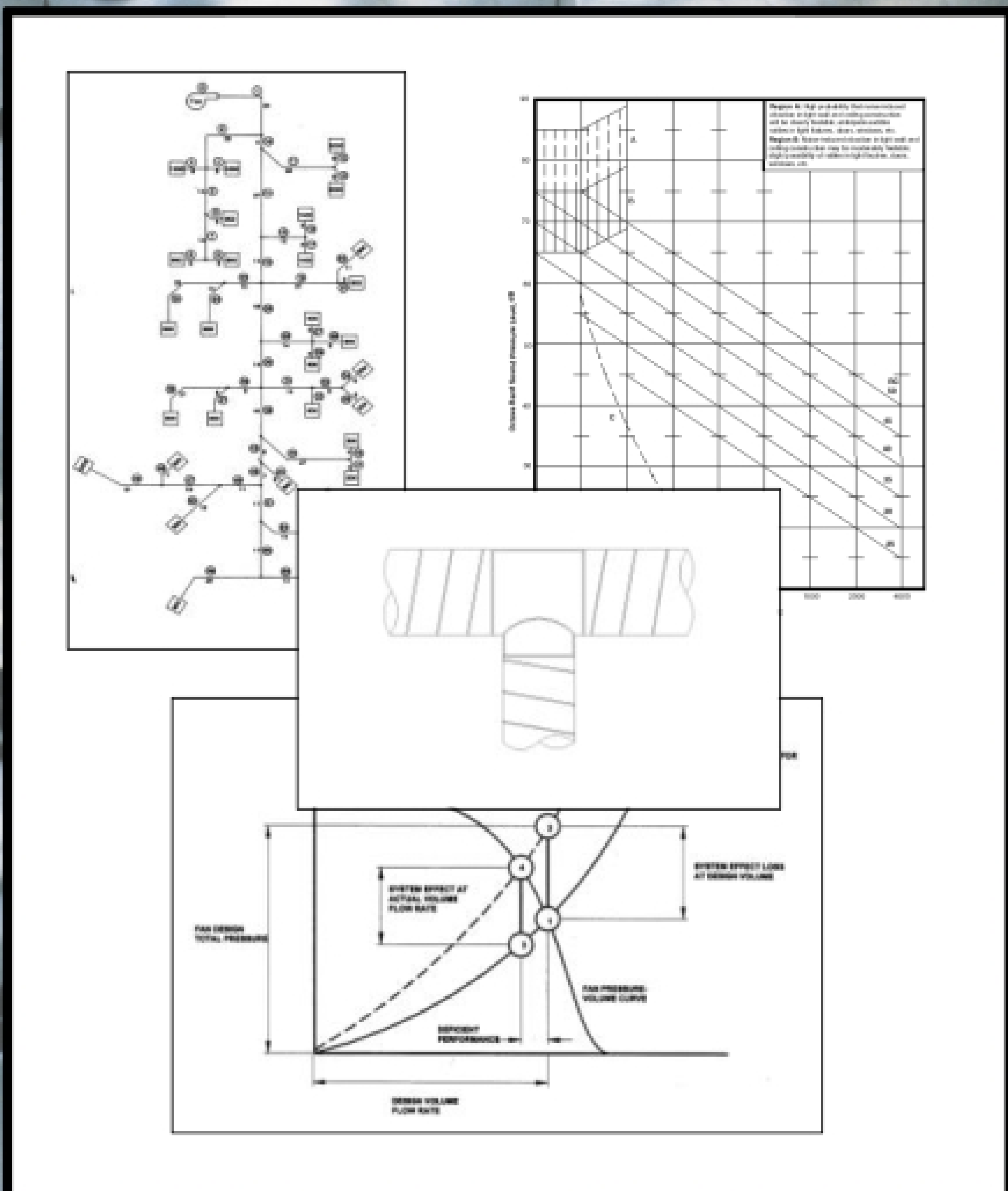
Home

McGill AirFlow's

Duct System Design Guide



McGill AirFlow Corporation

an enterprise of United McGill Corporation — Founded in 1951

Duct System Design Guide

First Edition

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Notice:

The performance data included in this Guide have been obtained from testing programs conducted in flow measurement laboratories and detailed in the reference test reports. The data are reprinted in this manual as a source of information for design engineers. McGill AirFlow Corporation assumes no responsibility for the performance of duct system components installed in the field.

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Acknowledgements

The following individuals have made significant contributions to this guide.

Patrick J. Brooks, P.E.
Gerard L. Iacouzze
Todd A. Talbott

Patrick J. Brooks, P.E.
Mr. Brooks has been both an employee of United McGill Corporation and a member of ASHRAE for 21 years. He holds BSME, MSME and MBA degrees from The Ohio State University. He was the principal investigator for ASHRAE Research Project 551, Laboratory Study to Determine Flow Resistance of HVAC Duct Fittings. He is a member of many ASHRAE committees and chairs the TC 5.2 Duct Design committee. He has reviewed many technical articles regarding duct design and air distribution.

Mr. Brooks authored or co-authored *Examining Duct Sizing Methods* in the May 1992 issue of Plant Engineering; *Comparing Duct Sizing Methods* in the June 1992 issue of Plant Engineering; *Duct Design Fundamentals* in Volume 37, Number 4 (April 1995) of the ASHRAE Journal, and *Designing Industrial Exhaust Systems* in Volume 43, Number 4 (April 2001) of the ASHRAE Journal. He also authors McGill AirFlow Corporation's technical bulletins and reports.

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Mr. Iacouzze's employment with McGill AirFlow Corporation spans 15 years. In addition to his contributions to this guide, he has served on the ASHRAE's TC 5.2 Duct Design and TC 6.3 Central Forced Air Heating and A/C Systems and Standards committees. He's the author and co-author of duct construction and standards manuals for the Sheet Metal and Air Conditioning Contractors' National Association (SMACNA) and the National Environmental Balancing Bureau (NEBB).

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INTRODUCTION

Foreword

These reference manuals have been compiled and written by engineers and consultants employed by United McGill Corporation and its subsidiary, McGill AirFlow Corporation. McGill AirFlow Corporation is the nation=s foremost producer of sheet metal duct and fitting components for air handling systems. In over 50 years of serving the mechanical system marketplace, McGill AirFlow has gained a technical expertise, which is unmatched in the industry. This publication shares a part of that expertise with the engineers, designers, contractors, and specifiers who design and install duct systems,

Although there is an abundance of published literature concerning mechanical system design, there is very little information specifically about duct system design condensed into a single document. What does exist is difficult to understand. There is a wealth of information available concerning air volume determination (heating and cooling loads), fan selection and specification, terminal and/or box selection, room air distribution, etc., but very little is written about the duct and fittings used in duct systems. This publication is an attempt to remedy that situation.

There are many methods for producing acceptable supply duct designs. In this notebook, three methods for positive pressure design will be addressed: equal friction, static regain and total pressure. This is not meant to exclude other design methods; however, these are the most popular. One notable method not included in this notebook is the T-Method developed by Dr. Robert Tsal (see Appendix A.9.7). It Is a complex design methodology that optimizes the owning costs of a duct system design. The design of negative pressure systems is a function of the intended application: return air, fume exhaust, particulate exhaust, etc.

Throughout this notebook, example systems and designs will be used to illustrate important concepts. Whenever possible, the same system layout will be used so that readers can become familiar with it and witness the result of various parameter, component, or operational changes of the system. In every case, the examples have been computer verified for accuracy.

For many years, McGill AirFlow Corporation has published engineering bulletins and engineering reports, which focus on a particular issue generally related to air handling system design. Many of these publications provide a complete discussion of important topics. Contact McGill AirFlow Corporation to obtain one of these notebooks.

In keeping with present industry convention, standard English units have been used throughout this notebook. For those who prefer metric units, conversion factors have been provided in **Appendix A.1.2**

An Overview of the Design Process

The design of supply duct systems can be approached in a simple and straightforward manner, using the following eight steps:

Step 1__Determine air volume requirements.

Step 2__Locate duct runs.

Step 3__Select design method.

Step 4__Determine duct sizes based on the design methodology.

Step 5__Determine system pressure requirements.

Step 6__Select fan according to proper guidelines.

Step 7__Analyze the design to improve balancing and reduce material cost.

Step 8__Analyze the life-cycle cost of the design.

Many experienced designers will stop work after completing the first six steps. The system may properly supply the design air requirements, but if the analysis steps are omitted, some very substantial savings in both equipment and operating costs may be overlooked.

The design of exhaust duct systems is also straightforward. However, these systems are often process oriented and may require customizing to suit individual applications. The basic design process for exhaust or negative pressure systems is similar to that for supply or positive pressure systems. The following steps, to be used when designing an exhaust system:

Step 1__Define the application and determine any operational and performance parameters.

Step 2__Determine capture velocities and air volume requirements.

Step 3__Locate duct runs.

Step 4__Determine duct sizes based on velocity constraints.

Step 5__Determine system pressure requirements.

Step 6__Select fan according to proper guidelines.

Step 7__Analyze the design to improve balancing and reduce material costs.

Step 8__Analyze the life-cycle cost of the design.

Several factors should be considered when developing the acoustical design and analysis of a supply or exhaust/return system. The following are suggested steps.

Step 1__Determine an acceptable noise criteria rating for the space.

Step 2__Determine the sound source - sound spectrum.

Step 3__Calculate the resultant sound levels criteria.

Step 4__Compare the resultant sound levels.

Step 5__Select the appropriate noise control product(s) needed to attain the noise criteria.

How to Use the Duct System Design Notebook

This notebook is arranged so that it will be useful to both the novice designer and the more experienced professional. The text and articles which form the main body of material are stepBby-step explanations of the design process. Most of the material is presented so that it can be understood even by those with limited technical training, although a basic knowledge of mathematics is assumed.

The notebook covers supply systems and exhaust/return systems. Each includes separate sections which discuss the design and the analysis of air handling systems. The design sections explain the procedures necessary to size duct and determine system pressure requirements. The analysis sections explain how to evaluate systems for which the preliminary design is already completed (or existing systems) and, where possible, how to make alterations which will improve the operation and/or reduce the cost of the system. Acoustical design and analysis of air handling systems has been included because designers must be concerned with all aspects of human comfort.

Throughout the text, there are many sample problems. These problems are provided to test the reader=s understanding of the material just presented. Sample problems will be especially useful to the novice designer who is encountering many of the concepts for the first time.

There is a substantial appendix, which is intended to be a reference for anyone doing design work. Many individuals are familiar with the basic design procedures but may need to review specific equations, determine loss coefficients, etc. The appendix is designed to be a quick source for this information. Specification information can also be found in the appendix along with computer applications and references.

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.

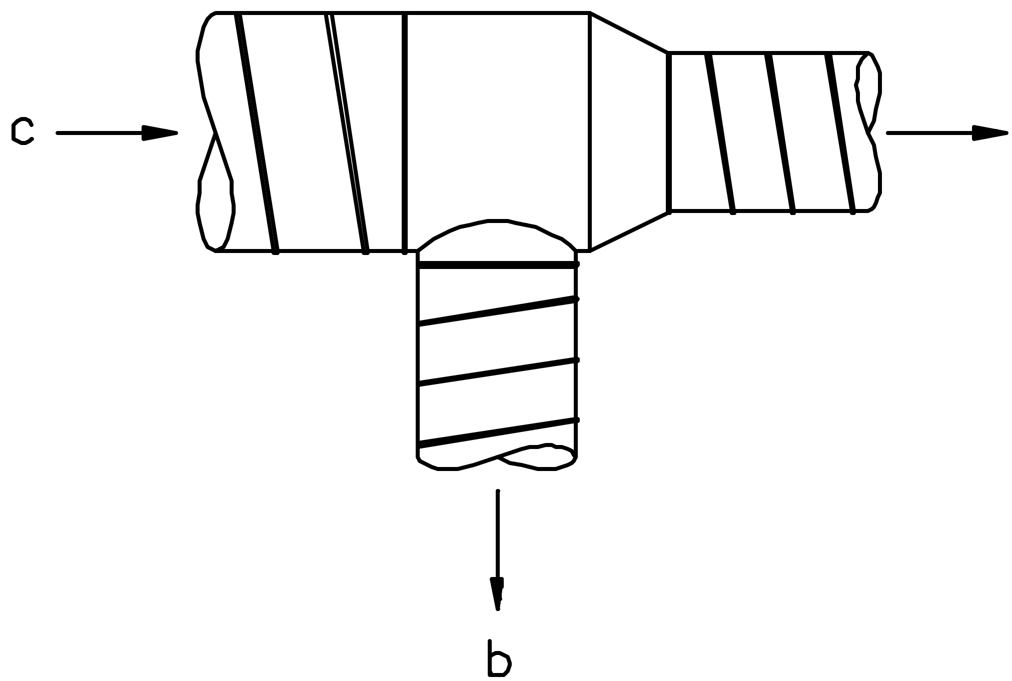

Figure 1.1
Diverging Flow

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

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

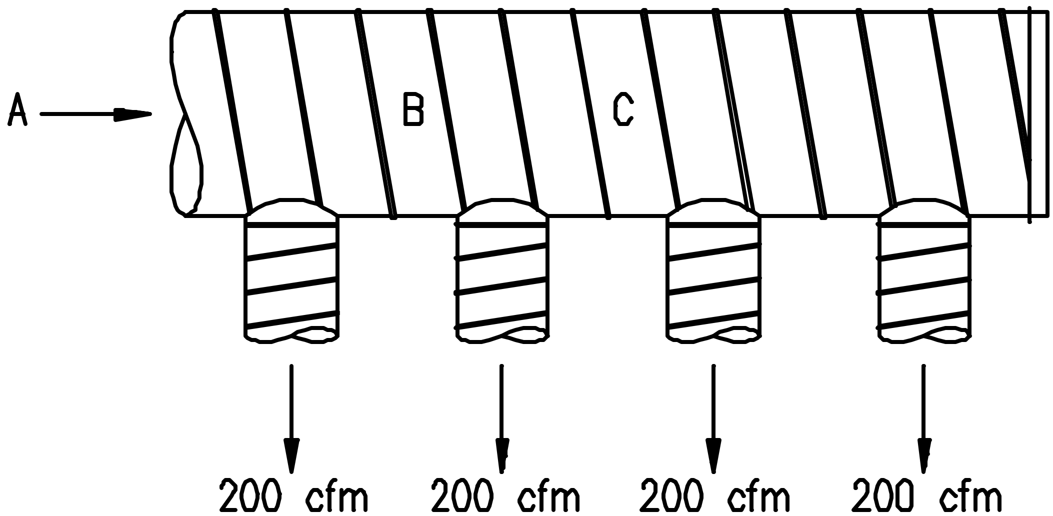

Figure 1.2
Multiple Diverging Flow

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**.

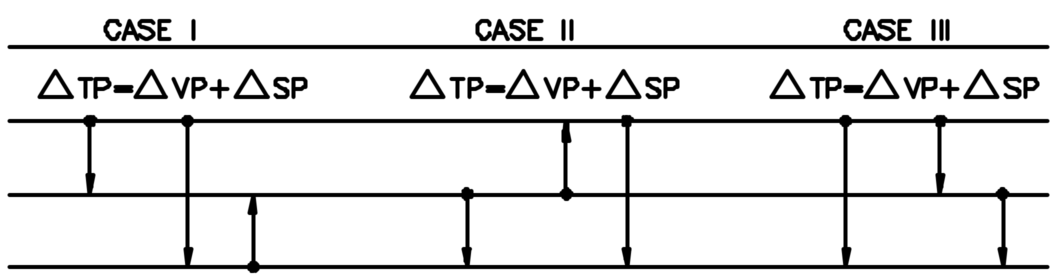

Figure 1.3
Conservation of Energy Relationship

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.

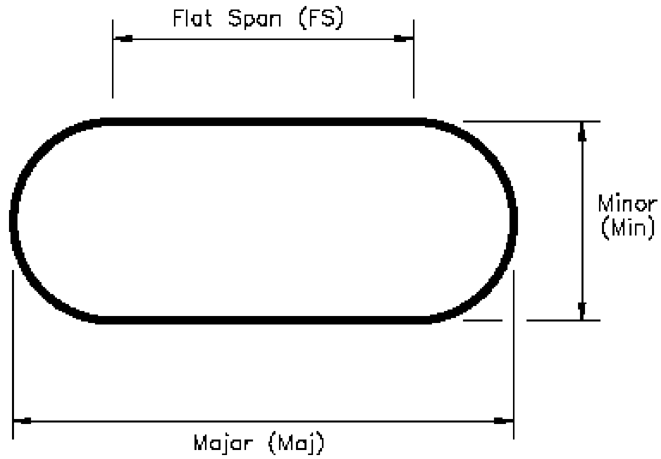


Figure 1.4
Flat Oval Dimensions

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**.

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

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**.

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

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 0.50 *inch wg* per 100 feet.

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 *inches wg* 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 12 *inches* x 31 *inches*.

If it is determined that there is room for a 14-*inch* minor axis duct, the required size would be 14 *inches* x 27 *inches* (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**:

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

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

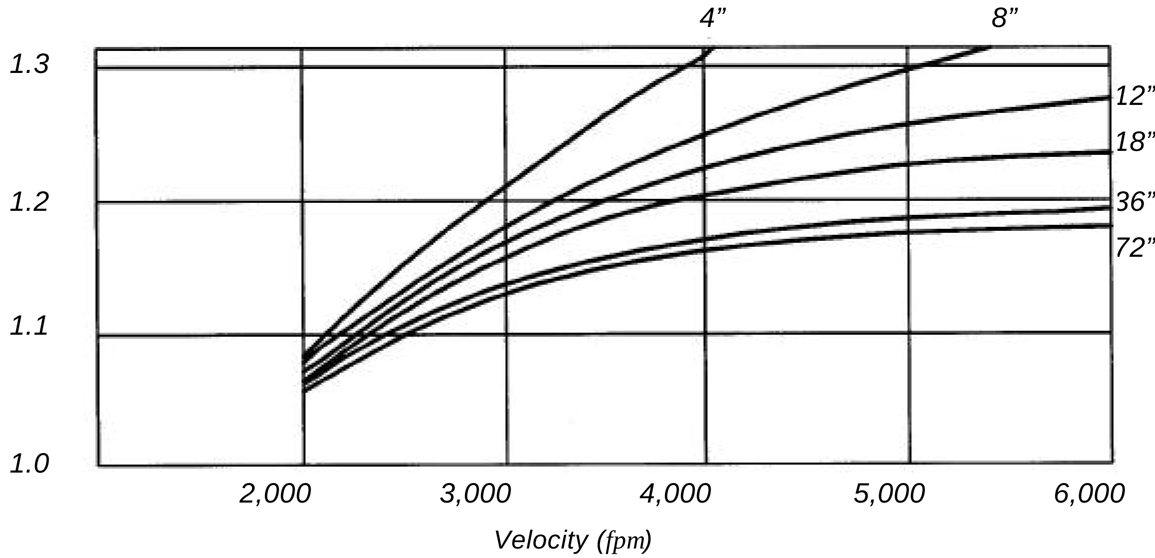


Figure 1.5
Correction Factors for Double-Wall Duct with Perforated Metal Inner Liner

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:

$$K_t = \left(\frac{530}{(T_a + 460)} \right)^{0.825}$$ **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:

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

Equation 1.12a can also be written as follows:

$$K_e = \left(\frac{b}{29.921} \right)^{0.9}$$ **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:

$$K_f = K_t \times K_e$$ **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 x 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 x 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:

$$K_h = \left(1 - \frac{(0.378 P_{ws})}{b}\right)^{0.9}$$ **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:

$$K_f = K_t \times K_e \times K_h$$ **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.

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

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:

$$\Delta TP = C \text{ x } VP \qquad \textbf{Equation 1.16b}$$

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:

$$C_b = \frac{\Delta TP_{c\text{-}b}}{VP_b}$$ **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:

$$\Delta TP_{c\text{-}b} = C_b \times VP_b$$ **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*)

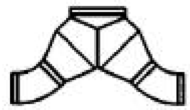
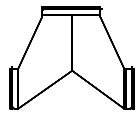
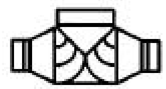
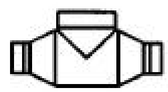
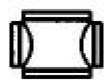
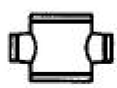
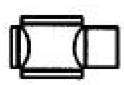
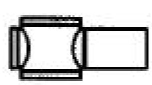
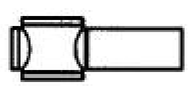
VP_c = Common velocity pressure (*inches wg*)

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**.

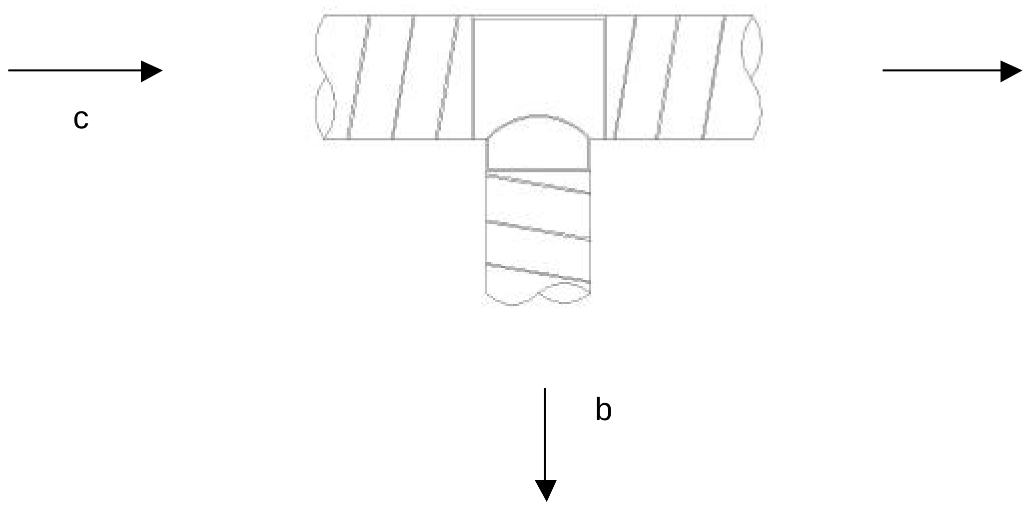
As is the case for elbows, a comparison of loss coefficients gives a good indication of relative fitting efficiencies. The following samples compare loss coefficients of various diverging-flow fittings.

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

$\Delta SP_{c-b} = VP_b\,(C_b + 1) - VP_c$ 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:

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

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

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

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

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

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

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

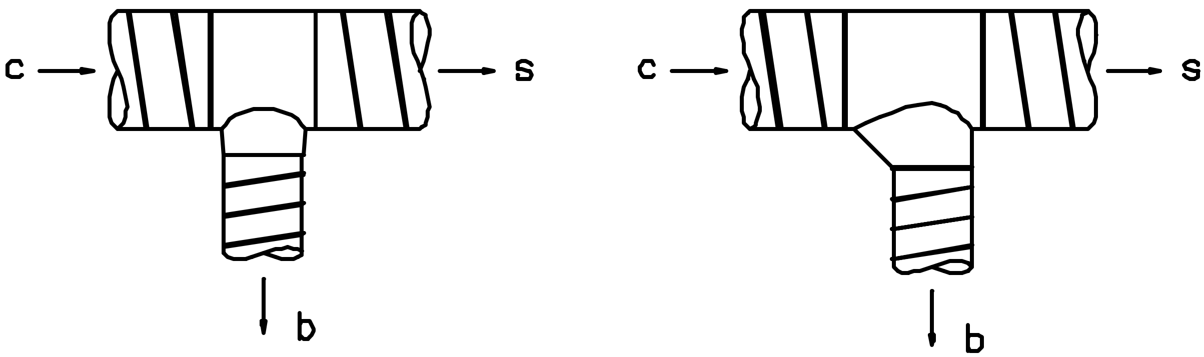
Determine: C_b - Interpolated from the ASHRAE table = 2.14

Answer: $\Delta TP_{c\text{-}b}$ = 2.14 x 0.08 = 0.17 *inches wg*

$\Delta SP_{c\text{-}b}$ = 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*

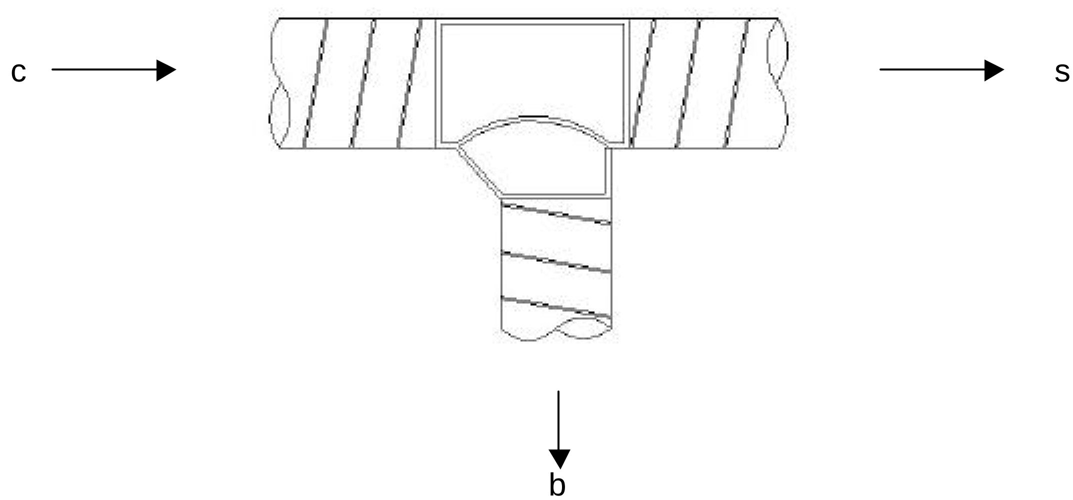
In the preceding problem, the static pressure loss for a LO-LOSS™ tee at the given conditions resulted in a negative number. Recall from **Section 1.3.1** that a pressure change expressed as a positive number is a loss, while a pressure change expressed as a negative number represents an increase in 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: $V_c, VP_c, V_b, VP_b, \dfrac{Q_b}{Q_c}, \dfrac{A_b}{A_c}$

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>

ΔSP_{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:

$$0.08 = 0.32 + -0.24$$
$$(\Delta TP = \Delta SP + \Delta VP)$$

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 = 1,592 *fpm* VP_c = 0.16 *inches wg*

V_s = 955 *fpm* VP_s = 0.06 *inches wg*

Q_s / Q_c = 0.60 A_s / A_c = 1.0

Determine: C_s, Interpolate from **ASHRAE Fitting SD5-9** = 0.20

Answer:

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

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

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.

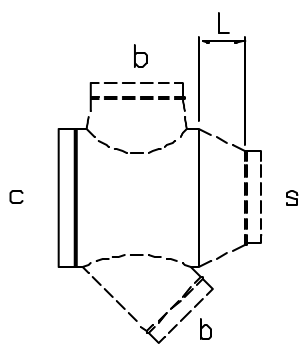


Figure 1.7
Reducing Fitting Construction

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.

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

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

where:

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

$DSP_{c\text{-}s}$ = 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

*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 $C_{r\text{-}o}$ = 0.17 and $C_{o\text{-}r}$ = 0.06.

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

$$\Delta TP_{c\text{-}s\{r\text{-}o\}} = 0.17 \times 0.28 = \underline{0.05 \textit{ inches wg}}$$

$$\Delta TP_{c\text{-}s\{o\text{-}r\}} = 0.06 \times 0.33 = \underline{0.02 \textit{ inches wg}}$$

$$\Delta SP_{c\text{-}s\{r\text{-}o\}} = 0.28\,(0.17 + 1) - 0.33 = \underline{-0.00\ (-0.002) \textit{ inches wg}}$$

$$\Delta SP_{c\text{-}s\{o\text{-}r\}} = 0.33\,(0.06 + 1) - 0.28 = \underline{-0.07 \textit{ inches wg}}$$

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, ΔVP = 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

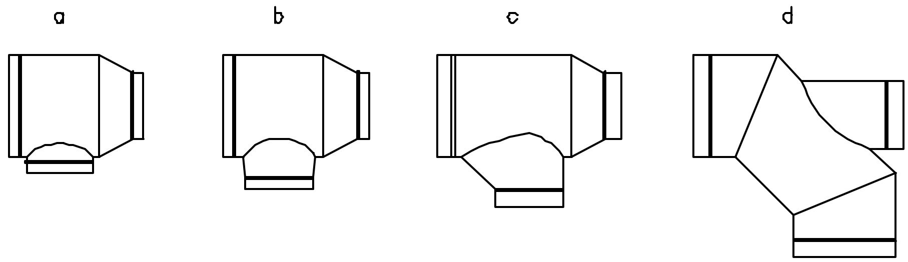
The tee-type diverging-flow fittings discussed in **Section 1.5.3** are generally used where the designer desires to direct a relatively small quantity of air at some angle relative to the main trunk duct, while maintaining a straight-through flow for the majority of the air. Occasionally, situations 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 90° 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°F ± 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**:

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

CHAPTER 2: Designing Supply Duct Systems

2.1 Determination of Air Volume Requirements

Modern buildings are constructed to minimize unintentional exchanges of inside and outside air. The environment inside is expected to be fresh and the temperature and humidity to be nearly constant throughout the year. To accomplish this, it is necessary to introduce conditioned and circulated air to every occupied portion of the building.

The key parameter which quantifies the required amount of air movement for a given occupancy situation is the air volume flow rate. In the English system, the air volume flow rate is expressed in units of cubic feet per minute (*cfm*).

The first step in designing a duct system is determining the required air volume flow rate, and a wealth of information is currently available on this topic. Properly calculating air volume flow rate requirements is rather complex and involves a number of factors. A few of those factors are geographic location (expected temperature and sunlight conditions), building orientation, glass area (radiant heat gain), thermal conductivity of exterior building surfaces, interior heat sources, and desired or required percent of fresh air intake.

Computation of the air volume flow rate is not within the scope of this design notebook. It will be assumed throughout that these values are known and that the air location and air volume flow rates necessary at each terminal device have been established. **Appendix A.9.2** provides an excellent source for information concerning heating and cooling load calculations and the determination of air volume flow rate requirements, and there are many computer programs available to speed the calculations. **Appendix A.9.3** discusses room air distribution, including selection of terminal devices as well as proper location for both heating and cooling.

2.2 Location of Duct Runs

The second step in designing a duct system is laying it out. To do that it is necessary to determine the location of the air handling unit and the terminal devices. **Appendix A.9.3** provides guidelines for locating air diffusers and return grilles and information regarding coverage and air circulation. The air handler will ideally be located in an area which is remote from noise-critical spaces or which is acoustically insulated from them.

Next, taking into account all known architectural, mechanical, and electrical obstructions, a single-line sketch is made, connecting the air handling unit to each terminal or volume control box. Make the duct runs as straight as possible. Each turn will require additional pressure. Always try to maintain the largest volume flow in the straight-through direction, relegating the lesser flows to the branches. This is also a good time to locate and indicate fire and smoke dampers, access doors, and any other duct accessories that may be required.

Here are a few suggestions that may be helpful:

1. Pay particular attention to the available space for installation of duct. Generally, it will be installed between the roof or upper floor support girders and the suspended ceiling. This space is usually not large to start with, and it will be even more restrictive if other trades have already installed their equipment. If flat oval or rectangular duct is used, remember to allow two to four extra inches for the required reinforcement. Round duct under normal positive HVAC pressures (0 to 10 *inches wg*) does not require reinforcement, but you will need to allow for any flanged connections for large duct.

2. Remember that duct will be largest near the air handling unit. If there are space problems, this is the area where they will most likely occur. Try to locate ducts between the girders, if possible, instead of locating them below the girders as larger ducts may be used. Consider multiple runs of small duct in lieu of a single run of large duct. Often this approach minimizes reinforcement and improves acoustics.

3. Duct systems carry noise as well as air. The largest noise source will be the fan. Some noise is also generated by the moving air stream and will propagate down the duct run from the fan. Fan noise will normally be most pronounced near the air handler, and will dissipate as the distance from the fan increases. Remember that fan noise will be propagated through both supply and return ducts. High velocity, high pressure loss fittings, and/or components located in the airstream (tie rods, extractors, etc.) will introduce duct-generated noise. It is good practice to anticipate where the duct noise will be greatest and locate those sections over public areas such as hallways, lobbies, cafeterias, or restrooms. For more information see **Chapters 7, 8** and **9**.

4. Flexible duct is often used for final connections to or from volume control boxes and terminal units. This duct has a very high pressure loss per unit length, and is virtually transparent to noise. It should be avoided whenever possible, and in no case should it be installed in sections longer than 5 feet. Bends, turns, and sags should be avoided, as they will substantially increase the pressure loss and may choke off the air supply to the device they are serving.

2.3 Selection of a Design Method

In the first two design steps, the air volume flow rate for each terminal and the location and routing of the duct were identified. Now the designer must determine the size of the duct, and select the appropriate fittings. These decisions will have a pronounced impact on the cost and operation (efficiency) of the system.

Selection of a design method is not truly a step in the design process, but it is listed here because it provides an opportunity to discuss several options that are available to the designer. Often the design method employed will be the one with which the designer is most familiar or most comfortable. Too often, when asked why they design systems in a certain way, designers respond, "because we've always done it that way."

Before we discuss the mechanics of an actual duct design, let's look at several design methods to compare their features. Terms used below are explained in subsequent selections.

2.3.1 Equal Friction Design

- Probably the most popular design method.
- Quick and easy to use; many nomographs and calculators are available.
- A friction loss per unit length is selected for all duct; this value is usually in the range of 0.05 to 0.2 *inches wg per 100 feet* of duct length.
- All duct is sized using the known air volume flow rates and the selected friction loss.
- Fitting losses can be calculated, but more often are estimated.
- The critical path (maximum pressure requirement) is often chosen by inspection unless a computer program is used to do the calculation.

- System pressure is usually calculated by multiplying the critical path length, plus an allowance for fitting losses, by the design friction loss per unit length.
- System analysis and/or optimization is generally not performed unless a computer program is used.
- Balancing is usually required and is accomplished with dampers or orifice plates.

2.3.2 Constant Velocity Design

- Quick and easy to use; many nomographs and calculators are available.
- A velocity is selected, which will be maintained throughout the system.
- All duct is sized using the known air volume flow rates and the selected velocity.
- Fitting losses can be calculated but more often are estimated (equivalent lengths).
- The critical path (maximum pressure requirement) is chosen by inspection.
- System pressure is calculated by adding the individual pressure losses for each section of the critical path (determined by chart, calculator, or nomograph), plus an allowance for fitting losses.
- System analysis and/or optimization is generally not performed.
- Balancing is usually required and is accomplished with dampers or orifice plates.

2.3.3 Velocity Reduction Design

- Similar to constant velocity design, except that instead of selecting a single, constant velocity for all duct sections, the velocity is systematically reduced in each downstream section.
- This method has questionable application, since there is no valid reason for continually reducing downstream velocities; in fact, it is counterproductive in certain situations.

2.3.4 Static Regain Design

- Probably the most difficult and time-consuming design method, but generally produces the most efficient system (lowest operating pressure, well balanced).
- An initial duct size (or velocity) is selected.
- All duct is sized so that the pressure loss in any duct section is equal to the regain of pressure regain caused by reducing the velocity from the upstream section to the downstream section.
- Fitting losses must be calculated (they are used in sizing the duct).
- The critical path is chosen by inspection or, more often, by computer.
- System pressure is determined by adding the individual (calculated) pressure losses

for all duct and fitting elements of the critical path.

- Static regain designs tend to be more self-balancing than other methods, but if balancing is required, it is usually accomplished with dampers or orifice plates.

2.3.5 Total Pressure Design

- Initially, the system is sized using one of the methods described above.
- The critical path(s) is determined.
- All non-critical legs, by definition, will have excess total pressure; if they are not redesigned or dampered, there will be a system imbalance.
- Duct sizes in the non-critical legs are reduced in size (velocities and pressures are increased) until there is no longer any excess total pressure in the section.
- Alternatively, or in addition, certain fitting types may be altered to reduce the excess total pressure (this generally involves the substitution of a less efficient and less costly fitting).
- Will always produce the lowest first-cost system for a given operating total pressure, and will self-balance the system and virtually eliminate dampering.
- The large number of iterations required by this method is best accomplished by use of a computer program.

2.3.6 Which Design Method?

For those with access to a computer and the necessary software, a total pressure design of a system initially designed by static regain will produce the most efficient, cost-effective system possible. Although the equal friction design is simple and straightforward, it generally will not result in the most efficient or cost-effective system. If systems are small or if the designer does not have access to a computer program, equal friction design with a low friction loss *per 100 feet* (0.05 *inches wg per100 feet* to 0.10 *inches wg per 100 feet*) will be most cost effective from a design time aspect.

In the following sections, both the equal friction and static regain design methods are explored. The total pressure method of design will be discussed in **Chapter 3**.

2.4 Equal Friction Design

2.4.1 Introduction

As mentioned earlier, equal friction is the most commonly used design method. In the following sections the duct is sized, the system pressure is determined, and the excess pressure is calculated. Duct sizing is the key part of this method of design since the other two steps are common to the other methods.

2.4.2 Duct Sizing

Equal friction design is based on the concept that the duct friction loss per unit length at any location in the system should always be the same. The first element of the design, therefore, is to select a friction loss per unit length for the system. Any value can be selected, but many designers prefer values between 0.05 to 0.20 *inches wg per 100 feet* of duct length. One method of choosing a friction loss per unit length is to determine an initial velocity and size the first section for this velocity. Then determine this section's friction loss rate from a friction loss chart or nomograph (duct calculator). The friction loss per unit length is then applied to other sections. Some suggested initial velocity values are given for round systems in **Table 2.1**. Alternatively, a friction loss per unit length is predetermined based on what the designer feels comfortable with, and then the friction loss chart or duct calculator is used to determine sizes.

Once the friction loss per unit length has been selected, one of the two parameters necessary to select duct sizes using the friction loss chart is fixed. (For a review of the use of friction loss charts, see **Section 1.4**.) Since it is assumed that the air volume flow rates for each terminal location are known, the air volume requirements can be determined for all duct sections in the system by applying the mass flow principles from **Section 1.2.2**.

Table 2.1
Suggested Initial Velocities in Round Systems

System Volume Flow Rate (*cfm*)	**Suggested Initial Velocity** (*fpm*)
0 to 15,000	1,500
15,000 to 30,000	2,500
30,000 to 70,000	3,000
70,000 to 100,000	3,500
above 100,000	4,000

Duct diameters are selected from the duct friction loss chart at the intersection of the appropriate unit friction loss per unit length line and the air volume flow rate line for the duct section in question.

Sample Problem 2-1

*Use the equal friction design method to size **Sample System 1** (**Figure 2.1**), with a friction loss of 0.20 inches wg per 100 feet. What is the diameter of each section? What would the diameters be if 0.10 inches wg per 100 feet had been selected as the design friction loss?*

Answer: First, determine the air volume flow rates for each duct in the system. These are shown in **Table 2.2**. Note that the system has been divided into numbered sections (in hexagons), which begin with a divided-flow fitting and end just before the next divided-flow fitting. The air volume flow rates for each section can be calculated by summing all downstream terminal requirements.

Table 2.2
Air Volume Flow Rates for Sample System 1

Section 1	20,600 *cfm*
Section 2, 3, 9	1,600 *cfm*
Section 4	17,400 *cfm*
Section 5	2,400 *cfm*
Section 6	15,000 *cfm*
Section 7	3,200 *cfm*
Section 8	11,800 *cfm*
Section 10	10,200 *cfm*
Section 11	3,600 *cfm*
Section 12	6,600 *cfm*

Next, calculate the duct sizes. For Section 1, refer to the duct friction loss chart in **Figure 2.2** and locate the intersection of the 0.20 *inches wg per 100 feet* friction loss line (horizontal axis), and the 20,600 *cfm* volume flow rate line (vertical axis). The 20,600 *cfm* line will have to be estimated. The intersection of these lines is approximately at the 38-*inch* duct diameter line. The friction loss chart also indicates that the air velocity in this section will be approximately 2,600 *fpm*.

Therefore, an air volume flow rate of 20,600 *cfm* will flow through a 38-*inch* diameter duct at a velocity of approximately 2,600 *fpm* and it will have a friction loss of about 0.20 *inches wg per 100 feet* of duct length.

Similarly, for Section 2 the intersection of the 0.20 *inches wg per 100 feet* friction loss line and the 1,600 *cfm* line is almost equidistant between the 14-*inch* and the 15-*inch* duct diameter lines. In this case, a 14.5-*inch* diameter duct would be the best choice. Some manufacturers will manufacture half sizes, but for simplicity this design will keep sizes in whole inch increments. Using a 14-*inch* duct to carry 1,600 *cfm* would actually require 0.23 *inches wg per 100 feet*, whereas a 15-*inch* duct would only require 0.17 *inches wg per 100 feet*. The larger duct size will be used. All remaining sections can be sized in a similar manner.

If the design were to be based on a friction loss of 0.10 *inches wg per 100 feet*, the procedure would be identical, except that the duct size would be selected at the intersection of the friction loss line representing 0.10 *inches wg per 100 feet* and the required air volume flow rate. In this case, Section 1 (20,600 *cfm*) would require a 44-*inch* diameter duct.

Figure 2.2 shows how these values are obtained from the friction loss chart. Dashed lines are shown on the chart for the 0.10 *inches wg per 100 feet* friction loss and the 0.20 *inches wg per 100 feet* friction loss. **Table 2.3** shows the duct sizes as determined by each design method.

Table 2.3
Equal Friction Duct Sizing for Sample System 1

Section	**Volume Flow Rate** (*cfm*)	**Duct Diameter** (*inches*)*	
		0.20 *inches wg per 100 feet*	0.10 *inches wg per 100 feet*
1	20,600	38	44
2,3,9	1,600	15	17
4	17,400	36	42
5	2,400	17	20
6	15,000	34	40
7	3,200	19	22
8	11,800	31	36
10	10,200	29	34
11	3,600	20	23
12	6,600	25	29

* Rounded to the nearest whole diameter

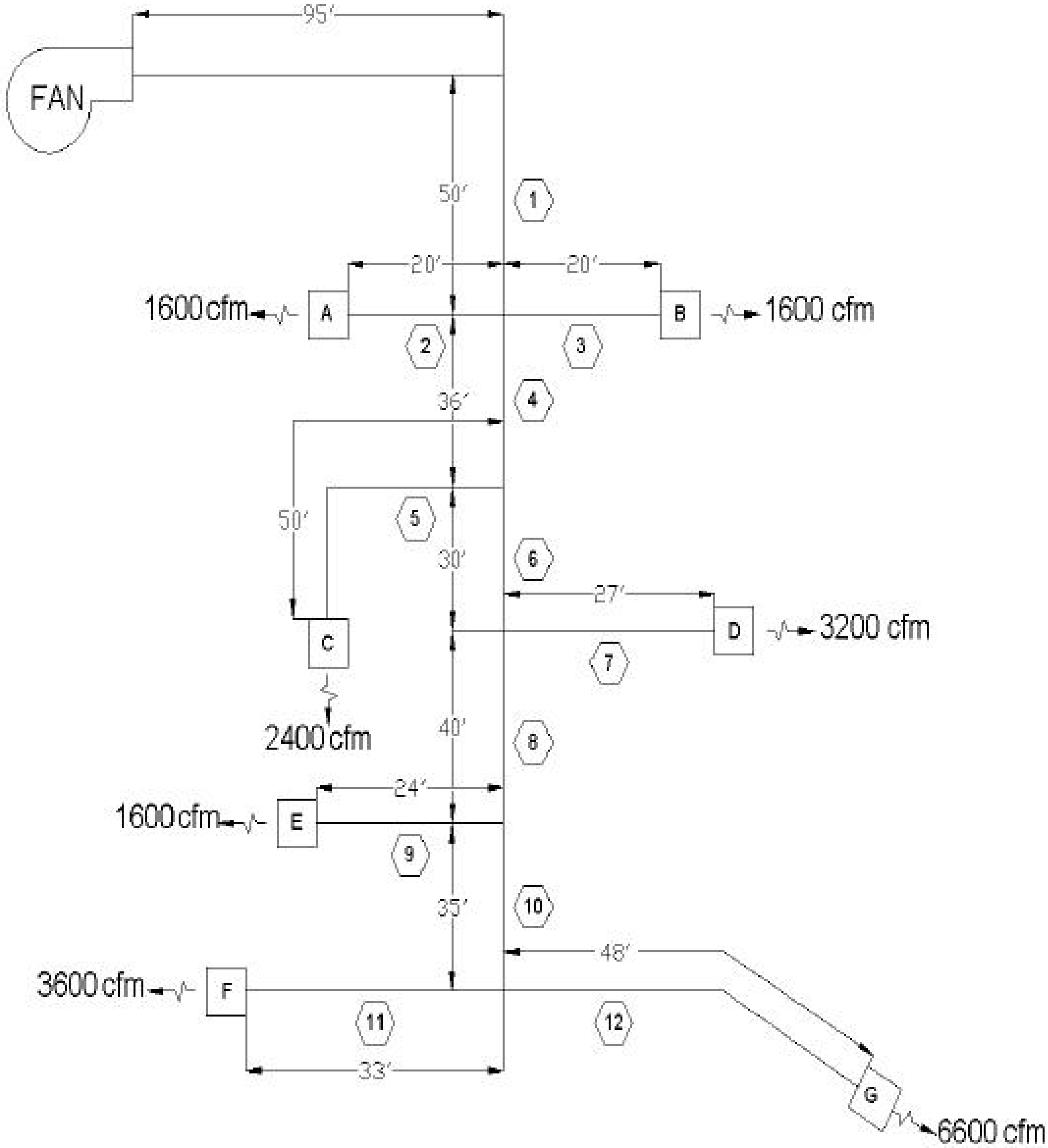


Figure 2.1
Sample System 1

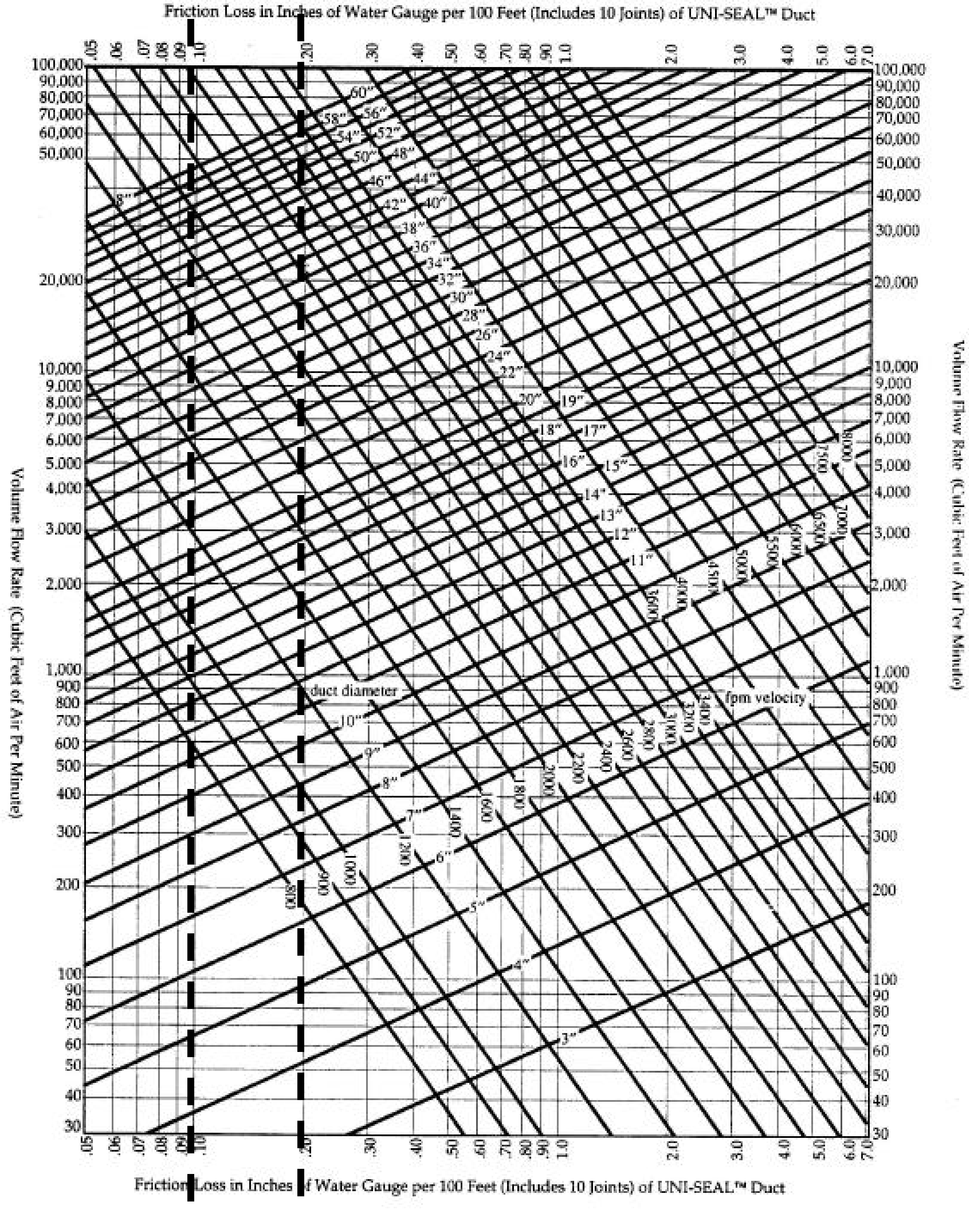


Figure 2.2
Friction Loss of Duct

From **Sample Problem 2-1**, a very important implication of the friction loss per unit length selection is apparent. When a higher (0.20 *inches wg per 100 feet*) friction loss per unit length is selected, the duct sizes in all sections are smaller than for the lower (0.10 *inches wg per 100 feet*) friction loss per unit length. Smaller duct sizes will result in a less expensive system to purchase and install. However, in subsequent sections, the selection of a higher friction loss per unit length will have negative consequences in terms of system pressure and operating costs and may result in substantially increased total costs.

2.4.3 Determination of System Pressure

Once the system air volume flow rate requirements are known, the duct can be sized. The designer knows that if the required volume of air flows through the ducts, it will do so at the unit pressure losses and velocities indicated by the friction loss chart. The next step is to make certain that the required volume flow rate of air will, in fact, flow through the ducts.

To supply the required volume flow rate in each duct section, the air handler must be capable of delivering the total system volume flow (20,600 *cfm* for **Sample System 1**). The air handler must supply this volume of air at a total pressure sufficient to overcome the resistance (frictional and dynamic) of the path having the highest pressure loss. This path is called the critical path or design leg, and the sum of all the total pressure losses from the air handler to the terminal of the critical path is called the system (total) pressure.

The next step in the design process is to identify the critical path and to determine its pressure loss.

System Pressure by Inspection

The simplest means of identifying the critical path and determining the system pressure is by inspection. Ideally, this should be done only by an experienced designer who has analyzed numerous systems and has a good feeling for the pressure losses associated with various system components.

It is often the case that the critical path is simply the longest path in a system. If frictional (duct) losses accounted for all the pressure losses, this would usually be the case; however, the dynamic (fitting) losses often are substantial and may cause one of the shorter paths to have a higher pressure loss.

In an equal friction design, all ducts should have the same pressure loss per unit length. Therefore, once the critical path is identified, the duct losses can be calculated by simply multiplying the design pressure loss (*inches wg per 100 feet*) by the total length of the critical path divided by 100 *feet*.

The critical path fitting losses should be calculated and added to the duct loss, although often only a very rough estimation is made. An acceptable compromise may be to ignore straight-through losses, which normally will be very small, but calculate and include the losses of any branches or abrupt-turn fittings that are in the critical path.

Sample Problem 2-2 will illustrate the inspection method for calculating the system pressure loss of **Sample System 1** (**Figure 2.1**).

Sample Problem 2-2

*Determine the critical path by inspection and estimate the system total pressure loss for **Sample System 1** (**Figure 2.1**), based on the 0.20 inches wg per 100 feet design. The fittings in Sections 1 and 5 are five-gore 90E elbows; all branch fittings are 90E conical tees; a bullhead tee with turning vanes is used at the Sections 10-11-12 junction; and there is a 45E, 3-gore elbow in Section 12.*

Answer: Because of the high volume flow rate requirement (6,600 *cfm*) and the distance from the fan, it is reasonable to assume that the critical path will be the duct run to Terminal G. The path includes Sections 1, 4, 6, 8, 10 and 12. The total center line length of these sections is 334 feet (95 + 50 + 36 + 30 + 40 + 35 + 48).

If the equal friction design is based on a friction loss of 0.20 *inches wg per 100 feet*, the total duct loss would be approximately:

334 feet H 0.20 *inches wg per 100 feet* = <u>0.67 *inches wg*</u>

Since bullhead tee losses are a function of volume flow rate and area ratios, it is necessary to know the volume flow rate and size of duct in Section 10 (common) and Section 12 (branch).

Q_b = 6,600 *cfm*	A_b = 3.41 *sq ft*	VP_b = 0.23 *inches wg*
Q_c = 10,200 *cfm*	A_c = 4.59 *sq ft*	VP_c = 0.31 *inches wg*
Q_b/Q_c = 0.65	A_b/A_c = 0.74	

Referring to **ASHRAE Fitting SD5-19**, the loss coefficient is approximately 0.67.

Substituting in **Equation 1.16b**, the total pressure loss is:

DTP_{c-b} = 0.23 x 0.67 = <u>0.15 *inches wg*</u>

In this case, the bullhead tee pressure loss is 22 percent of the duct loss and is therefore significant. It would also probably be wise to calculate the loss of the large elbow in Section 1. This is easily done by referring to **ASHRAE Fitting CD3-9**. At 2,616 *fpm*, a 38-*inch* diameter elbow will have a static pressure loss of approximately 0.05 *inches wg*.

($DTP = C$ x VP = 0.12 x 0.43 = 0.05 *inches wg*)

The loss for the 45E, 3-gore elbow in Section 12 is determined using **ASHRAE Fitting CD3-14** whereby DTP = 0.08 x 0.23 = 0.02 *inches wg*.

The total pressure loss of a path must also include the velocity pressure of the terminal section as well since this pressure will be lost when the air exits the section and must also be supplied by the fan. Therefore, a reasonable estimation of the pressure loss from the fan to Terminal G is:

0.67 + 0.15 + 0.05 + 0.02 + .23 = <u>1.12 *inches wg*</u>

If our critical path was selected correctly, this will be close to the system total pressure requirement, as determined by inspection.

System Pressure by Calculation

The best and most accurate way to determine system pressure losses is by calculation. There are many ways to approach these calculations, and many computer programs are now available to simplify the process. However, the longhand method is presented here since it will give the designer a better feel for the process.

In an equal friction design, the calculated pressure loss will differ from the estimated loss, as described in the preceding **Sample Problem 2-2**, in two ways: (1) the actual duct losses are used, and (2) all fitting losses are calculated and included.

Sample Problem 1 (**Figure 2.1**) will be used again for the analysis. **Table 2.4** is useful in collecting and analyzing the data necessary to properly calculate the system losses. It is grouped by section. Each section consists of duct and at least one branch fitting. The first section begins at the fan and ends just upstream of the first flow division; all subsequent sections begin with a flow division branch which is called the takeoff fitting, and ends just upstream of the subsequent downstream flow division. Elbows, constant area offsets, dampers, and other fittings that do not result in a change of area, shape, or air volume do not create new sections; their total pressure losses are included in the sections in which they occur.

For each section, the duct and fitting elements are listed, along with the following information:

1. Section velocity
2. Velocity pressure
3. Section velocity
4. Velocity pressure
5. Actual duct friction loss per 100 feet
6. Duct length
7. Actual duct loss
8. Fitting static pressure losses
9. Cumulative section static pressure loss
10. Fitting total pressure losses

Regarding 7, it is important to realize that, even in an equal friction design where the friction loss rate is supposedly constant, it will vary slightly from section to section. This is because the exact duct diameter required to move a given volume flow rate at the design pressure is seldom available.

Table 2.4
Sample System 1 - Equal Friction Design Data Sheet

SECT	ITEM REF[1]	ITEM	Q (cfm)	DIA. ($inches$)	AREA (ft^2)	V (fpm)	VP ($inches.\ wg$)	C	$\Delta P/100\ ft$ ($inches\ wg$)	L (ft)	ΔP DUCT ($inches\ wg$)	FITTING ΔSP ($inches\ wg$)	SECTION ΔSP ($inches\ wg$)	FITTING ΔTP ($inches\ wg$)	SECTION ΔTP ($inches\ wg$)
1	CD3-9	Duct 90 Elbow	20,600	38	7.88	2,616	0.43	0.12	0.20	145	0.29	0.05	0.34	0.05	0.34
4	SD5-25	Duct Straight T/O	17,400	36	7.07	2,462	0.38	0.14	0.19	36	0.07	0.00	0.07	0.05	0.12
6	SD5-10	Duct Straight T/O	15,000	34	6.31	2,379	0.35	0.13	0.19	30	0.06	0.02	0.08	0.05	0.11
8	SD5-10	Duct Straight T/O	11,800	31	5.24	2,251	0.32	0.13	0.19	40	0.08	0.00	0.08	0.04	0.12
10	SD5-10	Duct Straight T/O	10,200	29	4.59	2,224	0.31	0.13	0.20	35	0.07	0.03	0.10	0.04	0.11
2 and 3	SD5-25	Duct Con Cross	1,600	15	1.23	1,304	0.11	1.38	0.17	20	0.03	(0.17)	(0.14)	0.15	0.18
5	SD5-10 CD3-9	Duct Con Tee 90 Elbow	2,400	17	1.58	1,523	0.14	2.80 0.15	0.19	50	0.10	0.17 0.02	0.29	0.40 0.02	0.52
7	SD5-10	Duct Con Tee	3,200	19	1.97	1,625	0.16	1.60	0.19	27	0.05	0.08	0.13	0.26	0.31
9	SD5-10	Duct Con Tee	1,600	15	1.23	1,304	0.11	3.22	0.17	24	0.04	0.13	0.17	0.34	0.38
11	SD5-19	Duct Bullhead T/V	3,600	20	2.18	1,650	0.17	1.03	0.18	33	0.06	0.04	0.10	0.18	0.24
12	SD5-19 CD3-14	Duct Bullhead T/V 45 Elbow	6,600	25	3.41	1,936	0.23	0.67 0.08	0.19	48	0.09	0.28 0.02	0.39	0.16 0.02	0.27

1 – ITEM REF column lists the fitting designation as given by ASHRAE.

The velocities and pressure losses in the example above were read, as accurately as possible, from the friction loss chart in **Appendix A.4.1.1** or **Figure 2.2.** Nomographs, calculators or computers can also be used, and may provide slightly more accurate results. Note that the pressures attributable to the duct and all fittings within a section are totaled for each section. The **ASHRAE Duct Fitting Database** program was used to calculate the losses of fittings.

To determine the system pressure loss, it is again necessary to find the critical path. This time, however, there is no guesswork involved. Since actual pressure loss data is available for each section, the pressure requirement can be determined for all paths. The path with the highest pressure requirement (fan to terminal) is the critical path, and this pressure is the system pressure requirement. **Table 2.5** analyzes the terminals of **Sample System 1**. Remember that the velocity pressure in the terminal section must be added to determine the path's total pressure requirements.

Table 2.5
Path Total Pressures for Sample System 1 Equal Friction Design

Terminal	**Path (Sections)**	**Section Total Pressure Losses** (*inches wg*)	**Path Total Pressure Required** (*inches wg*)
A	1,2	0.34 + 0.18 + 0.11	0.63
B	1,3	0.34 + 0.18 + 0.11	0.63
C	1,4,5	0.34 + 0.12 + 0.52 + 0.14	1.12
D	1,4,6,7	0.34 + 0.12 + 0.11 + 0.31 + 0.16	1.04
E	1,4,6,8,9	0.34 + 0.12 + 0.11 + 0.12 + 0.38 + 0.11	1.18
F	1,4,6,8,10,11	0.34 + 0.12 + 0.11 + 0.12 + 0.11 + 0.24 + .17	1.21
G	1,4,6,8,10,12	0.34 + 0.12 + 0.11 + 0.12 + 0.11 + 0.27 + 0.23	1.30

From this analysis, we can see that the pressure required to deliver the design volume flow of air to Terminal G is greater than that required for any other path. The critical path, therefore, is from the fan to Terminal G, and the system total pressure requirement is 1.30 *inches wg*. To determine the system static pressure requirement, subtract the initial velocity pressure. In this case, the initial velocity was 2,616 *fpm* (*VP* = 0.43 *inches wg*), so the system static pressure is 0.87 *inches wg*.

The inspection method of **Sample Problem 2-2** provided the correct critical path but the pressure estimate (1.12 *inches wg*) was low by 0.18 *inches wg*. The main reason is because the straight-through losses were ignored. It was guesstimated that the path to Terminal G would be the critical path, which it was. However if the system pressure was estimated at 1.12 *inches wg* and the true pressure required at the fan is 1.30 *inches wg*, then the fan may be selected incorrectly to supply the lower pressure.

In analyzing real systems, it is important to note that there will be other losses, in addition to the duct and fitting losses discussed above. Probably most significant is the pressure loss associated with terminal devices. Typically, diffusers and registers have a pressure requirement ranging from 0.01 to 0.5 *inches wg*. If the system has volume control boxes, there must be sufficient residual pressure at the end of each path to power the box (or overcome the internal resistance of the device) and also to move the air through any low pressure or flexible duct on the downstream side of the box. Losses for the terminal devices should be added to the pressure requirements of each path prior to the determination of the critical path and system pressure.

2.4.4 Excess Pressure

In the preceding equal friction problem, the system operates at 1.30 *inches wg* to deliver air to the critical path (Terminal G). This is more than the pressure necessary to operate the other paths to Terminals A through F. If the system is turned on as designed, air will always follow the path of least resistance, and this will result in excess volume flow through the paths to Terminals A and B and progressively less air through the downstream terminals as the flow resistance increases. The critical path terminal will not get its required volume, while other terminals will get too much air.

Looking again at the pressure requirements for the system, the amount of excess pressure which is present at each terminal can be computed by calculating the difference between the system (critical path) pressure of 1.30 *inches wg* and the individual path pressures.

Table 2.6
Excess Pressure for Sample System 1 Equal Friction Design

Terminal	Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)
A	0.63	0.67
B	0.63	0.67
C	1.12	0.18
D	1.04	0.26
E	1.18	0.12
F	1.21	0.09
G	1.30	0.00

In order to balance the system, it is necessary to add resistance to each noncritical path, equal to the excess pressure of the path. In this way, each path essentially becomes a critical path and the design volume flow rate will be delivered to each terminal, as long as the air handler provides the total volume flow rate (20,600 *cfm*) at the correct initial velocity (2,616 *fpm*) and system total pressure (1.30 *inches wg*).

The simplest way to introduce resistance to a system is probably with the use of balancing dampers. These dampers can be adjusted to provide a pressure drop equal to the excess pressure for each flow path. Once adjusted, they should be permanently secured to avoid tampering, which will destroy the balance. Obviously, this method of balancing is appropriate only for inflexible constant-volume systems. Balancing is not as easily maintained in a variable air volume (VAV) system. Most often the balancing dampers in a VAV box are capable of handling excess pressure; however, the amount of excess pressure can affect VAV box performance.

Although the use of dampers is simple, it is not necessarily cost effective. **Chapter 3** will discuss an efficient means of balancing systems without the use of dampers.

2.5 Static Regain Design

2.5.1 Introduction

The key concept of static regain design is that the magnitude of the static pressure must be maintained as constant as possible throughout the system. At first this seems to be a contradiction, because static pressure is required to overcome duct friction, and duct friction is produced everywhere air flows over a duct surface. It seems only logical that the static pressure will progressively reduce as air flows from the fan to the terminals.

Remember, however, that static pressure and velocity pressure combine to yield total pressure, and it is the total pressure, not the static pressure, which is irretrievably lost as the air flows through the system. At any location where the duct area and/or volume flow rate change, there will usually be a change in air velocity. Velocity pressure will increase or decrease as the square of the velocity changes.

Referring to **Figure 1.3**, recall that as velocity pressure increases (Case II) there will always be a reduction of static pressure; however, a reduction in velocity pressure can be accompanied by a decrease (Case III) or an increase (Case I) in static pressure. Where static pressure increases, this phenomenon is called static regain, and is the basis of this design method. If the reduction in velocity pressure is greater than the total pressure loss, static regain will occur. If the total pressure loss is greater than the reduction in velocity pressure, then static regain can not occur. The following examples illustrate this concept:

1. In **Sample Problem 1-16**, a straight tee fitting produced a total pressure loss of 0.17 *inches wg*, upstream-to-branch. There was a reduction of 0.08 *inches wg* velocity pressure. This resulted in a static pressure loss of 0.09 *inches wg*. No static regain is possible in this situation.

2. **Sample Problem 1-17** substituted a conical tee and LO-LOSS™ tee under the same conditions. The LO-LOSS™ resulted in a total pressure loss of only 0.06 *inches wg*, upstream-to-branch. Since the velocity pressure was reduced by 0.08 inch wg., this condition resulted in a static pressure regain (increase) of 0.02 *inches wg* (which is written as a negative loss or -0.02 *inches wg*).

3. **Sample Problem 1-19** considered the straight-through leg of the straight tee (which would be the same for conical tee or LO-LOSS™ tee as well). Here the total pressure loss (upstream-to-downstream) was just 0.01 *inches wg*, but the velocity was reduced from 1,592 *fpm* to 955 *fpm*, resulting in a velocity pressure reduction of 0.10 *inches wg*. In this case there was a static pressure regain of 0.09 *inches wg*.

Static regain design uses the increase in static pressure, especially in straight-through legs of divided-flow fittings, to create a well-balanced system.

2.5.2 Duct Sizing

In static regain design, the first section (connecting to the fan or plenum) is sized for a desired velocity. This velocity may be any value, but the rules of thumb in **Table 2.1** are often used for round and flat oval ductwork. However, if silencers or rectangular ductwork are involved, velocities are generally held to 2,500 *fpm* maximum in order to minimize additional pressure drop.

To determine the optimum velocity, a cost analysis would have to be performed. This is demonstrated in **Section 3.5**.

All downstream duct is then sized so that the total pressure loss in each section is just equal to the reduction in velocity pressure such that the change in static pressure is zero from the beginning of one branch fitting or takeoff to the beginning of the next branch or takeoff fitting. Selecting a large duct size will result in greater regain in the fitting and less pressure loss in the duct and section=s fittings. By reducing the duct size, the static regain will be reduced (since the change in velocity pressure is reduced) and the duct and fitting pressure losses will be increased. When the velocity pressure reduction is about equal to the section=s duct and fitting losses, the section is correctly sized. This will result in the section static pressure loss being equal to approximately zero. Thus the goal of maintaining the same static pressure throughout the system is approached, which means the system will be balanced. However in many branch systems, it is almost impossible to decrease the velocity enough to offset the section=s losses without increasing the duct size above the upstream size. Therefore, when designing downstream sections, it is usual and practical to begin with the size of the section just upstream. The end result will be some imbalance in the system, which either must be dampered, or sections resized after the initial design to remove the excess pressure, which is the topic of **Chapter 3 Analyzing and Enhancing Positive Pressure Air Handling Systems**.

The following sample problem sizes the duct sections of **Sample System 1 (Figure 2.1)** using the Static Regain Design method.

Sample Problem 2-3

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*Refer to **Sample System 1** (**Figure 2.1**). Assume that Section 1 is sized at 38 inches and will carry 20,600 cfm. Use the static regain method to determine the size of Section 4.*

Answer: A 38-*inch* duct carrying 20,600 *cfm* will have a velocity of 2,616 *fpm*. First, consider keeping the downstream diameter (Section 4) the same size at 38 *inches*. Duct Section 4 is known to be 36 *feet* long and will carry 17,400 *cfm*. A 38-*inch* diameter duct at 17,400 *cfm* would have a velocity of 2,209 *fpm* and from the friction loss chart, a friction loss of 0.15 *inches wg per 100 feet*. The friction loss of the 36-*foot* duct section will be (36/100) H 0.15, or 0.05 *inches wg*. From ASHRAE Fitting **SD5-25**, at 2,209 *fpm* straight-through (downstream) and 2,616 *fpm* common (upstream), DTP_{c-s} = 0.04 *inches wg* across the straight-through portion of the cross fitting. The total section pressure loss is 0.09 *inches wg*. The velocity pressure drop, DVP_{c-s}, across the straight-through is 0.12 *inches wg*. Since the drop in velocity pressure exceeds the total pressure drop by 0.03 *inches wg*, this will be the static regain in Section 4. Since the static regain is greater than zero (in magnitude), we will try a smaller size. Try using 37-*inch* diameter duct.

A 37-*inch* diameter duct at 17,400 *cfm* will have a velocity of 2,330 *fpm*, and a friction loss of 0.17 *inches wg per 100 feet*. The friction loss of the 36-*foot* duct section will be (36/100) H 0.17, or 0.06 *inches wg*. DTP_{c-s} is 0.05 *inches wg* across the straight-through portion of the cross fitting. The total section pressure loss is 0.11 *inches wg*. The velocity pressure drop, DVP_{c-s}, across the straight-through is 0.09 *inches wg*. Since the drop in velocity pressure is 0.02 *inches wg* less than the total pressure drop, there is a static pressure loss of 0.02 *inches wg* and therefore no static regain.

To get a static regain (or pressure loss) equal to zero, we would need to use a duct size between 37 and 38 *inches* in diameter. This is usually not practical and either the 37- or 38-*inch* duct would be acceptable here. Use the size that either gives the closest to zero static regain. If the choices are about equal, use the size which results in the lowest cost, considering the cost of a reducer if it must be added to use a smaller size. For our example problem we will keep the 38-*inch* diameter for Section 4.

There are several important points to keep in mind when using static regain design:

1. The straight-through (downstream) duct sizes should generally not be made larger than common (upstream) duct sizes for the sole purpose of achieving a static regain. If it is not possible to achieve regain using a constant diameter duct (downstream the same size as upstream), then size the section the same as the upstream duct diameter and continue with the design, trying to achieve regain in the next downstream section.

2. When a section branches off the main trunk duct, it may not be possible to achieve regain with certain branch fittings. The straight tee in **Sample Problem 1-16** presents such an example. In these cases, substitution of a more efficient fitting may lower the total pressure loss enough that static regain is possible (see the LO-LOSS tee of **Sample Problem 1-17**). If not, use the largest duct size that is compatible with the space restrictions and minimum velocity criterion (see point 4, below).

3. Consider using a constant diameter duct until a reduction can be justified by less cost. Often a reducer and smaller size duct will cost more than keeping the same size if the reduction is only an *inch* or two and the size is large. The choice between the same size and next lower size should only occur if the static regain of the larger size is about equal in magnitude to the static pressure loss of the smaller size.

4. Select a minimum velocity, and do not use (larger) duct sizes that will cause the velocity to fall below this value. Especially on divided-flow fitting branches, a point may be reached where a larger duct size produces no measurable difference in the fitting pressure loss.

Table 2.7 is a worksheet that shows the iterations and results of a static regain analysis of **Sample System 1** (**Figure 2.1**). Section 1 is sized at 38 *inches*, the fittings are described in **Sample Problem 2-2**, and a minimum velocity of 900 *fpm* is to be maintained. Straight-through sections are treated first (Sections 4, 6, 8 and 10), followed by the branches (Sections 2, 3, 5, 7, 9, 11 and 12). An asterisk shows selected sizes. The **ASHRAE Duct Fitting Database** was used to determine the loss coefficients.

The following comments may be helpful in understanding how sizes were selected.

<u>Section 4</u>: See **Sample Problem 2-3**.

Section 6: The first size iteration is 38 *inches*, the upstream diameter. A slight overall static regain (DSP = -0.02 *inches wg*) is achieved, so a second iteration is made at 37 *inches*. This produces a section static pressure loss of 0.02 *inches wg*. Any duct size smaller than 37 *inches* will further increase the section static pressure loss. Therefore, the 38-*inch* duct is selected (see comment 3, above).

Section 8: The first iteration is again 38 *inches* which produces a section static regain (DSP = -0.04 *inches wg*). A 37-*inch* duct produces a slight static regain (DSP = -0.01 *inches wg*) , so we should not use 38-*inch* duct again since it would result in an even greater regain. A 36-*inch* duct produces a slight section static pressure loss (DSP = 0.01 *inches wg*), but this size is selected instead of the 37-*inch* duct (see comment 3, above).

Section 10: The upstream (Section 8) diameter of 36 *inches* is selected for the first iteration, and this result in a section static regain (DSP = -0.00 *inches wg* to two decimal places). Since this is what we want we will keep the 36-*inch* diameter.

Sections 2 and 3: These branches are identical, so they are considered together. Selecting the first iteration size for straight-through sections is a simple matter. It is usually the upstream section size or 1 or 2 *inches* smaller. Branches sizes are not as simple to select. In this case, the first iteration was selected at a size that would yield a velocity near the minimum. This resulted in a slight regain (DSP = -0.02 *inches wg*). A second iteration at 16-*inch* diameter resulted in a slight static pressure loss (DSP = 0.01 *inches wg*). This is closer to zero so this size is selected.

Section 5, 7, 9 and 11: The first iterations for these sections were also based on sizes the produced air flow close to the minimum velocity. In each of these cases there was already no static regain. Since we could not make the duct sizes larger without violated the minimum velocity constraint, these sizes were chosen.

Section 12: This section is a fairly long run and has a higher air volume flow rate than the other branches, but the first iteration was again based on the size that produced a velocity close to the minimum, which is 36 *inches*. This produced a section static pressure regain (DSP = -0.03 *inches wg*). A 34-*inch* duct resulted in a 0.03 *inches wg* section static pressure loss. Either 34-*inch* or 36-*inch* duct will be acceptable from a pressure loss standpoint, so a 34-*inch* diameter is selected to save in material cost.

There is more information given in **Table 2.7** than is actually necessary to perform the design calculations. This is done to show the relationships among static, velocity, and total pressure losses in fittings and sections.

For the bullhead tee, it is necessary to determine the loss coefficient from **ASHRAE Fitting SD5-19**. The static and total pressure losses can then be determined from **Equations 1.17b** and **1.17c**. The section static pressure loss (DSP_{sect}) is a summation of DP_{duct} (remember for duct, DSP = DTP) and DSP_{ftg}. For an ideal static regain design, this loss should be zero.

The section total pressure loss (DTP_{sect}) is important as a measure of total system energy consumption. The following equations will always apply, and can serve as a data check.

$$DTP_{sect} = DSP_{sect} + DVP_{sect} \qquad \textbf{Equation 2.1}$$

$$DTP_{sect} = DP_{duct} + SDTP_{ftgs} \qquad \textbf{Equation 2.2}$$

2.5.3 Determination of System Pressure

To determine the system pressure loss, it is necessary to find the critical path, as was done for the equal friction problem. Remember that the path with the highest pressure requirement (fan to terminal) is the critical path, and this pressure is the system pressure requirement. Therefore, it is necessary to identify the critical path and to determine its pressure loss. Determination of the total pressure losses for a system that has already been sized using the Static Regain design method is accomplished by adding the pressure losses for each section in a path from the fan to the terminal.

Table 2.8 represents the pressure loss analysis for **Sample System 1**, based on the static regain analysis of **Table 2.7**. The critical path is from the fan to Terminal C and requires 0.86 *inches wg* to deliver the required volume flow of air. The system static pressure requirement is 0.43 *inches wg, which* is determined by subtracting the initial velocity pressure of 0.43 *inches wg* from the system total pressure requirement.

2.5.4 Excess Pressure

The amount of excess pressure that is present at each terminal can be computed by calculating the difference between the system (critical path) pressure and the individual path pressures. Results of the evaluation of excess pressure for **Sample System 1** are shown in **Table 2.9**.

It is interesting to note that in the equal friction problem presented previously the terminals closest to the fan had the highest excess pressure and the terminals away from the fan had the highest path pressure requirements. However, static regain design creates the possibility of regain at each straight-through fitting, so sections remote from the fan (which have benefited from several straight-through regains) may actually have lower total pressure requirements than those sections closer to the fan. Also the static regain design required less overall pressure (0.86 *inches wg* compared to 1.30 *inches wg*) even though the first section sizes were the same for both. The static regain design was also much better balanced with excess pressures in the range of 0.05 *inches wg* to 0.13 *inches wg* compared to the equal friction design which has excess pressures in the range of 0.09 *inches wg* to 0.51 *inches wg*.

Excess pressures of 0.05 *inches wg* or less are considered negligible. Therefore, this static regain designed system is still not considered fully balanced and will require additional balancing dampers or balancing by use of VAV box dampers.

Table 2.7
Sample System 1 - Static Regain Design Data Sheet

SECT	ITEM REF	Q (cfm)	DIA. $(inches)$	AREA (ft^2)	V (fpm)	VP $(inches\ wg)$	C	$\Delta P/100\ ft$ $(inches\ wg)$	L (ft)	ΔP DUCT $(inches\ wg)$	FITTING ΔSP $(inches\ wg)$	SECTION ΔSP $(inches\ wg)$	FITTING ΔTP $(inches\ wg)$	SECTION ΔTP $(inches\ wg)$
1*	CD3-9	20600	38	7.88	2616	0.43	0.12	0.20	145	0.29	0.05	0.34	0.05	0.34
4a*	SD5-25	17400	38	7.88	2209	0.30	0.14	0.15	36	0.05	-0.08	-0.03	0.04	0.09
4b	SD5-25	17400	37	7.47	2330	0.34	0.14	0.17	36	0.06	-0.04	0.02	0.05	0.11
6a*	SD5-10	15000	38	7.88	1905	0.23	0.13	0.11	30	0.03	-0.05	-0.02	0.03	0.06
6b	SD5-10	15000	37	7.47	2009	0.14	0.13	0.07	30	0.04	-0.02	0.02	0.03	0.07
8a	SD5-10	11800	38	7.88	1498	0.14	0.15	0.07	40	0.03	-0.07	-0.04	0.02	0.05
8b	SD5-10	11800	37	7.47	1580	0.16	0.15	0.08	40	0.03	-0.05	-0.01	0.02	0.06
8c*	SD5-10	11800	36	7.07	1669	0.17	0.14	0.09	40	0.04	-0.03	0.01	0.02	0.06
10*	SD5-10	10200	36	7.07	1443	0.13	0.13	0.07	35	0.03	-0.03	-0.00	0.02	0.05
2 and 3 a	SD5-25	1600	18	1.77	905	0.05	6.68	0.07	20	0.01	-0.03	-0.02	0.34	0.36
2 and 3 b*	SD5-25	1600	16	1.40	1146	0.08	4.01	0.12	20	0.02	-0.02	0.01	0.33	0.35
5*	SD5-10 CD3-9	2400	22	2.64	909	0.05	6.75 0.14	0.06	50	0.02	0.10 0.01	0.13	0.35 0.01	0.38
7*	SD5-10	3200	25	3.41	939	0.05	3.33	0.05	27	0.01	0.01	0.03	0.18	0.19
9*	SD5-10	1600	18	1.77	905	0.05	3.72	0.07	24	0.02	0.07	0.08	0.19	0.21
11*	SD5-19	3600	27	3.98	905	0.05	1.51	0.04	33	0.01	-0.00	0.01	0.08	0.09
12a	SD5-19 CD3-14	6600	36	7.07	934	0.05	1.19 0.07	0.03	48	0.02	-0.05 0.00	-0.03	0.03 0.00	0.05
12b*	SD5-19 CD3-14	6600	34	6.31	1047	0.07	1.00 0.07	0.04	48	0.01	0.01 0.00	0.03	0.07 0.00	0.08

Table 2.8
Path Total Pressure for Sample System 1 Static Regain Design

Terminal	**Path (Sections)**	**Section Total Pressure Losses** (*inches wg*)	**Path Total Pressure Required** (*inches wg*)
A	1,2	0.34 + 0.35 + 0.08	0.77
B	1,3	0.34 + 0.35 + 0.08	0.77
C	1,4,5	0.34 + 0.09 + 0.38 + 0.05	0.86
D	1,4,6,7	0.34 + 0.09 + 0.06 + 0.19 + 0.05	0.73
E	1,4,6,8,9	0.34 + 0.09 + 0.06 + 0.06 + 0.21 + 0.05	0.81
F	1,4,6,8,10,11	0.34 + 0.09 + 0.06 + 0.06 + 0.05 + 0.09 + 0.05	0.74
G	1,4,6,8,10,12	0.34 + 0.09 + 0.06 + 0.06 + 0.05 + 0.08 + 0.07	0.75

Table 2.9
Excess Pressure for Sample System 1 Static Regain Design

Terminal	**Path Total Pressure Required** (*inches wg*)	**Excess Pressure** (*inches wg*)
A	0.77	0.09
B	0.77	0.09
C	0.86	0.00
D	0.73	0.13
E	0.81	0.05
F	0.74	0.12
G	0.75	0.11

CHAPTER 3: Analyzing and Enhancing Supply Duct Systems

3.1 Analyzing a Preliminary Supply Design

After a duct system has been initially designed using one of the methods discussed in **Section 2.3**, it should be reviewed to see if improvements can be made. The first step in analyzing a system is to determine the total pressure required to operate it. The method of determining the system's total pressure requirements is shown in **Section 2.4.3** for the equal friction design method and **Section 2.5.3** for the static regain design method. In both cases, each path's total pressure requirement is calculated and the path with the highest total pressure requirement is the system's design leg or critical path. This is shown in **Table 2.7** of **Section 2.5** and repeated below in **Table 3.1**.

TABLE 3.1
Path Total Pressure for Sample System 1 Static Regain Design

Terminal	**Path (Sections)**	**Section Total Pressure Losses** (*inches wg*)	**Path Total Pressure Required** (*inches wg*)
A	1,2	0.34 + 0.35 + 0.08	0.77
B	1,3	0.34 + 0.35 + 0.08	0.77
C	1,4,5	0.34 + 0.09 + 0.38 + 0.05	0.86
D	1,4,6,7	0.34 + 0.09 + 0.06 + 0.19 + 0.05	0.73
E	1,4,6,8,9	0.34 + 0.09 + 0.06 + 0.06 + 0.21 + 0.05	0.81
F	1,4,6,8,10,11	0.34 + 0.09 + 0.06 + 0.06 + 0.05 + 0.09 + 0.05	0.74
G	1,4,6,8,10,12	0.34 + 0.09 + 0.06 + 0.06 + 0.05 + 0.08 + 0.07	0.75

By analyzing a system using total pressure, the designer can more easily see areas of inefficiency. High pressure losses in sections or paths can be lowered by increasing sizes or using more efficient fittings. For sections or paths where there is too much pressure (unbalanced), the designer can reduce duct sizes, use less efficient fittings, or put in balancing devices such as dampers or orifices.

3.2 Balancing Equal Friction Designs

The amount of balancing any path requires is simply the path's excess pressure. To balance a system, three methods are often employed for equal friction in non-design legs: balancing dampers, orifice plates, and enhanced equal friction design.

3.2.1 Balancing Dampers

Balancing dampers are the most common method of balancing equal friction designs because calculations to determine the damper setting are not usually required when the system is being designed. The damper setting is usually determined by measuring the air volume flow rate (*cfm*) in the field after installation. The damper is adjusted until the required air volume for the terminal device is correct. The required angle of the balancing damper can be determined, however, if the excess pressure is known and the relationship between the damper angle and the loss coefficient for the damper is known.

For example, **Table 3-2** from ASHRAE Fitting **CD9-1** lists loss coefficients for round butterfly dampers of type shown in **Figure 3.1**.

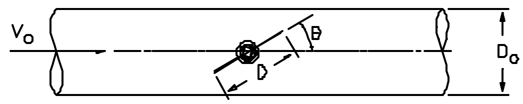


Figure 3.1
Round Butterfly Damper

Table 3.2
Round Butterfly Damper Loss Coefficients

	Loss Coefficient *C*											
	Damper Angle *q*											
***D*/*D*$_o$**	**0E**	**10E**	**20E**	**30E**	**40E**	**50E**	**60E**	**70E**	**75E**	**80E**	**85E**	**90E**
0.5	0.19	0.27	0.37	0.49	0.61	0.74	0.86	0.96	0.99	1.02	1.04	1.04
0.6	0.19	0.32	0.48	0.69	0.94	1.21	1.48	1.72	1.82	1.89	1.93	2.00
0.7	0.19	0.37	0.64	1.01	1.51	2.12	2.81	3.46	3.73	3.94	4.08	6.00
0.8	0.19	0.45	0.87	1.55	2.60	4.13	6.14	8.38	9.40	10.30	10.80	15.00
0.9	0.19	0.54	1.22	2.51	4.97	9.57	17.80	---	---	---	---	---
1.0	0.19	0.67	1.76	4.38	11.20	---	---	---	---	---	---	---

The required loss coefficient is determined from **Equation 1.16a**:

$$C = \frac{\Delta TP}{VP}$$

where *DTP* is the additional pressure required to balance the path (i.e., the path's excess pressure). For the equal friction sample problem in **Section 2.4**, assuming the dampers will be put in the terminal sections, the required loss coefficient and corresponding damper angle required to balance the paths for a D/D_o = 1.0 are shown in **Table 3.3**.

Table 3.3
Sample System No. 1 - Damper Angles

Terminal	**Path Excess Pressure** (*inches wg*)	**Velocity Pressure in Terminal Section** (*inches wg*)	**Required Loss Coefficient** ***C***	**Required** Damper Angle** ***q***
A	0.67	0.11	6.09	33E
B	0.67	0.11	6.09	33E
C	0.18	0.14	1.29	16E
D	0.26	0.16	1.63	19E
E	0.12	0.11	1.09	14E
F	0.09	0.17	0.53	07E
G*	0.00	N/A	N/A	N/A

*   Design leg, no dampers required
** Values interpolated to nearest degree

3.2.2 Orifice Plates

The size of an orifice plate required to balance the paths can also be determined if the excess pressure, velocity pressure, and corresponding loss coefficient are known. By assuming the number of holes equals one, ASHRAE Fitting **CD6-2** gives the data shown in **Table 3.4** to determine the loss coefficient of orifice plates.

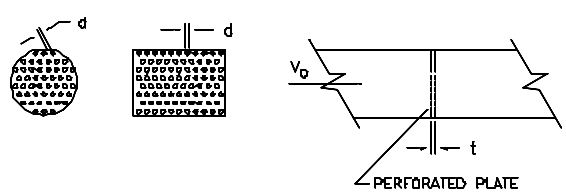


Figure 3.2
Orifice Plates

where (for *t/d* 0.015):

A_o = area of duct (ft^2)

A_{or} = orifice area = $p\,d^2/4$ (ft^2)

D = diameter of perforated hole (*inches*)

n = free area ratio of plate (dimensionless) = SA_{or}/A_o

t = plate thickness (*inches*)

Table 3.4
Orifice Plate Loss Coefficients

Loss Coefficient *C*										
$\frac{t}{d}$	*n*									
	0.20	**0.25**	**0.30**	**0.40**	**0.50**	**0.60**	**0.70**	**0.80**	**0.90**	**1.00**
0.015	51.50	30.00	18.20	8.25	4.00	2.00	0.97	0.42	0.13	0.00
0.200	48.00	28.00	17.40	7.70	3.75	1.87	0.91	0.40	0.13	0.01
0.400	46.00	26.50	16.60	7.40	3.60	1.80	0.88	0.39	0.13	0.01
0.600	42.00	24.00	15.00	6.60	3.20	1.60	0.80	0.36	0.13	0.01
0.800	34.00	19.60	12.20	5.50	2.70	1.34	0.66	0.31	0.12	0.02
1.000	31.00	17.80	11.10	5.00	2.40	1.20	0.61	0.29	0.11	0.02
1.400	28.40	16.40	10.30	4.60	2.25	1.15	0.58	0.28	0.11	0.03
2.000	27.40	15.80	9.90	4.40	2.20	1.13	0.58	0.28	0.12	0.04
4.000	27.70	16.20	10.00	4.60	2.25	1.20	0.64	0.35	0.16	0.08
6.000	28.50	16.60	10.50	4.80	2.42	1.32	0.70	0.40	0.21	0.12
8.000	30.00	17.20	11.10	5.10	2.58	1.45	0.80	0.45	0.25	0.16
10.000	31.00	18.20	11.50	5.40	2.80	1.57	0.89	0.53	0.32	0.20

For the equal friction sample, again assume the orifices will be put in terminal sections. The path excess pressures, terminal velocity pressures, and required loss coefficients are the same as those in **Section 3.2.1** for balancing with dampers. For orifice plate balancing, there is generally only one hole of diameter ***d***. The terminal diameters are in the range of 15 *inches* to 25 *inches*. An orifice thickness (t = **3** *inch*) is assumed. The required loss coefficient and corresponding orifice diameter to balance each terminal section are given in **Table 3.5**.

Table 3.5
Sample System No. 1 - Required Orifices

Terminal	Path Excess Pressure (*inches wg*)	Terminal Section Diameter (*inches wg*)	Required Loss Coefficient *C*	Required Open Area Assuming ***t/d*** = 0.015	Required Orifice Diameter ***d*** (*inches*)	Actual ***t/d***
A	0.67	15	6.09	0.45	8.9	0.028
B	0.67	15	6.09	0.45	8.9	0.028
C	0.18	17	1.29	0.67	13.9	0.018
D	0.26	19	1.63	0.64	15.2	0.016
E	0.12	15	1.09	0.69	12.5	0.020
F	0.09	20	0.53	0.78	14.6	0.017
G*	0.00	25	N/A	N/A	N/A	N/A

* design leg, no orifice plate required

3.2.3 Enhanced Equal Friction Design

Enhanced equal friction design incorporates equal friction duct sizing enhanced by total pressure duct balancing. The first step in using this method of balancing is to increase the friction loss per 100 *feet* in the non-design legs to a point where the path's pressure loss is equal to that of the design leg. This can be done systematically by decreasing duct diameters and determining the increased friction loss. The following shows the increase in friction loss and resulting decrease in sizes for non-design leg paths, required to balance the section. Note that decreasing branch sizes also affects the losses of the branch fitting and any in-line fittings (elbows). The following example in **Table 3.6** illustrates how decreasing duct size affects the overall pressure drop in terminals A and B of the equal friction sample problem in **Section 2.4.**

Given: V_c = 2,616 *fpm*, VP_c = 0.43 *inches wg* in Section 1.

Determine: Pressure increase in Sections 2 (Terminal A) and 3 (Terminal B) from a decrease in duct size.

Since the excess pressure in these sections is substantial (0.67 *inches wg*), begin downsizing with a diameter that will yield a velocity equal to upstream Section 1 or 2500 *fpm* maximum in order not to create excessive airflow noise in the branch feeding the outlet.

Table 3.6
Balancing by Duct Downsizing
Terminal A or B

Section Diameter (*inches*)	Velocity (*fpm*)	Duct Loss (*inches wg*)	Conical Tee Loss Coeff. (C_b)	FITTING ΔTP (*inches wg*)	SECTION ΔTP (*inches wg*)	Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)
15	1,304	0.03	1.38	0.15	0.18	0.63	0.67
11	2,424	0.15	1.36	0.27	0.42	1.13	0.17
12	2,037	0.10	0.49	0.28	0.38	0.98	0.32

It should be noted that downsizing is an art and not a science. Proficiency in hand calculating is gained with experience and various combinations. The reduction in diameter from 15 to 11 inches results in an excess pressure of only 0.17 *inches wg* compared to 0.51 *inches wg* pressure. This is much better, but still not balanced against the design leg. Further resistance to flow from the fan to either Terminal A or B is required in order to be the same as from the fan to the end of the design leg which is Terminal G. Keep in mind that downsizing can sometimes cause the cumulative total pressure to exceed the required excess pressure. Exceeding the required excess pressure in a non-design leg should be avoided since it will change the design leg.

Some systems may have a branch connection to a terminal device (i.e., VAV box or diffuser), consisting of a shorter duct length that is generally presized. The presize is to ensure proper operation of the device. Assume the equipment manufacturer's restrictions require a 12-*inch* diameter inlet for proper operation. **Table 3.6** then shows the terminal device must use an excess pressure of 0.32 inches wg in order to balance this branch.

A second step to further balance an equal friction design is to incorporate less efficient fittings to use the excess pressure. Several other types of fittings are evaluated in place of the conical tee in **Table 3.7** in order to see how they balance the branch.

Table 3.7
Balancing by Fitting Substitution
Terminal A or B

Branch Fitting Type	Section Diameter (*inches*)	Velocity (*fpm*)	Duct Loss (*inches wg*)	Loss Coeff (C_b)	FITTING ΔTP (*inches wg*)	SECTION ΔTP (*inches wg*)	Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)
Conical	12	2,037	0.10	1.07	0.28	0.38	0.98	0.32
Straight	12	2,037	0.10	1.81	0.47	0.57	1.17	0.13
Shop-Fabricated Manifold	12	2,037	0.10	1.95	0.50	0.60	1.20	0.10
Field-Fabricated Manifold	12	2,037	0.10	2.42	0.63	0.73	1.33	N/A

Increased values of loss coefficient (C) result in less efficient fittings. Notice the improved balancing with the use of a less efficient fitting in place of a conical cross. However if the fitting is too inefficient, it can create a new design leg at a higher total pressure requirement as is the case with the Field Fabricated Manifold Cross. Less efficient fittings are generally less expensive but this should always be verified.

Efficient fittings are required only in the design leg of the system. These design leg fittings rarely account for more than 10 percent of the branch fittings of a system. Therefore, using less efficient fittings in non-design legs could result in substantial savings as well as help balance the system. Additionally, using smaller duct will reduce material and installation costs. The best sequence to reduce cost and balance the fittings is to first reduce the branch sizes to increase the friction rate and use as much excess pressure as possible, then use less efficient fittings (junctions and elbows) to further balance the system. If the path is still more than 10% unbalanced with the design leg, use dampers to finalize the balancing in the field. Orifice plates should be avoided since they do not offer the flexibility of field adjustment.

In **Table 3.8**, the other non-design branches are evaluated using the enhanced equal friction design for the sample problem. Again duct downsizing was limited to approximately 2,500 *fpm* for terminal outlet airflow and acoustical performance. Duct system acoustics will be discussed in **Chapter 8**.

Table 3.8
Sample System 1 - Enhanced Equal Friction Design Data Sheet

SECT	ITEM REF	ITEM	Q (cfm)	DIA. ($inches$)	AREA (ft^2)	V (fpm)	VP ($inches\ wg$)	C	$\Delta P/100\ ft$ ($inches\ wg$)	L (ft)	ΔP DUCT ($inches\ wg$)	FITTING ΔSP ($inches\ wg$)	SECTION ΔSP ($inches\ wg$)	FITTING ΔTP ($inches\ wg$)	SECTION ΔTP ($inches\ wg$)
1	CD3-9	Duct 90 Elbow	20,600	38	7.88	2,616	0.43	0.12	0.20	145	0.29	0.05	0.34	0.05	0.34
4	SD5-25	Duct Straight T/O	17,400	36	7.07	2,462	0.38	0.14	0.19	36	0.07	0.00	0.07	0.05	0.12
6	SD5-10	Duct Straight T/O	15,000	34	6.31	2,379	0.35	0.13	0.19	30	0.06	0.02	0.08	0.05	0.11
8	SD5-10	Duct Straight T/O	11,800	31	5.24	2,251	0.32	0.13	0.19	40	0.08	0.01	0.08	0.04	0.12
10	SD5-10	Duct Straight T/O	10,200	29	4.59	2,224	0.31	0.13	0.20	35	0.07	0.03	0.10	0.04	0.11
2 and 3	SD5-25	Duct Con Cross	1,600	15	1.23	1,304	0.26	2.90	0.17	20	0.03	(0.01)	0.02	0.31	0.34
2 and 3	SD5-24	Duct Str Cross	1,600	12	0.79	2,037	0.11	1.81	0.26	20	0.13	0.30	0.43	0.47	0.60
5	SD5-10 CD3-9	Duct Con Tee 90 Elbow	2,400	17	1.58	1,523	0.14	2.80 0.15	0.19	50	0.10	0.15 0.02	0.27	0.4 0.02	0.52
5	SD5-10 CD3-9	Duct Con Tee 90 Elbow	2,400	16	1.40	1,719	0.18	2.05 0.16	0.26	50	0.13	0.21 0.03	0.37	0.38 0.03	0.54
7	SD5-10	Duct Con Tee	3,200	19	1.97	1,625	0.16	1.60	0.19	27	0.05	0.07	0.12	0.26	0.31
7	SD5-10	Duct Con Tee	3,200	13	1.40	2,292	0.33	0.79	0.43	27	0.12	0.27	0.39	0.26	0.38
9	SD5-10	Duct Con Tee	1,600	15	1.23	1,304	0.11	3.22	0.17	24	0.04	0.13	0.14	0.34	0.38
9	SD5-9	Duct Str Tee	1,600	15	1.23	1,304	0.11	4.29	0.17	24	0.04	0.25	0.29	0.45	0.50
11	SD5-19	Duct Bullhead T/V	3,600	20	2.18	1,650	0.17	1.03	0.18	33	0.06	0.04	0.10	0.18	0.24
12	SD5-19 CD3-14	Duct Bullhead T/V 45 Elbow	6,600	25	3.41	1,936	0.23	0.67 0.08	0.19	48	0.09	0.08 0.02	0.19	0.16 0.02	0.27

The rows that are shaded in **Table 3.8** are sections where smaller sizes and/or less efficient fittings were used in non-design legs to help balance the system. **Table 3.9** shows the before and after excess pressures to each of the terminals.

Table 3.9
Path Total Pressures for Sample System 1 Equal Friction Design

Terminal	Path (Sections)	Before Balancing		After Balancing	
		Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)	Path Total Pressure Required (*inches wg*)	Excess Pressure (*inches wg*)
A	1,2	0.79	0.51	1.17	0.13
B	1,3	0.79	0.51	1.17	0.13
C	1,4,5	1.12	0.18	1.18	0.12
D	1,4,6,7	1.04	0.26	1.26	0.04
E	1,4,6,8,9	1.18	0.12	1.28	0.02
F	1,4,6,8,10,11	1.21	0.09	1.21	0.09
G	1,4,6,8,10,12	1.30	0.00	1.30	0.00

Downsizing the duct and substituting a less efficient straight cross for the conical cross for the Terminal A and B sections 2 and 3, decreased the excess pressure from 0.51 *inches wg* to 0.13 *inches wg*. A rule of thumb is that if the excess pressure is less than 10 percent of the pressure required to operate the design, the path is considered well balanced. A designer may still want to consider dampers in these sections for fine tune adjustments since the excess pressure is just at 10 percent of the design total pressure. The other paths are fairly well balanced.

The enhanced equal friction design of **Sample System 1** proved to be quite beneficial. Overall, three duct sizes were reduced and less efficient (and less expensive) fittings were used in two places. Thus the redesigned system is balanced and will have a lower first cost of material without increasing the operating costs.

Downsizing alone improves the balancing the most by increasing the friction rate. Using less efficient fittings further improves balancing. However, in larger systems, often the substitution of less efficient fittings is more prominent in improving both balancing and reducing material cost. Again, the use of smaller ducts by virtue of the balancing process, results in lower first cost. In the case of using less efficient branch fittings, first cost is reduced further. Excess pressure reduction or better balancing means the system will deliver the designed airflow to the individual spaces or zones. The ductwork, by design, is less dependent on dampers and orifice plates. These devices only add to first cost and better balancing can eliminate them. Therefore, the enhanced equal friction design method can improve the systems balancing and reduce first cost.

3.3 Enhanced Static Regain Design

Enhanced static regain design is similar to the enhanced equal friction design just discussed in that it incorporates total pressure to balance the system. The difference, as the name implies, is that the duct downsizing and fitting changeouts are applied to static regain design instead of equal friction design. The comparative evaluation between static regain and equal friction in **Sample System 1** shows that if the friction loss in Section 1 of both designs are the same, the static regain design will have (1) lower overall design total pressure requirements, (2) larger duct sizes and, (3) significantly better balancing then the equal friction design. Enhanced static regain design is a more efficient design method than enhanced equal friction, especially when used with computer-aided duct design. Fewer duct downsizing and fitting selection considerations are required to balance the system because the static regain design is nearly balanced to begin with.

Sample System 2 in **Figure 3.3** uses the following design parameters to compare the difference between conventional equal friction and static regain designs with the enhanced static regain design. Although not shown, the enhanced equal friction design can exceed the performance/cost attributes of the static regain design.

Design parameters for **Sample System 2**:

1. No height restrictions for main trunk.
2. Height restriction of 12 *inches* for all other branches.
3. Required SP at outlets of 0.5 *inches wg*.
4. For equal friction and static regain design, all 90E tees are conical, all crosses are conical, and all 45E laterals are conical.
5. For enhanced static regain design, default fittings are substituted in non-design legs.

When height restrictions were not met by the design, flat oval duct was used. A friction loss factor of 0.10 *inches wg per100 ft* was used for the equal friction design, and the system static pressure for the static regain and enhanced static regain designs method was matched to this as closely as possible for an apples-to-apples evaluation. **Table 3.10** Comparison of System Operating Pressures and Sizes and **Table 3.11** Comparison of System Balancing show the results of this analysis. For equal overall pressure drop between designs, the enhanced static regain design yields smaller duct and fittings. Of the 59 sections, nearly 41 percent were reduced by the enhancing process, and the resultant average excess pressure of 0.06 *inches wg* was nearly 54 percent less than the equal friction method and 40 percent less than the static regain method. The enhanced static regain method also uses less expensive and less efficient fittings appropriate for the available pressure and this helps to further balance the system and reduce costs.

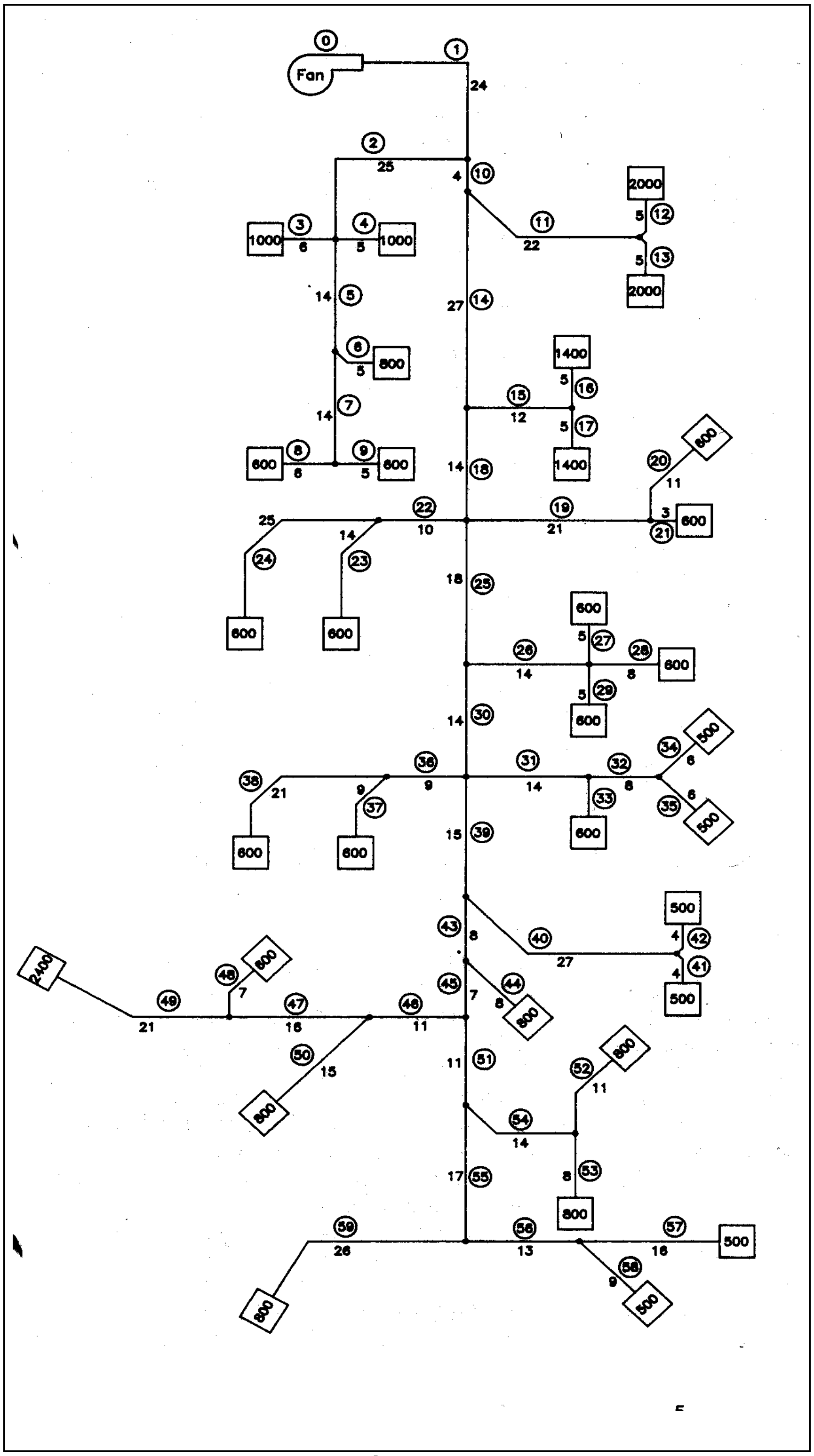


Figure 3.3
Sample System 2

Table 3.10
Sample System 2
Comparison of System Operating Pressures and Sizes

DESIGN METHOD	EQUAL FRICTION	STATIC REGAIN	ENHANCED STATIC REGAIN	
System Total Pressure (*inches wg*)	1.06	1.04	1.04	
System Static Pressure (*inches wg*)	0.78	0.76	0.76	**SECTION NUMBER**
Section Size: Round & Flat Oval (inches)	48	48	48	1
	12 x 45	12 x 45	12 x 45	2
	12 x 15	12	11	3
	12 x 15	12	11	4
	12 x 25	12 x 21	12 x 20	5
	12 x 14	11	10	6
	12 x 18	12 x 15	12 x 14	7
	12	9.5	10	8
	12	9.5	10	9
	46	46	46	10
	12 x 45	12 x 42	12 x 31	11
	12 x 25	12 x 21	12 x 18	12
	12 x 25	12 x 21	12 x 18	13
	43	45	45	14
	12 x 34	12 x 28	12 x 21	15
	12 x 20	12 x 15	12 x 15	16
	12 x 20	12 x 15	12 x 15	17
	40	44	44	18
	12 x 18	12 x 14	12	19
	12	9.5	9.5	20
	12	9.5	8.5	21
	12 x 18	12 x 14	11.5	22
	12	9.5	9	23
	12	9.5	9.5	24
	37	43	43	25
	12 x 25	12 x 20	12 x 42	26
	12	9.5	8	27
	12	9.5	8.5	28
	12	9.5	8	29
	36	42	42	30
	12 x 21	12 x 18	12 x 14	31
	12 x 15	12	11	32
	12	9.5	9.5	33
	11	8.5	8.5	34
	11	8.5	8.5	35
	12 x 18	12 x 14	11.5	36
	12	10.5	9	37
	12	9.5	9.5	38
	32	36	39	39
	12 x 15	12	11	40
	11	8.5	8.5	41
	11	8.5	8.5	42
	31	34	36	43
	12 x 14	11	9.5	44
	30	33	34	45
	12 x 42	12 x 37	12 x 34	46
	12 x 34	12 x 31	12 x 31	47
	12	9.5	10	48
	12 x 31	12 x 25	12 x 28	49
	12 x 14	11	10.5	50
	22	22	24	51
	12 x 14	11	11.5	52
	12	11	11.5	53
	12 x 21	12 x 18	12 x 18	54
	18	16	17	55
	12 x 15	12	12 x 14	56
	11	8.5	9	57
	11	8.5	9	58
	12 x 14	11	11.5	59
Number of Round Duct Sections	53	69	73	

* Friction loss factor for equal friction design was 0.10 *inches wg per 100 feet*.

Table 3.11
Sample System 2
Comparison of System Balancing

DESIGN METHOD	EQUAL FRICTION	STATIC REGAIN	ENHANCED STATIC REGAIN	TERMINAL SECTION
System Total Pressure (*inches wg*)	1.06	1.04	1.04	
System Static Pressure (*inches wg*)	0.78	0.76	0.76	
Path Excess Pressure (*inches wg*)	0.16	0.09	0.00	3
	0.16	0.09	0.00	4
	0.16	0.09	0.06	6
	0.16	0.01	0.10	7
	0.16	0.02	0.10	9
	0.24	0.19	0.11	12
	0.24	0.19	0.11	13
	0.18	0.12	0.11	16
	0.18	0.12	0.11	17
	0.15	0.07	0.02	20
	0.19	0.17	0.07	21
	0.18	0.13	0.03	23
	0.17	0.11	0.08	24
	0.12	0.12	0.00	27
	0.15	0.17	0.07	28
	0.12	0.12	0.00	29
	0.10	0.12	0.04	33
	0.12	0.12	0.09	34
	0.12	0.12	0.09	35
	0.13	0.14	0.06	37
	0.12	0.11	0.10	38
	0.12	0.13	0.05	41
	0.12	0.13	0.05	42
	0.13	0.20	0.08	44
	0.00	0.01	0.01	48
	0.00	0.03	0.01	49
	0.04	0.09	0.00	50
	0.06	0.05	0.09	52
	0.04	0.05	0.10	53
	0.09	0.00	0.08	57
	0.08	0.02	0.08	58
	0.07	0.03	0.09	59
Average Excess	0.13	0.10	0.06	
High Excess	0.24	0.20	0.11	
Average/System SP x 100	17%	13%	8%	
High/System SP x 100	31%	26%	14%	

Note: 0.00 (zero) excess pressure denotes a design leg path.

3.4 Fan Selection

Selection of the fan requires that all duct system resistance be evaluated. In general this resistance consists of the following:

1. Supply and return air duct and fitting losses.

2. In-line equipment losses (coils, filters, silencers, control dampers, terminal outlets, and VAV boxes.)

3. System inefficiencies (system effect) associated with improper fan inlet/outlet connections to the ductwork and inefficient ductwork layouts.

4. Nonstandard environmental conditions (temperature, humidity, and elevation).

System curves, which will be discussed in detail later in this chapter, determine overall flow resistance in the system for a given volume flow rate through the duct. Determination of supply air ductwork pressure losses is addressed fully in **Chapters 2 and 3**. Exhaust and/or return ductwork losses are addressed in detail in **Chapters 4 and 5.**

In-line equipment losses are easily obtained from the equipment manufacturers. Often data for air handlers incorporating a given fan type show an external static pressure (***ESP***) available for ductwork and components external to the unit. Internal losses can include coils, filters, and sometimes silencers. Air handler equipment schedules often list the total static pressure (***TSP***) value, which includes internal components and their associated losses in addition to the ***ESP*** available to overcome ductwork and duct component losses.

A fixed volume flow rate (*cfm*) through a fixed system layout and sizes results in a fixed total (***TP***) and static pressure loss (***SP***). Varying the volume flow rate will result in a change in the pressure loss as shown in **Equation 3.1** and **Figure 3.4.**

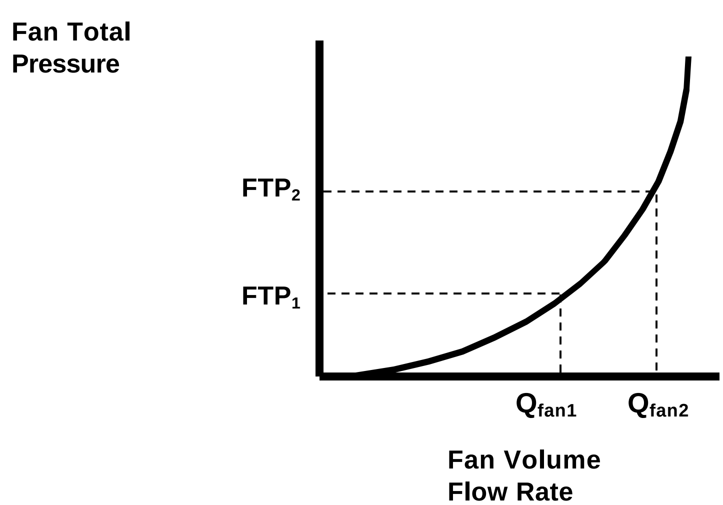


Figure 3.4
System Curve FTP versus Q

$$\frac{FTP_2}{FTP_1} = \left(\frac{Q_{fan2}}{Q_{fan1}}\right)^2$$ **Equation 3.1**

where:

FTP_1* and *FTP_2	=	Fan total pressure requirements at Q_{fan1} and Q_{fan2} respectively (*inches wg)*
Q_{fan1}* and *Q_{fan2}	=	Volume flow rate requirements for systems 1 and 2 respectively (*cfm*)

Sample Problem 3-1

For a given duct system, the fan total pressure is 2.0 inches wg and the volume flow rate is 15,000 cfm. Determine the resultant total pressure and volume flow rate if the system volume flow is increased by 10 percent.

Answer: Rearranging **Equation 3.1** for ***FTP_2*** and substituting the given values, the second condition can be determined. The volume flow rate for an increase of 10 percent is 16,500 *cfm* (1.1 x 15,000). Then

$$FTP_2 = FTP_1\left(\frac{Q_{fan2}}{Q_{fan1}}\right)^2 = 2.0\left(\frac{16{,}500}{15{,}000}\right)^2 = 2.42 \text{ inches wg}$$

Equation 3.1 shows that a 10 percent increase in volume flow rate will yield a 21 percent increase in static pressure for a given duct system. The duct system did not change in the above example problem. Trying to force more air through a given system with fixed duct sizes increases the total pressure requirement significantly. Having two sets of points will help define the system curve for the duct system. For more information on fan performance, see references in **Appendix A.9.2 and A.9.6**

The system curve is based on pressure losses for the supply, return and in-line components for a given volume flow rate. One other factor needs to be known in order to select a fan and that is the system effect factor (SEF).

3.4.1 System Effect Performance Deficiencies

The system effect factor (SEF) was developed to account for deficiencies in fan and system performance associated with improper flow conditions at the inlet and/or outlet of the fan. Fan equipment is normally rated with open inlets and a section of straight duct attached to the outlet. However, real-life installations often include improper outlet conditions, non-uniform inlet flow or swirl at the fan inlet. These conditions alter the aerodynamic characteristics of the fan and the full airflow potential is not realized. **Figure 3.5** gives a graphic presentation of how system effect causes deficient fan performance. Fan testing would be too expensive to model all the possible field conditions. Therefore, the duct designer needs to adjust the pressure calculation to account for these effects.

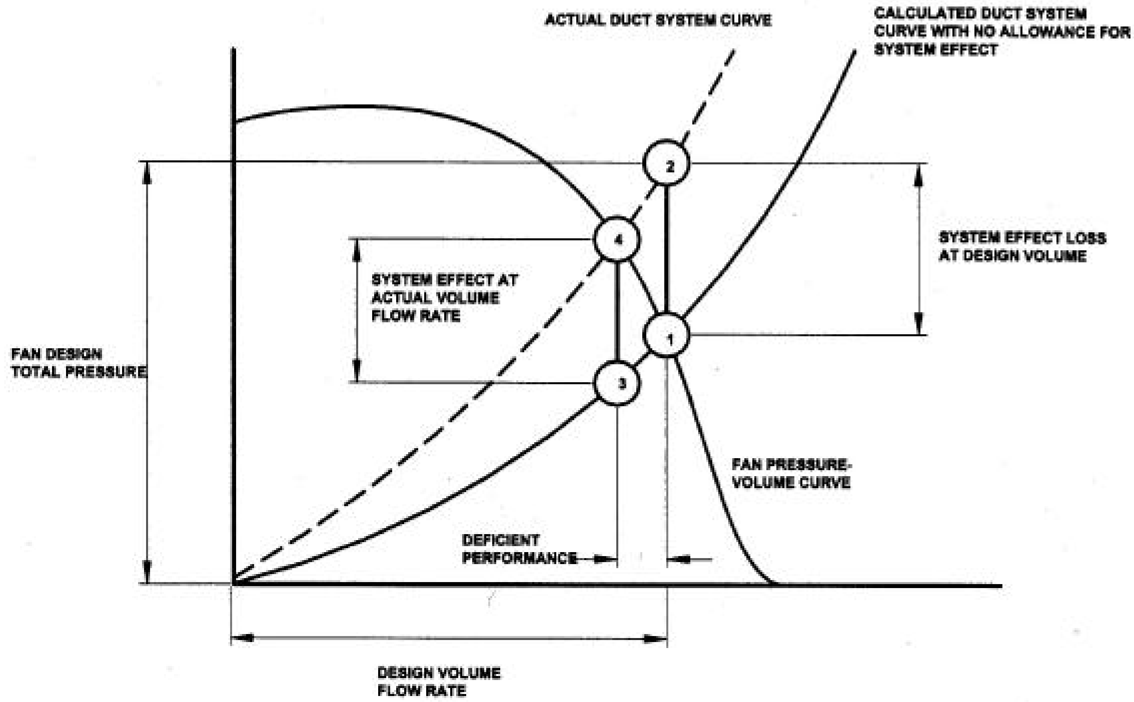


Figure 3.5
Duct System Effect

Point 1 in **Figure 3.5** represents the operating point on a system curve assuming no errors in calculating system resistance in ductwork and components. Proper fan selections require finding a fan performance curve, which passes through point 1. System effect causes added system resistance so a fan operating at a constant speed (rpm) with a higher system resistance will result in a volume flow rate deficiency shown by point 4. Achieving the design volume flow rate would require either a larger fan or increased brake horsepower (bhp) for the original fan. The actual operating point is point 2 on the actual system curve.

Further information on the description and calculation of system effect and system effect factors can be found in **Air Movement and Control Association (AMCA)** publication 201, **Fans and Systems and the ASHRAE 2000 HVAC Systems and Equipment Handbook** Chapter 18, Fans. Additional information for calculating fan inlet/outlet losses can be found **in ASHRAE 2001 Fundamentals Handbook**, Chapter 34, Duct Design (see Appendix A.9.2).

3.4.2 Duct Performance Deficiencies

As with fans, performance data for duct and fittings have much to do with flow conditions through these components based on their position in the system. Loss coefficient data for fittings are generally based on ideal flow conditions. Real-life layouts can often incorporate close-coupled fittings untested standard fittings, and customized fittings. Unlike fans, deficiencies in fittings' resistance are next to impossible to determine because of the variety of applications and factors influencing performance. Some of the more common arrangements have been tested.

These problems and problems associated with duct leakage all contribute to lessening the accuracy in determining a design system curve. Over the years, design engineers have established their own rules of thumb in developing safety factors to account for deficiencies. Many computer-aided duct designs which incorporate most of the available data have proven to be surprisingly accurate despite these deficiencies.

3.4.3 Fan Pressures

Fan selection is generally based on the fan total pressure or fan static pressure. These terms are exclusive to the fan industry and should not be used indiscriminately or confused with similar terms relating to duct system performance.

Fan Total Pressure (FTP) $= TP_{outlet} - TP_{inlet}$ **Equation 3.2**

$= (SP_{outlet} + VP_{outlet}) - (SP_{inlet} + VP_{inlet})$

Fan Static Pressure (FSP) $= FTP - VP_{outlet}$ **Equation 3.3**

$= SP_{outlet} - SP_{inlet} - VP_{inlet}$

The terms external static pressure (***ESP***) and total static pressure (***TSP***) often appear on air handling unit schedules along with various other performance factors associated with individual components that makeup the unit. The ***ESP*** includes all static pressure losses external to the equipment and, as a minimum, includes the static pressure losses associated with the supply and return air ductwork and any system effect. The **external static pressure loss** can also incorporate losses associated with filters, coils, and silencers for built-up air handling units. However, a majority of air handling units are packaged and include their own filters, coils, and silencers, leaving **external static pressures** for the interconnecting ductwork and any ductwork components (for example, VAV boxes, measuring stations, fire/smoke dampers, additional silencers, etc.). There is not a standard definition of external static pressure so make sure you check with the manufacture as to how much pressure is available for the ductwork system. Use manufacturers' instructions for selection of packaged air handling units. The type and size of fan to be selected depends on many factors and is discussed in detail in ASHRAE publications (**Appendix A.9.2**).

This discussion of fan selection is intended to ensure that all pressure losses are accounted for in determining the most accurate design operating parameters before selecting a fan performance curve. In **Figure 3.5**, point 2 on the system curve is the actual operating point of the system and the point through which the fan performance curve should intersect. If the original fan performance curve was used, the volume flow rate would be deficient by the difference between point 4 and point 2. Point 2, however, is on the actual system curve and needs to intersect a new fan performance curve (not shown on **Figure 3.5**) to supply an adequate volume of air. If we know where the actual system curve intersects the original fan performance curve we can determine the point on the new fan performance curve required using additional **fan laws**. *Remember though that the fan laws only apply to one system curve. You <u>can not</u> use the fan laws to determine the performance from one system curve to another.* That is, referring to **Figure 3.5**, knowing the Fan Total Pressure require for a given Volume Flow Rate such for point 4, you can determine point 2 operating parameters using the fan laws. However, you could not use the fan laws to determine the operating parameters of point from either points 1 or 3 because they are on a different system curve.

Equation 3.1 is a part of the **fan laws,** which govern the performance of fans and predict the fan performance at points of operation other than what was tested. The fan laws determine points of operation when changes are made in speed (rpm), volume flow rate (cfm), and brake horsepower (bhp), etc. The equations here assume that the fan wheel diameter and density ratios are unity. Here are the basic equations:

FAN LAWS

$$\frac{Q_2}{Q_1} = \frac{rpm_2}{rpm_1}$$ **Equation 3.4**

where:

Q	=	Volume flow rate of airflow (*cfm*)
rpm	=	Fan speed in revolutions per minute

$$\frac{bhp_2}{bhp_1} = \left(\frac{Q_2}{Q_1}\right)^3$$ **Equation 3.5**

where:

bhp	=	brake horsepower

Sample Problem 3-2

For Sample Problem 3-1, it is know that the design volume represented by point 1 was to be 15,000 cfm at a total pressure of 2 inches wg. The same fan is to be used as represented by the fan catalog performance curve. The measured total pressure is 2.25 inches wg at 14,660 cfm. What total pressure must the fan reach to maintain the 15,000 cfm design volume flow rate?

Answer: The measured values represent point 4 in **Figure 3.5**. Use **Equation 3.1,** rearranged to solve for FTP at point 2 in a similar manner to what was done in **Sample Problem 3-1**:

$$FTP_2 = 2.25 \left(\frac{15{,}000}{14{,}660}\right)^2 = 2.36 \ \textit{inches wg}$$

The fan total pressure must be increased 18 percent, from 2.00 to 2.36 inches wg, to attain the design volume flow rate that was deficient due to the system effect. This also represents an increase in power requirements of 18 percent.

3.5 Cost Optimization

Cost optimization in duct design involves minimizing the owning cost or the net present value of the sum of the initial cost of ductwork and equipment, and the operating cost that could go on for many years.

Here are factors to consider when calculating the net present value:

* Initial cost of material and equipment *Costs of maintenance

* Cost of installation
* Cost of field balancing
* Cost of energy
* Hours of operation
* Tax rates
* Insurance
* Cost of money
* Life of system

For simplification, the following assumptions are made in order to focus and compare the analysis:

1. The fan/motor drive combined efficiency is 75 percent.
2. The fan operates *50 weeks per year*, 7 *days per week*, 16 *hours per day* or 5,600 *hours* total *per year*.
3. Energy cost is $0.08*/kwh*.
4. The project life is 20 *years*.
5. The initial costs are of the duct material only.
6. The inflation rate is 3 percent per year.
7. Fan volume flow rate is constant.

In the following example, consideration is only given to the operating costs and initial cost of the ductwork. The systems annual operating cost can be expressed by **Equation 3.6**.

$$Cost/year = \left(\frac{Q_{fan} \ x \ FTP}{8{,}520 \ x \ eff} \right) x \ \frac{Hours}{Year} \ x \ \frac{\$}{kwh} \qquad \textbf{Equation 3.6}$$

where:

$Cost/year$ = System first year operating cost (*$*)

Q_{fan} = System volume flow rate (*cfm*)

FTP = System total operating pressure (*inches wg*)

$\frac{Hours}{Year}$ = Number of hours the system is in operation in one year

$\frac{\$}{kwh}$ = Cost of energy

eff = fan/motor drive combined efficiency

$8{,}520$ = a conversion factor to kwh (*kilowatt-hours*)

The object of the designer is to reduce or minimize the cost of owning the system. Looking at Equation **3.6**, there are several factors that can reduce the operating cost. The system volume flow rate (*cfm*) should be minimized by performing a thorough load analysis (refer to **Appendix A.9.2**). Variable air volume (VAV) systems can take into account variations in load requirements over periods of time to minimize airflow. Using efficient duct designs with low system effects by using proper connections at the fan can reduce the Fan Total Pressure requirements which has has a direct affect on operating cost. In the previous examples, it was shown that different design methods produce considerable differences in total pressure losses. These differences can result in significant operating cost savings. Selection of high efficiency fan/motor drive components for the system will also help to minimize operating cost.

Items in the equation that normally are out of the control of the designer are operating hours and the cost of energy. Normally, the owner dictates the number of hours the system will be operating annually. Cost of energy, unless it is being generated on site, is out of the control of the designer. Although there are certain times of the day when the cost of energy is lower, the owner may need to operate the duct system during that time. On the first cost side of the analysis, duct material costs are to be included. As was shown earlier, enhancing a duct system by balancing the airflow with smaller sizes and less efficient fittings can reduce first cost. Smaller duct and less efficient fittings are less expensive to purchase and to install. Ease of installation is not considered for this analysis, but it will be discussed later.

In order to calculate the owning cost or *net present value cost* (**NPVC**), the following equations and components are used.

Net present value cost is the sum of the first cost and the product of the present worth factor and the annual operating cost. The present worth factor is calculated from **Equation 3.7** as:

$$PWF = \left(\frac{(1 + IADR)^n - 1}{IADR\,(1 + IADR)^n} \right) \qquad \textbf{Equation 3.7}$$

where:

PWF	=	present worth factor
n	=	project life (*years*)
$IADR$	=	inflation adjusted discount rate (*percent*)

The inflation-adjusted discount rate is defined as:

$$IADR = \left(\frac{1 + ndr}{1 + ir} \right) - 1 \qquad \textbf{Equation 3.8}$$

where:

ndr	=	nominal discount rate (*percent*)
ir	=	Inflation rate (*percent*)

At a nominal discount rate of 20 percent and an inflation rate of 3 percent, the inflation-adjusted discount rate is:

$$IADR = \left(\frac{1+0.20}{1+0.03}\right) - 1 = 16.5 \textit{ percent}$$

The 3 percent inflation adjusted discount rate for discount rates of 30 percent and 5 percent are 26.2 percent and 1.9 percent respectively.

At an inflation adjusted discount rate of 16.5 percent, and a 20-*year* project life, **Equation 3.7** yields:

$$PWF = \left(\frac{(1+0.165)^{20} - 1}{0.165\,(1+0.165)^{20}}\right) = 5.78$$

The present worth factors for *IADR* = 26.2 percent and *IADR* = 1.9 percent are 3.78 and 16.51.

The net present value cost (***NPVC***) is determined by:

$$NPVC = (PWF \times \text{Annual Operating Cost}) + \text{First Cost} \qquad \textbf{Equation 3.9}$$

The net present value cost for the *ENHANCED STATIC REGAIN* design of **Figure 3.3** with an initial operating cost of $1,954 and first of $10,580 is determined from **Equation 3.9** as:

$$= (5.78 \times \$1{,}954) + \$10{,}580 = \$21{,}874$$

Table 3.12 compares the net present value costs for the three discount levels assuming a 3 percent annual inflation and a 20 year project life.

Table 3.12
System Cost Comparison

DESIGN METHOD	EQUAL FRICTION	STATIC REGAIN	ENHANCED STATIC REGAIN
System Total Pressure (*inches wg*)	1.06	1.04	1.04
System Static Pressure (*inches wg*)	0.78	0.76	0.76
First Cost	$13,548	$12,217	$10,580
Annual Operating Cost	$1,992	$1,954	$1,954
Net Present Value Cost at 30% Discount	$21,078	$19,603	$17,966
Net Present Value Cost at 20% Discount	$25,062	$23,511	$21,874
Net Present Value Cost at 5% Discount	$46,436	$44,478	$43,061

At system total pressure of 1.04 *inches wg*, the *ENHANCED STATIC REGAIN* design is 22 percent less expensive on a first cost basis than the equal friction design and, when using a 20 percent discount rate, and almost 13 percent less expensive on a net present value cost basis. The *ENHANCED STATIC REGAIN* design gives a lower material cost (first cost) at any system total pressure. Keep in mind that the lower first cost resulted because of the enhancement of duct and fittings. The smaller ducts and the less efficient fittings in non-design legs helped to reduce first cost.

Working under the assumption that the *ENHANCED STATIC REGAIN* design gives the lowest owning or net present value cost for any system, the owning cost may still not be minimized. Several design should be done (by computer) that vary the initial conditions, such as increasing or decreasing the initial velocity, so that the outcomes are different system total pressure requirements. The owning cost or net present value cost should be calculated for each and plotted as a function of the system total pressure requirement. The minimum value of this curve is the cost optimization point. For more information on Cost Optimization, see **Engineering Report No. 144, Computer-Aided Duct Design: Comparing the Methods**.

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.

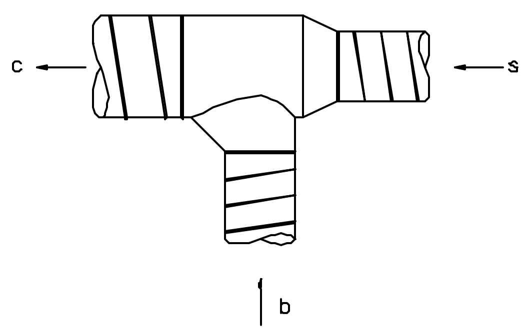


Figure 4.1
Converging Flow

$$Q_c = Q_b + Q_s$$ **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

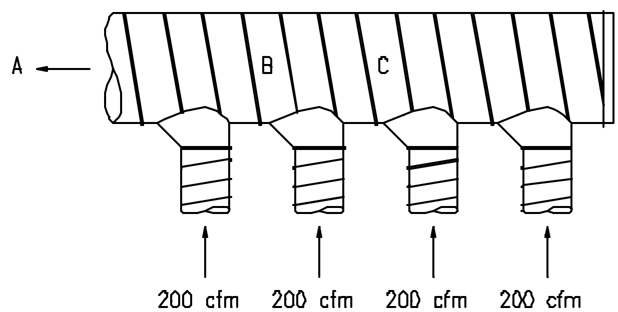


Figure 4.2
Multiple Converging Flow

*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: $DTP = DSP = C \times VP$

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:

$$DTP_{s\text{-}c} = C_s \text{ x } V$$ **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 x VP***) from **Equation 4.4**.

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

where:

$DSP_{s\text{-}c}$ = 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*.

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

where:

C_b = Branch loss coefficient

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

VP_b = Branch velocity pressure (*inches wg*)

Rewriting in terms of total pressure loss:

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

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

where:

ΔSP_{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.

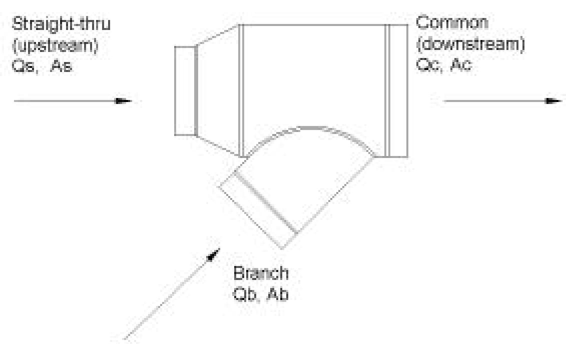


Figure 4.3
Definition of Parameters for a Typical Fitting

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.

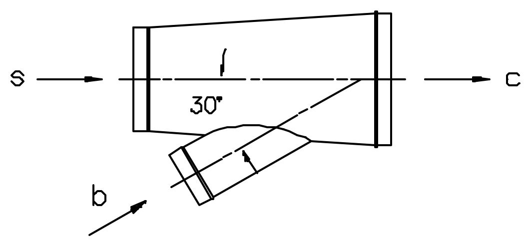
Since they are symmetrical, the loss coefficients for Y-branches, bullhead tees with vanes, and bullhead tees without vanes are calculated with the following rules:

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* Q_b = 2,500 *cfm* Q_c = 13,000 *cfm*
D_s = 25 *inches* D_b = 12 *inches* D_c = 27 *inches*

Answer: From **Equation 4.1**

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

From:

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

$$A_s = p25^2/576 = \underline{3.41\, square\ feet}$$

$$A_b = p12^2/576 = \underline{0.79\, square\ feet}$$

$$A_c = p27^2/576 = \underline{3.98\, square\ feet}$$

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:

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

$$C_b = \underline{0.12} \quad \textit{(Branch)}$$

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**:

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

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

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

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

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

Static pressure drops from **Equation 4.7**:

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

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.

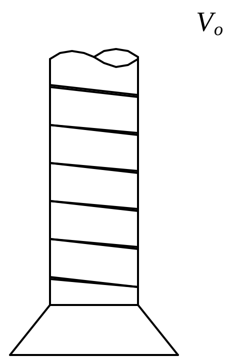


Figure 4.4
Hood

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**.

CHAPTER 5: Designing Exhaust Duct Systems

5.1 Defining the Application and Determining Parameters

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

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

5.2 Determining Capture Velocities and Air Volume Requirements

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

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

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

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

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

Plenum Velocity: Air velocity in the plenum.

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

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

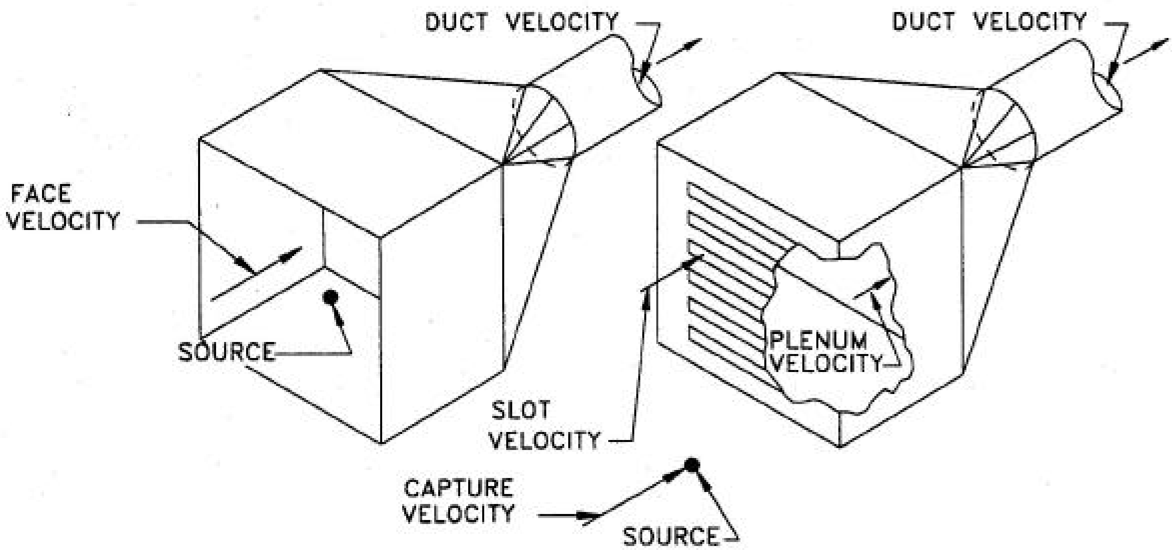


Figure 5.1
Hood Terminology
(Reprinted from Figure 3-1 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

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

1. Required velocity to capture or entrain the material or fumes (capture velocity).

2. Size and type of the hood entrance (dimensions).

3. Distance from the hood entrance to the process, which is producing the material or fumes to be exhausted.

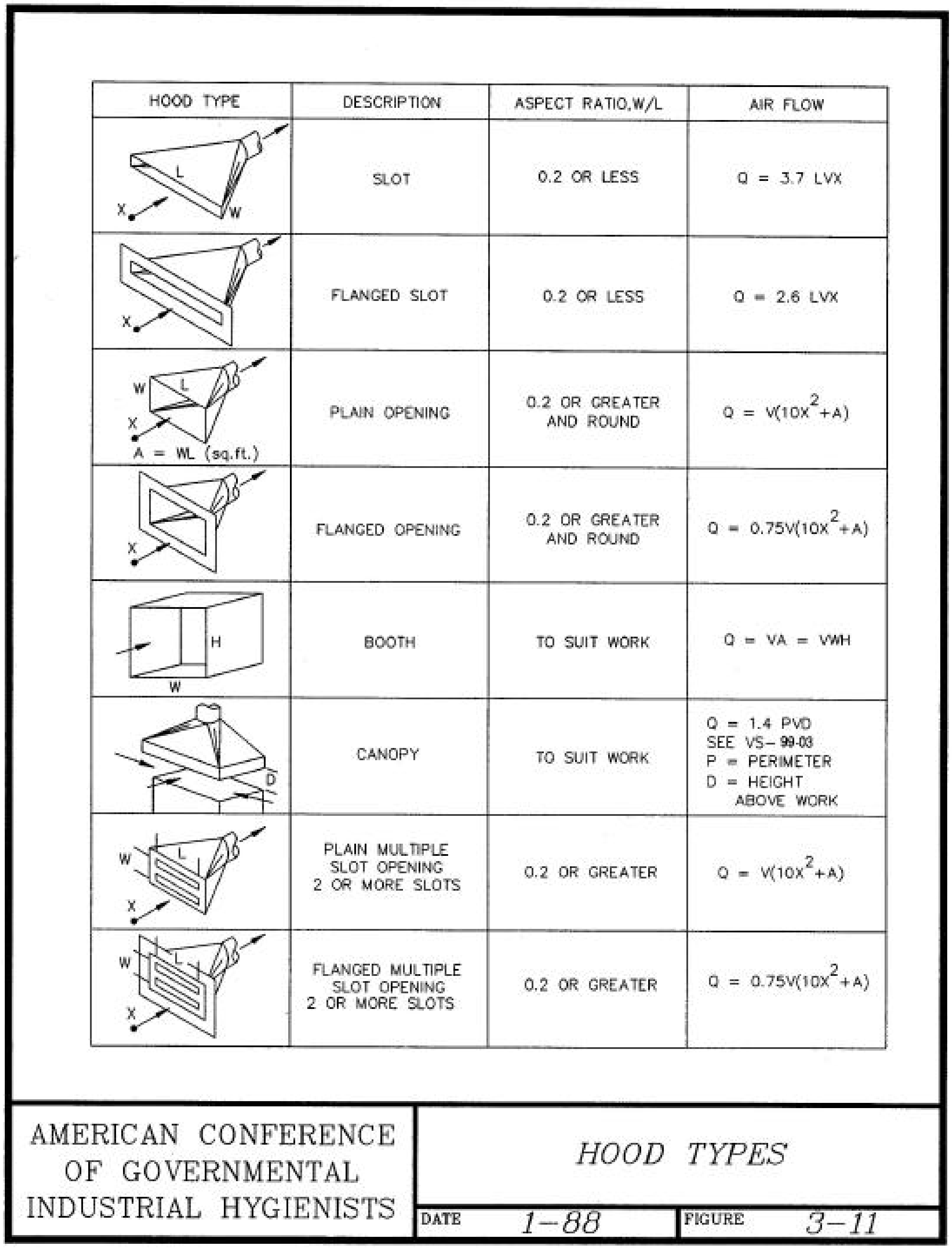
4. Miscellaneous factors such as crosscurrents, temperature, obstructions, and adjacent equipment.

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

Table 5.1
Typical Capture Velocities
(Reprinted from Table 3-1 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

Conditions of Dispersion of Contaminant	Examples	Capture Velocity (fpm)
Released with practically no velocity into quiet air.	Evaporation from tanks, degreasing, etc.	50-100
Released at low velocity into moderately still air.	Spray booths, intermittent container filling, low speed conveyor transfers, welding, plating, and pickling.	100-200
Active generation into zone of rapid air motion.	Spray painting in shallow booths, barrel filling, conveyor loading, and crushers.	200-500
Released at high initial velocity into zone of very rapid air motion.	Grinding, abrasive blasting, and tumbling.	500-2,000
In each category above, a range of capture velocities is shown and the proper choice of values depends on several factors:		
NOTES: Lower End of Range 1.Room air current minimal favorable to capture. 2. Contaminants of low toxicity or of nuisance value only. 3. Intermittent, low production. 4. Large hood: large air mass in motion.	Upper End of Range 1. Disturbing room air currents. 2. Contaminants of high toxicity. 3. High production, heavy use. 4. Small hood: local control only.	

Typical intake configurations are shown in **Figure 5.2** with various hood types, descriptions, aspect ratios, and air volume flow rate equations.



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

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HOOD TYPES

DATE *1–88* FIGURE *3–11*

Figure 5.2
Intake Configurations
(Reprinted from Figure 3-11 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

Sample Problem 5-1

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

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

$$Q = V \times A$$

$$= 100\ fpm \times (3\ ft \times 2\ ft)$$

$$= \underline{600\ cfm}$$

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

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

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

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

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

The mass flow rate is:

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

which can be rewritten as:

$$Q_{act} = \left(\frac{\rho_{std}}{\rho_{act}} \right) Q_{std} \qquad \textbf{Equation 5.1b}$$

where:

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

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

$$Q_{act} = Q_{std} \left(\frac{T_a + 460}{530} \right) \left(\frac{29.921}{b} \right) \qquad \textbf{Equation 5.2}$$

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

$$\rho_{act} = \left(\frac{\dfrac{b - p_w}{53.352} + \dfrac{p_w}{85.778}}{T_a + 460} \right) 70.53 \qquad \textbf{Equation 5.3}$$

where:

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

$$p_w = \frac{\beta W}{(0.62198 + W)}$$ **Equation 5.4**

where:

p_w = Water vapor partial pressure (*inches Hg*)

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

Sample Problem 5-2

---

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

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

$$Q = LHV$$

where:

L = Length (*ft*)

H = Height (*ft*)

V = Capture velocity (*fpm*)

Q_{std} = 4.4 *x* 1 *x* 100 = <u>440 *scfm*</u> (*standard cfm*)

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

$$\rho_{act} = \rho_{std}\left(\frac{530}{T_a + 460}\right)\left(\frac{\beta}{29.921}\right) = 0.075\left(\frac{530}{400 + 460}\right)\left(\frac{29.921}{29.92}\right)$$

$$\rho_{act} = \underline{0.046\ lb_m/ft^3}$$

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

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

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

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

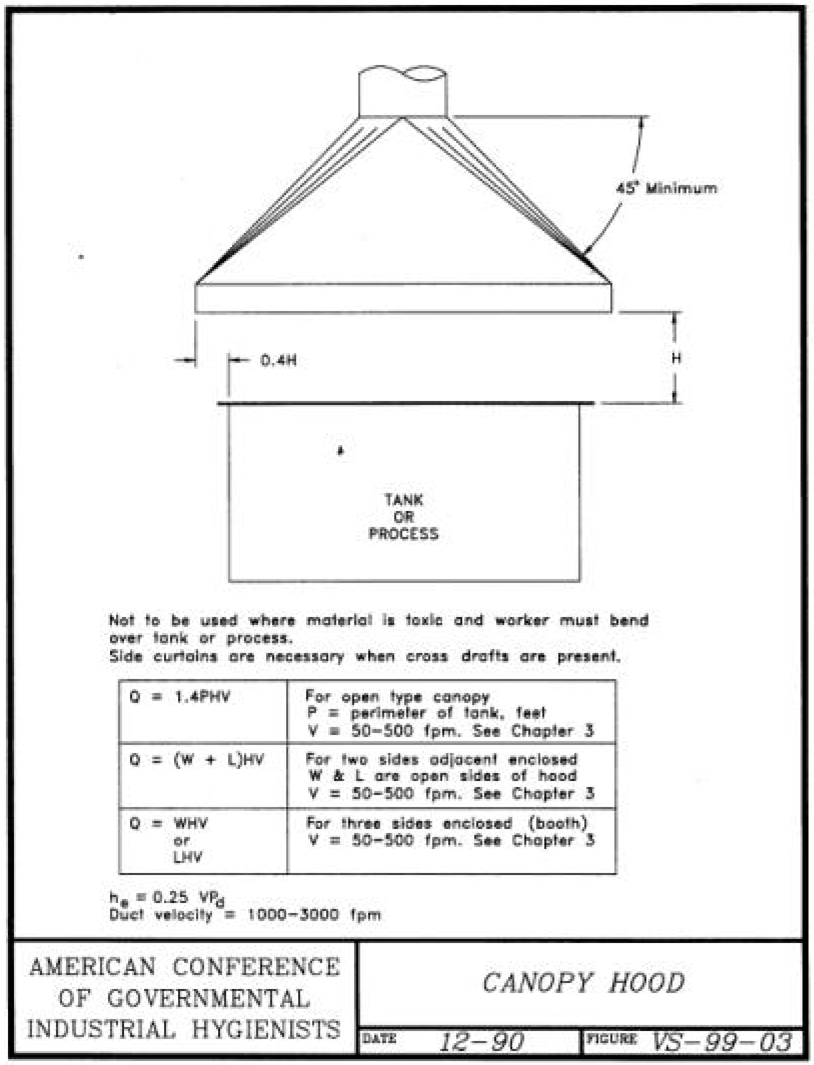

Figure 5.3
Canopy Hood
(Reprinted from Figure VS-99-03 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

5.3 Locating Duct Runs

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

5.4 Determining Duct Sizes Based on Velocity Constraints

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

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

Table 5.2
Duct Carrying Velocities
(Reprinted from Table 3-2 of the "Industrial Ventilation: A Manual of Recommended Practice," 22nd edition, 1995. Published by ACGIH®, Cincinnati, OH)

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

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

Sample Problem 5-3

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

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

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

Q_{act} = 717 *cfm* (from **Sample Problem 5-2**)

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

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

A = 0.398 *x* 144 = $\underline{57.3\ square\ inches}$

Determine the maximum diameter of the duct from:

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

Calculating for the diameter:

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

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

D = $\underline{8.5\ inches}$

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

5.5 Mixing of Two Air Streams

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

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

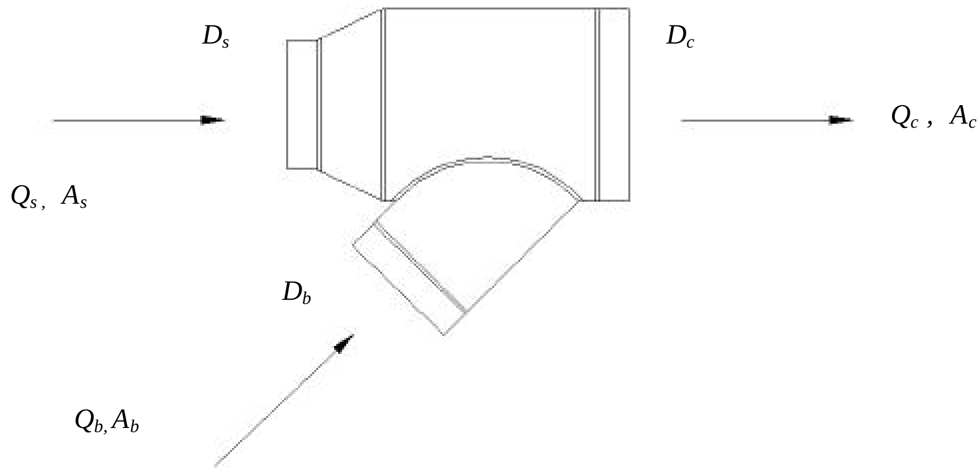

Figure 5.4
Mixing at Junction

The enthalpy of air can be expressed as:

$$h = C_pT + W(1{,}061 + 0.444T) \qquad \textbf{Equation 5.5}$$

where:

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

W = Humidity ratio (*lb$_m$ water vapor/lb$_m$ dry air*)

C_p = *Specific* heat of air (0.24 *Btu/lb$_m$ -°F*)

T = Temperature of the air (°*F*)

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

$$h_{s\,exit} = 0.24T_{s\,exit} + W_s(1061 + 0.444T_{s\,exit}) \qquad \textbf{Equation 5.6}$$

$$h_{bexit} = 0.24T_{bexit} + W_b(1061 + 0.444T_{bexit}) \qquad \textbf{Equation 5.7}$$

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

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

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

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

where:

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

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

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

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

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

where:

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

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

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

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

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

where:

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

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

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

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

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

$$\dot{m}_c = \dot{m}_s + \dot{m}_b \qquad \textbf{Equation 5.13}$$

where:

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

$\dot{m}_s$ = Mass flow rate in the straight-through (upstream) section (lb_m/min)

$\dot{m}_b$ = Mass flow rate in the branch sections (lb_m/min)

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

$$Q_c = \frac{\dot{m}_c}{r_c} \qquad \textbf{Equation 5.14}$$

where:

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

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

$$T_{exit} = (T_{inlet} - T_{amb})\, e^{-\frac{UPL}{60\dot{m}C_p}} + T_{amb} \qquad \textbf{Equation 5.15}$$

where:

U = Overall heat transfer coefficient ($Btu/hr\text{-}ft^2\text{-}°F$)

P = Perimeter of the duct (ft)

L = Length of the duct (ft)

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

C_p = Specific heat of air ($0.24\ Btu/lb_m\text{-}°F$)

60 = Conversion for lb_m/min to lb_m/hr

T_{exit} = Temperature exiting the duct ($°F$)

T_{inlet} = Temperature entering the duct ($°F$)

T_{amb} = Temperature of the air around the outside of the duct (°F)

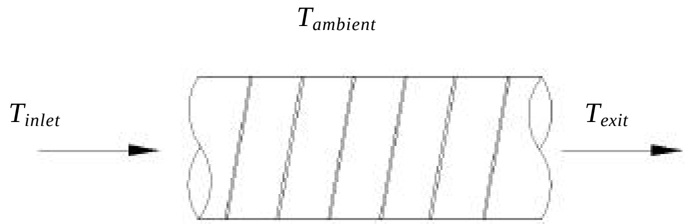

Figure 5.5
Duct Exit Temperature

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

5.6 Determining System Pressure Requirements

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

$$Q = A \times V \text{ and } Q = \frac{\dot{m}}{\mathbf{r}}$$ **Equation A.3**

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

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

1. Calculate the system pressure loss using the actual air volume (acfm), assuming standard conditions.

2. Multiply the total pressure by the appropriate density correction factor from **Appendix A.1.5**. This will be the new system pressure requirement for operation at the nonstandard conditions.

3. Recalculate horsepower requirements. Fan operation specifications should be based

on the corrected pressure and horsepower.

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

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

1. Calculate the volume requirements assuming standard conditions (*scfm*).

2. Multiply the standard volume by the inverse of the appropriate density factor from **Appendix A.1.5**. This will be the actual volume (*acfm*) requirement for operation at the nonstandard conditions.

3. All design and pressure calculations should be based on corrected volumes.

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

Sample Problem 5-4

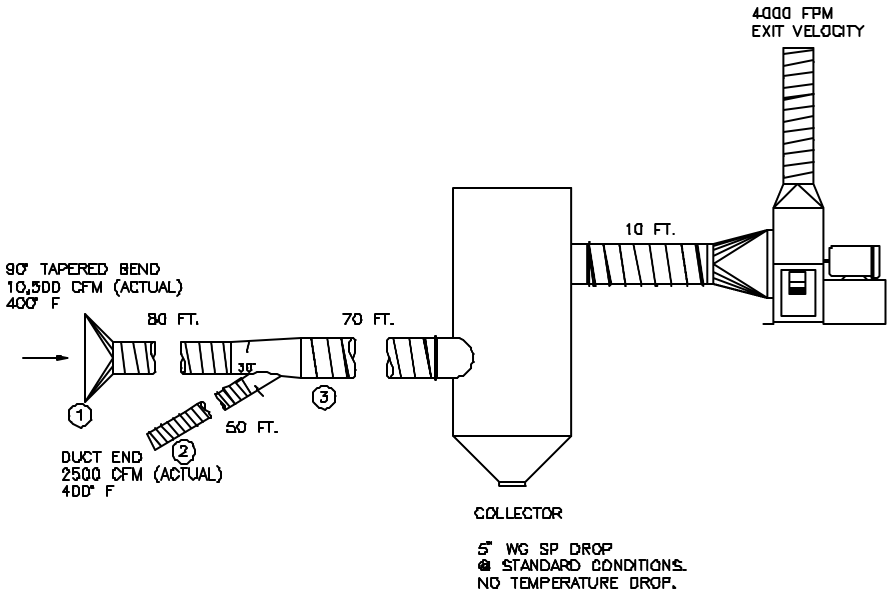

Figure 5.6
Exhaust System

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

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

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

Section 1: $A_{duct} = Q_{act}/V = 10{,}500\ cfm/4{,}000\ fpm = \underline{2.63\ ft^2}$
$= 378\ in^2$

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

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

$$V = Q/A = (10{,}500\ cfm)/(\ p\,(21)^2/4)(144\ \text{in}^2/\text{ft}^2))$$
$$= \underline{4{,}365\ fpm}$$

VP_s = (4,365 *fpm*/4,005) ^{2} = <u>1.19 *inches wg*</u>

<u>Section 2</u>: A_{duct} = Q_{act}/V = 2,500 *cfm*/4,000 *fpm* = <u>0.63 ft^2</u>
= 0.625 ft^2 x (144 in^2/ft^2) = <u>90.72 in^2</u>

D_{max} = $[4(90\ in^2)/\ p]^{0.5}$ = <u>10.75 *inches*</u>

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

V = Q/A = (2,500 *cfm*)/(*p* (10.5 in^2/4)/(144 in^2/ft^2))
= <u>4,158 *fpm*</u>

VP_b = (4,158 *fpm*/4,005) ^{2} = <u>1.08 *inches wg*</u>

<u>Section 3</u>: Q_{act} = 10,500 + 2,500 = <u>13,000 *cfm*</u>

A_{duct} = 13,000 *cfm*/4,000 *fpm* = <u>3.25 ft^2</u>
= 3.25 ft^2 x (144 in^2/ft^2) = <u>468 in^2</u>

D_{max} = $[4(468\ in^2)/\ p]^{0.5}$ = <u>24.41 *inches*</u>

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

V = (13,000 *cfm*)/(*p* (24 in^2/4)/(144 in^2/ft^2))
= <u>4,138 *fpm*</u>

VP_c = (4,138 *fpm*/4,005) ^{2} = <u>1.07 *inches wg*</u>

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

Table 5.3
Initial Sizing Information

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

* Based on standard conditions

<u>Part 2</u>: Determine the pressure losses for the branches and sections.

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

A_s = 2.41 ft^2; <u>A_s/A_c = 0.77</u>; <u>Q_b/Q_c = 0.19</u> <u>C_s = -0.06</u>

$A_b = 0.60\ ft^2$; $\underline{A_b/A_c = 0.19}$; $\underline{Q_s/Q_c = 0.81}$ $\underline{C_b = 0.11}$
$A_c = 3.14\ ft^2$;

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

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

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

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

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

Determine the section pressure losses:

Section 1:

Hood (entrance) Loss

From **Equation 4.8** $DTP_{h1} = C_{h1} \times VP_s$, where C_{h1} = 0.25 for a 90° tapered hood from an entrance loss chart (see **Appendix A.9.5**).

$$DTP_{h1} = 0.25(1.19) = \underline{0.30\ inches\ wg}$$

From **Equation 4.10**:

$$DSP_{h1} = (C_{h1} + 1)VP_s = (0.25 + 1) \times (1.19) = \underline{1.49\ inches\ wg}$$

Duct Friction Loss

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

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

Therefore:

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

The duct static pressure is determined from:

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

Losses for Entire Section

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

$DTP_{s\text{-}c}$ = -0.07 *inches wg*
$DSP_{s\text{-}c}$ = -0.19 *inches wg*

The total pressure loss for Section 1 is:

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

The static pressure loss for Section 1 is:

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

Section 2:

Plain Duct End (entrance) Loss

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

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

= 0.93(1.08) = 1.00 *inches wg*

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

Duct Friction Loss

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

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

Therefore:

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

The duct static pressure is determined from:

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

Losses for Entire Section

The branch losses from previous calculations are:

$DTP_{b\text{-}c}$ = 0.12 *inches wg*
$DSP_{b\text{-}c}$ = 0.11 *inches wg*

The total pressure loss for Section 2 is:

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

The static pressure loss for Section 2 is:

$$DSP_{Section\ 2} = DSP_{h2} + DSP_{duct\ 2} + DSP_{b\text{-}c} = 2.08 + 1.04 + 0.11)$$
$$= \underline{3.23\ inches\ wg}$$

Section 3:

Duct Friction Loss

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

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

Therefore:

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

The duct static pressure is determined from:

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

Collector Loss

$$DTP_{\text{collector}} = 5.00\ inches\ wg$$
$$DSP_{\text{collector}} = 5.00\ inches\ wg$$

Losses for Entire Section

The total pressure loss for Section 3 is:

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

The static pressure loss for Section 3 is:

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

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

Table 5.4
Pressure Losses

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

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

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

Path (1-3): *DTP* = 1.03 + 5.62 = 6.65 *inch wg*
DSP = 2.11 + 5.62 = 7.73 *inches wg*

Path (2-3): *DTP* = 2.16 + 5.62 = 7.78 *inches wg*
DSP = 3.23 + 5.62 = 8.85 *inches wg*

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

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

Excess total pressure:

Path (1-3): 7.78 - 6.65 = 1.13 *inches wg*

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

Calculate the Total and Static Pressure of the Fan:

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

$$TP_{exit} = VP_{exit}$$

$$= \left(\frac{4{,}000}{4{,}005}\right)^2 = 1.07 \ \ inches \ \ wg$$

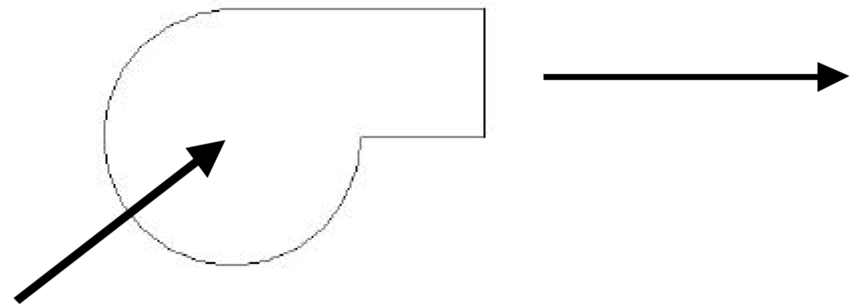

Figure 5.7
Fan

$Fan\ TP = TP_{out} - TP_{in}$ = 1.07 - (-7.78) = <u>8.85 *inches wg*</u>
$Fan\ SP = SP_{out} - SP_{in} - VP_{in}$ = 0 - (-8.85) - 1.07 = <u>7.78 *inches wg*</u>

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

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

$Fan\ TP_{actual}$ = *Fan TP x (density correction factor)*
$Fan\ SP_{actual}$ = *Fan SP x (density correction factor)*

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

$Fan\ TP_{actual}$ = 8.85(0.51) = <u>4.51 *inches wg*</u>
$Fan\ SP_{actual}$ = 7.78(0.51) = <u>3.97 *inches wg*</u>

CHAPTER 6: Analyzing Exhaust Duct Systems

6.1 Fitting Selection

If a system is exhausting abrasive particulate, the designer must address accelerated abrasive actions on the fitting walls caused by the angular impact of the particulate. These actions are best controlled by the proper choice of fitting types. Fittings that provide a gradual change in airflow direction limit the additional abrasion due to angular impact. The following recommendations apply in these situations:

1. If the particulate is not abrasive, use elbows that have a minimum centerline radius of 1.5 x diameter. If the particulate is abrasive, use elbows that have a minimum centerline radius of 2.5 x diameter. If the amount of abrasive particulate warrants it, consider using a flat back elbow with a wear plate that is replaceable. It should still that have a minimum centerline radius of 2.5 x diameter.

2. For abrasive material, converging-flow fitting branches should enter the main at no greater than a 30E angle. If there is any type of particulate or fumes being exhausted, converging-flow fitting branches should never enter the main at greater than a 45E angle. Converging-flow fitting branches entering the main at a 90E angle are not recommended and should never be used except in low pressure return air systems.

3. Tapered body converging-flow fittings should be considered when particulate fallout will create a hazardous situation or will promote plugging in the duct system.

6.2 Balancing the System

In the previous chapter, system pressure requirements were determined for an exhaust duct system (**Sample Problem 5-4**). This determination of system pressure also illustrated the excess pressure or imbalance in the system. If a system has branches with excess pressure, the flow through the nondesign branches will be greater than the design amount and the flow in the design will be less than the design amount. Airflow will travel to the path with least resistance. To get the correct flow through each branch, the system needs to be balanced. There are several ways to balance a system to obtain the correct airflow.

6.2.1 Using Dampers

One way to balance a system is to add dampers in the nondesign legs. Dampers restrict the flow and cause the pressure in each nondesign branch to increase to the point of excess pressure and thereby balance the system. However a damper can hinder the flow of certain materials through the duct system, causing erosion, particulate buildup, or other negative effects. **Section 3.2.1** discusses balancing supply systems using dampers. Similar concepts apply to return and exhaust systems.

6.2.2 Using Corrected Volume Flow Rate

The other method of balancing is *correcting the air volume flow rate.* Airflow in nondesign legs is increased until that branch or sections excess pressure is zero (thus also becoming a design leg) by matching the systems pressure requirement. **Equation 6.1** can be used to estimate the amount of airflow necessary to balance a section:

$$cfm_{corrected} = cfm_{inlet}\sqrt{\frac{|\,Fan_{inlet}\ SP\,|}{|\,Fan_{inlet}\ SP\,| - |\,Excess\ SP\,|}} \qquad \textbf{Equation 6.1}$$

where:

***cfm**corrected*	=	Corrected volume flow rate (*cfm*)
***cfm**inlet*	=	Inlet volume flow rate (*cfm*)
Fan**inlet**SP	=	Inlet to fan static pressure (*inches wg*)
Excess SP	=	Excess static pressure in branch (*inches wg*)

Sample Problem 6-1 illustrates how **Equation 6.1** is used.

Sample Problem 6-1

What is the corrected volume flow rate for section 1 of Sample Problem 5-4? This section's inlet airflow volume is 10,500 cfm, the fan inlet static pressure is -8.85 inches wg, and the excess pressure is 1.13 inches wg.

Answer: From **Equation 6.1**:

$$cfm_{corrected} = 10{,}500\sqrt{\frac{|-8.85|}{|-8.85|-|1.13|}}$$

= 11,242 *cfm*

[All analysis are done assuming standard conditions.] Enter the corrected cfm for better balancing in the branch. The corrected cfm is greater than the original by about 16.6 percent. This increase in volume flow rate will change the design and should be accounted for in the design. This method of balancing requires exact layouts, and changes in the system are not recommended since they will upset the balance. Particulate or dust accumulation should not occur if the design conditions are maintained. **Table 6.1** summarizes the results of the increased airflow. The system is more balanced but it is not significant at this point. With the redesign there is only 0.96 *inches wg* of excess pressure in section 1 compared to 1.13 *inches wg* without the extra airflow. We would need even more air, so another iteration would need to be done to further increase the airflow. Already though we have increased the fan inlet total pressure requirement to -8.42 *inches wg* from the -7.78 *inches wg* without the additional airflow and the airflow has increased by 742 *cfm* as well. Both of these directly affect the fan power requirements. Therefore, increasing the airflow may eventually balance the system, but the cost of the increased power consumption may not be worth it.

6.2.3 Using Smaller Duct Sizes and Less Efficient Fittings

Probably the most economical way to balance an exhaust system is similar to what we used in **Section 3.2.3**. By using smaller duct sizes in the nondesign legs of the system, we increase the friction rate and typically increase the dynamic losses of any fittings because of the higher velocities encountered. To do this, increment the duct size smaller until it creates a new design leg, then keep the previous size. The result will be a better-balanced system, with the benefit of smaller sizes that are less expensive and easier to install***; at no additional operating cost***. The system will operate with the same airflow volume and total pressure requirements as in the original design.

Sample Problem 6-2

Resize section 1 of Sample Problem 5-4 to produce a balanced system.

Answer: Three iterations were required to balance the system from Sample Problem 5-4. Again all calculations are done at standard conditions.

The first iteration begins with decreasing the size in section 1 of the system to 20 *inches* from 21 *inches*. This reduced the excess pressure in section 1 to 0.71 *inches wg*. Note that the fan inlet pressure requirement (the negative of the highest total cumulative *DTP*) actually decreased slightly to -7.71 *inches wg* from -7.78 *inches wg* in Sample Problem 5-4. That is because when we change the size (or airflow) of one of the branches of a converging flow fitting, the loss coefficient of both branches is changed. For this first iteration with the smaller size in section 1, the straight-through loss coefficient, C_s, increased from -0.07 to 0.01 while the branch coefficient, C_b, decreased from 0.11 to 0.05. A summary of these results is given in **Table 6-2a**.

For the second iteration, the size in section 1 was decreased to 19 *inches*. This reduced the excess pressure in section 1 to 0.13 *inches wg*. Again, the fan inlet total pressure requirement decrease slightly, this time to -7.60 *inches wg*. For this iteration, the straight-through loss coefficient, C_s, increased to 0.07 while the branch coefficient, C_b, decreased -0.05. A summary of these results is given in **Table 6-2b**.

For the third iteration, the size in section 1 was decreased to 18 *inches*. This changed the design leg to sections 1-3 and caused the excess pressure to be 0.80 *inches wg*, but in section 2. The fan inlet total pressure requirement increased to 8.22 *inches wg*, which is more than the original design. For this iteration, the straight-through loss coefficient, C_s, increased to 0.17 while the branch coefficient, C_b, decreased -0.22. A summary of these results is given in **Table 6-2c.**

Because the third iteration changed the design leg (and increased the system's pressure requirements), we should revert back to the second iteration where a 19 *inches* diameter duct produced the best balancing while not increasing the operation cost. The duct system cost would also be reduced, as 80 *feet* of 19-*inch* ductwork would cost less than 80 *feet* of 21-*inch* ductwork.

Table 6.1
Redesign of Sample Problem 5-4, Increased Airflow in Section 1

SECTION					Inlet			Duct	Fitting	Branch			Other		Section Total		Cumulative Pressure		Section Excess Pressure
No.	Size (*inches*)	Volume Flow Rate (*cfm*)	Velocity (*fpm*)	VP (*in wg*)	Loss Coeff.	*DTP* (*in wg*)	*DSP* (*in wg*)	*DP* (*in wg*)	*DP* (*in wg*)	Loss Coeff.	*DTP* (*in wg*)	*DSP* (*in wg*)	Loss Coeff.	*DP* (*in wg*)	*DTP* (*in wg*)	*DSP* (*in wg*)	*DTP* (*in wg*)	*DSP* (*in wg*)	*DTP* (*in wg*)
1	21	11242	4674	1.36	0.25	0.34	1.70	0.94	n/a	-0.06	-0.08	-0.25			1.20	2.39	7.46	8.66	0.96
2	10.5	2500	4158	1.08	0.93	1	2.08	1.04	n/a	0.11	0.12	0.23			2.16	3.35	8.42	9.62	n/a
3	24	13742	4374	1.19	n/a	n/a	n/a	0.69	n/a				4.67	5.57	6.26	6.26	6.26	6.26	n/a

Table 6.2a - Iteration 1
Redesign of Sample Problem 5-4, Decreased Size in Section 1

SECTION					Inlet			Duct	Fitting	Branch			Other		Section Total		Cumulative Pressure		Section Excess Pressure
No.	Size (*inches*)	Volume Flow Rate (*cfm*)	Velocity (*fpm*)	VP (*in wg*)	Loss Coeff.	*DTP* (*in wg*)	*DSP* (*in wg*)	*DP* (*in wg*)	*DP* (*in wg*)	Loss Coeff.	*DTP* (*in wg*)	*DSP* (*in wg*)	Loss Coeff.	*DP* (*in wg*)	*DTP* (*in wg*)	*DSP* (*in wg*)	*DTP* (*in wg*)	*DSP* (*in wg*)	(*in wg*)
1	20	10500	4813	1.44	0.25	0.36	1.81	1.01	n/a	0.01	0.01	-0.36			1.39	2.45	7.01	8.07	0.71
2	10.5	2500	4158	1.08	0.93	1	2.08	1.04	n/a	0.05	0.05	0.04			2.09	3.16	7.71	8.78	n/a
3	24	13000	4138	1.07	n/a	n/a	n/a	0.62	n/a				4.67	5.00	5.62	5.62	5.62	5.62	n/a

Table 6.2b - Iteration 2
Redesign of Sample Problem 5-4, Decreased Size in Section 1

SECTION					Inlet			Duct	Fitting	Branch			Other		Section Total		Cumulative Pressure		Section Excess Pressure
No.	Size	Volume Flow Rate	Velocity	VP	Loss	ΔTP	ΔSP	ΔP	ΔP	Loss	ΔTP	ΔSP	Loss	ΔP	ΔTP	ΔSP	ΔTP	ΔSP	
	(inches)	*(cfm)*	*(fpm)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*
1	19	10500	5333	1.77	0.25	0.36	2.22	1.29	n/a	0.07	0.12	-0.58			1.86	2.93	7.48	8.55	0.13
2	10.5	2500	4158	1.08	0.93	1	2.08	1.04	n/a	-0.05	-0.05	-0.07			1.98	3.05	7.60	8.67	n/a
3	24	13000	4138	1.07	n/a	n/a	n/a	0.62	n/a				4.67	5.00	5.62	5.62	5.62	5.62	n/a

Table 6.2c - Iteration 3
Redesign of Sample Problem 5-4, Decreased Size in Section 1

SECTION					Inlet			Duct	Fitting	Branch			Other		Section Total		Cumulative Pressure		Section Excess Pressure
No.	Size	Volume Flow Rate	Velocity	VP	Loss	ΔTP	ΔSP	ΔP	ΔP	Loss	ΔTP	ΔSP	Loss	ΔP	ΔTP	ΔSP	ΔTP	ΔSP	
	(inches)	*(cfm)*	*(fpm)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	**Coeff.**	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*	*(in wg)*
1	18	10500	5942	2.20	0.25	0.55	2.75	1.67	n/a	0.17	0.37	-0.76			2.60	3.66	8.22	9.28	n/a
2	10.5	2500	4158	1.08	0.93	1	2.08	1.04	n/a	-0.22	-0.24	-0.25			1.80	2.87	7.42	8.49	0.80
3	24	13000	4138	1.07	n/a	n/a	n/a	0.62	n/a				4.67	5.00	5.62	5.62	5.62	5.62	n/a

6.3 Specifying and Selecting a Fan

The information needed to specify and select a fan was shown in **Sample Problem 5-4**. Fan manufacturers catalog various information about the performance of their fans. This performance is based on the fan laws and tests run by the manufacturer. Fan data from the manufacturer is normally in terms of cfm and static pressure; however, some manufacturers use total pressure in their catalog. For more information about fans, fan testing, and performance, see AMCA publications (**Appendix A.9.6**).

From **Sample Problem 5-4**, the required fan total pressure is:

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

From a fan manufacturer's data at standard conditions, the fan needs to be sized for 8.85 *inches wg* total pressure at a volume airflow rate of 13,000 *cfm*, the required fan static pressure is:

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

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

Corrections for nonstandard conditions are expressed by the following:

$$Fan\ TP_{actual} = Fan\ TP \times Density\ correction\ factor$$
$$Fan\ SP_{actual} = Fan\ SP \times Density\ correction\ factor$$

For the conditions of **Sample Problem 5-4**, a nonstandard temperature of 400EF, barometric pressure of 24.90 inches Hg, and an elevation of 5,000 feet above sea level, the density correction factor from **Appendix A.1.5** is 0.51.

$$Fan\ TP_{actual} = 8.85(0.51) = \underline{4.51\ inches\ wg}$$
$$Fan\ SP_{actual} = 7.78(0.51) = \underline{3.97\ inches\ wg}$$

In summary, for sizing purposes, a fan capable of providing 13,000 cfm at 7.78 inch wg static pressure for standard conditions is needed for the system in **Sample Problem 5-4**. When the temperature is 400EF, the pressure is 24.90 inch Hg, and the elevation is 5,000 feet, the system will actually operate at a static pressure of 3.97 *inches wg* with the specified fan.

Balancing the system with dampers should not change the fan requirements as a damper in section 1 would be adjusted so the pressure requirements for the section 1-3 path are the same as those for the section 1-2 path, which is the design leg. Balancing the system by increasing airflow in one of the branches could significantly increase the cost (both first and operating) of the fan as both the airflow and fan inlet total pressure requirements increase. Using smaller sizes of duct to increase pressure loss in nondesign legs to balance the system, as was shown in **Sample Problem 6-2**, should have minimal affect on the fan requirements. For this system design the fan total pressure requirements would be:

$$Fan\ TP = \mathrm{TP_{out}} - \mathrm{TP_{in}} = 1.07 - (-7.60) = \underline{8.67\ inches\ wg}$$
$$Fan\ SP = SP_{\mathrm{out}} - SP_{\mathrm{in}} - VP_{\mathrm{in}} = 0 - (-8.67) - 1.07 = \underline{7.60\ inches\ wg}$$

Correcting for nonstandard conditions results in:

$$Fan\ TP_{actual} = 8.67(0.51) = \underline{4.42\ inches\ wg}$$
$$Fan\ SP_{actual} = 7.60(0.51) = \underline{3.88\ inches\ wg}$$

Therefore the same fan would probably be selected as in the unbalanced system design.

Investigate other considerations concerning fans, such as fan orientation and system effect prior to the selection. In exhaust systems, temperature, corrosion, erosion, and expansion may also affect system design and fan selection. Consult fan manufacturers for more specific application and selection information.

6.4 System Considerations

When analyzing exhaust systems, there are special considerations that are not found in systems such as those serving office buildings or commercial shopping centers. With the high temperatures, corrosive atmospheres, erosion, and various materials being transported, an industrial duct system must be able to resist those influences. Duct system performance from the standpoint of structural integrity, economic fitting selection, and available duct materials are just a few of the items for consideration. Reinforcement recommendations for spiral duct are located in **Appendix A.2.3**.

CHAPTER 7: Acoustical Fundamentals

7.1 Overview

The complete design of an air handling system requires that acoustical aspects be considered. It is not sufficient to simply deliver the proper volume flow rate at a design pressure. It must be done without creating excessive noise levels or tonal components.

Our discussion of acoustical air handling systems begins with a presentation of some fundamental acoustical concepts. An understanding of the fundamentals is necessary to analyze, evaluate, and remedy noise problems. In this chapter, sound power, sound pressure, decibels, frequency, loudness, and weighting are defined. **Chapter 8**, "Duct System Acoustics" includes a review of fan noise, natural attenuation, and generated and radiated duct noise.

Chapter 9 discusses room acoustics, including room characteristics and the criteria for determining allowable noise levels for various occupancy/use situations. Air terminal noise and design criteria are also covered.

Chapter 10 pulls everything from the previous chapters together and guides the designer in the proper steps to perform an acoustical analysis of a duct system.

7.2 Sound Power and Sound Pressure

The terms *sound power* and *sound pressure* sound similar, and they can be confusing to those not familiar with acoustics. Sound power, as the name implies, is a quantification of the actual acoustical power generated by a sound source. It is expressed in the power unit of watts. Sound power cannot be directly measured.

Sound pressure is a measurable fluctuation of the ambient air pressure generated by a sound source. It is expressed in the pressure unit of *Newtons per square meter* (or *Pascals*). The measured sound pressure will depend on a number of factors, including the magnitude of the sound power, the pressure measurement location with respect to the source, and the conditions along the propagation path from the source to the measurement location.

Have you ever noticed a vacuum cleaner sounding louder in a bathroom than say, the bedroom or outside? Or that it is louder nearer to it than far away? The vacuum cleaner has a constant noise generating ability (constant sound power), but the noise heard (sound pressure) is dependent on the environment and the proximity to the unit.

The range of possible sound power and sound pressure magnitudes is very large. For example, the sound power of a very faint noise at the lower limit of human audibility is 0.000000000001 (1×10^{-12}) *watts*. The sound power generated by a Saturn V rocket at lift-off is on the order of 100,000,000 ($1 \times 10^{+8}$) *watts*. Similarly, sound pressures can range from 0.00002 (2×10^{-5}) *Pascals* to 100,000 ($1 \times 10^{+5}$) *Pascals*.

Obviously, working with these very large and very small numbers would be very cumbersome, and since our ears don't hear variations in sound unless there are large differences in sound pressure, a logarithmic definition of sound power and sound pressure is used. When expressed in this fashion, the quantities are known as sound power level (abbreviated PWL or Lw) and sound pressure level (abbreviated SPL or Lp). The unit for both is the *decibel*. See **Equations 7.1** and **7.2** for definitions of sound power level and sound pressure level, respectively.

$$Lw = 10 \log_{10} \left(\frac{W}{Wref} \right)$$ **Equation 7.1**

where:

Lw	=	Sound power level (*dB*)
W	=	Sound power of the sound source (*watts*)
$Wref$	=	Standard reference sound power (1×10^{-12} *watts* or 1 *picawatt*)

$$Lp = 10 \log_{10} \left(\frac{P}{Pref} \right)^2$$ **Equation 7.2**

where:

Lp	=	Sound pressure level (*dB*)
P	=	measured sound pressure (*N/m^2 or Pascals*)
$Pref$	=	Standard reference sound pressure (2×10^{-5} *Pascals*, which is the threshold of youthful hearing)

Most people have some familiarity with sound levels measured in decibels. The sound level meter directly measures the local sound pressure through the use of a pressure transducer (microphone). In simple terms, the meter's pressure reading is converted to a sound pressure level using Equation 7.2.

Table 7.1 presents some typical sound sources and their approximate sound pressure levels.

Table 7.1
Sound Source and Sound Pressure Level

Source	**Lp** (*dB*)
Threshold of Hearing	0
Whisper	30
Normal Speech	60
Passing Truck	100
Pipe Organ (sforzando)	130
Jet Engine (near field)	160

Current governmental regulations state that continued exposure to sound pressure levels in excess of 85 *dB* can result in hearing impairment. Any exposure to levels exceeding 140 *dB* can result in permanent hearing damage.

Since sound levels are logarithmic quantities, an increase or decrease of only a few decibels is significant. For example, a sound pressure level change of 6 *dB* represents a doubling (or halving) of the sound pressure. Our ears, however, will not detect this change as a doubling or halving of the subjective loudness of the sound. Subjective reactions to changes in sound level are summarized in **Table 7.2**.

Table 7.2
Subjective Reactions

Change from Ambient Level	Subjective Reactions
+/- 1 *dB*	Not Detectable
+/- 3 *dB*	Just Detectable by Most People
+/- 10 *dB*	Perceived as Doubling/Halving of Loudness

Sound levels, whether it is sound pressure or sound power, cannot be added directly because they are logarithmic quantities. For example, two noise sources individually producing a sound pressure of 100 *dB* at a certain point in space will not produce 200 dB when operated simultaneously. Actually, the combined sound pressure level is 103 *dB*. The 100 dB sound pressure level is the resultant of 2 *Pascals* of sound pressure as derived in the following equation:

$$Lp = 10\,\log_{10}\left(\frac{2}{2\ x\ 10^{-5}}\right)^2 = 100\ dB$$

Adding two 100 dB sound pressures levels logarithmically is as follows:

$$Lp = 10\,\log_{10}\left[\left(\frac{2}{2\ x\ 10^{-5}}\right)^2 + \left(\frac{2}{2\ x\ 10^{-5}}\right)^2\right] = 103\ dB$$

therefore:

$$\sum Lp = 10\ \log_{10}\sum\left(\frac{P}{Pref}\right)^2 \qquad \textbf{Equation 7.3}$$

Combining sound pressure or sound power levels can involve extensive calculations. However, the following rule-of-thumb guidelines are helpful in making fairly accurate manual calculations.

Determining the difference between two of the levels and adding the adjustments shown in **Table 7.3** to the higher of the two levels can combine any number of sound levels.

Table 7.3
Simplified Decibel Addition

Difference in Levels	Add to Higher Level
0 - 1 *dB*	3 *dB*
2 - 4 *dB*	2 *dB*
5 - 9 *dB*	1 *dB*

Whenever the difference between two sound levels is 10 *dB* or more, the louder level masks the quieter source and there is no contribution to the overall level by the second source. **Sample Problem 7-1** provides an example of decibel addition.

Sample Problem 7-1

A listener is simultaneously subjected to the following sound pressure levels: 51 dB, 53 dB, 49 dB, 45 dB, 36 dB, 31 dB, 25 dB, and 24 dB. What is the overall sound pressure level?

Answer:

The sound levels are added in groups of two, in accordance with **Table 7.3** and the results of these groups are then coupled in like manner until a single sound level is attained. The additions can be carried out in any order and results should be identical (or should vary by no more than 1 *dB*).

(51 + 53) *dB* = 53 + 2 = 55 *dB*
(49 + 45) *dB* = 49 + 2 = 51 *dB*
(36 + 31) *dB* = 36 + 1 = 37 *dB*
(25 + 24) *dB* = 25 + 3 = 28 *dB*

Adding these results together:

(55 + 51) *dB* = 55 + 2 = 57 *dB*
(37 + 28) *dB* = 37 + 1 = 38 *dB*

Finally, (57 + 38) *dB* = 57 + 0 = 57 *dB*, the overall level.

7.3 Frequency

The *frequency* of a sound is determined by the number of sound waves (pressure fluctuations) produced per unit of time. Frequency can be correlated to the *pitch* of the sound and is measured in *cycles per second* or *Hertz* (*Hz*). Humans are capable of hearing sounds from about 20 *Hz* to 20,000 *Hz*.

As a point of reference, middle C on a piano keyboard has a frequency of approximately 260 *Hz*. Moving up the keyboard, each octave C will have a frequency twice the value of the lower octave. Similarly, moving down the keyboard, each octave C will have a frequency one half that of the upper octave. This relationship is true regardless of which tone is selected as a starting point.

Frequencies which are important from an acoustical standpoint can be grouped into octave bands, each with a defining center frequency that is twice the frequency of the next lower band center frequency and one half the frequency of the next higher band center frequency. The eight octave bands are shown in **Table 7.4**.

Table 7.4
Octave Bands

Band No.	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
Range (*Hz*)	45 to 90	90 to 180	180 to 355	355 to 710	710 to 1,400	1,400 to 2,800	2,800 to 5,600	5,600 to 11,200
Center Frequency (*Hz*)	63	125	250	500	1,000	2,000	4,000	8,000

Note that in the higher frequencies, the octave band ranges are much wider than in the lower frequencies. This indicates that our ability to distinguish constant increment frequency differentials is reduced as frequency increases.

7.4 Wavelength

The wavelength of sound is the distance between successive points of compression or rarefaction in the sound carrying medium, usually air. Two sound waves are drawn in **Figure 7.1**. Both waves are of the same frequency, one cycle per second or 1 Hz. One of the waves has a higher amplitude than the other wave, and therefore, is louder.

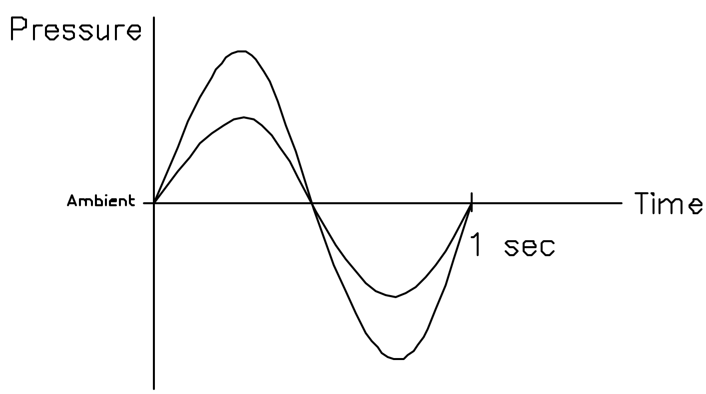

Figure 7.1
Graph of Two Sound Waves

The relationship between wavelength and frequency is shown in **Equation 7.4**.

$$l = \frac{c}{f}$$ **Equation 7.4**

where,

l = Wavelength (*ft*)

c = speed of sound (*feet per second*)

f = frequency of wave (*Hz*)

As discussed in **Section 7.3** the frequency of a sound is determined by the number of sound waves produced per unit of time. Since one complete wavelength is equal to one cycle, and we use the "second" as our unit of time measurement, a 63 Hz frequency sound has 63 wavelengths pass a point every second. This means that low frequency sounds have longer wavelengths than high frequency sounds. See **Table 7.5** for the wavelengths of the octave band center frequencies at room temperature. Since the speed of sound is a function of air density, changes in temperature result in changes in wavelength.

Table 7.5
Wavelengths of Octave Band Center Frequencies

Frequency (Hz)	Wavelength (feet)
63	17.9
125	9.0
250	4.5
500	2.3
1000	1.1
2000	0.6
4000	0.3
8000	0.1

7.5 Loudness

Loudness can be defined as the intensive attribute of an auditory sensation. It is a function of both sound pressure level and frequency. Various metrics have been created to measure loudness, but all recognize that low frequency noise is more tolerable than similar levels at higher frequencies.

As a result of numerous surveys made with a wide range of human subjects, equal loudness contours have been created which provide an indication of the actual pure tone sound pressure levels at various frequencies which are judged to be equal in loudness to a reference tone at 1,000 Hz. For example, in the first octave band (63 *Hz*), a 61 *dB* tone is considered to have the same loudness as a 40 *dB* tone at 1,000 *Hz*. In the seventh octave band (4,000 *Hz*), a 33 *dB* level is judged to be equal to the same 40 *dB* tone at 1,000 *Hz*. Thus the frequency of a sound will have a substantial bearing on how loud it is perceived to be.

Figure 7.2 presents typical equal loudness contours.

Free-Field Loudness Contours for Pure Tones
(Reference 2)

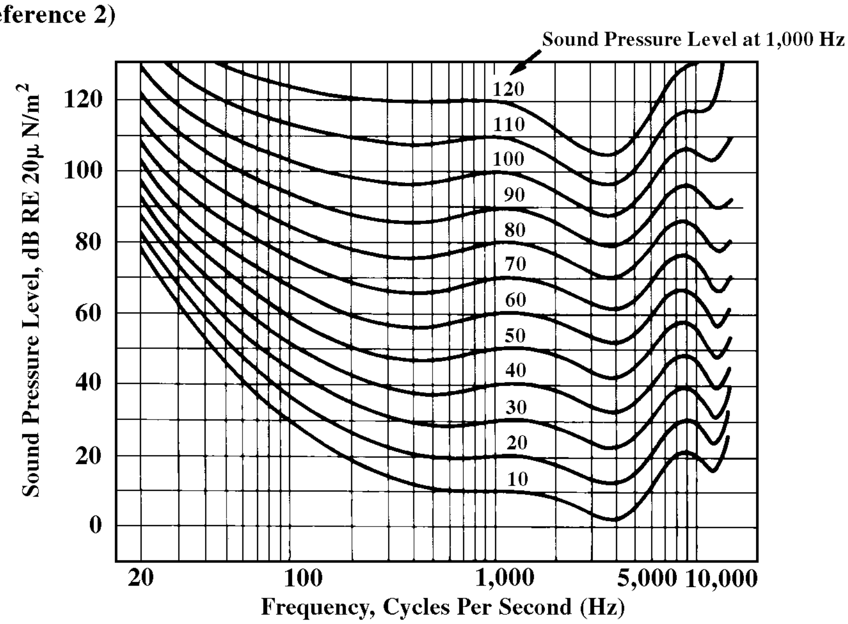

Figure 7.2
Equal Loudness Contours

7.6 Weighting

When sound pressure levels are measured in the various octave bands, it is often useful to *weight* the levels in accordance with their perceived contribution to loudness. Since, as indicated above, humans are less sensitive to noise with lower frequencies, the measured levels at these frequencies are reduced to reflect this. High frequency noises contribute substantially to annoyance and should not be reduced from their measured levels. In fact, the human ear is so sensitive to noises at 2,000 and 4,000 *Hz* that the most common weighting system, A-weighting, actually increases these levels slightly.

The A-weighting system requires that adjustments be made to the measured sound pressure levels at each frequency, in accordance with their relative contribution to annoyance or loudness. In this way, the A-weighted sound levels will more nearly reflect the response characteristics of the human ear. **Table 7.5** presents the A-weighting adjustments as a function of frequency.

Table 7.5
A-Weighting

Octave Band Number	Frequency (*Hz*)	Adjustment (*dB*)
1	63	-26
2	125	-16
3	250	-9
4	500	-3
5	1,000	0
6	2,000	+1
7	4,000	+1
8	8,000	-1

Sample Problem 7-2

*Refer to the unweighted sound levels given in **Sample Problem 7-1**. Assume that these are the eight-octave band levels measured for a single sound source. Determine the overall A-weighted sound pressure level of the source.*

Answer:

First, each of the frequency components must be A-weighted. Next, they can be added, using decibel (logarithmic) addition, in a similar fashion to the previous problem. The A-weighted levels are as follows:

OB	Unweighted Lp (*dB*)	A-weight Adjustment (*dB*)	A-weighted Lp (*dBA*)
1	51	-26	25
2	53	-16	37
3	49	-9	40
4	45	-3	42
5	36	0	36
6	31	+1	32
7	25	+1	26
8	24	-1	23

Now, add the A-weighted frequency components to arrive at an overall A-weighted Lp.

(25 + 37) *dB* = 37 + 0 = 37 *dB*
(40 + 42) *dB* = 42 + 2 = 44 *dB*
(36 + 32) *dB* = 36 + 2 = 38 *dB*
(26 + 23) *dB* = 26 + 2 = 28 *dB*

Adding these results together:

(37 + 44) *dB* = 44 + 1 = 45 *dB*
(38 + 28) *dB* = 38 + 0 = 38 *dB*

Finally, (45 + 38) *dB* = 45 + 1 = 46 *dBA*, the overall A-weighted level.

Because the A-weighting system is a relatively simple metric, and because it accounts for the sensitivity of human hearing as a function of frequency, the Occupational Safety and Health Administration (OSHA) adopted its use for setting noise limits for noisy working environments. In regard to working conditions, specific requirements and guidelines are located in OSHA Standard 29CFR, Part 1910, Subpart G “Occupational Health and Environmental Control”

Many local codes and regulations have adopted guidelines similar to OSHA. Almost all outdoor noise regulations utilize A-weighting as the metric for conformance.

CHAPTER 8: Duct System Acoustics

There are two propagation paths that should concern the design professional. One is the noise which propagates through the duct system, and the other is the airborne noise radiated away from the duct into the surrounding spaces.

There is usually a tradeoff between these two propagation paths. A rigid and well-constructed duct system will contain much of the noise within the system. Usually this is desirable since there are several attenuation mechanisms available for noise propagating down a duct path. Nonmetallic duct systems, or those made of lightweight or poorly constructed materials, will radiate much of the in-duct noise to the surrounding spaces. While this can produce significant in-duct attenuation, it also can be the source for annoying radiated noise problems in the areas through which the duct passes.

In this chapter, we will first examine the primary noise source in HVAC systems: the air handler or fan. After that, we will discuss the methods of natural attenuation in duct systems, followed by a discussion of airflow generated noise. Finally, there is a presentation of noise radiation into and out of duct systems.

8.1 Fan Noise

The most reliable source of fan noise data is the manufacturer of the fan. A fan manufacturer should be able to provide test results from a laboratory testing their fan to Air-Movement and Control Association (AMCA) Standard 300*, Reverberant Room Method for Sound Testing of Fa*ns. AMCA Standard 300 specifies test setup requirements and calculation methods for rating the noise output of fans. AMCA is a trade organization currently comprised of fan, damper, and silencer manufacturers. One of AMCA's goals is to standardize the test methods and requirements for member manufacturers so that all members are employing the same guidelines in rating their product.

Note that a fan manufacturer can not possibly test all of the various combinations of fan size and flow rates, so AMCA has produced AMCA Standard 301, *Methods for Calculating Fan Sound Ratings from Laboratory Test Data*. This standard allows a fan manufacturer to predict sound power levels of a fan at various speeds and for fans of a different size, but geometrically proportional. Therefore, it is important to make sure the test data is for a product identical to that being considered. It is possible that the data for a fan you are considering has had its sound power levels estimated using AMCA 301. Avoid assumptions if possible. Paragraph 5.1, *Setup Categories*, of AMCA 300 allows various test configurations to acquire the appropriate data.

When using manufacturer's information, it is also important to note whether the data is for total or ducted sound power levels. Total sound power includes contributions from the inlet and the discharge ports, as well as noise radiated from the motor, drive train, and equipment casings. We are concerned with the noise propagated into the duct system, and the total sound power levels will probably be at least 3 dB higher than these ducted levels.

Like that of stand-alone fans, sound power levels for packaged air-conditioning equipment also should be obtained from the manufacturer. The manufacturer's data should be from laboratory tests done in accordance with Air-Conditioning and Refrigeration Institute (ARI) Standard 260. ARI is a manufacturer's trade organization including manufacturers of central air-conditioning and commercial refrigeration equipment. ARI Standard 260 uses AMCA Standard 300 as the primary method of obtaining sound power levels, and incorporates some items specific to packaged units. ARI 260 also allows the estimation of sound power levels for untested units.

8.2 Natural Attenuation

Even with no specific provisions for airborne noise control, such as lined duct or silencers, duct systems provide natural attenuation of noise via several mechanisms: duct wall losses, elbow reflections, sound power splits, and terminal end reflections. Together, these natural attenuations can provide significant broad-band noise reduction and may actually eliminate the need for expensive attenuators.

8.2.1 Duct Wall Losses

Whenever sound waves or any pressure fluctuations travel through a confined space such as a duct system, some component of the pressure will be transmitted to the surrounding surface. This will cause the surface to vibrate slightly and thereby dissipate a fraction of the energy from the incident pressure wave. In the case of air flowing through a duct, the energy transmitted to the surface is a function of the shape and size of the duct and the frequency of the duct-borne sound.

These losses are not the same as direct radiation of sound from inside the duct to the surrounding spaces, which is discussed in **Section 8.4.1**. The natural attenuation mechanism of duct wall loss assumes that the duct walls are massive enough to contain most of the duct-borne noise and that the energy transfer is accomplished by transforming incident sound waves into duct surface vibrations, which are then radiated to the surrounding spaces as acoustical energy at a much lower level.

Natural attenuation for round and rectangular duct is shown in **Tables 8.1** and **8.2**. These losses are expressed in terms of decibels per foot. They would appear to be significant in long lengths of duct. However, duct's natural attenuation can never reduce the noise level below the generated noise level of air inside the duct.

Table 8.1
Sound Attenuation in Straight Circular Ducts

Diameter (*inches*)	**Attenuation** (*dB/ft*)						
	Octave Band Center Frequency (*Hz*)						
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**
D <= 7	0.03	0.03	0.05	0.05	0.10	0.10	0.10
7 < D <= 15	0.03	0.03	0.03	0.05	0.07	0.07	0.07
15 < D <= 30	0.02	0.02	0.02	0.03	0.05	0.05	0.05
30 < D <= 60	0.01	0.01	0.01	0.02	0.02	0.02	0.02

Table 8.2
Sound Attenuation in Unlined Rectangular
Sheet Metal Ducts

Duct Size *(in.x in.)*	**Perimeter/ Area** *(ft/ft²)*	**Attenuation** *(dB/ft)*			
		Octave Band Center Frequency *(Hz)*			
		63	**125**	**250**	**>250**
6x6	8.0	0.30	0.20	0.10	0.10
12x12	4.0	0.35	0.20	0.10	0.06
12x24	3.0	0.40	0.20	0.10	0.05
24x24	2.0	0.25	0.20	0.10	0.03
48x48	1.0	0.15	0.10	0.07	0.02
72x72	0.7	0.10	0.10	0.05	0.02

The attenuation values shown in Table 8.2 apply only to rectangular sheet metal ducts that have gages selected according to SMACNA duct construction standards.
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Natural attenuation for flat oval ducts has not been investigated thoroughly, but McGill AirFlow has sponsored enough testing to warrant the following rule of thumb: use the natural attenuation for a round duct having a diameter equal to the flat oval duct's minor axis. For example, to estimate the natural attenuation experienced by a 12x36 flat oval duct, use the attenuation values for a 12 inch diameter duct.

8.2.2 Elbow Reflections

When sound waves traveling in a duct encounter a hard metal elbow, a portion of the sound wave is reflected back in the direction of propagation. **Tables 8.3**, **8.4** and **8.5** provide attenuation values for elbows as a function of elbow diameter (minor axis for flat oval) and frequency. See **Figure 8.1** for a description of each elbow type.

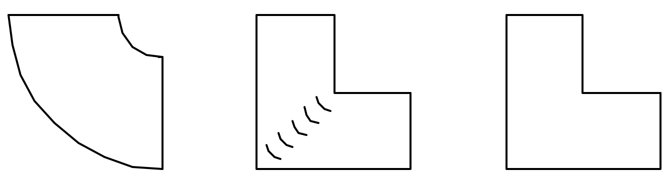

Figure 8.1
Rectangular Elbows
Radiused, mitered with vanes, mitered without vanes (from left to right)

Elbows with bend angles less than 90 have sound power level reductions proportional to the actual bend angle divided by 90. For example, a 45 degree elbow will have approximately one half (45/90) the attenuation of a 90 degree elbow.

Table 8.3
Insertion Loss of Radiused Rectangular Elbows

fw = f x w (f = center frequency, *kHz*, and w = width, *inches*)	**Insertion Loss** *(dB)*
fw < 1.9	0
1.9 <= fw < 3.8	1
3.8 <= fw < 7.5	2
fw > 7.5	3

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Table 8.4
Insertion Loss of Unlined and Lined Rectangular Mitered Elbows with Turning Vanes

fw = f x w (f = center frequency, *kHz*, and w = width, *inches*)	**Insertion Loss** *(dB)*	
	Unlined Elbows	**Lined Elbows**
fw < 1.9	0	0
1.9 <= fw < 3.8	1	1
3.8 <= fw < 7.5	4	4
7.5 <= fw < 15	6	7
fw > 15	4	7

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Table 8.5
Insertion Loss of Unlined and Lined Rectangular Mitered Elbows Without Turning Vanes

fw = f x w (f = center frequency, *kHz*, and w = width, *inches*)	**Insertion Loss** (*dB*)	
	Unlined Elbows	**Lined Elbows**
fw < 1.9	0	0
1.9 <= fw < 3.8	1	1
3.8 <= fw < 7.5	5	6
7.5 <= fw < 15	8	11
15 <= fw < 30	4	10
fw > 30	3	10

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8.2.3 Sound Power Splits

Perhaps the most significant natural attenuation mechanism is sound power splits. In the same way that air will split at a divided-flow fitting, the airborne sound power energy in watts (not sound power level in *dB*) will also be divided. This energy division will be proportional to the cross-sectional area of the straight-through (downstream) flow path of interest, divided by the total of the cross-sectional areas of all downstream flow paths at a particular junction.

For example, if a 12-*inch* common duct has a 5-*inch* branch and an 11-*inch* straight-through, then the sound energy will split as the following ratios:

$$Branch = \frac{A_b}{A_b + A_s} = \frac{\frac{\pi 5^2}{4}}{\frac{\pi 5^2}{4} + \frac{\pi 11^2}{4}} = 0.17$$

$$Main = \frac{A_s}{A_b + A_s} = \frac{\frac{\pi 11^2}{4}}{\frac{\pi 5^2}{4} + \frac{\pi 11^2}{4}} = 0.83$$

To convert this energy split to a sound power level reduction in decibels, use **Equations 8.1** and **8.2**:

$$\Delta Lw_{c-b} = 10 \log [A_b/(A_b + A_s)] \qquad \textbf{Equation 8.1}$$

$$\Delta Lw_{c-s} = 10 \log [A_s/(A_b + A_s)] \qquad \textbf{Equation 8.2}$$

where:

ΔLw_{c-b} = Sound power level reduction, common (upstream) to reference branch (*dB*)

ΔLw_{c-s} = Sound power level reduction, common (upstream) to straight-through (downstream) (*dB*)

A_b = Branch cross-sectional area (ft^2)

A_s = Straight-through cross-sectional area (downstream and nonreference branches) (ft^2)

For the above example, the sound power level reduction for the branch path would be 10 log 0.17 (or 8 *dB*). The straight-through path would have a sound power level reduction of 10 log 0.83 (or 1 *dB*). Sound level reductions due to power splits apply to all frequencies. **Table 8.6** provides a quick reference of sound power level reductions for various area ratios.

Table 8.6
Duct Branch Sound Power Reduction

$A_b / (A_b + A_s)$	ΔLw	$A_b / (A_b + A_s)$	ΔLw
1.00	0	0.100	10
0.80	1	0.080	11
0.63	2	0.063	12
0.50	3	0.050	13
0.40	4	0.040	14
0.32	5	0.032	15
0.25	6	0.025	16
0.20	7	0.020	17
0.16	8	0.016	18
0.12	9	0.012	19

8.2.4 End Reflections

When there is a significant change of area at the termination of a duct run, some of the low-frequency acoustical energy is reflected back into the duct, in the direction of propagation, due to the change in acoustical impedance of the air stream. That is, the multitude of air molecules in a much larger space (compared to a duct) cannot transfer energy as quickly as it is being delivered by a duct. This situation occurs when a open-ended duct discharges air directly into a large (large compared to the duct) room.

The end reflection effect is reduced or eliminated when a diffuser, register, or other terminal device is placed at the duct opening. When flexible duct is used as a final run to a terminal, the end reflection effect is essentially negated. Given current design practices, this mechanism will not provide significant attenuation for many systems. However, where low frequency noise is anticipated to be a problem, special designs which make use of the end reflection phenomenon can provide a cost-effective solution.

Tables 8.7 and **8.8** provide expected attenuation for end reflection as a function of frequency and duct diameter. These values assume the discharge is a open-ended duct (no diffuser), and that there are at least 3 to 5 diameters of straight rigid duct upstream of the discharge. If the duct is rectangular, use **Equation 8.3** to calculate the diameter to use in the tables.

$$D = \sqrt{4A/p}$$

Equation 8.3

Though some low aspect diffusers agree reasonably using **Equation 8.3** and **Tables 8.7** and **8.8**, it is recommended that when diffusers are placed at the duct termination, 6 dB be subtracted from the values shown. After the adjustment, any resulting negative values should be adjusted to zero.

Table 8.7
Duct End Reflection Loss -- Duct Terminated in Free Space

Diameter (*inches*)	**End Reflection Loss** (*dB*)					
	Octave Band Center Frequency (*Hz*)					
	63	**125**	**250**	**500**	**1000**	**2000**
6	20	14	9	5	2	1
8	18	12	7	3	1	0
10	16	11	6	2	1	0
12	14	9	5	2	1	0
16	12	7	3	1	0	0
20	10	6	2	1	0	0
24	9	5	2	0	0	0
28	8	4	1	0	0	0
32	7	3	1	0	0	0
36	6	3	1	0	0	0
48	5	2	1	0	0	0
72	3	1	0	0	0	0

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Table 8.8
Duct End Reflection Loss -- Duct Terminated Flush with Wall

Diameter (*inches*)	**End Reflection Loss** (*dB*) **Octave Band Center Frequency** (*Hz*)				
	63	**125**	**250**	**500**	**1000**
6	18	13	8	4	1
8	16	11	6	2	1
10	14	9	5	2	1
12	13	8	4	1	0
16	10	6	2	1	0
20	9	5	2	1	0
24	8	4	1	0	0
28	7	3	1	0	0
32	6	2	1	0	0
36	5	2	1	0	0
48	4	1	0	0	0
72	2	1	0	0	0

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8.3 Airflow-Generated Noise

Noise is generated by air flowing over duct surfaces. Although this noise may not always be audible, especially in areas where the fan noise is substantial, there are several conditions that will increase the levels of airflow-generated noise. The following situations should be avoided to minimize generated noise: (1) high velocities (>2,000 *fpm* for rectangular duct, >3,000 *fpm* for round and flat oval – except initial velocities as given in **Table 2.1**), (2) airflow turbulence, (3) obstructions in the airstream (tie rods, extractors, etc.), and (4) abrupt-turn fittings, especially those without turning vanes.

Duct and all types of fittings can create airflow-generated noise. **Appendix A.9.10** provides a summary of the generated noise properties of duct and concludes that it is directly proportional to both air velocity and duct diameter. In most systems, where both velocity and diameter have small magnitudes at discharge locations, the levels of duct self-noise will not contribute to the overall in-duct sound power level.

The reference in Appendix **A.9.10** contains a procedure for estimating the noise generation of several types of fittings. If the airflow-generated noise of a fitting is at least 10 *dB* below the residual section's sound power level, it will not contribute to the overall sound power level and can be ignored.

It is advisable to calculate the airflow-generated noise levels for all sections, regardless of their location relative to the fan. These levels should be compared to the residual fan sound power levels determined from an acoustical analysis. If the levels are within 10 *dB*, they will contribute to the overall noise level and should be added to the residual section's sound power levels, using simplified decibel addition as shown in **Table 7.3**. Sample Problem 8-1 shows the proper procedures.

Generally, as long as velocities are kept at reasonable levels and care is taken in the selection of optimum fittings, the airflow-generated noise of normal fitting components can be ignored. As we shall see later, this is not always the case for duct silencers.

Sample Problem 8-1

---

*For the system shown in Figure 8.1, calculate the sound power levels at Terminals A, B, C, D, and E, using the fan noise levels from Section 1001 shown in **Table 8.9**. Take into account natural attenuation and fitting generated noise.*

Answer: **Tables 8.9** to **8.12** show the resultant natural attenuation for each section up to the diffuser. **Tables 8.9** to **8.12** also account for the airflow-generated noise levels determined in accordance with **Appendix A.9.2**. The resulting sound power levels for each outlet are determined in the final step (row) of each table.

Fan sound power levels (*Lw*) are given in **Table 8.9** and are used for section 1001, which is located immediately downstream of the fan. There is no power split attenuation for section 1001; however, this section does have natural attenuation in the duct and elbow.

The attenuation of each section is subtracted from the residual sound power level of the previous (upstream) section, then the generated noise level is added using the method shown in **Table 7.3**. In this way, the sound power can be determined at any location in the system. Note that the residual sound power entering section 1003 is the attenuated level from section 1001. Also, the end reflection correction values for each terminal is 6 *dB* less than those found in **Table 8.8**, due to the diffusers.

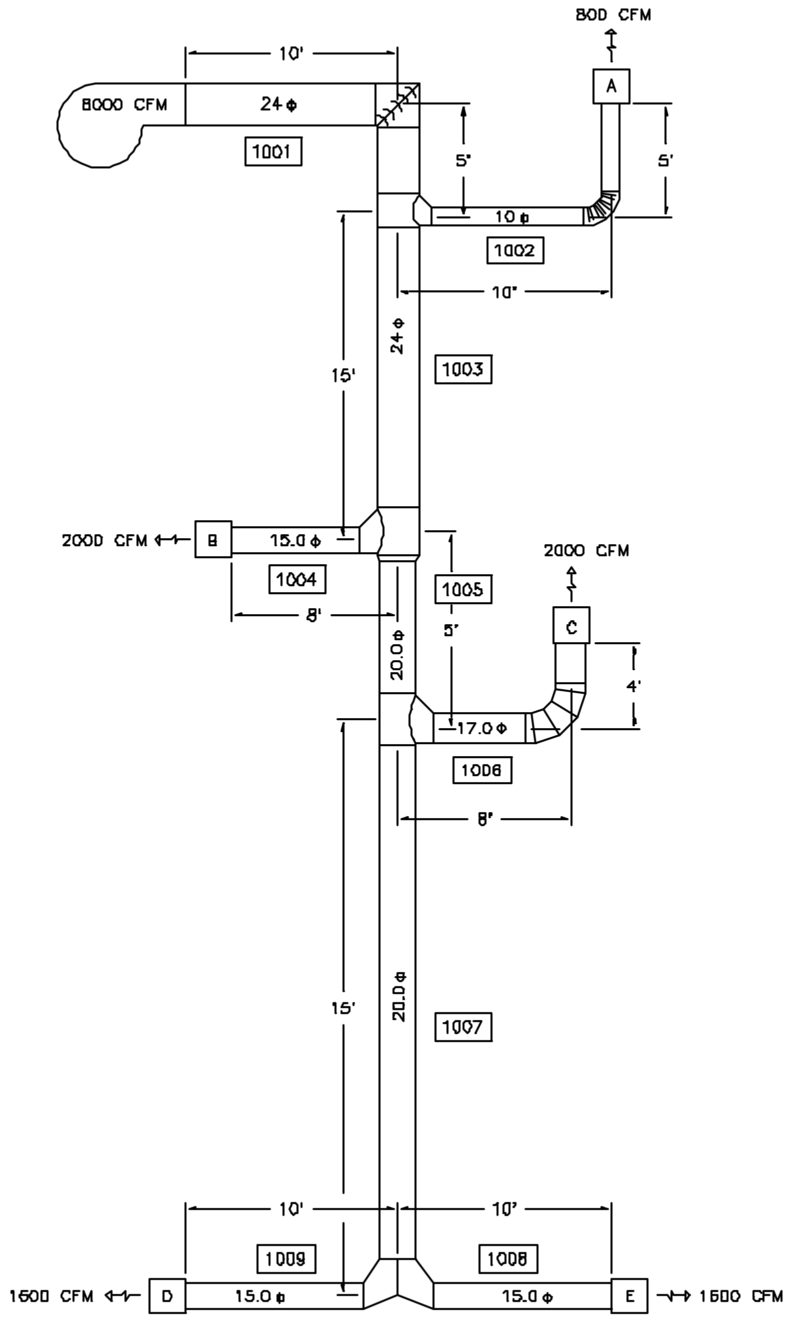

Figure 8.2
Sample Problem 8-1

Table 8.9
Single-Wall Natural Attenuation Acoustical Analysis, Outlet A

Outlet A	**Octave Band/Center Frequency** (*Hz*)							
Comment	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Section 1001								
1. Fan *Lw* values	89	89	90	83	78	72	68	63
2. Duct, 24 *inches*, *L*=15 *ft*	0	0	0	0	1	1	1	1
3. Mitered 90E elbow with Vanes, 24 in. diameter	0	1	2	3	3	3	3	3
Remaining	89	88	88	80	74	68	64	59
4. Duct *GNL* at 2546 *fpm*	67	62	51	47	46	43	35	35
5. Elbow *GNL* at 2546 *fpm*	63	64	63	59	53	43	30	13
Resultant (Section 1001)	89	88	88	80	74	68	64	59
Section 1002								
6. LoLoss™ tee, branch 10 *inches* (sound power split)	8	8	8	8	8	8	8	8
Remaining	81	80	80	72	66	60	56	51
7. LoLoss™ tee, Branch *GNL*	40	39	37	35	32	28	23	18
Remaining	81	80	80	72	66	60	56	51
8. Duct, 10 *inches* *L*=15 *ft*	0	0	0	1	1	1	1	1
9. Pleated 1.5 CLR, 90E elbow, 10 *inches* diameter	0	0	1	2	3	3	3	3
Remaining	81	80	79	69	62	56	52	47
10. Duct *GNL* at 1467 *fpm*	35	30	28	27	26	24	20	20
11. Elbow *GNL* at 1467 *fpm*	0	0	0	0	0	0	0	0
Resultant 2 (Section 1002) Outlet A, *Lw* (*dB*)	81	80	79	69	62	56	52	47

Table 8.10
Single-Wall Natural Attenuation Acoustical Analysis, Outlet B

Outlet B Comment	Octave Band Center Frequency (*Hz*) 63	125	250	500	1000	2000	4000	8000
Section 1003 (resultant of 1001)								
1. Entering Sound Power	89	88	88	80	74	68	64	59
2. Duct, 24 *inches*, *L*=15 *ft*	0	0	0	0	1	1	1	1
Mitered 90E Elbow w/Vanes 24 *inch*-diameter	0	1	2	3	3	3	3	3
3. LoLoss™ Tee, Main, 24 *inches* (sound power split)	1	1	1	1	1	1	1	1
Remaining	88	86	85	76	69	63	59	54
4. LoLoss™ Tee Main *GNL*	48	39	37	35	32	28	23	18
5. Duct *GNL* at 2292 *fpm*	64	59	49	45	44	41	32	32
Resultant 3 (Section 1003)	88	86	85	76	69	63	59	54
Section 1004								
1 Duct, 15 *inches*, *L*=8 *ft*	0	0	0	0	1	1	1	1
2 LoLoss™ Tee Branch, 15 *inches* (sound power split)	4	4	4	4	4	4	4	4
Remaining	84	82	81	72	64	58	54	49
3 LoLoss™ Tee, Branch *GNL*	40	39	37	34	30	25	20	14
4 Duct *GNL* at 1629 *fpm*	43	39	37	33	35	33	23	23
Resultant 4 (Section 1004) Outlet B, *Lw* (*dB*)	84	82	81	72	64	58	54	49

Table 8.11
Single-Wall Natural Attenuation Acoustical Analysis, Outlet C

Outlet C	**Octave Band Center Frequency** (*Hz*)							
Comment	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Section 1005								
1. Entering Sound Power resultant 3 (Section 1003)	88	86	85	76	69	63	59	54
2. Duct, 20 *inches*, *L* = 5 *ft*	0	0	0	0	0	0	0	0
3. LoLoss™ Tee, Main (sound power split)	2	2	2	2	2	2	2	2
Remaining	86	84	83	74	67	61	57	52
4. LoLoss™ Tee, Main *GNL*	44	43	41	38	34	29	24	18
5. Duct *GNL* at 2384 *fpm*	58	55	50	45	48	46	33	33
Resultant 5 (Section 1005)	86	84	83	74	67	61	57	52
Section 1006								
1. Duct, 17 *inches*, *L* = 12 *ft*	0	0	0	0	1	1	1	1
2. 5-gore 90E elbow 1.5 CLR	0	1	2	3	3	3	3	3
3. LoLoss™ Tee, Branch 17 *inches* (sound power split)	4	4	4	4	4	4	4	4
Remaining	82	79	77	67	59	53	49	44
4. LoLoss™ Tee, Branch GNL	42	40	38	35	31	27	21	15
5. Duct *GNL* at 1269 *fpm*	34	31	30	26	29	28	18	18
6. Elbow *GNL* at 1269 *fpm*	0	0	0	0	0	0	0	0
Resultant 6 (Section 1006) Outlet C, *Lw* (*dB*)	82	79	77	67	59	53	49	44

Table 8.12 Single-Wall Natural Attenuation Acoustical Analysis, Outlet D/E								
Outlet D/E	**Octave Band Center Frequency** (*Hz*)							
Comment	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Section 1007								
1. Entering Sound Power, Resultant 5 (Section 1005)	86	84	83	74	67	61	57	52
2. Duct 20 *inches*, *L* = 15 *ft*	0	0	0	0	1	1	1	1
3. LoLoss™ Tee, Main (sound power split)	2	2	2	2	2	2	2	2
Remaining	84	82	81	72	64	58	54	49
4. LoLoss™ Tee, Main *GNL*	45	43	41	38	34	30	24	18
5. Duct *GNL* at 1467 *fpm*	41	39	35	31	33	31	21	21
Resultant 7 (Section 1007)	84	82	81	72	64	58	54	49
Sections 1008 and 1009								
1. Duct 15 *inches*, *L* = 10 *ft*	0	0	0	1	1	1	1	1
2. Vee fitting branches, 15 *inches*	3	3	3	3	3	3	3	3
Remaining	81	79	78	68	60	54	50	45
3. Vee Fitting Branches, *GNL*	56	52	46	40	33	25	16	6
4. Duct *GNL* at 1304 *fpm*	33	31	29	27	28	26	18	18
Resultant 7 (Sections 1008 and 1009) Outlet D/E, *Lw* (*dB*)	81	79	78	68	60	54	50	45

As you can see, the natural attenuation in a duct system can be significant, and it is due to the natural attenuation of the duct, elbows, power splits (branch fittings), and end reflections. Sometimes, a potential noise problem can be alleviated by re-routing the duct.

Chapter 10 will take the analysis of **Sample Problem 8.1** one step further by comparing the natural attenuated noise level to the desired noise level criteria for each outlet to determine the amount of additional attenuation required via lined duct and fittings and silencers.

8.4 Radiated Duct Noise

8.4.1 Break-Out Noise

In addition to the fan noise and airflow-generated noise propagated within a duct system, another concern is the noise that radiates from HVAC duct to the surrounding spaces. This is often referred to as break-out noise, and it can be a critical design parameter whenever duct passes through or over an acoustically sensitive area. A radiated noise problem is likely to exist if the local in-duct sound power level at any frequency, minus the duct transmission loss, exceeds or is within 3 to 5 *dB* of the noise criteria (*NC*) level of the critical space. Noise criteria will be discussed in a subsequent section. Use **Tables 8.13** to **8.16** to determine the transmission loss of ducts when noise is from inside the duct and radiating outward.

Table 8.13
Break-Out Transmission Loss of Single-Wall Rectangular Duct

Duct Size *(in x in)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)*							
		63	125	250	500	1000	2000	4000	8000
12 x 12	24	21	24	27	30	33	36	41	45
12 x 24	24	19	22	25	28	31	35	41	45
12 x 48	22	19	22	25	28	31	37	43	45
24 x 24	22	20	23	26	29	32	37	43	45
24 x 48	20	20	23	26	29	31	39	45	45
48 x 48	18	21	24	27	30	35	41	45	45
48 x 96	18	19	22	25	29	35	41	45	45

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Table 8.14
Break-Out Transmission Loss of Double-Wall Rectangular Duct[1,2,3]

Duct Size *(in x in)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)*							
		63	125	250	500	1000	2000	4000	8000
24 x 24	22	20	23	26	29	35	42	44	45

1.Data based on McGill AirFlow Corporation testing and ASHRAE tables.
2.Double-wall duct is Rectangular-k27® with 1-1/2 inch insulation sandwiched between a 24 gage perforated inner liner and a 22 gage outer solid shell (total compressed thickness is about 1-3/8 inches); perforated inner liner is 23 percent open area.
3. data from 5 ft length duct

Table 8.15
Break-Out Transmission Loss of Single-Wall Round Duct

Diameter *(inches)*	Length *(feet)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)* 63	125	250	500	1000	2000	4000
Long Seam Ducts									
8	15	26	>45	(53)	55	52	44	35	34
14	15	24	>50	60	54	36	34	31	25
22	15	22	>47	53	37	33	33	27	25
32	15	22	(51)	46	26	26	24	22	38
Spiral Wound Ducts									
8	10	26	>48	>64	>75	72	56	56	46
14	10	26	>43	>53	55	33	34	35	25
26	10	24	>45	50	26	26	25	22	36
26	10	16	>48	53	36	32	32	28	41
32	10	22	>43	42	28	25	26	24	40

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Table 8.16
Break-Out Transmission Loss of Single-Wall Flat Oval Duct

Duct Size *(in x in)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)* 63	125	250	500	1000	2000	4000	8000
6 x 12	24	31	34	37	40	43	--	--	--
6 x 24	24	24	27	30	33	36	--	--	--
12 x 24	24	28	31	34	37	--	--	--	--
12 x 48	22	23	26	29	32	--	--	--	--
24 x 48	22	27	30	33	--	--	--	--	--
24 x 96	20	22	25	28	--	--	--	--	--
48 x 96	18	28	31	--	--	--	--	--	--

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Note that in **Tables 8.13** and **8.16**, the data are for duct lengths of 20 feet, but the values may be used for the cross-section shown regardless of length. In **Table 8.15**, if the transmission loss listed is preceded by the “>” sign, the actual transmission loss may be higher than shown. This is due to a limitation in the laboratory testing facilities that acquired the data. Data in parenthesis has a greater uncertainty than usual. The references in **Appendix A.9.2** and **A.9.10** give additional information for estimating break-out noise.

8.4.2 Break-In Noise

Just as noise from inside the duct transmits outward, ambient noise can be transmitted into a duct. This is known as break-in noise. When the in-duct noise is 10 dB or more than the break-in noise, the break-in noise can be ignored. However, at locations where fan noise and aerodynamically generated noise are minimal, significant levels of break-in noise can propagate inside the duct and radiate to surrounding spaces.

The ability of ducts to resist break-in noise is quantified as break-in transmission loss. Break-in transmission loss data is located in **Tables 8.17** to **8.19**.

Note that in **Tables 8.17** and **8.19**, the data are for duct lengths of 20 feet, but the values may be used for the cross-section shown regardless of length. In **Table 8.18**, if the transmission loss listed is preceded by the ">" sign, the actual transmission loss may be higher than shown. This is due to a limitation in the laboratory testing facilities that acquired the data. Data in parenthesis has a greater uncertainty than usual. The references in **Appendix A.9.2** and **A.9.10** give additional information for estimating break-in noise.

Table 8.17
Break-In Transmission Loss of Single-Wall Rectangular Duct

Duct Size *(in x in)*	**Gage**	**Transmission Loss** *(dB)*							
		Octave Band Center Frequency *(Hz)*							
		63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
12 x 12	24	16	16	16	25	30	33	38	42
12 x 24	24	15	15	17	25	28	32	38	42
12 x 48	22	14	14	22	25	28	34	40	42
24 x 24	22	13	13	21	26	29	34	40	42
24 x 48	20	12	15	23	26	28	36	42	42
48 x 48	18	10	19	24	27	32	38	42	42
48 x 96	18	11	19	22	26	32	38	42	42

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Table 8.18
Break-in Transmission Loss of Single-Wall Round Duct

Diameter *(inches)*	Length *(feet)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)*						
			63	125	250	500	1000	2000	4000
Long Seam Ducts									
8	15	26	>17	(31)	39	42	41	32	31
14	15	24	>27	43	43	31	31	28	22
22	15	22	>28	40	30	30	30	24	22
32	15	22	(35)	36	23	23	21	19	35
Spiral Wound Ducts									
8	10	26	>20	>42	>59	>62	53	43	26
14	10	26	>20	>36	44	28	31	32	22
26	10	24	>27	38	20	23	22	19	33
26	10	16	>30	>41	30	29	29	25	38
32	10	22	>27	32	25	22	23	21	37

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Table 8.19
Break-in Transmission Loss of Single-Wall Flat Oval Duct

Duct Size *(in x in)*	Gage	Transmission Loss *(dB)* Octave Band Center Frequency *(Hz)*							
		63	125	250	500	1000	2000	4000	8000
12 x 6	24	18	18	22	31	40	--	--	--
24 x 6	24	17	17	18	30	33	--	--	--
24 x 12	24	15	16	25	34	--	--	--	--
48 x 12	22	14	14	26	29	--	--	--	--
48 x 24	22	12	21	30	--	--	--	--	--
96 x 24	20	11	22	25	--	--	--	--	--
96 x 48	18	19	28	--	--	--	--	--	--

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8.4.3 Nonmetal Ducts

A special word of caution is in order when nonmetal ducts are being considered, especially in or near acoustically sensitive areas. These products, such as fiberglass duct , duct board, or flexible duct, have negligible mass and therefore will have very little transmission loss. A substantial amount of the noise inside a duct of this type will be immediately radiated to the surrounding spaces. Although manufacturers of these products claim high in-duct attenuation, due to the absorptive wall surfaces, very few publish data on break-in or break-out transmission loss.

CHAPTER 9: Room Acoustics

In the previous section, we discussed the various sources of duct-borne noise and the natural attenuation mechanisms of duct systems. If the fan sound power level and the duct system size and configuration are known, one can calculate the residual sound power level at any outlet in the system. The next step is to determine the sound pressure level that this sound power will generate in the room or area that the duct is serving, and whether this is in conformance with acceptable using design criteria.

9.1 Air Terminal Noise

Since most duct runs will terminate with some type of register or diffuser, it is first necessary to calculate the generated noise caused by the air flowing across the terminal device. Acoustical and airflow data is usually obtained in accordance with ASHRAE Standard 70, *Method of Testing for Rating the Performance of Air Outlets and Inlets*. The data should present noise levels as a function of frequency and air volume or velocity. These levels should be treated as generated sound power levels and added to the residual sound power levels at the outlet, using decibel addition.

If the diffuser sound power is within 10 *dB* of the residual sound power level in the duct, it will increase the sound power level being emitted in the space. Additionally, poor flow conditions at the diffuser or register entrance can substantially increase the generated noise levels.

If the velocity profile is not uniform across the diffuser entrance, sound power levels may increase by as much as 12 *dB* above the manufacturer's predicted levels. This situation can often be corrected with flow straighteners. If flexible duct is used as a final connection to the terminal device, it should be run as straight as possible. Any offset greater than one fourth of the diffuser collar diameter over a length of twice the diffuser diameter will increase the noise levels by as much as 15 *dB* greater than the manufacturer's predicted levels.

9.2 ASHRAE Room Effect Equation

ASHRAE endorses a simple procedure for determining the sound pressure levels that will result from sound power emitted at the terminal of an HVAC duct. The equations are for normal rooms, which mean that there is assumed to be a certain amount of sound-absorbing surfaces and furnishings in the space. It is possible that extremely hard or soft spaces will have a slightly higher or lower sound pressure level, respectively; however, ASHRAE claims that the method will be accurate to within "2 *dB*.

Equation 9.1 is used to predict the sound pressure level at any distance (*r*) from a terminal outlet. The sound pressure level in a room is a function of frequency, room volume and the distance from the terminal outlet to a specified point in the room. The calculation must be repeated for each octave band center frequency (*f*).

$$Lp = Lw - 5\log(V) - 3\log(f) - 10\log(r) + 25dB \qquad \textbf{Equation 9.1}$$

where:

Lp = Room sound pressure level (dB re 20×10^{-5} Pascals)

Lw = Source sound power level (dB re 1×10^{-12} watts)

V = Room volume (ft^3)

f = Octave band center frequency (Hz)

r = Reference distance (ft)

For multiple terminals, the sound pressure level at the reference point is calculated for each terminal, using the appropriate Lw for that outlet and volume of the room. The resultant levels are then added using decibel corrections presented in **Appendix 9.10**.

Sample Problem 9-1

*A 14-ft H 20-ft H 8-ft-high room is supplied by two diffusers. The diffusers are mounted on the longitudinal centerline of the room, 5 feet from either end wall (10 feet between diffusers). See **Figures 9.1** and **9.2** for the plan and elevation views, respectively. The residual sound power level in the duct just upstream of the diffusers and the regenerated noise level of the diffusers are shown in **Table 9.1**. What will be the octave band sound pressure levels at a height of 5 feet from the floor, on the longitudinal centerline, 8 feet from the end of the room?*

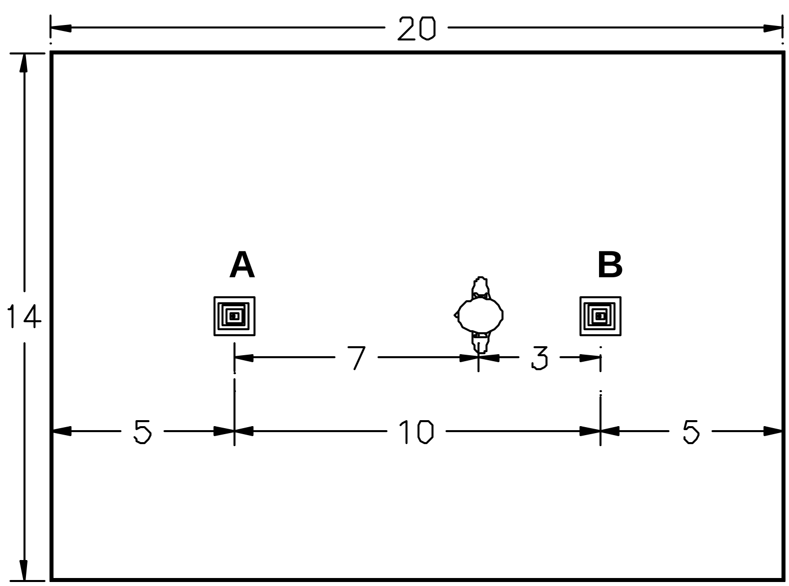

Figure 9.1
Plan View of Room for Sample Problem 9-1

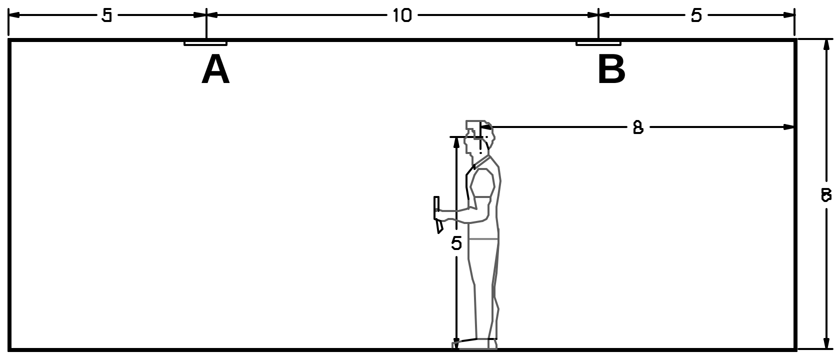

Figure 9.2
Elevation View of Room for Sample Problem 9-1

Table 9.1
Sample Problem 9-1 Noise Levels

Frequency (*Hz*)	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Residual Duct *Lw*	35	38	46	31	46	41	33	26
Diffuser *Lw*	33	35	36	36	35	33	27	18

Answer: The sound pressure from each terminal is calculated, using **Equation 9.1**, and added together. The terminal closest to the reference location is *A* and the other is *B*.

The reference location is a point 5 *feet* above the floor (3 *feet* below the ceiling) and, in plan view, located midway between the long walls, 3 *feet* from one terminal and 7 *feet* from the other. The reference distance from either terminal can be calculated from simple geometry:

$$r\,(A) = \sqrt{3^2 + 3^2} = 4.2 \quad feet$$

$$r\,(B) = \sqrt{3^2 + 7^2} = 7.6 \quad feet$$

The distance term [*10 log(r)*] is: Terminal [A]: 10 log (4.2) = 6.2 = 6 *dB*
Terminal [B]: 10 log (7.6) = 8.8 = 9 *dB*

The room volume is 14 H 20 H 8 = 2,240 ft^3, therefore, The volume term [*5 log(V)*] is 5 log (2,240) = 16.7 = 17 *dB*

To determine the overall source sound power entering the room, it is necessary to add (decibel addition) the residual duct sound power level and the generated sound power level of the terminal devices. The resultant sound power level from each terminal is then adjusted by the factors in **Equation 9.1** to determine the sound pressure level at the reference location.

First, calculate the sound pressure level resulting from the noise exiting Terminal A. This step is summarized in **Table 9.2**.

Table 9.2
Sound Pressure Levels Due To Noise From Terminal A

Description	Data (dB) Frequency (Hz)							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Residual Duct *Lw*	35	38	46	31	46	41	33	26
Diffuser *Lw*	33	35	36	36	35	33	27	18
Resultant *Lw*	37	40	46	37	46	42	34	27
Room Effect:								
-*5 log (V)*	-17	-17	-17	-17	-17	-17	-17	-17
-*3 log (f)*	-5	-6	-7	-8	-9	-10	-11	-12
-*10 log (r)*	-6	-6	-6	-6	-6	-6	-6	-6
+25 *dB*	+25	+25	+25	+25	+25	+25	+25	+25
Lp due to A (use **Equation 9.1**)	34	36	41	31	39	34	25	17

Next, calculate the sound pressure level resulting from the noise exiting Terminal B. This step is summarized in **Table 9.3**.

Table 9.3
Sound Pressure Levels Due To Noise From Terminal B

Description	Data (dB)							
	Frequency (Hz)							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Residual Duct *Lw*	35	38	46	31	46	41	33	26
Diffuser *Lw*	33	35	36	36	35	33	27	18
Resultant *Lw*	37	40	46	37	46	42	34	27
Room Effect:								
-5 log (V)	-17	-17	-17	-17	-17	-17	-17	-17
-3 log (f)	-5	-6	-7	-8	-9	-10	-11	-12
-10 log (r)	-9	-9	-9	-9	-9	-9	-9	-9
+25 *dB*	+25	+25	+25	+25	+25	+25	+25	+25
Lp due to B (use **Equation 9.1**)	31	33	38	28	36	31	22	14

The resultant sound pressure at the reference point, due to the sound power level from both terminals, is the logarithmic sum of the two levels, *Lp* (resultant). (See **Table 7.3** for decibel addition rules). The result is summarized in **Table 9.4**.

Table 9.4
Sound Pressure Levels Due To Noise From Terminals A and B

Description	Data (dB)							
	Frequency (Hz)							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Lp due to **A**	34	36	41	31	39	34	25	17
Lp due to **B**	31	33	38	28	36	31	22	14
Lp Resultant	**36**	**38**	**43**	**33**	**41**	**36**	**27**	**19**

9.3 Design Criteria

Now that we are able to calculate fan sound power, duct attenuations, generated diffuser noise, and convert sound power level to a sound pressure level in a room, we must determine whether the sound pressure level is suitable for the noise criteria in the space. For this purpose, various criteria have been introduced in an attempt to relate a spectrum of noise to various occupancy situations.

9.3.1 NC Rating Method

In the past, the most commonly accepted criteria were the noise criteria (*NC*) curves, which were introduced in 1957. These curves are shown in **Figure 9.3**.

Note that all NC curves slope downward, left to right, indicating that the allowable sound pressure levels are higher in the lower frequencies. This is consistent with the auditory sensitivity of the human ear (see **Section 7.5**).

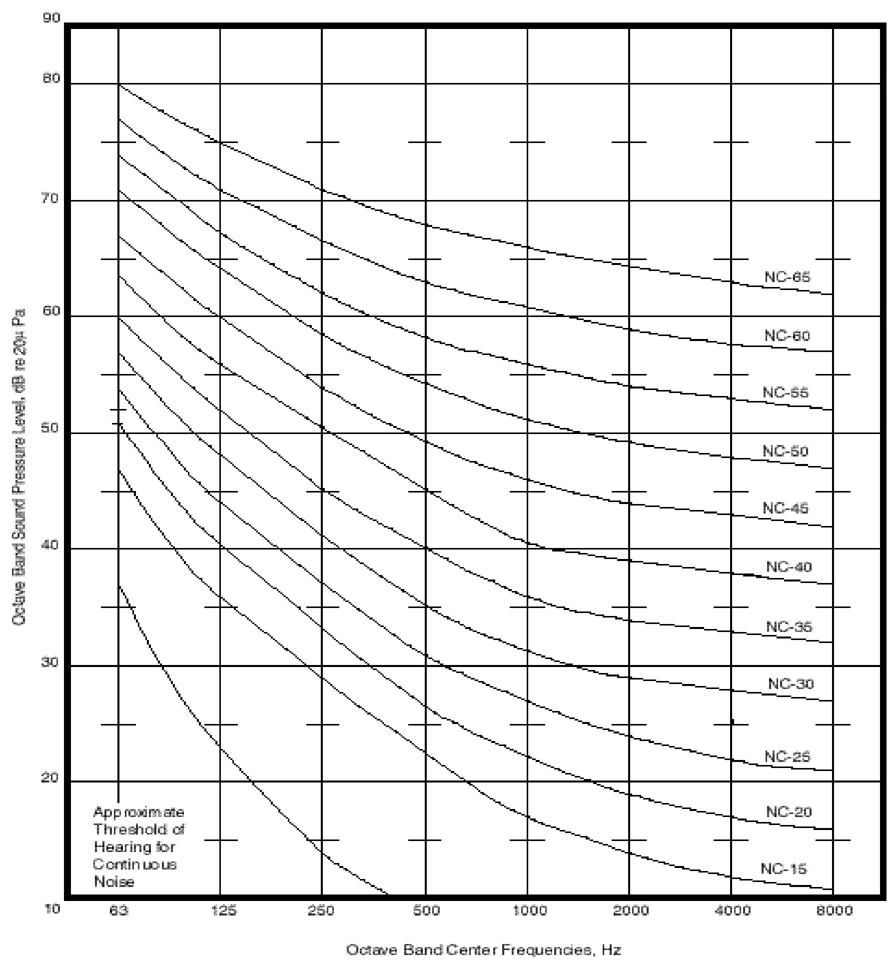

Figure 9.3
Noise Criteria Curves
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When determining the noise criteria of a space, the actual sound pressure levels are plotted over the standard curves. The *NC* rating of a particular noise source is generally taken as the value of the lowest *NC* curve that is closest to the highest point of the actual noise spectrum. *NC* ratings should always be in increments of 5, though interpolation between curves is often done. Usage of *NC* curves often results in an unbalanced noise spectrum since the highest sound pressure dictates the *NC* rating, and no requirement is specified for the relationship between nearby frequencies.

As an example of how to use the *NC* rating method, the calculated sound pressure levels of the office background noise from **Sample Problem 9-1** is plotted in **Figure 9.4**. The lowest *NC* curve that is closest to the highest noise spectrum sound level is at 1,000 *Hz*. At that point the spectrum touches the *NC-40* curve. Therefore, the room would have an *NC-40* rating.

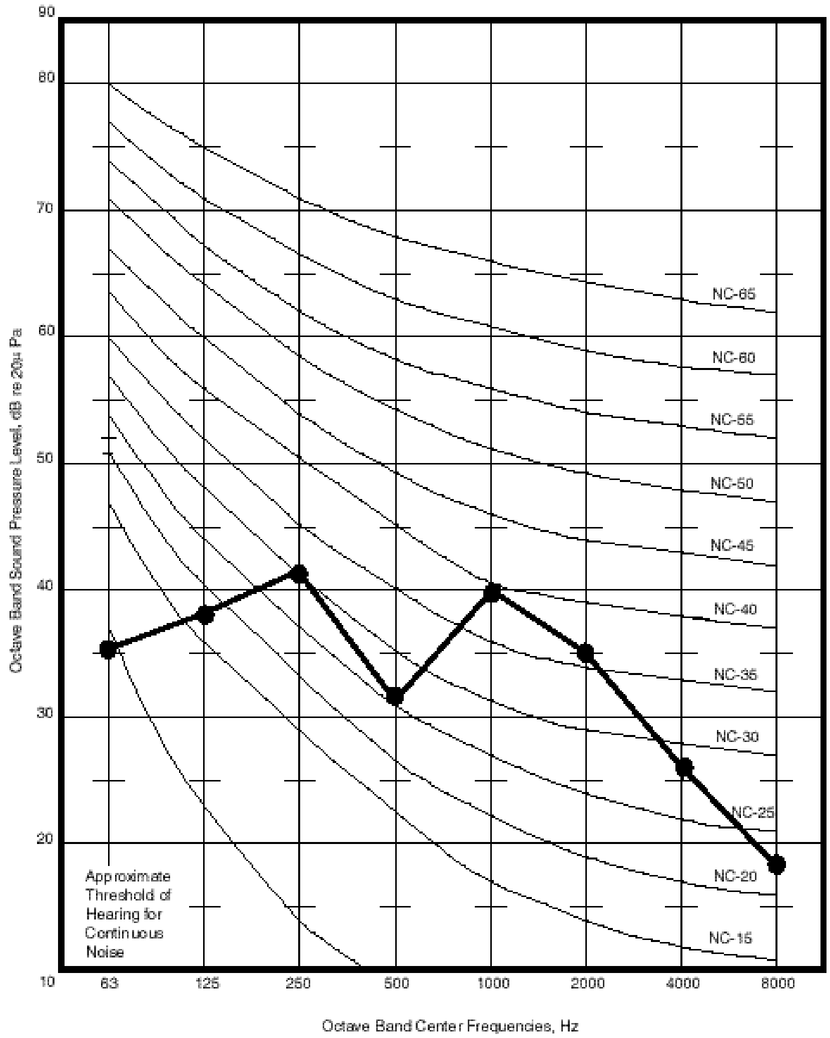

Figure 9.4
NC Rating Determination for Office Background Noise

The *NC* curves are actually plots of allowable sound pressures as a function of frequency. See **Table 9.5** for the sound pressure data that make up each *NC* curve. **Table 9.5** may be used in place of plotting measured sound pressure data on an *NC* curve. A determination can be made of the room *NC* rating using **Table 9.5** by finding the lowest NC criterion that has all of its sound pressure levels greater than the spectrum data, beginning with the NC-15 and working up the table.

Table 9.5
Sound Pressure Level Data for *NC* curves

***NC* Criterion**	**Sound Pressure Data Point on Curve (dB)**							
	Frequency (Hz)							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
NC-65	80	75	71	68	66	64	63	62
NC -60	77	71	67	63	61	59	58	57
NC -55	74	67	62	58	56	54	53	52
NC -50	71	64	58	54	51	49	48	47
NC -45	67	60	54	49	46	44	43	42
NC -40	64	56	50	45	41	39	38	37
NC -35	60	52	45	40	36	34	33	32
NC -30	57	48	41	35	31	29	28	27
NC -25	54	44	37	31	27	24	22	21
NC -20	51	40	33	26	22	19	17	16
NC -15	47	36	29	22	17	14	12	11

Sample Problem 9-2

The following sound pressure levels were recorded by a sound level meter in a classroom.

63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
58	53	53	44	40	36	37	25

*Using **Table 9.5**, what NC criterion does the room meet?*

Answer: Determine NC criterion for each data point.

63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
NC-35	NC-40	**NC-45**	NC-40	NC-40	NC-40	NC-40	NC-30

From the table, we see that the NC rating is dictated by the data in the 250 Hz band. The answer is **NC-45**.

9.3.2. RC Rating Method

Although the *NC* rating method is still widely used, the room criteria (*RC*) rating method is preferred and is often required in acoustical specifications because it reveals tonal components that go unaccounted for in the *NC* rating method. The *RC* rating method also accounts for the acoustical energy produced in the low frequencies (16 and 31.5 *Hz*), which may result in perceptible vibration of building components. **Figure 9.5** is a plot of the family of *RC* curves.

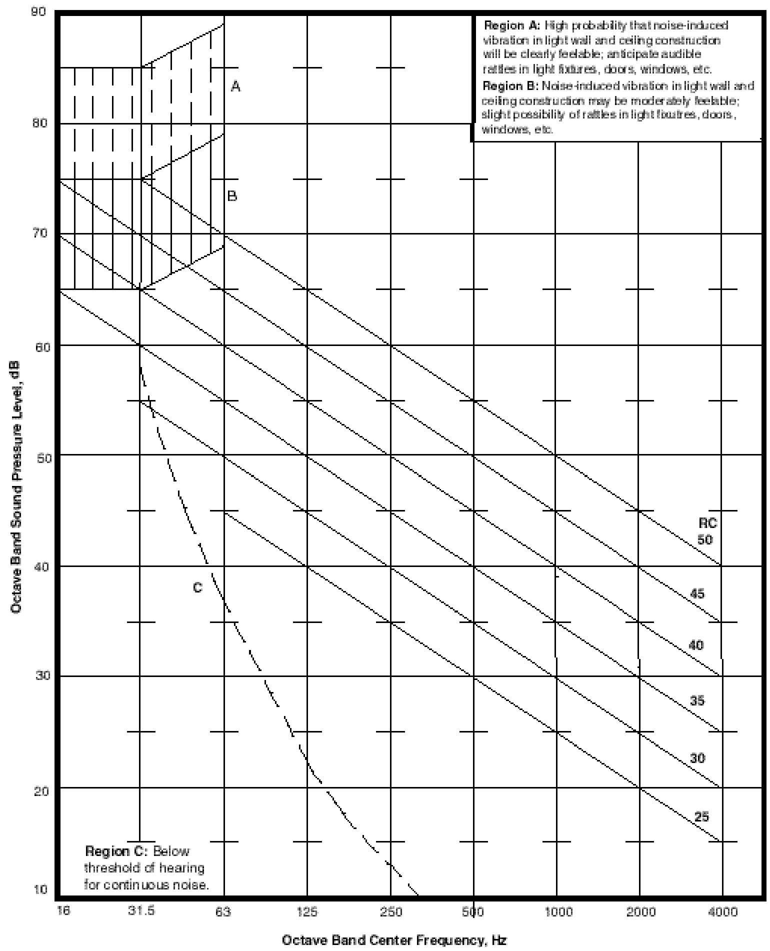

Figure 9.5
Room Criteria Curves

RC rating procedures are more involved than the *NC* rating, but they are fairly simple to follow. The following procedure is used to obtain RC ratings.

RC Rating Method Procedure

1. Plot the sound pressure level data on the *RC* curve.

2. Calculate the arithmetic average of the sound pressure level data in the 500, 1000, and 2000 *Hz* frequencies. Round to nearest dB. This is the numerical RC rating.

3. Draw a line with a slope of -5 *dB* in the frequency range of 31.5 to 4,000 *Hz*, passing through 1,000 *Hz* at the value calculated in step 2. This is the reference curve.

4. For the frequency range from 31.5 to 500 *Hz*, draw a line 5 *dB* above and parallel to the reference curve. Draw a second line 3 *dB* above and parallel to the reference curve, extending from 1,000 to 4,000 *Hz*. The range between the reference curve and the 5 dB and 3 dB curves represents the maximum permitted deviation of the noise spectrum to receive a neutral rating.

5. Determine the quality of the sound by observing how the shape of the spectrum deviates from the boundary limits based on the results of step 4. That is, note the relevant sound quality descriptor from **Table 9.6**.

6. Assign the spectrum a complete *RC* rating; using the value determined in step 2 and the descriptor in step 5.

Table 9.6
Sound Quality Descriptors for RC Rating Procedure

Sound Quality	**Descriptor**	**Description**
Neutral Spectrum	(N)	Sound pressure level data falls within the 5 dB and 3 dB curves and reference curves established in step 4
Rumbly Spectrum	(R)	Sound pressure level data exceeds the boundary established between the 5 dB and 3 dB curves and reference curve (31.5 to 500 *Hz*)
Hissy Spectrum	(H)	Sound pressure level data exceeds the boundary established between the 5 dB and 3 dB curves and reference curve (1,000 to 4,000 *Hz*)
Tonal Spectrum	(T)	A prominent sound pressure level data point in any octave band exceeds the boundaries established between the 5 dB and 3 dB curves and reference curve by more than 3 *dB*
Acoustically Induced Perceptible Vibration	(RV)	Sound pressure level data occurs in cross-hatched region of *RC* curve

Just like the *NC* curves, *RC* curves are actually plots of sound pressure levels. See **Table 9.7** for the sound pressure data that make up each *RC* curve.

Table 9.7
Sound Pressure Level Data for *RC* curves

***RC* Criteria**	**Sound Pressure Data Point on Curve (dB)**								
	Frequency (Hz)								
	16	**31.5**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**
RC -50	!	!	70	65	60	55	50	45	40
RC -45	!	!	65	60	55	50	45	40	35
RC -40	!	!	60	55	50	45	40	35	30
RC -35	!	60	55	50	45	40	35	30	25
RC -30	60	55	50	45	40	35	30	25	20
RC -25	55	50	45	40	35	30	25	20	15

1. Shaded areas represent noise that is below the threshold for hearing for continuous noise exposure.
2. ! symbol indicates a serious vibration condition may exist if sound pressure level is more than 65 dB.

Sample Problem 9-3

Consider the sound pressure level data calculated in Sample Problem 9-1. What is the RC rating?

Answer: Follow the steps outlined in the previous section.

Step 1: The data is plotted on the RC graph in **Figure 9.6**.

Step 2: Average of the sound pressure levels in the 500, 1000 and 200 Hz bands is (33+41+36)/3, or 37. RC rating is 37. Now we need to determine the sound quality descriptor.

Step 3: The –5 dB slope reference curve is shown as a solid line in **Figure 9.6**. The line is located between the RC-35 and RC-40 curves.

Step 4: The two limit curves are shown as dashed lines parallel to the solid reference line from Step 3.

Step 5: Since sound pressure levels in the 1000 and 2000 octave bands exceed the limit (dashed lines) established for sound pressure levels between 1000 and 4000 Hz, the sound quality descriptor is (H) for hissy.

Step 6: The complete RC rating is **RC-37(H)**.

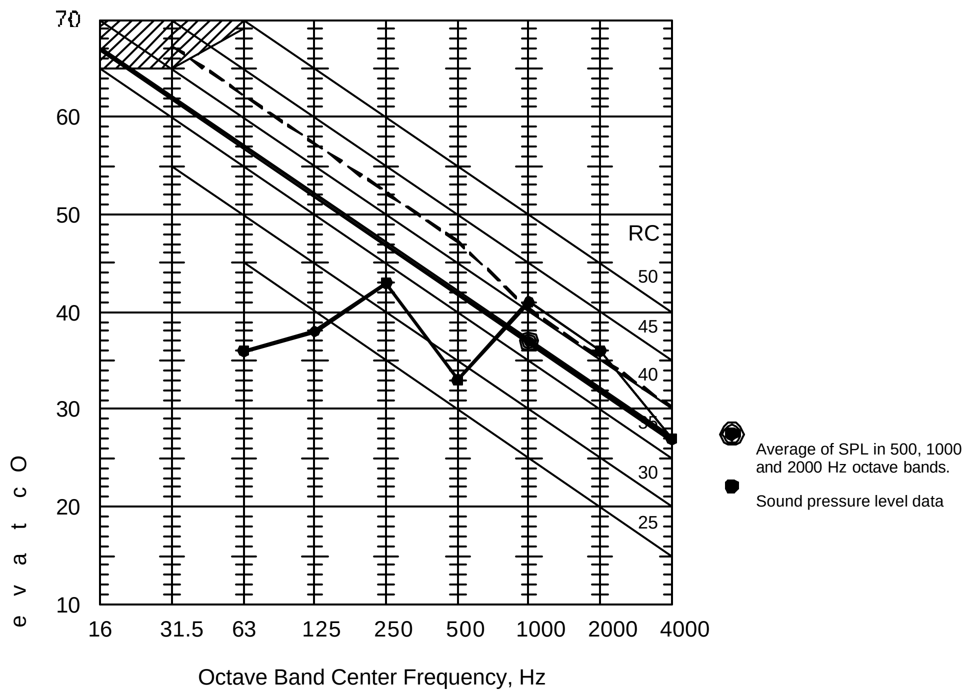

Figure 9.6
Plot of sound pressure level data for Sample Problem 9-2

9.3.3 Criteria Design Guidelines

The allowable *RC/NC*-curve for a space is generally chosen by the building's owner or architect; however, **Table 9.8** provides design guidelines for achieving acceptable RC neutral, or RC(N), background sound based on room function.

Table 9.8
Design Guidelines for HVAC-Related Background Sound in Rooms

Environment	RC(N)[a,b] Criterion	Environment	RC(N)[a,b] Criterion
Residences, Apartments, Condominiums:	25-35	**Laboratories (w/fume-hoods)**	
		Testing/research, minimal Speech communication:	45-55
		Research, extensive telephone use, speech communication:	40-50
		Group teaching:	35-45
Hotels/Motels		**Church, Mosque, Synagogue**	
Individual rooms or suites:	25-35	General assembly:	25-35
Meeting/banquet rooms:	25-35	With critical music programs:[c]	
Corridors, lobbies:	35-45		
Service/support areas:	35-45		
Office Buildings		**Schools**[e]	
Executive and private offices:	25-35	Classrooms up to 750 ft^2:	40 (max)
Conference rooms:	25-35	Classrooms over 750 ft^2:	35 (max)
Tele-conference rooms:	25 (max)	Large lecture rooms, without speech amplification:	35 (max)
Open-plan offices:	30-40		
Corridors and lobbies:	40-45		
Hospitals and Clinics		**Libraries:**	30-40
Private rooms:	25-35		
Wards:	30-40		
Operating rooms:	25-35		
Corridors & public areas:	30-40		
Performing Arts Spaces		**Courtrooms**	
Drama theaters:	25 (max)	Unamplified speech:	25-35
Concert and recital halls:[c]		Amplified speech:	30-40
Music teaching studios:	25 (max)		
Music practice rooms:	35 (max)		
Indoor Stadiums, Gymnasiums			
Gymnasiums and natatoriums:[d]	40-50		
Large seating-capacity spaces with speech amplification:[d]	45-55		

NOTES:

a: The values and ranges are based on judgment and experience, not on quantitative evaluations of human reactions. They represent general limits of acceptability for typical building occupancies. Higher or lower values may be appropriate and should be based on a careful analysis of economics, space usage, and user needs.

b: When the quality of sound in the space is important, specify criteria in terms of RC(N). If the quality of the sound in the space is of secondary concern, the criteria may be specified in terms of NC or NCB levels of similar magnitude.

c: An experienced acoustical consultant should be retained for guidance on acoustically critical spaces (below RC 30) and for all performing arts spaces .

d: RC or NC criteria for these spaces need only be selected for the desired speech and hearing conditions.

e: There is evidence that HVAC-related sound criteria for schools, such as those listed in this Table, are too high and impede the learning process for children in the primary grades whose vocabulary is limited, or whose 1st language is not English. Some educators and others believe that the HVAC-related background sound should not exceed RC 25 (N).

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9.3.4 Other Criteria

The *NCB* and *RC Mark II* are two other noise rating methods for specifying noise criteria, but they have yet to achieve the acceptance of the *RC* or *NC* methods due to their complexity. You can learn more about these methods in **Appendix. A.9.2**.

9.4 Spectrum Shape

Regardless of whether the *RC* or *NC* rating method is used, it is important to have a balanced noise spectrum. A complete acoustical analysis of the HVAC system will allow the designer to control the noise spectrum so that it closely approximates the shape of the *RC* or *NC* curve. HVAC systems without balanced noise spectrums are likely to have poor sound quality and can cost a building owner money by making it difficult to find occupants for noisy areas of the building. For example, a slight humming noise may cause a loss in productivity even though the over-all sound pressure levels are within a specified NC criterion. This is why it is important to use the RC or other rating method that considers spectrum shape.

If there is a problem with spectrum shape, sometimes noise is added to the environment to provide a more balanced noise spectrum. This is called noise-masking, and can consist of a discretely located sound system, providing broad-band sound to the environment. This method can also be very effective at masking private conversations in an open office environment. Usually, this is an inconvenient and expensive solution. It is always better to evaluate potential noise problems during the design stage before they become cost problems.

CHAPTER 10: Supplemental Attenuation

10.1 Calculating Attenuation Requirements

Using the information presented in the previous sections, a complete acoustical analysis may be performed on an HVAC system. The objective of this analysis is to calculate the noise levels at the duct outlets and to determine if those levels will meet the design criteria of the space being supplied. The attenuation required is the difference between the actual level and the desired criterion level at each frequency.

As an example, consider the spectrum shown in **Figure 9.4** (from **Sample Problem 9-1**). If the design criterion for this space is *NC*-35, the system would be considered unacceptable from a specification standpoint, since the levels at 1,000 and 2,000 *Hz* both exceed the prescribed *NC*-35 limits. The spectrum would meet an *NC*-35 criterion with the addition of 5 *dB* attenuation at 1,000 *Hz* and 2 *dB* of attenuation at 2,000 *Hz*. To achieve an *NC*-25 criterion, would require 6 *dB* additional attenuation at 250 *Hz*, 2 *dB* at 500 *Hz*, 14 *dB* at 1,000, 12 *dB* at 2,000 *Hz* and 5 *dB* at 4,000 *Hz*. To see how these values were determined numerically, refer to **Tables 10.1** and **10.2**.

Table 10.1
Determining Attenuation Required
To Meet NC-35 Criterion For Sample Problem 9-1

Item	**Sound pressure levels** *(dB)*							
	Frequency (*Hz*)							
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
Room Noise (**Sample Prob. 9-1**)	36	38	43	33	41	36	27	19
NC-35 (data from **Table 9.5**)	60	52	45	40	36	34	33	32
Difference[1] (Attenuation Required)	0	0	0	0	5	2	0	0

1 – difference is set equal to zero when the sound pressure level from the room noise is less than the NC value.

Table 10.2
Determining Attenuation Required
To Meet NC-25 Criterion for Sample Problem 9-1

Item	Sound pressure levels *(dB)* Frequency (*Hz*) 63	125	250	500	1000	2000	4000	8000
Room Noise (**Sample Prob. 9-1**)	36	38	43	33	41	36	27	19
NC-25 (data from **Table 9.5**)	54	44	37	31	27	24	22	21
Difference[1] (Attenuation Required)	0	0	6	2	14	12	5	0

1 – difference is set equal to zero when the sound pressure level from the room noise is less than the NC value.

When the residual sound power levels at the terminals exceed the noise criteria levels at any frequency, additional attenuation must be provided. There are several methods for providing this attenuation as discussed in the remainder of this chapter.

10.2 Double-Wall Insulated and Single-Wall Lined Duct Noise Control

A highly effective method to reduce the noise levels propagated in a duct system is to provide an absorptive insulation medium inside the duct. The insulation material is usually fiberglass, and the duct/liner combination must be constructed so that the inner, nominal duct dimension is maintained.

The most effective insulated duct system consists of an inner perforated metal round duct shell, sized to the nominal duct diameter, surrounded by at least 1-*inch* of fiberglass insulation, and enclosed in a standard-gauge solid outer duct pressure shell. This construction is called double-wall duct and McGill AirFlow Corporation manufacturers this product under the trade name, ACOUSTI-k27®.

The outer shell on double-wall round duct systems provides high transmission loss, thereby reducing the radiated noise to the surrounding environment. Furthermore, the inner perforated shell will provide a smooth airflow surface and protect against insulation erosion. For restricted space applications, double-wall flat oval and rectangular duct are acceptable alternatives, but transmission loss is usually reduced.

There will be a slight increase in duct pressure loss when the inner perforated shell is used due to the increased roughness of the surface. **Section 1.4.4** of **Chapter 1** provides a method for calculating these losses. Double-wall duct with a solid inner shell has the same pressure loss as a single-wall duct with the same inner dimensions.

Tests have shown that the airflow-generated noise produced by perforated duct liners is no greater than that of standard single-wall ducts (see **Section 8.3**).

10.2.1 Attenuation of Insulated Duct Systems

Double-wall or lined single-wall duct attenuates more noise than a single-wall duct. This increased attenuation is called *insertion loss* and is the amount of additional noise control above the amount provided by the natural attenuation of single-wall duct. An extensive database of insertion loss values for insulated duct is given in **Appendix A.5.1**. Data are presented in terms of decibels (insertion loss) per foot of duct length for 1-inch, 2-inch, and 3-inch insulation thickness. Insertion loss (dB/ft) is given for six typical diameters at velocities of -4,000, -2,000, 0, 2000 and 4,000 *fpm*. The insertion loss for double-wall (with a perforated inner wall) and lined rectangular duct is also given in **Appendix A.5.1**

To calculate the total insertion loss attained by a lined duct, the insertion losses from **Appendix A.5.1** are added to the natural attenuation data of the single-wall duct from **Section 8.2.1**.

Double-wall or lined single-wall duct may be located near the fan to reduce the noise generated by the fan throughout the system. This strategy does not always work as there may be certain outlets that require additional attenuation. Therefore, if a problem is anticipated at a particular outlet location, double-wall or lined single-wall duct can be used upstream of terminals or diffusers along that outlet path.

Sample Problem 10-1 evaluates how much double-wall duct is needed to satisfy an NC-45 requirement for the noisiest outlet (Outlet B) in **Sample Problem 8-1**.

Sample Problem 10-1

---

*Refer again to **Sample Problem 8-1**. What will be the attenuated resultant sound pressure levels be in a room fed by terminal B if 3-inch-thick insulated duct is used in sections 1001, 1003, and 1004? (NOTE: All fittings have a solid inner shell and do not contribute additional attenuation. The only fittings with a perforated inner liner having significant attenuation are double-wall elbows. See Table A.29 for double-wall elbow attenuation as a function of elbow diameter and insulation thickness) The room is 12ft x 15ft x 8ft and sound pressure levels will be measured 8 feet from the terminal outlet. The required room NC criterion is NC-45.*

Answer: Insertion loss values (dB/ft) for double-wall duct at a velocity of +2,000 *fpm*, were obtained from **Appendix A.5.1, Table A.26**, and added to sections 1001, 1003 and 1004 leading to Outlet B so that the resultant *Lw* values are equal to or less than that required to meet an *NC* of 45.

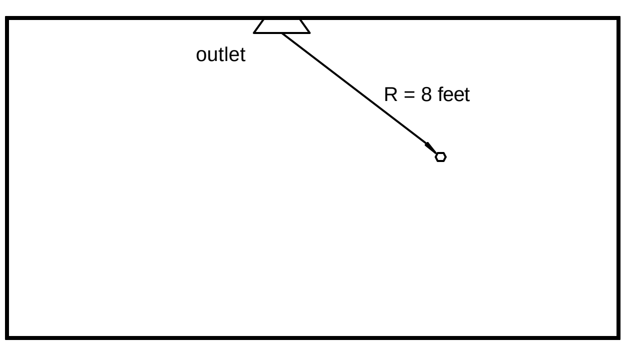


Figure 10.1
Room Elevation View

Table 10.3

1-Inch Double-Wall, Acoustical Analysis, Outlet B of Sample Problem 8-1

Outlet B	**Octave Band Center Frequency** (*Hz*)							
	63	**125**	**250**	**500**	**1k**	**2k**	**4k**	**8k**
Section 1001								
1. Fan *Lw* values	89	89	90	83	78	72	68	63
2. Single-wall duct, 24 *inches*, *L*=15 *ft*	0	0	0	0	1	1	1	1
3. Mitred 90E elbow with vanes, 24 in. dia.	0	1	2	3	3	3	3	3
4. Double-wall duct (ACOUSTI-k27®), 24 *inches*, *L*=15 *ft*, Insertion Loss *(dB)*	5	10	18	30	21	16	15	14
Remaining	84	78	70	50	53	52	49	45
5. Duct *GNL* at 2546 *fpm*	67	62	51	47	46	43	35	35
6. Elbow *GNL* at 2546 *fpm*	63	64	63	59	53	43	30	13
Resultant (Section 1001)	84	78	71	60	56	53	49	45
Section 1003								
1. LoLoss™ Tee, Main, 24 *inches* (sound power split)	1	1	1	1	1	1	1	1
2. Single-wall duct, 24 *inches*, *L*=15 *ft*	0	0	0	0	1	1	1	1
3. Double-wall duct (ACOUSTI-k27®), 24 *inches*, *L*=15 *ft*, Insertion Loss *(dB)*	5	10	18	30	21	16	15	14
Remaining	78	67	52	29	33	35	32	29
4. LoLoss™ Tee Main *GNL*	48	39	37	35	32	28	23	18
5. Duct *GNL* at 2292 *fpm*	64	59	49	45	44	41	32	32
Resultant (Section 1003)	78	68	54	45	45	42	35	34
Section 1004								
1. LoLoss™ Tee Branch, 15 *inches* (sound power split)	4	4	4	4	4	4	4	4
2. Single-wall Duct, 15 *inches*, *L*=8 *ft*	0	0	0	0	1	1	1	1
3. Double-wall duct (ACOUSTI-k27®), 15 *inches*, *L*=8 *ft*, Insertion Loss *(dB)*	3	6	11	18	14	11	10	9
Remaining	71	58	39	23	26	26	20	20
4. LoLoss™ Tee, Branch *GNL*	40	39	37	34	30	25	20	14
5. Duct *GNL* at 1629 *fpm*	43	39	37	33	35	33	23	23
Resultant (Section 1004), Outlet B, *Lw*	71	58	43	37	37	35	26	25
Resultant due to Room Effect (**Equation 9.1**)	66	52	36	29	28	25	15	13
Sound pressure levels for *NC* = 45	67	60	54	49	46	44	43	42
Outlet B, Attenuation Requirement (*dB*)	0	0	0	0	0	0	0	0

If we needed slightly more attenuation, we might consider replacing the elbow with a double-wall, 1.5 centerline radius elbow with perforated inner liner (see double-wall elbow data in **Table A.29**).

Keep in mind that Outlet B represents the worst-case noise problem for the system. Selecting double-wall insulated duct in sections leading to Outlet B will not necessarily solve noise problems at the other outlets. Similarly, an acoustical analysis must be performed for each outlet

It is important to note from **Table 10.1** data that the airflow generated noise level of the duct and fittings now contribute to the overall noise of the system especially in the mid and upper frequencies. Attenuation devices can only reduce the noise level to as low as the highest *GNL* in a given section.

10.3 Silencers

A silencer is a device that can be placed directly in the air-stream to attenuate duct-borne noise. It is essentially a series of perforated baffles or bullets, usually with internal sound absorbent material. Baffle silencers are manufactured in a variety of configurations, but often have a central element that splits the airflow. Both the central element and the perimeter walls provide absorptive surfaces. Most rectangular silencers use baffles, while most round silencers utilize bullets.

The acoustical performance of duct silencers is generally described in terms of insertion loss. Insertion loss is a measure of the noise level reduction in a duct when a silencer is inserted into a duct, compared to the level in the same duct without the silencer. Any disturbance or obstruction in an air-stream can be a source of generated noise, and this is also the case with silencers. If the generated noise level of the silencer at any frequency is within 10 dB of the attenuated sound power level (after subtracting the silencer insertion loss), it is necessary to add (logarithmically) the generated noise level to the attenuated sound power level. This will determine the actual sound power immediately downstream of the silencer.

By design, baffle and bullet type silencers block a portion of the air-stream. They will generally reduce the local cross-sectional area of the air-stream and increase the velocity. Therefore, they cause additional pressure drops which must be added to the pressure drop of the section in which they are located. Manufacturers should always include values for insertion loss, regenerated noise and pressure drop in their literature.

Typical values for McGill AirFlow Corporation silencer insertion loss, regenerated noise, and pressure drop are presented in **Appendix A.5.4**. Silencers are often placed downstream of an air-handling unit to attenuate duct-borne noise, and this close proximity to the fan exit can cause excessive pressure loss above the published amount due to the uneven airflow distribution entering the silencer.

Silencers can be purchased with varying percent open areas depending on the type of noise control desired and allowable pressure drop. Tests for insertion loss should be conducted in accordance to **ASTM Standard E477**, *Standard Method for Testing Duct Liner Materials and Prefabricated Silencers for Acoustical and Airflow Performance*.

Silencers can be placed individually at any point in a duct system, or sometimes they can be stacked and grouped to fill a large area, such as the cross-section of a fan plenum. There are three main types of HVAC duct silencers so each category is addressed separately.

10.3.1 Dissipative Silencers

This is the most common silencer type and is typically used in a wide variety of commercial and industrial applications. The silencers utilize an absorptive, usually fibrous, material contained in baffles or bullets with a perforated sheet that allows sound energy to be absorbed into the fibrous material. This type of silencer is also called an absorptive or passive silencer. The insertion loss is accomplished from the conversion of sound energy into frictional forces induced upon the absorptive material. Dissipative silencers are simple, relatively inexpensive, and generally effective for broadband attenuation. The silencers are more effective when length is increased and/or baffle and bullet dimensions are increased, but unfortunately, the result is higher pressure drop and generated noise level. A couple of simple designs are shown in **Figures 10.2** and **10.3**.

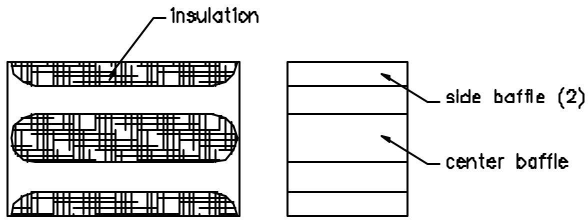


Figure 10.2
Rectangular Dissipative Silencer

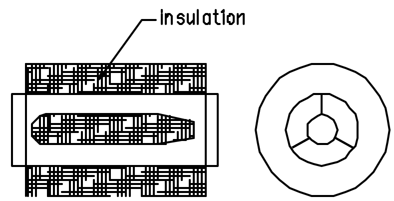


Figure 10.3
Round Dissipative Silencer

10.3.2 Reactive Silencers

Reactive silencers operate under the same principal as the Helmholtz resonator. There are specially sized chambers constructed of sheet metal that are sized specifically for attenuating entering noise. The sound waves are essentially trapped in these chambers, attenuating themselves out. Since reactive silencers are not filled or packed with fiberglass insulation, they are commonly called no-fill or pack-less silencers. Reactive silencers are used in applications that might be laden with high moisture content, highly corrosive gases, or require 100 percent fiber-free air. These silencers are more expensive than passive silencers and do not attenuate as well as a passive silencer of equal cross-section and length. A couple of examples of reactive silencers are shown in **Figures 10.4** and **10.5**.

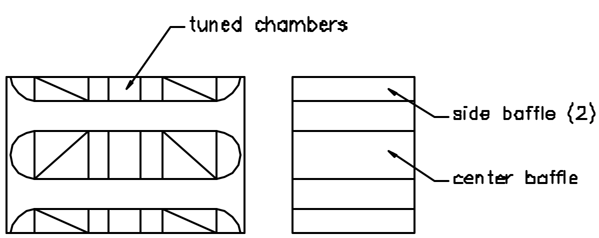


Figure 10.4
Rectangular Reactive Silencer

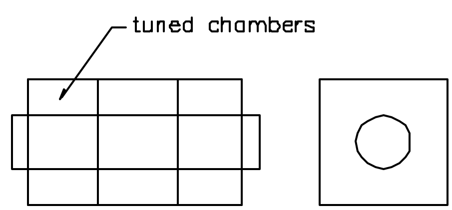


Figure 10.5
Round Reactive Silencer

10.3.3 Active Silencers

Active noise control or noise cancellation technology has been around for many years. It is only recently that the technology has been affordable for widespread use. The technology uses four basic components to attenuate noise. The pickup microphone records the sound pressure levels inside the duct; the controller analyzes the readings and generates a noise that is out of phase with the incoming noise; the loudspeaker produces the out of phase noise to cancel the incoming sound waves; finally, the error microphone records the downstream noise to document the effectiveness of the noise cancellation, and communicates to the controller to make necessary adjustments. See **Figure 10.6** for a schematic of the active silencer.

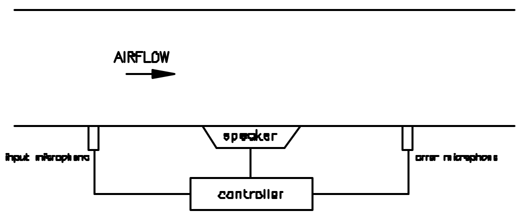


Figure 10.6
Active Silencer Components

While the technology encompassed by the active silencer is impressive, the practicality of its usage remains limited due to several issues. The active silencer is a high maintenance item compared to reactive and passive silencers, both of which contain no moving or electrical parts. The active silencer is relatively expensive and is only effective at the low frequencies. Therefore, their use is limited to critical environments where low frequency is of utmost importance, and the mid to high frequencies are taken care of via traditional methods (lined duct and/or silencers).

10.4 No-Bullet Silencers

The previous section discussed the application of rectangular baffle or round bullet silencers, which by definition have a central element that obstructs the airflow. This blockage not only causes airflow turbulence, but it reduces the cross-sectional area of the air passage, increasing the air velocity and pressure loss. The combination of these effects can result in substantial pressure losses if in-duct velocities are not controlled properly.

A no-bullet silencer provides attenuation similar to that of a bullet silencer but without the bullet. This means that a no-bullet silencer can be substituted for a section of straight duct, and it will provide attenuation with no additional pressure loss. The silencer is essentially a double-wall duct with a 6-*inch* or 8-*inch* lining and provides a high degree of low frequency attenuation.

A no-bullet duct silencer in combination with a double-wall insulated elbow will provide exceptional broadband attenuation with a pressure loss equal to a single-wall duct of the same configuration. Since this type of silencer has no additional pressure drop (greater than a single-wall duct of the same configuration) and no elements in the airstream, there is no additional generated noise.

Insertion loss values for no-bullet silencers can be found in **Appendix A.5.4.1**. **Sample Problem 10-2** illustrates how effective a no-bullet silencer is in attenuating the noise to Outlet B in **Sample Problem 8-1**. See **Figure 10.7** to see a sketch of a no-bullet silencer.

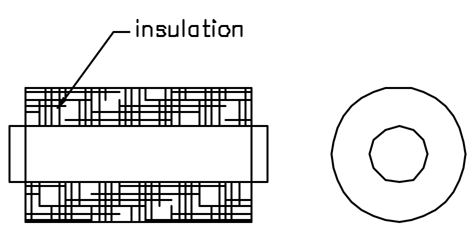


Figure 10.7
No-bullet Round Silencer

Sample Problem 10-2

*Refer to **Sample Problem 10-1** and **Sample Problem 8-1**. Will Outlet B meet NC-45 if a model CSF-HV-L35 silencer is used in section 1001 and is the only form of noise control? All other conditions same as in **Sample Problem 10-1**.*

Answer: Using a **CSF-HV-L35** round silencer (see **Appendix A.5.4.1**), the system will produce the results in **Table 10.4**.

Table 10.4
No-bullet Silencer, Acoustical Analysis, Outlet B of Sample Problem 8-1

	Sound Level *(dB)*							
Comment ***(dB)***	**Octave Band Center Frequency** (*Hz*)							
	63	**125**	**250**	**500**	**1k**	**2k**	**4k**	**8k**
Section 1001								
1. Fan *Lw* values	89	89	90	83	78	72	68	63
2. Single-wall duct, 24 *inches*, *L*=15 *ft*	0	0	0	0	1	1	1	1
3. Mitred 90E elbow w Vanes, 24 in. dia.	0	1	2	3	3	3	3	3
4. 24 *inch*, model CSF-HV-L35 silencer insertion loss at 2546 *fpm*	7	10	20	25	24	19	17	14
Remaining	82	78	68	55	50	49	47	45
5. Duct *GNL* at 2546 *fpm*	67	62	51	47	46	43	35	35
6. Elbow *GNL* at 2546 *fpm*	63	64	63	59	53	43	30	13
Resultant (Section 1001)	82	78	69	61	56	51	47	45
Section 1003								
1. LoLoss™ Tee, Main, 24 *inches*	1	1	1	1	1	1	1	1
2. Single-wall duct, 24 *inches*, *L*=15 *ft*	0	0	0	0	1	1	1	1
Remaining	81	77	68	60	54	49	45	43
3. LoLoss™ Tee Main *GNL*	48	39	37	35	32	28	23	18
4. Duct *GNL* at 2292 *fpm*	64	59	49	45	44	41	32	32
Resultant (Section 1003)	81	77	68	60	54	50	45	43
Section 1004								
1. LoLoss™ Tee Branch, 15 *inches*	4	4	4	4	4	4	4	4
2. Single-wall duct, 15 *inches*, *L*=8 *ft*	0	0	0	0	1	1	1	1
Remaining	77	73	64	56	49	45	40	38
3. LoLoss™ Tee, Branch *GNL*	40	39	37	34	30	25	20	14
4. Duct *GNL* at 1629 *fpm*	43	39	37	33	35	33	23	23
Resultant (Section 1004), Outlet B, *Lw*	77	73	64	56	49	45	40	38
Resultant due to Room Effect (**Equation 9.1**)	72	67	57	48	40	35	29	26
Sound pressure levels for *NC* = 45	67	60	54	49	46	44	43	42
Outlet B, Attenuation Requirement (*dB*)	5	7	3	0	0	0	0	0

Alone the CSF-HV-L35 silencer cannot meet the conditions. However, it has already been demonstrated that substituting 3-inch ACOUSTI-k27® for the silencer in section 1001 - **Table 10.1**, would take care of the additional attenuation requirements to achieve on *NC*-45 of Outlet B.

10.5 Computer Analysis

A thorough evaluation of the acoustical performance of a duct system requires a rigorous analysis. This analysis must address all the duct system acoustics, room acoustics, and supplemental attenuation. A methodology used in previous sections evaluates the acoustical performance of various noise control solutions. The optimum noise control solution is that which yields the lowest initial and operating costs to the customer.

Quite frequently, when a noise problem is anticipated in a duct system, a common practice is to insulate the entire system and hope that the noise problem is eliminated. This approach may not work and is almost never the lowest cost solution. In fact, it could introduce too much attenuation in some frequencies, resulting in hiss or rumble noise problems due to unmasked tonal components. Worse yet, it might not introduce enough noise control, causing additional cost to the owner after the system has been installed. There is just no way to be sure the proper noise control will be applied without performing an acoustical analysis on the entire single-wall duct system.

Two important points to remember are:

1. A large amount of noise control treatment will introduce additional system pressure losses, especially significant if baffle silencers are used. An increase in pressure loss will increase the operating cost of the system. If a fan has already been selected, the amount of available fan pressure must be considered.

2. Select noise control that reduces noise at the outlets, resulting in a noise spectrum that closely matches the shape of the *NC* or *RC* sound pressure level curve.

As noted, a single-wall acoustical analysis will allow the designer to select the minimum noise control required for the system. Sometimes the proper amount of noise control is determined easily. Other times, when the HVAC system is large and noise problems exist at outlets throughout, calculations can become time consuming and laborious. Computer programs such as the one in **Appendix A.8.1** are designed to reduce the calculation time and provide a multitude of designs and initial costs to be evaluated.

A computer analysis examined two noise control options for the duct system shown in **Figure 8-2, Sample Problem 8-1**. The results are noted in **Table 10.5**. In option 1, a silencer and insulated elbow combination is used in duct section 1001 along with some double-wall duct. This type of noise treatment yields a large amount of low-frequency to mid-frequency noise control without introducing a large pressure drop. Option 2 incorporates only double-wall duct with insulation thicknesses of 1 and 3 *inches*. Option 2 was considered because it used the minimum amount of insulated duct without incorporating a silencer. Both options meet the desired acoustical criterion.

Table 10.5
Comparison of Noise Control Options

Section Number	Option 1	Option 2
1001	Silencer[1]	3-inch DW
1002 (Outlet A)	2-inch DW	2-inch DW
1003	3-inch DW	3-inch DW
1004 (Outlet B)	3-inch DW	3-inch DW
1005	3-inch DW	3-inch DW
1006 (Outlet C)	SW	1-inch DW
1007	SW	SW
1008 (Outlet E)	SW	SW
1009 (Outlet D)	SW	SW
Initial Cost	$3,863	$4,636

[1]Model CEF-HV-L55 round elbow duct silencer
SW = single-Wall duct and fittings,
DW = double-Wall duct and fittings

Option 1 has the lowest initial cost and is more desirable unless there is some constraint that will not allow the use of the chosen silencer. Only a computer analysis will allow proper selection of noise control devices in meeting both design criteria and the budget.

Appendix

A.1 General Information

The following section provides the designer with general information that is useful in designing and analyzing duct systems. It is intended to supplement the material found in the text and assist with calculations. This information consists of a glossary of terms, conversion tables, area/diameter table and equations, duct shape conversion tables, nonstandard condition correction factors, and velocity/velocity pressures.

A.1.1 Glossary of Terms

A

A-Weighting. A weighting system that makes adjustments to frequencies based on each frequency's relative annoyance to human hearing, and results in a single number rating that may be used for comparison.

Acoustically Insulated Duct. A duct consisting of a solid metal outer shell, thermal/acoustical insulation, and a metal inner liner (often perforated). See double-wall duct.

Acoustically Lined Duct. A duct consisting of a solid metal outer shell and thermal/acoustical insulation liner.

Acoustics. The scientific study of sound, especially of its generation, propagation, perception, and interaction with materials and other forms of radiation.

Active Silencer. A type of silencer that utilizes electronics and a loud-speaker to produce sound waves out of phase with the noise the silencer is trying to attenuate. These silencers are effective at low frequencies.

Air Cleaning Device. A device that removes contaminants from an air stream (for example, a cyclone, dryer, scrubber, electrostatic precipitator, baghouse, or filter).

Air Handler. A device in a duct system which moves the air.

ASHRAE Room Effect Equation. A relationship used to calculate the sound pressure level in a normally furnished room.

Aspect Ratio. For rectangular duct, it is the ratio of the large side to the small side; for flat oval duct, it is the ratio of the major axis to the minor axis. For example, 12-inch × 48-inch rectangular or flat oval duct has an aspect ratio of 4.

B

Balancing. A process of directing airflow to individual terminals.

Barometric Pressure. The pressure of the surrounding atmosphere as obtained from a barometer.

Bellmouth. A rounded fitting used to connect a duct to an air handler or plenum.

BFI. Blade frequency increment is a correction factor (in decibels) that is applied to a single frequency when determining sound power level.

Break-In Noise. Sound energy that radiates into a duct system through its duct walls from the surrounding area.

Break-Out Noise. Sound energy that radiates out of a duct system through its duct walls to the surrounding area.

C

Capture Velocity. The air velocity required to capture (entrain) the material or fumes conveyed by an exhaust system.

Carrying Velocity. The minimum velocity required to avoid material fallout.

Close-Coupled Fittings. Two or more fittings which cause substantial increase in pressure loss because of their close proximity to each other.

Computer-Aided Duct Design. The use of computer programs to design duct systems.

Conical. A fitting with a tapered tap. The large end connects to the trunk duct.

Conservation of Energy. A law stating that the total energy per unit volume of air flowing in a duct system equals the sum of the static, kinetic, and potential energy.

Conservation of Mass. A law stating that the mass flowing into a control volume must equal the mass flowing out of the control volume.

Constant Velocity Design. A method of sizing duct using a fixed velocity.

Continuity Equation. A relationship that predicts the duct volume flow rate by knowing the duct area and velocity ($Q = VA$).

Converging Flow. In return/exhaust systems, a situation in which air flowing from an straight-through (upstream) and branch duct combine to form a common (downstream) airflow.

Cooling Load. The amount of energy needed to cool a building (expressed in Btu).

Corrected cfm. A recalculated volume flow rate that accounts for excess pressure.

Cost Optimization. Designing a duct system so that it will offer the lowest present value (initial equipment costs plus operating costs).

Coupling. A fitting that connects one duct to another.

Cross. A fitting that has taps opposite one another across the body of the fitting.

Cycle. The complete movement of a particle in periodic motion from the zero point, through maximum and minimum compression, and back to the zero rest point. The amount that these movements occur per second is the frequency they occur, or the number of cycles per second, also known as Hertz.

D

Darcy-Weisbach. An equation for determining the pressure loss in duct.

Decibel. A sound level measurement.

Density. Pounds mass per cubic foot of a substance.

Design Leg. The duct path with the maximum resistance to airflow. It will have an excess pressure of zero.

Design Path. See Design Leg.

Design Process. A procedure for designing duct systems.

Diffuser. A type of terminal device that purposefully spreads and mixes the impinging air flow as it delivers air into the conditioned space.

Dissipative Silencer. A type of silencer that utilizes an absorptive media that converts acoustical energy into frictional forces thereby producing a reduction in noise level from the entry to the exit of the silencer.

Diverging Flow. In supply systems, a situation in which air flowing from an common (upstream) duct splits to form a branch and straight-through (downstream) airflow.

Double-Wall. Usually refers to duct that is made up of an inner and outer duct, separated with a 1, 2 or 3 inch annular space that is filled with fiberglass.

Downstream. The location on a fitting that is just past a branch take-off in the direction of air flow.

Duct. A device that conveys air from one point to another (it can be round, flat oval, rectangular, or square).

Duct Class. A way of classifying ductwork according to material abrasion characteristics.

Duct Design. The process of determining size, shape, pressure, material, support, and operation of a system to convey conditioned air.

Duct Velocity. The speed at which air travels through a duct.

Duct Wall Loss. The dissipation of sound energy traveling through a duct caused when a fraction of that energy is transmitted to the surface of the duct. The amount of sound energy lost is affected by the size and shape of the duct and the frequency of the sound.

Dynamic Losses. A pressure drop in a duct system caused by changes in airflow direction, volume, or shape.

E

Elbow Reflection. The natural sound attenuation caused when sound is reflected by the metal wall of an elbow or other fitting that changes the direction of airflow.

End Reflection. The natural sound attenuation caused when a significant change of area at the end of a duct run reflects part of the acoustical energy back into the duct.

Enhanced Pressure Design. A method of design which decreases duct sizes and substitutes less efficient fittings in nondesign legs or branches to reduce excess pressure after one of the duct design methods is used.

Enthalpy. The amount of energy per pound of dry air.

Entrance. A device (such as a hood) that permits air to enter the duct system.

Equal Friction. A pressure drop of constant magnitude.

Equal Friction Design. A method of sizing duct based on a constant friction loss per 100 feet.

Equivalent Round Diameter. The diameter of round duct that has the same pressure loss per unit length at the same volume flow rate as a flat oval or rectangular duct.

Excess Pressure. The difference between the design leg pressure and that of any other leg.

Exhaust. Ductwork on the suction side of a fan.

Expander. A fitting which increases in size in the direction of airflow.

F

Face Velocity. Air velocity at the opening of a hood, obtained by dividing the volume flow rate by the component face area.

Fan. A device that moves air through ductwork at a constant volume flow rate.

Fan Laws. Laws that govern the performance of fans and can be used to predict a fan's performance at other points of operation than those tested.

Fan Noise. The amount of sound power emitted by a fan for each octave band.

Fan Pressure. The pressure needed to overcome the path of greatest resistance in a duct system.

Flat Oval. A duct shape with two flat parallel sides and two semicircular sides, fabricated by stretching round duct.

Flat Span. The difference between the major and minor axes in a flat oval duct.

Frequency. The number of sound waves (cycles) produced per unit of time. Frequency is measured in cycles per second or Hertz (Hz).

Friction Loss. The energy loss in a duct system caused by air moving over a rough surface such as a duct wall (see Darcy-Weisbach).

G

Grille. A terminal device that spreads the air flow coming into or out of it into an angular pattern. A grille has no means for regulating air flow.

H

Heating Load. The amount of energy needed to heat a building (expressed in Btu).

Heel-Tapped Elbow. An elbow which has a tap on its heel.

Hertz. Unit of measurement in acoustics equal to cycles per second.

Hiss. A descriptor used in the RC Rating System, hiss refers to the sound in the approximate frequency range from 1000 to 4000 Hz that is above a calculated threshold.

Hood. An inlet device that captures air or particulate and guides it into the duct system.

Humidity Ratio. The ratio of pounds of water vapor per pound of dry air.

I

Inlet. A device to get air into a duct system.

L

Lateral Fitting. A branch fitting with a tap at an angle less than 90°. These fittings are generally recommended for use in material handling systems.

Lined Duct. Duct that has its interior wall lined with sound-absorbing material to attenuate noise.

Loss Coefficient. A dimensionless parameter that is determined experimentally and is a measure of a fitting's efficiency.

Loudness. The auditory sensation caused by the sound pressure level and the frequency composition of a sound.

M

Manifold Fitting. A tap or taps that are attached directly to a duct without the use of separate fitting bodies.

Mass Flow Rate. The mass of a substance flowing past a fixed point per unit length of time.

Mechanical System. Heating, ventilating, and air conditioning components that form a system to provide heating and cooling.

Mixed Air Temperature. The air temperature in an exhaust/return duct system downstream of two converging air streams with different temperatures or energy levels.

Multiple Round. Several runs of round duct used as an alternative to flat oval or rectangular duct with a large aspect ratio.

N

Natural Attenuation. The inherent ability of a duct system to absorb or dissipate sound.

NC Rating Method. A series of noise criteria curves that represent the auditory sensitivity of the human ear and can be used to evaluate sound pressure levels in different spaces.

Negative Pressure. Air pressure in a duct that is below atmospheric pressure.

No-Bullet Silencer. Type of round silencer that does not have an internal tubular device (bullet) to help attenuate noise. These silencers have minimal pressure drop, but with the use of thickly lined outer shells, are able to attenuate a good amount of noise.

Noise. Sound that we do not want to hear.

Nonstandard Conditions. When the temperature is above or below 70°F or the elevation is above sea level. Usually, corrections should be made for temperatures above 120°F or below 30°F and for elevations above 2,000 feet.

O

Octave Band. A range of frequencies in which the upper limit frequency is twice the lower limit frequency.

Offset. A single fitting or two or more elbows in a duct run that change elevation to avoid interference with another object.

Orifice Plate. A fixed diameter device used to balance a duct system by increasing resistance to airflow in the duct section in which it is installed.

P

Pitch. See *Frequency.*

Plenum Velocity. Air velocity in a plenum.

Positive Pressure. Air pressure in a duct that is above atmospheric pressure.

Power Split. The natural sound attenuation caused when a divided-flow fitting splits the sound energy. It is the ratio of the cross-sectional area of one of the downstream paths to the sum of the cross-sectional areas of all downstream paths.

R

RC Rating Method. A series of room criteria curves that is often used as an alternative to NC curves. Unlike the NC rating method, the RC rating method takes into account the shape of the noise spectrum.

Reactive Silencer. A type of silencer that does not contain fibrous material, and works on the principle of the Helmholtz resonator.

Rectangular Duct. A duct shape with two sets of parallel sides.

Reducer. A fitting that decreases in size in the direction of airflow.

Regenerated Noise. Noise created by air flowing over duct surfaces.

Register. A terminal device that is used to spread air flow to the conditioned environment. A register is essentially a grille with a means for dampening the amount of air that can pass through it.

Return Duct. Ductwork on the suction side of the fan that brings air back to the fan. Although identical in philosophy to exhaust, return air moves at much lower velocities (500 to 2,000 fpm).

Rumble. An excess of low-frequency noise.

S

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

Single-wall. Usually refers to duct that is made of sheet metal and has no fiberglass liner attached to it.

Sound Power. A quantification of the actual acoustical power generated by a sound source (expressed in watts).

Sound Pressure. A measurable fluctuation of the ambient air pressure caused by the emission of sound power (expressed in Pascals).

Specific Heat. Amount of energy per pound mass per degree Fahrenheit.

Specific Sound Power Level (Kw). For a specific type of fan, this is the generic sound power level for each octave band.

Spectrum. In acoustics, the audible range of frequencies of interest.

Split Fitting. A fitting (such as a Y-branch or bullhead tee) that divides air into two branches, each with a 30° or 90° angle to the main and a 60° to 180° angle between the taps.

Standard Air. Air that has a density of 0.075 lbm/ft^3.

Static Pressure. A measure of the static energy of the air flowing in a duct system.

Static Regain. An increase in duct pressure caused when a reduction in velocity converts velocity pressure into static pressure.

Static Regain Design. A method of sizing duct to keep pressure regain in the downstream duct or branch section equal to the static pressure loss of the upstream section.

Straight Tee. A fitting with a tap that has a constant cross-sectional area and is perpendicular to the straight-through portion.

Straight-Through. The downstream side of a diverging-flow fitting, the upstream side of a converging-flow fitting, or an in-line reducer or expander.

Supply Duct. Ductwork on the discharge side of a fan.

System Effect Factor. A factor that accounts for the deficiencies in fan and system performance associated with improper flow conditions at the inlet or outlet of the fan.

System Pressure. The pressure needed to provide air to the terminus of the design leg plus pressure to return air to the air handler.

T

Terminal. A device that permits conditioned or nonconditioned air to enter or exit a duct system.

T-Method. A method of duct design which optimizes material, pressure, and cost of operation.

Total Pressure. The total energy in a duct system. It is the sum of static pressure and velocity pressure.

Transition. A duct fitting that connects duct of different sizes or shapes (for example, a reducer, expander, or square-to-round).

U

Upstream. The location a fitting that is just before a branch take-off in the direction of air flow.

V

Velocity Pressure. A measure of the kinetic energy of the air flowing in a duct system. For standard air, it is calculated from the square of the ratio of the velocity to the quantity of 4,005.

Velocity Ratio. The ratio of branch velocity to upstream velocity.

Velocity Reduction Design. A method of sizing duct using a systematically reduced velocity.

W

Wavelength. In acoustics, it is the distance between the successive points of compression or rarefaction in the sound carrying medium.

Weighting. A system whereby each frequency of interest is adjusted with a weighting factor. The most common weighting system in acoustics is the A-weighting system.

A.1.2 Conversion Tables

Length

1 meter	= 39.37 inches	= 3.281 ft	= 1.094 yd
1 inch	= 25.4 mm	= 2.54 cm	= 0.254 m
1 foot	= 304.8 mm	= 30.48 cm	= 0.3048 m

Area

1 square inch	= 645.2 mm^2	= 6.452 cm^2	= 0.06452 m^2
1 square foot	= 0.0929 m^2	= 929 cm^2	
1 square meter	= 10.76 ft^2		

Volume

1 cubic foot	= 0.0283 m^3	= 28.317 liters
1 liter	= 0.0353 ft^3	= 0.2642 gal
1 gallon	= 3.785 liters	= 0.003785 m^3
1 cubic meter	= 35.314 ft^3	

Weight (mass)

1 ounce	= 28.35 grams
1 pound	= 0.4536 kg

1 kilogram = 2.2046 lb

Density

1 lb/ft^3 = 16.02 kg/m^3

Specific Volume

1 ft^3/lb = 0.0624 m^3/kg

Pressure

1 pascal (Pa)	= 1 newton/meter^2; 1,000 pascals		= 1kPa
1 psi	= 6,895 Pa	= 6.9 kPa	
1 bar	= 14.5 psi	= 10^5 Pa; 1 millibar	= 100 Pa
1 inch H_2O	= 0.0361 psi	= 248.84 N/m^2	= 2.488 millibar
1 kgf	= 9.807 newtons		

A.1.3 Area/Diameter Tables and Equations

A.1.3.1 Round Duct Size and Area

Table A.1 Diameter/Area

Nominal Diameter (inches)	Area (sq ft)	Nominal Diameter (inches)	Area (sq ft)	Nominal Diameter (inches)	Area (sq ft)	Nominal Diameter (inches)	Area (sq ft)
3.0	0.047	12.5	0.85	29.0	4.59	58.0	18.35
3.5	0.066	13.0	0.92	30.0	4.91	60.0	19.63
4.0	0.087	13.5	0.99	31.0	5.24	62.0	20.97
4.5	0.11	14.0	1.07	32.0	5.56	64.0	22.34
5.0	0.14	14.5	1.15	33.0	5.94	66.0	23.76
5.5	0.16	15.0	1.22	34.0	6.31	68.0	25.22
6.0	0.19	16.0	1.40	35.0	6.68	70.0	26.73
6.5	0.23	17.0	1.58	36.0	7.07	72.0	28.27
7.0	0.27	18.0	1.77	37.0	7.47	74.0	29.87
7.5	0.31	19.0	1.97	38.0	7.88	76.0	31.50
8.0	0.35	20.0	2.18	40.0	8.73	78.0	33.18
8.5	0.39	21.0	2.41	42.0	9.62	80.0	34.91
9.0	0.44	22.0	2.64	44.0	10.56	82.0	36.67
9.5	0.49	23.0	2.89	46.0	11.54	84.0	38.48
10.0	0.55	24.0	3.14	48.0	12.56		
10.5	0.60	25.0	3.41	50.0	13.64		
11.0	0.66	26.0	3.69	52.0	14.75		
11.5	0.72	27.0	3.98	54.0	15.90		
12.0	0.78	28.0	4.28	56.0	17.10		

$$A = pD^2/576$$ **Equation A.1**

where

A = Cross-sectional area of round duct (ft^2)

D = Duct diameter (inches)

A.1.3.2 Spiral Flat Oval Duct

Table A.2 Flat Oval Duct Table

Nominal Size	Approx. Equiv. Rect.	*Equiv. Round	Gauge	Cross Sectional Area Sq Ft	Surface Area Per Lineal Foot Sq Ft/Ft	Wt. Per Lineal Foot Lbs/Ft	1” K-27Ò	2” K-27Ò	3” K-27Ò
3 x 8 *	3 x 8	5.1	24	0.15	1.57	1.81	X	X	X
3 x 9 *	3 x 9	5.6	24	0.19	1.83	2.12	X	X	X
3 x 11 *	3 x 11	6.0	24	0.22	2.09	2.42	X	X	X
3 x 12 *	3 x 12	6.4	24	0.25	2.36	2.73	X	X	X
3 x 14 *	3 x 14	6.7	24	0.29	2.62	3.03	X	X	X
3 x 17 *	3 x 17	7.3	24	0.36	3.14	3.63	X	X	X
3 x 19 *	3 x 19	7.6	24	0.39	3.40	3.93	X	X	
3 x 22 *	3 x 22	8.1	24	0.46	3.63	4.20			
4 x 7 *	4 x 7	5.6	24	0.17	1.57	1.81	X	X	X
4 x 9 *	4 x 8	6.2	24	0.22	1.83	2.12	X	X	X
4 x 10 *	4 x 10	6.7	24	0.26	2.09	2.42	X	X	X
4 x 12 *	4 x 12	7.2	24	0.31	2.36	2.73	X	X	X
4 x 13 *	4 x 13	7.6	24	0.35	2.62	3.03	X	X	X
4 x 15 *	4 x 15	8.0	24	0.39	2.88	3.33	X	X	X
4 x 17 *	4 x 16	8.3	24	0.44	3.14	3.63	X	X	X
4 x 18 *	4 x 18	8.7	24	0.48	3.40	3.93	X	X	X
4 x 20 *	4 x 20	9.0	24	0.52	3.67	4.24	X	X	X
4 x 21 *	4 x 21	9.3	24	0.57	3.93	4.54	X	X	X
5 x 8 *	5 x 8	6.6	24	0.25	1.83	2.12	X	X	X
5 x 10 *	5 x 9	7.3	24	0.30	2.09	2.42	X	X	X
5 x 11 *	5 x 11	7.9	24	0.35	2.36	2.73	X	X	X
5 x 13 *	5 x 13	8.4	24	0.41	2.62	3.03	X	X	X
5 x 14 *	5 x 14	8.8	24	0.46	2.88	3.33	X	X	X
5 x 16 *	5 x 16	9.3	24	0.52	3.14	3.63	X	X	X
5 x 18 *	5 x 17	9.7	24	0.57	3.40	3.93	X	X	X
5 x 19 *	5 x 19	10.0	24	0.63	3.67	4.24	X	X	X
5 x 21 *	5 x 21	10.4	24	0.68	3.93	4.54	X		
7 x 18 *	7 x 17	11.7	24	0.80	3.67	4.24	X	X	
7 x 20 *	7 x 19	12.2	24	0.88	3.93	4.54	X	X	
7 x 21	7 x 21	12.6	24	0.95	4.19	4.84	X	X	
8 x 10 +*	8 x 9	8.9	24	0.44	2.36	2.73	X	X	
8 x 11 +*	8 x 10	9.8	24	0.52	2.62	3.03	X	X	X
8 x 13 +*	8 x 12	10.5	24	0.61	2.88	3.33	X	X	X
8 x 14 +*	8 x 13	11.2	24	0.70	3.14	3.63	X	X	X
8 x 16 +*	8 x 15	11.8	24	0.79	3.40	3.93	X	X	X
8 x 17 +*	8 x 17	12.3	24	0.87	3.67	4.24	X	X	X
8 x 19 +*	8 x 18	12.9	24	0.96	3.93	4.54	X	X	X
8 x 21 +	8 x 20	13.4	24	1.05	4.19	4.84	X	X	X

Nominal Size	Approx. Equiv. Rect.	*Equiv. Round	Gauge	Cross Sectional Area Sq Ft	Surface Area Per Lineal Foot Sq Ft/Ft	Wt. Per Lineal Foot Lbs/Ft	1” K-27Ò	2” K-27Ò	3” K-27Ò
8 x 22 +	8 x 22	13.9	24	1.13	4.45	5.14	X	X	X
8 x 24 +	8 x 23	14.3	24	1.22	4.71	5.44	X	X	X
8 x 25	8 x 25	14.7	22	1.31	4.97	6.99	X	X	X
8 x 27	8 x 26	15.1	22	1.40	5.24	7.37	X	X	X
8 x 30	8 x 30	15.9	22	1.57	5.76	8.10	X	X	X
8 x 32	8 x 31	16.3	22	1.66	6.02	8.46	X	X	X
8 x 33	8 x 33	16.6	22	1.75	6.28	8.83	X	X	X
8 x 35	8 x 34	17.0	22	1.83	6.54	9.20	X	X	
8 x 36	8 x 36	17.3	22	1.92	6.81	9.57	X	X	X
8 x 38	8 x 37	17.6	22	2.01	7.07	9.94	X		
8 x 39	8 x 39	17.9	22	2.09	7.33	10.31	X	X	X
8 x 43	8 x 42	18.5	22	2.27	7.85	11.04	X	X	X
8 x 46	8 x 45	19.1	22	2.44	8.38	11.78	X	X	X
8 x 49	8 x 49	19.7	20	2.62	8.90	14.74	X	X	X
8 x 52	8 x 52	20.2	20	2.79	9.42	15.60	X	X	X
8 x 55	8 x 55	20.7	20	2.97	9.95	16.48	X	X	X
8 x 58	8 x 58	21.1	20	3.14	10.47	17.34	X	X	X
8 x 61	8 x 61	21.6	20	3.32	11.00	18.22	X	X	X
8 x 65	8 x 64	22.1	20	3.49	11.52	19.08	X	X	X
8 x 68	8 x 68	22.5	20	3.67	12.04	19.94	X	X	X
8 x 71	8 x 71	22.9	18	3.84	12.57	27.10	X	X	X
10 x 45	10 x 44	21.5	22	2.95	8.38	11.78	X	X	X
10 x 51	10 x 50	22.7	20	3.38	9.42	15.60	X	X	X
10 x 54	10 x 54	23.3	20	3.60	9.95	16.48	X	X	X
10 x 57	10 x 57	23.9	20	3.82	10.47	17.34	X	X	X
10 x 60	10 x 60	24.4	20	4.04	11.00	18.22	X	X	X
10 x 63	10 x 63	25.0	20	4.25	11.52	19.08	X	X	X
10 x 67	10 x 66	25.5	20	4.47	12.04	19.94	X	X	X
10 x 70	10 x 69	26.0	20	4.69	12.57	20.82	X	X	X
10 x 73	10 x 73	26.4	18	4.91	13.09	28.22	X	X	X
10 x 76	10 x 76	26.9	18	5.13	13.61	29.34	X	X	
10 x 79	10 x 79	27.3	18	5.35	14.14	30.49	X		
11 x 14 *	11 x 13	12.8	24	0.90	3.40	3.93			
11 x 16 *	11 x 14	13.6	24	1.02	3.67	4.24			
11 x 17 *	11 x 16	14.3	24	1.14	3.93	4.54			
11 x 19	11 x 18	15.0	24	1.26	4.19	4.84			
11 x 20	11 x 19	15.7	24	1.38	4.45	5.14			
11 x 22	11 x 21	16.3	24	1.50	4.71	5.44			
11 x 24	11 x 22	16.8	24	1.62	4.97	5.75			
11 x 25	11 x 24	17.4	22	1.74	5.24	7.37			
12 x 14 +*	12 x 12	13.0	24	0.92	3.40	3.93			
12 x 15 +*	12 x 14	13.8	24	1.05	3.67	4.24			
12 x 17 +*	12 x 15	14.6	24	1.18	3.93	4.54		X	X
12 x 18 +	12 x 17	15.4	24	1.31	4.19	4.84	X	X	X
12 x 20 +	12 x 18	16.1	24	1.44	4.45	5.14	X	X	X
12 x 21 +	12 x 20	16.7	24	1.57	4.71	5.44	X	X	X
12 x 23 +	12 x 22	17.4	24	1.70	4.97	5.75	X	X	X

Nominal Size	Approx. Equiv. Rect.	*Equiv. Round	Gauge	Cross Sectional Area Sq Ft	Surface Area Per Lineal Foot Sq Ft/Ft	Wt. Per Lineal Foot Lbs/Ft	1” K-27Ò	2” K-27Ò	3” K-27Ò
12 x 25 +	12 x 23	18.0	22	1.83	5.24	7.37	X	X	X
12 x 26 +	12 x 25	18.5	22	1.96	5.50	7.73	X	X	X
14 x 34 +	14 x 33	22.9	22	3.05	7.07	9.94	X		
14 x 36 +	14 x 35	23.4	22	3.21	7.33	10.31	X	X	X
14 x 39 +	14 x 38	24.4	22	3.51	7.85	11.04	X	X	X
14 x 42	14 x 41	25.3	22	3.82	8.38	11.78	X	X	X
14 x 45	14 x 44	26.1	22	4.12	8.90	12.51	X	X	X
14 x 49	14 x 47	26.9	20	4.43	9.42	15.60	X	X	X
14 x 52	14 x 51	27.7	20	4.73	9.95	16.48	X	X	X
14 x 55	14 x 54	28.4	20	5.04	10.47	17.34	X	X	X
14 x 58	14 x 57	29.1	20	5.35	11.00	18.22	X	X	X
14 x 61	14 x 60	29.8	20	5.65	11.52	19.08	X	X	X
14 x 64	14 x 63	30.5	20	5.96	12.04	19.94	X	X	X
14 x 67	14 x 67	31.3	20	6.26	12.57	20.82	X	X	X
14 x 71	14 x 70	31.7	18	6.57	13.09	28.22	X	X	X
14 x 74	14 x 73	32.3	18	6.87	13.61	29.34	X	X	X
14 x 77	14 x 76	32.9	18	7.18	14.14	30.49	X	X	
14 x 80	14 x 79	33.4	18	7.48	14.66	31.61			
16 x 22	16 x 20	19.5	24	2.09	5.24	6.05	X	X	
16 x 24	16 x 22	20.3	24	2.27	5.50	6.35	X	X	
16 x 25	16 x 23	21.0	22	2.44	5.76	8.10	X	X	
16 x 27	16 x 25	21.7	22	2.62	6.02	8.46	X	X	
16 x 29	16 x 27	22.3	22	2.79	6.28	8.83	X	X	X
16 x 30	16 x 28	22.9	22	2.97	6.54	9.20	X	X	
16 x 32	16 x 30	23.6	22	3.14	6.81	9.51	X	X	X
16 x 33	16 x 32	24.1	22	3.32	7.07	9.94	X		
16 x 35	16 x 33	24.7	22	3.49	7.33	10.31	X	X	X
16 x 38	16 x 36	25.8	22	3.84	7.85	11.04	X	X	X
16 x 41	16 x 40	26.8	22	4.19	8.38	11.78	X	X	X
16 x 44	16 x 43	27.7	22	4.54	8.90	12.51	X	X	X
16 x 47	16 x 46	28.6	22	4.89	9.42	13.24	X	X	X
16 x 51	16 x 49	29.5	20	5.24	9.95	16.48	X	X	X
16 x 54	16 x 52	30.3	20	5.59	10.47	17.34	X	X	X
20 x 26	20 x 24	23.6	22	3.05	6.28	8.83			
20 x 28	20 x 25	24.4	22	3.27	6.54	9.20			
20 x 29	20 x 27	25.2	22	3.49	6.81	9.51			
20 x 31	20 x 28	25.9	22	3.71	7.07	9.94			
20 x 33	20 x 30	26.6	22	3.93	7.33	10.31	X	X	
20 x 36	20 x 33	27.9	22	4.36	7.85	11.04	X	X	
20 x 39	20 x 37	29.1	22	4.80	8.38	11.78	X	X	
20 x 42	20 x 40	30.3	22	5.24	8.90	12.51	X	X	
20 x 45	20 x 43	31.4	22	5.67	9.42	13.24	X	X	
20 x 48	20 x 46	32.5	22	6.11	9.95	13.99	X	X	
20 x 51	20 x 49	33.5	20	6.54	10.47	17.34	X	X	
20 x 55	20 x 53	34.4	20	6.98	11.00	18.22	X	X	
20 x 58	20 x 56	35.3	20	7.42	11.52	19.08	X	X	
20 x 61	20 x 59	36.2	20	7.85	12.04	19.94	X	X	

Nominal Size	Approx. Equiv. Rect.	*Equiv. Round	Gauge	Cross Sectional Area Sq Ft	Surface Area Per Lineal Foot Sq Ft/Ft	Wt. Per Lineal Foot Lbs/Ft	1” K-27Ò	2” K-27Ò	3” K-27Ò
20 x 64	20 x 62	37.1	20	8.29	12.57	20.82	X	X	
20 x 67	20 x 65	37.9	20	8.73	13.09	21.68	X	X	
20 x 70	20 x 69	38.7	20	9.16	13.61	22.54	X	X	
20 x 73	20 x 72	39.4	18	9.60	14.14	30.49	X		
20 x 77	20 x 75	40.2	18	10.04	14.66	31.61	X	X	
20 x 80	20 x 78	40.9	18	10.47	15.18	32.73	X		
22 x 35	22 x 32	28.7	22	4.56	7.85	11.04	X		
22 x 38	22 x 35	30.0	22	5.04	8.38	11.78	X		
22 x 41	22 x 38	31.3	22	5.52	8.90	12.51	X		
22 x 44	22 x 41	32.5	22	6.00	9.42	13.24	X		
22 x 47	22 x 45	33.7	22	6.48	9.95	13.99	X		
22 x 50	22 x 48	34.8	22	6.96	10.47	14.72	X		
22 x 53	22 x 51	35.8	20	7.44	11.00	18.22	X		
22 x 57	22 x 54	36.8	20	7.92	11.52	19.08	X		
22 x 60	22 x 58	37.8	20	8.40	12.04	19.94	X		
22 x 63	22 x 61	38.7	20	8.88	12.57	20.82	X		
22 x 66	22 x 64	39.6	20	9.36	13.09	21.68	X		
30 x 74	30 x 70	49.2	18	14.08	15.19	32.75			
30 x 77	30 x 73	50.1	18	14.70	15.69	33.83			
30 x 80	30 x 76	51.1	18	15.33	16.19	34.91			
30 x 83	30 x 80	52.0	18	15.95	16.69	35.95			

Notes: * = 6 –foot standard length
+ = available in UNI-RIB®

A.1.4 Duct Shape Conversion Tables

Table A.3 Round/Rectangular Equivalents

Duct Diameter (inches)	Rectangular Size (inches)	Aspect Ratio														
		1.00	**1.25**	**1.50**	**1.75**	**2.00**	**2.25**	**2.50**	**2.75**	**3.00**	**3.50**	**4.00**	**5.00**	**6.00**	**7.00**	**8.00**
6	Width		6													
	Height		5													
7	Width	6	8													
	Height	6	6													
8	Width	7	9	9	11											
	Height	7	7	6	6											
9	Width	8	9	11	11	12	14									
	Height	8	7	7	6	6	6									
10	Width	9	10	12	12	14	14	15	17							
	Height	9	8	8	7	7	6	6	6							
11	Width	10	11	12	14	14	16	18	17	18	21					
	Height	10	9	8	8	7	7	7	6	6	6					
12	Width	11	13	14	14	16	16	18	19	21	21	24				
	Height	11	10	9	8	8	7	7	7	7	6	6				
13	Width	12	14	15	16	18	18	20	19	21	25	24	30			
	Height	12	11	10	9	9	8	8	7	7	7	6	6			
14	Width	13	14	17	18	18	20	20	22	24	25	28	30	36		
	Height	13	11	11	10	9	9	8	8	8	7	7	6	6		
15	Width	14	15	17	18	20	20	23	25	24	28	28	35	36	42	
	Height	14	12	11	10	10	9	9	9	8	8	7	7	6	6	
16	Width	15	16	18	19	20	23	23	25	27	28	32	35	42	42	48
	Height	15	13	12	11	10	10	9	9	9	8	8	7	7	6	6
17	Width	16	18	20	21	22	25	25	28	27	32	32	35	42	49	48
	Height	16	14	13	12	11	11	10	10	9	9	8	7	7	7	6
18	Width	16	19	21	23	24	25	28	28	30	32	36	40	42	49	56
	Height	16	15	14	13	12	11	11	10	10	9	9	8	7	7	7
19	Width	17	20	21	23	24	27	28	30	30	35	36	40	48	49	56
	Height	17	16	14	13	12	12	11	11	10	10	9	8	8	7	7
20	Width	18	20	23	25	26	27	30	30	33	35	40	45	48	56	56
	Height	18	16	15	14	13	12	12	11	11	10	10	9	8	8	7
21	Width	19	21	24	26	28	29	30	33	33	39	40	45	54	56	64
	Height	19	17	16	15	14	13	12	12	11	11	10	9	9	8	8
22	Width	20	23	26	26	28	32	33	36	36	39	44	50	54	56	64
	Height	20	18	17	15	14	14	13	13	12	11	11	10	9	8	8
23	Width	21	24	26	28	30	32	35	36	39	42	44	50	54	63	64
	Height	21	19	17	16	15	14	14	13	13	12	11	10	9	9	8
24	Width	22	25	27	30	32	34	35	39	39	42	48	55	60	63	72
	Height	22	20	18	17	16	15	14	14	13	12	12	11	10	9	9
25	Width	23	25	29	30	32	36	38	39	42	46	48	55	60	70	72
	Height	23	20	19	17	16	16	15	14	14	13	12	11	10	10	9
26	Width	24	26	30	32	34	36	38	41	42	46	52	55	66	70	72
	Height	24	21	20	18	17	16	15	15	14	13	13	11	11	10	9
27	Width	25	28	30	33	36	38	40	41	45	49	52	60	66	70	80
	Height	25	22	20	19	18	17	16	15	15	14	13	12	11	10	10

		Aspect Ratio														
Duct Diameter (inches)	**Rectangular Size (inches)**	**1.00**	**1.25**	**1.50**	**1.75**	**2.00**	**2.25**	**2.50**	**2.75**	**3.00**	**3.50**	**4.00**	**5.00**	**6.00**	**7.00**	**8.00**
28	Width	26	29	32	35	36	38	43	44	45	49	56	60	66	77	80
	Height	26	23	21	20	18	17	17	16	15	14	14	12	11	11	10
29	Width	27	30	33	35	38	41	43	44	48	53	56	65	72	77	88
	Height	27	24	22	20	19	18	17	16	16	15	14	13	12	11	11
30	Width	27	31	35	37	40	43	45	47	48	53	60	65	72	77	88
	Height	27	25	23	21	20	19	18	17	16	15	15	13	12	11	11
31	Width	28	31	35	39	40	43	45	50	51	56	60	70	78	84	88
	Height	28	25	23	22	20	19	18	18	17	16	15	14	13	12	11
32	Width	29	33	36	39	42	45	48	50	54	56	60	70	78	84	96
	Height	29	26	24	22	21	20	19	18	18	16	15	14	13	12	12
33	Width	30	34	38	40	44	47	50	52	54	60	64	75	78	91	96
	Height	30	27	25	23	22	21	20	19	18	17	16	15	13	13	12
34	Width	31	35	39	42	44	47	50	52	57	60	64	75	84	91	96
	Height	31	28	26	24	22	21	20	19	19	17	16	15	14	13	12
35	Width	32	36	39	42	46	50	53	55	57	63	68	75	84	91	104
	Height	32	29	26	24	23	22	21	20	19	18	17	15	14	13	13
36	Width	33	36	41	44	48	50	53	55	60	63	68	80	90	98	104
	Height	33	29	27	25	24	22	21	20	20	18	17	16	15	14	13
38	Width	35	39	44	47	50	54	58	61	63	67	72	85	96	105	112
	Height	35	31	29	27	25	24	23	22	21	19	18	17	16	15	14
40	Width	37	41	45	49	52	56	60	63	66	70	76	90	96	105	120
	Height	37	33	30	28	26	25	24	23	22	20	19	18	16	15	15
42	Width	38	43	48	51	56	59	63	66	69	74	80	90	102	112	120
	Height	38	34	32	29	28	26	25	24	23	21	20	18	17	16	15
44	Width	40	45	50	54	58	61	65	69	72	81	84	95	108	119	128
	Height	40	36	33	31	29	27	26	25	24	23	21	19	18	17	16
46	Width	42	48	53	56	60	65	68	72	75	84	88	100	114	126	136
	Height	42	38	35	32	30	29	27	26	25	24	22	20	19	18	17
48	Width	44	49	54	60	62	68	70	74	78	88	92	105	120	126	136
	Height	44	39	36	34	31	30	28	27	26	25	23	21	20	18	17
50	Width	46	51	57	61	66	70	75	77	81	91	96	110	120	133	144
	Height	46	41	38	35	33	31	30	28	27	26	24	22	20	19	18
52	Width	48	54	59	63	68	72	78	83	84	95	100	115	126	140	152
	Height	48	43	39	36	34	32	31	30	28	27	25	23	21	20	19
54	Width	49	55	62	67	70	77	80	85	90	98	104	120	132	147	160
	Height	49	44	41	38	35	34	32	31	30	28	26	24	22	21	20
56	Width	51	58	63	68	74	79	83	88	93	102	108	125	138	147	160
	Height	51	46	42	39	37	35	33	32	31	29	27	25	23	21	20
58	Width	53	60	66	70	76	81	85	91	96	105	112	130	144	154	168
	Height	53	48	44	40	38	36	34	33	32	30	28	26	24	22	21
60	Width	55	61	68	74	78	83	90	94	99	109	116	130	144	161	
	Height	55	49	45	42	39	37	36	34	33	31	29	26	24	23	
62	Width	57	64	71	75	82	88	93	96	102	112	120	135	150	168	
	Height	57	51	47	43	41	39	37	35	34	32	30	27	25	24	
64	Width	59	65	72	79	84	90	95	99	105	116	124	140	156		
	Height	59	52	48	45	42	40	38	36	35	33	31	28	26		
66	Width	60	68	75	81	86	92	98	105	108	119	128	145	162		
	Height	60	54	50	46	43	41	39	38	36	34	32	29	27		
68	Width	62	70	77	82	90	95	100	107	111	123	132	150	168		
	Height	62	56	51	47	45	42	40	39	37	35	33	30	28		

Duct Diameter (inches)	Rectangular Size (inches)	Aspect Ratio														
		1.00	**1.25**	**1.50**	**1.75**	**2.00**	**2.25**	**2.50**	**2.75**	**3.00**	**3.50**	**4.00**	**5.00**	**6.00**	**7.00**	**8.00**
70	Width	64	71	80	86	92	99	105	110	114	126	136	155			
	Height	64	57	53	49	46	44	42	40	38	36	34	31			
72	Width	66	74	81	88	94	101	108	113	117	130	140	160			
	Height	66	59	54	50	47	45	43	41	39	37	35	32			
74	Width	68	76	84	91	98	104	110	116	123	133	144	165			
	Height	68	61	56	52	49	46	44	42	41	38	36	33			
76	Width	70	78	86	93	100	106	113	118	126	137	148	165			
	Height	70	62	57	53	50	47	45	43	42	39	37	33			
78	Width	71	80	89	95	102	110	115	121	129	140	152				
	Height	71	64	59	54	51	49	46	44	43	40	38				
80	Width	73	83	90	98	104	113	118	124	132	144	156				
	Height	73	66	60	56	52	50	47	45	44	41	39				
82	Width	75	84	93	100	108	115	123	129	135	147	160				
	Height	75	67	62	57	54	51	49	47	45	42	40				
84	Width	77	86	95	103	110	117	125	132	138	151	164				
	Height	77	96	63	59	55	52	50	48	46	43	41				
86	Width	79	88	98	105	112	119	128	135	141	154	168				
	Height	79	70	65	60	56	53	51	49	47	44	42				
88	Width	80	90	99	107	116	124	130	138	144	158					
	Height	80	72	66	61	58	55	52	50	48	45					
90	Width	82	93	102	110	118	126	133	140	147	161					
	Height	82	74	68	63	59	56	53	51	49	46					
92	Width	84	94	104	112	120	128	138	143	150	165					
	Height	84	75	69	64	60	57	55	52	50	47					
94	Width	86	96	107	116	124	131	140	146	153	168					
	Height	86	77	71	66	62	58	56	53	51	48					
96	Width	88	99	108	117	126	135	143	151	159						
	Height	88	79	72	67	63	60	57	55	53						
98	Width	90	100	111	119	128	137	145	154	162						
	Height	90	80	74	68	64	61	58	56	54						
100	Width	91	103	113	123	132	140	148	157	165						
	Height	91	82	75	70	66	62	59	57	55						
102	Width	93	105	116	124	134	142	153	160	168						
	Height	93	84	77	71	67	63	61	58	56						
104	Width	95	106	117	128	136	146	155	162							
	Height	95	85	78	73	68	65	62	59							

A.1.5 Nonstandard Condition Correction Factors

Correction Factors for Temperature

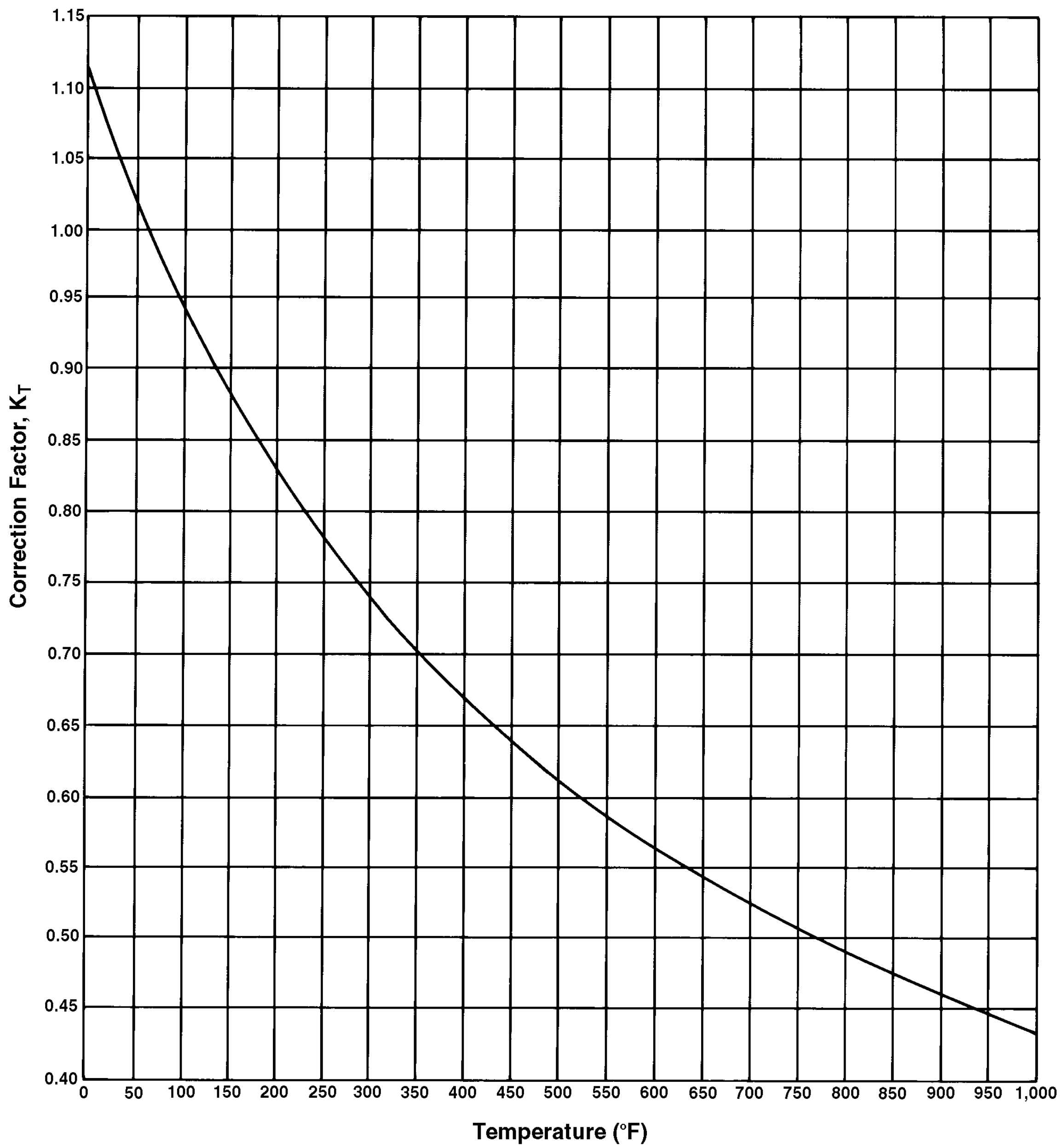


Figure A.1 Correction Factors for Temperature

Correction Factors for Elevation

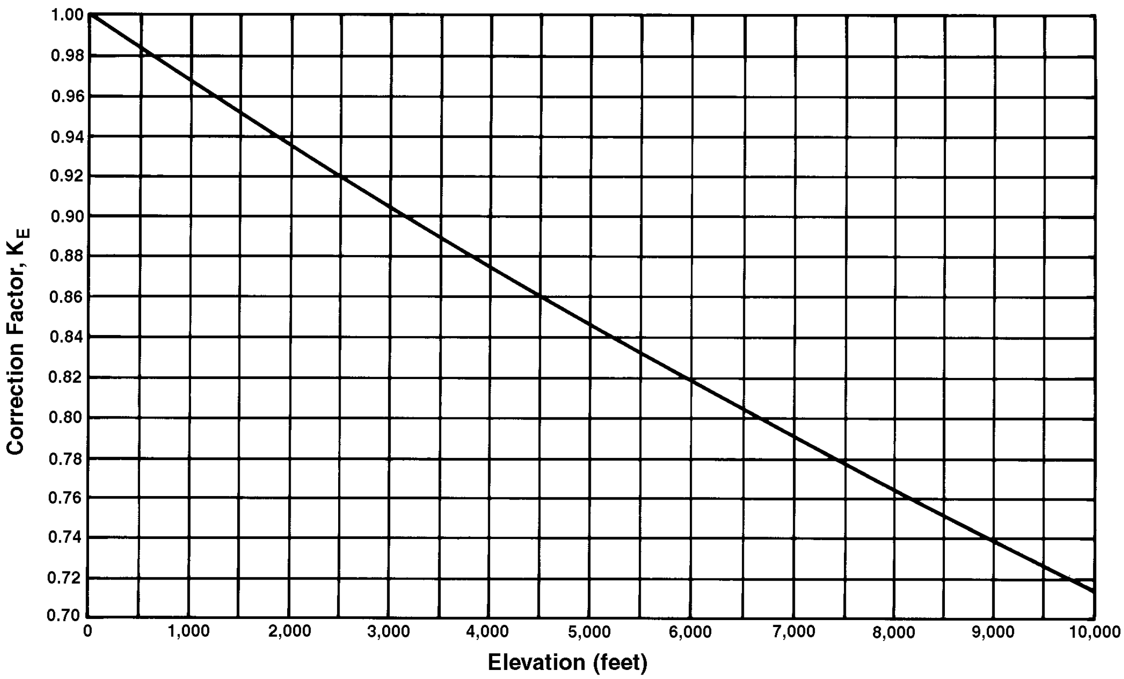


Figure A.2 Correction Factors for Elevation

Table A.4 Correction Factors for Density(p_a /p_{std})

Altitude (ft)		**Sea Level**	**1,000**	**2,000**	**3,000**	**4,000**	**5,000**	**6,000**	**7,000**	**8,000**	**9,000**	**10,000**
Barometer (Hg)		29.92	28.86	27.82	26.82	25.84	24.90	23.98	23.09	22.22	21.39	20.58
(inches wg)		407.5	392.8	378.6	365.0	351.7	338.9	326.4	314.3	302.1	291.1	280.1
AIR TEMPERATURE (°F)	-40°	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.93	0.90	0.87
	0°	1.15	1.11	1.07	1.03	0.99	0.95	0.91	0.89	0.85	0.82	0.79
	40°	1.06	1.02	0.99	0.95	0.92	0.88	0.85	0.82	0.79	0.76	0.73
	70°	1.00	0.96	0.93	0.89	0.86	0.83	0.80	0.77	0.74	0.71	0.69
	100°	0.95	0.92	0.88	0.85	0.81	0.78	0.75	0.73	0.70	0.68	0.65
	150°	0.87	0.84	0.81	0.78	0.75	0.72	0.69	0.67	0.65	0.62	0.60
	200°	0.80	0.77	0.74	0.71	0.69	0.66	0.64	0.62	0.60	0.57	0.55
	250°	0.75	0.72	0.70	0.67	0.64	0.62	0.60	0.58	0.56	0.53	0.51
	300°	0.70	0.67	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.50	0.48
	350°	0.65	0.62	0.60	0.58	0.56	0.54	0.52	0.51	0.49	0.47	0.45
	400°	0.62	0.60	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.44	0.42
	450°	0.58	0.56	0.54	0.52	0.50	0.48	0.46	0.45	0.43	0.42	0.40
	500°	0.55	0.53	0.51	0.49	0.47	0.45	0.44	0.43	0.41	0.39	0.38
	550°	0.53	0.51	0.49	0.47	0.45	0.44	0.42	0.41	0.39	0.38	0.36
	600°	0.50	0.48	0.46	0.45	0.43	0.41	0.40	0.39	0.37	0.35	0.34
	700°	0.46	0.44	0.43	0.41	0.39	0.38	0.37	0.35	0.34	0.33	0.32
	800°	0.42	0.40	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29
	900°	0.39	0.37	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27
	1000°	0.36	0.35	0.33	0.32	0.31	0.30	0.29	0.28	0.27	0.26	0.25

A.1.6 Velocity and Velocity Pressure

Table A.5
Velocity Pressures and Corresponding Velocities at Standard Conditions

VP (inches wg)	V (fpm)	VP (inches wg)	V (fpm)	VP (inches wg)	V (fpm)
0.01	400	0.35	2,369	0.69	3,327
0.02	566	0.36	2,403	0.70	3,351
0.03	694	0.37	2,436	0.71	3,375
0.04	801	0.38	2,439	0.72	3,398
0.05	896	0.39	2,501	0.73	3,422
0.06	981	0.40	2,533	0.74	3,445
0.07	1,060	0.41	2,564	0.75	3,468
0.08	1,133	0.42	2,596	0.76	3,491
0.09	1,202	0.43	2,626	0.77	3,514
0.10	1,266	0.44	2,657	0.78	3,537
0.11	1,328	0.45	2,687	0.79	3,560
0.12	1,387	0.46	2,716	0.80	3,582
0.13	1,444	0.47	2,746	0.81	3,605
0.14	1,499	0.48	2,775	0.82	3,627
0.15	1,551	0.49	2,804	0.83	3,649
0.16	1,602	0.50	2,832	0.84	3,671
0.17	1,651	0.51	2,860	0.85	3,692
0.18	1,699	0.52	2,888	0.86	3,714
0.19	1,746	0.53	2,916	0.87	3,736
0.20	1,791	0.54	2,943	0.88	3,757
0.21	1,835	0.55	2,970	0.89	3,778
0.22	1,879	0.56	2,997	0.90	3,799
0.23	1,920	0.57	3,024	0.91	3,821
0.24	1,962	0.58	3,050	0.92	3,841
0.25	2,003	0.59	3,076	0.93	3,862
0.26	2,042	0.60	3,102	0.94	3,883
0.27	2,081	0.61	3,128	0.95	3,904
0.28	2,119	0.62	3,154	0.96	3,924
0.29	2,157	0.63	3,179	0.97	3,944
0.30	2,194	0.64	3,204	0.98	3,965
0.31	2,230	0.65	3,229	0.99	3,985
0.32	2,266	0.66	3,254	1.00	4,005
0.33	2,301	0.67	3,278	1.02	4,045
0.34	2,335	0.68	3,303		

A.2 Product Information

A.2.1 Metal Properties

A.2.1.1 Gauge, Thickness, and Weight

Table A.6 Approximate Weight and Thickness of Steel and Aluminum

Gauge	Galvanized and Paintable Galvanized Steel		Nongalvanized Carbon Steel		Stainless Steel (304 or 316)		Aluminum 3003-H14*	
	Nominal Thickness (Inches)	Nominal Weight (lb/sq ft)	Nominal Thickness (Inches)	Nominal Weight (lb/sq ft)	Nominal Thickness (Inches)	Nominal Weight (lb/sq ft)	Nominal Thickness (Inches)	Nominal Weight (lb/sq ft)
28	0.0187	0.781	0.0149	0.625	0.0156	0.656	0.025	0.356
26	0.0217	0.906	0.0179	0.750	0.0188	0.788	0.032	0.456
24	0.0276	1.156	0.0239	1.000	0.0250	1.050	0.040	0.570
22	0.0336	1.406	0.0299	1.250	0.0313	1.313	0.050	0.713
20	0.0396	1.656	0.0359	1.500	0.0375	1.575	0.063	0.898
18	0.0516	2.156	0.0478	2.000	0.0500	2.100	0.080	1.140
16	0.0635	2.656	0.0598	2.500	0.0625	2.625	0.090	1.283
14	0.0785	3.281	0.0747	3.125	0.0781	3.281		
12	0.1084	4.531	0.1046	4.375	0.1094	4.594		
10	0.1382	5.781	0.1345	5.625	0.1406	5.906		

* Nominal thickness is based on approximately 1.5 times the galvanized gauge thickness for equal strength and stiffness.

Table A.7 Approximate Weight of UNI-SEAL™ Spiral Round Duct

Nominal Diameter (inches)	Cross-Sectional Area (sq ft)	10" wg SMACNA		28 Gauge (0.019 inch)	26 Gauge (0.022 inch)	24 Gauge (0.028 inch)	22 Gauge (0.034 inch)	20 Gauge (0.040 inch)	18 Gauge (0.052 inch)	16 Gauge (0.064 inch)	14 Gauge (0.080 inch)
		Standard Gauge	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)
3	0.05	28	0.7	0.7	0.8	1.1	1.3	1.5	2.0	—	—
3.5	0.07	28	0.8	0.8	1.0	1.2	1.5	1.8	2.3	—	—
4	0.09	28	1.0	1.0	1.1	1.4	1.7	2.0	2.6	—	—
4.5	0.11	28	1.1	1.1	1.2	1.6	1.9	2.3	2.9	—	—
5	0.14	28	1.2	1.2	1.4	1.8	2.1	2.5	3.3	—	—
5.5	0.16	28	1.3	1.3	1.5	1.9	2.4	2.8	3.6	—	—
6	0.20	28	1.4	1.4	1.7	2.1	2.6	3.0	3.9	4.8	—
6.5	0.23	28	1.5	1.5	1.8	2.3	2.8	3.3	4.3	5.2	—
7	0.27	28	1.7	1.7	1.9	2.5	3.0	3.5	4.6	5.6	—
7.5	0.31	28	1.8	1.8	2.1	2.6	3.2	3.8	4.9	6.0	—
8	0.35	28	1.9	1.9	2.2	2.8	3.4	4.0	5.2	6.4	—
8.5	0.39	28	2.0	2.0	2.3	3.0	3.6	4.3	5.6	6.8	—
9	0.44	28	2.1	2.1	2.5	3.2	3.9	4.5	5.9	7.2	—
9.5	0.49	28	2.3	2.3	2.6	3.3	4.1	4.8	6.2	7.6	—

Nominal Diameter (inches)	Cross-Sectional Area (sq ft)	10" wg SMACNA		28 Gauge (0.019 inch)	26 Gauge (0.022 inch)	24 Gauge (0.028 inch)	22 Gauge (0.034 inch)	20 Gauge (0.040 inch)	18 Gauge (0.052 inch)	16 Gauge (0.064 inch)	14 Gauge (0.080 inch)
		Standard Gauge	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)
10	0.55	28	2.3	2.3	2.7	3.4	4.1	4.9	6.5	8.0	—
10.5	0.60	28	2.4	2.4	2.8	3.6	4.3	5.1	6.9	8.4	—
11	0.66	28	2.5	2.5	2.9	3.7	4.5	5.4	7.2	8.8	—
11.5	0.72	28	2.6	2.6	3.1	3.9	4.8	5.6	7.5	9.2	—
12	0.79	28	2.8	2.8	3.2	4.1	5.0	5.8	7.8	9.6	—
12.5	0.85	28	2.9	2.9	3.3	4.3	5.2	6.1	8.2	10.0	—
13	0.92	28	3.0	3.0	3.5	4.4	5.4	6.3	8.5	10.4	—
13.5	0.99	28	3.1	3.1	3.6	4.6	5.6	6.6	8.8	10.8	—
14	1.07	28	3.2	3.2	3.7	4.8	5.8	6.8	9.2	11.2	—
14.5	1.15	26	3.9	3.3	3.9	4.9	6.0	7.1	9.5	11.6	—
15	1.23	26	4.0	3.4	4.0	5.1	6.2	7.3	9.8	12.0	—
16	1.40	26	4.3	3.7	4.3	5.4	6.6	7.8	10.5	12.8	—
17	1.58	26	4.5	3.9	4.5	5.8	7.0	8.3	11.1	13.6	—
18	1.77	26	4.8	4.1	4.8	6.1	7.4	8.8	11.8	14.4	—
19	1.97	26	5.1	4.4	5.1	6.5	7.9	9.3	12.4	15.2	—
20	2.18	26	5.3	4.6	5.3	6.8	8.3	9.7	13.1	16.1	—
21	2.41	26	5.6	4.8	5.6	7.1	8.7	10.2	13.7	16.9	—
22	2.64	26	5.9	5.1	5.9	7.5	9.1	10.7	14.4	17.7	—
23	2.89	26	6.1	5.3	6.1	7.8	9.5	11.2	15.0	18.5	—
24	3.14	26	6.4	—	6.4	8.2	9.9	11.7	15.7	19.3	23.6
25	3.41	24	8.5	—	6.7	8.5	10.3	12.2	16.3	20.1	24.6
26	3.69	24	8.8	—	6.9	8.8	10.7	12.7	17.0	20.9	25.6
27	3.98	24	9.2	—	—	9.2	11.2	13.1	17.7	21.7	26.6
28	4.28	24	9.5	—	—	9.5	11.6	13.6	18.3	22.5	27.6
29	4.59	24	9.9	—	—	9.9	12.0	14.1	19.0	23.3	28.5
30	4.91	24	10.2	—	—	10.2	12.4	14.6	19.6	24.1	29.5
31	5.24	24	10.5	—	—	10.5	12.8	15.1	20.3	24.9	30.5
32	5.59	24	10.9	—	—	10.9	13.2	15.6	20.9	25.7	31.5
33	5.94	24	11.2	—	—	11.2	13.6	16.1	21.6	26.5	32.5
34	6.31	24	11.6	—	—	11.6	14.1	16.6	22.2	27.3	33.5
35	6.68	24	11.9	—	—	11.9	14.5	17.0	22.9	28.1	34.4
36	7.07	24	12.2	—	—	12.2	14.9	17.5	23.5	28.9	35.4
37	7.47	24	12.6	—	—	12.6	15.3	18.0	24.2	29.7	36.4
38	7.88	24	12.9	—	—	12.9	15.7	18.5	24.8	30.5	37.4
40	8.73	24	13.6	—	—	13.6	16.5	19.5	26.1	32.1	39.4
42	9.62	24	14.3	—	—	14.3	17.4	20.4	27.5	33.7	41.3
44	10.56	22	18.2	—	—	—	18.2	21.4	28.8	35.3	43.3
46	11.54	22	19.0	—	—	—	19.0	22.4	30.1	36.9	45.3
48	12.57	22	19.8	—	—	—	19.8	23.4	31.4	38.5	47.2

Nominal Diameter (inches)	Cross-Sectional Area (sq ft)	10" wg SMACNA		28 Gauge (0.019 inch)	26 Gauge (0.022 inch)	24 Gauge (0.028 inch)	22 Gauge (0.034 inch)	20 Gauge (0.040 inch)	18 Gauge (0.052 inch)	16 Gauge (0.064 inch)	14 Gauge (0.080 inch)
		Standard Gauge	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)	Nominal Weight (lb/ft)
50	13.64	22	20.7	—	—	—	20.7	24.3	32.7	40.1	49.2
52	14.75	22	21.5	—	—	—	21.5	25.3	34.0	41.7	51.2
54	15.90	22	22.3	—	—	—	22.3	26.3	35.3	43.3	53.1
56	17.10	22	23.1	—	—	—	23.1	27.3	36.6	44.9	55.1
58	18.35	22	24.0	—	—	—	24.0	28.2	37.9	46.5	57.1
60	19.63	22	24.8	—	—	—	24.8	29.2	39.2	48.1	59.0
62	20.97	18	40.5	—	—	—	26.5	31.2	40.5	49.7	61.0
64	22.34	18	41.8	—	—	—	27.4	32.2	41.8	51.4	63.0
66	23.76	18	43.1	—	—	—	28.2	33.2	43.1	53.0	64.9
68	25.22	18	44.4	—	—	—	—	34.2	44.4	54.6	66.9
70	26.73	18	45.8	—	—	—	—	35.3	45.8	56.2	68.9
72	28.27	18	47.1	—	—	—	—	36.3	47.1	57.8	70.8
74	29.87	18	48.4	—	—	—	—	37.3	48.4	59.4	72.8
76	31.50	18	49.7	—	—	—	—	38.3	49.7	61.0	74.8
78	33.18	18	51.0	—	—	—	—	39.3	51.0	62.6	76.7
80	34.91	18	52.3	—	—	—	—	40.3	52.3	64.2	78.7
82	36.67	18	53.6	—	—	—	—	41.3	53.6	65.8	80.7
84	38.48	18	54.9	—	—	—	—	42.3	54.9	67.4	82.7
86	40.34	18	56.2	—	—	—	—	43.3	56.2	69.0	84.6
88	42.24	18	57.5	—	—	—	—	44.3	57.5	70.6	86.6

A.2.1.2 ASTM Standards

A.2.1.2.1 **ASTM A924** General Requirements for Steel Sheet, Metallic - Coated by the Hot-Dip Process.

A.2.1.2.2 **ASTM A653** Steel Sheet, Zinc-Coated (Galvanized) or Zinc-Iron Alloy-Coated (Galvannealed) by the Hot-Dip Process.

A.2.1.2.3 **ASTM A167** Specification for Stainless and Heat-Resisting Chromium-Nickel Plate, Sheet, and Strip.

A.2.1.2.4 **ASTM A480** General Requirements for Flat-Rolled Stainless Heat-Resisting Steel Plate, Sheet and Strip.

A.2.1.2.5 **ASTM B209** Specification for aluminum and aluminum-alloy sheet and plate.

A.2.1.2.6 **ASTM A366** Specification for steel sheet, carbon, cold rolled, commercial quality.

A.2.1.2.7 **ASTM A619** Specification for steel sheet, carbon, cold rolled, drawing quality.

A.2.1.2.8 **ASTM E477** Standard Test Method for measuring acoustical and airflow performance of duct liner materials and prefabricated silencers.

A.2.2 United McGill Corporation Manufacturing Standards

Table A.8 Single-Wall, Round Duct and Fittings

Single-Wall, Round Duct¾ Available Sizes, Materials, and Thicknesses				
Construction	**Diameters**	**Lengths**[1]	**Materials**[2,3]	**Thicknesses**
UNI-SEAL™ Duct (spiral lockseam)	3-84 inches	1-20 feet	Galvanized Steel	28-14 gauge
			Stainless Steel	26-20 gauge
	3-60 inches[4]		Aluminum	0.025-0.063 inch
UNI-RIB® Duct (spiral lockseam with standing rib)	9-60 inches	1-20 feet	Galvanized Steel	28-26 gauge
			Aluminum	0.025-0.032 inch
Longitudinal Seam Duct (solid welded)	8-90 inches	1-5 feet	Galvanized Steel	20-10 gauge
			Stainless Steel	22-10 gauge
			Aluminum	0.040-0.080 inch

[1] Standard lengths of round UNI-SEAL™ duct and UNI-RIB® duct are 10, 12, and 20 feet; longer lengths are available. Standard lengths of Longitudunal Seam duct are 4 and 5 feet.
[2] Round duct is also available in nongalvanized carbon steel and paintable galvanized steel. Stainless steel is generally type 304 or 316. Aluminum is generally type 3003-H14.
[3] UNI-COAT® duct (polyvinyl-chloride-coated galvanized steel) is available in diameters of 4-84 inches.
[4] Aluminum UNI-SEAL™ duct is available with diameters greater than 60 inches on special order.

Single-Wall, Round Fittings¾ Available Sizes, Materials, and Thicknesses			
Construction[1]	**Diameters**	**Materials**[2,3]	**Thicknesses**
UNI-SEAL™ Fittings (solid welded, spot welded and bonded, or standing seam)	3-90 inches	Galvanized Steel	26-10 gauge
		Stainless Steel	26-20 gauge
		Aluminum	0.040-0.080 inch

[1] Includes the fitting line formerly known as United McGill's UNI-WELD™ fittings. This fitting line is available through 60 inches in diameter (galvanized steel only).
[2] Round fittings are also available in nongalvanized carbon steel and paintable galvanized steel. Stainless steel is generally type 304 or 316. Aluminum is generally type 3003-H14.
[3] UNI-COAT® fittings (polyvinyl-chloride-coated galvanized steel) are available in diameters of 4-60 inches. Diameters greater than 60 inches are available on special order.

Table A.9 Double-Wall, Round Duct and Fittings

Double-Wall, Round Duct¾ Available Sizes, Materials, and Thicknesses

Construction	Inner Liner Diameters — Insulation Thickness (inches): 1	2	3	Lengths[1]	Materials[2,3]	Outer Shell Thicknesses
ACOUSTI-k27® Duct (spiral lockseam)	3-82 inches	3-80 inches	3-78 inches	1-20 feet	Galvanized Steel	28-14 gauge
					Stainless Steel	26-20 gauge
	3-58 inches[4]	3-56 inches[4]	3-54 inches[4]		Aluminum	0.025-0.063 inch
UNI-RIB-k27® Duct (spiral lockseam with standing rib)	7-58 inches	5-56 inches	3-54 inches	1-20 feet	Galvanized Steel	28-26 gauge
					Aluminum	0.025-0.032 inch
Longitudinal Seam-k27® Duct (solid welded)	6-88 inches	4-86 inches	3-84 inches	1-5 feet	Galvanized Steel	20-10 gauge
					Stainless Steel	22-10 gauge
					Aluminum	0.040-0.080 inch

[1] Standard length of round ACOUSTI-k27® duct and UNI-RIB-k27® duct are 10, 12, and 20 feet; longer lengths are available. Standard lengths of Longitudunal Seam-k27® duct are 4 and 5 feet.
[2] Round duct is also available in nongalvanized carbon steel and paintable galvanized steel. Stainless steel is generally type 304 or 316. Aluminum is generally type 3003-H14.
[3] UNI-COAT-k27® duct (polyvinyl-chloride-coated galvanized steel) is available on special order.
[4] Aluminum ACOUSTI-k27® duct is available with outside diameters greater than 60 inches on special order.

Double-Wall, Round Fittings¾ Available Sizes, Materials, and Thicknesses

Construction[1]	Inner Liner Diameters — Insulation Thickness (inches): 1	2	3	Materials[2,3]	Outer Shell Thicknesses
ACOUSTI-k27® Fittings (solid welded, spot welded and bonded, or standing seam)	3-88 inches	3-86 inches	3-84 inches	Galvanized Steel	26-10 gauge
				Stainless Steel	26-10 gauge
	3-58 inches[4]	3-56 inches[4]	3-54 inches[4]	Aluminum	0.040-0.080 inch

[1] Includes the fitting line formerly known as United McGill's UNI-WELD-k27® fittings. This fitting line is available through 60 inches in diameter (outer shell) and only with galvanized steel outer shells.
[2] Round fittings are also available in nongalvanized carbon steel and paintable galvanized steel. Stainless steel is generally type 304 or 316. Aluminum is generally type 3003-H14.
[3] UNI-COAT-k27® fittings (polyvinyl-chloride-coated galvanized steel) are available on special order.
[4] Aluminum ACOUSTI-k27® fittings are available with outside diameters greater than 60 inches on special order.

Table A.10 Single-Wall, Flat Oval Duct and Fittings

Single-Wall, Flat Oval Duct¾ Available Lengths, Materials, and Thicknesses			
Construction	**Lengths[1]**	**Materials[2]**	**Thicknesses**
UNI-SEAL™ Duct (spiral lockseam)	1-12 feet	Galvanized Steel	28-18 gauge
		Stainless Steel	26-20 gauge
		Aluminum	0.025-0.063 inch
Longitudinal Seam Duct (solid welded)	1-5 feet	Galvanized Steel	20-10 gauge
		Stainless Steel	22-10 gauge
		Aluminum	0.040-0.080 inch

[1] Some spiral flat oval sizes are available only to lengths of 6 feet. Other sizes may be available in lenghts greater than 12 feet.
[2] Duct is also available in nongalvanized carbon steel and paintable steel. Stainless steel is generally type 304 or 316. Aluminum is generally type 3003-H14.

Single-Wall, Flat Oval Fittings¾ Available Materials, and Thicknesses		
Construction[1]	**Materials[2]**	**Thicknesses**
UNI-SEAL™ Fittings (solid welded, spot welded and bonded, or standing seam)	Galvanized Steel	26-10 gauge
	Stainless Steel	22-10 gauge
	Aluminum	0.040-0.080 inch

[1] Includes the fitting line formerly known as United McGill's UNI-WELD™ fittings. This fitting line is available in galvanized steel only.
[2] Fittings are also available in nongalvanized carbon steel and paintable galvanized steel. Stainless steel is generally type 304 or 316. Aluminum is generally type 3003-H14.

Table A.11 Double-Wall, Flat Oval Duct and Fittings

Double-Wall, Flat Oval Duct¾ Available Lengths, Materials, and Thicknesses			
Construction	**Lengths[1]**	**Materials[2]**	**Thicknesses**
ACOUSTI-k27® Duct (spiral lockseam)	1-12 feet	Galvanized Steel	28-18 gauge
		Stainless Steel	26-20 gauge
		Aluminum	0.025-0.063 inch
Longitudinal Seam-k27® Duct (solid welded)	1-5 feet	Galvanized Steel	20-10 gauge
		Stainless Steel	22-10 gauge
		Aluminum	0.040-0.080 inch

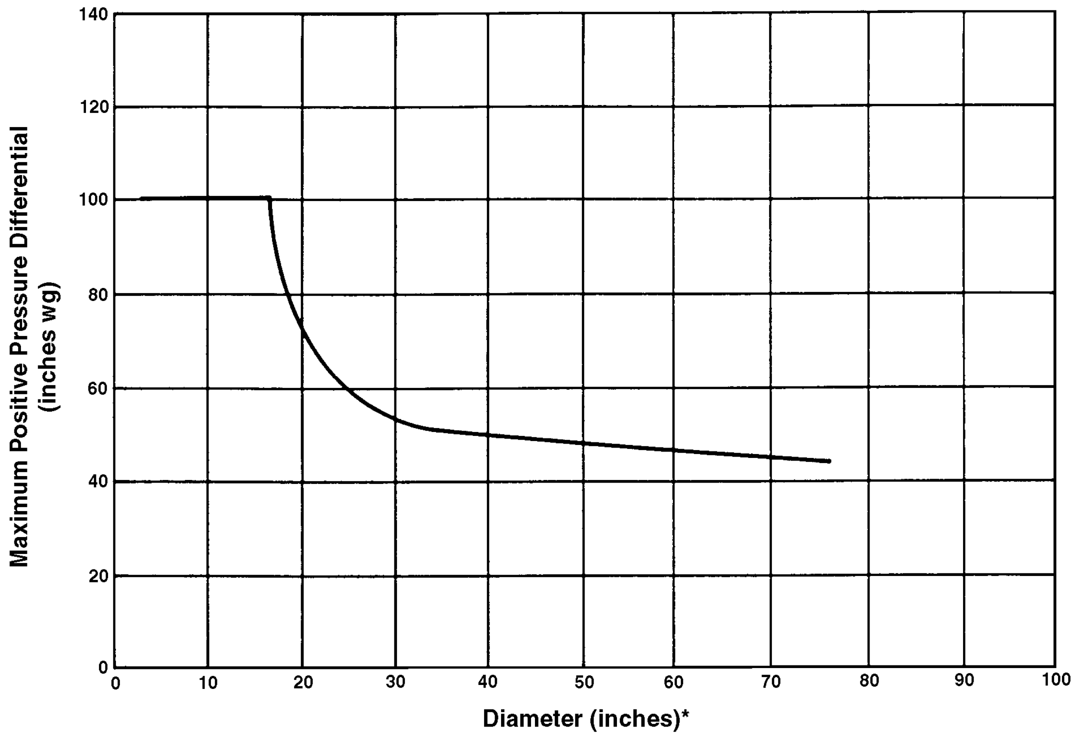
[1] Some spiral flat oval sizes are available only to lengths of 6 feet. Other sizes may be available in lengths greater than 12 feet. Standard lengths on Longitudinal Seam-k27® duct are 4 and 5 feet.
[2] Duct is also available in nongalvanized carbon steel and paintable steel. Stainless steel is generally type 304 or 316. Aluminum is generally type 3003-H14.

Double-Wall, Flat Oval Fittings¾ Available Materials, and Thicknesses		
Construction[1]	**Materials[2]**	**Thicknesses**
ACOUSTI-k27® Fittings (solid welded, spot welded and bonded, or standing seam)	Galvanized Steel	26-10 gauge
	Stainless Steel	22-10 gauge
	Aluminum	0.040-0.080 inch

[1] Includes the fitting line formerly known as United McGill's UNI-WELD-k27® fittings. This fitting line is available in galvanized steel outer shells.
[2] Fittings are also available in nongalvanized carbon steel and paintable galvanized steel. Stainless steel is generally type 304 or 316. Aluminum is generally type 3003-H14.

Table A.12 Suggested Maximum Positive Pressure Differential for UNI-SEAL™ Spiral Round Duct and Fittings

Suggested Maximum Positive Pressure Differential for UNI-SEAL™ Spiral Round Duct and Fittings



1. These recommendations refer to the use of high-pressure duct and fittings (steel only).
2. Systems that handle elevated positive pressure differentials require special installation practices.
3. Avoid large flat surfaces (for example, end caps) at elevated pressures even if the pressure/diameter relationship lies within the lower bounds of the above curve (a 60-inch diameter end cap at 46 inches wg positive pressure differential has a distributed force over its flat face of 4,695 pounds).
4. System leakage rates will exceed those normally expected at typical positive pressure differentials.

*Use the outer shell dimension for double-wall duct.

Table A.13 Reinforcement Specifications for Flat Oval Duct

MINOR	MAJOR	0.5 in.wg.	1 in.wg.	2 in.wg.	3 in.wg.	4 in.wg.	6 in.wg.	10 in.wg.
3	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
4	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
5	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
6	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
7	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
8	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
9	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
10	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	8	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
4	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
5	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
6	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
7	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
8	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
9	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
10	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	9	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
4	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
5	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
6	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
7	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
8	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
9	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
10	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"

14	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	10	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
4	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
5	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
6	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
7	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
8	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
9	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
10	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	11	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	12	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
4	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
5	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
6	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
7	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
8	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
9	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
10	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
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20	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	12	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
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4	13	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
5	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
6	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
7	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
8	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
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10	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"

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14	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
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18	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
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22	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	13	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	14	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
4	14	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
5	14	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
6	14	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
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14	14	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
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22	14	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	14	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	15	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
4	15	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
5	15	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
6	15	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
7	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
8	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
9	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
10	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
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20	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	15	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	16	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
4	16	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
5	16	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
6	16	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
7	16	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
8	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"

9	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
10	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	16	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	17	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
4	17	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
5	17	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
6	17	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
7	17	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
8	17	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
9	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
10	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	17	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	18	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
4	18	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
5	18	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
6	18	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
7	18	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
8	18	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
9	18	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
10	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
11	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	18	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	19	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
4	19	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
5	19	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
6	19	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"

7	19	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
8	19	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
9	19	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
10	19	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
11	19	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
12	19	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	19	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	19	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	19	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	19	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	19	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	19	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	20	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"
4	20	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
5	20	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
6	20	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
7	20	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
8	20	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
9	20	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
10	20	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
11	20	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
12	20	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
14	20	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	20	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	20	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	20	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	20	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	20	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	21	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"
4	21	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"
5	21	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
6	21	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
7	21	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
8	21	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
9	21	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
10	21	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
11	21	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
12	21	"NR"	"NR"	"NR"	"NR"	"B6"	"NR"	"C4"
14	21	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	21	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	21	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	21	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	21	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	21	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	22	"NR"	"C10"	"C6"	"D5"	"E5"	"E4"	"E2.5"
4	22	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"

5	22	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"
6	22	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
7	22	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
8	22	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
9	22	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
10	22	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
11	22	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"
12	22	"NR"	"NR"	"NR"	"NR"	"B6"	"NR"	"C4"
14	22	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
16	22	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	22	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	22	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	22	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	22	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	23	"NR"	"C10"	"C6"	"D5"	"E5"	"E4"	"E2.5"
4	23	"NR"	"C10"	"C6"	"D5"	"E5"	"E4"	"E2.5"
5	23	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"
6	23	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"
7	23	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
8	23	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
9	23	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
10	23	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
11	23	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
12	23	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
14	23	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
16	23	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	23	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	23	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	23	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	23	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
3	24	"B10"	"C10"	"D6"	"D5"	"E5"	"F4"	"F2.5"
4	24	"NR"	"C10"	"C6"	"D5"	"E5"	"E4"	"E2.5"
5	24	"NR"	"C10"	"C6"	"D5"	"E5"	"E4"	"E2.5"
6	24	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"
7	24	"NR"	"B10"	"C8"	"D6"	"D5"	"E4"	"E3"
8	24	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
9	24	"NR"	"B10"	"C8"	"C6"	"C5"	"D4"	"E3"
10	24	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
11	24	"NR"	"NR"	"B8"	"C6"	"C5"	"C4"	"D3"
12	24	"NR"	"NR"	"NR"	"B8"	"C6"	"C5"	"D4"
14	24	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"	"C4"
16	24	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
18	24	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
20	24	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
22	24	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"
24	24	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"C5"

3	25	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
4	25	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
5	25	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
6	25	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
7	25	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
8	25	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
9	25	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
10	25	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
11	25	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
12	25	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
14	25	"NR"	"NR"	"NR"	"NR"	"B10"	"C5"	"D5"
16	25	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"
18	25	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
20	25	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
22	25	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
24	25	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
3	26	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
4	26	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
5	26	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
6	26	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
7	26	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
8	26	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
9	26	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
10	26	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
11	26	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
12	26	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
14	26	"NR"	"NR"	"NR"	"NR"	"B10"	"C6"	"D5"
16	26	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"
18	26	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
20	26	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
22	26	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
24	26	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
3	27	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
4	27	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
5	27	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
6	27	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
7	27	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
8	27	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
9	27	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
10	27	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
11	27	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
12	27	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
14	27	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
16	27	"NR"	"NR"	"NR"	"NR"	"B10"	"C6"	"D5"
18	27	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"NR"
20	27	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"

22	27	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
24	27	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
3	28	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
4	28	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
5	28	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
6	28	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
7	28	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
8	28	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
9	28	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
10	28	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
11	28	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
12	28	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
14	28	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
16	28	"NR"	"NR"	"NR"	"NR"	"B10"	"C6"	"D5"
18	28	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"
20	28	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
22	28	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
24	28	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
3	29	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
4	29	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
5	29	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
6	29	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
7	29	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
8	29	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
9	29	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
10	29	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
11	29	"NR"	"C10"	"C10"	"D8"	"D6"	"E5"	"E3"
12	29	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
14	29	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
16	29	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
18	29	"NR"	"NR"	"NR"	"NR"	"B10"	"C6"	"D5"
20	29	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"
22	29	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
24	29	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
3	30	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
4	30	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
5	30	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
6	30	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
7	30	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
8	30	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
9	30	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
10	30	"NR"	"C10"	"D8"	"D6"	"E6"	"E5"	"F3"
11	30	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
12	30	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
14	30	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
16	30	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"

18	30	"NR"	"NR"	"NR"	"NR"	"B10"	"D5"	"D5"
20	30	"NR"	"NR"	"NR"	"NR"	"B10"	"B6"	"C5"
22	30	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
24	30	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
3	31	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
4	31	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
5	31	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
6	31	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
7	31	"C10"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
8	31	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
9	31	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
10	31	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
11	31	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
12	31	"NR"	"NR"	"C8"	"D6"	"E6"	"E5"	"F3"
14	31	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
16	31	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
18	31	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
20	31	"NR"	"NR"	"NR"	"NR"	"B10"	"D5"	"D5"
22	31	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"
24	31	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
3	32	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
4	32	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
5	32	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
6	32	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
7	32	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
8	32	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
9	32	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
10	32	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
11	32	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
12	32	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
14	32	"NR"	"NR"	"C10"	"D6"	"D6"	"E5"	"E3"
16	32	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
18	32	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
20	32	"NR"	"NR"	"NR"	"NR"	"B10"	"D5"	"D5"
22	32	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"
24	32	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"	"NR"
3	33	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
4	33	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
5	33	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
6	33	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
7	33	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
8	33	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
9	33	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
10	33	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
11	33	"NR"	"C10"	"D8"	"E6"	"F5"	"F5"	"F3"
12	33	"NR"	"C10"	"D8"	"E6"	"F5"	"F5"	"F3"

14	33	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
16	33	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
18	33	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
20	33	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
22	33	"NR"	"NR"	"NR"	"NR"	"B10"	"D5"	"D5"
24	33	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"
3	34	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
4	34	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
5	34	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
6	34	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
7	34	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
8	34	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
9	34	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
10	34	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
11	34	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
12	34	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
14	34	"NR"	"C10"	"C8"	"D6"	"E6"	"F5"	"F3"
16	34	"NR"	"NR"	"C10"	"D6"	"D6"	"E5"	"E3"
18	34	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
20	34	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
22	34	"NR"	"NR"	"NR"	"NR"	"B10"	"C6"	"D5"
24	34	"NR"	"NR"	"NR"	"NR"	"NR"	"B6"	"C5"
3	35	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
4	35	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
5	35	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
6	35	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
7	35	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
8	35	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
9	35	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
10	35	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
11	35	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
12	35	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
14	35	"NR"	"C10"	"C8"	"E6"	"E5"	"F5"	"F3"
16	35	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
18	35	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
20	35	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
22	35	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
24	35	"NR"	"NR"	"NR"	"NR"	"B10"	"C6"	"D5"
3	36	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
4	36	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
5	36	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
6	36	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
7	36	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
8	36	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
9	36	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
10	36	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"

11	36	"C10"	"C10"	"E8"	"E6"	"F5"	"F4"	"G3"
12	36	"NR"	"C10"	"E8"	"E6"	"E5"	"F5"	"G3"
14	36	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
16	36	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
18	36	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
20	36	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
22	36	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
24	36	"NR"	"NR"	"NR"	"NR"	"B10"	"C6"	"D5"
3	37	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
4	37	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
5	37	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
6	37	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
7	37	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
8	37	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
9	37	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
10	37	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
11	37	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
12	37	"C10"	"C10"	"E8"	"E6"	"F5"	"F4"	"G3"
14	37	"NR"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"
16	37	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
18	37	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
20	37	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
22	37	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
24	37	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
3	38	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
4	38	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
5	38	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
6	38	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
7	38	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
8	38	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
9	38	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
10	38	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
11	38	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
12	38	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
14	38	"NR"	"D10"	"E8"	"E6"	"E5"	"F4"	"G3"
16	38	"NR"	"C10"	"D8"	"E6"	"E5"	"F4"	"F3"
18	38	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
20	38	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
22	38	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
24	38	"NR"	"NR"	"NR"	"C8"	"C8"	"D5"	"D4"
3	39	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
4	39	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
5	39	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
6	39	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
7	39	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
8	39	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"

9	39	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
10	39	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
11	39	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
12	39	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
14	39	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
16	39	"NR"	"D10"	"E8"	"E6"	"E5"	"F4"	"G3"
18	39	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
20	39	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
22	39	"NR"	"NR"	"C10"	"C8"	"D6"	"E5"	"E3"
24	39	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
3	40	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
4	40	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
5	40	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
6	40	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
7	40	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
8	40	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
9	40	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
10	40	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
11	40	"C10"	"E10"	"E5"	"F5"	"G5"	"G4"	"H3"
12	40	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
14	40	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
16	40	"NR"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"
18	40	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
20	40	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
22	40	"NR"	"NR"	"C10"	"E6"	"D6"	"E5"	"E3"
24	40	"NR"	"NR"	"C10"	"C8"	"D6"	"D5"	"E4"
3	41	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
4	41	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
5	41	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
6	41	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
7	41	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
8	41	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
9	41	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
10	41	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
11	41	"C10"	"E10"	"E5"	"F5"	"F5"	"G4"	"H3"
12	41	"C10"	"E10"	"E5"	"F5"	"F5"	"G4"	"H3"
14	41	"C10"	"D10"	"E5"	"F5"	"F5"	"G4"	"H3"
16	41	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
18	41	"NR"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"
20	41	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
22	41	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
24	41	"NR"	"NR"	"C10"	"D8"	"D6"	"E5"	"E3"
3	42	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
4	42	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
5	42	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
6	42	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"

7	42	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
8	42	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
9	42	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
10	42	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
11	42	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
12	42	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
14	42	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
16	42	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
18	42	"C10"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"
20	42	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
22	42	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
24	42	"NR"	"C10"	"C10"	"D8"	"D6"	"E5"	"E3"
3	43	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
4	43	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
5	43	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
6	43	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
7	43	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
8	43	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
9	43	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
10	43	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
11	43	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
12	43	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
14	43	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
16	43	"C10"	"D10"	"E6"	"E6"	"F5"	"G4"	"H3"
18	43	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
20	43	"NR"	"C10"	"E8"	"E6"	"F5"	"F4"	"G3"
22	43	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
24	43	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
3	44	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
4	44	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
5	44	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
6	44	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
7	44	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
8	44	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
9	44	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
10	44	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
11	44	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
12	44	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
14	44	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
16	44	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
18	44	"C10"	"C10"	"E8"	"E6"	"F5"	"F4"	"G3"
20	44	"NR"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"
22	44	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
24	44	"NR"	"C10"	"C8"	"D6"	"E6"	"E5"	"F3"
3	45	"E8"	"D6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
4	45	"E8"	"D6"	"F4"	"G4"	"G3"	"H2.5"	"I2"

5	45	"E8"	"D6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
6	45	"E8"	"D6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
7	45	"E8"	"D6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
8	45	"D10"	"D6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
9	45	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
10	45	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
11	45	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
12	45	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
14	45	"C10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
16	45	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
18	45	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
20	45	"NR"	"C10"	"E8"	"E6"	"F5"	"F4"	"G3"
22	45	"NR"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"
24	45	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
3	46	"E8"	"F5"	"G4"	"G3"	"H3"	"I2.5"	"I2"
4	46	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
5	46	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
6	46	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
7	46	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
8	46	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
9	46	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
10	46	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
11	46	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
12	46	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
14	46	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
16	46	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
18	46	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
20	46	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
22	46	"NR"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"
24	46	"NR"	"C10"	"D8"	"E6"	"E5"	"F5"	"F3"
3	47	"E8"	"F5"	"G4"	"G3"	"H3"	"I2.5"	"I2"
4	47	"E8"	"F5"	"G4"	"G3"	"H3"	"I2.5"	"I2"
5	47	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
6	47	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
7	47	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
8	47	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
9	47	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
10	47	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
11	47	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
12	47	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
14	47	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
16	47	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
18	47	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
20	47	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
22	47	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
24	47	"NR"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"

3	48	"E8"	"F5"	"G4"	"G3"	"H3"	"I2.5"	"I2"
4	48	"E8"	"F5"	"G4"	"G3"	"H3"	"I2.5"	"I2"
5	48	"E8"	"F5"	"G4"	"G3"	"H3"	"I2.5"	"I2"
6	48	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
7	48	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
8	48	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
9	48	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
10	48	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
11	48	"E8"	"E6"	"F4"	"G4"	"G3"	"H2.5"	"I2"
12	48	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
14	48	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
16	48	"D10"	"E8"	"F5"	"F4"	"G4"	"H3"	"H2.5"
18	48	"C10"	"E10"	"E6"	"F5"	"G5"	"G4"	"H3"
20	48	"C10"	"D10"	"E6"	"F5"	"F5"	"G4"	"H3"
22	48	"C10"	"D10"	"E8"	"E6"	"F5"	"F4"	"G3"
24	48	"NR"	"C10"	"E8"	"E6"	"E5"	"F4"	"G3"
3	49	"E10"	"F6"	"G4"	"H4"	"H3"	"I2.5"	"I2"
4	49	"E10"	"F6"	"G4"	"H4"	"H3"	"I2.5"	"I2"
5	49	"E10"	"F6"	"G4"	"H4"	"H3"	"I2.5"	"I2"
6	49	"E10"	"F6"	"G4"	"H4"	"H3"	"I2.5"	"I2"
7	49	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
8	49	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
9	49	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
10	49	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
11	49	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
12	49	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
14	49	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
16	49	"D10"	"E10	"F6"	"G5"	"H5"	"H4"	"I3"
18	49	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
20	49	"C10"	"D10"	"F8"	"F6"	"G5"	"G4"	"H3"
22	49	"C10"	"D10"	"E8"	"F6"	"G6"	"G4"	"H3"
24	49	"C10"	"C10"	"E10"	"E6"	"G6"	"G5"	"H4"
3	50	"E10"	"F5"	"H5"	"H4"	"H3"	"I2.5"	"I2"
4	50	"E10"	"F5"	"H5"	"H4"	"H3"	"I2.5"	"I2"
5	50	"E10"	"F5"	"H5"	"H4"	"H3"	"I2.5"	"I2"
6	50	"E10"	"F5"	"H5"	"H4"	"H3"	"I2.5"	"I2"
7	50	"E10"	"F5"	"H5"	"H4"	"H3"	"I2.5"	"I2"
8	50	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
9	50	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
10	50	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
11	50	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
12	50	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
14	50	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
16	50	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
18	50	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
20	50	"C10"	"E10"	"F8"	"F6"	"G5"	"G4"	"H3"

22	50	"C10"	"D10"	"E8"	"F6"	"G6"	"H5"	"H3"
24	50	"NR"	"D10"	"E10"	"E6"	"F6"	"G5"	"H4"
3	51	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
4	51	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
5	51	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
6	51	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
7	51	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
8	51	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
9	51	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
10	51	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
11	51	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
12	51	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
14	51	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
16	51	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
18	51	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
20	51	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
22	51	"C10"	"E10"	"F8"	"F6"	"G5"	"F3"	"H3"
24	51	"C10"	"D10"	"E8"	"F6"	"G6"	"H5"	"H3"
3	52	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
4	52	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
5	52	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
6	52	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
7	52	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
8	52	"E10"	"F5"	"H5"	"H4"	"H3"	"I2.5"	"I2"
9	52	"E10"	"F6"	"H5"	"H4"	"H3"	"H3"	"I2"
10	52	"E10"	"F6"	"G5"	"H5"	"H4"	"H3"	"I2.5"
11	52	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
12	52	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
14	52	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
16	52	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
18	52	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
20	52	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
22	52	"C10"	"E10"	"F8"	"F6"	"G5"	"G4"	"H3"
24	52	"C10"	"D10"	"E8"	"F6"	"G6"	"H5"	"H3"
3	53	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
4	53	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
5	53	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
6	53	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
7	53	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
8	53	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
9	53	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
10	53	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
11	53	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
12	53	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
14	53	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
16	53	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"

18	53	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
20	53	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
22	53	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
24	53	"C10"	"E10"	"F8"	"F6"	"G5"	"G4"	"H3"
3	54	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
4	54	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
5	54	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
6	54	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
7	54	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
8	54	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
9	54	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
10	54	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
11	54	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
12	54	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
14	54	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
16	54	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
18	54	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
20	54	"D10"	"E10"	"F5"	"G5"	"H5"	"H4"	"I3"
22	54	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
24	54	"C10"	"E10"	"F8"	"F6"	"G5"	"G4"	"H3"
3	55	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
4	55	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
5	55	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
6	55	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
7	55	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
8	55	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
9	55	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
10	55	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
11	55	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
12	55	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
14	55	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
16	55	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
18	55	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
20	55	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
22	55	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
24	55	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
3	56	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
4	56	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
5	56	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
6	56	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
7	56	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
8	56	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
9	56	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
10	56	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
11	56	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
12	56	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"

14	56	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
16	56	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
18	56	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
20	56	"D10"	"E10"	"F6"	"E6"	"H5"	"H4"	"I3"
22	56	"D10"	"E10"	"F6"	"E6"	"H5"	"H4"	"I3"
24	56	"D10"	"E10"	"F6"	"E6"	"H5"	"H4"	"I3"
3	57	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
4	57	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
5	57	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
6	57	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
7	57	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
8	57	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
9	57	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
10	57	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
11	57	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
12	57	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
14	57	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
16	57	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
18	57	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
20	57	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
22	57	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
24	57	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
3	58	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
4	58	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
5	58	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
6	58	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
7	58	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
8	58	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
9	58	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
10	58	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
11	58	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
12	58	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
14	58	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
16	58	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
18	58	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
20	58	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
22	58	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
24	58	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
3	59	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
4	59	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
5	59	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
6	59	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
7	59	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
8	59	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
9	59	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
10	59	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"

11	59	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
12	59	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
14	59	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
16	59	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
18	59	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
20	59	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
22	59	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
24	59	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
3	60	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
4	60	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
5	60	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
6	60	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
7	60	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
8	60	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
9	60	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
10	60	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
11	60	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
12	60	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
14	60	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
16	60	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
18	60	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
20	60	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
22	60	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
24	60	"D10"	"E10"	"F6"	"G5"	"H5"	"H4"	"I3"
3	61	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
4	61	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
5	61	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
6	61	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"I2"
7	61	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
8	61	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
9	61	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
10	61	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
11	61	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
12	61	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
14	61	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
16	61	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
18	61	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2.5"
20	61	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
22	61	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
24	61	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I3"
3	62	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
4	62	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
5	62	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
6	62	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
7	62	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"I2"
8	62	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"

9	62	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
10	62	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
11	62	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
12	62	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
14	62	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
16	62	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
18	62	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
20	62	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
22	62	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
24	62	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
3	63	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
4	63	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
5	63	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
6	63	"F8"	"G5"	"G3"	"H3"	"I3"	"I2.5"	"J2"
7	63	"F8"	"G5"	"G3"	"H3"	"I3"	"I2.5"	"J2"
8	63	"F8"	"G5"	"G3"	"H3"	"I3"	"I2.5"	"I2"
9	63	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
10	63	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
11	63	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
12	63	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
14	63	"F8"	"G6"	"H4"	"H4"	"I3"	"I2.5"	"I2"
16	63	"E10"	"F6"	"H5"	"H4"	"H4"	"I2.5"	"I2"
18	63	"E10"	"F6"	"H5"	"H4"	"H4"	"I2.5"	"I2"
20	63	"E10"	"F6"	"H5"	"H4"	"H4"	"I2.5"	"I2.5"
22	63	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
24	63	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
3	64	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"J2"
4	64	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
5	64	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
6	64	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
7	64	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
8	64	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
9	64	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"I2"
10	64	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
11	64	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
12	64	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
14	64	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
16	64	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
18	64	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
20	64	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
22	64	"E10"	"F8"	"G5"	"H5"	"H4"	"F3"	"I2.5"
24	64	"E10"	"F8"	"G5"	"H5"	"H4"	"F3"	"I2.5"
3	65	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
4	65	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"J2"
5	65	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
6	65	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"

7	65	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
8	65	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
9	65	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
10	65	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
11	65	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
12	65	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
14	65	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
16	65	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
18	65	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
20	65	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
22	65	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2.5"
24	65	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
3	66	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
4	66	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
5	66	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"J2"
6	66	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
7	66	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
8	66	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
9	66	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
10	66	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
11	66	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"I2"
12	66	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
14	66	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
16	66	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
18	66	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
20	66	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
22	66	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
24	66	"E10"	"F8"	"G5"	"H5"	"H4"	"H3"	"I2.5"
3	67	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
4	67	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
5	67	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
6	67	"F6"	"H4"	"H3"	"I3"	"I2.5"	"I2.5"	"NA"
7	67	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
8	67	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
9	67	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
10	67	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
11	67	"F8"	"G5"	"G3"	"H3"	"I3"	"I2.5"	"J2"
12	67	"F8"	"G5"	"G3"	"H3"	"I3"	"I2.5"	"J2"
14	67	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
16	67	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
18	67	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
20	67	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
22	67	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
24	67	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
3	68	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
4	68	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"

5	68	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
6	68	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
7	68	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
8	68	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
9	68	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
10	68	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
11	68	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
12	68	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
14	68	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
16	68	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
18	68	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
20	68	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
22	68	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
24	68	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
3	69	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
4	69	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
5	69	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
6	69	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
7	69	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
8	69	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
9	69	"F8"	"G6"	"G3"	"H3"	"I2.5"	"I2.5"	"J2"
10	69	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
11	69	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
12	69	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
14	69	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
16	69	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"I2"
18	69	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
20	69	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
22	69	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
24	69	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
3	70	"E10"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
4	70	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
5	70	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
6	70	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
7	70	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
8	70	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
9	70	"F6"	"H4"	"H3"	"I3"	"I2.5"	"J2"	"NA"
10	70	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
11	70	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
12	70	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
14	70	"F8"	"G6"	"G3"	"H3"	"I3"	"I2.5"	"J2"
16	70	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
18	70	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
20	70	"F8"	"G6"	"H4"	"H3"	"I3"	"I2.5"	"I2"
22	70	"E10"	"F6"	"H5"	"H4"	"H3"	"I2.5"	"I2"
24	70	"E10"	"E6"	"H5"	"H4"	"H3"	"I2.5"	"I2"

3	71	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
4	71	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
5	71	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
6	71	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
7	71	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	71	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	71	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	71	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	71	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
12	71	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
14	71	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
16	71	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
18	71	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
20	71	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
22	71	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
24	71	"F10"	"G8"	"H6"	"I5"	"I5"	"I4"	"J2.5"
3	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
4	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
5	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
6	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
7	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	72	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	72	"G10"	"H8"	"I5"	"I4"	"I3"	"I3"	"K2.5"
14	72	"G10"	"H8"	"I5"	"I4"	"I3"	"I3"	"K2.5"
16	72	"G10"	"H8"	"I5"	"I4"	"I3"	"I3"	"K2.5"
18	72	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
20	72	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
22	72	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
24	72	"F10"	"G8"	"H6"	"I5"	"I5"	"I4"	"J2.5"
3	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
4	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
5	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
6	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
7	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	73	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
14	73	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
16	73	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
18	73	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
20	73	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"

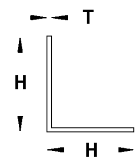
22	73	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
24	73	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
3	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
4	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
5	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
6	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
7	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	74	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	74	"H8"	"H6"	"I5"	"I3"	"J3"	"J2.5"	"L2"
14	74	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
16	74	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
18	74	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
20	74	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
22	74	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
24	74	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
3	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
4	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
5	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
6	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
7	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
14	75	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
16	75	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
18	75	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
20	75	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
22	75	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
24	75	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
3	76	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
4	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
5	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
6	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
7	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
14	76	"H8"	"I5"	"I4"	"I3"	"J3"	"J2.5"	"L2"
16	76	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"

18	76	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
20	76	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
22	76	"F10"	"H8"	"H5"	"I5"	"I3"	"I3"	"J2.5"
24	76	"F10"	"H8"	"H5"	"I5"	"I3"	"I3"	"J2.5"
3	77	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
4	77	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
5	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
6	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
7	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
14	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
16	77	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
18	77	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
20	77	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
22	77	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
24	77	"G10"	"H8"	"H5"	"I5"	"I4"	"I3"	"J2.5"
3	78	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
4	78	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
5	78	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
6	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
7	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
14	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
16	78	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
18	78	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"L2"
20	78	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
22	78	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
24	78	"F10"	"H8"	"H5"	"I5"	"I4"	"I3"	"K2.5"
3	79	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
4	79	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
5	79	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
6	79	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
7	79	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
8	79	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	79	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	79	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	79	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	79	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"

14	79	"H8"	"H6"	"I3"	"I3"	"J2.5"	"J2.5"	"L2"
16	79	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
18	79	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
20	79	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
22	79	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
24	79	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
3	80	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
4	80	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
5	80	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
6	80	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
7	80	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
8	80	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
9	80	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
10	80	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	80	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	80	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
14	80	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
16	80	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
18	80	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
20	80	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
22	80	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
24	80	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
3	81	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
4	81	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
5	81	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
6	81	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
7	81	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
8	81	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
9	81	"H8"	"H6"	"I4"	"I3"	"J2.5"	"J2.5"	"L2"
10	81	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
11	81	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	81	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
14	81	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
16	81	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
18	81	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
20	81	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
22	81	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
24	81	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
3	82	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
4	82	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
5	82	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
6	82	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
7	82	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
8	82	"H6"	"I5"	"I4"	"J3"	"J2.5"	"K2"	"NA"
9	82	"H6"	"H6"	"I4"	"J3"	"J2.5"	"K2"	"NA"
10	82	"H8"	"H6"	"I4"	"I3"	"J3"	"K2"	"L2"

11	82	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
12	82	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
14	82	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
16	82	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
18	82	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
20	82	"H8"	"H6"	"I4"	"I3"	"J3"	"J2.5"	"L2"
22	82	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
24	82	"G10"	"H8"	"I5"	"I4"	"I3"	"J3"	"K2.5"
99	99	"Not Listed"	"Not Listed"	"Not Listed"	"Not Listed"	"Not Listed"	"Not Listed"	"Not Listed"

Table A.14 Reinforcement Key
Spiral Flat Oval Duct Gauge/Reinforcement

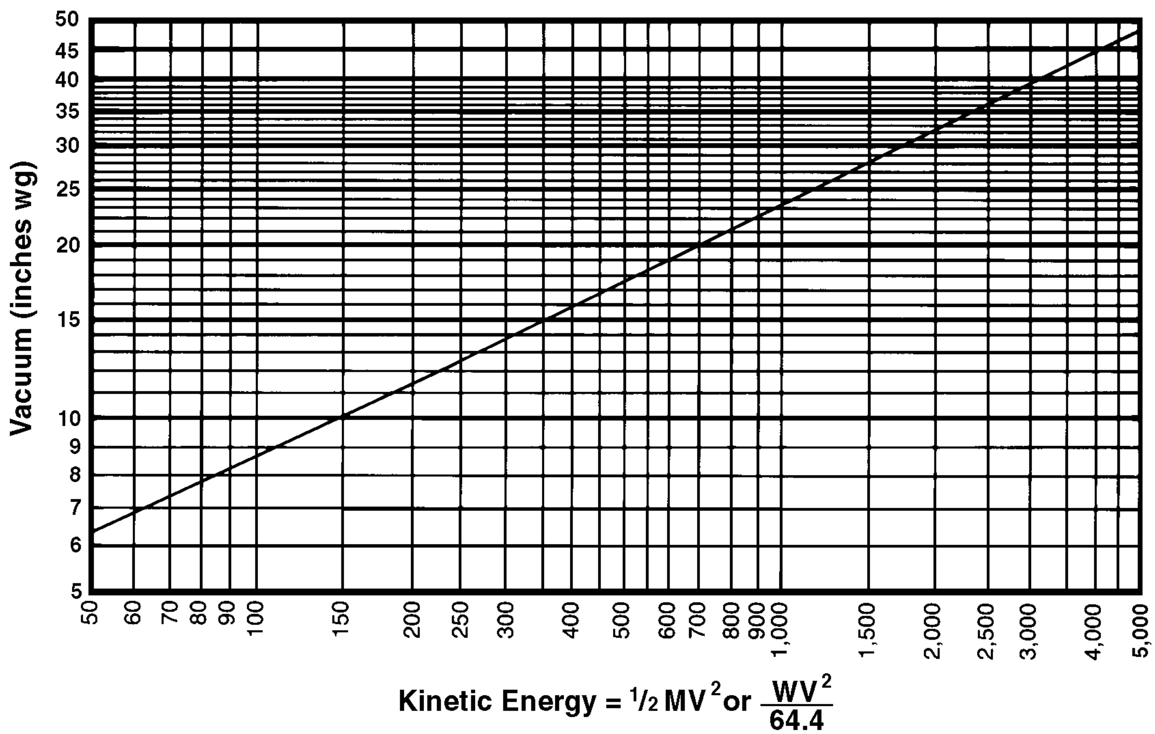


REINF. CLASS	EI*	ANGLE H x T (MIN)	WT LF
A	0.43	Use C	
B	1.0	Use C	
C	1.9	C1 x 16 ga. C3/4 x 1/8	0.40 0.57
D	2.7	H3/4 x 1/8 C1 x 1/8	0.57 0.80
E	6.5	C1 1/4 x 12 ga. H1 x 1/8	0.90
F	12.8	H1 1/4 x 1/8	1.02
G	15.8	1 1/2 x 1/8	1.23
H	22 (+) 26.4 (-)	1 1/2 x 3/16 2 x 1/8	1.78 1.65
I	69	C2 x 3/16 2 1/2 x 1/8	2.44 2.10
J	80	H2 x 3/16 C2 x 1/4 2 1/2 x 1/8 (+)	2.44 3.20 2.10
K	103	2 1/2 x 3/16	3.10
L	207	H2 1/2 x 1/4	4.10

* Effective El is number listed times 10^5 before adjustment for bending moment capacity. Plus (+) or minus (-) is a pressure mode restriction. Both modes are accepted when neither is given. C and H denote cold formed and hot rolled ratings; when neither is listed, either may be used.

Table A.15 Duct Collapse

Vacuum developed by sudden block in rapidly moving air streams



Under certain conditions, ductwork on the positive-pressure side of a fan can collapse. Duct collapse can be caused, for example, by a fire damper slamming shut. Due to inertia, the column of air downstream of the closed damper continues to move rapidly, creating a tremendous vacuum behind it. Use the chart below to estimate the potential negative pressure. Calculate the kinetic energy of the weight of air in the ductwork between the fire damper and the first divided-flow fitting or outlet. "W" is weight of air in pounds, and "V" is velocity in feet per second. Using a negative pressure relief damper just downstream of the fire damper can protect the duct system collapse caused by this type of stress. Refer to United McGill's *Engineering Bulletin Volume 2, Number 9* for a detailed discussion of duct collapse.

A.2.3 Industrial Manufacturing Standards

Table A.16 Class A Negative Pressure Material Handling Systems Round Spiral Duct Gauge/Reinforcement Recommendations

Duct Diameter (inches)	0 to -10 inches wg Gauge/Reinforcement	-10 to -20 inches wg Gauge/Reinforcement	-20 to -30 inches wg Gauge/Reinforcement	-30 to -40 inches wg Gauge/Reinforcement
3 – 7	26 ga.	26 ga.	26 ga.	26 ga.
7 ½- 8	26 ga.	26 ga.	26 ga.	24 ga.
8 ½- 12	24 ga.	24 ga.	22 ga.	22 ga.
12 ½- 15	24 ga.	22 ga.	22 ga.	20 ga.
16 – 18	22 ga.	20 ga.	20 ga.	18 ga.
19 – 22	22 ga.	18 ga.	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.
23 –26	20 ga.	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.	18 ga. W/AR 6 ft. c/c or 16 ga. W/AR 12 ft. c/c
27 – 34	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.	18 ga. W/AR 6 ft. c/c or 16 ga. W/AR 12 ft. c/c	18 ga. W/AR 4 ft. c/c or 16 ga. W/AR 12 ft. c/c
35 – 42	18 ga. W/FL or 16 ga.	18 ga. W/FL + AR 6 ft. c/c or 16 ga. W/FL	18 ga. W/FL + AR 4 ft. c/c or 16 ga. W/FL	16 ga. W/FL
44 – 50	18 ga. W/FL + AR 6 ft. c/c or 16 ga. W/FL	18 ga. W/FL + AR 4 ft. c/c or 16 ga. W/FL	16 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL
52 – 60	18 ga. W/FL + AR 6 ft. c/c or 16 ga. W/FL	18 ga. W/FL + AR 4 ft. c/c or 16 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL
62 – 72	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	14 ga. W/FL	14 ga. W/FL + AR 6 ft. c/c

Notes:

1. W/AR ## ft. c/c = single angle ring reinforcement at maximum ## feet on center spacing should be used.
2. W/FL = fully welded flange angle rings should be used as joint connectors at maximum 12-foot spacing.
3. Class A = return air, hot air and nonflammable gaseous emissions.

Table A.17 Class B Round Spiral Duct Gauge/Reinforcement Recommendations Negative Pressure Material Handling Systems

Duct Diameter (inches)	0 to -10 inches wg Gauge/Reinforcement	-10 to -20 inches wg Gauge/Reinforcement	-20 to -30 inches wg Gauge/Reinforcement	-30 to -40 inches wg Gauge/Reinforcement
3 – 8	24 ga.	24 ga.	24 ga.	24 ga.
8 ½- 12	22 ga.	22 ga.	22 ga.	22 ga.
12 ½- 15	22 ga.	22 ga.	22 ga.	20 ga.
16 – 18	22 ga.	20 ga.	20 ga.	18 ga.
19 – 22	20 ga.	18 ga.	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.
23 –26	20 ga.	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.	18 ga. W/AR 6 ft. c/c or 16 ga. W/AR 12 ft. c/c
27 – 34	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.	18 ga. W/AR 6 ft. c/c or 16 ga. W/AR 12 ft. c/c	18 ga. W/AR 4 ft. c/c or 16 ga. W/AR 12 ft. c/c
35 – 42	18 ga. W/FL or 16 ga.	18 ga. W/FL + AR 6 ft. c/c or 16 ga. W/FL	18 ga. W/FL + AR 4 ft. c/c or 16 ga. W/FL	16 ga. W/FL
44 – 50	18 ga. W/FL + AR 6 ft. c/c or 16 ga. W/FL	18 ga. W/FL + AR 4 ft. c/c or 16 ga. W/FL	16 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL
52 – 60	18 ga. W/FL + AR 6 ft. c/c or 16 ga. W/FL	18 ga. W/FL + AR 4 ft. c/c or 16 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL
62 – 72	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	14 ga. W/FL	14 ga. W/FL + AR 6 ft. c/c

Notes:

1. W/AR ## ft. c/c = single angle ring reinforcement at maximum ## feet on center spacing should be used.
2. W/FL = fully welded flange angle rings should be used as joint connectors at maximum 12-foot spacing.
3. Class B = flammable gaseous emissions, liquid sprays and mists, and nonabrasive particulate in light and moderate concentrations.

Table A.18 Class C Negative Pressure Material Handling Systems Round Spiral Duct Gauge/Reinforcement Recommendations

Duct Diameter (inches)	0 to -10 inches wg Gauge/Reinforcement	-10 to -20 inches wg Gauge/Reinforcement	-20 to -30 inches wg Gauge/Reinforcement	-30 to -40 inches wg Gauge/Reinforcement
3 – 8	22 ga.	22 ga.	22 ga.	22 ga.
8 ½- 15	20 ga.	20 ga.	20 ga.	20 ga.
16 – 18	20 ga.	20 ga.	20 ga.	18 ga.
19 – 22	18 ga.	18 ga.	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.
23 – 26	18 ga.	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.	18 ga. W/AR 6 ft. c/c or 16 ga. W/AR 12 ft. c/c
27 – 30	18 ga.	18 ga. W/AR 12 ft. c/c or 16 ga.	18 ga. W/AR 6 ft. c/c or 16 ga W/AR 12 ft. c/c	18 ga. W/AR 4 ft. c/c or 16 ga. W/AR 12 ft. c/c
31 – 34	16 ga.	16 ga.	16 ga. W/AR 12 ft. c/c	16 ga. W/AR 12 ft. c/c
35 – 42	16 ga.	16 ga. W/FL	16 ga. W/FL	16 ga. W/FL
44 – 50	16 ga. W/FL	16 ga. W/FL	16 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL
52 – 60	16 ga. W/FL	16 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL
62 – 72	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	16 ga. W/FL + AR 6 ft. c/c or 14 ga. W/FL	14 ga. W/FL	14 ga. W/FL + AR 6 ft. c/c

Notes:

1. W/AR ## ft. c/c = single angle ring reinforcement at maximum ## feet on center spacing should be used.
2. W/FL = fully welded flange angle rings should be used as joint connectors at maximum 12-foot spacing.
3. Class C = nonabrasive particulate in high concentrations, moderately abrasive particulate in low and moderate concentrations, and highly abrasive particulate in low concentrations.

Table A.19 Class D Negative Pressure Material Handling Systems Round Spiral Duct Gauge/Reinforcement Recommendations

Duct Diameter (inches)	0 to -10 inches wg Gauge/Reinforcement	-10 to -20 inches wg Gauge/Reinforcement	-20 to -30 inches wg Gauge/Reinforcement	-30 to -40 inches wg Gauge/Reinforcement
3 – 8	20 ga.	20 ga.	20 ga.	20 ga.
8 ½- 18	18 ga.	18 ga.	18 ga.	18 ga.
19 – 22	16 ga.	16 ga.	16 ga.	16 ga.
23 – 26	16 ga.	16 ga.	16 ga.	16 ga. W/AR 12 ft. c/c
27 – 30	16 ga.	16 ga.	16 ga. W/FL or 14 ga.	16 ga. W/FL or 14 ga.
31 – 46	14 ga. W/FL	14 ga. W/FL	14 ga. W/FL	14 ga. W/FL + AR 6 ft. c/c
48 – 72	14 ga. W/FL	14 ga. W/FL	14 ga. W/FL	14 ga. W/FL + AR 6 ft. c/c

Notes:

1. W/AR ## ft. c/c = single angle ring reinforcement at maximum ## feet on center spacing should be used.
2. W/FL = fully welded flange angle rings should be used as joint connectors at maximum 12-foot spacing.
3. Class D = moderately abrasive particulate in high concentrations and highly abrasive particulate in moderate and high concentrations.

A.3 Derivations

This section discusses several topics which are not necessary for designing or analyzing duct systems. They are presented as additional information to give more insight.

A.3.1 Derivation of the Continuity Equation

When applied to a control volume, the law of conservation of mass states that the time rate of accumulation of mass within the control volume (*cv*) is equal to the excess of the incoming rate of flow over the outgoing rate of flow (see **Figure A.3**).

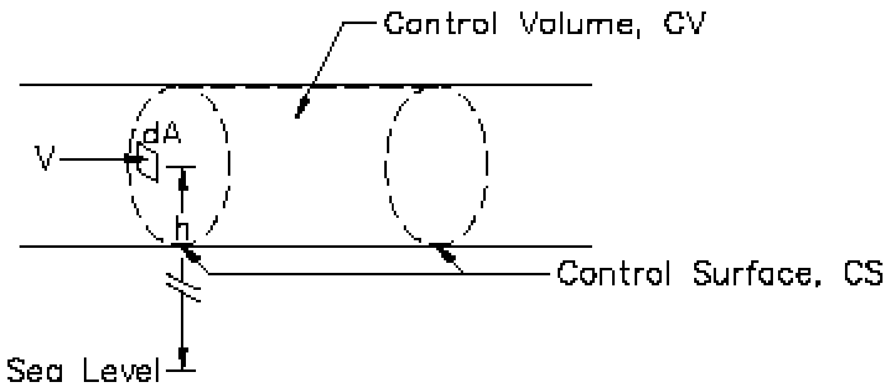


Figure A.3 Control Volume Analysis

Using control volume formulation, this is expressed as

$$\frac{dm_{cv}}{dt} = \frac{\partial}{dt}\int_{cv} \rho \, dv + \int_{cs} \rho \vec{V} \, dA \qquad \textbf{Equation A.2a}$$

where

$\frac{dm_{cv}}{dt}$	=	time rate of mass change within the control volume
$\frac{\text{¶}}{dt}$	=	partial derivative with respect to time
ò_{cv}	=	control volume integral
ò_{cs}	=	control surface integral
$\vec{V}$	=	fluid velocity vector
r	=	fluid density

For steady flow (i.e., fluid properties at every point are independent of time), **Equation A.2a** reduces to

$$0 = \int_{CS} \rho \vec{V} \, dA \qquad \textbf{Equation A.2b}$$

Since any fluid property remains constant for one-dimensional, steady-state flow, **r** and $\vec{V}$ can be brought out in front of the integral:

$$0 = \rho \vec{V} \int_{CS} dA = \rho \vec{V} \int_{in}^{out} dA \qquad \textbf{Equation A.2c}$$

Since there is only one section where fluid enters and one section where fluid leaves the control volume, solving the integral in **Equation A.2c** results in the continuity equation:

$$(\rho A V)_{out} - (\rho A V)_{in} = 0 \qquad \textbf{Equation A.2d}$$

or

$$(\rho A V)_{out} = (\rho A V)_{in} = \dot{m}$$

where

$\dot{m}$ = mass flow rate

and the velocity is perpendicular to the control surfaces.

Normally we are not concerned with mass flow rates when designing duct systems. When density remains relatively constant throughout a system, volume flow rates are generally the only information necessary to size ductwork. The volume rate of flow is calculated by dividing the mass flow rate by the density, giving the resulting simplified form of the continuity equation:

$$Q = \frac{\dot{m}}{\rho} = V \times A \quad \text{(usually written } Q = AV\text{)} \qquad \textbf{Equation A.3}$$

A.3.2 Derivation of DTP = DSP + DVP

We must begin with the first law of thermodynamics, the conservation of energy. A system of constant mass can be expressed as

$$\dot{Q} + \dot{W} = \left(\frac{dE}{dt} \right)_{system} \qquad \textbf{Equation A.4}$$

where

$\dot{Q}$ = rate of heat transfer into or out of the system

$\dot{W}$ = rate of work done by the system (shaft work)

$\left(\frac{dE}{dt} \right)_{system}$ = rate of change in total energy of the system where the total energy of the system is given by

$$E_{system} = \int_{mass(system)} e\,dm + \int_{volume(system)} e\rho\,dv \qquad \textbf{Equation A.5}$$

and

$$e = u + \frac{v^2}{2} + gz \qquad \text{or the energy per unit mass.}$$

Note that for e, u is the internal energy of the system, $\frac{v^2}{2}$ represents the kinetic energy, and gz the potential energy. Therefore, the control volume formulation for the first law of thermodynamics is

$$\left(\frac{dE}{dt}\right)_{system} = \frac{\partial}{\partial t}\int_{CV} e\rho\,dv + \int_{CS} e\rho\vec{V}\cdot dA = \dot{Q} + \dot{W} \qquad \textbf{Equation A.6a}$$

Again, for steady-flow where extensive properties are independent of time and for a system with no shaft work $(\dot{W})$ or no heat transfer $(\dot{Q})$, **Equation A.6a** reduces to

$$0 = \int_{CS_{1-2}} e\rho\vec{V}\cdot dA \qquad \textbf{Equation A.6b}$$

Combining the internal energy terms on the left-hand side and the other like terms on the right-hand side

$$\rho u_1 - \rho u_2 = (P_1 - P_2) + \left(\frac{\rho V_1^2}{2} - \frac{\rho V_2^2}{2}\right) + (\rho g Z_1 - \rho g Z_2) \qquad \textbf{Equation A.7a}$$

We will let the term ΔP_t denote the irreversible (unrecoverable) loss of internal energy due to real systems or the quantity (ρu_1 - ρu_2).

$$\Delta P_t = (P_1 - P_2) + \left(\frac{\rho V_1^2}{2} - \frac{\rho V_2^2}{2}\right) + (\rho g Z_2 - \rho g Z_2) \qquad \textbf{Equation A.7b}$$

For an ideal case no energy would be lost, only converted to different forms of static, kinetic, or potential energy. These terms can further be identified as

$(P_1 - P_2)$ is the component of loss due to static pressure changes

$\left(\frac{\rho V_1^2}{2} - \frac{\rho V_2^2}{2}\right)$ is the component of loss due to velocity pressure changes

$(\mathbf{r} z_1 - \mathbf{r} z_2)$ is the component of loss due to changes in elevation or potential energy

ΔP_t is the loss of energy (total pressure loss) which is irrecoverable as air flows from point 1 to point 2. Note that u_2 will always be less than u_1, and thus ΔP_t will always be a positive number (positive loss).

Elevation changes are generally negligible in HVAC design except for the case of high-rise buildings which are served by a single air handling unit. Therefore, **Equation A.7b** reduces to

$$\Delta P_t = \left(P_1 - P_2\right) + \left(\frac{\rho V_1^2}{2} - \frac{\rho V_2^2}{2}\right)$$ **Equation A.7c**

The commonly used English unit for measuring pressure and pressure loss in duct design is *inches water gauge (wg)*. Density is usually expressed in units of lbm per cubic feet and velocity in feet per minute. Conversions need to be applied such that when these units are used, pressure is expressed as inches wg. When **Equation A.7c** is expressed in units of inches wg, it is usually written as

$$\Delta TP_{1-2} = \Delta SP_{1-2} + \Delta VP_{1-2}$$ **Equation A.8a**

$$\Delta TP_{1-2} = TP_1 - TP_2$$ **Equation A.8b**

where

D*TP*$_{1\text{-}2}$	=	Total pressure loss (inches wg) (ΔP_t converted to inches wg)
	=	TP_1 - TP_2
D*SP*$_{1\text{-}2}$	=	Change in static pressure (inches wg) (P_1 - P_2 converted to inches wg)
D*VP*$_{1\text{-}2}$	=	Change in velocity pressure (inches wg) ($\rho V_1^2/2$ - $\rho V_2^2/2$ converted to inches wg)
***TP*$_1$**	=	SP_1 + VP_1 or total pressure at Point 1 (inches wg)
***TP*$_2$**	=	SP_2 + VP_2 or total pressure at Point 2 (inches wg)
***SP*$_1$**	=	Static pressure at Point 1 (inches wg)
***SP*$_2$**	=	Static pressure at Point 2 (inches wg)
***VP*$_1$**	=	Velocity pressure at Point 1 (inches wg)
***VP*$_2$**	=	Velocity pressure at Point 2 (inches wg)

A.3.3 Derivation of the Velocity Pressure Equation

When density is given in lbm/ft^3 and velocity in fpm, the velocity pressure (**Equation A.9a**) must be multiplied by several conversion factors to express it in inches wg:

$$VP = \rho \frac{V^2}{2}$$ **Equation A.9a**

$$VP = \frac{\left(\rho \frac{lbm}{ft^3}\right) \times \left(V \frac{ft}{min} \times \frac{1\ min}{60\ sec}\right)^2}{2 \times \frac{1}{32.174}\ lbm\ \frac{ft}{lb_f}\ sec^2} \times \frac{0.1922\ \ inches\ \ wg}{\left(\frac{lb_f}{ft^2}\right)}$$

or

$$VP = \frac{\rho V^2}{1{,}206{,}243.5}$$ inches wg (where ρ is in lbm/ft^3 and V is in fpm)

More common use is to express this equation as

$$VP = \rho\left(\frac{V}{1{,}097}\right)^2 \text{ inches wg} \qquad \textbf{Equation A.9b}$$

When standard density (0.075 lbm/ft^3) is used or assumed **Equation A.9b** can be written as

$$VP = 0.075\left(\frac{V}{1{,}097}\right)^2 = \left(\frac{V}{4{,}005}\right)^2 \text{ inches wg} \qquad \textbf{Equation A.9c}$$

A.3.4 Darcy-Weisbach Equation for Calculating Friction Loss

When friction loss data is not available, the Darcy-Weisbach equation can be used to calculate the pressure loss as shown by the following formula:

$$\Delta TP_{friction} = f\left(\frac{12L}{D_h}\right)VP \qquad \textbf{Equation A.10}$$

where

$\mathbf{D}\boldsymbol{TP}_{\mathbf{friction}}$ = Friction loss (inches wg)

$\boldsymbol{f}$ = Friction factor (dimensionless)

$\boldsymbol{L}$ = Duct length (ft)

$\boldsymbol{D}_{\mathbf{h}}$ = Hydraulic diameter (inches)

For most flows encountered in duct, the friction factor depends on the surface roughness and Reynolds Number. The Colebrook equation is often used to calculate the friction factor (for flow in the transitional roughness zone, normally found in HVAC duct systems):

$$\frac{1}{\sqrt{f}} = -2\,\log_{10}\left[\frac{12\,\in}{3.7\,D_h} + \frac{2.51}{Re\,\sqrt{f}}\right] \qquad \textbf{Equation A.11}$$

where

I = Material absolute roughness factor (ft)

Re = Reynolds Number (dimensionless)

The Reynolds Number is given by **Equation A.12a**:

$$Re = \frac{D_h V}{720\,v} \qquad \textbf{Equation A.12a}$$

and for standard air

$$Re = 8.56\,D_h V \qquad \textbf{Equation A.12b}$$

where

$\boldsymbol{D_h}$ = Hydraulic diameter (inches)

$\boldsymbol{V}$ = Velocity (fpm)

V = Kinematic viscosity (ft²/sec)

A simplified formula for calculating *f* is shown in **Equation A.13**:

$$f' = 0.11\left(\frac{12 \in}{D_h} + \frac{68}{Re}\right)^{0.25}$$ **Equation A.13**

$If \;\; f' \geq 0.018\,,\; f = f'$

$If \;\; f' < 0.018\,,\; f = 0.85\, f' + 0.0028$

The hydraulic diameter is given by **Equation A.14**:

$$D_h = \frac{4A}{P}$$ **Equation A.14**

where

A = Duct cross-sectional area (square inches)

P = Duct perimeter (inches)

For round duct the hydraulic diameter is equal to the duct diameter.

Note that if actual values of temperature and barometric pressure are used to calculate the parameters in the Darcy-Weisbach equation, no additional corrections need to be made. However, if standard conditions are assumed and nonstandard conditions occur outside of the range of 50°F to 90°F or above 5,000 feet elevation, factors developed in **Section 1.4.5** should be used to correct the data.

Sample Problem A-1

Recalculate the friction loss of ***Sample Problem 1-5****, using a surface roughness of* ***e*** *= 0.0003 feet.*

Answer:

The velocity in the duct was found to be 1,400 fpm from **Sample Problem 1-5**.

From Equation 1.5:

$$VP = \left(\frac{V}{4{,}005}\right)^2 = \left(\frac{1{,}400}{4{,}005}\right)^2 = 0.12 \;\; inch \;\; wg$$

From Equation A.14:

$$D_h = D = 18 \;\; inches$$

From Equation A.12b:

$$Re = 8.56\, D_h V = 8.56 \times 18 \times 1{,}400 = 215{,}712$$

From Equation A.13:

$$f' = 0.11\left(\frac{12 \in}{D_h} + \frac{68}{Re}\right)^{0.25} = 0.11\left(\frac{12 \times 0.0003}{18} + \frac{68}{215,712}\right)^{0.25} = 0.0166$$

$$f < 0.018 \;:\; f = 0.85 \times f' + 0.0028 = 0.85 \times 0.0166 + 0.0028 = 0.0169$$

From Equation A.10:

$$\Delta TP_{friction} = f\left(\frac{12\,L}{D_h}\right)VP = 0.0169\left(\frac{12 \times 150}{18}\right) \times 0.12 = 0.20 \;\; inch \;\; wg$$

Thus the pressure loss would have been slightly underestimated by using a surface roughness of ε = 0.0003 feet.

A.3.5 Surface Roughness

Surface roughness can have a substantial impact on the friction loss of duct. Some materials have very low values of ε, such as aluminum (ε = 0.0001 feet). Other materials, such as fiberglass duct liner, may have surface roughness values as high as 0.01 feet (100 times as rough). To show the effect of surface roughness, **Sample Problem 1-5** is recalculated for ε = 0.01 feet.

Sample Problem A-2

Recalculate the friction loss of ***Sample Problem A-1****, using a value of e = 0.01 feet.*

Answer:

VP, D_h and *Re* are the same as in **Sample Problem A-1**.

From Equation A.13:

$$f' = 0.11\left(\frac{12 \times 0.01}{18} + \frac{68}{215,712}\right)^{0.25} = 0.0318$$

$$f' > 0.018 \;:\; f = f' = 0.0318$$

From Equation A.10:

$$\Delta TP_{friction} = f\left(\frac{12\,L}{D_h}\right)VP = 0.0318\left(\frac{12 \times 150}{18}\right) \times 0.12 = 0.38 \;\; inch \;\; wg$$

The friction loss of fiberglass-lined duct almost doubles that of unlined galvanized steel surface. **Table A.21** lists surface roughness ranges for various duct materials.

Table A.21 Duct Surface Roughness

Duct Material	e (ft)
Uncoated Carbon Steel	0.00015
PVC Plastic Pipe	0.00015-0.0003
Aluminum	0.00015-0.0002
Galvanized Steel, Longitudinal Seams, 4-foot Joints	0.00016-0.00032
Galvanized Steel, Spiral Seam, 12-foot Joints	0.00018-0.00038
Galvanized Steel, Longitudinal Seams, 2.5-foot Joints	0.0005
Fibrous Glass Duct, Rigid	0.003
Fibrous Glass Duct Liner, Air Side With Facing Material	0.005
Flexible Duct, Metallic	0.004-0.007
Flexible Duct, All Types of Fabric and Wire	0.0035-0.015
Concrete	0.001-0.01
Fibrous Glass Duct Liner Air Side Spray Coated	0.015

A.3.6 Derivation of Pressure Loss in Supply Fittings

$$DSP_{c\text{-}b} = VP_b(C_b + 1) - VP_c \text{ and } DSP_{c\text{-}s} = VP_s(C_s + 1) - VP_c$$

From Equation 1.4:

$$\Delta TP_{c-b} = \Delta SP_{c-b} + \Delta VP_{c-b}$$

$$\textit{Solving for } \Delta SP_{c-b}$$

$$\Delta SP_{c-b} = \Delta TP_{c-b} - \Delta VP_{c-b} \qquad \textbf{Equation A.15a}$$

From Equation 1.17a:

$$\Delta TP_{c-b} = C_b VP_b$$

and since $\Delta VP_{c\text{-}b} = VP_c - VP_b$, these values of $\Delta TP_{c\text{-}b}$ and $\Delta VP_{c\text{-}b}$ can be substituted into **Equation A.15a** to obtain

$$\Delta SP_{c-b} = C_b VP_b - (VP_c - VP_b) \qquad \textbf{Equation A.16a}$$

Gathering the VP_b terms on the right-hand side, we get

$$\Delta SP_{c-b} = VP_b(C_b + 1) - VP_c \qquad \textbf{Equation A.17a}$$

The same concepts can be used to show

$$\Delta SP_{c-s} = VP_s(C_s - 1) + VP_c$$

by substituting d or b in **Equation A.15a** and **Equation 1.17b**.

A.3.7 Derivation of Pressure Loss in Exhaust Fittings

$$\boldsymbol{DSP_{b-c} = VP_b(C_b - 1) + VP_c \;\; and \;\; DSP_{s-c} = VP_s(C_s - 1) + VP_c}$$

From Equation 4.4a:

$$\Delta TP_{b-c} = \Delta SP_{b-c} + \Delta VP_{b-c}$$

$$Solving \;\; for \; \Delta SP_{b-c}$$

$$DSP_{b-c} = DTP_{b-c} - DVP_{b-c}$$ **Equation A.15b**

From Equation 4.17b:

$$\Delta TP_{b-c} = C_b VP_b$$

and since $\Delta VP_{b-c} = VP_b - VP_c$, these values of ΔTP_{b-c} and ΔVP_{b-c} can be substituted into **Equation A.15b** to obtain

$$\Delta SP_{b-c} = C_b VP_b - \left(VP_b - VP_c\right)$$ **Equation A.16b**

Gathering the VP_b terms on the right-hand side, we get

$$\Delta SP_{b-c} = VP_b\left(C_b - 1\right) + VP_c$$ **Equation A.17b**

The same concepts can be used to show

$$\Delta SP_{s-c} = VP_s\left(C_s - 1\right) + VP_c$$

by substituting *s* or *b* in **Equation A.15b** and **Equation 4.17b**.

A.3.8 Negative Loss Coefficients

The negative loss coefficients of certain fittings are derived from experimental data. Note that two separate test programs produced consistent data: "An Experimental Study of the Pressure Losses in Converging Flow Fittings Used in Exhaust Systems," C.F. Sepsey and D.B. Pies, The Ohio State Research Foundation, 1972; and ASHRAE Research Project 551-TRP, "Laboratory Study to Determine Flow Resistance of HVAC Duct Fittings." Recall **Equation 4.16a**, which shows the relationship between measured total pressures and calculated loss coefficients. See **Figure A.4** for a simplified diagram of the lab test setup.

$$C_s = \frac{\Delta T_{s-c}}{VP_s}$$ **Equation 4.16a**

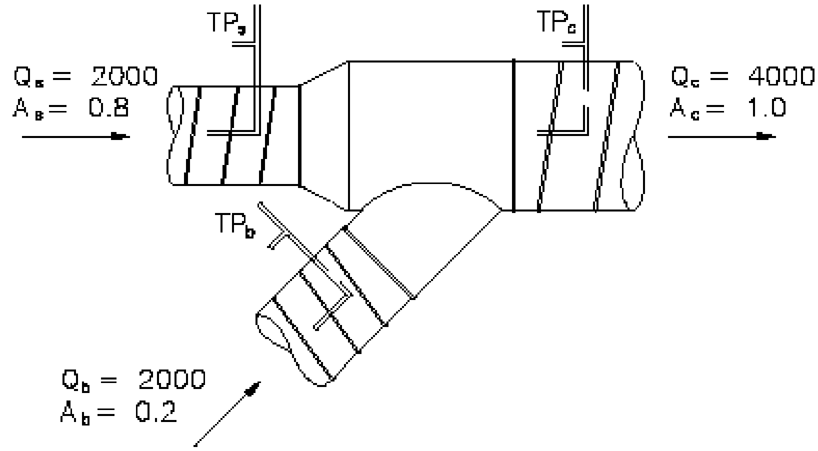


Figure A.4
Lab Test Setup to Measure Converging-Flow Parameters

Note that if the total pressure measured in the straight-through (upstream) (TP_s) is greater than the total pressure measured common (downstream) (TP_c), then the change in total pressure ($\Delta TP_{s\text{-}c} = TP_s - TP_c$) will be negative. Therefore, the upstream loss coefficient (C_s) will be negative. Does this make sense? Energy conservation still exists if we look at the proper control volume, i.e., one that includes all paths entering and exiting the control volume. **Sample Problem A-3** illustrates the correct control volume analysis.

Sample Problem A-3

The downstream area of a 30° lateral fitting is 1 square foot. The upstream area is 0.8 square foot and the branch area is 0.2 square foot. The upstream and branch volume flow rates are 2,000 cfm each. Show that energy is being lost in the control volume.

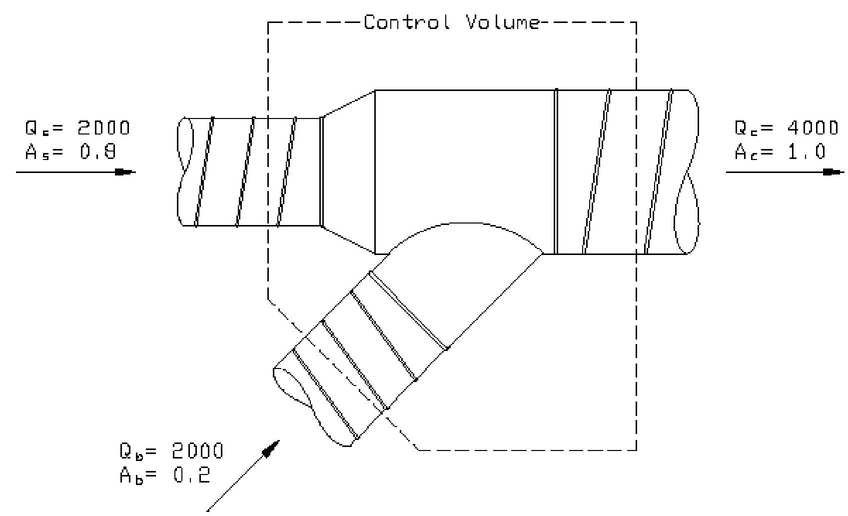


Figure A.5

Answer:

Calculate the relevant parameters.

The upstream to downstream area ratio is

$$\frac{A_s}{A_c} = \frac{0.8}{1.0} = 0.8$$

The branch to downstream area ratio is

$$\frac{A_b}{A_c} = \frac{\mathbf{0.2}}{\mathbf{1.0}} = \mathbf{0.2}$$

Also

$$\boldsymbol{Q_c = Q_s + Q_b = 2{,}000 + 2{,}000 = 4{,}000\ cfm}$$

From Equation 4.1

$$V = \frac{Q}{A} = V_s = \frac{2{,}000}{0.8} = 2{,}500\ \ fpm$$

$$V_b = \frac{2{,}000}{0.2} = 10{,}000\ \ fpm$$

The branch-to-downstream volume flow rate is

$$\frac{Q_b}{Q_c} = \frac{2{,}000}{4{,}000} = 0.5$$

From **ED5-2**

C_s = -3.56 C_b = 0.84

From Equation 4.5

$$VP_s = \left(\frac{V_s}{4{,}005}\right)^2 = \left(\frac{2{,}500}{4{,}005}\right)^2 = 0.39\ \ inch\ \ wg$$

$$VP_b = \left(\frac{10{,}000}{4{,}005}\right)^2 = 6.23\ \ inch\ \ wg$$

From Equation 4.15b

$TP_s - TP_c = \Delta TP_{s\text{-}c} = C_s VP_s = -3.56 \times 0.39 = -1.39$ inches wg

$TP_b - TP_c = \Delta TP_{b\text{-}c} = C_b VP_b = 0.84 \times 6.23 = 5.23$ inches wg

Thus the pressure increase from the straight-through (upstream) to the common (downstream) seemingly violates the conservation of energy. To realize what is going on, look at the energy states at the only three points of interest of our control volume. First, put a pseudo-fan at each point straight-through (upstream), branch, common (downstream) and calculate the necessary fan energy (hp) to pull the air. To calculate the required fan energy, use **Equation A.18**.

$$Fan\ \ hp = \frac{\left|TP\right| \times Q}{6{,}356}$$ **Equation A.18**

where 6,356 is a conversion constant to convert (in wg × cfm) to horsepower.

Now assume that the common (downstream) pressure (TP_c) is -10 inches wg.

$\Delta TP_{s\text{-}c} = TP_s - TP_c = -1.39$ inches wg and

$TP_c = -10$ inches wg, therefore, we can solve for TP_s:

$TP_s = \Delta TP_{s\text{-}c} + TP_c = -1.39 + (-10) = -11.39$ inches wg

Likewise

$TP_b = \Delta TP_{b\text{-}c} + TP_c = 5.23 + (-10) = -4.77$ inches wg

Fan energy required in the straight-through (upstream) section:

$$Fan\ \ hp_s = \frac{11.39 \times 2{,}000}{6{,}356} = 3.58\ hp$$

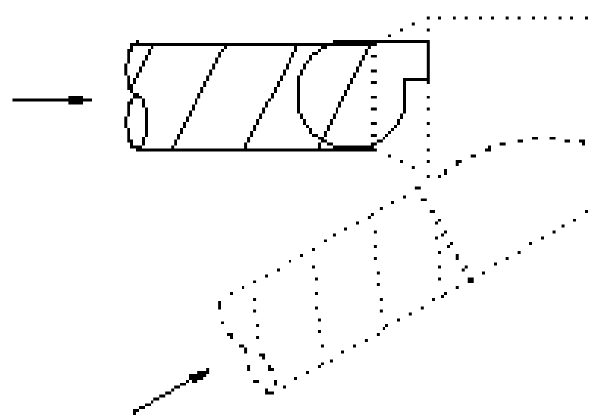


Figure A.6

Fan energy required in the branch section:

$$Fan\ \ hp = \frac{4.77 \times 2{,}000}{6{,}356} = 1.50\ hp$$

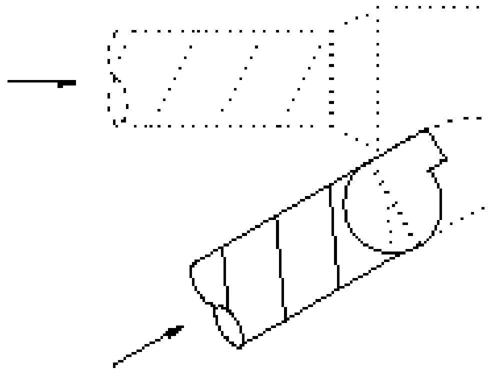


Figure A.7

The fan energy required in the common (downstream) section:

$$Fan\ \ hp_c = \frac{10 \times 4,000}{6,356} = 6.29\ hp$$

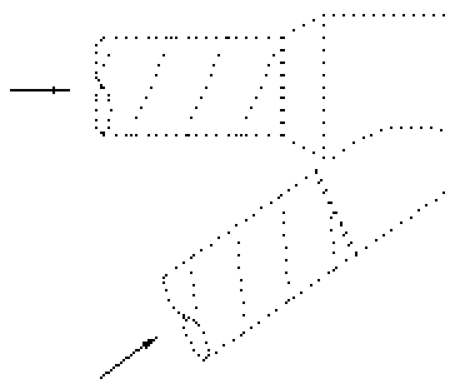


Figure A.8

For an ideal control volume, i.e., one with no losses, the downstream fan energy required would equal the energy (fan hp) required in the branch plus the energy required in the upstream. For example

Fan hp_{ideal} = Fan hp_b + Fan hp_s

= 1.50 hp + 3.58 hp

= 5.08 hp

But the fan energy required is

Fan hp_c = 6.29 hp

Hence, our process is not ideal; there are losses in the control volume that consume energy. Therefore, negative loss coefficients do not violate the law of conservation of energy.

A.3.9 Derivation of the Inlet Static Pressure Loss Equation

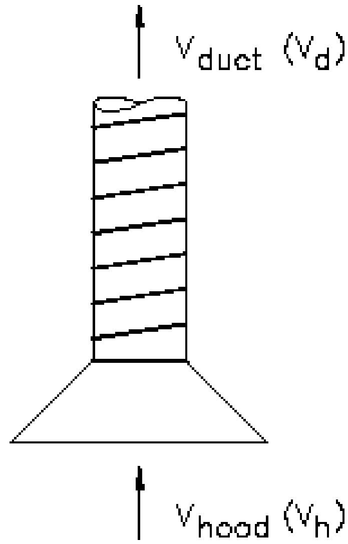


Figure A.9 Simple Hood

Consider the upstream of a simple hood shown in **Figure A.9** to be just outside the inlet of the hood at the same distance such that $V_h = 0$. Applying **Equation 4.3** yields

$\Delta TP_{h\text{-}d} = \Delta SP_{h\text{-}d} + \Delta VP_{h\text{-}d}$ or solving for $\Delta SP_{h\text{-}d}$

$\Delta SP_{h\text{-}d} = \Delta TP_{h\text{-}d} - \Delta VP_{h\text{-}d}$

From Equation 4.15b

$\Delta TP_h = C_h \, VP_d$

These values of $\Delta TP_{h\text{-}d}$ and $\Delta VP_{h\text{-}d}$ can be substituted into the equation above to obtain

$\Delta SP_{h\text{-}d} = C_h \, VP_d - (VP_h - VP_d)$

Gathering VP_d terms on the right-hand side, and since $VP_h = 0$, we get

$\Delta SP_{h\text{-}d} = VP_d \, (C_h + 1)$

where

C_h = Hood loss coefficient

Compound Slotted Hood

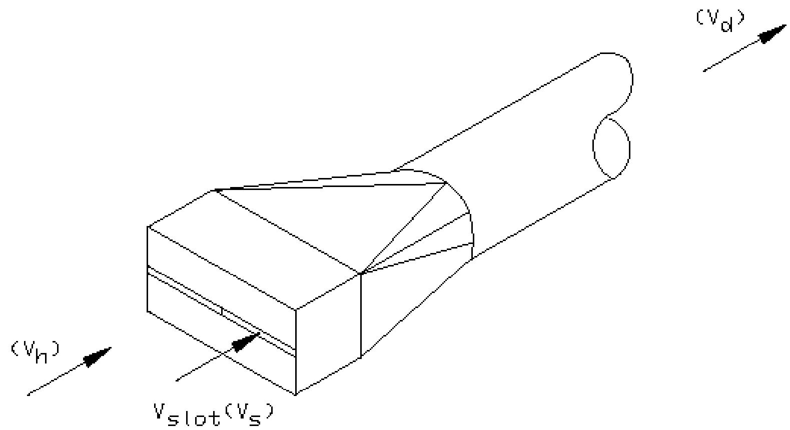


Figure A.10 Compound Slotted Hood

Once again, consider the upstream of the compound slotted hood shown in **Figure A.10** to be just outside the inlet at the same distance such that $V_h = 0$. Also, consider a cross-section S just inside the hood past the slot. At this point the velocity downstream of the slot is just V_s. The total pressure drop of the slot is given by an application of **Equation 4.3**:

$\Delta TP_{h\text{-}s} = \Delta SP_{h\text{-}s} + \Delta VP_{h\text{-}s}$

The total pressure drop across the hood section is now given by

$\Delta TP_{s\text{-}d} = \Delta SP_{s\text{-}d} + \Delta VP_{s\text{-}d}$

Combining the two equations results in the following:

$\Delta TP_{h\text{-}d} = \Delta TP_{h\text{-}s} + \Delta TP_{s\text{-}d} = \Delta SP_{h\text{-}s} + \Delta SP_{s\text{-}d} + \Delta VP_{h\text{-}s} + \Delta VP_{s\text{-}d}$

Also

$\Delta SP_{h\text{-}s} = SP_h - SP_s$; $\Delta VP_{h\text{-}s} = VP_h - VP_s$

$\Delta SP_{s\text{-}d} = SP_s - SP_d$; $\Delta VP_{s\text{-}d} = VP_s - VP_d$

From

$\Delta TP_{s\text{-}d} = C_h\, VP_d$ and $\Delta TP_{h\text{-}s} = C_s\, VP_s$

Substituting these values into the following

$\Delta TP_{h\text{-}d} = \Delta TP_{h\text{-}s} + \Delta TP_{s\text{-}d} = \Delta SP_{h\text{-}s} + \Delta SP_{s\text{-}d} + \Delta VP_{h\text{-}s} + \Delta VP_{s\text{-}d}$

$C_h\, VP_d + C_s\, VP_s = (SP_h - SP_s) + (SP_s - SP_d) + (VP_h - VP_s) + (VP_s - VP_d)$

or simplifying

$C_h\, VP_d + C_s\, VP_s = SP_h - SP_d + VP_h - VP_d$

Since

$SP_h - SP_d = \Delta SP_{h\text{-}d}$

This can be substituted into the above equation. Then solving for $\Delta SP_{h\text{-}d}$, this substitution gives

$\Delta SP_{h\text{-}d} = C_h \; VP_d + C_s \; VP_s - VP_h + VP_d$

Gathering VP_d terms on the right-hand side, and since $VP_h = 0$, then

$\Delta SP_{h\text{-}d} = VP_d(C_h + 1) + C_s \; VP_s$

A.4 Pressure Loss Data

A.4.1 Duct

A.4.1.1 Single-Wall Duct Friction Loss Chart

[See next page]

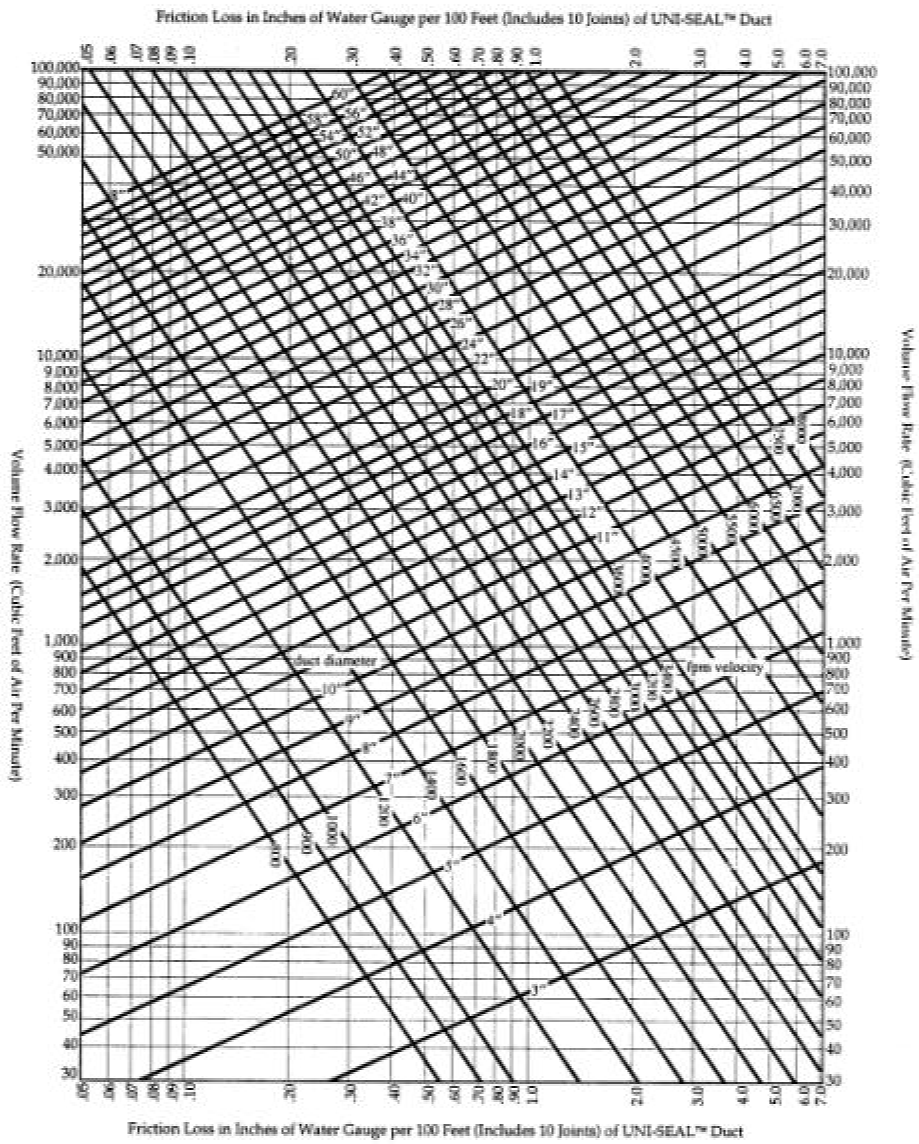


Figure A.11 Friction Loss of UNI-SEAL™ Duct

A.4.1.2 k-27® Correction Factor

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

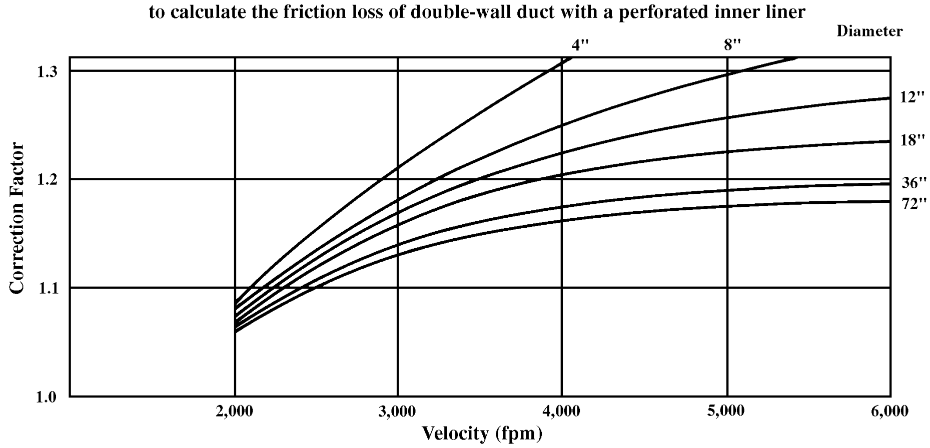


Figure A.12 k-27® Correction Factor

4.2 Fittings

A.4.2.1 Round-to-Flat Oval and Flat Oval-to-Round Transitions, C vs. Velocity

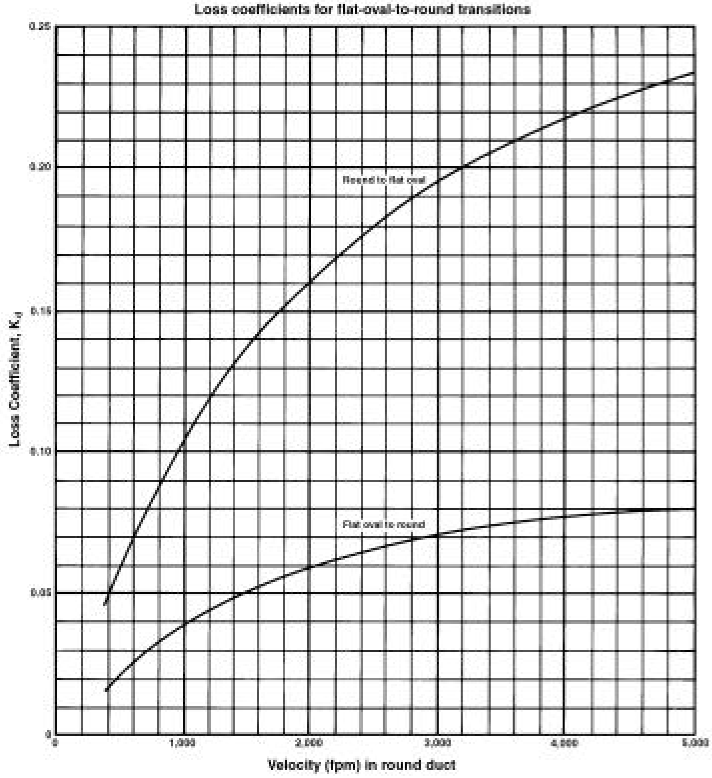


Figure A.24 Loss Coefficients for Round-to-Flat Oval and Flat Oval-to-Round Transitions A.4.2.2 Coupling, C vs. Diameter

Table A.22 Loss Coefficients for Round Duct Coupling

Nominal Diameter (inches)	Loss Coefficient
3	0.030
4	0.019
5	0.014
6	0.010
7	0.0081
8	0.0067
9	0.0055
10	0.0047
12	0.0036
14	0.0028
16	0.0023
18	0.0019
20	0.0016
22	0.0014
24	0.0012
26	0.0011
28	0.00097
30	0.00087
32	0.00079
34	0.00072
36	0.00066

A.5 Acoustical Data

A.5.1 Double-Wall Duct

A.5.1.1 Round Duct

Table A.24 Insertion Loss of 1-inch-thick Liner

Duct Diameter (inches)	Insulation Thickness (inches)	Velocity (fpm)	Octave Band							
			1 63 Hz	2 125 Hz	3 250 Hz	4 500 Hz	5 1,000 Hz	6 2,000 Hz	7 4,000 Hz	8 8,000 Hz
6	1	-4,000	0.25	0.58	1.29	2.60	2.23	1.56	1.30	0.62
		-2,000	0.25	0.54	1.11	2.21	2.57	1.85	1.46	0.95
		0	0.25	0.50	0.93	1.82	2.51	2.14	1.62	1.28
		2,000	0.23	0.48	0.87	1.69	2.31	1.99	1.56	1.27
		4,000	0.20	0.45	0.81	1.56	2.11	1.84	1.50	1.26
12	1	-4,000	0.19	0.47	1.19	2.44	1.91	1.29	1.10	0.47
		-2,000	0.19	0.43	1.01	2.05	2.25	1.58	1.26	0.80
		0	0.19	0.39	0.83	1.66	2.19	1.87	1.42	1.13
		2,000	0.17	0.37	0.77	1.53	1.99	1.72	1.36	1.12
		4,000	0.14	0.34	0.71	1.40	1.79	1.57	1.30	1.11
18	1	-4,000	0.16	0.42	1.09	2.29	1.64	1.06	0.92	0.33
		-2,000	0.16	0.38	0.91	1.90	1.98	1.35	1.08	0.66
		0	0.16	0.34	0.73	1.51	1.92	1.64	1.24	0.99
		2,000	0.14	0.32	0.67	1.38	1.72	1.49	1.18	0.98
		4,000	0.11	0.29	0.61	1.25	1.52	1.34	1.12	0.97
24	1	-4,000	0.13	0.34	0.99	2.14	1.36	0.83	0.74	0.20
		-2,000	0.13	0.30	0.81	1.75	1.70	1.12	0.90	0.53
		0	0.13	0.26	0.63	1.36	1.64	1.41	1.06	0.86
		2,000	0.11	0.24	0.57	1.23	1.44	1.26	1.00	0.85
		4,000	0.08	0.21	0.51	1.10	1.24	1.11	0.94	0.84
36	1	-4,000	0.03	0.16	0.78	1.82	0.75	0.31	0.35	0.00
		-2,000	0.03	0.12	0.60	1.43	1.09	0.60	0.51	0.24
		0	0.03	0.08	0.42	1.04	1.03	0.89	0.67	0.57
		2,000	0.00	0.06	0.36	0.91	0.83	0.74	0.61	0.56
		4,000	0.00	0.03	0.30	0.78	0.63	0.59	0.55	0.55
48	1	-4,000	0.00	0.00	0.57	1.51	0.17	.00	0.02	0.00
		-2,000	0.00	0.00	0.39	1.12	0.51	0.12	0.14	0.04
		0	0.00	0.00	0.21	0.73	0.45	0.41	0.30	0.29
		2,000	0.00	0.00	0.15	0.60	0.25	0.26	0.24	0.28
		4,000	0.00	0.00	0.09	0.47	0.00	0.11	0.18	0.27

Table A.25 Insertion Loss of 2-inch-thick Liner

Duct Diameter (inches)	Insulation Thickness (inches)	Velocity (fpm)	Octave Band							
			1 63 Hz	2 125 Hz	3 250 Hz	4 500 Hz	5 1,000 Hz	6 2,000 Hz	7 4,000 Hz	8 8,000 Hz
6	2	-4,000	0.36	0.81	1.62	3.00	2.19	1.45	1.28	0.67
		-2,000	0.36	0.77	1.44	2.61	2.53	1.74	1.44	1.00
		0	0.36	0.73	1.26	2.22	2.47	2.03	1.60	1.33
		2,000	0.34	0.71	1.20	2.09	2.27	1.88	1.54	1.32
		4,000	0.31	0.68	1.14	1.96	2.07	1.73	1.48	1.31
12	2	-4,000	0.31	0.71	1.52	2.85	1.87	1.18	1.08	0.52
		-2,000	0.31	0.67	1.34	2.46	2.21	1.47	1.24	0.85
		0	0.31	0.63	1.16	2.07	2.15	1.76	1.40	1.18
		2,000	0.29	0.61	1.10	1.94	1.95	1.61	1.34	1.17
		4,000	0.26	0.58	1.04	1.81	1.75	1.46	1.28	1.16
18	2	-4,000	0.27	0.63	1.41	2.69	1.60	0.95	0.90	0.38
		-2,000	0.27	0.59	1.23	2.30	1.94	1.24	1.06	0.71
		0	0.27	0.55	1.05	1.91	1.88	1.53	1.22	1.04
		2,000	0.25	0.53	0.99	1.78	1.68	1.38	1.16	1.03
		4,000	0.22	0.50	0.93	1.65	1.48	1.23	1.10	1.02
24	2	-4,000	0.23	0.56	1.31	2.53	1.33	0.73	0.73	0.25
		-2,000	0.23	0.52	1.13	2.14	1.67	1.02	0.89	0.58
		0	0.23	0.48	0.95	1.75	1.61	1.31	1.05	0.91
		2,000	0.21	0.46	0.89	1.62	1.41	1.16	0.99	0.90
		4,000	0.18	0.43	0.83	1.49	1.21	1.01	0.93	0.89
36	2	-4,000	0.14	0.38	1.10	2.22	0.71	0.21	0.33	0.00
		-2,000	0.14	0.34	0.92	1.83	1.05	0.50	0.49	0.29
		0	0.14	0.30	0.74	1.44	0.99	0.79	0.65	0.62
		2,000	0.12	0.28	0.68	1.31	0.79	0.64	0.59	0.61
		4,000	0.09	0.25	0.62	1.18	0.59	0.49	0.53	0.60
48	2	-4,000	0.06	0.21	0.90	1.91	0.13	0.00	0.04	0.00
		-2,000	0.06	0.17	0.72	1.52	0.47	0.01	0.12	0.00
		0	0.06	0.13	0.54	1.13	0.41	0.30	0.28	0.33
		2,000	0.04	0.11	0.48	1.00	0.21	0.15	0.22	0.33
		4,000	0.00	0.08	0.42	0.87	0.00	0.00	0.16	0.31

Table A.26 Insertion Loss of 3-inch-thick Liner

Duct Diameter (inches)	Insulation Thickness (inches)	Velocity (fpm)	Octave Band							
			1 63 Hz	2 125 Hz	3 250 Hz	4 500 Hz	5 1,000 Hz	6 2,000 Hz	7 4,000 Hz	8 8,000 Hz
6	3	-4,000	0.46	1.01	1.95	3.40	2.15	1.34	1.26	0.71
		-2,000	0.46	0.97	1.77	3.01	2.49	1.63	1.42	1.04
		0	0.46	0.93	1.59	2.62	2.43	1.92	1.58	1.37
		2,000	0.44	0.91	1.53	2.49	2.23	1.77	1.52	1.36
		4,000	0.41	0.88	1.47	2.36	2.03	1.62	1.46	1.35
12	3	-4,000	0.41	0.92	1.84	3.25	1.83	1.05	1.06	0.57
		-2,000	0.41	0.88	1.66	2.86	2.17	1.34	1.22	0.90
		0	0.41	0.84	1.48	2.47	2.11	1.63	1.38	1.23
		2,000	0.39	0.82	1.42	2.34	1.91	1.48	1.32	1.22
		4,000	0.36	0.79	1.36	2.21	1.71	1.33	1.26	1.21
18	3	-4,000	0.37	0.84	1.75	3.09	1.56	0.84	0.88	0.43
		-2,000	0.37	0.80	1.56	2.70	1.90	1.13	1.04	0.76
		0	0.37	0.76	1.38	2.31	1.84	1.42	1.20	1.09
		2,000	0.35	0.74	1.32	2.18	1.64	1.27	1.14	1.08
		4,000	0.32	0.71	1.26	2.05	1.44	1.12	1.08	1.07
24	3	-4,000	0.33	0.75	1.64	2.93	1.29	0.63	0.71	0.29
		-2,000	0.33	0.71	1.46	2.54	1.63	0.92	0.87	0.62
		0	0.33	0.67	1.28	2.15	1.57	1.21	1.03	0.95
		2,000	0.31	0.65	1.22	2.02	1.37	1.06	0.97	0.94
		4,000	0.28	0.62	1.16	1.89	1.17	0.91	0.91	0.93
36	3	-4,000	0.24	0.58	1.43	2.62	0.67	0.10	0.31	0.00
		-2,000	0.24	0.54	1.25	2.23	1.01	0.39	0.47	0.33
		0	0.24	0.50	1.07	1.84	0.95	0.68	0.63	0.66
		2,000	0.22	0.48	1.01	1.71	0.75	0.53	0.57	0.65
		4,000	0.19	0.45	0.95	1.58	0.55	0.38	0.51	0.64
48	3	-4,000	0.15	0.41	1.22	2.31	0.09	0.00	0.06	0.00
		-2,000	0.15	0.37	1.04	1.92	0.43	0.09	0.10	0.05
		0	0.15	0.33	0.86	1.53	0.37	0.20	0.26	0.38
		2,000	0.13	0.31	0.80	1.40	0.17	0.05	0.20	0.37
		4,000	0.10	0.28	0.74	1.27	0.00	0.00	0.14	0.36

A.5.1.2 Flat Oval Duct (use Round using minor axis)

A.5.1.3 Rectangular Duct

Table A.27 Insertion Loss for Rectangular Sheet Metal Ducts with 1-inch Fiberglass Lining

Internal Cross-Sectional Dimensions (inches)	Insertion loss (dB/ft)					
	Octave Band Center Frequency (Hz)					
	125	250	500	1000	2000	4000
6 × 6	0.6	1.5	2.7	5.8	7.4	4.3
6 × 10	0.5	1.2	2.4	5.1	6.1	3.7
6 × 12	0.5	1.2	2.3	5.0	5.8	3.6
6 × 18	0.5	1.0	2.2	4.7	5.2	3.3
8 × 8	0.5	1.2	2.3	5.0	5.8	3.6
8 × 12	0.4	1.0	2.1	4.5	4.9	3.2
8 × 16	0.4	0.9	2.0	4.3	4.5	3.0
8 × 24	0.4	0.8	1.9	4.0	4.1	2.8
10 × 10	0.4	1.0	2.1	4.4	4.7	3.1
10 × 16	0.4	0.8	1.9	4.0	4.0	2.7
10 × 20	0.3	0.8	1.8	3.8	3.7	2.6
10 × 30	0.3	0.7	1.7	3.6	3.3	2.4
12 × 12	0.4	0.8	1.9	4.0	4.1	2.8
12 × 18	0.3	0.7	1.7	3.7	3.5	2.5
12 × 24	0.3	0.6	1.7	3.5	3.2	2.3
12 × 36	0.3	0.6	1.6	3.3	2.9	2.2
15 × 15	0.3	0.7	1.7	3.6	3.3	2.4
15 × 22	0.3	0.6	1.6	3.3	2.9	2.2
15 × 30	0.3	0.5	1.5	3.1	2.6	2.0
15 × 45	0.2	0.5	1.4	2.9	2.4	1.9
18 × 18	0.3	0.6	1.6	3.3	2.9	2.2
18 × 28	0.2	0.5	1.4	3.0	2.4	1.9
18 × 36	0.2	0.5	1.4	2.8	2.2	1.8
18 × 54	0.2	0.4	1.3	2.7	2.0	1.7
24 × 24	0.2	0.5	1.4	2.8	2.2	1.8
24 × 36	0.2	0.4	1.2	2.6	1.9	1.6
24 × 48	0.2	0.4	1.2	2.4	1.7	1.5
24 × 72	0.2	0.3	1.1	2.3	1.6	1.4
30 × 30	0.2	0.4	1.2	2.5	1.8	1.5
30 × 45	0.2	0.3	1.1	2.3	1.6	1.4
30 × 60	0.2	0.3	1.1	2.2	1.4	1.3
30 × 90	0.1	0.3	1.0	2.1	1.3	1.2

Internal Cross-Sectional Dimensions (inches)	Insertion loss (dB/ft)					
	Octave Band Center Frequency (Hz)					
	125	**250**	**500**	**1000**	**2000**	**4000**
36 × 36	0.2	0.3	1.1	2.3	1.6	1.4
36 × 54	0.1	0.3	1.0	2.1	1.3	1.2
36 × 72	0.1	0.3	1.0	2.0	1.2	1.2
36 × 108	0.1	0.2	0.9	1.9	1.1	1.1
42 × 42	0.2	0.3	1.0	2.1	1.4	1.3
42 × 64	0.1	0.3	0.9	1.9	1.2	1.1
42 × 84	0.1	0.2	0.9	1.8	1.1	1.1
42 × 126	0.1	0.2	0.9	1.7	1.0	1.0
48 × 48	0.1	0.3	1.0	2.0	1.2	1.2
48 × 72	0.1	0.2	0.9	1.8	1.0	1.0
48 × 96	0.1	0.2	0.8	1.7	1.0	1.0
48 × 144	0.1	0.2	0.8	1.6	0.9	0.9

Notes:

1. Values based on measurements of surface-coated duct liners of 1.5 lb/ft^3 density.
2. For the specific materials tested, liner density had a minor effect over the nominal range of 1.5 to 3 lb/ft^3.
3. Add natural attenuation (Table 7 in Chapter 46 of ASHRAE's 1999 *HVAC Applications* Handbook) to obtain total attenuation.

Table A.28 Insertion Loss for Rectangular Sheet Metal Ducts with 2-inch Fiberglass Lining

Internal Cross-Sectional Dimensions (inches)	Insertion loss (dB/ft)					
	Octave Band Center Frequency (Hz)					
	125	250	500	1000	2000	4000
6 × 6	0.8	2.9	4.9	7.2	7.4	4.3
6 × 10	0.7	2.4	4.4	6.4	6.1	3.7
6 × 12	0.6	2.3	4.2	6.2	5.8	3.6
6 × 18	0.6	2.1	4.0	5.8	5.2	3.3
8 × 8	0.6	2.3	4.2	6.2	5.8	3.6
8 × 12	0.6	1.9	3.9	5.6	4.9	3.2
8 × 16	0.5	1.8	3.7	5.4	4.5	3.0
8 × 24	0.5	1.6	3.5	5.0	4.1	2.8
10 × 10	0.6	1.9	3.8	5.5	4.7	3.1
10 × 16	0.5	1.6	3.4	5.0	4.0	2.7
10 × 20	0.4	1.5	3.3	4.8	3.7	2.6
10 × 30	0.4	1.3	3.1	4.5	3.3	2.4
12 × 12	0.5	1.6	3.5	5.0	4.1	2.8
12 × 18	0.4	1.4	3.2	4.6	3.5	2.5
12 × 24	0.4	1.3	3.0	4.3	3.2	2.3
12 × 36	0.4	1.2	2.9	4.1	2.9	2.2
15 × 15	0.4	1.3	3.1	4.5	3.3	2.4
15 × 22	0.4	1.2	2.9	4.1	2.9	2.2
15 × 30	0.3	1.1	2.7	3.9	2.6	2.0
15 × 45	0.3	1.0	2.6	3.6	2.4	1.9
18 × 18	0.4	1.2	2.9	4.1	2.9	2.2
18 × 28	0.3	1.0	2.6	3.7	2.4	1.9
18 × 36	0.3	0.9	2.5	3.5	2.2	1.8
18 × 54	0.3	0.8	2.3	3.3	2.0	1.7
24 × 24	0.3	0.9	2.5	3.5	2.2	1.8
24 × 36	0.3	0.8	2.3	3.2	1.9	1.6
24 × 48	0.2	0.7	2.2	3.0	1.7	1.5
24 × 72	0.2	0.7	2.0	2.9	1.6	1.4
30 × 30	0.2	0.8	2.2	3.1	1.8	1.6
30 × 45	0.2	0.7	2.0	2.9	1.6	1.4
30 × 60	0.2	0.6	1.9	2.7	1.4	1.3
30 × 90	0.2	0.5	1.8	2.6	1.3	1.2

Internal Cross-Sectional Dimensions (inches)	Insertion loss (dB/ft)					
	Octave Band Center Frequency (Hz)					
	125	**250**	**500**	**1000**	**2000**	**4000**
36 × 36	0.2	0.7	2.0	2.9	1.6	1.4
36 × 54	0.2	0.6	1.9	2.6	1.3	1.2
36 × 72	0.2	0.5	1.8	2.5	1.2	1.2
36 × 108	0.2	0.5	1.7	2.3	1.1	1.1
42 × 42	0.2	0.6	1.9	2.6	1.4	1.3
42 × 64	0.2	0.5	1.7	2.4	1.2	1.1
42 × 84	0.2	0.5	1.6	2.3	1.1	1.1
42 × 126	0.1	0.4	1.6	2.2	1.0	1.0
48 × 48	0.2	0.5	1.8	2.5	1.2	1.2
48 × 72	0.2	0.4	1.6	2.3	1.0	1.0
48 × 96	0.1	0.4	1.5	2.1	1.0	1.0
48 × 144	0.1	0.4	1.5	2.0	0.9	0.9

Notes:

1. Values based on measurements of surface-coated duct liners of 1.5 lb/ft^3 density.
2. For the specific materials tested, liner density had a minor effect over the nominal range of 1.5 to 3.0 lb/ft^3.
3. Add natural attenuation (Table 7 in Chapter 46 of ASHRAE's 1999 *HVAC Applications* Handbook) to obtain total attenuation.

A.5.2 Elbows

Table A.29 Insertion Loss of Double-Wall, 1.5 Centerline Radius Round Elbows with a Perforated Inner Liner (dB)

Elbow Diameter (inches)	Insulation Thickness (inches)	Octave Band							
		1 63 Hz	2 125 Hz	3 250 Hz	4 500 Hz	5 1,000 Hz	6 2,000 Hz	7 4,000 Hz	8 8,000 Hz
6	1	0	3	4	8	15	23	18	12
12		1	1	4	8	17	17	12	12
18		0	3	4	10	15	14	13	14
24		0	1	4	8	13	11	14	13
36		0	1	3	9	10	10	11	8
48		0	1	5	9	9	11	11	10
6	2	1	5	10	16	24	24	22	14
12		2	3	8	14	23	19	13	15
18		0	4	8	18	18	16	14	19
24		0	3	7	18	17	13	16	18
36		0	3	9	18	12	12	14	13
48		1	4	11	13	10	13	13	13
6	3	1	9	16	22	27	24	22	14
12		4	6	14	17	23	21	13	15
18		0	6	12	22	18	16	14	19
24		0	4	9	21	19	15	19	20
36		0	5	14	22	15	15	19	16
48		2	6	15	14	11	14	16	13

Insertion loss testing of ACOUSTI-k27® elbows with airflow has also been conducted. The test data indicate little if any change of insertion loss under condition of airflow.

A.5.3 Single-Wall Round Duct GNL

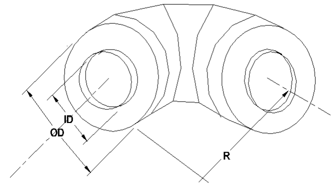
Table A.30 Single-Wall Duct Generated Noise Level

Duct Size (inches)	In-Duct Velocity (fpm)	Generated Noise Level (dB)							
		Octave Band Center Frequency (Hz)							
		63	125	250	500	1000	2000	4000	8000
6	2,000	32	32	32	31	28	28	28	28
	4,000	57	45	44	45	44	40	46	46
12	2,000	55	45	41	40	40	36	27	27
	4,000	59	53	55	54	57	58	53	52
18	2,000	60	52	49	42	47	45	29	29
	4,000	64	61	58	55	57	58	52	52
24	2,000	64	58	47	43	42	38	29	29
	4,000	69	68	60	56	56	57	51	51

A.5.4 Silencers

A.5.4.1 Round

McGill AirPressure Corporation



CEF-HV-L20

Circular, Elbow, Fiber-Filled
High Velocity Silencer

Availability
Diameters from 3 to 26 inches, in 1 inch increments; 26 to 60 inch, in 2 inch

Table 1: Insertion Loss

ID (in.)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
6	-3000 to 3000	3	8	13	16	24	26	26	20
12	-3000 to 3000	3	6	13	15	20	22	15	13
18	-3000 to 3000	1	6	12	16	16	15	13	17
24	-3000 to 3000	1	4	9	17	18	16	18	18
36	-3000 to 3000	1	5	12	16	15	15	18	14
48	-3000 to 3000	2	5	13	13	12	14	15	14

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies.
Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Dimensions

ID (in.)	OD (in.)	Elbow Radius
3 to 60	ID + 6	1.5(ID) + 9

INSERT ELBOW FIGURES FROM DKE SILENCER DATA SHEET HERE

Airflow Generated Sound Power

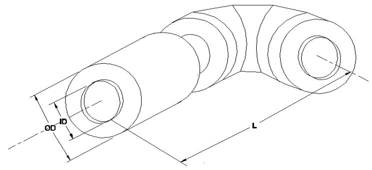
This silencer does not have internal components that would cause generated noise. The results of laboratory testing indicate indiscernible differences between the noise generated by the silencer and the noise generated by the connecting duct.

Table 3: Pressure Loss

ID (in.)	Total Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm)				
			1000	1500	2000	2500	3000
3	15	0.92	0.06	0.13	0.23	0.36	0.52
6	25	0.60	0.04	0.08	0.15	0.23	0.34
12	50	0.38	0.02	0.05	0.09	0.15	0.21
18	85	0.25	0.02	0.04	0.06	0.10	0.14
24	140	0.18	0.01	0.03	0.04	0.07	0.10
36	255	0.12	0.01	0.02	0.03	0.05	0.07
48	510	0.10	0.01	0.01	0.02	0.04	0.06
60	2015	0.08	0.00	0.01	0.02	0.03	0.04

Note: Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

McGill AirPressure Corporation



CEF-HV-L55

Circular, Elbow, Fiber-Filled High Velocity Silencer

Availability
Diameters from 3 to 26 inches, in 1 inch increments; 26 to 60 inch, in 2 inch

Table 1: Insertion Loss

ID (in.)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
6	-3000	15	22	35	33	38	45	38	25
	0	12	22	33	32	39	45	39	24
	3000	13	20	33	32	39	46	40	26
12	-3000	11	16	31	29	41	45	30	24
	0	10	16	29	29	41	44	27	36
	3000	10	15	28	29	42	45	28	25
18	-3000	4	17	36	36	39	31	28	28
	0	3	15	32	33	39	31	26	27
	3000	2	14	27	32	37	32	29	24
24	-3000	9	15	31	44	40	29	31	31
	0	6	15	28	42	42	29	32	30
	3000	8	13	28	38	42	32	35	29
36	-3000	11	12	31	40	40	32	33	30
	0	8	13	28	38	42	32	35	29
	3000	10	11	28	34	43	36	38	29
48	-3000	13	21	41	46	27	28	29	24
	0	9	22	37	44	28	28	30	23
	3000	12	18	37	40	28	31	32	22

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies.
Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Dimensions

ID (in.)	Straight Portion OD (in.)	Elbow Portion OD (in.)	Minimum Centerline Length[2] (in.)	Elbow Radius
3	ID + 12	ID + 6	1.5(ID + 6) +3(ID) + 9	1.5(ID) + 9
4	ID + 12	ID + 6	1.5(ID + 6) +3(ID) + 9	1.5(ID) + 9
5	ID + 12	ID + 6	1.5(ID + 6) +3(ID) + 9	1.5(ID) + 9
6 to 16	ID + 12	ID + 6	1.5(ID + 6) +3(ID) + 9	1.5(ID) + 9
17 to 30	ID + 12	ID + 6	1.5(ID + 6) +3(ID) + 7	1.5(ID) + 9
32 to 60	ID + 16	ID + 6	1.5(ID + 6) +3(ID) + 7	1.5(ID) + 9

Notes: (1) Any length of duct may be used between straight and elbow sections, but the standard length is 4 inches.

Airflow Generated Sound Power

This silencer does not have internal components that would cause generated noise. The results of laboratory testing indicate indiscernible differences between the noise generated by the silencer and the noise generated by the connecting duct.

Table 3: Pressure Loss

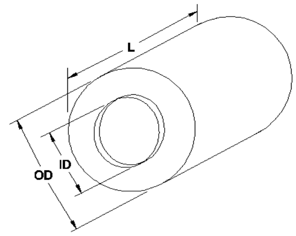
ID (in.)	Total Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm)				
			1000	1500	2000	2500	3000
3	35	0.96	0.06	0.13	0.24	0.37	0.54
6	55	0.62	0.04	0.09	0.15	0.24	0.35
12	125	0.40	0.02	0.06	0.10	0.16	0.22
18	230	0.27	0.02	0.04	0.07	0.11	0.15
24	370	0.20	0.01	0.03	0.05	0.08	0.11
36	1050	0.13	0.01	0.02	0.03	0.05	0.07
48	1860	0.11	0.01	0.02	0.03	0.04	0.06
60	2890	0.09	0.01	0.01	0.02	0.04	0.05

Note: Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

McGill AirPressure Corporation

CSF-HV-L15

Circular, Straight, Fiber-Filled High Velocity Silencer



Availability
Diameters from 3 to 26 inches, in 1 inch increments; 26 to 60 inch, in 2 inch increments. Custom lengths available.

Table 1: Insertion Loss

ID (in.)	OD (in.)	Length (in.)	Face Velocity (fpm)	Insertion Loss (dB)							
				63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
6	12	21	- 5000	1	2	2	3	3	2	2	1
			0	2	2	3	4	4	4	3	3
			5000	1	2	2	3	2	2	1	1
12	18	42	- 5000	0	2	4	5	3	3	2	2
			0	1	4	5	7	7	8	5	3
			5000	0	2	4	2	3	3	2	2
18	24	63	- 5000	1	3	7	7	8	10	8	8
			0	1	4	8	11	11	8	6	6
			5000	1	3	7	7	8	10	8	8
24	30	84	- 5000	1	4	7	14	8	6	4	3
			0	3	5	8	16	12	7	6	5
			5000	1	4	7	14	8	6	4	3
36	42	126	- 5000	0	3	7	14	10	6	2	2
			0	2	5	10	19	7	7	7	8
			5000	0	3	7	14	10	6	2	2
48	54	168	- 5000	0	1	5	2	5	5	0	0
			0	2	4	12	18	5	5	5	5
			5000	0	1	5	2	5	5	0	0

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies.
Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities. Length equal to 3.5 times the diameter.

Airflow Generated Sound Power

This silencer does not have internal components that would cause generated noise. The results of laboratory testing indicate indiscernible differences between the noise generated by the silencer and the noise generated by the connecting duct.

Table 2: Pressure Loss

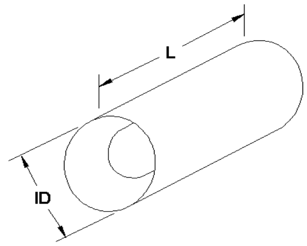
ID (in.)	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)				
			Face Velocity (fpm)				
			1000	2000	3000	4000	5000
6	10	0.034	0.00	0.01	0.02	0.03	0.05
12	30	0.024	0.00	0.01	0.01	0.02	0.04
18	65	0.020	0.00	0.00	0.01	0.02	0.03
24	130	0.017	0.00	0.00	0.01	0.02	0.03
36	295	0.014	0.00	0.00	0.01	0.01	0.02
48	660	0.014	0.00	0.00	0.01	0.01	0.02

Note: Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

McGill AirPressure Corporation

CSF-HV-L20

Circular, Straight, Fiber-Filled High Velocity Silencer



Availability
Diameters from 12 to 60 inch, in 2 inch increments.
Custom lengths available.

Table 1: Insertion Loss

ID (in.)	Length (in.)	Face Velocity (fpm)	Insertion Loss (dB)							
			63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
12	40	- 3000	1	2	7	13	15	9	9	8
		0	1	2	9	15	15	11	8	9
		3000	1	2	7	13	13	11	8	8
24	78	- 3000	4	4	11	18	15	9	9	9
		0	4	5	11	20	16	9	9	7
		3000	3	4	9	19	15	9	9	9
36	118	- 3000	4	4	11	20	14	10	8	14
		0	3	5	10	17	15	9	9	9
		3000	3	4	11	18	14	9	8	9
48	156	- 3000	5	5	12	20	14	8	7	6
		0	5	5	12	21	12	6	8	5
		3000	4	5	11	18	14	6	8	6
60	196	- 3000	6	6	14	18	12	5	6	5
		0	7	6	15	20	12	6	7	5
		3000	4	5	15	18	12	7	8	6

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities. Length equal to approximately 3.25 times the diameter.

Table 2: Airflow Generated Sound Power

ID (inches)	Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
24	-3000	71	67	65	60	59	58	55	48
	- 1500	63	53	47	41	42	35	28	23
	+1500	61	53	43	43	44	39	33	23
	+3000	69	64	61	57	58	60	58	50

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment.

Table 3: Face Area Adjustment Factor

Silencer Diameter (inches)						
12	18	24	34	48	68	96
-6	-3	0	+3	+6	+9	+12

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

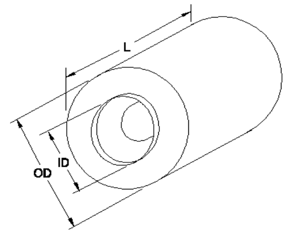
ID (inches)	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
			Face Velocity (fpm)					
			1000	1500	2000	2500	3000	4000
12	30	0.55	0.03	0.08	0.14	0.21	0.31	0.55
12 T	35	0.42	0.03	0.06	0.10	0.16	0.24	0.42
24	115	0.55	0.03	0.08	0.14	0.21	0.31	0.55
24 T	130	0.42	0.03	0.06	0.10	0.16	0.24	0.42
36	275	0.55	0.03	0.08	0.14	0.21	0.31	0.55
36 T	310	0.42	0.03	0.06	0.10	0.16	0.24	0.42
48	585	0.55	0.03	0.08	0.14	0.21	0.31	0.55
48 T	650	0.42	0.03	0.06	0.10	0.16	0.24	0.42
60	1020	0.55	0.03	0.08	0.14	0.21	0.31	0.55
60 T	1130	0.42	0.03	0.06	0.10	0.16	0.24	0.42

Notes: T denotes silencer with tail cone. Weights rounded up to nearest 5 lbs. Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

McGill AirPressure Corporation

CSF-HV-L30

Circular, Straight, Fiber-Filled



Availability
Diameters from 12 to 60 inch, in 2 inch increments. Custom lengths available.

Table 1: Insertion Loss

ID (in.)	OD (in.)	Length (in.)	Face Velocity (fpm)	Insertion Loss (dB)							
				63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
12	20	32	-4000	8	6	18	35	36	37	21	16
			0	3	4	18	31	32	35	22	14
			4000	4	6	14	22	30	34	19	12
24	32	48	-4000	6	6	17	31	34	27	15	12
			0	5	7	16	28	35	26	18	13
			4000	8	6	18	31	34	26	15	13
36	44	72	-4000	5	5	16	28	33	26	18	13
			0	5	6	14	22	31	23	17	14
			4000	5	6	15	20	29	25	17	14
48	56	96	-4000	12	11	19	35	31	22	15	15
			0	7	9	19	33	27	22	16	14
			4000	8	10	16	27	25	23	15	12
60	68	120	-4000	12	14	20	35	31	20	12	12
			0	13	13	22	37	31	21	10	14
			4000	9	13	21	32	30	21	12	13

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities. Length equal to 2 times the diameter or 32 inches, whichever is longer.

Table 2: Airflow Generated Sound Power

ID (inches)	Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
24	-4000	69	64	65	65	63	61	55	55
	-2000	67	55	53	48	49	38	31	32
	2000	63	55	47	48	51	46	38	31
	4000	71	64	61	61	64	64	62	55

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment.

Table 3: Face Area Adjustment Factor

Silencer Diameter (inches)						
12	18	24	34	48	68	96
-6	-3	0	+3	+6	+9	+12

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

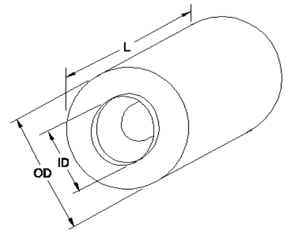
ID (inches)	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm)					
			1000	2000	3000	4000	5000	6000
12	60	0.25	0.02	0.06	0.14	0.25	0.39	0.56
12 T	65	0.21	0.01	0.05	0.12	0.21	0.33	0.47
24	155	0.25	0.02	0.06	0.14	0.25	0.39	0.56
24 T	165	0.21	0.01	0.05	0.12	0.21	0.33	0.47
36	330	0.25	0.02	0.06	0.14	0.25	0.39	0.56
36 T	350	0.21	0.01	0.05	0.12	0.21	0.33	0.47
48	630	0.25	0.02	0.06	0.14	0.25	0.39	0.56
48 T	675	0.21	0.01	0.05	0.12	0.21	0.33	0.47
60	1220	0.25	0.02	0.06	0.14	0.25	0.39	0.56
60 T	1285	0.21	0.01	0.05	0.12	0.21	0.33	0.47

Notes: T denotes silencer with tail cone. Weights rounded up to nearest 5 lbs. Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

McGill AirPressure Corporation

CSF-HV-L30

Circular, Straight, Fiber-Filled



Availability
Diameters from 12 to 60 inch, in 2 inch increments. Custom lengths available.

Table 1: Insertion Loss

ID (in.)	OD (in.)	Length (in.)	Face Velocity (fpm)	Insertion Loss (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
12	20	32	-4000	8	6	18	35	36	37	21	16
			0	3	4	18	31	32	35	22	14
			4000	4	6	14	22	30	34	19	12
24	32	48	-4000	6	6	17	31	34	27	15	12
			0	5	7	16	28	35	26	18	13
			4000	8	6	18	31	34	26	15	13
36	44	72	-4000	5	5	16	28	33	26	18	13
			0	5	6	14	22	31	23	17	14
			4000	5	6	15	20	29	25	17	14
48	56	96	-4000	12	11	19	35	31	22	15	15
			0	7	9	19	33	27	22	16	14
			4000	8	10	16	27	25	23	15	12
60	68	120	-4000	12	14	20	35	31	20	12	12
			0	13	13	22	37	31	21	10	14
			4000	9	13	21	32	30	21	12	13

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities. Length equal to 2 times the diameter or 32 inches, whichever is longer.

Table 2: Airflow Generated Sound Power

ID (inches)	Face Velocity (fpm)	Airflow Generated Sound Power Level (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
24	-4000	69	64	65	65	63	61	55	55
	-2000	67	55	53	48	49	38	31	32
	2000	63	55	47	48	51	46	38	31
	4000	71	64	61	61	64	64	62	55

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment.

Table 3: Face Area Adjustment Factor

Silencer Diameter (inches) 12	18	24	34	48	68	96
-6	-3	0	+3	+6	+9	+12

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

ID (inches)	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm) 1000	2000	3000	4000	5000	6000
12	60	0.25	0.02	0.06	0.14	0.25	0.39	0.56
12 T	65	0.21	0.01	0.05	0.12	0.21	0.33	0.47
24	155	0.25	0.02	0.06	0.14	0.25	0.39	0.56
24 T	165	0.21	0.01	0.05	0.12	0.21	0.33	0.47
36	330	0.25	0.02	0.06	0.14	0.25	0.39	0.56
36 T	350	0.21	0.01	0.05	0.12	0.21	0.33	0.47
48	630	0.25	0.02	0.06	0.14	0.25	0.39	0.56
48 T	675	0.21	0.01	0.05	0.12	0.21	0.33	0.47
60	1220	0.25	0.02	0.06	0.14	0.25	0.39	0.56
60 T	1285	0.21	0.01	0.05	0.12	0.21	0.33	0.47

Notes: T denotes silencer with tail cone. Weights rounded up to nearest 5 lbs. Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

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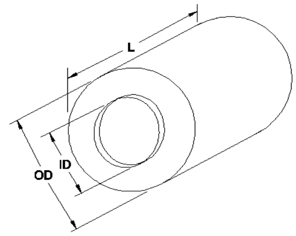


CSF-HV-L35

Circular, Straight, Fiber-Filled
High Velocity Silencer

Availability
Diameters from 3 to 26 inches, in 1 inch increments; 26 to 60 inch, in 2 inch increments. Custom lengths available.

Table 1: Insertion Loss

ID (in.)	OD (in.)	Length (in.)	Face Velocity (fpm)	Insertion Loss (dB)							
				63Hz	125Hz	250Hz	500Hz	1000Hz	2000	4000Hz	8000Hz
6	18	18	- 5000	5	10	19	25	25	18	10	8
			0	5	9	19	24	25	22	12	9
			5000	5	7	18	22	24	20	14	9
12	24	36	- 5000	5	9	19	21	29	20	13	9
			0	5	10	16	20	27	21	13	26
			5000	5	8	15	19	28	22	15	13
18	30	54	- 5000	3	10	21	25	26	17	14	13
			0	3	10	21	22	26	17	13	12
			5000	2	10	18	20	26	17	16	12
24	36	72	- 5000	9	11	23	30	24	18	14	15
			0	6	11	21	27	23	17	15	15
			5000	7	10	19	24	24	20	18	14
36	52	126	- 5000	13	16	29	34	15	14	11	12
			0	9	17	26	31	14	14	12	12
			5000	10	15	23	28	15	17	14	11
48	64	168	- 5000	13	17	27	28	11	11	8	8
			0	9	18	25	25	11	11	9	8
			5000	10	16	22	22	11	13	11	8

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities. For diameters from 3 to 30 in., length equal to 3 times the diameter or 18 inches, whichever is longer. For diameters greater than 30 inches, length equal to 3.5 times the diameter.

Airflow Generated Sound Power

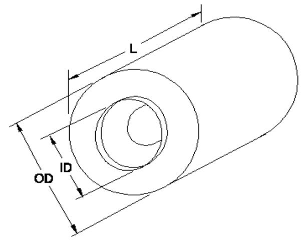
This silencer does not have internal components that would cause generated noise. The results of laboratory testing indicate indiscernible differences between the noise generated by the silencer and the noise generated by the connecting duct.

Table 2: Pressure Loss

ID (in.)	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)				
			Face Velocity (fpm)				
			1000	2000	3000	4000	5000
3	20	0.039	0.00	0.01	0.02	0.04	0.06
6	30	0.024	0.00	0.01	0.01	0.02	0.04
12	75	0.017	0.00	0.00	0.01	0.02	0.03
18	150	0.014	0.00	0.00	0.01	0.01	0.02
24	235	0.012	0.00	0.00	0.01	0.01	0.02
36	800	0.010	0.00	0.00	0.01	0.01	0.02
48	1355	0.010	0.00	0.00	0.01	0.01	0.02
60	2015	0.010	0.00	0.00	0.01	0.01	0.02

Note: Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

McGill AirPressure Corporation



CSF-HV-L45

Circular, Straight, Fiber-Filled
High Velocity Silencer

Availability
Diameters from 12 to 60 inches, in 2 inch increments. Standard lengths shown in Table 1.

Table 1: Insertion Loss

ID (in)	OD (In)	Length (in)	Face Velocity (fpm)	Insertion Loss (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
12	24	36	- 4000	11	17	25	29	36	35	23	15
			0	11	15	23	29	40	40	30	24
			4000	9	13	22	25	33	33	24	21
24	40	72	- 4000	10	16	18	25	31	25	13	10
			0	10	15	17	24	32	25	14	10
			4000	9	14	15	22	30	25	13	10
36	52	108	- 4000	10	17	20	26	29	20	15	12
			0	11	16	19	25	29	21	15	11
			4000	9	14	17	23	29	23	16	12

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities. Length equal to 2 times the diameter or 36 inches, whichever is longer.

Table 2: Airflow Generated Sound Power

ID (inches)	Face Velocity (fpm)	Airflow Generated Sound Power Level (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
24	-4000	75	73	67	63	59	62	61	55
	- 2000	61	58	48	43	44	46	44	39
	- 1000	45	41	30	27	29	31	29	26
	+1000	41	34	34	35	33	31	26	28
	+2000	57	45	47	50	49	47	42	39
	+4000	72	61	59	64	65	64	61	54

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment.

Table 3: Face Area Adjustment Factor

Silencer Diameter (inches)						
12	18	24	34	48	68	96
-6	-3	0	+3	+6	+9	+12

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

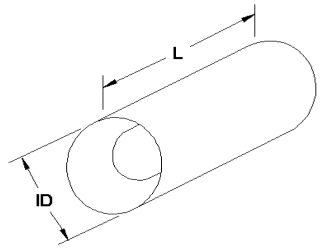
ID	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm) 1000	2000	3000	4000	4500
12	80	0.33	0.02	0.08	0.19	0.33	0.42
12 T	85	0.29	0.02	0.07	0.16	0.29	0.37
24	260	0.33	0.02	0.08	0.19	0.33	0.42
24 T	270	0.29	0.02	0.07	0.16	0.29	0.37
36	515	0.33	0.02	0.08	0.19	0.33	0.42
36 T	535	0.29	0.02	0.07	0.16	0.29	0.37

Note: T denotes silencer with tail cone. Weights rounded up to nearest 5 lbs. Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

McGill AirPressure Corporation

CSF-MV-L25

Circular, Straight, Fiber-Filled
Medium Velocity Silencer



Availability
Diameters from 12 to 60 inch, in 2 inch increments.
Length equal to approximately 3.25 times the diameter.
Custom lengths also available.

Table 1: Insertion Loss

ID (in.)	Length (in.)	Face Velocity (fpm)	Insertion Loss (dB)							
			63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
12	40	- 2500	5	2	12	20	28	14	13	10
		0	4	3	10	20	26	15	13	12
		2500	2	2	9	17	22	14	10	11
24	78	- 2500	6	8	18	27	28	14	10	13
		0	5	8	16	23	26	14	12	11
		2500	3	4	14	22	25	14	10	11
36	118	- 2500	3	6	14	22	24	15	10	11
		0	3	6	14	22	24	15	10	11
		2500	4	6	14	21	26	15	10	11
48	156	- 2500	8	11	19	27	26	9	11	11
		0	7	10	17	27	26	9	9	10
		2500	5	8	16	22	25	10	9	9
60	196	- 2500	8	13	22	28	24	7	8	11
		0	8	8	20	30	25	5	10	9
		2500	7	12	22	30	24	6	11	10

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power

ID (inches)	Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
24	-2500	67	58	54	53	56	53	46	38
	- 1500	63	47	48	44	45	37	25	23
	+1500	58	50	46	43	42	36	27	20
	+2500	66	57	52	52	55	53	48	41

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment.

Table 3: Face Area Adjustment Factor

Silencer Diameter (inches)						
12	18	24	34	48	68	96
-6	-3	0	+3	+6	+9	+12

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

ID (inches)	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
			Face Velocity (fpm)					
			500	1000	1500	2000	2500	3000
12	30	1.11	0.02	0.07	0.16	0.28	0.43	0.62
12 T	35	0.77	0.01	0.05	0.11	0.19	0.30	0.43
24	130	1.11	0.02	0.07	0.16	0.28	0.43	0.62
24 T	155	0.77	0.01	0.05	0.11	0.19	0.30	0.43
36	320	1.11	0.02	0.07	0.16	0.28	0.43	0.62
36 T	370	0.77	0.01	0.05	0.11	0.19	0.30	0.43
48	705	1.11	0.02	0.07	0.16	0.28	0.43	0.62
48 T	815	0.77	0.01	0.05	0.11	0.19	0.30	0.43
60	1265	1.11	0.02	0.07	0.16	0.28	0.43	0.62
60 T	1470	0.77	0.01	0.05	0.11	0.19	0.30	0.43

Notes: T denotes silencer with tail cone. Weights rounded up to nearest 5 lbs. Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

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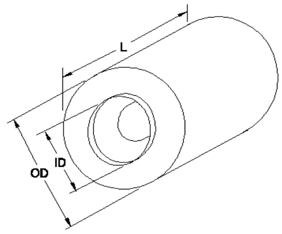

CSF-MV-L45

Circular, Straight, Fiber-Filled

Availability
Diameters from 12 to 60 inch, in 2 inch increments. Custom lengths available.

Table 1: Insertion Loss

ID (in.)	OD (in.)	Length (in.)	Face Velocity (fpm)	Insertion Loss (dB)							
				63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
12	20	32	-2500	5	10	26	31	32	35	35	28
			0	4	8	23	35	33	36	36	30
			2500	4	5	17	35	37	38	35	27
24	32	48	-2500	7	9	26	33	37	38	25	17
			0	6	10	21	34	37	39	28	18
			2500	4	9	21	34	39	39	26	22
36	44	72	-2500	9	15	27	36	33	31	22	16
			0	10	13	25	39	37	32	26	18
			2500	10	11	22	41	39	33	25	19
48	56	96	-2500	10	17	27	35	33	24	15	16
			0	10	14	25	39	34	24	21	18
			2500	10	12	22	39	38	27	23	18
60	68	120	-2500	11	19	31	36	29	20	13	15
			0	11	17	29	40	31	21	18	16
			2500	11	13	26	41	33	24	21	17

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities. Length equal to 2 times the diameter or 32 inches, whichever is longer.

Table 2: Airflow Generated Sound Power

ID (in.)	Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
24	-2500	72	64	63	66	62	59	55	54
	-1000	66	54	52	48	50	37	31	31
	1000	62	53	47	48	50	44	38	31
	2500	73	67	60	61	63	63	61	55

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment.

Table 3: Face Area Adjustment Factor

Silencer Diameter (inches)						
12	18	24	34	48	68	96
-6	-3	0	+3	+6	+9	+12

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

ID (in.)	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
			Face Velocity (fpm)					
			500	1000	1500	2000	2500	3000
12	65	0.98	0.02	0.06	0.14	0.24	0.38	0.55
12 T	70	0.64	0.01	0.04	0.09	0.16	0.25	0.36
24	170	0.98	0.02	0.06	0.14	0.24	0.38	0.55
24 T	195	0.64	0.01	0.04	0.09	0.16	0.25	0.36
36	370	0.98	0.02	0.06	0.14	0.24	0.38	0.55
36 T	425	0.64	0.01	0.04	0.09	0.16	0.25	0.36
48	755	0.98	0.02	0.06	0.14	0.24	0.38	0.55
48 T	870	0.64	0.01	0.04	0.09	0.16	0.25	0.36
60	1505	0.98	0.02	0.06	0.14	0.24	0.38	0.55
60 T	1710	0.64	0.01	0.04	0.09	0.16	0.25	0.36

Notes: T denotes silencer with tail cone. Weights rounded up to nearest 5 lbs. Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

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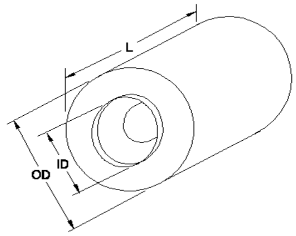

CSF-MV-L55

Circular, Straight, Fiber-Filled
Medium Velocity Silencer

Availability
Diameters from 12 to 60 inches, in 2 inch increments. Standard lengths shown in Table 1.

Table 1: Insertion Loss

ID. (inches)	OD (Inches)	Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
				63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
12	24	36	- 2000	14	19	32	39	39	38	29	22
			0	13	16	28	36	48	46	27	23
			2000	11	14	26	31	37	37	29	23
24	40	48	- 2000	11	19	26	33	39	37	20	14
			0	9	17	23	32	41	39	24	19
			2000	6	15	20	29	39	38	27	24
36	52	72	- 2000	13	22	29	34	42	31	21	17
			0	12	20	27	33	43	31	19	17
			2000	10	16	24	29	39	32	22	17

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities. Length equal to 2 times the diameter or 36 inches, whichever is longer.

Table 2: Airflow Generated Sound Power

ID (inches)	Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
24	- 2000								
	- 1000	51	45	38	34	33	34	33	29
	+1000	51	40	35	40	41	34	31	31
	+2000								

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment.

Table 3: Face Area Adjustment Factor

Silencer Diameter (inches)						
12	18	24	34	48	68	96
-6	-3	0	+3	+6	+9	+12

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

ID (inches)	Weight (lbs)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
			Face Velocity (fpm)					
			250	500	1000	1500	2000	2500
12	85	1.21	0.00	0.02	0.08	0.17	0.30	0.47
12 T	90	0.94	0.00	0.01	0.06	0.13	0.23	0.37
24	280	1.21	0.00	0.02	0.08	0.17	0.30	0.47
24 T	290	0.94	0.00	0.01	0.06	0.13	0.23	0.37
36	575	1.21	0.00	0.02	0.08	0.17	0.30	0.47
36 T	630	0.94	0.00	0.01	0.06	0.13	0.23	0.37

Notes: T denotes silencer with tail cone. Weights rounded up to nearest 5 lbs. Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

A.5.4 Silencers

A.5.4.2 Rectangular

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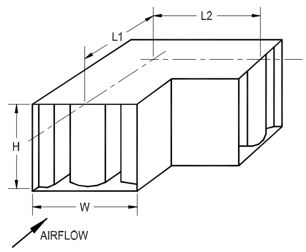

REF-LV-L33

Rectangular, Elbow, Fiber-filled
Low Velocity Silencer

Availability
L1 and L2: 2 ft and greater
W: 11.5-12.5, 23-25, 35-38, 46-50
H: any length (72 inch practical limit)

QuickRating = P17-L33-M88

See bottom of page for explanation.

Table 1: Insertion Loss

L1 x L2 (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36 x 36	- 1500	6	11	17	27	32	32	18	10
	0	6	11	16	26	31	31	21	13
	1500	5	9	15	23	28	27	18	10

Test data based on a 24Wx24Hx36L1x36L2 unit utilizing ASTM E 477 test method. Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
- 1500	(54)	(47)	39	38	37	40	39	33
- 1000	(46)	(40)	30	(33)	40	37	32	36
- 500	(45)	(32)	(27)	21	22	23	29	37
500	(43)	32	(25)	(24)	21	22	28	35
1000	(48)	(37)	26	33	33	33	28	26
1500	(55)	57	43	50	54	62	63	57

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Weight = 4.3 lb/ft^3

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

L1 x L2 (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36 x 36	2.80	0.04	0.10	0.17	0.27	0.39	0.70

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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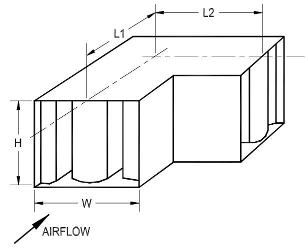

REF-LV-L38

Rectangular, Elbow, Fiber-filled Low Velocity Silencer

Availability
L1 and L2: 2 ft and greater
W: 11.5-12.5, 23-25, 35-38, 46-50
H: any length (72 inch practical limit)

QuickRating = P24-L38-M114

See bottom of page for explanation.

Table 1: Insertion Loss

L1 x L2 (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36 x 60	- 1250	7	14	20	35	40	35	21	10
	0	7	13	18	36	42	36	26	16
	1250	6	11	17	32	38	31	22	12

Test data based on a 24Wx24Hx36L1x60L2 unit utilizing ASTM E 477 test method. Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
- 1500	(54)	(47)	39	38	37	40	39	33
- 1000	(46)	(40)	30	(33)	40	37	32	36
- 500	(45)	(32)	(27)	21	22	23	29	37
500	(43)	32	(25)	(24)	21	22	28	35
1000	(48)	(37)	26	33	33	33	28	26
1500	(55)	57	43	50	54	62	63	57

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Weight = 4.3 lb/ft^3

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

L1 x L2 (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36 x 60	3.78	0.06	0.13	0.24	0.37	0.53	0.94

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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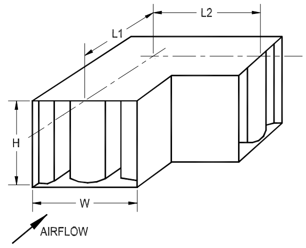

REF-MV-L39

Rectangular, Elbow, Fiber-filled Medium Velocity Silencer

Availability
L1 and L2: 2 ft and greater
W: 11.5-12.5, 23-25, 35-38, 46-50
H: any length (72 inch practical limit)

QuickRating = P15-L39-M110

See bottom of page for explanation.

Table 1: Insertion Loss

L1 x L2 (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36 x 36	- 1500	5	12	26	33	33	31	21	13
	0	5	10	24	32	40	38	24	15
	1500	5	9	22	28	36	34	22	15

Test data based on a 24Wx24Hx36L1x36L2 unit utilizing ASTM E 477 test method. Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
- 1500	58	55	44	45	48	55	54	47
- 1000	46	40	34	43	44	41	35	36
1000	(51)	40	34	40	43	43	38	34
1500	62	56	45	45	49	57	55	48

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.8 lb/ft^3

Table 4: Pressure Loss

L1 x L2 (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	2000	2500	3000
36 x 36	2.35	0.04	0.15	0.33	0.59	0.92	1.32

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled "QuickRating Guide" for further information.

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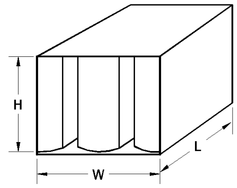

RSF-LV-L31

Rectangular, Straight, Fiber-Filled
Low Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 7-8, 14.5-15.5, 29-31, 44-47
H: any length (72 inch practical limit)

5 ft QuickRating = P17-L31-M111

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-1250	2	7	9	24	30	28	11	12
	0	2	6	8	19	27	20	16	12
	+1250	4	6	9	22	27	21	17	11
60	-1250	5	12	17	36	46	45	17	15
	0	6	11	14	29	44	36	23	15
	+1250	6	10	15	33	43	35	26	14
84	-1250	8	18	23	47	53	55	29	19
	0	7	16	20	37	54	47	37	23
	+1250	7	17	21	41	54	44	37	21
120	-1250	9	22	31	47	53	(60)	35	23
	0	8	19	27	46	56	53	48	27
	+1250	9	17	29	49	58	51	50	26

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1250	49	45	49	48	53	55	47	41
-750	32	36	30	32	28	30	22	27
+750	36	30	28	28	29	25	16	17
+1250	53	39	39	38	41	46	41	36

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 6.7 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36	2.55	0.04	0.09	0.16	0.25	0.36	0.64
60	2.78	0.04	0.10	0.17	0.27	0.39	0.69
84	3.03	0.05	0.11	0.19	0.30	0.43	0.76
120	3.43	0.05	0.12	0.21	0.33	0.48	0.86

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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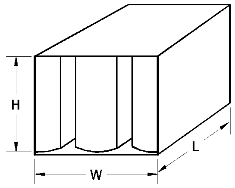

RSF-LV-L37

Rectangular, Straight, Fiber-Filled
Low Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 13-14, 26-28, 52-56

5 ft QuickRating = P29-L37-M89

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-1250	3	9	13	20	20	16	11	10
	0	3	9	13	21	23	17	16	17
	+1250	3	8	12	19	21	15	13	12
60	-1250	4	16	20	32	34	23	15	12
	0	5	16	20	34	38	24	21	21
	+1250	5	14	18	30	35	23	18	16
84	-1250	5	22	27	35	40	30	19	15
	0	7	22	27	38	46	31	26	25
	+1250	7	19	25	34	42	30	23	19
120	-1250	7	29	35	48	55	38	24	17
	0	9	29	36	52	(60)	42	34	29
	+1250	9	26	33	48	58	39	30	22

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1250	59	57	53	53	53	54	46	36
-500	57	48	43	38	35	31	26	24
+500	58	38	33	27	28	28	27	26
+1250	60	52	43	42	45	46	40	34

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.6 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm)					
		250	500	750	1000	1250	1500
36	3.34	0.01	0.05	0.12	0.21	0.33	0.47
60	4.73	0.02	0.07	0.17	0.29	0.46	0.66
84	5.79	0.02	0.09	0.20	0.36	0.56	0.81
120	7.13	0.03	0.11	0.25	0.44	0.69	1.00

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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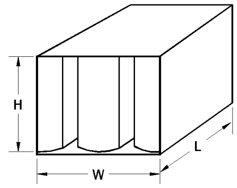

RSF-LV-L43

Rectangular, Straight, Fiber-Filled
Low Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 6-6.5, 11.5-12.5, 23-25, 35-38, 46-50
H: any length (72 inch practical limit)

5 ft QuickRating = P32-L43-M148

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	- 1000	7	11	20	28	37	35	27	18
	0	7	10	18	29	38	37	28	19
	1000	7	9	17	27	37	37	29	20
60	- 1000	9	15	26	42	51	48	37	22
	0	9	14	24	40	49	52	37	26
	1000	8	13	22	40	50	51	38	25
84	- 1000	11	17	29	44	51	52	48	30
	0	10	16	27	44	50	54	49	32
	1000	9	15	25	45	50	53	51	33
120	-1000	13	22	35	52	51	55	59	38
	0	12	19	32	53	53	56	(60)	43
	1000	11	18	30	53	52	56	58	43

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1000	46	41	50	50	57	55	48	42
-500	44	33	41	45	51	52	39	23
+500	34	29	29	29	30	27	24	22
+1000	45	43	38	35	37	39	35	33

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 7.9 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36	3.96	0.06	0.14	0.25	0.39	0.56	0.99
60	5.09	0.08	0.18	0.32	0.50	0.71	1.27
84	6.39	0.10	0.22	0.40	0.62	0.90	1.59
120	6.94	0.11	0.24	0.43	0.68	0.97	1.73

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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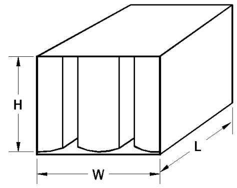

RSF-MV-L24

Rectangular, Straight, Fiber-Filled Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38

5 ft QuickRating = P08-L24-M93

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2000	1	5	7	15	28	24	12	10
	0	2	3	8	15	19	20	17	11
	+2000	2	2	6	11	17	20	17	13
60	-2000	2	11	14	31	34	36	17	14
	0	3	9	14	29	36	35	24	14
	+2000	3	7	11	21	31	33	23	16
84	-2000	3	13	19	41	42	48	19	12
	0	4	10	22	39	47	42	27	13
	+2000	4	10	15	29	42	40	27	17
120	-2000	9	20	23	50	(60)	(60)	31	21
	0	7	15	23	44	(60)	53	38	21
	+2000	9	16	16	34	59	55	38	25

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
- 2000	54	53	57	54	60	62	54	47
- 1000	37	41	45	39	43	42	38	27
+1000	39	36	35	37	33	36	30	32
+2000	53	52	50	48	46	55	48	45

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 6.2 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	2000	2500	3000
36	1.05	0.02	0.07	0.15	0.26	0.41	0.59
60	1.25	0.02	0.08	0.18	0.31	0.49	0.70
84	1.87	0.03	0.12	0.26	0.47	0.73	1.05
120	1.98	0.03	0.12	0.28	0.49	0.77	1.11

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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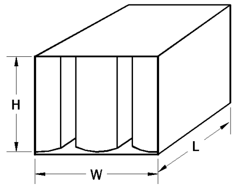

RSF-MV-L27

Rectangular, Straight, Fiber-Filled Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 8.5-9.5, 17-19, 34-38

5 ft QuickRating = P07-L27-M81

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2000	1	4	8	20	19	19	16	21
	0	4	3	11	19	20	15	18	16
	+2000	4	4	8	14	19	15	14	13
60	-2000	4	7	14	32	33	27	20	23
	0	5	7	18	31	32	23	21	19
	+2000	5	6	13	23	31	22	19	17
84	-2000	4	10	18	40	46	32	19	19
	0	7	9	21	39	48	25	25	19
	+2000	5	8	16	28	46	27	22	18
120	-2000	7	12	27	50	54	43	26	25
	0	9	10	30	49	57	34	31	20
	+2000	8	10	22	38	54	35	28	17

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
- 2000	58	52	58	56	59	60	55	47
- 1000	39	38	47	43	47	44	36	27
+1000	32	32	33	33	34	38	27	25
+2000	61	49	46	45	45	55	50	44

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 6.4 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	2000	2500	3000
36	1.08	0.02	0.07	0.15	0.27	0.42	0.61
60	1.10	0.02	0.07	0.15	0.27	0.43	0.62
84	1.30	0.02	0.08	0.18	0.32	0.51	0.73
120	1.68	0.03	0.10	0.24	0.42	0.65	0.94

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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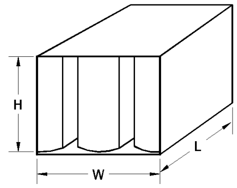

RSF-MV-L33

Rectangular, Straight, Fiber-Filled Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P09-L33-M70

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2500	5	7	12	19	18	13	14	11
	0	4	7	12	17	17	11	10	7
	2500	4	7	12	16	17	11	9	8
60	-2500	7	11	18	30	29	17	18	15
	0	6	10	17	27	27	16	13	10
	2500	5	10	17	26	27	16	12	10
84	-2500	11	14	24	39	39	22	21	17
	0	10	14	23	36	37	20	16	12
	2500	8	14	23	35	38	20	14	12
120	-2500	15	19	32	52	50	28	24	19
	0	14	18	31	49	47	26	18	14
	2500	11	18	31	48	49	26	17	15

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-2500	59	58	57	56	58	62	56	44
-1000	37	41	45	39	43	42	38	27
1000	39	36	35	37	33	36	30	32
2500	61	59	56	50	51	55	51	34

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.8 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	2000	2500	3000
36	0.91	0.01	0.06	0.13	0.23	0.35	0.51
60	1.43	0.02	0.09	0.20	0.36	0.56	0.80
84	2.17	0.03	0.14	0.30	0.54	0.85	1.22
120	3.28	0.05	0.20	0.46	0.82	1.28	1.84

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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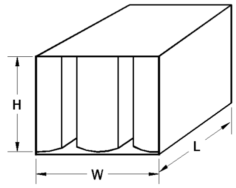

RSF-MV-L41

Rectangular, Straight, Fiber-Filled
Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P19-L41-M82

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	- 1500	6	7	13	20	20	16	14	14
	0	4	7	14	20	20	17	14	14
	1500	4	8	14	19	18	15	14	14
60	- 1500	9	11	21	32	32	22	18	17
	0	8	11	22	32	32	21	17	16
	1500	7	12	22	29	29	21	18	17
84	- 1500	12	14	28	43	43	29	21	20
	0	11	14	30	44	44	27	20	19
	1500	9	15	30	40	40	26	22	20
120	-1500	16	19	36	57	57	37	27	23
	0	15	19	39	58	60	36	27	22
	1500	12	21	40	53	54	35	29	24

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1500	57	53	50	54	52	51	44	29
750	55	48	42	46	42	34	22	29
750	52	40	38	33	40	33	24	25
1500	57	51	46	44	49	48	44	29

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.9 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		250	500	750	1000	1500	1750
36	2.28	0.01	0.04	0.08	0.14	0.32	0.44
60	3.06	0.01	0.05	0.11	0.19	0.43	0.58
84	4.55	0.02	0.07	0.16	0.28	0.64	0.87
120	6.81	0.03	0.11	0.24	0.42	0.96	1.30

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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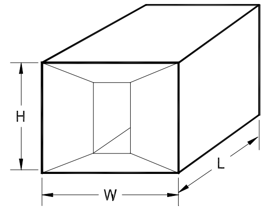
RSF-PV-L45

Rectangular, Straight, Fiber-Filled Plenum Velocity Silencer



Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P54-L45-M97

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-750	8	6	13	17	34	20	17	14
	0	8	5	13	19	22	16	17	14
	750	6	4	12	17	21	16	14	10
60	-750	12	14	24	35	40	30	25	18
	0	12	13	24	36	41	28	24	18
	750	11	13	21	33	38	26	19	13
84	-750	14	17	34	47	50	40	28	17
	0	12	15	37	49	54	34	27	17
	750	11	17	30	46	51	32	22	14
120	-750	17	26	40	57	(60)	54	45	29
	0	15	23	39	56	(60)	42	39	26
	750	17	28	32	53	(60)	43	31	20

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-750	(54)	48	45	44	44	43	42	39
-500	(53)	(41)	36	37	38	31	33	34
500	(52)	(40)	(29)	28	27	28	32	35
750	(54)	46	38	35	37	40	41	39

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.5 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm) 250	500	750	1000	1250	1500
36	7.20	0.03	0.11	0.25	0.45	0.70	1.01
60	8.73	0.03	0.14	0.31	0.54	0.85	1.22
84	11.00	0.04	0.17	0.39	0.69	1.07	1.54
120	13.05	0.05	0.20	0.46	0.81	1.27	1.83

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at100 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled “QuickRating Guide” for further information.

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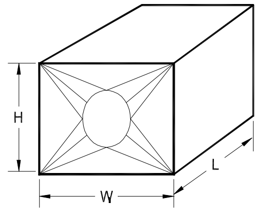
RSF-PV-L47

Rectangular, Straight, Fiber-Filled Plenum Velocity Silencer



Availability
L: 5 ft
W: 23.5-24.5, 47-48, 94-96

5 ft QuickRating = P60-L47-M87

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
60	-1000	11	13	24	33	35	24	20	10
	0	12	15	25	33	36	21	18	13
	+1000	11	13	23	30	35	22	20	15

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1000	57	57	55	52	50	50	47	44
-500	53	42	37	36	33	25	26	32
+500	52	40	29	25	23	28	32	35
+1000	54	55	47	42	42	47	44	36

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm) 250	500	750	1000	1250	1500
60	9.59	0.04	0.15	0.34	0.60	0.93	1.02

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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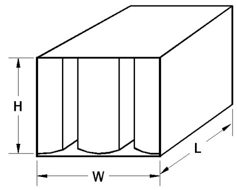

RSF-PV-L51

Rectangular, Straight, Fiber-Filled Plenum Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50
H: any length (72 inch practical limit)

5 ft QuickRating = P80-L51-M128

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-500	6	11	19	27	32	29	21	18
	0	6	10	19	28	30	30	22	16
	500	6	10	19	28	32	32	21	18
60	-500	10	18	29	41	46	44	27	21
	0	10	17	28	40	44	44	29	21
	500	8	16	27	39	45	44	27	22
84	-500	14	23	39	47	51	48	33	25
	0	13	26	37	48	50	48	35	27
	500	10	23	36	46	51	48	33	27
120	-500	18	29	47	56	(60)	(60)	40	30
	0	17	32	46	58	(60)	(60)	46	32
	500	13	29	45	59	(60)	(60)	44	33

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-500	58	44	38	48	43	39	25	32
500	51	41	36	32	35	28	24	27

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight =6.0 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		250	500	750	1000	1250	1500
36	10.41	0.04	0.16	0.37	0.65	1.01	1.46
60	12.45	0.05	0.19	0.44	0.78	1.21	1.75
84	14.40	0.06	0.22	0.51	0.90	1.40	2.02
120	17.71	0.07	0.28	0.62	1.10	1.73	2.48

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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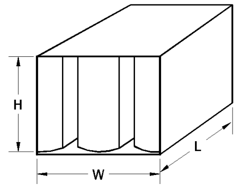

RSN-LV-L23

Rectangular, Straight, No-Fill
Low Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P22-L23-M35

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-1250	5	7	13	21	9	8	4	2
	0	5	6	10	18	9	7	7	8
	1250	4	5	12	19	9	5	4	5
60	-1250	6	8	14	23	10	8	5	2
	0	5	6	10	18	10	8	8	8
	1250	5	6	13	20	9	6	5	6
84	-1250	9	12	18	27	12	10	6	2
	0	7	9	12	21	13	10	9	9
	1250	6	9	16	23	11	7	6	7
120	-1250	12	16	23	33	15	12	7	3
	0	11	13	15	27	16	13	11	11
	1250	7	14	21	28	13	9	7	8

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1250	61	46	41	45	47	48	48	43
-500	59	39	36	45	39	27	25	25
+500	58	39	32	40	38	27	26	27
+1250	59	46	41	45	48	48	47	42

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 6.2 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36	2.99	0.05	0.10	0.19	0.29	0.42	0.75
60	3.51	0.05	0.12	0.22	0.34	0.49	0.88
84	4.48	0.07	0.16	0.28	0.44	0.63	1.12
120	5.48	0.09	0.19	0.34	0.53	0.77	1.37

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled "QuickRating Guide" for further information.

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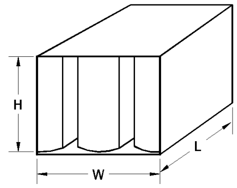

RSN-LV-L27

Rectangular, Straight, No-Fill
Low Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P22-L27-M42

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-1250	3	7	11	18	15	8	5	4
	0	3	6	10	16	13	8	5	4
	1250	5	7	12	16	15	8	5	4
60	-1250	4	8	12	19	16	9	6	4
	0	4	7	10	16	15	9	6	4
	1250	6	9	13	17	17	9	5	5
84	-1250	6	12	15	22	19	11	7	4
	0	5	10	12	19	18	11	7	5
	1250	7	13	16	20	20	11	7	6
120	-1250	8	16	19	27	24	13	8	4
	0	8	14	15	24	23	14	8	6
	1250	9	19	21	24	24	15	9	7

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1250	61	46	41	45	47	48	48	43
-500	59	39	36	45	39	27	25	25
+500	58	39	32	40	38	27	26	27
+1250	59	46	41	45	48	48	47	42

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.1 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36	2.99	0.05	0.10	0.19	0.29	0.42	0.75
60	3.51	0.05	0.12	0.22	0.34	0.49	0.88
84	4.48	0.07	0.16	0.28	0.44	0.63	1.12
120	5.48	0.09	0.19	0.34	0.53	0.77	1.37

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled “QuickRating Guide” for further information.

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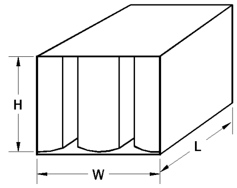

RSN-LV-L29

Rectangular, Straight, No-Fill
Low Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P22-L29-M45

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-1250	4	7	12	19	15	9	6	4
	0	4	6	11	17	13	9	6	5
	1250	6	8	13	17	16	9	6	5
60	-1250	6	8	13	20	16	10	7	4
	0	4	7	12	17	15	10	7	5
	1250	7	10	14	18	18	10	6	6
84	-1250	8	12	16	24	19	12	8	5
	0	7	10	13	20	18	12	8	6
	1250	9	14	17	21	21	13	8	7
120	-1250	10	16	21	29	24	15	10	6
	0	13	14	16	25	23	15	10	7
	1250	12	21	23	26	26	16	10	8

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1250	61	46	41	45	47	48	48	43
-500	59	39	36	45	39	27	25	25
+500	58	39	32	40	38	27	26	27
+1250	59	46	41	45	48	48	47	42

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.1 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36	2.99	0.05	0.10	0.19	0.29	0.42	0.75
60	3.51	0.05	0.12	0.22	0.34	0.49	0.88
84	4.48	0.07	0.16	0.28	0.44	0.63	1.12
120	5.48	0.09	0.19	0.34	0.53	0.77	1.37

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled “QuickRating Guide” for further information.

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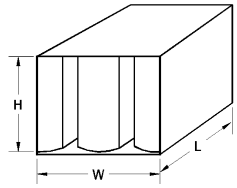

RSN-MV-L19

Rectangular, Straight, No-Fill
Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P09-L19-M28

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2000	4	7	10	12	11	8	9	7
	0	4	6	9	10	10	8	9	8
	2000	3	4	7	9	9	6	6	6
60	-2000	6	8	11	12	12	9	9	7
	0	4	7	10	11	11	9	9	8
	2000	4	5	7	9	9	6	6	7
84	-2000	8	12	14	14	15	11	11	8
	0	6	10	11	12	14	11	11	9
	2000	6	8	9	9	11	8	8	8
120	-2000	10	16	17	18	18	13	14	9
	0	8	14	14	15	17	14	14	11
	2000	10	12	12	13	13	10	10	9

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-2000	50	42	43	42	47	56	53	38
-1000	(46)	(37)	(32)	37	32	32	33	27
1000	(48)	(39)	(31)	36	35	40	40	33
2000	52	47	41	43	48	57	54	39

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 4.9 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1500	2000	2500
36	1.16	0.02	0.04	0.07	0.16	0.29	0.45
60	1.37	0.02	0.05	0.09	0.19	0.34	0.53
84	1.75	0.03	0.06	0.11	0.25	0.44	0.68
120	2.13	0.03	0.07	0.13	0.30	0.53	0.83

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled “QuickRating Guide” for further information.

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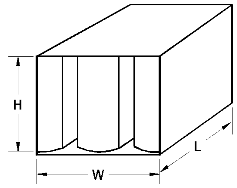

RSN-MV-L20

Rectangular, Straight, No-Fill
Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P12-L20-M36

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2000	5	6	7	11	14	14	14	10
	0	4	4	6	9	10	11	9	7
	2000	4	5	5	9	10	10	7	5
60	-2000	7	8	9	13	15	16	16	12
	0	5	6	8	12	12	14	11	8
	2000	6	6	7	10	12	12	8	6
84	-2000	8	11	11	16	18	19	20	14
	0	6	8	9	15	15	17	13	9
	2000	6	9	9	12	14	15	10	7
120	-2000	9	12	13	18	19	22	22	16
	0	7	10	10	17	17	19	15	11
	2000	7	11	10	13	16	18	11	8

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-2000	53	62	55	51	54	58	53	37
-1000	(49)	(52)	(43)	42	39	42	42	35
+1000	(52)	(46)	38	41	41	45	45	38
+2000	53	51	48	50	57	62	58	41

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 7.2 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	1750	2000	3000
36	1.66	0.03	0.10	0.23	0.32	0.41	0.93
60	1.86	0.03	0.12	0.26	0.36	0.46	1.04
84	2.77	0.04	0.17	0.39	0.53	0.69	1.55
120	2.94	0.05	0.18	0.41	0.56	0.73	1.65

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled “QuickRating Guide” for further information.

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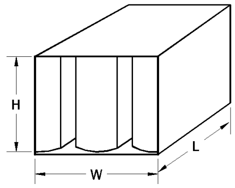

RSN-MV-L21

Rectangular, Straight, No-Fill Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P09-L21-M35

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2000	5	7	10	19	9	8	4	0
	0	4	6	7	18	7	7	5	4
	2000	6	6	9	19	9	8	5	3
60	-2000	6	9	11	20	9	9	5	0
	0	4	6	8	18	7	7	6	4
	2000	7	7	10	20	10	8	5	3
84	-2000	9	13	14	23	11	11	6	0
	0	6	9	9	21	9	9	7	5
	2000	8	10	13	23	12	10	7	3
120	-2000	12	17	17	29	14	13	7	0
	0	9	13	11	27	12	11	8	6
	2000	10	16	17	28	15	13	9	4

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-2000	50	42	43	42	47	56	53	38
-1000	(46)	(37)	(32)	37	32	32	33	27
1000	(48)	(39)	(31)	36	35	40	40	33
2000	52	47	41	43	48	57	54	39

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.4 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1500	2000	2500
36	1.16	0.02	0.04	0.07	0.16	0.29	0.45
60	1.37	0.02	0.05	0.09	0.19	0.34	0.53
84	1.75	0.03	0.06	0.11	0.25	0.44	0.68
120	2.13	0.03	0.07	0.13	0.30	0.53	0.83

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . QuickRatings for rectangular silencers may only be compared to other rectangular silencers. The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled “QuickRating Guide” for further information.

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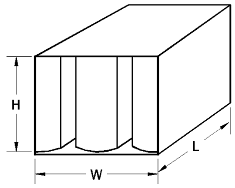

RSN-MV-L27

Rectangular, Straight, No-Fill
Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P12-L27-M43

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2000	7	10	11	16	15	13	9	4
	0	6	6	6	9	10	11	8	2
	2000	7	8	9	14	13	13	8	6
60	-2000	10	13	14	18	16	15	10	5
	0	7	8	7	12	12	14	9	3
	2000	9	10	12	16	15	16	10	7
84	-2000	13	17	17	22	19	18	12	6
	0	10	11	8	15	15	17	11	4
	2000	12	14	15	19	18	20	12	8
120	-2000	13	19	20	24	20	20	13	7
	0	12	14	9	17	17	19	12	4
	2000	12	19	18	21	20	24	14	9

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-2000	53	62	55	51	54	58	53	37
-1000	(49)	(52)	(43)	42	39	42	42	35
+1000	(52)	(46)	38	41	41	45	45	38
+2000	53	51	48	50	57	62	58	41

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 7.2 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	1750	2000	3000
36	1.66	0.03	0.10	0.23	0.32	0.41	0.93
60	1.86	0.03	0.12	0.26	0.36	0.46	1.04
84	2.77	0.04	0.17	0.39	0.53	0.69	1.55
120	2.94	0.05	0.18	0.41	0.56	0.73	1.65

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled "QuickRating Guide" for further information.

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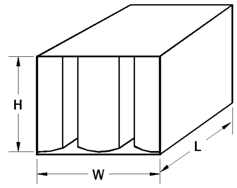

RSN-MV-L33

Rectangular, Straight, No-Fill
Medium Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P12-L33-M56

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2000	7	10	11	19	19	15	11	5
	0	8	7	9	16	14	14	11	6
	2000	7	9	10	17	15	15	9	6
60	-2000	11	13	15	22	20	17	12	6
	0	10	10	11	20	17	18	13	7
	2000	10	11	13	20	18	18	11	8
84	-2000	14	17	18	27	24	20	15	7
	0	14	14	13	24	21	22	15	8
	2000	13	16	16	24	21	23	14	9
120	-2000	15	19	21	30	26	23	16	8
	0	17	17	14	28	24	25	18	10
	2000	14	21	19	27	23	27	15	11

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-2000	53	62	55	51	54	58	53	37
-1000	(49)	(52)	(43)	42	39	42	42	35
+1000	(52)	(46)	38	41	41	45	45	38
+2000	53	51	48	50	57	62	58	41

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 7.5 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm) 500	1000	1500	1750	2000	3000
36	1.66	0.03	0.10	0.23	0.32	0.41	0.93
60	1.86	0.03	0.12	0.26	0.36	0.46	1.04
84	2.77	0.04	0.17	0.39	0.53	0.69	1.55
120	2.94	0.05	0.18	0.41	0.56	0.73	1.65

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled "QuickRating Guide" for further information.

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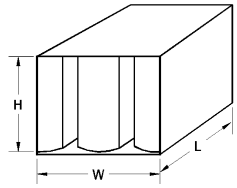

RSN-PV-L37

Rectangular, Straight, No-Fill
Plenum Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P70-L37-M54

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-750	6	10	19	28	15	11	12	9
	0	5	9	14	21	12	11	12	12
	750	6	9	18	25	15	11	10	11
60	-750	9	13	22	29	17	12	12	9
	0	6	10	14	21	13	12	12	12
	750	7	11	19	26	16	12	11	11
84	-750	13	18	27	34	20	14	15	10
	0	9	14	17	25	16	15	14	14
	750	8	16	24	31	19	15	14	13
120	-750	13	18	35	42	25	17	18	12
	0	9	14	21	31	20	18	18	17
	750	8	16	32	38	23	19	17	16

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-750	(51)	(41)	33	40	47	53	49	43
-250	(45)	(35)	31	29	29	30	32	25
+250	(45)	(35)	31	29	29	30	32	25
+750	49	42	41	45	48	53	49	40

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 6.4 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		250	500	750	1000	1250	1500
36	9.53	0.04	0.15	0.33	0.59	0.93	1.34
60	11.18	0.04	0.17	0.39	0.70	1.09	1.57
84	14.24	0.06	0.22	0.50	0.89	1.39	2.00
120	17.39	0.07	0.27	0.61	1.08	1.69	2.44

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled “QuickRating Guide” for further information.

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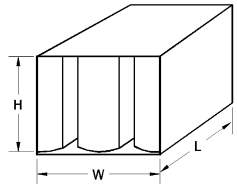

RSN-PV-L38

Rectangular, Straight, No-Fill Plenum Velocity Silencer

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P70-L38-M53

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-750	6	10	20	22	16	13	13	10
	0	5	8	14	12	10	9	9	10
	750	6	9	19	20	15	13	11	12
60	-750	9	13	23	23	18	15	14	10
	0	5	9	14	13	11	10	9	10
	750	7	11	20	21	17	15	13	12
84	-750	13	18	29	27	21	18	17	11
	0	8	12	17	15	14	12	11	11
	750	9	16	25	24	20	18	16	14
120	-750	16	24	36	33	26	21	21	13
	0	11	18	21	18	17	15	14	14
	750	12	25	34	30	24	23	20	17

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-750	(51)	(41)	33	40	47	53	49	43
-250	(45)	(35)	31	29	29	30	32	25
+250	(45)	(35)	31	29	29	30	32	25
+750	49	42	41	45	48	53	49	40

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 6.4 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		250	500	750	1000	1250	1500
36	9.53	0.04	0.15	0.33	0.59	0.93	1.34
60	11.18	0.04	0.17	0.39	0.70	1.09	1.57
84	14.24	0.06	0.22	0.50	0.89	1.39	2.00
120	17.39	0.07	0.27	0.61	1.08	1.69	2.44

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 1000 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 1000 fpm. See the sheet titled “QuickRating Guide” for further information.

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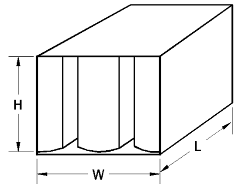


RSV-LV-L25

Rectangular, Straight, Low Velocity Silencer with Vapor Barrier

Availability
L: 3 ft and greater (sections if L>12ft)
W: 7-8, 14-16, 28-32, 42-48
H: any length (72 inch practical limit)

5 ft QuickRating = P17-L25-M77

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-1250	3	7	10	10	18	18	12	12
	0	3	6	9	12	21	14	14	12
	+1250	3	6	10	10	16	14	15	10
60	-1250	7	9	13	20	30	29	18	17
	0	6	9	14	24	31	23	22	19
	+1250	4	8	12	24	30	23	21	13
84	-1250	10	14	20	29	41	38	23	20
	0	7	14	21	30	41	31	27	23
	+1250	6	12	20	32	41	32	29	20
120	-1250	11	22	29	36	50	48	29	26
	0	8	20	29	40	52	39	33	27
	+1250	6	17	26	44	53	40	36	26

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1250	49	45	49	48	53	55	47	41
-750	32	36	30	32	28	30	22	27
+750	36	30	28	28	29	25	16	17
+1250	53	39	39	38	41	46	41	36

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Weight = 6.7 lb/ft^3

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

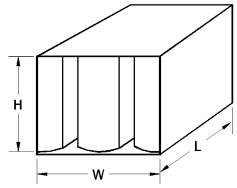
Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36	2.55	0.04	0.09	0.16	0.25	0.36	0.64
60	2.78	0.04	0.10	0.17	0.27	0.39	0.69
84	3.03	0.05	0.11	0.19	0.30	0.43	0.76
120	3.43	0.05	0.12	0.21	0.33	0.48	0.86

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

McGill AirPressure Corporation



RSV-LV-L35

Rectangular, Straight, Fiber-filled Low Velocity Silencer with **FDA Approved** Vapor Barrier

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P19-L35-M42

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	- 1500	6	9	9	11	10	10	10	10
	0	5	8	9	10	10	10	11	10
	1500	5	8	9	10	9	9	9	9
60	- 1500	10	14	15	17	15	13	12	12
	0	9	13	15	16	16	13	13	12
	1500	9	12	14	15	15	12	11	11
84	- 1500	14	18	20	23	20	17	14	14
	0	13	17	20	22	22	16	15	14
	1500	12	15	19	20	21	15	13	13
120	-1500	18	24	26	30	27	22	18	17
	0	17	23	27	29	30	22	20	17
	1500	16	21	25	28	28	20	18	15

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1500	57	53	50	54	52	51	44	29
750	55	48	42	46	42	34	22	29
750	52	40	38	33	40	33	24	25
1500	57	51	46	44	49	48	44	29

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Weight = 5.9 lb/ft^3

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		250	500	750	1000	1500	1750
36	2.28	0.01	0.04	0.08	0.14	0.32	0.44
60	3.06	0.01	0.05	0.11	0.19	0.43	0.58
84	4.55	0.02	0.07	0.16	0.28	0.64	0.87
120	6.81	0.03	0.11	0.24	0.42	0.96	1.30

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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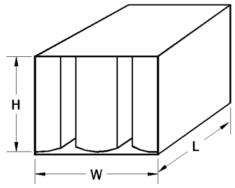


RSV-LV-L36

Rectangular, Straight, Low Velocity Silencer with Vapor Barrier

Availability
L: 3 ft and greater (sections if L>12ft)
W: 6-6.5, 11.5-12.5, 23-25, 35-38, 46-50
H: any length (72 inch practical limit)

5 ft QuickRating = P32-L36-M89

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	- 1000	7	9	11	12	21	23	19	16
	0	7	9	11	13	19	21	18	14
	1000	6	8	10	12	17	21	18	14
60	- 1000	9	12	17	26	37	34	23	16
	0	11	13	19	26	33	33	23	17
	1000	8	12	16	24	32	33	25	18
84	- 1000	11	15	23	35	50	47	34	26
	0	14	15	23	35	45	47	34	25
	1000	12	14	22	35	46	47	37	25
120	-1000	12	20	30	41	52	53	43	33
	0	12	18	30	42	48	52	42	30
	1000	12	17	29	43	47	53	44	31

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB) 63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1000	46	41	50	50	57	55	48	42
-500	44	33	41	45	51	52	39	23
+500	34	29	29	29	30	27	24	22
+1000	45	43	38	35	37	39	35	33

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 7.9 lb/ft^3

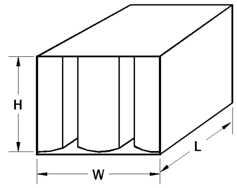
Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.) Face Velocity (fpm) 500	750	1000	1250	1500	2000
36	3.96	0.06	0.14	0.25	0.39	0.56	0.99
60	5.09	0.08	0.18	0.32	0.50	0.71	1.27
84	6.39	0.10	0.22	0.40	0.62	0.90	1.59
120	6.94	0.11	0.24	0.43	0.68	0.97	1.73

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled "QuickRating Guide" for further information.

McGill AirPressure Corporation



RSV-LV-L38

Rectangular, Straight, Low Velocity Silencer with **FDA Approved** Vapor Barrier

Availability
L: 3 ft and greater (sections if L>12ft)
W: 6-6.5, 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P32-L38-M68

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-1250	5	9	13	20	16	11	11	9
	0	4	8	14	23	14	13	11	11
	+1250	4	9	12	20	12	12	12	11
60	-1250	9	13	21	30	24	16	15	15
	0	7	12	21	35	24	18	17	20
	+1250	7	12	19	30	21	17	16	18
84	-1250	12	17	28	42	30	21	18	17
	0	11	17	29	47	30	26	19	23
	+1250	11	18	26	41	26	24	19	23
120	-1250	15	23	37	57	39	27	21	19
	0	15	22	38	63	38	35	22	28
	+1250	15	25	34	54	33	33	23	29

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1000	46	41	50	50	57	55	48	42
-500	44	33	41	45	51	52	39	23
+500	34	29	29	29	30	27	24	22
+1000	45	43	38	35	37	39	35	33

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Weight = 7.9 lb/ft^3

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	750	1000	1250	1500	2000
36	3.96	0.06	0.14	0.25	0.39	0.56	0.99
60	5.09	0.08	0.18	0.32	0.50	0.71	1.27
84	6.39	0.10	0.22	0.40	0.62	0.90	1.59
120	6.94	0.11	0.24	0.43	0.68	0.97	1.73

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled "QuickRating Guide" for further information.

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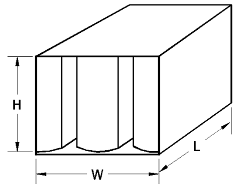

RSV-LV-L41

Rectangular, Straight Low Velocity Silencer with Vapor Barrier

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P19-L41-M81

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	- 1500	6	7	13	20	20	16	14	14
	0	4	7	14	20	20	17	14	14
	1500	4	8	14	19	18	15	14	14
60	- 1500	9	11	21	32	32	22	18	17
	0	8	11	22	32	32	21	17	16
	1500	7	12	22	29	29	21	18	17
84	- 1500	12	14	28	43	43	29	21	20
	0	11	14	30	44	44	27	20	19
	1500	9	15	30	40	40	26	22	20
120	-1500	16	19	36	57	57	37	27	23
	0	15	19	39	58	60	36	27	22
	1500	12	21	40	53	54	35	29	24

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
-1500	57	53	50	54	52	51	44	29
750	55	48	42	46	42	34	22	29
750	52	40	38	33	40	33	24	25
1500	57	51	46	44	49	48	44	29

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.9 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		250	500	750	1000	1500	1750
36	2.28	0.01	0.04	0.08	0.14	0.32	0.44
60	3.06	0.01	0.05	0.11	0.19	0.43	0.58
84	4.55	0.02	0.07	0.16	0.28	0.64	0.87
120	6.81	0.03	0.11	0.24	0.42	0.96	1.30

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled "QuickRating Guide" for further information.

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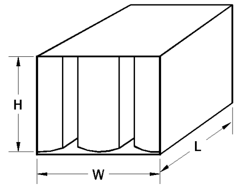

RSV-MV-L22

Rectangular, Straight, Medium Velocity Silencer with Vapor Barrier

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38

5 ft QuickRating = P08-L22-M56

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	-2000	1	7	5	9	15	17	13	11
	0	2	6	5	7	15	14	16	12
	+2000	3	4	6	7	13	13	16	10
60	-2000	5	11	9	15	22	25	19	16
	0	6	10	8	12	24	20	23	16
	+2000	3	8	9	14	22	20	24	15
84	-2000	6	14	14	23	32	37	27	26
	0	6	13	14	20	30	35	32	26
	+2000	3	11	15	22	32	28	31	19
120	-2000	8	14	17	34	44	46	27	26
	0	7	16	20	31	49	39	33	26
	+2000	4	13	22	33	48	39	37	24

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
- 2000	54	53	57	54	60	62	54	47
- 1000	37	41	45	39	43	42	38	27
+1000	39	36	35	37	33	36	30	32
+2000	53	52	50	48	46	55	48	45

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 6.2 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	2000	2500	3000
36	1.05	0.02	0.07	0.15	0.26	0.41	0.59
60	1.25	0.02	0.08	0.18	0.31	0.49	0.70
84	1.87	0.03	0.12	0.26	0.47	0.73	1.05
120	1.98	0.03	0.12	0.28	0.49	0.77	1.11

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

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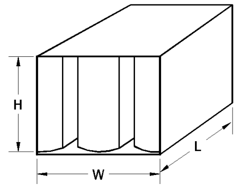


RSV-MV-L25

Rectangular, Straight, Medium Velocity

Availability
L: 3 ft and greater (sections if L>12ft)
W: 8.5-9.5, 17-19, 34-38
H: any length (72 inch practical limit)

5 ft QuickRating = P07-L25-M60

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	- 2000	8	4	7	12	16	16	11	9
	0	6	4	6	11	15	13	13	9
	2000	4	3	8	13	16	13	12	8
60	- 2000	6	6	12	20	22	19	11	9
	0	6	7	13	18	23	17	17	15
	2000	7	6	12	19	24	18	18	15
84	- 2000	12	8	16	29	30	22	11	9
	0	8	8	17	26	33	20	18	17
	2000	9	7	19	28	35	22	20	18
120	-2000	18	14	22	41	41	30	17	14
	0	14	12	23	34	41	26	24	23
	2000	13	10	25	39	43	27	28	23

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
- 2000	58	52	58	56	59	60	55	47
- 1000	39	38	47	43	47	44	36	27
+1000	32	32	33	33	34	38	27	25
+2000	61	49	46	45	45	55	50	44

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 6.4 lb/ft^3

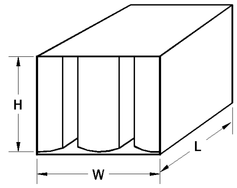
Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	2000	2500	3000
36	1.08	0.02	0.07	0.15	0.27	0.42	0.61
60	1.10	0.02	0.07	0.15	0.27	0.43	0.62
84	1.30	0.02	0.08	0.18	0.32	0.51	0.73
120	1.68	0.03	0.10	0.24	0.42	0.65	0.94

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled “QuickRating Guide” for further information.

McGill AirPressure Corporation



RSV-MV-L27

Rectangular, Straight Medium Velocity Silencer with Vapor Barrier

Availability
L: 3 ft and greater (sections if L>12ft)
W: 11.5-12.5, 23-25, 35-38, 46-50

5 ft QuickRating = P09-L27-M37

See bottom of page for explanation.

Table 1: Insertion Loss

Length (inches)	Face Velocity (fpm)	Insertion Loss (dB)							
		63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
36	- 2000	5	7	8	8	9	10	9	8
	0	5	7	8	8	9	9	9	8
	2000	4	6	7	7	7	8	8	7
60	- 2000	7	11	12	13	14	13	12	11
	0	7	10	11	12	14	13	12	11
	2000	6	9	10	11	12	11	10	9
84	- 2000	11	14	16	17	19	17	14	12
	0	11	14	15	16	19	17	14	13
	2000	9	12	13	15	17	14	12	11
120	-2000	15	19	22	23	24	21	16	14
	0	16	18	20	22	25	21	17	15
	2000	13	17	18	20	22	18	14	13

Note that ASTM inter-laboratory testing has shown insertion loss may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be greater than shown due to limitations in laboratory equipment and/or facilities.

Table 2: Airflow Generated Sound Power Level

Face Velocity (fpm)	Airflow Generated Sound Power Level (dB)							
	63Hz	125Hz	250Hz	500Hz	1000Hz	2000Hz	4000Hz	8000Hz
- 2000	51	49	50	51	53	54	46	37
-1000	37	41	45	39	43	42	38	27
1000	39	36	35	37	33	36	30	32
2000	53	48	47	45	47	51	44	35

Note that ASTM inter-laboratory testing has shown that generated noise may vary as much as 6 dB in the 63hz band, and 3 dB for all other frequencies. Data in parenthesis () may be less than shown due to limitations in laboratory equipment and/or facilities.

Table 3: Face Area Adjustment Factor

Silencer cross-sectional area (square feet)							
1	2	4	8	16	32	64	128
-6	-3	0	+3	+6	+9	+12	+15

Look up silencer cross-sectional area in table. Add adjustment to each octave band airflow generated sound power level from Table 2.

Weight = 5.8 lb/ft^3

Table 4: Pressure Loss

Length (inches)	Loss Coefficient	Dynamic Pressure Loss (in.wg.)					
		Face Velocity (fpm)					
		500	1000	1500	2000	2500	3000
36	0.92	0.01	0.06	0.13	0.23	0.36	0.52
60	1.43	0.02	0.09	0.20	0.36	0.56	0.80
84	2.17	0.03	0.14	0.30	0.54	0.85	1.22
120	3.28	0.05	0.20	0.46	0.82	1.28	1.84

Note: Shaded regions represent a design condition that may have negative consequences for acoustically sensitive applications.

The QuickRating is a designation used for comparing different silencer models to note differences in energy consumption (pressure loss), low frequency performance, and mid-frequency performance . The P rating is the pressure drop at 1000 fpm where PXX is the pressure drop in hundredths of an in.wg. The LYY rating is the total insertion loss, YY dB, of the 63, 125 and 250 Hz octave bands at 0 fpm. The MZZ rating is the total insertion loss, ZZ dB, of the 500, 1000 and 2000 Hz octave bands at 0 fpm. See the sheet titled "QuickRating Guide" for further information.

A.6 Duct Installation

A.6.3 UNI-COAT® Duct and Fittings

A.7 Specifying Duct Systems

A.7.1 Recommended Specifications for Commercial and Industrial Duct Systems

A.7 Specifying Duct Systems

A.7.2 Construction Specification Institute (CSI), Alexandria, Virginia

http://www.thebluebook.com/nj/htm/0015363500002.shtml

A.7.3 Production Systems for Architects and Engineers (American Institute of Architects), Washington, DC

http://www.aia.org/

A.7.4 Master Specification System for Design Professionals and the Building/Construction Industry (Masterspec)

http://www.arcomnet.com/visitor/masterspec/ms.html

A.8 USING COMPUTERS FOR DUCT DESIGN

A.8.1 UNI-DUCT® Design Program

UNI-DUCT® Duct System Design Software

McGill AirFlow Corporation can save you money by working with you to design efficient, economical duct systems. We can use our state-of-the-art UNI-DUCT computer program to perform time-consuming design work for you. It dramatically reduces your engineering time, while designing duct systems that have the lowest possible material costs and operating costs. The UNI-DUCT program gives you the most efficiently balanced and sized duct system designs available.

UNI-DUCT® Program Features

Key		
Key	S	**Supply Program Only**
	E	**Exhaust Program Only**

S	Optimized static regain design method that uses all available total pressure to balance the system and reduce duct cost (see McGill AirFlow's Engineering Report No. 144)
	Unparalleled database that includes many types of fittings with performance data derived from actual laboratory testing
	User-selected maximum and minimum velocities
	Round or flat oval presizing of any (or all) sections
	Half-inch duct sizes through 15 inches
	Round and/or flat oval designs
	Automatically designs with flat oval whenever user-selected height restriction is exceeded
S	Fitting default selection (automatically uses less expensive fittings for better balancing where appropriate in nondesign legs)
	Fitting selection override at any location in the system

E	Component pressure losses entered as either in-line losses or loss coefficients	
S	Allows entry of outlet static pressure requirements	
E	Entry losses entered as either total pressure losses or loss coefficients	
S	User-selected diversity factors for any nonoutlet section	
S	System can be designed to operate at a selected target pressure	
E	Maintains minimum design carrying velocities	
E	Corrects for nonstandard elevation and temperature	
S	An acoustical program that does the following:	
	a.	Accepts user-defined fan sound power levels or estimates the fan sound power levels by established practices
	b.	Accepts entry of additional attenuation and regenerated noise at any point in the system
	c.	Calculates all forms of attenuation and noise regeneration due to system components and airflow conditions
	d.	Calculates insertion loss of 1-, 2-, and 3-inch-thick lined duct and elbows
	e.	Determines path attenuation requirements for each terminal section based on user-selected NC levels
S	Products match SMACNA high- and/or low-pressure gauge requirements	
E	Entry of duct material handling class	
E	Automatic selection of duct gauge and reinforcement based on material handling class and duct pressure	
	Input data listings	
Sectional analysis report provides the following data for all sections:		
	a.	Duct size
	b.	Volume flow
	c.	Velocity
S	d.	Static or total pressure drop (or regain) through the takeoff or branch fitting
	e.	Cumulative pressure drop through all duct fittings

	f.	Duct pressure drop
	g.	Component pressure drops
	h.	Cumulative static or total pressure drop (or regain) for the section
	i.	Outlet or inlet static pressure drop
	j.	System static or total pressure at the entrance to each section

	System analysis report
	Acoustical analysis report
	Report format for either static or total pressure (user selected)
	Separate report of excess pressure at all terminal or entry sections (provides an indication of balancing and locates the system design leg)
	Average and highest excess pressure in system shown on balancing report
	Complete bill of material
	Complete project information
	Report includes suggested specification
	Unprecedented design guarantee

A.8 USING COMPUTERS FOR DUCT DESIGN

A.8.2 ASHRAE Duct Fitting Database Program

Available at ASHRAE's online bookstore at http://www.ashrae.org/

Duct Fitting Database

This menu-driven database includes loss coefficient tables for 228 round and rectangular duct fittings. A given fitting may be accessed to obtain loss coefficient data and associated pressure loss when appropriate flow rate and fitting information are entered. Table data included for supply, exhaust and common (supply/return) duct functions. Pictorial outlines of each fitting are included in documentation and may be displayed on screen. Input/output screen and fitting tables can be printed. 3.5 inch disk. (ISBN 883413-57-5)
Pages/Length: 88 Pubyear: 1994

Publisher: ASHRAE

Member Price - $79.00

Non-Member Price - $99.00

Units: Dual
Code: 94601

A.8 USING COMPUTERS FOR DUCT DESIGN

A.8.3 ASHRAE Algorithms for HVAC Acoustics

Algorithms for HVAC Acoustics, Douglas D. Reynolds and Jeffrey M. Bledsoe, American Society of Heating, Refrigerating and Air-Conditoning Engineers, Inc. (ASHRAE), Copyright 1991.

Available at ASHRAE's online bookstore at http://www.ashrae.org/

A.9 REFERENCES

A.9.1 HVAC Systems Duct Design, Sheet Metal and Air Conditioning Contractors National Association (SMACNA), 1990 Chantilly, Virginia. http://www.smacna.org/

A.9.2 ASHRAE Handbooks, American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), Atlanta, Georgia. http://www.ashrae.org/

A.9.3 Air Diffusion Council Publications (ADC), Chicago, Illinois. http://www.flexibleduct.org/

A.9.4 American Society for Testing and Materials (ASTM), Philadelphia, Pennsylvania. http://www.astm.org/

A.9.5 American Conference of Governmental Industrial Hygienists (ACGIH), Cincinnati, Ohio. http://www.acgih.org/

A.9.6 Air Movement and Control Association (AMCA), Arlington Heights, Illinois. http://www.amca.org/

A.9.7 T-Method by Robert Tsal (see reference ASHRAE Fundamentals Handbook).

A.9.8 ER 151: Flat Oval--The Alternative to Rectangular (see Part IV EDRM-Supplementary Topics by United McGill Corporation).

A.9.9 ER 102: Effect of Spacing Tees (see above noted reference).

A.9.10 Noise Control for Buildings and Manufacturing Plants, Bolt, Beranek and Newman, Inc., 1989.