equation 1.16a**

where:

C = Fitting loss coefficient

DTP = Change in total pressure of air flowing through the fitting (*inches wg*)

VP = Velocity pressure of air flowing through the fitting (*inches wg*)

Once the loss coefficient for a particular fitting or class of fittings has been experimentally determined, the total pressure loss for any flow condition can be determined. Rewriting **Equation 1.16a**, we obtain:

$$\Delta TP = C \ x \, VP$$ **Equation 1.16b**

From this equation, it can be seen that the total pressure loss is directly proportional to both the loss coefficient and the velocity pressure. Higher loss coefficient values or increases in velocity will result in higher total pressure losses for a fitting. A less efficient fitting will have a higher loss coefficient (i.e., for a given velocity, the total pressure loss is greater).

1.5.2 Elbows

Table 1.1 shows typical loss coefficients for 8-inch diameter elbows of various construction.

Table 1.1
Loss Coefficient Comparisons for Abrupt-Turn Fittings

90E Elbows, 8-inch Diameter	
Fitting	**Loss Coefficient**
Die-Stamped/Pressed, 1.5 Centerline Radius	0.11
Five-Piece, 1.5 Centerline Radius	0.22
Mitered with Turning Vanes	0.52
Mitered	1.24

From **Equation 1.4**, we can determine the pressure loss:

$$DTP = DSP + DVP$$

Since the elbow diameter and volume flow rate are constant, the continuity equation **(Equation 1.1)** tells us that the velocity will be constant. From **Equation 1.5**, the velocity pressure is a direct function of velocity, and so $\Delta VP = 0$. Therefore, $\Delta SP = \Delta TP$.

Note that whenever there is no change in velocity, as is the case in duct and constant diameter elbows, the change in static pressure is equal to the change in total pressure.

Sample Problem 1-14

What is the total pressure loss of an 8-inch diameter die-stamped elbow carrying 600 cfm? What is the static pressure loss?

Answer: From **Equation 1.1**:

$$Q = A \times V \quad or \quad V = Q/A$$

$$A = \pi D^2/576 = \pi (8)^2 / 576 = 0.35\, ft^2$$

$$V = 600 / 0.35$$

$$V = 1{,}714\, fpm$$

From **Equation 1.5** or **Appendix A.1.6**:

$$VP = \left(\frac{1{,}714}{4{,}005}\right)^2 = \underline{0.18\, inches\ wg}$$

From **Table 1.1**:

$C = 0.11$ (die-stamped elbow)

From **Equation 1.16b**:

$$\Delta TP = C \times VP = 0.11 \times 0.18 = \underline{0.02\, inches\ wg}$$

Sample Problem 1-15

A designer is trying to determine which 8-*inch* elbow to select for a location, which will have a design velocity of 1,714 *fpm*. What will be the implications, in terms of pressure loss, if the designer chooses (1) a die-stamped elbow, (2) a five-gore elbow, (3) a mitered elbow with turning vanes, or (4) a mitered elbow without turning vanes?

Answer: From **Sample Problem 1-14**, we calculated the total pressure loss of an 8-*inch* die-stamped elbow at 1,714 *fpm* to be 0.02 *inches wg*.

For the other elbows, we can determine the pressure loss from loss coefficients given in **Table 1.1**:

$C_2 = 0.22$; $C_3 = 0.52$; $C_4 = 1.24$

From **Equation 1.16b**:

$DTP_2 = DSP_2 = 0.22 \times 0.18 =$ <u>0.04 *inches wg*</u> (five-gore)

$DTP_3 = DSP_3 = 0.52 \times 0.18 =$ <u>0.09 *inches wg*</u> (mitered with turning vanes)

$DTP_4 = DSP_4 = 1.24 \times 0.18 =$ <u>0.22 *inches wg*</u> (mitered without turning vanes)

Therefore, using a five-gore elbow will increase the total pressure loss by 100 percent, but it will be a very modest 0.04 *inches wg*. Using the mitered elbow with vanes would result in a 350 percent increase over the die-stamped elbow, or a 125 percent increase over the five-gore elbow. The mitered elbow without turning vanes would have a loss of 0.22 *inches wg*, which is a tenfold increase over the die-stamped elbow.

The increased pressure losses associated with the use of less efficient fittings may or may not be critical to the operation of the system, depending on the location of the fittings. Succeeding chapters will note when there could be locations in a system where it is desirable to increase the losses of certain fittings. In general, unless the system has been carefully analyzed to determine the location of the critical path(s) and the excess pressures present in other paths, it is wise to always select fittings with the lowest pressure drop.

The loss coefficients of most elbows vary as a function of diameter. The **ASHRAE Duct Fitting Database Program** (**Appendix A.8.2**) presents loss coefficients as a function of diameter for various elbow constructions. The loss coefficient drops sharply as diameters increase through approximately 24 *inches*, then only slightly from 24 *inches* through 60 *inches*. Also, eliminating turning vanes in mitered elbows more than doubles the pressure loss.

Flat Oval Elbows

Although the use of equivalent duct lengths as a measure of dynamic fitting losses is usually strongly discouraged, it provides acceptable approximations in the case of flat oval elbows. Data indicates that flat oval 90° elbows (hard or easy bend), with 1.5 centerline radius bends, have a pressure loss approximately equal to the friction loss of a flat oval duct with an identical cross section and a length equal to nine times the elbow major axis dimension, calculated at the same air velocity that is flowing through the elbow.

For example, a 12-*inch* x 31-*inch* flat oval elbow would have a pressure loss approximately equal to that of a 12-*inch* x 31-*inch* flat oval duct, 23 *feet* long (9 x 31 *inches*) at the same velocity.

For flat oval elbows that do not have a 1.5 centerline radius bend, use the loss coefficient for a round elbow of similar construction, with the diameter equal to the flat oval minor axis.

Rectangular Elbows (see ASHRAE=s Duct Fitting Database Program)

Acoustically Lined/Double-wall Elbows

For acoustically lined elbows or double-wall elbows with either a solid or perforated metal inner liner, the losses are the same as for standard single-wall elbows with dimensions equal to the metal inner liner dimensions of the acoustically lined or double-wall elbow.

Elbows With Bend Angles Less Than 90°

For elbows constructed with bend angles less than 90°, multiply the calculated pressure loss for a 90° elbow by the correction factor given in **Table 1.2.**

Table 1.2
Elbow Bend Angle Correction Factor

Angle	CF_{elb}
22.5°	0.31
30°	0.45
45°	0.60
60°	0.78
75°	0.90

1.5.3 Diverging-Flow Fittings: Branches

The pressure losses in diverging-flow fittings are somewhat more complicated than elbows, for two reasons: (1) there are multiple flow paths and (2) there will almost always be velocity changes.

First, consider the case of air flowing from the common (upstream) section to the branch. Referring to **Figure 1.1**, this is from c to b. (Refer to **Appendix A.1.1** for clarification of *upstream* and *downstream*.)

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

$$C_b = \frac{\Delta TP_{c\text{-}b}}{VP_b} \qquad \textbf{Equation 1.17a}$$

where:

C_b = Branch loss coefficient

$\Delta TP_{c\text{-}b}$ = Total pressure loss, common-to-branch (*inches wg*)

VP_b = Branch velocity pressure (*inches wg*)

Rewriting in terms of total pressure loss:

$$\Delta TP_{c\text{-}b} = C_b \times VP_b \qquad \textbf{Equation 1.17b}$$

Therefore, the total pressure loss of air flowing into the branch leg of a diverging-flow fitting is directly proportional to the branch loss coefficient and the branch 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, diverging-flow fittings almost always have velocity changes associated with them. If ΔVP is not zero, then the total and static pressure losses cannot be equal **(Equation 1.4)**.

For diverging-flow fittings, the static pressure loss of air flowing into the branch leg can be determined from **Equation 1.17c**:

$$\Delta SP_{c-b} = VP_b\,(C_b + 1) - VP_c \qquad \textbf{Equation 1.17c}$$

where:

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

VP_b = Branch velocity pressure (*inches wg*)

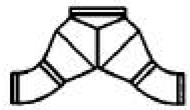
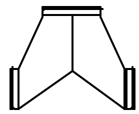
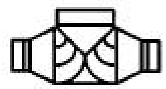
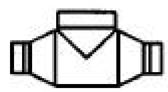
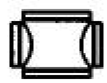
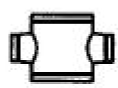
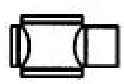
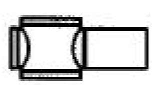
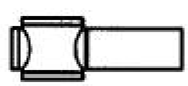
VP_c = Common velocity pressure (*inches wg*)

C_b = Branch loss coefficient (dimensionless)

Equation 1.17c is derived from **Equations 1.17a** and **1.17b**, as shown in **Appendix A.3.6**.

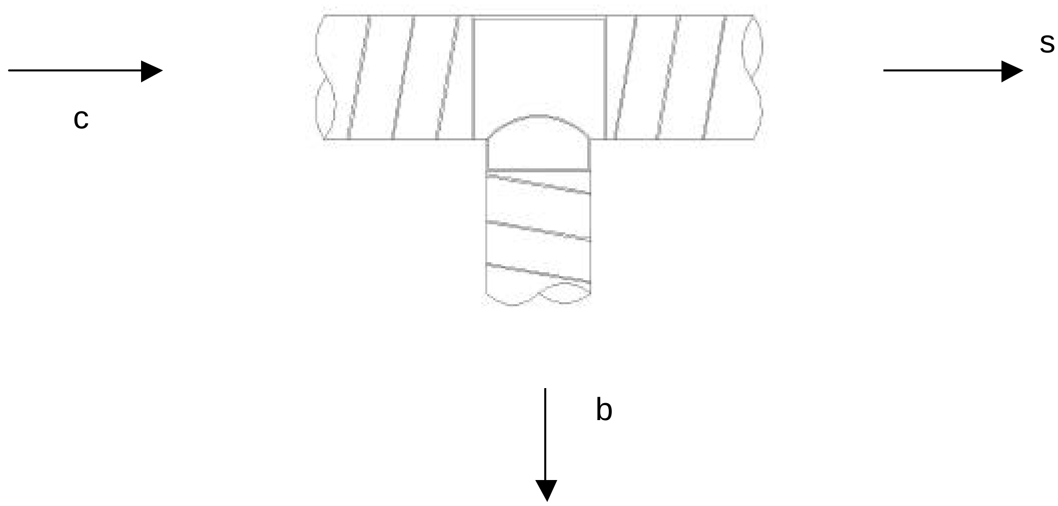
As is the case for elbows, a comparison of loss coefficients gives a good indication of relative fitting efficiencies. The following samples compare loss coefficients of various diverging-flow fittings.

Table 1.3
Loss Coefficient Comparisons for Diverging-Flow Fittings

Fitting		Loss Coefficient (C_b)
Y-Branch plus 45° Elbows		0.22
Vee Fitting		0.30
Tee with Turning Vanes plus Branch Reducers (Bullhead Tee with Vanes)		0.45
Tee plus Branch Reducers		1.08
Capped Cross with Straight Branches		4.45
Capped Cross with Conical Branches		4.45
Capped Cross with 1-*foot* Cushion Head		5.4
Capped Cross with 2-*foot* Cushion Head		6.0
Capped Cross with 3-*foot* Cushion Head		6.4
The loss coefficient, C_b, is for a V_b/V_c ratio of approximately 1.0.		

Sample Problem 1-16

What is the total pressure loss for flow from c to b in the straight tee shown below? What is the static pressure loss?



Answer: $\Delta TP_{c\text{-}b} = C_b \times VP_b$ From **Equation 1.17b**

$\Delta SP_{c\text{-}b} = VP_b\,(C_b + 1) - VP_c$ From **Equation 1.17c**

Reference: **ASHRAE Duct Fitting Database Number SD5-9**

Given:

	$Q_c = 5{,}000\ cfm$	$D_c = 24\ inches$
	$Q_b = 2{,}000\ cfm$	$D_b = 18\ inches$
	$Q_s = 3{,}000\ cfm$	$D_s = 24\ inches$

Calculate:

$$V_c,\ VP_c,\ V_b,\ VP_b,\ \frac{Q_b}{Q_c},\ \frac{A_b}{A_c}$$

$$V_c = \frac{Q_c}{A_c} = \frac{5{,}000}{3.14} = \underline{1{,}592\ fpm}$$

$$VP_c = \left(\frac{V_c}{4{,}005}\right)^2 = \left(\frac{1{,}592}{4{,}005}\right)^2 = \underline{0.16\ inches\ wg}$$

$$V_b = \frac{Q_b}{A_b} = \frac{2{,}000}{1.77} = \underline{1{,}130\ fpm}$$

$$VP_b = \left(\frac{V_b}{4{,}005}\right)^2 = \left(\frac{1{,}130}{4{,}005}\right)^2 = \underline{0.08\ inches\ wg}$$

$$\frac{Q_b}{Q_c} = \frac{2{,}000}{5{,}000} = \underline{0.40}$$

$$\frac{A_b}{A_c} = \frac{1.77}{3.14} = \underline{0.56}$$

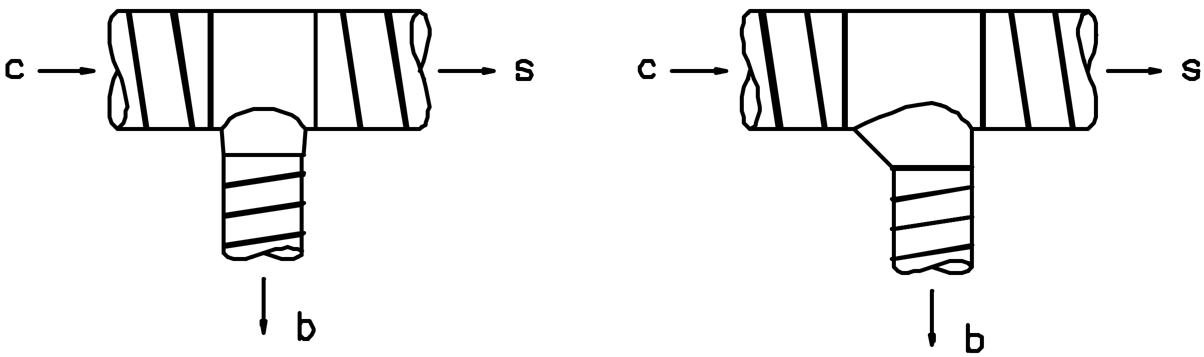
Determine: C_b - Interpolated from the ASHRAE table = 2.14

Answer: ΔTP_{c-b} = 2.14 x 0.08 = 0.17 *inches wg*

ΔSP_{c-b} = 0.08 (2.14 + 1) - 0.16 = 0.09 *inches wg*

Sample Problem 1-17

What would be the static and total pressure losses in ***Sample Problem 1-16****, if a conical tee were substituted for the straight tee? A LO-LOSS™ tee?*



Reference: Conical Tee: **ASHRAE Fitting SD5-10**
LO-LOSS™ Tee: **ASHRAE Fitting SD5-12**

Determine: C_b (conical) = 1.35

C_b (LO-LOSS™) = 0.79

Answer: ΔTP_{c-b} (conical) = 0.11 *inches wg*

ΔSP_{c-b} (conical) = 0.03 *inches wg*

ΔTP_{c-b} (LO-LOSS™) = 0.06 *inches wg*

ΔSP_{c-b} (LO-LOSS™) = -0.02 *inches wg*

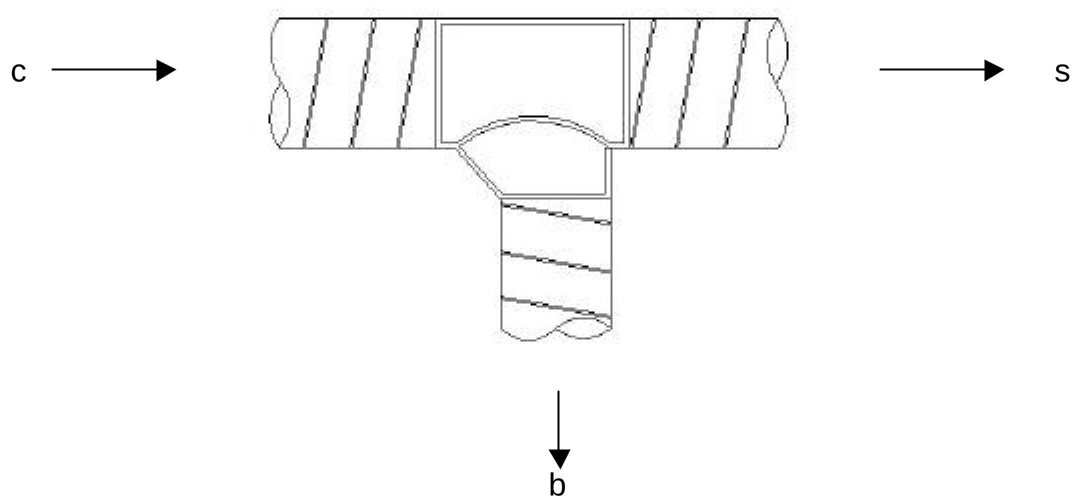
In the preceding problem, the static pressure loss for a LO-LOSS™ tee at the given conditions resulted in a negative number. Recall from **Section 1.3.1** that a pressure change expressed as a positive number is a loss, while a pressure change expressed as a negative number represents an increase in pressure. This pressure increase is a common phenomenon in air handling systems and is known as *static regain*. It occurs for the LO-LOSS™ fitting because the decrease in velocity pressure is greater than the total pressure loss of the fitting.

Total Pressure Losses versus Static Pressure Losses

Just as total pressure represents the total energy present at any point in a system, the total pressure loss of a fitting represents the true energy loss of the fitting for a given flow situation. Static pressure losses are useful for certain design methods, as we shall see later; however, they do not give an accurate indication of fitting efficiency. **Sample Problem 1-18** illustrates this concept.

Sample Problem 1-18

Change the branch size in Sample 1-17 from 18 inches diameter to 12 inches diameter and recalculate the total and static pressure losses of the LO-LOSS™ tee:



Reference: **ASHRAE Fitting SD5-12**

Given:

Q_c = 5,000 *cfm* D_c = 24 *inches*

Q_b = 2,000 *cfm* D_b = 12 *inches*

Q_s = 3,000 *cfm* D_s = 24 *inches*

Calculate: $V_c, VP_c, V_b, VP_b, \frac{Q_b}{Q_c}, \frac{A_b}{A_c}$

V_c = <u>1,592 *fpm*</u> VP_c = <u>0.16 *inches wg*</u>

V_b = <u>2,548 *fpm*</u> VP_b = <u>0.40 *inches wg*</u>

Q_b / Q_c = <u>0.40</u> A_b / A_c = <u>0.25</u>

Determine: C_b, Interpolate from **ASHRAE Fitting SD5-12** = <u>0.21</u>

Answer: $\Delta TP_{c-b} = C_b x VP_b$ (from **Equation 1.17b**) = 0.21x0.40
= <u>0.08 *inches wg*</u>

$\Delta SP_{c-b} = \mathbf{VP_b (C_b + 1) - VP_c}$ (from **Equation 1.17c**) = 0.40(0.21+1)-0.16
= <u>0.32 *inches wg*</u>

In this problem, the total pressure loss is 0.08 *inches wg*, but the static pressure loss is 0.32 *inches wg*. If one were to look only at the static pressure, this would seem to be a very inefficient fitting. However, notice that due to flow and pressure conditions in the system, the velocity increased as the air moved from the common section into the branch, resulting in a velocity pressure increase of 0.24 *inches wg*.

This situation is shown in **Figure 1.3**, Case II. The apparently large decrease in static pressure was caused by a large increase in velocity pressure. As **Equation 1.4**, this becomes:

$$
\begin{aligned}
0.08 &= 0.32 + -0.24 \\
(\Delta TP &= \Delta SP + \Delta VP)
\end{aligned}
$$

Conversely, when certain flow conditions are present, it is possible for a fitting to have a small static pressure loss but a relatively large total pressure loss. It is always advisable to calculate the total pressure loss in order to determine the total energy consumption of a fitting.

Manifold Fittings

The single-branch fittings discussed thus far are assumed to be factory fabricated, and constructed as a separate unit from the duct to which they would be attached. Occasionally, it is desirable to construct a manifold fitting, with a tap attached directly to the duct. This construction will generally result in a less efficient fitting, especially if the manifold is constructed in the field.

Flat Oval Diverging-Flow Fittings

Diverging-flow fittings of similar construction generally exhibit the same pressure loss for the same volume flow rate ratios and area ratios. Flat oval fittings exhibit similar pressure losses as round fittings. Testing is under way to develop a database of loss coefficients for flat oval diverging-flow fittings. Until this data is available, use the same loss coefficients as for the same construction of round.

Acoustically Lined and Double-wall Diverging-Flow Fittings

Whether a fitting has been acoustically lined or has a perforated metal inner shell, the difference in surface roughness is accounted for in the friction loss determination, since all friction loss calculations are base on fitting-to-fitting centerline dimensions. Therefore there is no need to increase the dynamic loss of a diverging flow-fitting that is either acoustically lined or one that has an inner metal shell, even if the shell is perforated. Determine the loss coefficient of the fitting as if it were a single-wall fitting with the dimension of the inner liner or metal inner shell.

Rectangular Diverging-Flow Fittings (see ASHRAE=s Duct Fitting Database Program)

1.5.4 Diverging-Flow Fittings: Straight-Throughs, Reducers, and Transitions

Straight-Throughs

The *straight-through* (downstream) leg of a diverging-flow fitting is that path followed by air flowing from c to s, as represented in **Figure 1.1**. The straight-through may have a constant diameter, such that $D_c = D_s$, or there may be a *reducer* attached to the straight-through, such that $D_c > D_s$.

In the case of a constant diameter straight-through, there will always be a velocity reduction caused by a reduced volume (after the branch) flowing through the same diameter duct. If a reducer is attached to the straight-through, it can be sized to reduce, maintain, or increase the downstream velocity relative to the common velocity.

Dynamic losses associated with air flowing straight through a diverging-flow fitting and/or a reducer is very slight. This is understandable, since there is little physical disturbance of the airflow. The total pressure loss in a straight-through leg or reducer is often only a few hundredths of an *inch wg*.

Perhaps the most important phenomenon associated with the straight-through flow situation is the potential for static regain. This situation is illustrated in **Figure 1.3**, Case I. A large reduction in velocity pressure and a small reduction in total pressure must (by **Equation 1.4**) result in an increase in static pressure, or static regain. The regain will be equal in magnitude to the velocity pressure loss minus the total pressure loss. Of course, if the total pressure loss is greater than the reduction in velocity pressure, as shown in **Figure 1.3**, Case III, there can be no static regain.

Referring again to **Sample Problem 1-16**, we see that the velocity in the constant diameter straight-through leg is reduced from 1,592 *fpm* (VP_c = 0.16 *inches wg*) in the duct before the straight-through to 955 *fpm* (VP_s = 0.06 *inches wg*) in the duct after the straight-through, due to the reduced volume flow. This is a velocity pressure reduction of DVP_{c-s} = 0.10 *inches wg*. If we assume a total pressure loss of DTP_{c-s} = 0.01 *inches wg*, then from **Equation 1.4** we get:

$$0.01 = DSP_{c-s} + 0.10 \text{ or } DSP_{c-s} = -0.09 \textit{ inches wg}$$

The negative result indicates a static regain, or that the static pressure at point **s** will be 0.09 *inches wg* higher than the static pressure at point **c**.

The loss coefficient data for reducers and straight-throughs is found in the **ASHRAE Duct Fitting Database**. When using loss coefficients to determine straight-through losses, **Equations 1.17b** and **1.17c** are rewritten as follows:

$$DTP_{c-s} = C_s \times VP_s \qquad \textbf{Equation 1.18a}$$

$$DSP_{c-s} = VP_s\,(C_s + 1) - VP_c \qquad \textbf{Equation 1.18b}$$

where:

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

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

VP_c = Common velocity pressure (*inches wg*)

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

C_s = Straight-through loss coefficient

Sample Problem 1-19

Calculate the total and static pressure losses for the straight-through portion of the straight tee in Sample Problem 1-16,

Reference: ASHRAE Fitting **SD5-9**

Calculate: V_c, VP_c, V_s, VP_s, $\dfrac{Q_s}{Q_c}$, $\dfrac{A_s}{A_c}$

V_c = <u>1,592 *fpm*</u> VP_c = <u>0.16 *inches wg*</u>

V_s = <u>955 *fpm*</u> VP_s = <u>0.06 *inches wg*</u>

Q_s / Q_c = <u>0.60</u> A_s / A_c = <u>1.0</u>

Determine: C_s, Interpolate from **ASHRAE Fitting SD5-9** = <u>0.20</u>

Answer:

$DTP_{c\text{-}s}$ = $\boldsymbol{C_s \times VP_s}$ (**Equation 1.18a**) = 0.20 x 0.06
= <u>0.01 *inches wg*</u>

$DSP_{c\text{-}s}$ = $\boldsymbol{VP_s (C_s + 1) - VP_c}$ (**Equation 1.18b**) = 0.06(0.20 + 1) - 0.16
= <u>-0.09 *inches wg*</u>

Reducers

A stand-alone reducer will cause the velocity to increase, since after the reducer, the same volume of air will be flowing through a smaller diameter duct (see **Sample Problem 1-2**). The use of a reducer on its own is not consistent with any design methods presented in this manual, and should be fairly rare in most duct systems. However, when this fitting is used, the losses are calculated using the same charts and in the same manner as the straight-throughs. Losses are a function of upstream and downstream velocity.

Reducing fittings should be constructed as shown in **Figure 1.7**, such that the length of the taper portion (L) is equal to the difference between the common diameter and the straight-through diameter (D_c - D_s). Verify this with manufacturer's dimension sheets. Since reducers are very efficient fittings, the use of a longer taper section will not necessarily provide a significant improvement in performance.

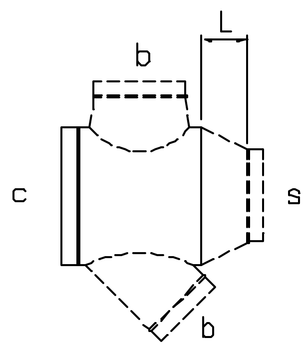


Figure 1.7
Reducing Fitting Construction

Transitions

Transitions between round and flat oval duct also produce dynamic pressure losses. As with other fittings, these losses can be quantified in terms of a loss coefficient. The loss coefficient for round-to-flat oval or flat oval-to-round transitions depends on the flat oval aspect ratio (major axis/minor axis), the direction of airflow, and the air velocity.

When round duct transitions to flat oval, the flat oval minor axis dimension is usually less than the original round diameter, while the flat oval major axis dimension is greater than the round diameter. The reverse is true in transitions from flat oval to round. Therefore, round/flat oval transitions usually involve both a reducer effect (round to flat oval minor or flat oval major to round) and an enlarger effect (round to flat oval major or flat oval minor to round).

The change in dimension involving the flat oval major axis is normally much greater than the change to/from the flat oval minor axis. Therefore, in round-to-flat oval transitions the enlarger effect predominates while in flat oval-to-round transitions the reducer effect predominates. Dynamic losses which result from expanding areas (decreasing velocities) are always more severe than losses from reducing areas (increasing velocities). Therefore the flat oval-to-round transition is more efficient than the round-to-flat oval fitting.

Figure A.24 in **Appendix A.4.2** is a plot of loss coefficient (C_S) versus round duct velocity, for both round-to-flat oval and flat oval-to-round transitions. The curves are valid for any size flat oval and will be conservative for transitions involving flat oval with a low aspect ratio. Use the appropriate loss coefficient value in the following equations to determine static and total pressure losses for transition fittings.

$$DTP_{c-s} = C_s \ x \ VP_s \qquad \textbf{Equation 1.19a}$$

$$DSP_{c-s} = VP_s \ (C_s + 1) - VP_c \qquad \textbf{Equation 1.19b}$$

where:

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

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

VP_c = Common velocity pressure (*inches wg*)

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

C_s = Straight-through loss coefficient

Sample Problem 1-20

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*In **Sample Problem 1-10**, we determined that a 12-inch H 31-inch flat oval duct would have the same pressure loss per unit length as a 20-inch round duct. What would be the impact on total and static pressure losses in a 100-foot section of 20-inch round duct carrying 5,000 cfm if 40 feet of this duct were replaced by a 12-inch H 31-inch flat oval duct? Assume the first 30 feet of duct is round, next 40 feet is flat oval, and the last 30 feet is round.*

Answer:

Since the duct sizes are equivalent, 40 *feet* of 12-*inch* H 31-*inch* flat oval would have the same pressure loss per 100 *feet* as the section of 20-*inch* round duct it replaced. The addition of two transitions, one round-to-flat oval at the start of the

40-*foot* section and the other flat oval-to-round at the transition back to round duct would cause the only change in pressure loss.

The velocity in the 20-*inch* round duct carrying 5,000 *cfm* is 2,294 *fpm* (VP = 0.33 *inches wg*). The flat oval duct cross-sectional area from Appendix A.1.3 is 2.36 ft^2; therefore the velocity in the flat oval duct is 2,119 *fpm* (VP = 0.28 *inches wg*). From Appendix A.4.3.4, the loss coefficients at 2,294 *fpm* are C_{r-o} = 0.17 and C_{o-r} = 0.06.

Substituting into **Equations 1.19a** and **1.19b**

$DTP_{c-s\{r-o\}}$ = 0.17 x 0.28 = <u>0.05 *inches wg*</u>

$DTP_{c-s\{o-r\}}$ = 0.06 x 0.33 = <u>0.02 *inches wg*</u>

$DSP_{c-s\{r-o\}}$ = 0.28 (0.17 + 1) - 0.33 = <u>-0.00 (-0.002) *inches wg*</u>

$DSP_{c-s\{o-r\}}$ = 0.33 (0.06 + 1) - 0.28 = <u>-0.07 *inches wg*</u>

The flat oval section will therefore increase the total pressure loss by an additional 0.07 *inches wg* (0.05 + 0.02), due to the combined effects of both transitions. As expected, since there was no net change in velocity in the round duct, DVP = 0 and (by **Equation 1.4**) the combined static pressure loss (0.07 *inches wg*) is equal to the combined total pressure loss.

1.5.5 Miscellaneous Fittings

Heel-Tapped Elbows

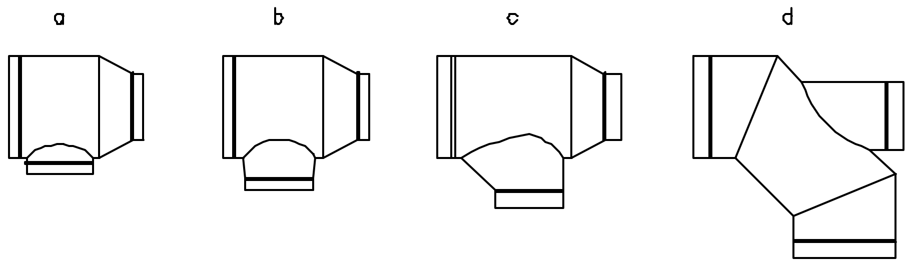
The tee-type diverging-flow fittings discussed in **Section 1.5.3** are generally used where the designer desires to direct a relatively small quantity of air at some angle relative to the main trunk duct, while maintaining a straight-through flow for the majority of the air. Occasionally, situations arise where the main air stream must be diverted at some angle, while a smaller quantity of air is required in a straight-through direction. In these situations, the use of a heel-tapped elbow will generally result in lower pressure losses, in both common and branch directions.

ASHRAE Fitting **SD5-21** presents loss coefficients (C_b) for both the straight-through tap and the elbow section of heel-tapped elbows as a function of velocity ratio. Use **Equations 1.17b** and **1.17c** for determining the total and static pressure losses. If 65 percent or more of the airflow is diverted, then it is advisable to use a heel-tapped elbow.

Sample Problem 1-21

A diverging-flow fitting must be selected which will split 10,000 cfm volume flow rate such that 7,000 cfm will flow at a 90E angle relative to the upstream direction, and 3,000 cfm will continue in the same direction as the upstream flow.

Compare the performance of a (a) straight tee, (b) conical tee, (c) LOLOSS™ tee, and (d) heel-tapped elbow, and select the most efficient fitting for this situation. Assume it is desired to *maintain approximately constant velocity:*



Given:

Q_c =	10,000 *cfm*	Q_b =	7,000 *cfm*	Q_s =	3,000 *cfm*
D_c =	28 *inches*	D_b =	23 *inches*	D_s =	15 *inches*
A_c =	4.28 *sq ft*	A_b =	2.89 *sq ft*	A_s =	1.23 *sq ft*
V_c =	2,336 *fpm*	V_b =	2,422 *fpm*	V_s =	2,439 *fpm*
VP =	0.34 *inches wg*	VP_b =	0.37 *inches wg*	VP_s =	0.37 *inches wg*
Q_b/Q_c =	0.70	Q_s/Q_c =	0.30		
A_b/A_c =	0.68	A_s/A_c =	0.29		

Reference:

Straight Tee	**ASHRAE Fitting SD5-9**
Conical Tee	**ASHRAE Fitting SD5-10**
LO-LOSS™Tee	**ASHRAE Fitting SD5-12**
Heel-tapped Elbow	**ASHRAE Fitting SD5-21**

Determine:

Coefficients	C_b	C_s
Straight Tee	1.15	0.13
Conical Tee	0.62	0.13
LO-LOSS™ Tee	0.33	0.13
Heel-tapped Elbow	0.65	0.09

Answer:

Pressure Losses (*inches wg*)	DTP_{c-b}	DTP_{c-s}	DSP_{c-b}	DSP_{c-s}
Straight Tee	0.43	0.05	0.46	0.08
Conical Tee	0.23	0.05	0.26	0.08
LO-LOSS™ Tee	0.12	0.05	0.15	0.08
Heel-tapped Elbow	0.24	0.03	0.27	0.06

In this case, the heel-tapped elbow and the conical tee will have nearly the same total pressure loss in the 90E bend direction. The heel-tapped elbow provides a significant performance increase over the straight tee but has a higher loss than the LO-LOSS™ tee branch. The straight-through leg of the heel-tapped elbow is slightly more efficient than the straight-throughs of the tees.

The best fitting though, based strictly on efficiency, would appear to be the LO-LOSS™ tee. However, bear in mind that all three tee fittings will require substantial straight-through reducers (28 *inches* to 15 *inches*) which will generally be at least 12 *inches* long (**Figure 1.7**) and will add substantially to the cost of the fitting. If a compromise between cost and performance is desired, the heel-tapped elbow may still be the best choice for these flow situations. Also, increased loss may help balance the leg in which this fitting resides.

Note that in Sample **Problem 1-21** the total pressure is lower than the static pressure by 0.03 *inches wg* in all cases, due to the increase in velocity pressure (0.03 *inches wg*) from common to both straight-through and branch. In the form of **Equation 1.4:**

$$DTP_{c-b} = DSP_{c-b} - 0.03$$

$$DTP_{c-s} = DSP_{c-s} - 0.03$$

Crosses

Cross fittings are those which have two taps located at the same cross section of a main or trunk duct. Usually these taps will be constructed so that they discharge air in diametrically opposed directions.

The pressure loss of the taps on a cross fitting depends, to a large extent, on the cross-sectional area reduction of the straight-through duct. For straight-through area reductions of less that 20 percent, the branch losses through either tap of a cross fitting will be the same as those for a single-branch fitting with identical tap construction. For example, a conical cross (a cross with conical taps) that has a straight-through of 24 *inches*, reducing to 22 *inches*, would have an area reduction of 16 percent. Since this is less than 20 percent, the branch losses for either tap would be found from the direct-read charts in ASHRAE Fitting**s SD5-9** through **SD5-17**.

For crosses where the straight-through area reduction exceeds 20 percent, use the loss coefficients presented in ASHRAE Fitting**s SD5-23** through **SD5-26**. The three curves shown are for varying percentage area reductions and apply for all tap constructions. Use interpolation to find loss coefficients for area reductions between these curves. Use **Equations 1.17b** and **1.17c** for determining the total and static pressure losses of each branch.

Split Fittings

Split fittings are diverging-flow fittings where the air divides into two branches, each of which turns at 90E to the main. There is no straight-through leg. The most common types of split fittings are the Vee, Y-branch and the bullhead tee.

The Y-branches are the most efficient split fittings; however, they are more expensive than Vee's and may be more expensive than bullhead tees. Bullhead tees should always be specified with turning vanes. ASHRAE Fittings **SD5-18**, **19** and **21** presents drawings and loss coefficients data for Y-branches, bullhead tees with turning vanes, bullhead tees without turning vanes, and capped crosses. The capped cross is discussed in the following section. Use **Equations 1.17b** and **1.17c** for determining the total and static pressure losses.

Capped Fittings

Capped fittings are those in which the main or straight-through is completely closed off. The most common use for a capped fitting is in locations where it is expected that future expansion will require additional ducting, at which time the cap can be removed and the duct run continued. In general, the use of capped fittings is strongly discouraged. For both single-branch fittings and crosses, the performance is severely degraded if the main is capped.

Where these fittings are unavoidable **SD5-2** includes a curve for capped crosses. Use **Equations 1.17b** and **1.17c** for determining the total and static pressure losses.

Close-Coupled Fittings

After air flows through any type of fitting, a certain length of straight duct is required to re-establish the flow profile of the airstream. Simply stated, it takes a certain distance for air to recover from a disturbance produced by a fitting. If the airstream encounters a second fitting before it has had a chance to recover from a previous disturbance, the effect of the second fitting will be more pronounced than if it had been located in a long run of straight duct.

Generally, two elbows in series will have the same loss as the sum of the individual elbows. The exception to this is when the second elbow has an additional change in direction such that the air is not flowing parallel to the first flow. For this case, as much as an additional 100 percent of the combined losses should be added, unless the elbows are at least 10 diameters apart.

The loss of two tees in series is a function of the spacing between the tees, although the loss coefficient of the upstream tee is not significantly affected. The loss coefficient of the downstream tee actually decreases at half-diameter spacing. At two-diameter spacing, however, the downstream loss coefficient is significantly higher. The loss coefficient gradually decreases back to its original value at 10 diameters. To account for the increased pressure loss at two-diameter spacing, add 100 percent of the calculated loss. This can be decreased as the diameter spacing between tees becomes greater. **Appendix A.9.9** has a more detailed discussion of the effect of spacing of tees.

Couplings

Slip couplings, which are inserted inside duct sections and are therefore exposed to the air stream, are generally used to join two adjacent duct sections. Fittings, which connect directly to duct sections, do not require couplings, and fittings which are connected directly to other fittings usually have an outside coupling. Losses associated with duct couplings are very low. When slip couplings are separated by 10 to 20 *feet* of duct, their effects are negligible.

However, in the event it is necessary to calculate the loss due to duct couplings, **Appendix A.4.2** presents a table of loss coefficients versus coupling diameter. Use **Equations 1.17b** and **1.17c** for determining the total and static pressure losses. As can be seen from the loss coefficient values, it is normally quite acceptable to ignore these losses when calculating system pressure losses. Experience has shown that even poorly made and undersized couplings have negligible losses. The resulting loss coefficient may be two to three times that of a slip coupling, but this is still a very low value.

Offsets

Offsets are required to change the location of a duct run horizontally, vertically or both. This is most often necessary to avoid interference with some obstruction along the duct run. Offsets are usually constructed with two or more elbows, joined by a suitable length of straight duct. Due to the almost limitless number of offsets that could be created, there are no tables or charts in this manual for the calculation of these losses. It is suggested that offset losses be obtained by adding the losses of the individual elbows and duct, which form the offset and, if necessary, adding a factor for any close-coupling effects that may exist.

Bellmouths

The bellmouth fitting is used as an intake or entrance to a duct, usually from a plenum or fan housing. There is a substantial advantage in having a smooth radiused entrance, as opposed to a square-edged entrance. Loss coefficients for bellmouths are presented in ASHRAE Fittings **SD1-1**, **SD1-2**, and **SD1-3**. To calculate the total and static pressure losses, use (transition loss) **Equations 1.19a** and **1.19b.**

Expanders

Increasing the duct diameter, upstream-to-downstream, in a supply air system is not a recommended design practice. Abrupt expanders are very inefficient fittings in that their loss coefficients are always 1.0 or greater. This means that the entire upstream velocity head is lost and unrecoverable. This can be shown by substituting a unity loss coefficient into **Equations 1.18a** and **1.18b**:

$$DTP_{c\text{-}s} = 1.0 \times VP_c = VP_c$$

$$DSP_{c\text{-}s} = VP_c\,(1.0 - 1.0) + VP_s = VP_s$$

Therefore, although the static pressure loss may be small, the total pressure loss is equal to (at least) the entire upstream velocity pressure.

Exits

Exits are fittings that discharge air into the surrounding environment. Refer to ASHRAE Fittings **SD1-1** and **SD1-2** (plenums), **SD2-1** to **SD2-6** (atmosphere), and **SD7-1** to **SD7-5** (fans) for loss coefficients of round exits. Refer to **SR1-1** (plenums), **SR2-1** to **SR2-6** (atmosphere), and **SR7-1** to **SR7-17** (fans) for loss coefficients of rectangular exits. Increasing the duct size at an exit is advantageous in minimizing pressure loss.

Obstructions

In-line losses common to supply systems also must be taken into account. Refer to ASHRAE's Duct Fitting Database **CD9-1** to **CD9-3** for loss coefficients of round dampers, **CR9-1** to **CR9-7** for loss coefficients of rectangular dampers, **CD8-1** to **CD8-8** for loss coefficients of round silencers, **CR8-1** to **CR8-4** for loss coefficients of rectangular silencers, **CR8-5** to **CR8-8** for loss coefficients of coils, **CR8-9** to **CR8-11** for loss coefficients of VAV boxes, and **CD6-1** to **CD6-4** for loss coefficients of other round obstructions.

1.5.6 Nonstandard Conditions

In **Section 1.4.5** equations were given for correcting the calculated friction loss of a system for nonstandard conditions of temperature and/or elevation. Since velocity pressure is a function of air density (see **Appendix A.3.3**), and since all dynamic fitting losses are a function of velocity pressure, an additional correction must be made to the calculated fitting losses whenever there are substantial variations from standard conditions. If a density-corrected velocity pressure (**Appendix A.3.3**) is used to calculate all dynamic fitting losses, then no further corrections (except friction loss corrections) are required.

If the pressure losses are calculated assuming standard conditions, the results can be corrected by multiplying by the ratio of actual density diverging by standard density. For most HVAC applications, this ratio can be calculated as shown in **Equation 1.20**:

$$\frac{\rho_{act}}{\rho_{std}} = \left[\frac{530}{(T_a + 460)}\right] x \left[\frac{\beta}{29.921}\right] \qquad \textbf{Equation 1.20}$$

where:

ρ_{act} = Actual density (lbm/ft^3)

ρ_{std} = Standard density (lbm/ft^3)

T_a = Actual air temperature (°F)

β = Actual barometric pressure (*inches Hg*)

The corrected total, static, and velocity pressure can be calculated as follows:

$$\Delta TP_{act} = \Delta TP_{std}\ x \left(\frac{\rho_{act}}{\rho_{std}}\right) \qquad \textbf{Equation 1.21}$$

$$\Delta SP_{act} = \Delta SP_{std}\ x \left(\frac{\rho_{act}}{\rho_{std}}\right) \qquad \textbf{Equation 1.22}$$

$$\Delta VP_{act} = \Delta VP_{std}\ x \left(\frac{\rho_{act}}{\rho_{std}}\right) \qquad \textbf{Equation 1.23}$$

where:

ΔTP_{act} = Total pressure loss at actual conditions (*inches wg*)

ΔTP_{std} = Total pressure loss at standard conditions (*inches wg*)

ΔSP_{act} = Static pressure loss at actual conditions (*inches wg*)

ΔSP_{std} = Static pressure loss at standard conditions (*inches wg*)

ΔVP_{act} = Change in velocity pressure calculated at actual conditions (*inches wg*)

ΔVP_{std} = Change in velocity pressure calculated at standard conditions (*inches wg*)

Density correction factors are tabulated in **Appendix A.1.5.** Often, for systems operating at normal HVAC temperatures (70° ± 30°F) and elevations less than 1,500 *feet* above sea level, these corrections can be neglected.

When **Equations 1.21** and/or **1.22** are applied to the aggregate pressure loss of an entire system instead of to the individual fitting components, the resulting pressures are not necessarily accurate. This is because the friction correction factors (**Section 1.4.5**) are calculated in a different manner from the dynamic loss correction factors discussed above. The accuracy depends on the ratio of duct length versus number of fittings and on the deviation from standard conditions.

Sample Problem 1-22

Determine the effect on the total and static pressure losses of the straight tee of **Sample Problem 1-16** *if the air temperature is 55°F and the elevation is 5,000 feet.*

Answer:

From **Sample Problem 1-16** at standard conditions:

$$\Delta TP_{c\text{-}b,std} = \underline{0.17\ inches\ wg}$$

$$\Delta SP_{c\text{-}b,std} = \underline{0.09\ inches\ wg}$$

From **Appendix A.1.5**:

$$\frac{\rho_{act}}{\rho_{std}} = \underline{0.86} \quad \text{(interpolation required)}$$

From **Equation 1.21**:

$$\Delta TP_{c\text{-}b,act} = \Delta TP_{c\text{-}b,std}\ x\ \frac{\rho_{act}}{\rho_{std}} = 0.17 \text{ x } 0.86 = \underline{0.15\ inches\ wg}$$

From **Equation 1.22**:

$$\Delta SP_{c\text{-}b,act} = \Delta SP_{c\text{-}b,std}\ x\ \frac{\rho_{act}}{\rho_{std}} = 0.09 \text{ x } 0.86 = \underline{0.08\ inches\ wg}$$