# HVAC SYSTEMS
# DUCT DESIGN

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**SHEET METAL AND AIR CONDITIONING CONTRACTORS'
NATIONAL ASSOCIATION, INC.
www.smacna.org**

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FOURTH EDITION – MAY 2006

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**4201 Lafayette Center Drive**
**Chantilly, VA 20151–1209**
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# HVAC SYSTEMS DUCT DESIGN

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**SHEET METAL AND AIR CONDITIONING CONTRACTORS' NATIONAL ASSOCIATION, INC.**

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# FOREWORD

In keeping with its policy of disseminating information and providing standards of design and construction, the Sheet Metal and Air Conditioning Contractors' National Association, Inc. (SMACNA), offers this comprehensive fundamental *HVAC Systems-Duct Design* manual as part of our continuing effort to upgrade the quality of work produced by the heating, ventilating and air conditioning (HVAC) industry. This manual presents the basic methods and procedures required to design HVAC air distribution systems. It does not deal with the calculation of air conditioning loads or room air ventilation quantities.

This manual is part one of a three set HVAC Systems Library. The second manual is the SMACNA *HVAC Systems-Applications* manual that contains information and data needed by designers and installers of more specialized air and hydronic HVAC systems. The third manual is the *HVAC Systems-Testing, Adjusting and Balancing* manual, a recently-updated publication on air and hydronic system testing and balancing.

The HVAC duct system designer is faced with many considerations after the load calculations are completed and the type of distribution system is determined. This manual provides not only the basic engineering guidelines for the sizing of HVAC ductwork systems, but also related information in the areas of:

a. Materials
b. Methods of Construction
c. Economics of Duct Systems
d. Duct System Layout
e. Pressure Losses
f. Fan Selection
g. Duct Leakage
h. Acoustic Considerations
i. Duct Heat Transfer
j. Testing, Adjusting and Balancing

With emphasis on energy conservation, the designer must balance duct size with the space allocated and duct system pressure loss. Duct pressure loss increases fan power and associated operating costs. Materials, equipment, and construction methods must be carefully chosen to achieve the most advantageous balance between both initial and life cycle costing considerations. This manual is designed to offer both the HVAC system designer and installer detailed information on duct design, materials and construction methods. Both U. S. and metric units have been provided in examples, calculations, and tables.

The SMACNA *HVAC Systems-Duct Design* manual is intended to be used in conjunction with the American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE) *Fundamentals* Handbook. The basic fluid flow equations are not included here, but may be found in the ASHRAE handbook. Practical applications of these equations are included through use of reference tables and examples. Some sections of this manual have been reprinted with permission from various ASHRAE publications. Another important source of HVAC systems information is the Air Movement and Control Association International, Inc. (AMCA). SMACNA and the entire HVAC industry owe these two organizations a debt of gratitude for continued investments testing and development of HVAC standards.

Although most HVAC systems are constructed to pressure classifications between minus 3 inches water gage (wg) to 10 inches wg, (-750 to 2,500 Pascals (Pa), the design methods, tables, charts, and equations provided in this text may be used to design other duct systems operating at higher pressures and temperatures. Air density correction factors for both higher altitudes and temperatures are included.

SMACNA recognizes that this manual will be expanded and updated as new material becomes available. We will continue to provide the HVAC industry with the latest construction methods and engineering data from recognized sources including SMACNA research programs and the services of local SMACNA Chapters and SMACNA Contractors.

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CHAPTER 1

# INTRODUCTION

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## 1.1 SCOPE

This manual provides the duct system designer with the technical information required to design a complete air distribution system. This text has been extensively revised and updated, and now includes key portions of the previously separate Duct Design Home Course Study material.

## 1.2 HOW TO USE THIS MANUAL

This manual is divided into chapters that address each step of duct system design, layout, and sizing.

**CHAPTER 1 INTRODUCTION**
Brief introduction to the history, related building codes, smoke control, and system types for duct installations in commercial facilities.

**CHAPTER 2 ECONOMICS OF DUCT SYSTEMS**
How duct sizing and system layout impacts the economics for a project.

**CHAPTER 3 ROOM AIR DISTRIBUTION**
All of the design considerations related to room air distribution and indoor air quality.

**CHAPTER 4 GENERAL APPROACH TO DUCT DESIGN**
Basics of duct system design, including pressure losses, duct and diffuser noise, and basic system balancing issues.

**CHAPTER 5 DUCT DESIGN FUNDAMENTALS**
Fundamental elements of fan curves, pressure loss calculations, duct leakage, and duct heat gains and losses.

**CHAPTER 6: FAN–DUCT CONNECTION PRESSURE LOSSES**
Issues related to the transition from supply and return fans to the ductwork.

**CHAPTER 7 DUCT SIZING PROCEDURES**
Pressure loss design information for duct components including fittings, diffusers, registers, and duct transitions.

**CHAPTER 8 PRESSURE LOSS OF SYSTEM COMPONENTS**
Tables and graphs to estimate pressure drop for each component in a duct system.

**CHAPTER 9 PROVISIONS FOR TESTING, ADJUSTING, AND BALANCING**
How a new air distribution system should be balanced. Common duct system testing and balancing procedures.

**CHAPTER 10 DESIGNING FOR SOUND AND VIBRATION**
Noise generation and methods to reduce system noise.

**CHAPTER 11 DUCT SYSTEM CONSTRUCTION**
Construction and how duct material selection can improve indoor air quality.

**CHAPTER 12 SPECIAL DUCT SYSTEMS**
Kitchen and dishwasher exhaust ducts, and duct systems for corrosive and noxious gases.

**APPENDIX**
Tables and charts of duct design information in an easy to use format.

## 1.3 PURPOSE

The purpose of any heating, ventilating, and air conditioning (HVAC) duct system is to provide:

a. Thermal comfort

b. Humidity control

c. Ventilation air

d. Air filtration

However, a poorly designed or constructed HVAC duct system may result in a system that is costly to operate, noisy, and does not meet occupant comfort requirements.

This manual and associated SMACNA publications will assist both the system designer and the installer to provide an HVAC system that meets all these basic requirements.

## 1.4 HISTORY OF AIR DUCT SYSTEMS

Over 2,000 years ago, both the Greeks and Romans used masonry and terra cotta pipe to distribute flue gases from a central heating source to indirectly heat interior rooms and baths. The use of flues and ducts eventually disappeared until the twelfth century when heating fireplaces were moved from the center of a great-room to a sidewall, and chimneys or flues were used again.

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In 1550 a German named Georgius Agricola, completed a book describing his many inventions, methods, and procedures to ventilate deep mines. Using ducts and ventilating fans made from wood, these primitive systems were powered by windmills, humans, horses, and finally by running water.

In the early 1800s, most air ducts used for heating and ventilation were masonry and supply fans were driven by steam engines. However, many buildings were still ventilated by stack effect which did not require powered fans. The galvanized coating of steel using zinc did not occur until the 1890s and tin or zinc-coated sheet metal was not commonly used until around World War I. Some of the larger blower and centrifugal fan manufacturers went into business during and after the 1860s.

At the first meeting of American Society of Heating, Refrigerating and Air-Conditioning Engineers (then ASH&VE) in 1894, a discussion of metal versus wood air conduits was ended after it was reported that galvanized iron ducts generally had replaced wooden ones. It was not until 1922 that ASH&VE published the first "Guide." This was a handbook on current engineering practice in heating and ventilating and included tables and charts that had evolved during the years. The Guide included a chart on "Synthetic Air" that offered a "*means of determining the percentage of perfect ventilation by considering all known factors that make up the air conditions in a room.*" This was the first official chart published that addressed indoor air quality.

In the late 1800s, mechanical ventilation systems provided 100 percent outside air for ventilation, re-circulation of air was considered unhealthy. Centrifugal fans were used for the HVAC supply air duct while the natural stack effect of chimney ducts was used for relief or exhaust air. In 1908 the HVAC industry advised that a minimum of 30 cfm (15 L/s) of outside air per person should be used, with up to 60 cfm (30 L/s) recommended for hospitals and places of assembly. Many states adopted these early ventilation air guidelines.

In 1936 ASHRAE research suggested 7 cfm (3.5 L/s) of outside ventilation air per person could be used when the space was over 500 $ft^3$ (50 $m^3$), but recommended 25 cfm (12.5 L/s) when the space was reduced to 100 $ft^3$ (10 $m^3$). In the 1970s outside ventilation air was reduced further to 5 cfm (2.5 L/s) to reduce building energy usage during a period of very high oil and gas prices. In the early 1990s, most code agencies adopted ASHRAE Standard 62-1989 that increased the minimum ventilation rates to between 15 cfm (7.5 L/s) and 60 cfm (30 L/s) per person after problems developed with "sick building syndrome." This was primarily caused by inadequate outside ventilation in the heavily-insulated and tightly-sealed building construction that had become standard. Hospital operating areas were still designed to provide 100 percent outside ventilation air.

## 1.5 GENERAL REQUIREMENTS

An HVAC duct system is a structural assembly designed to convey air between specific points. To provide this function, the duct assembly must meet certain fundamental performance characteristics. Elements of the duct system include an envelope of sheet metal or other material, reinforcements, seams, and joints. Practical performance requirements and construction standards must be established for:

a. Dimensional stability – deformation and deflection.

b. Containment of the air being conveyed.

c. Vibration.

d. Noise generation, transmission, and attenuation.

e. Exposure to damage, weather, temperature extremes, flexure cycling, chemical corrosion, and other in-service conditions.

f. Structural support.

g. Emergency conditions including fire and seismic occurrence.

h. Heat gain and loss of the air stream.

i. Dirt and contaminants collecting on duct interior walls and duct liners.

In establishing limitations for these factors, consideration must be given to the effects of pressure differential across the duct wall, airflow friction losses, dynamic losses, air velocities, air leakage, and the inherent strength of the duct components. A design and construction criterion which meets both an economic budget and desired performance must be determined.

## 1.6 HVAC SYSTEMS LIBRARY

In addition to this *HVAC Systems Duct Design* manual, there are many other SMACNA publications available that relate to the design and installation of HVAC sys-

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tems. A partial listing of the more relevant publications with a brief description follows. These texts and guides may be ordered from SMACNA using our web site http://smacna.org.

### Related SMACNA Publications:

**HVAC Air Duct Leakage Test Manual**
A companion to *HVAC Duct Construction Standards – Metal and Flexible*. Duct leakage test procedures, recommendations on use of leakage testing, types of test apparatus, and test setup.

**HVAC Duct Construction Standards - Metal and Flexible**
The *HVAC Duct Construction Standards – Metal and Flexible* is primarily for commercial and institutional projects. Rectangular, round, oval and flexible duct constructions for positive or negative pressures up to 10 in. water gage (2500 Pa).

**HVAC Systems – Testing, Adjusting and Balancing (TAB)**
Standard procedures, methods, and equipment required to properly balance both air and water systems.

**Indoor Air Quality Manual**
This manual identifies indoor air quality (IAQ) problems as currently defined. It also contains methods and procedures used to solve IAQ problems and the equipment and instrumentation necessary.

**Fire, Smoke, and Radiation Damper Guide for HVAC Systems**
Installation guidelines for all types of fire and smoke dampers and smoke detectors.

**TAB Procedural Guide**
The *TAB Procedural Guide* is intended for trained TAB technicians to assure that the appropriate procedures are employed in an effective manner. HVAC system air and water side adjusting and balancing. Variable air volume, multi-zone, dual duct, and exhaust air systems are examples of systems specifically covered.

**Fibrous Glass Duct Construction Standards**
Performance characteristics for fibrous glass board as well as specifications for closures and illustrations of how to construct the full range of fittings. Details for connections to equipment and air terminals, hanger schedules, reinforcement requirements, fabrication of rectangular duct and fittings, closures of seams and joints, channel and tie rod reinforcements, plus hangers and supports.

**HVAC Systems – Commissioning Manual**
Practical how-to commissioning guide for contractors, owners, and engineers for new buildings, and re-commissioning for existing buildings. Separate chapters are devoted to the different levels of commissioning, including basic, comprehensive, and critical systems commissioning. The appendix contains a sample HVAC Systems Commissioning Specification.

**Guidelines for Roof Mounted Outdoor Air-Conditioner Installations**
Guidelines for installation of roof-mounted outdoor air-conditioner equipment. Supplement to the unit manufacturer's specific installation instructions. Waterproofing illustrations and reminders covers curb and roof penetrations and sealings, as well as the interface between the roof and the location at which the unit, piping, electrical wiring, or sheet metal ductwork pass through the roof.

**HVAC Seismic Manual**
Guidelines for HVAC system installations in areas subject to seismic activities.

**HVAC Sound and Vibration Manual**
Components and installation methods to reduce sounds and vibrations in HVAC systems.

## 1.7 CODES AND ORDINANCES

### 1.7.1 HVAC System Codes

In the private sector, each new construction or renovation project normally is governed by state laws or local ordinances that require compliance with specific health, safety, and property protection guidelines. In addition, recent federal legislation is requiring more energy efficient building and system designs to reduce our nation's dependence on foreign energy supplies and to protect the environment.

Figure 1-1 illustrates the relationship between laws, ordinances, codes, and standards that can affect the design, and construction of HVAC duct systems. However, this may not include all applicable regulations and standards for a specific locality. Specifications for federal government construction are promulgated by the Federal Construction Council, the General Services Administration, the Department of the Navy, the Veterans Administration, and other agencies.

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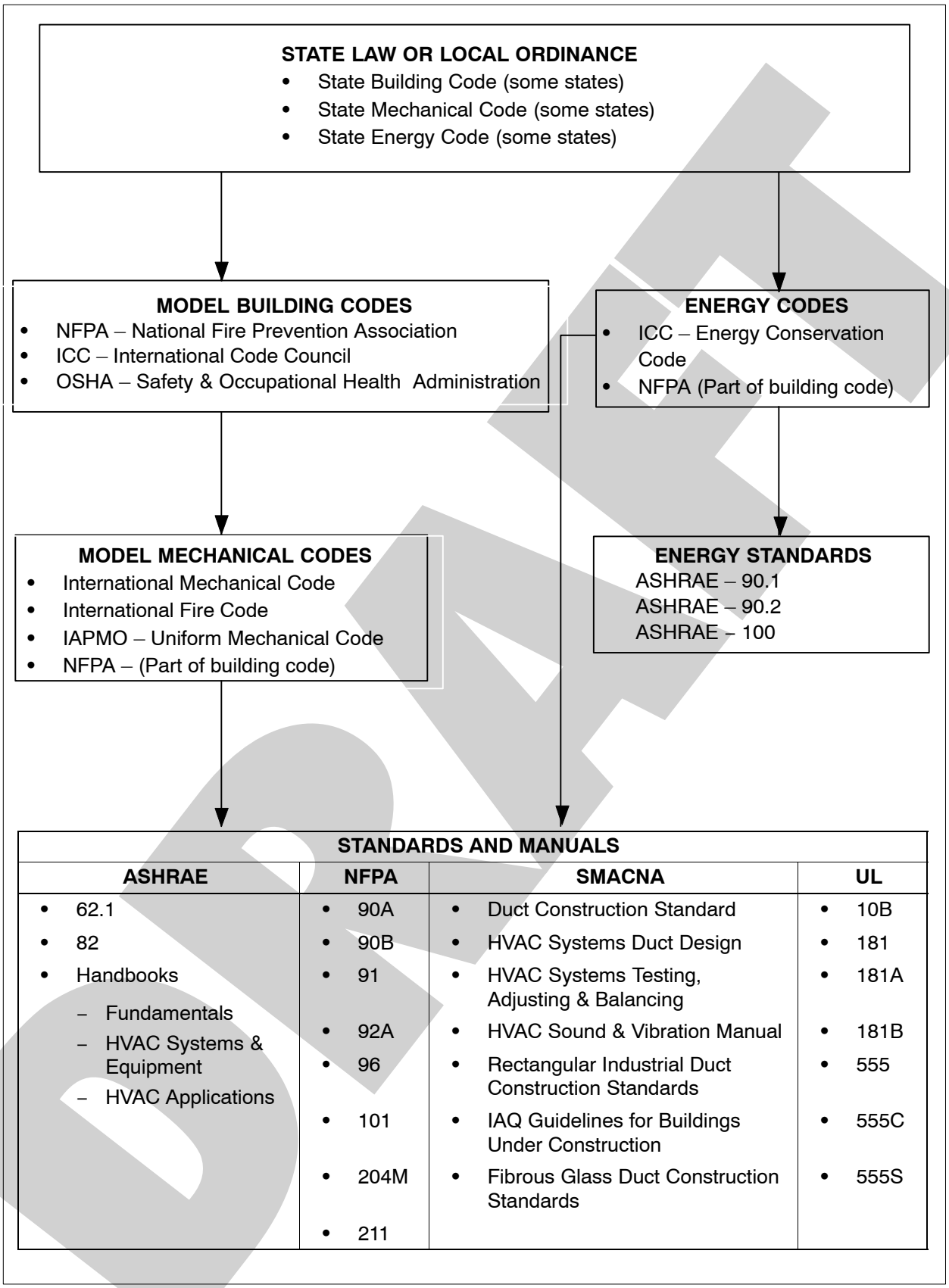


| STANDARDS AND MANUALS | | | |
|---|---|---|---|
| **ASHRAE** | **NFPA** | **SMACNA** | **UL** |
| • 62.1 | • 90A | • Duct Construction Standard | • 10B |
| • 82 | • 90B | • HVAC Systems Duct Design | • 181 |
| • Handbooks | • 91 | • HVAC Systems Testing, Adjusting & Balancing | • 181A |
| – Fundamentals | | | |
| – HVAC Systems & Equipment | • 92A | • HVAC Sound & Vibration Manual | • 181B |
| – HVAC Applications | • 96 | • Rectangular Industrial Duct Construction Standards | • 555 |
| | • 101 | • IAQ Guidelines for Buildings Under Construction | • 555C |
| | • 204M | • Fibrous Glass Duct Construction Standards | • 555S |
| | • 211 | | |

**FIGURE 1–1 U.S.A. BUILDING CODES AND ORDINANCES**

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Model code changes require long cycles for approval by both industry consensus and legislation. Since the development of safety codes, energy codes, and installation standards proceed independently, the most recent edition of a code or standard may not have been adopted by a local jurisdiction. HVAC designers must know which code version governs their design. If a provision is in conflict with the design intent, the designer should resolve the issue with the local building official. New or different construction methods can be accommodated by the provisions for equivalency that are incorporated into these codes.

### 1.7.2 Fire and Smoke Codes

Fire and smoke control designs are covered in Chapter 52 of the 2003 *ASHRAE Handbook - HVAC Applications*. The designer should consider flame spread, smoke development, and toxic gas production from duct and duct insulation materials. Code documents for ducts in certain locations within buildings rely on a material requirement that is the generally accepted criteria of 25 flame spread and 50 smoke development. However, certain duct construction protected by extinguishing systems may be accepted with higher levels of combustibility by code officials. Combustibility and toxicity ratings are normally based on tests of materials.

National, state, and local codes usually require fire or smoke dampers where ducts penetrate fire-rated walls, floors, ceilings, and partitions or smoke barriers. **Any required fire, radiation, or smoke damper must be clearly identified on the plans by the duct designer and accessible for servicing.** Before specifying dampers for installation in any vertical shaft or as part of a smoke evacuation system, consult with local authorities having jurisdiction. Also review NFPA 92A *Recommended Practice for Smoke Control Systems*.

The codes, however, do not prescribe or dictate how these penetrations are to be protected. The responsibility for the specification of the details (materials and methods) is delegated to the design professionals.

One or more of the following national codes usually apply to most duct system installations:

a. International Building Code and International Mechanical Code by International Code Council (ICC). Both codes are now used in most states as replacement for BOCA, ICBO, and SBCCI regional building codes.

b. Uniform Mechanical Code (UMC) by the International Association of Plumbing and Mechanical Officials (IAPMO)

c. California State Mechanical Code (CMC)

d. Federal Energy Code for energy efficiency in new commercial and multi-family high-rise facilities.

e. National Fire Protection Association (NFPA).

*NOTE: Federal, state, and local codes or ordinances may modify or replace these listed codes.*

## 1.8 HVAC DUCT SYSTEM TYPES

### 1.8.1 Commercial Duct Systems

ASHRAE divides the application of commercial HVAC systems into single-path systems and dual-path systems, and single-zone and multiple-zone systems. In addition, these systems can be designed to provide a constant airflow or a variable airflow to each zone. An HVAC duct system provides complete sensible and latent cooling capacity in the conditioned air supplied by the system.

The term *zone* implies the provision or the need for separate thermostatic control, while the term *room* implies a partitioned area that may or may not require a separate control.

Single-path systems contain the main heating and cooling coils in a series-flow air path. A common duct distribution system provides a common air temperature to all terminal devices

**Single-Path Systems:**

Single duct, constant volume
Reheat systems, single duct
Variable air volume (VAV)
Variable air volume, reheat
Single duct, VAV induction
Single duct, VAV fan powered
Constant fan, intermittent fan

Dual-path systems contain the main heating and cooling coils in parallel-flow air paths with a separate cold and warm air distribution duct. A dual-path system may also use a separate supply duct to each zone, with the amount of air supplied to that zone controlled at the main supply fan.

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**Dual-Path Systems:**

Dual duct, constant volume
Multi-zone

In recent years, most commercial HVAC supply air ductwork has been designed for a maximum pressure of 6 in. wg (1500 Pa) to save energy and operating costs. During the 1960s, systems designed at 10 in. wg (2500 Pa) were common with fan horsepower of 100 HP (75 kW) or greater. When fan laws are studied in later chapters you will find that a 10 percent increase in system pressure or airflow will increase fan power by 33 percent. The reverse is also true, which is why variable air systems are popular in areas where utility costs are high.

Since zones do not usually have design peak cooling or heating demands during most of the year, even a very small reduction in airflow during non-peak load periods can provide significant energy and utility savings.

A new application of commercial "duct" systems is the under-floor air distribution (UFAD) system. Originally used with constant-volume packaged units serving raised-floor computer rooms, these systems have been adopted for general space conditioning to allow more flexibility for space alterations in large, open space tenant offices and to provide a more accessible path for wiring and cabling changes. When planning any under-floor air distribution system, special care is required to address air leakage—both from the floor tiles and into vertical spaces and wall partitions, seismic bracing issues and to comply with code-mandated fire and smoke separations. Air leakage from the UFAD system into exterior spaces presents an increased potential for mold problems, especially during cooling. Excessive air leakage from the plenum also makes achieving satisfactory air balance problematic. Designers are cautioned that the space below the raised floor is classified as a supply plenum and should be treated the same as a duct for code purposes. The two most commonly overlooked code compliance issues are the practice of extending gypsum board into the supply plenum and neglecting to use code-compliant wire.

### 1.8.2 Industrial Duct Systems

Duct systems in offices and warehouses, when used for comfort applications, fall into the category of commercial HVAC duct design. In research- and laboratory-type facilities, unusual airflow velocities or temperature conditions may be encountered. If additional information on the design of these special systems is required, SMACNA has other industrial duct system design manuals available.

## 1.9 SMOKE CONTROL SYSTEMS

Smoke is recognized as the major killer in fire situations. Smoke often migrates to other building areas that are remote from where the fire is actually located. Stairwells and elevator shafts can become smoke filled, blocking occupant evacuation and inhibiting fire fighting. As a solution to this smoke problem, organizations in the United States and Canada have developed the concept of smoke control. Some of the information found in this section has been condensed from the ASHRAE publication *Design of Smoke Control Systems for Buildings.*

Smoke control systems may use fans and ducts to control airflow and pressure differences between zones to regulate smoke movement. The primary objective of a smoke control system is to reduce deaths and injuries from smoke.

A smoke control system is designed to produce a safe escape route, a safe refuge area, or both. It is obvious that a smoke control system can meet these objectives even if a small amount of smoke infiltrates protected areas. However, for most areas, smoke control systems are designed on the basis that no smoke infiltrations occur.

There are situations where energy conservation methods can potentially defeat a smoke control system. The smoke control system must be designed to override the local temperature controls for a variable air volume HVAC system so that the air supply required to pressurize non-fire spaces is available.

Automatic activation of a smoke control system should be considered with the primary activation from a smoke detector located in the building space. A smoke control system should be equipped with a control center with easy access for the fire department where the smoke control system can be manually overridden.

### 1.9.1 Smoke Movement

The major driving forces causing smoke movement in a building are:

- Stack effect
- Buoyancy
- Expansion
- Wind
- HVAC systems

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Generally in a fire situation, smoke movement will be caused by a combination of these driving forces.

### 1.9.2 Stack Effect

When it is cold outside, there often is an upward movement of air within building shafts, such as stairwells, elevator shafts, and mechanical shafts. This phenomenon is referred to as *"stack effect."* The air in the building has a buoyant force when it is warmer and less dense than the outside air. This buoyant force causes air to rise within vertical shafts of buildings. The significance of normal stack effect is greater when outside temperatures are low and building shafts are tall. However, stack effect can even exist in a one-story building.

When the outside air is warmer than the building interior, a downward airflow frequently exists in these vertical shafts. The downward airflow is called *reverse stack effect*.

### 1.9.3 Buoyancy

High-temperature smoke from a fire has a buoyancy force due to its reduced density. In a building with leakage paths in the ceiling of the room containing the fire, this buoyancy-induced pressure causes smoke movement to the floor above the fire floor. In addition, this pressure causes smoke to move through any leakage paths in the walls or around the doors of the fire compartment. As smoke travels away from the fire, its temperature drops due to heat transfer and dilution. The effect of buoyancy generally decreases with distance from the fire.

### 1.9.4 Expansion

In addition to buoyancy, the energy released by a fire can cause smoke movement due to air expansion. In a fire compartment with only one access or window opening, building air will flow into the fire compartment and hot smoke will flow out of the fire compartment.

### 1.9.5 Wind

Wind can have a pronounced effect on smoke movement within a building. Frequently in fire situations, a window breaks in the fire compartment. If the window is on the leeward side of the building, the negative pressure caused by the wind vents the smoke from the fire compartment. This can greatly reduce smoke movement throughout the building. However, if the broken window is on the windward side, the wind forces the smoke throughout the fire floor and even to other floors. This both endangers the lives of building occupants and hampers fire fighting. Pressures induced by the wind can be relatively large and can easily dominate air movement throughout the building.

### 1.9.6 HVAC Systems

Before the development of smoke control systems, HVAC duct systems and supply fans were shut down when fires were discovered.

In the early stages of a fire, HVAC systems can serve as an aid to fire detection. When a fire starts in an unoccupied portion of a building, HVAC systems can transport the smoke to a space where people can smell the smoke and be alerted to the fire. However, as the fire progresses, HVAC systems will transport smoke to every area that the system serves, thus endangering life in all spaces. HVAC systems also supply air to the fire space, which aids combustion.

These are the reasons early HVAC systems traditionally were shut down when fires were discovered. Although shutting down an HVAC system prevents it from supplying air to the fire, this does not prevent smoke movement through the supply and return air ducts, airshafts, and other building openings due to stack effect, buoyancy, or wind pressure.

### 1.9.7 Smoke System Design

Building codes contain design parameters for the design of safe and economical smoke control systems. The designer has an obligation to adhere to any smoke control design criteria specified in the appropriate codes and standards. However, such criteria should be evaluated to determine if their use would result in an ineffective system. If necessary, the designer may need to seek a waiver from local smoke control codes.

The five design parameters that must be established are: (1) leakage areas, (2) weather data, (3) pressure differences, (4) airflow, and (5) number of open doors in the smoke control system.

Current state-of-the-art smoke design gives little consideration to local wind patterns and weather data. Further information on smoke control duct systems may be found in the SMACNA *HVAC System Applications* manual and ASHRAE *Design of Smoke Control Systems for Buildings*.

## 1.10 INDOOR AIR QUALITY

The definition of good indoor air quality is complex. In addition to thermal comfort and adequate outside

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ventilation air, the condition of light, sound, and subjective irritants may become part of any indoor air quality evaluation. With today's much tighter building construction, higher concentrations of irritants are possible from interior finishes including paints, glues, solvents, and carpeting.

As stated earlier, using outside air in varying amounts has been a concern to HVAC system designers for over 100 years. Regardless of the percentage or amounts of outside air introduced to the occupied spaces, or the filtration methods used, the HVAC duct system designer must be sure the required amount of ventilation air reaches every occupied space. This can become an even greater problem when designing variable air volume (VAV) systems.

### 1.10.1 Calculating Outside Air Percentage

Determining the outside air percentage for an HVAC duct system that is required to supply 100 percent outside air is easy since the total system airflow and the ventilation airflow are the same. Both local and national building codes define ventilation air requirements based on occupancy and room type.

For all other systems, the percentage of outside air can be calculated as follows: Assume that the area served by the HVAC system normally will have the maximum design number of people occupying the space. Multiply the ventilation rate times the number of people. Divide this amount by the HVAC system airflow and multiply by 100 percent. This will provide the percentage outside air required.

*Example 1:*

A space heated and cooled by a 4000 cfm (2000 L/s) HVAC duct system is normally occupied by 30 people. The local ventilation code requires 20 cfm (10 L/s) of outside air per person. Calculate the percentage of outside air required.

30 people × 20 cfm (10 L/s) = 600 cfm (300 L/s)

$$\frac{600 \text{ cfm } (300 \text{ L/s})}{4000 \text{ cfm } (2000 \text{ L/s})} \times = 15 \text{ percent outside air}$$

### 1.10.2 Measuring Air Percentage

Depending on the location of the HVAC duct system components; there are two methods for calculating the percentage of outside air based on actual measured data:

- Direct measurement of airflows
- Direct measurement of temperatures

### 1.10.3 Airflow Measurement

Total system and outside airflows may be measured by making duct traverses using a Pitot tube and manometer. The total system airflow also may be found by totaling the air volumes from the terminal air outlets obtained by using a flow measuring hood or anemometer. Corrections must be made to account for HVAC system leakage.

### 1.10.4 Temperature Measurement

When air temperature measurements are carefully made at points that are representative of the respective airflows, Equation 1-1 may be used to calculate the percentage of outside ventilation air or the mixed air temperature that is needed.

**Equation 1–1**

$$T_m = \frac{X_0 T_0 + X_r T_r}{100}$$

Where:

$T_m$ = Temperature of mixed air degree F (degree C)

$X_0$ = Percentage of outside air

$T_0$ = Temperature of outside air degree F (degree C)

$X_r$ = Percentage of return air

$T_r$ = Temperature of return air degree F (degree C)

## 1.11 VENTILATION RATES

Before ducts, louvers, and fans related to the ventilation air portion of any HVAC system can be designed and installed, the required ventilation air must be determined. Unfortunately, ventilation air-flow rates are determined by sometimes conflicting building codes, not load calculations.

Since conditioning any outside ventilation air has a high energy and associated utility cost, buildings energy codes do not favor excess ventilation air. Since there are multiple building, health, and life safety codes that may apply depending on occupancy type, it is very important to identify which of these codes will take precedence for your specific application.

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CHAPTER 2

# ECONOMICS OF DUCT SYSTEMS

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## 2.1 SCOPE

It is natural to want the most for the least cost, but building owners need to realize the annual cost to maintain and operate their facility is the direct result of their first-cost construction decisions. With today's high-energy costs, the higher operating and maintenance costs for a low efficiency HVAC system can quickly exceed any initial first-cost construction savings.

It is important for the HVAC system designer to provide building owners with alternative system designs during initial budget development, to compare the operating and maintenance costs for alternative designs.

The 2003 *ASHRAE Handbook - HVAC Applications* has a more detailed analysis of HVAC system economics, which is summarized below:

## 2.2 RESPONSIBILITIES

Any duct system requires many different individuals working together to make a successful installation. Typically, commercial duct systems will have a separate designer and an installation contractor. The installation contractor may also use a separate duct fabricator, and the project specifications may require the services of a totally separate balancing contractor. Each of these individuals has specific functions and each expects certain functions to be performed by the others when submitting their bids.

The following outline provides a minimum level of responsibilities and functions that each is expected by the others to provide.

### 2.2.1 Duct System Designer Responsibilities

a. Match the fan to the system pressure losses.

b. Designate the pressure class for construction of each duct system and duct segment and clearly identify these in the contract document.

c. Evaluate the leakage potential for ducts conforming to SMACNA Standards and Guidelines and supplement with deletions and additions that may be prudent and economical for this specific project. Check the location of all ducts, type of service, connections to other equipment dampers and accessories in the system, tolerances on air balance, and the performance objectives. Account for leakage in equipment such as fans, coils, and volume regulating boxes, in addition to all duct leakage.

d. Specify the amount and manner of leakage testing, if testing is required, and clearly indicate the acceptance criteria.

e. Reconcile all significant inconsistencies between performance specifications and prescription specifications before releasing contract documents for construction.

f. Avoid ambiguity created by references to non-specific editions of SMACNA or other documents specified. At a minimum, the full title and edition of all referenced documents should be noted.

g. The duct designer needs to understand that a "single line" duct drawing does not provide sufficient information to describe the complexity of today's duct systems. Unusual fittings to avoid structural components or ducts located in highly congested ceiling spaces must be detailed on the drawings. Any duct fittings that are not clearly designated on the design drawings cannot be included in the sheet metal contractor's initial base bid.

h. Make the contract documents reflect a clear scope of work that is known to conform to applicable codes and regulations, including those addressing energy conservation.

i. Require adequate submittals and record keeping, while making sure all work in progress conforms to the contract documents in a timely manner.

### 2.2.2 Ductwork Installer Responsibilities

a. Comply with the contract documents.

b. Provide all required pre-construction and after-installation submittals.

c. Report discovery of conflicts and ambiguities in a timely manner.

d. Seal duct as specified.

e. Examine the leakage criteria, the specified duct construction classes, and the testing and balancing specifications for consistency.

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f. Select duct construction and sealing methods that are appropriate and compatible, giving due consideration to the size of the system.

g. Maintain quality control over all workmanship. Provide protection from the elements for all materials stored on the job site.

h. Schedule any required leakage or control system tests in a timely manner, with appropriate notice to project management.

## 2.3 INITIAL SYSTEM COSTS

The first consideration during any new HVAC duct system selection is the initial cost. A careful evaluation of all cost variables must be completed if maximum economy is to be achieved. The designer has a significant influence on these initial costs when specifying the duct system material, operating pressure, duct sizes and complexity, fan horsepower, sound attenuation, and space requirements for ductwork and equipment.

When evaluating equipment purchases or comparing alternate system layouts, it is helpful to take into account the "time value of money." For example, it is fairly easy to know first hand that a dollar will purchase more materials today than in ten years. We do not normally think of the installation of a HVAC system as an "investment." However, the system owner could have earned interest income if the purchase money had just been left in the bank. Also, if the system owner borrowed the money for the purchase, there is a yearly interest cost associated with the installed cost.

Life cycle costing is a method to view any equipment purchase as a fixed cost each year, during the life of the equipment. Since this costing method includes the effect of compound interest, the annual cost of the equipment over its lifetime can be converted into "today's" dollars.

Table 2-1 provides life cycle cost factors that will give a uniform annual cost for any equipment or system purchase including interest expense. To use this table, it is necessary to estimate the life of the equipment and the annual rate of interest (or inflation) that would apply.

The useful life for an HVAC duct system is normally considered to be the life of the building, which can minimize the annual effect of duct system first-cost, in comparison with those elements of an HVAC system having a shorter useful life, including fans, VAV boxes, and controls.

*Example 2-1*

Find the annual life cycle cost for a $50,000.00 HVAC system, with a 25-year life, using an estimated interest rate of 4 percent, compounded annually.

| | Compound Interest or Inflation Annual Rate | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| **Years** | **2%** | **3%** | **4%** | **5%** | **6%** | **8%** | **10%** | **12%** |
| **2** | 0.51505 | 0.52261 | 0.53019 | 0.53781 | 0.54544 | 0.56077 | 0.57619 | 0.59169 |
| **3** | 0.34675 | 0.35353 | 0.36035 | 0.36721 | 0.37411 | 0.38803 | 0.40212 | 0.41635 |
| **4** | 0.26262 | 0.26903 | 0.27549 | 0.28201 | 0.28859 | 0.30192 | 0.31547 | 0.32923 |
| **6** | 0.17853 | 0.18459 | 0.19076 | 0.19702 | 0.20336 | 0.21632 | 0.22961 | 0.24323 |
| **8** | 0.13651 | 0.14246 | 0.14853 | 0.15472 | 0.16104 | 0.17402 | 0.18744 | 0.20130 |
| **10** | 0.11133 | 0.11723 | 0.12329 | 0.12951 | 0.13587 | 0.14903 | 0.16275 | 0.17698 |
| **12** | 0.09456 | 0.10046 | 0.10655 | 0.11283 | 0.11928 | 0.13269 | 0.14676 | 0.16144 |
| **14** | 0.08260 | 0.08853 | 0.09467 | 0.10102 | 0.10759 | 0.12129 | 0.13575 | 0.15087 |
| **16** | 0.07365 | 0.07961 | 0.08558 | 0.09227 | 0.09895 | 0.11298 | 0.12782 | 0.14339 |
| **18** | 0.06670 | 0.07271 | 0.07899 | 0.08555 | 0.09236 | 0.10670 | 0.12193 | 0.13794 |
| **20** | 0.06116 | 0.06722 | 0.07358 | 0.08024 | 0.08719 | 0.10185 | 0.11746 | 0.13388 |
| **25** | 0.05122 | 0.05743 | 0.06401 | 0.07095 | 0.07823 | 0.09803 | 0.11017 | 0.12750 |
| **30** | 0.04465 | 0.05102 | 0.05783 | 0.06505 | 0.07265 | 0.08883 | 0.10608 | 0.12414 |
| **35** | 0.04000 | 0.04654 | 0.05358 | 0.06107 | 0.06897 | 0.08580 | 0.10369 | 0.12232 |
| **40** | 0.03656 | 0.04326 | 0.05052 | 0.05828 | 0.06646 | 0.08386 | 0.10226 | 0.12130 |
| **50** | 0.03182 | 0.03887 | 0.04655 | 0.05478 | 0.06344 | 0.08174 | 0.10086 | 0.12042 |

**Table 2–1 Annual Life Cycle Cost Factors**

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The life cycle cost factor from Table 2-1 for the present value of $50,000.00, spread out over 25-years at 4 percent annual interest, is $3201.00.

Annual life cycle cost = 0.06401 × $50,000

= $3201.00

## 2.4 ANNUAL OWNING COSTS

### 2.4.1 Initial Costs

There are many costs associated with owning and operating any commercial building. The initial costs should be amortized over the equipment life as part of the annual owning costs.

| Item | Percentage |
|---|---|
| Financing (New) | 44% |
| Maintenance Operation | 30% |
| Initial Construction | 20% |
| Indirect Construction | 2% |
| Land | 2% |
| A/E Fees | 2% |
| Miscellaneous | 1% |
| | 100% |

**Table 2–2 Cost of Owning and Operating a Typical Commercial Building**

### 2.4.2 Interest

### 2.4.3 Taxes

a. Property and real estate taxes.

b. Local taxes.

c. Other building taxes.

### 2.4.4 Insurance

## 2.5 ANNUAL OPERATING COSTS

### 2.5.1 Annual Energy Costs

a. Energy and fuel costs.

b. Water charges.

c. Sewer charges.

### 2.5.2 Annual Maintenance Costs

a. Maintenance Contract.

b. General housekeeping.

c. Parts and filters.

d. Refrigerant, oil, and grease.

e. Cleaning and painting.

f. Testing and rebalancing.

g. Waste disposal.

h. Chemicals for water treatment.

### 2.5.3 Maintenance Staff

The annual salaries of building engineers and maintenance staff should not be included in the above annual maintenance cost. This should be considered a separate cost item.

## 2.6 OPERATION COSTS

A duct system does not normally require any allowance for annual maintenance expense, but a duct system can have a significant impact on annual energy costs. Fan size and power required to move air through any duct system is directly related to system total pressure. Chapter 6 and 7 will provide several methods to calculate these system pressure losses and associated fan power requirements.

Since most HVAC system fans are required by building code to operate continuously when the building is occupied, the energy requirements for various air distribution systems is a major contributor to the total building utility costs. Reducing duct velocities and static pressure losses can minimize fan energy. However, this has a direct bearing on system first-cost, since extra space for enlarged ductwork throughout the building, and larger HVAC equipment rooms might be required. It is extremely important for the HVAC system designer to adequately investigate and calculate the impact of system operating costs, versus first cost.

For example, computations have confirmed that a continuously operating HVAC system costs three cents per cfm (six cents per L/s) per 0.25 in. wg (62 Pa) static pressure annually, based on nine cents per kWh cost of electrical energy. Therefore, a 0.25 in. wg (62 PA) increase in static pressure for a 100,000 cfm (50,000 L/s) system would add $3000.00 to the cost of the HVAC operation for one year. An increase in HVAC system operating static pressure will also increase the first-cost of the system.

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1. Energy and Fuel Service Costs
   a. Fuel service, storage, handling, piping, and distribution costs
   b. Electrical service entrance and distribution equipment costs
   c. Total energy plant

2. Heat-Producing Equipment
   a. Boilers and furnaces
   b. Steam-water converters
   c. Heat pumps or resistance heaters
   d. Make-up air heaters
   e. Heat producing equipment auxiliaries

3. Refrigeration Equipment
   a. Compressors, chillers, or absorption units
   b. Cooling towers, condensers, well water supplies
   c. Refrigeration equipment auxillaries

4. Heat Distribution Equipment
   a. Pumps, reducing valves, piping, piping insulatation, etc.
   b. Terminal units or devices
   c. Geothermal ground loops on wells.

5. Cooling Distribution Equipment
   a. Pumps, piping, piping insulation, condensate drains, etc.
   b. Terminal units, mixing boxes, diffusers, grilles, etc.
   c. Geothermal ground loops on wells.

6. Air treatment and Distribution Equipment
   d. Air heaters, humidifiers, dehumidifiers, filters, etc.
   e. Fans, ducts, duct insulation, dampers, etc.
   f. Exhaust and return systems

7. System and Controls Automation
   a. Terminal or zone controls
   b. System program control
   c. Alarms and indicator system

8. Building Construction and Alteration
   a. Mechanical and electric space
   b. Chimneys and flues
   c. Building insulation
   d. Solar radiation controls
   e. Acoustical and vibration treatment
   f. Distribution shafts, machinery foundations, furring

**Table 2–3 Initial Cost Systems**

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## 2.7 CONTROLLING COSTS

Some rule-of-thumb industry practices that can lower first-costs are:

a. Use the minimum number of fittings possible. Fittings are expensive, and the pressure loss of fittings is far greater than straight duct sections. For example, one 24 × 24 in. (600 × 600 mm) radius elbow with an R/W ratio of 1.0 has a pressure loss equivalent to 29 feet (8.8 m) of straight duct.

b. Use a semi-extended plenum where possible, *see* Chapter 7.

c. Seal ductwork to minimize air leakage. This can also reduce equipment and ductwork sizes.

d. Use round duct where space and initial cost allow, round ductwork has the lowest duct friction loss for a given perimeter.

e. Maintain the aspect ratio as close to 1-to-1 as possible when sizing rectangular ductwork to minimize friction loss.

## 2.8 DUCT ASPECT RATIOS

It is very important to understand the impact of aspect ratios of rectangular ducts on initial cost and annual operating cost. Table 2-4 contains an aspect ratio example of different straight duct sizes that will convey the same airflow at the same duct pressure friction loss. When comparing the weight of the higher aspect ratio ducts per foot (meter), keep in mind the cost of labor and material will also be greater.

The cost for different types of ductwork and the use of taps instead of divided-flow fittings can materially affect installation costs as shown in Figure 2-1. Figures 2-2 and 2-3 show how relative costs may vary with aspect ratios. Caution must be used with any table or chart since duct construction materials and methods, system operating pressures, and duct system location can all affect these cost relationships considerably.

## 2.9 PRESSURE CLASSIFICATION AND LEAKAGE

The HVAC system designer needs to indicate the operating pressures for the various sections of each duct system on the plans. This recommendation is noted in all SMACNA publications and is required so each system segment will have the structural strength for the indicated pressure classification, while keeping initial construction costs as low as possible. Each advancement to the next higher duct pressure class will increase duct system construction costs.

Since the installed cost for any duct system varies greatly due to labor rates, cost of materials, and local variables, it is virtually impossible to present specific cost data. Therefore, a system of relative cost has been provided. Considering the lowest pressure classification, 0.0 to 0.5 in. wg (0 to 125 Pa) static pressure as a base (1.0), the tabulation in Table 2-5 will give the designer a better appreciation of the relative cost of the various pressure classifications.

Table 2-5 is based on galvanized sheet metal ductwork sealed in accordance with the minimum classifications as listed in Standard Duct Sealing Requirements table in the SMACNA *HVAC Duct Construction Standards – Metal and Flexible*.

The amount of duct air leakage should be determined in advance by the HVAC system designer, so the estimated leakage can be added to the system airflow total to be used for selecting the system supply air fan. The amount of duct air leakage in terms of cfm/100 $ft^2$ (L/s / $m^2$) is based on the amount of ductwork in each "seal class."

| Duct Dimensions | | Duct Area | | Aspect Ratio | Metal Thickness | | | Duct Weight* | |
|---|---|---|---|---|---|---|---|---|---|
| in. | mm. | sq. in. | sq. m. | | ga | in. | mm. | lbs./ft | kgs./m |
| 24 (diam.) | 600 (diam.) | 452 | 0.28 | — | 26 | 0.022 | 0.55 | 5.70 | 8.35 |
| 22 × 22 | 550 × 550 | 484 | 0.30 | 1 : 1 | 26 | 0.022 | 0.55 | 6.64 | 9.73 |
| 30 × 16 | 750 × 400 | 480 | 0.30 | 1.9 : 1 | 26 | 0.022 | 0.55 | 6.95 | 10.71 |
| 44 × 12 | 1100 × 400 | 528 | 0.33 | 3.7 : 1 | 22 | 0.034 | 0.85 | 13.12 | 19.21 |
| 60 × 10 | 1500 × 250 | 600 | 0.38 | 6 : 1 | 20 | 0.040 | 1.00 | 19.32 | 28.28 |
| 80 × 8 | 2000 × 200 | 640 | 0.40 | 10 : 1 | 18 | 0.052 | 1.31 | 31.62 | 46.29 |

**Table 2–4 Aspect Ratio Example**

* Duct Weight based on 2 in. wg (500 Pa) Pressure Classification, 4 foot (1.22 m) Reinforcement Spacing. Weight of Reinforcement and Hanger materials not included.

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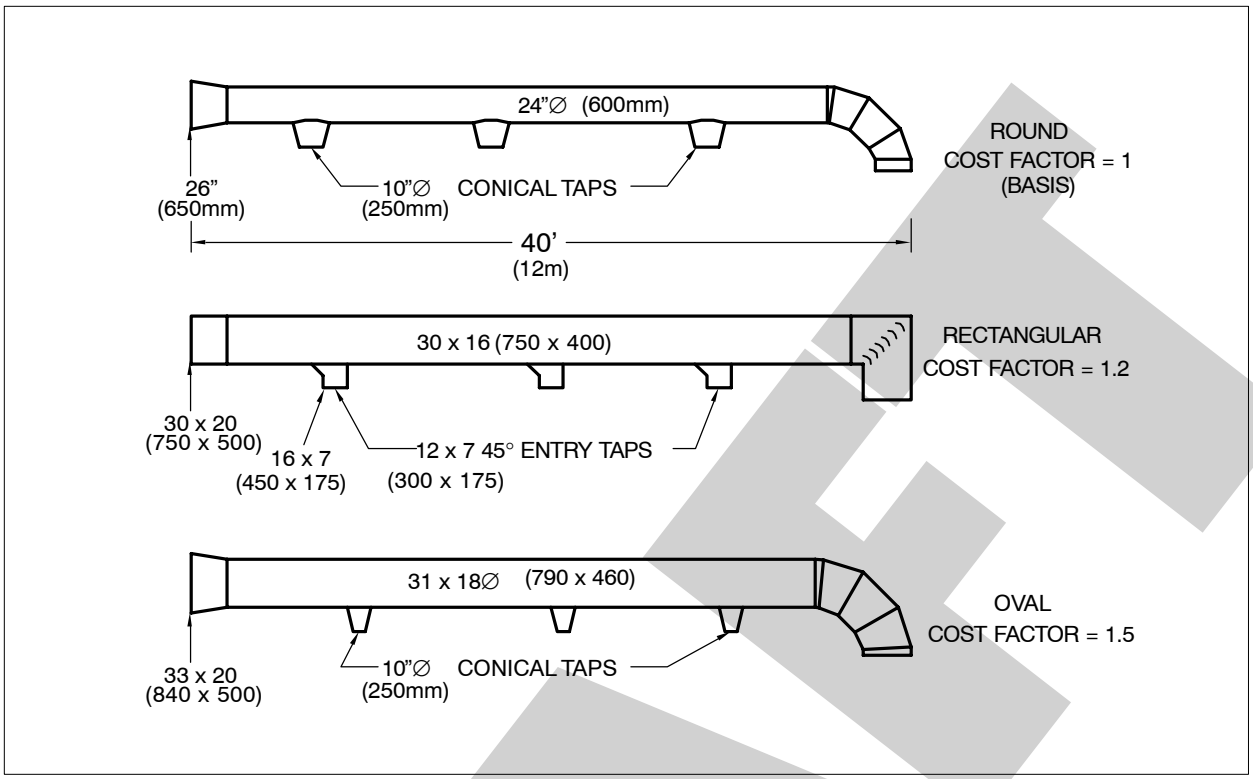


**FIGURE 2–1 RELATIVE COSTS OF DUCT SEGMENTS INSTALLED**

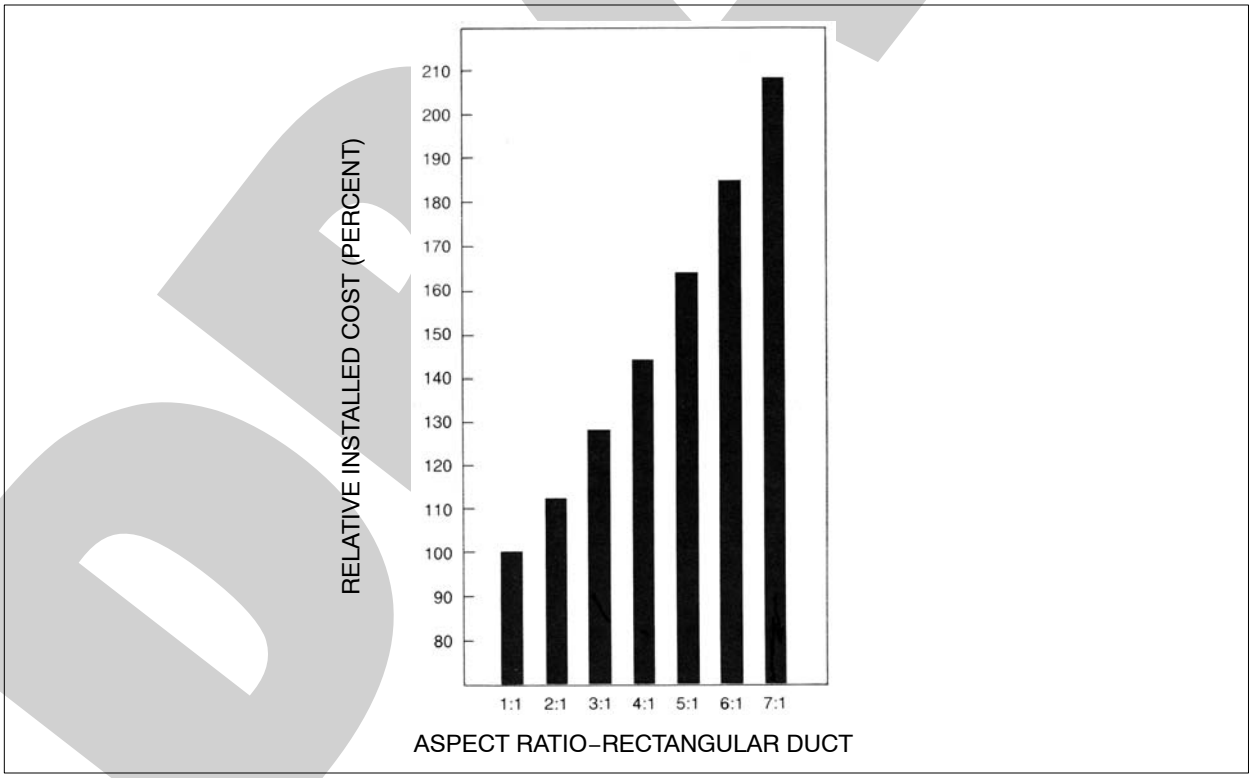


**FIGURE 2–2 RELATIVE INSTALLED COST VERSES ASPECT RATIO**

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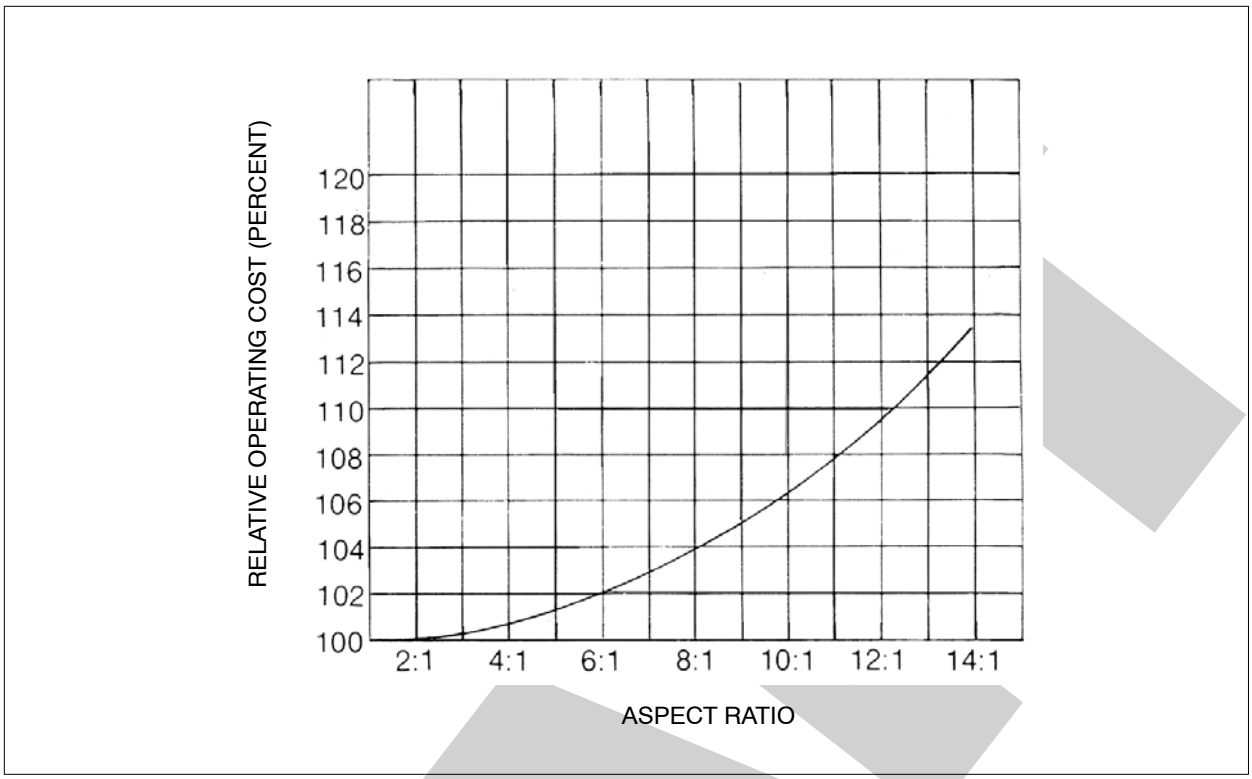


**FIGURE 2–3 RELATIVE OPERATING COST VERSES ASPECT RATIO**

Additional information may be found in the SMACNA *HVAC Air Duct Leakage Test Manual*, and in the *ASHRAE Handbook - Fundamentals*. It is important to note that a one percent air leakage rate for large HVAC duct systems is almost impossible to attain, and a large unsealed duct system may develop leakage well above 30 percent of total system airflow. Generally, a significant portion of the total HVAC system "duct" leakage actually occurs at HVAC equipment casings and the designer, not the HVAC contractor, has responsibility for equipment air leakage rates. The cost of sealing ductwork can add approximately 5 to 10 percent to the HVAC duct system fabrication and installation costs.

| Duct Pressure Class | | Cost Ratio |
|---|---|---|
| In. wg | Pa | |
| 0 - 0.5 | 0 - 125 | 1.00 |
| 0.5 - 1.0 | 125 - 150 | 1.05 |
| 1.0 - 2.0 | 250 - 500 | 1.15 |
| 2.0 - 3.0 | 500 - 750 | 1.40 |
| 3.0 - 4.0 | 750 - 1000 | 1.50 |
| 4.0 - 6.0 | 1000 -1500 | 1.60 |
| 6.0 - 10.0 | 1500 -2500 | 1.80 |

**Table 2–5 Relative Duct System Costs (Fabrication and installation of same size duct)**

## 2.10 COST OF FITTINGS

The "Duct Design Tables and Charts" in the Appendix contain fitting loss coefficient data for the HVAC system designer. However, the fitting that gives the lowest dynamic loss may also be the most expensive to make. A rectangular duct fitting having a higher aspect ratio might cost only slightly more than a square fitting and much less than some round fittings.

For example, using a 5 ft (1.5 m) section of ductwork as a base, the relative cost of a simple full radius elbow of constant cross-sectional area is approximately 4 to 8 times that of a straight section of ductwork. The relative cost of a square-throated elbow with turning vanes might be even greater.

The HVAC system designer should keep in mind that much of the straight ductwork fabricated today is done by automated equipment with fabrication labor reduced to a minimum. However, fittings—transitions, offsets, elbows, etc.—must still be individually hand-formed and assembled. As the number of fittings relative to the straight duct increases, over-all labor cost also increases.

| Equipment Item | Median Years | Equipment Item | Median Years | Equipment Item | Median Years |
|---|---|---|---|---|---|
| Air conditioners | | Air terminals | | Air-cooled condensers | 20 |
| Window unit | 10 | Diffusers, grilles, and registers | 27 | Evaporate condensers | 20 |
| Residential single or split package | 15 | Induction and fan-coil units | 20 | Insulation | |
| Commercial through-the-wall | 15 | VAV and double-duct boxes | 20 | Molded | 20 |
| Water-cooled package | 15 | Air washers | 17 | Blanket | 24 |
| Heat pumps | | Duct work | 30 | Pumps | |
| Residential air-to-air | 15 | Dampers | 20 | Base-mounted | 20 |
| Commercial air-to-air | 15 | Fans | | Pipe-mounted | 10 |
| Commercial water-to-air | 19 | Centrifugal | 25 | Sump and well | 10 |
| Roof-top air condtioners | | Axial | 20 | Condensate | 15 |
| Single-zone | 15 | Propeller | 15 | Reciprocating engines | 20 |
| Multizone | 15 | Ventilating roof-mouted | 20 | Steam turbines | 30 |
| Boilers, hot water (steam) | | Coils | | Electric motors | 18 |
| Steel water-tube | 24 | DX, water, or steam | 20 | Motor starters | 17 |
| Steel fire-tube | 24 | Electric | 15 | Electric transformers | 30 |
| Cast iron | 35 | Heat Exchangers | | Controls | |
| Electric | 15 | Shell-and-tube | 24 | Pneumatic | 20 |
| Burners | 21 | Reciprocating compressors | 20 | Electric | 16 |
| Furnaces | | Package chillers | | Electronic | 15 |
| Gas- or oil-fired | 18 | Reciprocating | 20 | Valve actuators | |
| Unit heaters | | Centrifugal | 23 | Hydraulic | 15 |
| Gas or electric | 13 | Absorption | 23 | Pneumatic | 20 |
| Hot water or steam | 20 | Cooling towers | | Self-contained | 10 |
| Radiant heaters | | Galvanized metal | 20 | | |
| Electric | 10 | Wood | 20 | | |
| Hot water or steam | 25 | Ceramic | 34 | | |

**Table 2–6 Estimated Equipment Service Lives**

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CHAPTER 3

# ROOM AIR DISTRIBUTION

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## 3.1 SCOPE

Before addressing room air distribution systems and system components, it is helpful to understand how human comfort factors should affect distribution system design.

Of the many human comfort factors and principles, the ones that can be controlled or at least affected by environmental systems are air temperature, radiant temperature, humidity, air movement, and noise. The important personal factors are clothing and activity level.

Much of this chapter was taken from ASHRAE's Standard 55-2004 Standard *Thermal Environmental Conditions for Human Occupancy.* The basic aim of these standards is to specify conditions that are thermally accepted to 80 percent or more of the occupants. This standard now includes adjustments that can be made for clothing, activity, air movement, and temperature drifts that give greater flexibility and have energy saving potential. In addition, ASHRAE's Standard introduces limits on the thermal non-uniformities in the space to decrease the probability of local discomfort.

## 3.2 COMFORT

### 3.2.1 Physiological Principals

An elementary understanding of physiological principles is essential to the person who designs or works with environmental and HVAC systems used for the thermal comfort of the human occupants of a building. An in-depth study of the subject is complex, including such elements as how the body reacts to cold or heat under various conditions and clothing. There are also many considerations when a controlled, artificial environment is provided for building occupants who depend on the surrounding air to sustain their lives.

The human body through its process of metabolism, breathing, and perspiring gives off varying amounts of heat and moisture. Persons in good health normally maintain a body temperature of 98.6 degrees F (37 degrees C), as long as a reasonable equilibrium with the surroundings is controlled. The body gains heat from external sources by radiation from the sun or other hot objects, and by conduction and convection from the surrounding air.

The body loses heat by:

a. *Conduction* when the body temperature is higher than that of the surrounding air. The type and amount of clothing can greatly influence this conduction loss.

b. *Convection* to the layer of air immediately adjacent to the skin. If air is circulated past the body, the convection heat transfer process is speeded up allowing the person to feel cooler even though the surrounding temperature does not change.

c. *Radiation* to surrounding surfaces such as walls, ceiling, floor and windows.

d. *Evaporation* of perspiration from the skin. If the surrounding air and surfaces are warmer than the skin temperature, evaporation is the only method the body has to lose heat because it can be gaining heat from conduction, convection, and radiation.

The evaporation of perspiration is increased as the relative humidity of the air decreases. This allows the body to maintain a satisfactory heat loss at dry bulb temperatures of 120 degrees F (49 degrees C) (or higher), if the relative humidity remains below 20 percent. However, a lower dry bulb temperature of 95 degrees F (35 degrees C) is uncomfortable, or even harmful to some people, when the relative humidity remains high.

The final analysis of human comfort is based on the condition of the air surrounding the body and the temperatures of the nearby surfaces. The goal of the HVAC system designer is to provide a system that can maintain that comfort in the artificial environment, while people are engaged in selected activities.

| Activity | Metabolic Rate in Met Units |
|---|---|
| Reclining | 0.8 |
| Seated, quietly | 1.0 |
| Sedentary activity (office, dwelling, lab, school) | 1.2 |
| Standing, relaxed | 1.2 |
| Light activity, standing (shopping, lab, light industry) | 1.6 |
| Medium activity, standing (shop assistant, domestic work, machine work) | 2.0 |
| High activity (heavy machine work, garage work) | 3.0 |

**Table 3–1 Metabolic Rates of Typical Tasks**

### 3.2.2 Environmental Indices

In choosing optimal conditions for comfort and health, knowledge of the energy expended during the course



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of routine physical activities is necessary, since body-heat production increases in proportion to exercise intensity. This is illustrated in Table 3-1. One "met" is defined as 18.4 Btu/h $ft^2$ (58 $W/m^2$) and is equal to the energy produced per unit surface area of a seated person at rest. The external surface area of an average male adult is 19 $ft^2$ (1.77 $m^2$).

In order to evaluate the sensation of comfort for the human body, three classes of environmental indices are used: direct, rationally derived, and empirical.

a. Direct indices include:
   - Dry bulb temperature
   - Dew point temperature
   - Wet bulb temperature
   - Relative humidity
   - Air motion

b. Rationally derived indices are:
   - Mean radiant temperature
   - Operative temperature
   - Humid operative temperature
   - Heat stress index
   - Index of skin wetness

c. Empirical indices are:
   - Effective temperature
   - Black globe temperature
   - Corrected effective temperature
   - Wet bulb globe temperature index
   - Wind chill index

### 3.2.3 Definitions

a. Mean radiant temperature: The uniform surface temperature of a radiantly-black enclosure in which an occupant would exchange the same amount of radiant heat as in the actual non-uniform space.

b. Operative temperature: The uniform temperature of a radiantly-black enclosure in which an occupant would exchange the same amount of heat by radiation, plus convection as the actual non-uniform environment.

c. At air speeds of 80 fpm (0.4 m/s) or less and mean radiant temperature of less than 120 degrees F (49 degrees C) operative temperature is approximately the simple average of the air's and person's radiant temperatures, and is equal to the adjusted dry-bulb temperature.

d. Humid operative temperature: The uniform temperature of an environment at 100 percent relative humidity, when a person will exchange the same heat from their skin surface by radiation, convection, and conductance, through clothing and evaporation in the actual environment.

e. Heat stress index: This rational index is the ratio of total evaporative heat loss required for thermal equilibrium or the sum of metabolism plus dry heat load to maximum evaporation to the environment itself.

f. Index of skin wetness: The ratio of observed skin sweating to the maximum allowed by the environment as defined by humidity, air movement, and clothing. Skin wetness is more closely related to the sense of discomfort or unpleasantness than temperature sensation.

g. Effective temperature: The uniform temperature of a radiantly-black enclosure of 50 percent relative humidity in which an occupant would experience the same comfort, physiological strain, and heat exchange as in the actual environment with the same air motion.

h. Black-globe temperature: The equilibrium temperature of a 6 in. (150 mm) diameter black globe which has been used as a single temperature index describing the combined physical effect of dry-bulb temperature, air movement, and radiant heat received from various surrounding areas.

i. Corrective effective temperature: Black globe temperature is substituted for the dry-bulb temperature of the original Effective Temperature Scale to correct for effects of any intense radiant heat source in the surrounding environment. The globe temperature is a measure of the operative temperature in an enclosed space in which comfort may be maintained by radiant heat.

j. Wet-bulb globe temperature index: Used as a weighted average of the dry bulb, a naturally convected wet bulb, and globe temperature.

k. Wind chill indices: An index that describes the rate of heat loss from a liter-size cylinder of water at 91.4 degrees F (33 degrees C) as a function of ambient temperature and wind velocity.

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Rather than express wind chill level as a numerical index, common practice has been to use the corresponding equivalent wind-chill temperature. This is the ambient temperature in a calm, under 4 mph (1.79 m/s) wind, that would yield the same body heat loss.

l. Throw: The *throw* of a jet is the distance from the outlet device to a point in the airstream where the maximum velocity in the stream cross-section has been reduced to a selected terminal velocity. Data for the throw of a jet from various outlets are generally given by each manufacturer for isothermal-jet conditions, without boundary walls interfering with the jet. Throw data certified under Air Diffusion Council (ADC) Equipment Test Code must be taken under isothermal conditions. ASHRAE's Standard also includes specifications for reporting throw data.

m. Throw distance: The *throw distance* of a jet is denoted by the symbol $T_v$, where the subscript indicates the terminal velocity for which the throw is given.

n. Characteristic room length: The *characteristic room length* (L) is the distance from the outlet device to the nearest boundary wall in the principal horizontal direction of the airflow. However, where air injected into the room does not impinge on a wall surface but mixes with air from a neighboring outlet. The characteristic length (L) is one-half the distance between outlets, plus the distance the mixed jets must travel downward to reach the occupied zone.

o. Midplane: The *midplane* between outlets can be considered the module line when outlets serve equal modules throughout a space. Characteristic length consideration can then be based on module dimensions.

### 3.2.4 The Thermal Environment

The environmental parameters of temperature, radiation, humidity, and air movement necessary for thermal comfort depend upon the occupant's clothing and activity level. In the majority of applications of ASHRAE, the building occupants will be sedentary or slightly active and wearing typical indoor clothing. The perception of comfort, temperature, and thermal acceptability are related to one's metabolic heat pro-

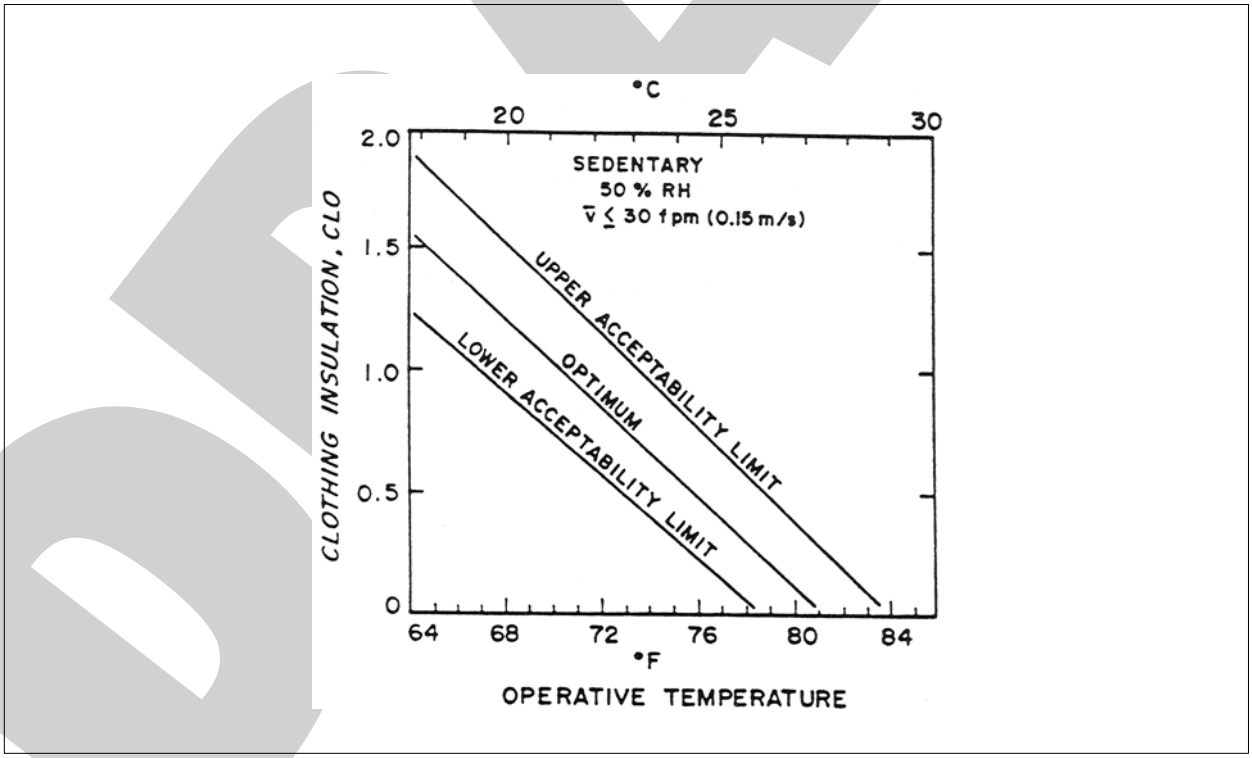


**FIGURE 3–1 CLOTHING INSULATION NECESSARY FOR VARIOUS LEVELS OF COMFORT AT A GIVEN TEMPERATURE DURING LIGHT, MAINLY SEDENTARY ACTIVITIES**

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duction, its transfer to the environment, and the resulting physiological adjustments and body temperatures.

Clothing, through its insulation properties, is an important modifier of body heat loss and comfort. Clothing insulation can be described in terms of its clo-value ($I_{cl}$). A heavy-wool business suit with normal accessories has an insulation value of about 1 clo. A pair of walking shorts is about 0.08 clo. A definition of "clo value" is the numerical representation of a person's clothing thermal resistance; 1 clo = 0.88 $ft^2 \cdot h \cdot F/Btu$ (0.155 $m^2$ ºC/W). The operative temperatures and clo values corresponding to the optimum sensation of neutral, and the 80 percent thermal acceptability limits of the ASHRAE's Standard are given in Figure 3-1.

Environmental temperature and humidity often fluctuate as the control system adjusts the heating and air conditioning equipment. Studies show that allowable fluctuating limits stated in the ASHRAE Comfort Standard are conservative; and that, in the thermal comfort range, the rate of change of temperature should not exceed 4 degrees F (2 degrees C) per hour, if the peak-to-peak variation in the dry bulb temperature cycle is 2 degrees F (1 degrees C) or greater.

For mean radiant temperature fluctuations, the rate of 3 degrees F (1.7 degrees C) per hour should not be exceeded if the peak-to-peak variation in mean radiant temperature is 1.5 degrees F (0.8 degrees C) or greater. Humidity limits are quite broad, usually posing no problem for modern control systems; the rate of *relative humidity* change should not exceed 20 percent per hour if the peak-to-peak variation in humidity is 10 percent or greater. However, below 30 percent humidity in the winter and above 60 percent humidity in the summer can adversely affect building materials, furnishings, and paper products.

### 3.2.5 Sedentary People – Indoor Clothing

#### 3.2.5.1 Temperature

The insulation value of clothing worn by people indoors is influenced by the season and outside weather conditions. During summer months, typical clothing has insulation values ranging from 0.35 to 0.6 clo. The winter heating season brings a change to thicker, heavier clothing. A typical winter outfit would have an insulation value range from 0.8 to 1.2 clo.

Between seasons, the clothing may likely consist of medium-weight slacks or shirt, or long-sleeved shirt or blouse, having an insulation value in the rate of 0.6 to 0.8 clo.

Because of the seasonal clothing habits of building occupants, the temperature range for comfort in summer is higher than for winter. The operative temperature range when at least 80 percent of the sedentary or near sedentary (≤1.2 mets) occupants will find the environment thermally acceptable, are given in Table 3-2.

The zones overlap in the 73 to 75 degree F (23 to 24 degree C) range. In this region, people in summer dress would tend to feel slightly cool, while those in winter clothing would have a slightly warm sensation. The boundaries of each comfort zone are not as sharp as depicted in Figure 3-1, due to individual clothing and activity differences. Table 3-2 also gives the acceptable operative temperature range for sedentary persons in minimal clothing.

| Season | Description of Typical Clothing | clo | Optimum Operative Temperature | Operative Temperature Range for 80% Thermal Acceptability[b] |
|---|---|---|---|---|
| Winter | Heavy slacks, long sleeve and sweater | 0.9 | 71°F (22°C) | 68° - 75°F (20° - 24°C) |
| Summer | Light slacks and short sleeve shirt | 0.5 | 76°F (24°C) | 73° - 79°F (23° - 26°C) |
| | Minimal | 0.05 | 81°F (27°C) | 79° - 84°F (26° - 29°C) |

**Table 3–2 Operative Temperatures for Thermal Acceptability or Sedentary or slightly Active persons (≤ 1.2 Mets) at 50 percent Relative Humidity[a]**

*NOTES:*

[a]Other than clothing there are no season or sex variations on the temperatures of Table 3-2. For infants, certain elderly people and individuals who are physically disabled, the lower limits of Table 3-2 should be avoided.
[b]Slow air movement ≤ 30 fpm (0.15 m/s) and 50 percent RH.

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### 3.2.5.2 Humidity

Humidity is described in terms of dew-point temperature. In the zone occupied by sedentary or near sedentary people, the dew-point temperature should not be less than 35 degrees F (2 degrees C) or greater than 62 degrees F (17 degrees C). The thermal effect of humidity on the comfort of sedentary persons is small (Figure 3-1). The upper and lower dew-point limits are based on considerations of comfort, respiratory health, mold growth, and other moisture related phenomena. It should be noted that humidification in winter may need to be limited, to prevent condensation on building surfaces and windows.

### 3.2.5.3 Air Movement

Within the thermally-acceptable temperature ranges of Table 3-2 and Figure 3-1, there is no minimum air movement that is necessary for thermal comfort. Further, the maximum average air movement allowed in the occupied zone is lower in winter than in summer.

Winter: The average air movement in the occupied zone shall not exceed 30 fpm (0.15 m/s). If the temperature is less than the optimum or neutral sensation temperature, the maintenance of low air movements is important to prevent local draft discomfort.

Summer: The average air movement in the occupied zone shall not exceed 50 fpm (0.25 m/s). However, the comfort zone can be extended above 79 degrees F (26 degrees C) if the average air movement is increased 30 fpm (0.15 m/s) for each degree F (0.6 degree C) of increased temperature to a maximum temperature of 82.5 degrees F (28 degrees C), and air movement of 160 fpm (0.8 m/s) (Figure 12-3). Loose paper, hair, and other light objects may start to be disturbed at air movements of 160 fpm (0.8 m/s).

### 3.2.5.4 Mean Radiant Temperature

The mean radiant temperature is as important as air temperature in affecting heat loss and comfort. For indoor environments when air movement is low, the operative temperature is approximately the average of air temperature and mean radiant temperature. When the mean radiant temperature in the occupied zone differs from the air temperature, the air temperature or mean radiant temperature shall be adjusted to keep the operative temperature within the appropriate comfort zone.

## 3.2.6 Non–Steady State

### 3.2.6.1 Temperature Cycling

If the peak variation in operative temperature exceeds 2 degrees F (1 degree C), the rate of temperature change shall not exceed 4 degrees F (2.2 degrees C) per hour. There are no restrictions on the rate of temperature change if the peak-to-peak is 2 degrees F (1 degree C) or less.

### 3.2.6.2 Temperature Drifts or Ramps

Temperature drifts and ramps are monotonic, steady, non-cyclical temperature changes. Drifts refer to passive temperature changes, and ramps refer to actively controlled temperature changes. Slow rates of operative temperature change of approximately 1 degree F (0.6 degree C) per hour during the occupied period are acceptable, provided the temperature during a drift or ramp does not extend beyond the comfort zone by more than the 1 degree F (0.6 degree C) for longer than one hour.

## 3.2.7 Non–Uniformity

### 3.2.7.1 Vertical Temperature Difference

Air temperature in an enclosed space generally increases from floor to ceiling. If this increment is sufficiently large, local warm discomfort can occur at the head and cold discomfort at the feet, although the body as a whole is thermally neutral. Therefore, to prevent local discomfort, the vertical air temperature difference within the occupied zone, measured at the 4 in. (100 mm) and 67 in. (1700 mm) levels shall not exceed 5.4 degrees F (3 degrees C).

### 3.2.7.2 Radiant Asymmetry

Asymmetric radiation from hot and cold surfaces and from direct sunlight, can cause local discomfort and reduce the thermal acceptability level of the space. In general, people are more sensitive to asymmetric radiation caused by a warm ceiling than caused by hot and cold vertical surfaces. Thus, to limit local discomfort, radiant-temperature-asymmetry in the vertical direction should be less than 9 degrees F (5 degrees C), and in the horizontal direction less than 18 degrees F (10 degrees C). The radiant-temperature-asymmetry in the vertical direction is the difference in plane radiant temperature of the upper and lower parts of the space, with respect to a small horizontal plane 2 ft (600 mm) above the floor. In the horizontal direction, it is the difference in plane-radiant-temperature in opposite directions from a small vertical plane 2 ft (600 mm) above the floor.

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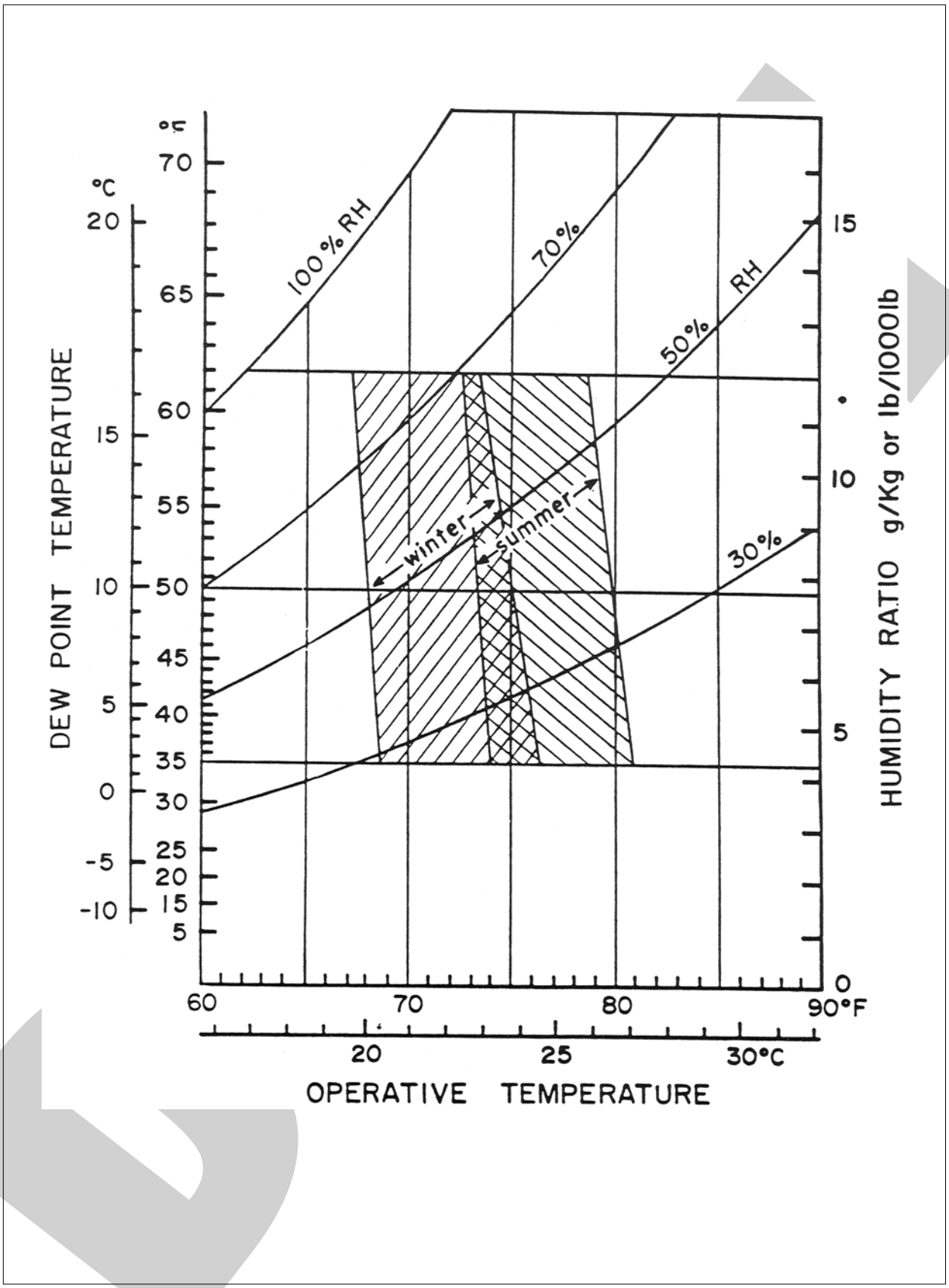


**FIGURE 3–2 COMFORT ZONE**

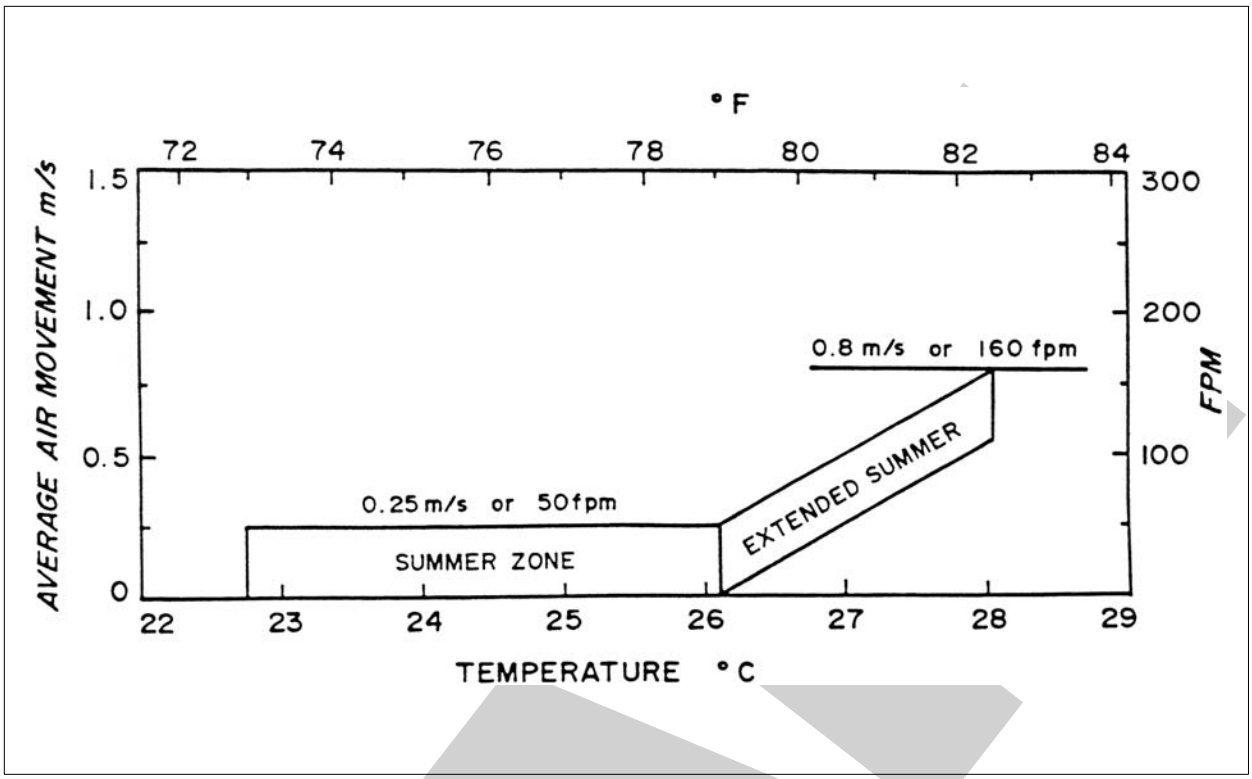


**FIGURE 3–3 RANGE OF AVERAGE AIR MOVEMENTS PERMITTED IN THE SUMMER AND EXTENDED SUMMER ZONES**

#### 3.2.7.3 Floor Temperatures

To minimize foot discomfort, the surface temperature of the floor for people wearing appropriate indoor footwear shall be between 66 degrees F (19 degrees C) and 84 degrees F (29 degrees C).

### 3.2.8 Sedentary People: Non–Typical Clothing

Comfort conditions for clothing levels different from those of the preceding subsection can be determined approximately by lowering the temperature ranges of Table 3-2 or Figure 3-2 by 1 degree F (0.6 degree C) for each 0.1 clo of increased clothing. At lower temperatures, comfort depends on the maintenance of a reasonably uniform distribution of clothing insulation over the entire body. For sedentary occupancy of more than an hour, the minimum operative temperature shall not be less than 65 degrees F (18 degrees C). The insulation of ensembles can be estimated from the sum of the clo values of the individual items worn, as described in Table 3-3, and multiplying the sum by 0.82. A rough approximation of the "clo" value may also be estimated from the clothing weight using the relationship of 0.15 clo /lb of clothing.

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| MEN | |
|---|---|
| **Clothing** | **clo** |
| Underwear | |
| Sleeveless | 0.06 |
| T Shirt | 0.09 |
| Briefs | 0.05 |
| Long underwear upper | 0.10 |
| Long underwear lower | 0.10 |
| Torso | |
| Shirt | |
| Light, short sleeve | 0.14 |
| Light, long sleeve | 0.22 |
| Heavy, short sleeve | 0.25 |
| Heavy, long sleeve | 0.29 |
| (Plus 5 percent tie or turtleneck) | |
| Vest | |
| Light | 0.25 |
| Heavy | 0.29 |
| Trousers | |
| Light | 0.26 |
| Heavy | 0.32 |
| Sweater | |
| Light | 0.20 |
| Heavy | 0.37 |
| Jacket | |
| Light | 0.22 |
| Heavy | 0.49 |
| Footwear | |
| Socks | |
| Ankle Length | 0.04 |
| Knee High | 0.10 |
| Shoes | |
| Sandals | 0.02 |
| Oxfords | 0.04 |
| Boots | 0.08 |

| WOMEN | |
|---|---|
| **Clothing** | **clo** |
| Underwear | |
| Bras and Panties | 0.05 |
| Half Slip | 0.13 |
| Full Slip | 0.19 |
| Long underwear upper | 0.10 |
| Long underwear lower | 0.10 |
| Torso | |
| Blouse | |
| Light | 0.20 |
| Heavy | 0.29 |
| Dress | |
| Light | 0.22 |
| Heavy | 0.70 |
| Skirt | |
| Light | 0.10 |
| Heavy | 0.22 |
| Slacks | |
| Light | 0.26 |
| Heavy | 0.44 |
| Sweater | |
| Light | 0.17 |
| Heavy | 0.37 |
| Jacket | |
| Light | 0.17 |
| Heavy | 0.37 |
| Stockings | |
| Any Length | 0.01 |
| Panty Hose | 0.01 |
| Shoes | |
| Sandals | 0.02 |
| Pumps | 0.04 |
| Boots | 0.08 |

**Table 3–3 Clo Units for Individual Items of Clothing = 0.82 (Σ Individual Items)**

### 3.2.9 Active People

The comfort zone temperatures of Table 3-2 and Figure 3-2 shall be decreased when the average steady state activity level of the occupants is higher than sedentary or slightly active (1.2 met). The acceptable operative temperature for activity depends on both the time average activity level, and the clothing insulation. The minimum allowable operative temperature for these activities is 59 degrees F (15 degrees C). Table 3-2 gives the "met" levels for some common activities. It must also be remembered that people may remove some of their normal clothing when performing hard or heavy work.

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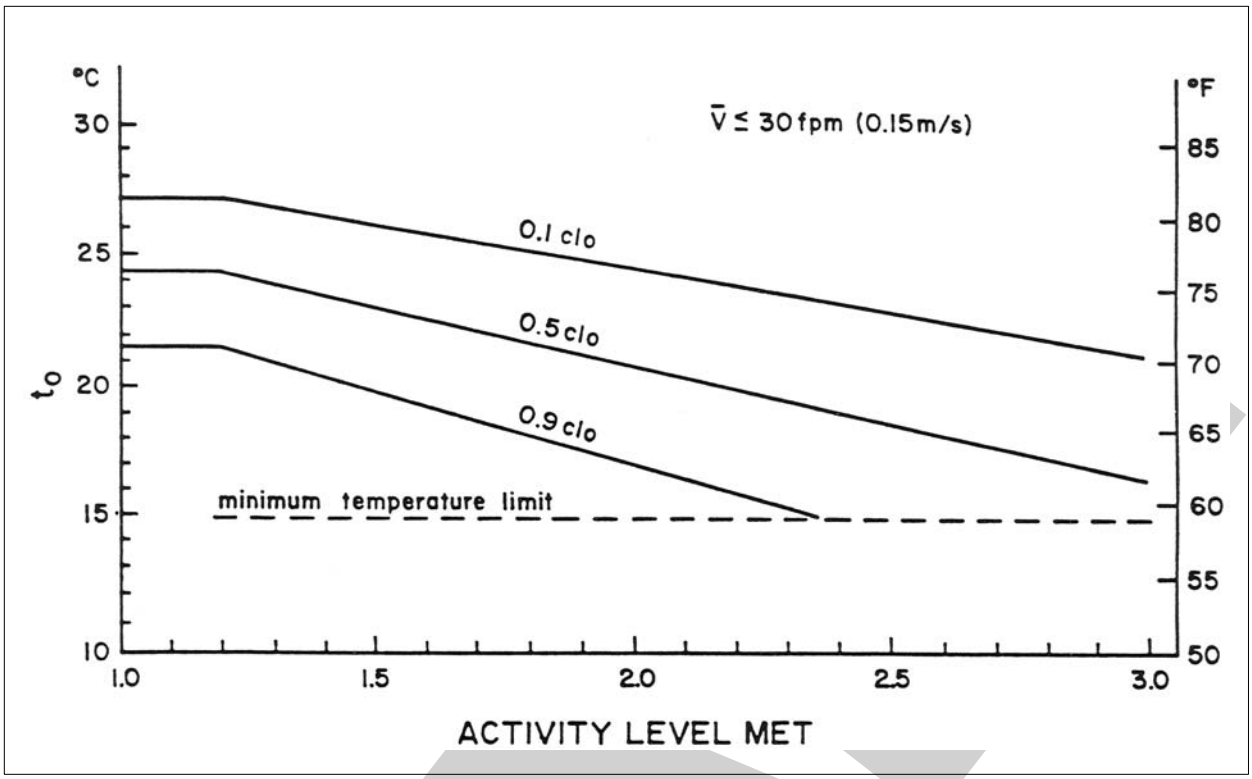


**FIGURE 3–4 OPTIMUM OPERATIVE TEMPERATURES FOR ACTIVE PEOPLE IN LOW AIR MOVEMENT ENVIRONMENTS**

### 3.2.10 Maintaining Comfort Conditions

An understanding of the principles of room air distribution helps in the selection, design, control, and operation of HVAC air duct systems. However, the real evaluation of air distribution in a space requires an affirmative answer to the question: "Are the occupants comfortable?" The object of good air distribution in any HVAC system is to create the proper combination of temperature, humidity, and air motion in the occupied zone of the conditioned room from the floor to 6 ft (2m) above floor level.

To obtain comfort conditions within this zone, standard limits have been established as an acceptable *effective draft temperature*. This term includes air temperature, air motion, relative humidity, and the physiological effects of these on the human body. Any variation from accepted standards of one of these elements can cause discomfort to occupants. Lack of uniform conditions within the space or excessive fluctuation of conditions in the same part of the space may produce similar negative effects.

Although the percentage of room occupants who object to certain conditions may change, Figures 3-5 and 3-6 provide insight into possible objectives of room-air distribution. These show that a person tolerates higher velocities and lower temperatures at ankle level than at neck level. Because of this, conditions in the zone extending from approximately 30 to 60 in. (0.75 to 1.5 m) above the floor are more critical than conditions nearer the floor.

Room-air velocities less than 50 fpm (0.25 m/s) are acceptable; however, Figure 3-5 and 3-6 show that even higher velocities may be acceptable to some occupants. ASHRAE recommends elevated air speeds at elevated air temperatures. No minimum air speeds are recommended for comfort, although air speeds below 20 fpm (0.1 m/s) are usually imperceptible.

Figure 3-5 shows that up to 20 percent of occupants will not accept an ankle-to-sitting-level gradient of about 4 degrees F (2 degrees C). Poorly designed or operated systems in heating mode can create this condition, which emphasizes the importance of proper selection and operation of perimeter systems.

To define the difference (è) in effective draft temperature between any point in the occupied zone and the control condition, the following equation is used:

**Equation 3–1**

$$\theta = (t_x - t_{c)} - a(V_x - b)$$

Where (I-P):

$\theta$ = effective draft temperature, degree F

$t_x$ = local airstream dry-bulb temperature, degree F

$t_c$ = average room dry-bulb temperature, degree F

$V_x$ = local airstream velocity, fpm

$a$ = 0.07

$b$ = 30

Where (SI):

$\theta$ = effective draft temperature, degree C

$t_x$ = local airstream dry-bulb temperature, degree C

$t_c$ = average room dry-bulb temperature, degree C

$V_x$ = local airstream velocity, m/s

$a$ = 8

$b$ = 0.15

Equation 3-1 accounts for the feeling of "coolness" produced by air motion and is used to establish the neutral line in Figures 3-5 and 3-6. In summer, the local airstream temperature, $t_x$, is below the control temperature. Hence, both temperature and velocity terms are negative when velocity, $V_x$, is greater than 30 fpm (0.15 m/s), both add to the feeling of coolness. If in winter, $t_x$, is above the control temperature, any air velocity above 30 fpm (0.15 m/s) subtracts from the feeling of warmth produced by $t_x$. Therefore, it is usually possible to have zero difference in effective temperature between location, x, and the control point in winter, but not in summer.

### 3.2.11 Air Diffusion Performance Index

#### 3.2.11.1 Comfort Criteria

A high percentage of people are comfortable in sedentary (office) occupations where the effective draft temperature ($\theta$), as defined in Equation 3-1, is between -3 and + 2 degrees F (-1.7 and + 1.1 degrees C), and the air velocity is less than 70 fpm (0.35 m/s). If multiple measurements of air velocity and air temperature were made throughout the occupied zone of an office, the Air Diffusion Performance Index (ADPI) would be defined, as the percentage of locations where measurements were taken that meet the previous specifications on effective draft temperature and air velocity. If the ADPI is maximum, (approaching 100 percent) the most desirable conditions are achieved.

ADPI is based only on air velocity and effective draft temperature, a combination of local temperature differences from the room average, and not directly related to the level of dry-bulb temperature or relative humidity. These and similar effects, such as mean radiant temperature, must be accounted for separately according to ASHRAE recommendations, *see* ASHRAE Standard 55.

The ADPI is a measure of cooling mode conditions, *see* ASHRAE Standard 113. Heating conditions can be evaluated using ASHRAE Standard 55.

#### 3.2.11.2 Load Considerations

These recommendations cover cooling loads of up to 80 Btu/h·ft$^2$ (250 W/m$^2$) of floor surface. The loading is distributed uniformly over the floor up to about 7 Btu/h·ft$^2$ (22 W/m$^2$). Lighting contributes about 10 Btu/h·ft$^2$ (31 W/m$^2$) and the remainder is supplied by a concentrated load against one wall that simulated a business machine or a large sun-loaded window. Over this range of data the maximum ADPI condition is lower for the highest loads; however, the optimum design condition changes only slightly with the load.

| Diffuser Type | Characteristic Length, *L* |
|---|---|
| High Sidewall Grille | Distance to wall perpendicular to jet |
| Circular Ceiling Diffuser | Distance to closest wall or intersecting air jet |
| Sill Grille | Length of room in the direction of the jet flow |
| Ceiling Slot Diffuser | Distance to wall or midplane between outlets |
| Light Troffer Diffusers | Distance to midplane between outlets, plus distance from ceiling to top of occupied zone |
| Perforated, Louvered Ceiling Diffusers | Distance to wall or midplane between outlets |

**Table 3–4 Characteristic Room Lengths for Diffusers**

#### 3.2.11.3 Design Conditions

The quantity of air must be known from other design specifications. If it is not known, the solution must be obtained by a trial and error technique.

The devices for which data were obtained are high-sidewall grille, sill grill, two-and four-slot ceiling dif-

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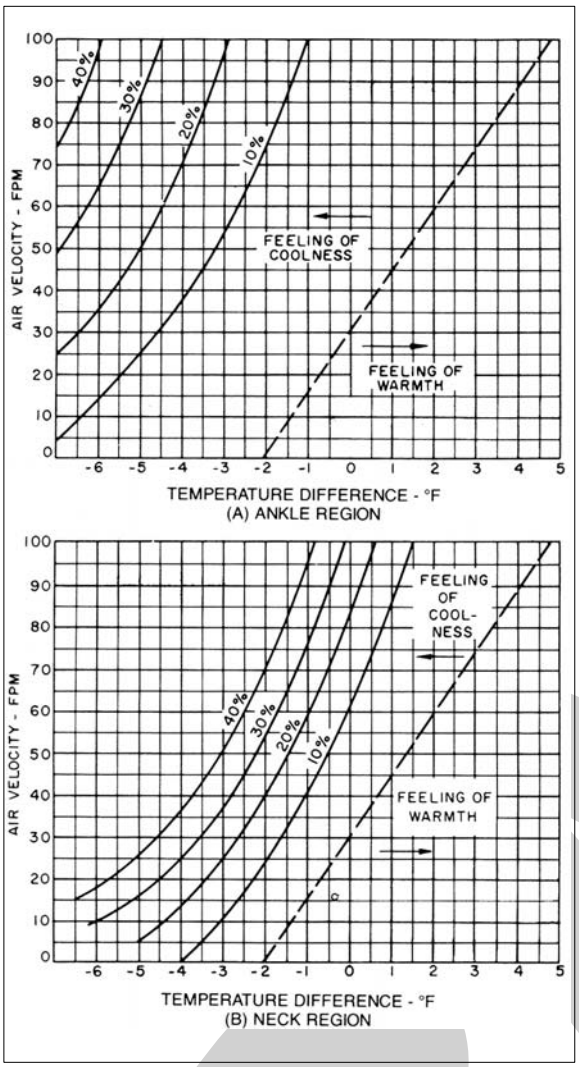


**FIGURE 3–5 PERCENTAGE OF OCCUPANTS OBJECTING TO DRAFTS IN AIR–CONDITIONED ROOMS (I–P)**

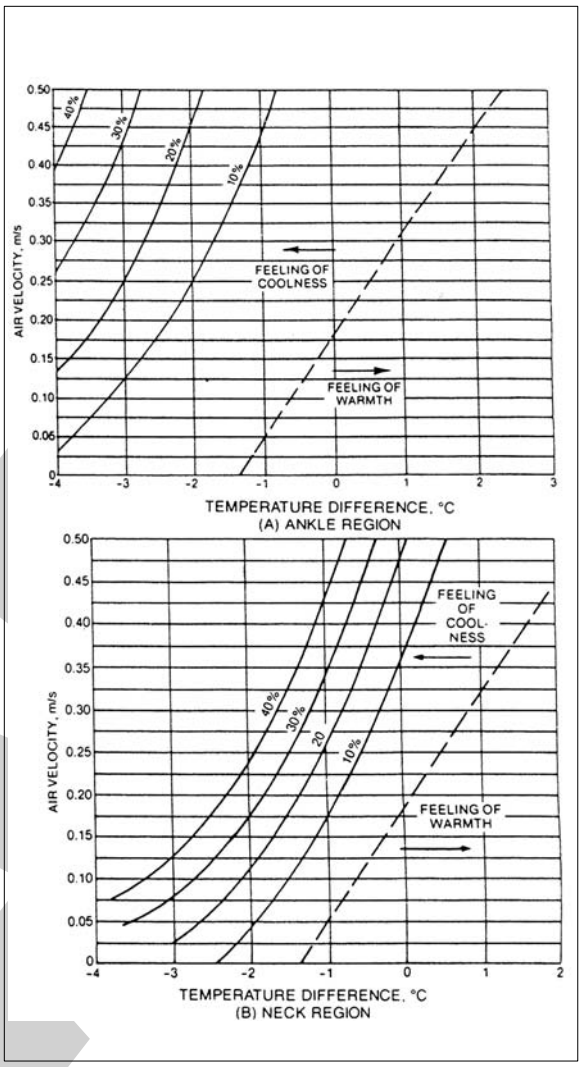


**FIGURE 3–6 PERCENTAGE OF OCCUPANTS OBJECTING TO DRAFTS IN AIR–CONDITIONED ROOMS (SI)**

fusers, cone-type circular ceiling diffusers, light-troffer diffusers, and square-faced perforated and louvered ceiling diffusers. Table 3-5 summarizes the results of the recommendations on values of $T_V/L$, by giving the value of $T_V/L$ when the ADPI is a maximum for various loads, as well as a range of values $T_V/L$ where ADPI is above a minimum specified value.

### 3.2.11.4 Outlet Type Selection

No criteria have been established for choosing among the six types of outlets to obtain an optimum ADPI. Referring to Table 3-5, as the room BTU/h·ft$^2$ values decrease, the ADPI values will increase. Most of the outlets tested, under these recommendations, will have ADPI values greater than 90, for room loads below 40 BTU/h·ft$^2$ (126 W/m$^2$)

### 3.2.11.5 Design Procedure

a. Determine the air volume requirements and room size.

b. Select the tentative outlet type and location within room.

c. Determine the room's characteristic length (L) (Table 3-4).

d. Select the recommended $T_V/L$ ratio from Table 3-5.

e. Calculate the throw distance ($T_V$) by multiplying the recommended $T_V/L$ ratio from Table 3-5 by the room length (L).

| Terminal Device | Room Load Btu/h ft$^2$ | Room Load W/m$^2$ | $T_{0.25}$/L for Max. ADPI | Maximum ADPI | For ADPI Greater Than | Range of $T_{0.25}$/L |
|---|---|---|---|---|---|---|
| High | 80 | 250 | 1.8 | 68 | – | – |
| Sidewall | 60 | 190 | 1.8 | 72 | 70 | 1.5-2.2 |
| Grilles | 40 | 125 | 1.6 | 78 | 70 | 1.2-2.3 |
| | 20 | 65 | 1.5 | 85 | 80 | 1.0-1.9 |
| Circular | 80 | 250 | 0.8 | 76 | 70 | 0.7-1.3 |
| Ceiling | 60 | 190 | 0.8 | 83 | 80 | 0.7-1.2 |
| Diffusers | 40 | 125 | 0.8 | 88 | 80 | 0.5-1.5 |
| | 20 | 65 | 0.8 | 93 | 90 | 0.7-1.3 |
| Sill Grille | 80 | 250 | 1.7 | 61 | 60 | 1.5-1.7 |
| Straight | 60 | 190 | 1.7 | 72 | 70 | 1.4-1.7 |
| Blades | 40 | 125 | 1.3 | 86 | 80 | 1.2-1.8 |
| | 20 | 65 | 0.9 | 95 | 90 | 0.8-1.3 |
| Sill Grille | 80 | 250 | 0.7 | 94 | 90 | 0.8-1.5 |
| Spread | 60 | 190 | 0.7 | 94 | 80 | 0.6-1.7 |
| Blades | 40 | 125 | 0.7 | 94 | – | – |
| | 20 | 65 | 0.7 | 94 | – | – |
| Ceiling | 80 | 250 | 0.3* | 85 | 80 | 0.3-0.7 |
| Slot | 60 | 190 | 0.3* | 88 | 80 | 0.3-0.8 |
| Diffusers (for $T_{100}$/L) | 40 | 125 | 0.3* | 91 | 80 | 0.3-1.1 |
| | 20 | 65 | 0.3* | 92 | 80 | 0.3-1.5 |
| Light | 60 | 190 | 2.5 | 86 | 80 | <3.8 |
| Troffer | 40 | 125 | 1.0 | 92 | 90 | <3.0 |
| Diffusers | 20 | 65 | 1.0 | 95 | 90 | <4.5 |
| Perforated and Louvered Ceiling Diffusers | 11-51 | 35-160 | 2.0 | 96 | 90 | 1.4-2.7 |
| | – | – | – | – | 80 | 1.0-3.4 |

**Table 3–5 Air Diffusion Performance Index (ADPI)**

*Given for $T_{0.5}$/L

f. Locate the appropriate outlet size from manufacturer's catalog.

g. Ensure that this outlet meets other imposed specifications, such as noise and static pressure.

*Example 3-1 (I-P):*

Specifications:

Room Size: 20 × 12 ft with 9 ft ceiling.

Type Device: High sidewall grille, located at the center of 12 ft end wall, 9 in. from ceiling.

Loading: Uniform, 10 Btu/h·ft$^2$ or 2400 Btu/h.

Air Volume: 1 cfm/ft$^2$ or 240 cfm for the one outlet.

Data Required:

Characteristic length: (L) = 20 ft (length of room: Table 3-4.)

Recommended $T_V$/L = 1.5 (Table 3-6)

Throw to 50 fpm = $T_{50}$ = 1.5 × 20 =30 ft

Refer to manufacturer's catalog data for a size that gives a isothermal throw to 50 fpm. The manufacturer

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recommends the following sizes when blades are straight, and discharging 240 cfm: 16 × 4 in., 12 × 5 in. or 10 × 6 in.

*Example 3-1 (SI):*

Specifications:

Room size: 6000 by 4000 mm with 2500 mm high ceiling.

Type Device: High sidewall grille located at the center of 4000 mm end wall, 230 mm from ceiling.

Loading: Uniform, 30 W/m$^2$ or 720 W.

Air Volume: 0.5 L/s per m$^2$ or 120 L/s per outlet.

Data Required:

Characteristic length L = 6000 mm (length of room: Table 3-5).

Recommended $T_V/L = 1.5$ (Table 3-6)

Throw to 0.25 m/s = $T_{0.25} = 1.5 \times 6 = 9$m

Refer to manufacturer's catalog data for a size that gives a isothermal throw to 0.25 m/s. The manufacturer recommends the following sizes when blades are straight and discharging 120 L/s: 400 × 100 mm, 300 × 125 mm or 250 × 125 mm.

## 3.3 AIR DISTRIBUTION FUNDAMENTALS

### 3.3.1 Air Diffusion

Conditioned air is normally supplied to air outlets at velocities much greater than those acceptable in the occupied zone. Conditioned air temperature may be above, below, or equal to the air. Proper air diffusion calls for the entrainment of room air by the primary airstream outside the zone of occupancy, to reduce airmotion and temperature differences to acceptable limits before the air enters the occupied zone.

This process of entrainment of secondary air into the primary air is an essential part of air distribution to create total air movement within the room. This process will also tend to overcome natural convection and radiation effects within the room, thereby eliminating stagnant-air areas and reducing temperature differences to acceptable levels before the air enters the occupied zone.

### 3.3.2 Surface (Coanda) Effect

Drawing A and B of Figure 3-7 illustrate the Coanda effect phenomenon. Since turbulent-jet airflow from a grille or diffuser is dynamically unstable, it may veer rapidly back and forth. When the jet airflow veers close to a parallel and adjacent wall or ceiling, the surface interrupts the flow path on the side as shown in Figure 3-7 (B).

The result is, no more secondary air is flowing on that side to replace the air being entrained with the turbu-

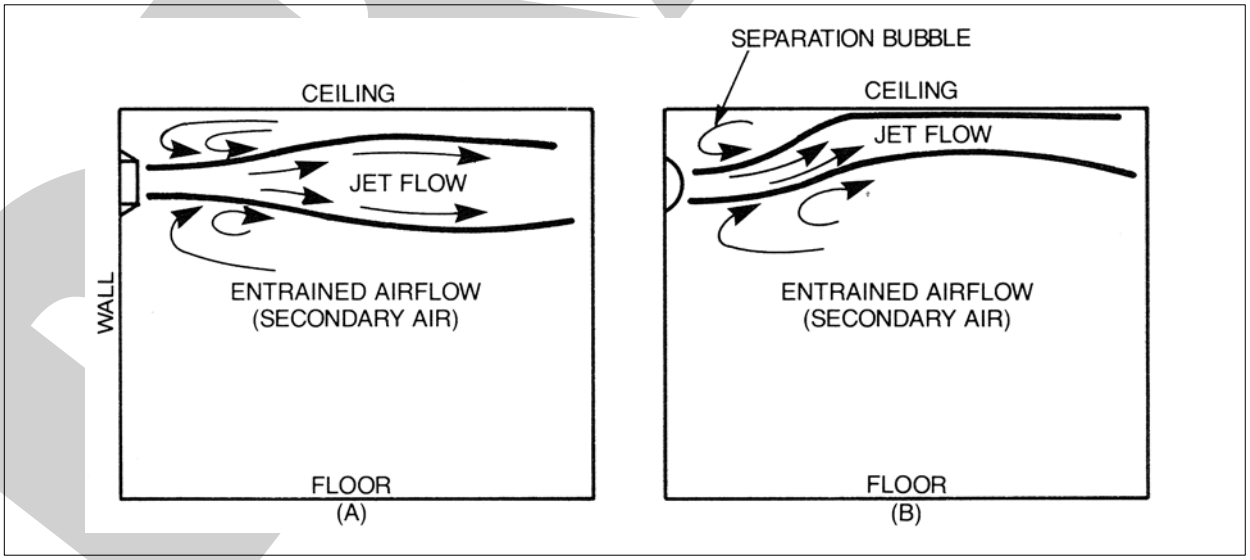


**FIGURE 3–7 SURFACE (COANDA) EFFECT**

lent-jet airflow. This causes a lowering of the pressure on that side of the outlet device, creating a low-pressure bubble that causes the turbulent-jet airflow to become stable and remain attached to the adjacent surface, throughout the length of the throw. The surface effect counteracts the drop of cool, horizontally-projected airstreams.

Ceiling diffusers exhibit surface effect to a high degree, because a circular air pattern blankets the entire ceiling area surrounding each outlet. Slot diffusers, which discharge the airstream across the ceiling, exhibit surface effect only if they are long enough to blanket the ceiling area. Grilles exhibit varying degrees of surface effect, depending on the spread of the particular air pattern.

In many installations, the outlets must be mounted on an exposed duct and discharge the airstream into free space. In this type of installation, the airstream entrains air on both its upper and lower surfaces. As a result, a higher rate of entrainment is obtained and the throw is shortened by about one-third. Airflow per unit area for these outlets can be increased to offset this effect. Because there is no surface effect from ceiling diffusers installed on the bottom of exposed ducts, the air drops rapidly to the floor. Therefore, temperature differentials in air conditioning systems must be restricted to a range of 15 to 20 degree F (8 to 11 degree C). Airstreams from slot diffusers and grilles show a marked tendency to drop because of the lack of surface effect.

### 3.3.3 Smudging

Smudging can be a problem with ceiling and slot diffusers. Dirt particles held in suspension in the secondary room air are subjected to turbulence at the outlet face. This turbulence, along with surface effect, is primarily responsible for smudging. Smudging can be expected in areas of high pedestrian traffic, including lobbies and retail stores. When ceiling diffusers are installed on smooth ceilings (such as plaster, mineral tile, and metal pan), smudging is usually in the form of a narrow band of discoloration around the diffuser. Anti-smudge rings may reduce this type of smudging. On highly-textured-ceiling surfaces, such as rough plaster and sprayed-on composition, smudging often occurs over a more extensive area.

### 3.3.4 Sound Level

The sound level of an outlet is a function of the discharge velocity and the transmission of system noise, which is a function of the size of the outlet. Higher-frequency sounds can be the result of excessive outlet velocity, but may also be generated in the duct by the moving airstream. Lower-pitched sounds are generally the result of mechanical equipment noise transmitted through the duct system and outlet.

Try to avoid using an outlet damper as a method to control system air balance. When possible, use balancing dampers located in the ductwork branches to reduce or eliminate noise from partially-closed damper blades located in outlets.

The cause of higher frequency sounds can be pinpointed as outlet or system sounds by removing the outlet during operation. A reduction in sound level indicates the outlet is causing noise. If the sound level remains essentially unchanged, the system is at fault. Chapter 10 of this manual, SMACNA's *HVAC Sound and Vibration Manual,* and the *ASHRAE Handbook - HVAC Applications* have more information on design criteria, acoustic treatment, and selection procedures.

### 3.3.5 Effect of Blades

Blades can affect grille performance if their depth is at least equal to the distance between the blades. If the blade ratio is less than one, effective control of the airstream discharged from the grille by means of the blades is impossible. Increasing the blade ratio above two has little or no effect, so blade ratios should be between one and two.

A grille discharging air uniformly forward, blades in the straight position, has a spread of 14 to 24 degrees, depending on the type of outlet, duct approach, and discharge velocity. Turning the blades influences the direction and throw of the discharged airstream.

A grille with diverging blades, vertical blades with uniformly increasing angular deflection from the centerline to a maximum of each end of 45 degrees, has a spread of about 60 degrees which reduces the throw considerably. With increasing divergence, the quantity of air discharged by a grille for a given upstream total pressure decreases.

A grille with converging blades, vertical blades with uniformly decreasing angular deflection from the centerline, has a slightly higher throw than a grille with straight blades, but the spread is approximately the same for both settings. The airstream converges slightly for a short distance in front of the outlet and then spreads more rapidly than air discharged from a grille with straight blades.

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In addition to vertical blades that normally spread the air horizontally, horizontal blades may spread the air vertically. Spreading the air vertically risks hitting beams or other obstructions or blowing primary air at excessive velocities into the occupied zone. Vertical deflection may increase adherence to the ceiling and reduce the drop.

In spaces with exposed beams, the outlets should be located below the bottom of the lowest beam level, preferably low enough to employ an upward air path. The air path should be upward sufficiently to miss the beams and to prevent the primarily or induced airstream from striking furniture and obstacles and producing objectionable drafts.

### 3.3.6 Duct Approaches to Outlets

How the airstream is introduced into the outlet is important. To obtain the correct air diffusion, the velocity of the airstream must be as uniform as possible over the entire connection to the duct and must be perpendicular to the outlet face. No air outlet can compensate for airflow from an improper duct approach.

A wall grille installed at the end of a long horizontal duct and a ceiling outlet at the end of a long vertical duct, receives the air perpendicularly and at a uniform velocity over the entire duct cross section, if the system is designed carefully. However, few outlets are installed in this way. Most sidewall outlets are installed either at the end of vertical ducts or in the side of horizontal ducts. In addition, most ceiling outlets are attached either directly to the bottom of horizontal ducts or to special vertical takeoff ducts that connect the outlet with the horizontal duct. For these cases, special devices for directing and equalizing the airflow are necessary for proper direction and diffusion of the air.

### 3.3.7 Stack Heads

Tests conducted with stack heads indicated that splitters or turning vanes in the elbows at the top of the vertical stacks were needed, regardless of the shape of the elbows, whether rounded, square, or expanding. *Note: Splitters and turning vanes are beneficial at the top of stack heads to align airflow into the outlets. However, turning vanes are not normally used in small, low-velocity duct fittings (even square-throated elbows) except immediately prior to outlets.* Cushion chambers at the top of stack heads are not beneficial. Figure 3-8 shows the direction of flow, diffusion, and velocity as measured 12 in. (300 mm) from opening of the air for various stack heads tested, expanding from a 14 × 6 in. (350 × 150 mm) stack to a 14 × 9 in. (350 × 225 mm) opening without a grille. The air velocity for each was 500 fpm (2.5 m/s) in the stack below the elbow but the direction of flow and the diffusion pattern indicate performance obtained with non-expanding elbows of similar shapes, for velocities from 200 to 400 fpm (1 to 2 m/s).

In tests conducted with 3 × 10 in. (75 × 250 mm), 4 × 9 in. (100 × 225 mm), and 6 × 6 in. (150 × 150 mm), side outlets in a 6 × 20 in. (150 × 500 mm) horizontal duct at duct velocities of 200 to 1400 fpm (1 to 7 m/s) in the horizontal duct section, multiple-curved deflectors produced the best flow characteristics. Vertical guide strips in the outlet were not as effective as curved deflectors. A single scoop-type deflector at the outlet did not improve the flow pattern obtained from a plain outlet and was undesirable.

### 3.3.8 Branch Takeoffs

SMACNA duct fitting research at ETL Laboratories, and the SMACNA "bubble" airflow research video, have shown from duct traverse pressure readings and visual observation using entrained soap-bubbles, that airflow in branch ducts has a non-uniform profile. Regardless of the type of device used and the type of tap or branch fitting, most of the airflow is concentrated in the downstream portion of the branch duct. The upstream portion of the branch duct contains either reverse flow back toward the main duct or has a swirling turbulence.

Designers are especially cautioned to avoid the use of so called extractors–devices with turning vanes that are intended to capture air in the main duct and route it into branch ducts. The airflow research clearly illustrated that extractors create a number of problems-increased static pressure, excessive turbulence in branch ducts, balancing, etc. The airflow research found that the best branch fitting takeoff is a simple 45 degree tap.

## 3.4 OUTLET LOCATION

The building's use, size, and construction type must be considered in designing the air distribution system and in selecting the type and location of the supply outlets. Location and selection of the supply outlets is further influenced by the interior design of the building, local sources of heat gain or loss and outlet performance and design.

Local sources of heat gain or loss promote convection currents or cause stratification and may determine both the type and location of the supply outlets.

An HVAC system operating in the cooling mode performs best when generated heat is removed at its

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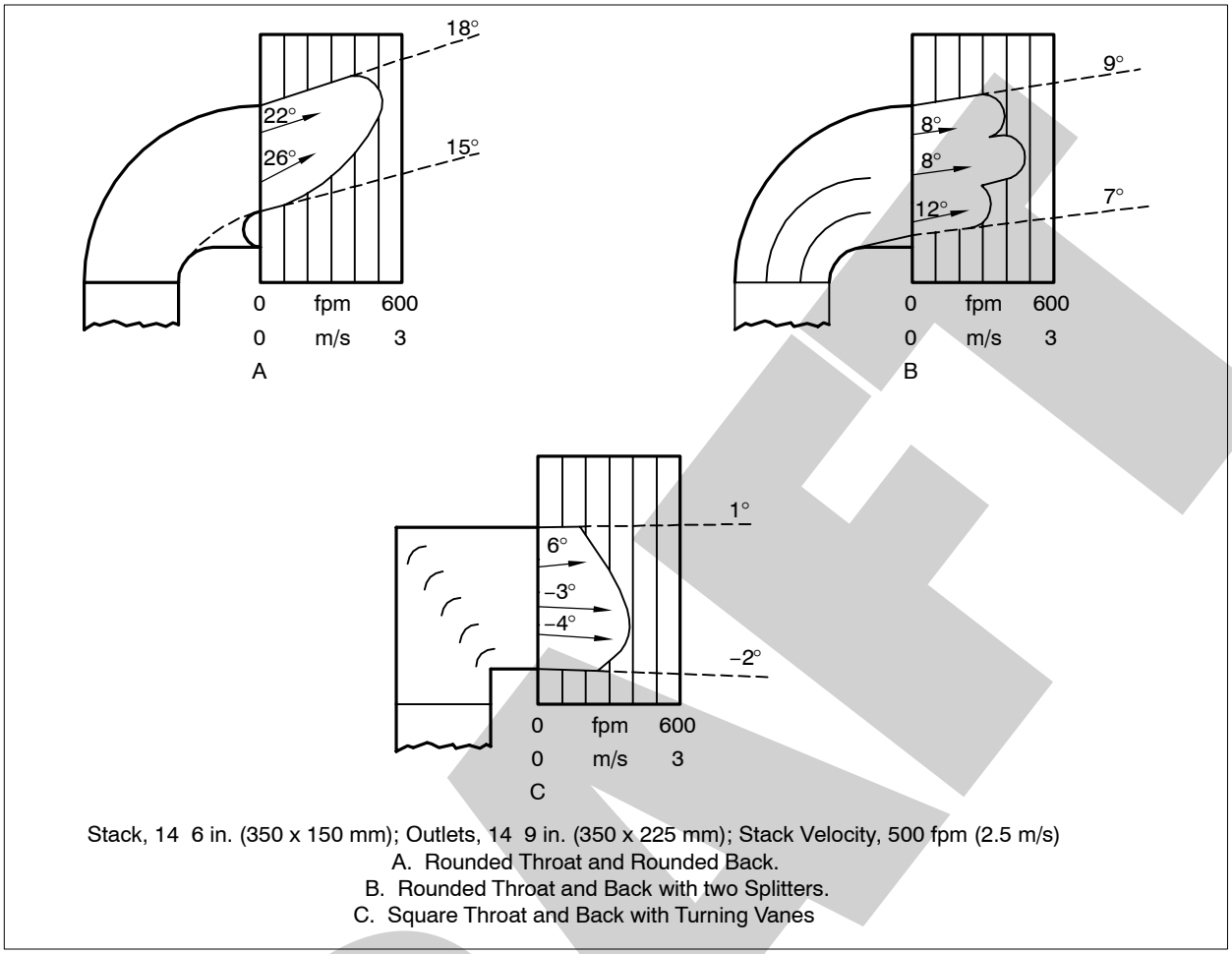


Stack, 14 6 in. (350 x 150 mm); Outlets, 14 9 in. (350 x 225 mm); Stack Velocity, 500 fpm (2.5 m/s)
A. Rounded Throat and Rounded Back.
B. Rounded Throat and Back with two Splitters.
C. Square Throat and Back with Turning Vanes

**FIGURE 3–8 OUTLET VELOCITY AND AIR DIRECTION DIAGRAMS FOR STACK HEADS WITH EXPANDING OUTLETS**

source, rather than distributed throughout the conditioned space. Heat from solar and miscellaneous loads, including machinery and floor or desk-mounted lamps, are difficult to remove at the source.

However, return air flowing over ceiling-mounted lighting fixtures keeps most of this heat from being distributed into the conditioned space. Combination return air/lighting fixtures increase the HVAC system efficiency and improve light output and lamp life. The manufacturers of fixtures, ceiling grids, and grilles provide airflow rate, pressure drop, and heat removal performance information of their products.

Outlets should be located to neutralize any undesirable convection currents set up by a concentrated load. If a concentrated heat source is located at the occupancy level of the room, the heating effect can be counteracted by directing cool air toward the heat source or by locating an exhaust or return grille adjacent to the heat source. The second method is more economical. Where lighting loads are in the heavy range of 5 W/ft$^2$ (54 W/m$^2$) and ceilings are above 15 ft (4.6 m), the outlets should be located below the lighting load, and the stratified warm air should be removed by an exhaust or return fan. An exhaust fan is recommended if the wet-bulb temperature of the air is above that of the outdoors. A return fan is recommended if it is below this temperature.

These methods reduce the requirements for supply air. Enclosed lights produce more savings than exposed lights, since a considerable portion of the energy is radiant.

Based upon the analysis of ASHRAE outlet performance tests by Straub et al. (1955, 1957) the following are selection consideration for outlet types in Groups A to E, s*ee* Figures 3-9 to 3-13.

### 3.4.1 Group A Outlets

Outlets mounted in or near the ceiling with horizontal air discharge should not be used with temperature dif-

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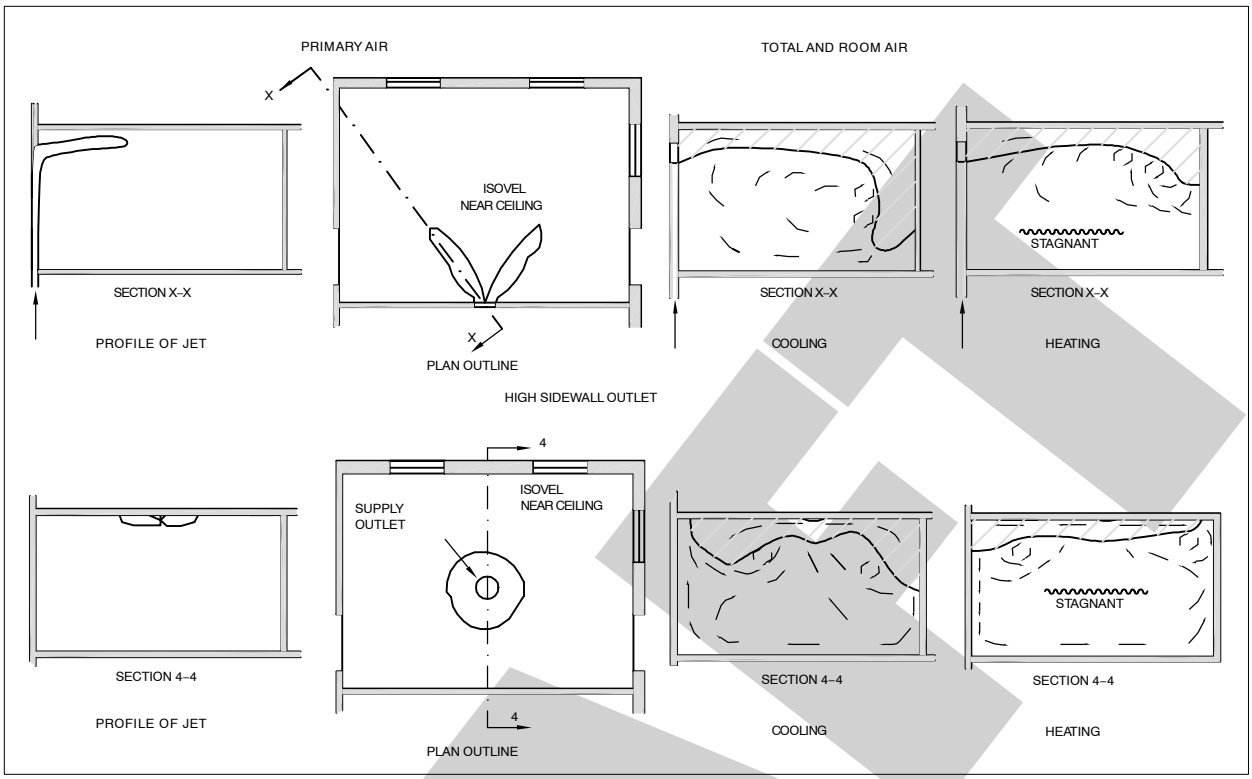


**FIGURE 3–9 AIR MOTION CHARACTERISTICS OF GROUP A OUTLETS**

ferentials exceeding 25 degrees F (14 degrees C) during heating. Consequently, Group A outlets should be used for heating in buildings located in regions where winter heating is only a minor problem, and in northern latitudes solely for interior spaces. However, these outlets are particularly suited for cooling and can be used with high airflow rates and large temperature differentials. They are usually selected for their cooling characteristics.

The performance of these outlets is affected by various factors. Blade-deflection settings reduce throw and drop by changing air from a single straight jet to a wide-spreading or fanned-out jet. Accordingly, a sidewall outlet with 0 degree deflection has a longer throw and more drop than a ceiling diffuser with a single 360 degree angle of deflection. Sidewall grilles and similar outlets with other deflection settings may have performance characteristics between these two extremes.

Wide deflection settings also cause a surface effect that increases the throw and decreases the drop. To prevent smudging, the total air should be directed away from the ceiling, but this is rarely practical except for very high ceilings. For optimum air diffusion in areas without high ceilings total air should scrub the ceiling surface.

Drop increases and throw decreases, with larger cooling temperature differentials. For a constant temperature differential, airflow rate affects drop more than velocity. Therefore, to avoid drop, several small outlets may be better in a room than of one large outlet.

| Type of Outlet | Floor Space | | Approximate Maximum Air Changes/Hour For 10 Ft. (3 m) Ceiling Height |
|---|---|---|---|
| | CFM/per Sq. Foot | l/s per Sq. Meter | |
| Grilles & Registers | 0.6 to 1.2 | 3 to 6 | 7 |
| Slot Diffusers | 0.8 to 2.0 | 4 to 10 | 12 |
| Perforated Panel | 0.9 to 3.0 | 5 to 15 | 18 |
| Ceiling Diffuser | 0.9 to 5.0 | 5 to 25 | 30 |
| Perforated Ceiling | 1.0 to 10.0 | 5 to 50 | 60 |

**Table 3–6 Guide for Selection of Supply Outlets**

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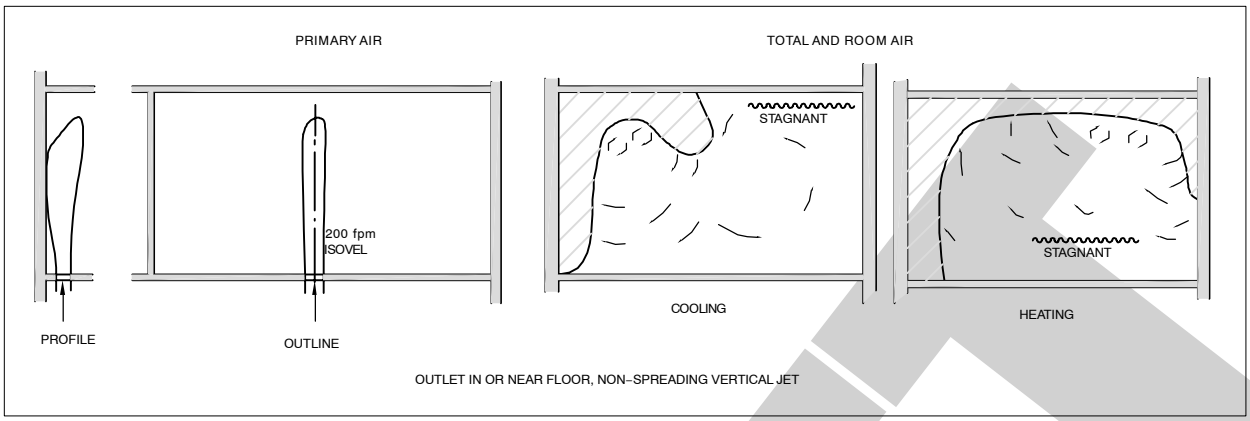


**FIGURE 3–10 AIR MOTION CHARACTERISTICS OF GROUP B OUTLETS**

With "Isothermal Jets", the throw may be selected for a portion of the distance between the outlet and wall or, preferably, for the entire distance. For outlets in opposite walls, the throw should be one-half the distance between the walls. Following the above recommendations, the air drops before striking the opposite wall or the opposing airstream. To counteract specific sources of heat gain, or provide higher air motion in rooms with high ceilings, it may be necessary to select a longer throw. In no case should the drop exceed the distance from the outlet to the 5 ft (2 m) level. Refer to Chapter 33 of the 2005 ASHRAE *Fundamentals Handbook* for a more detailed review of room airflow patterns, jets, and air diffusion.

To maintain maximum ventilation effectiveness with ceiling diffusers, throws should be kept as long as possible. With VAV designs, some overthrow at maximum design volumes will be desirable. The highest induction can be maintained at reduced flows. Adequate induction by a ceiling-mounted diffuser prevents short-circuiting of unmixed supply air between the supply outlet and ceiling-mounted return.

### 3.4.2 Group B Outlets

In selecting Group B outlets, it is important to provide enough throw to project the air high enough for proper cooling in the occupied zone. An increase of supply air velocity improves air diffusion during both heating and cooling. Also, during heating and cooling, a terminal velocity of about 150 fpm (0.75 m/s) is found at the same distance from the floor. Therefore, outlets should be selected with throw based on terminal velocity of 150 fpm (0.75 m/s).

With outlets installed near the exposed wall, the primary air is drawn toward the wall, resulting in a surface effect. This scrubbing of the wall increases heat gain or loss. To reduce scrubbing, outlets should be installed some distance from the wall or the supply air should be deflected at an angle away from the wall. However, the distance should not be too large, nor the angle too wide, to prevent the air from dropping into the occupied zone before maximum projection has been reached. A distance of 6 in. (150 mm) and an angle of 15 degrees are satisfactory.

These outlets do not counteract natural convection currents unless sufficient outlets are installed around the perimeter of the space, preferably in locations of greatest heat gain or loss (under windows). The effect of drapes and blinds must be considered with outlets installed near windows. If installed correctly, outlets of this type handle large airflow rates with uniform air motion and temperatures.

### 3.4.3 Group C Outlets

Group C outlets can be used for heating, even with severe heat-load conditions. High supply velocities produce better room-air diffusion than lower velocities but velocity is not critical in selecting these units for heating.

For cooling, the outlets should be used with temperature differentials of less than 15 degrees F (8 degrees C) to achieve the required projection. With higher temperature differentials, supply-air velocity is not sufficient to project the total air up to the desired level.

These outlets have been used successfully for residential heating but they may also offer a solution for applications where heating requirements are severe and cooling requirements are moderate.

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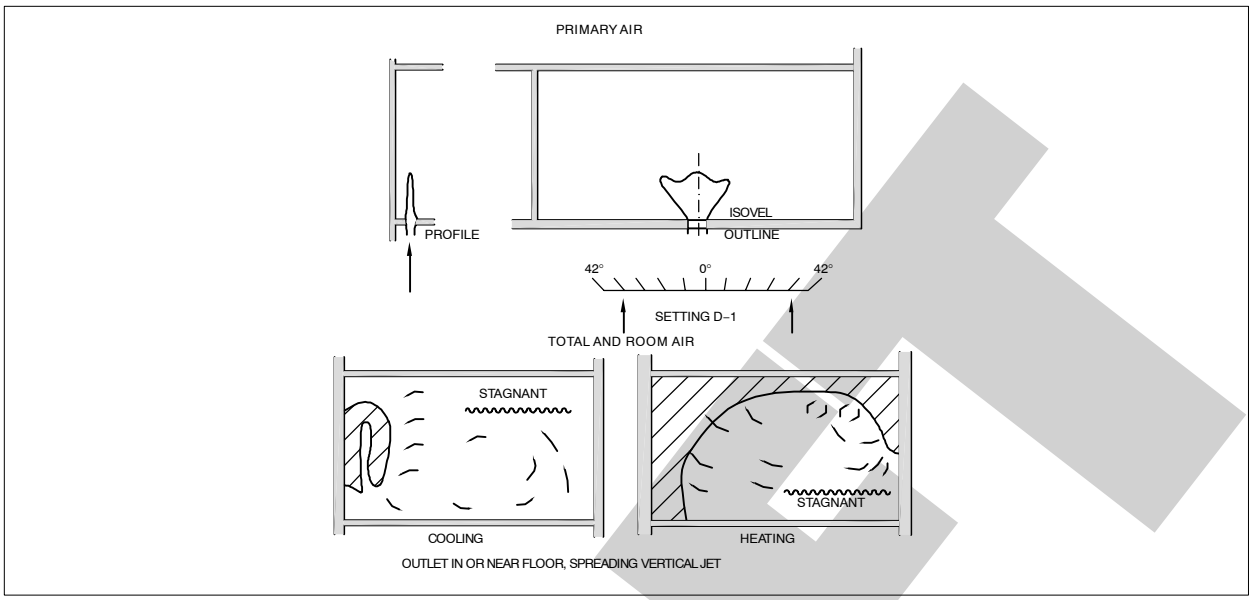


**FIGURE 3–11 AIR MOTION CHARACTERISTICS OF GROUP C OUTLETS**

### 3.4.4 Group D Outlets

Group D outlets direct high-velocity total air into the occupied zone and, therefore, are not recommended for comfort applications, particularly for summer cooling. If used for heating, outlet velocities should not be higher than 300 fpm (1.5 m/s) or air velocities in the occupied zone will be excessive. These outlets have been applied successfully to process installations where controlled air velocities are desired.

### 3.4.5 Group E Outlets

The heating and cooling diagrams for Group E outlets show different throws that become critical considerations in selecting and applying these outlets. Since the total air enters the occupied zone for both cooling and heating, outlets are used for either cooling or heating, but seldom for both.

During cooling, temperature differential, supply air velocity, and airflow rate have considerable influence

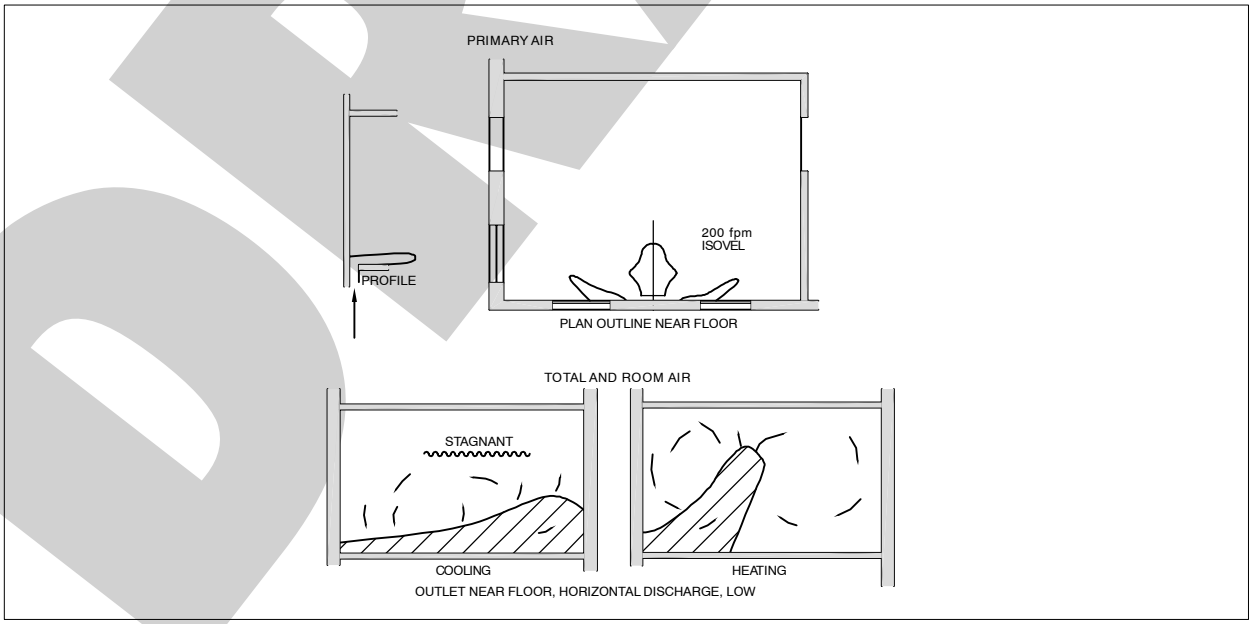


**FIGURE 3–12 AIR MOTION CHARACTERISTICS OF GROUP D OUTLETS**

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on projection. Therefore, low values of each should be selected.

During heating, selection of the correct supply air velocity is important to project the warm air into the occupied zone. Temperature differential is also critical because a small temperature differential reduces variation of the throw during the cyclic operation of the supply-air temperature. Blade setting for deflection is as important here as for Group B and C outlets.

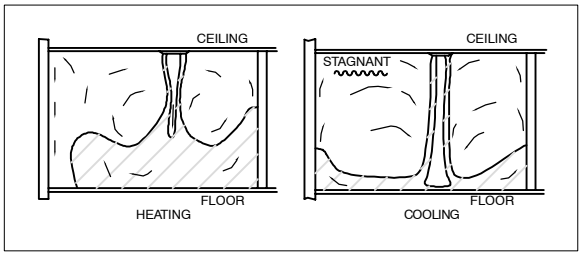


**FIGURE 3–13 AIR MOTION CHARACTERISTICS OF GROUP E OUTLETS**

## 3.5 OUTLET CRITERIA

Outlets with higher induction rates move air short distances but have rapid-temperature equalization. Ceiling diffusers with radial patterns have shorter throws and obtain more rapid temperature equalization than slot diffusers. Grilles, which have long throws, have the lowest diffusion and induction rates. Therefore, in those cases, round or square ceiling diffusers deliver more air to a given space than grilles and slot diffuser outlets that require room velocities of 25 to 35 fpm (0.13 to 0.18 m/s). In some spaces, higher room-velocities can be tolerated or the ceilings may be high enough to permit a throw long enough to result in the recommended room-air velocities.

Outlets with high induction characteristics can also be used successfully in air-conditioning systems with low supply air temperatures and resulting high temperature differentials between room air temperature and supply air temperatures. Therefore, ceiling diffusers may be used in systems with cooling temperature differentials up to 30 to 35 degrees F (17 to 19 degrees C) and still provide satisfactory temperature equalizations within the spaces. Slot diffusers may be used in systems with cooling temperature differentials as high as 25 degree F (14 degrees C). Grilles may generally be used in well-designed systems with cooling temperature differentials up to 20 degrees F (11 degrees C).

## 3.6 GRILLE AND REGISTER APPLICATIONS

Properly selected grilles operate satisfactorily from high-side and perimeter locations in the sill, or floor. Ceiling-mounted grilles which discharge the airstream down are generally not acceptable in comfort air-conditioning installations in interior zones and may cause drafts in perimeter applications.

### 3.6.1 High Side Walls

The use of a double-deflection grille usually provides the most satisfactory solution. The vertical face blades of a well-designed grille deflect the air approximately 50 degrees to either side and amply cover the conditioned space. The rear blades deflect the air at least 15 degrees in the vertical plane, which is ample to control the elevation of the discharge pattern.

### 3.6.2 Perimeter Installations

The grille selected must fit the specific job. When small grilles are used, adjustable-blade grilles improve the coverage of perimeter surfaces. Where the perimeter surface can be covered with long grilles, the fixed-blade grille is satisfactory. Where grilles are located more than 8 in. (200 mm) from the perimeter surface, it is usually desirable to deflect the air stream toward the perimeter wall. This can be done with adjustable- or fixed-deflecting-blade grilles.

### 3.6.3 Ceiling Installations

Ceiling installations generally are limited to grilles having curved blades, which provide a horizontal pattern. Curved-blade grilles may also be used satisfactorily in high-sidewall or perimeter installations.

### 3.6.4 Slot Diffuser Applications

A slot diffuser is an elongated outlet consisting of single or multiple numbers of slots. It is usually installed in long, continuous lengths. Outlets with dimensional aspect ratios of 25 to 1 or greater and a maximum height of approximately 3 in. (80 mm), generally meet the performance criteria for slot diffusers.

### 3.6.5 High Side Wall Installation

The perpendicular-flow slot diffuser is best suited to high side wall installations and perimeter installations in sills, and floors. The air discharged from a perpendicular-slot diffuser will not drop if the diffuser is located within 6 to 12 in. (150 to 300 mm) of the ceiling and is long enough to establish surface effect. Under

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these conditions, air travels along the ceiling to the end of the throw. If the slot diffuser is mounted 1 to 2 ft (300 to 600 mm) below the ceiling, an outlet that deflects the air up to the ceiling must be used to achieve the same result. If the slot is located more than 2 ft (600 mm) below the ceiling, premature drop of cold air into the occupied zone will probably result.

### 3.6.6 Ceiling Installation

The parallel-flow slot diffuser is ideal for ceiling installation because it discharges across the ceiling. The perpendicular-flow slot diffuser may be mounted in the ceiling; however, the downward-discharge pattern may cause localized areas of high air motion. This device performs satisfactorily when installed adjacent to a wall or over an unoccupied or temporarily-occupied area. Care should be exercised in using perpendicular-flow slot diffuser in a downward-discharge pattern because variations of supply-air temperature cause large variations in throw.

### 3.6.7 Sill Installation

The perpendicular-flow slot diffuser is well suited to sill installation, but it may also be installed in the floor. When the diffuser is located within 8 in. (200 mm) of the perimeter wall, the discharged air may be either directed straight toward the ceiling or deflected slightly toward the wall. When the diffuser distance from the wall is greater than 8 in. (200 mm), the air should generally be deflected toward the wall at an angle of approximately 15 degrees. Deflections as great as 30 degrees may be desirable in some cases. The air should not be deflected away from the wall into the occupied zone. To perform satisfactorily, outlets of this type must be used only in installations with carefully designed duct and plenum systems.

Slot diffusers are generally equipped with accessory devices for uniform supply-air discharge along the entire length of the slot. While accessory devices help correct the airflow pattern, proper approach conditions for the airstream are also important for satisfactory performance. When the plenum supplying a slot diffuser is being designed, the traverse velocity in the plenum should be less than the discharge velocity or as recommended by the manufacturer and experience.

If tapered ducts are used for introducing supply air into the diffuser, they should be sized to maintain a velocity of approximately 500 fpm (2.5 m/s) and tapered to maintain constant static pressure.

### 3.6.8 Air–Light Fixtures

Slot diffusers having a single-slot discharge, and nominal 2, 3, and 4 ft (600, 900, and 1200 mm) lengths are available for use in conjunction with recessed fluorescent light troffers. A diffuser mates with a light fixture and is entirely concealed from the room. It discharges air through suitable openings in the fixture and is available with fixed or adjustable air-discharge patterns, air-distribution plenum, inlet dampers for balancing, and inlet collars suitable for flexible duct connections. Light fixtures adapted for slot diffusers are available in styles to fit common ceiling constructions. Various slot diffuser and light fixture manufacturers can furnish products compatible with each other's equipment.

## 3.7 CEILING DIFFUSER APPLICATIONS

### 3.7.1 Ceiling Installations

Ceiling diffusers should be mounted in the center of the space they serve, when they discharge the supply air in all directions. Multi-pattern diffusers can be used in the center of the space or adjacent to partitions, depending on the discharge pattern. By using different inner assemblies, their air pattern can be changed to suit particular requirements.

### 3.7.2 Side Wall Installations

Half-round diffusers, when installed high in interior side walls, should generally discharge the air toward the ceiling.

### 3.7.3 Exposed Duct Installation

Some ceiling diffuser types, particularly stepped-down units, perform satisfactorily on exposed ducts. Spiral and perforated-round duct are typical for exposed-duct applications. Consult manufacturers' catalogs for specific types.

### 3.7.4 Adjustable Pattern Diffusers

Surface effect is important in the performance of adjustable-pattern diffusers. In fact, this effect is so pronounced that usually only two discharge patterns are possible with adjustable-pattern diffusers mounted directly on the ceiling. When the diffuser is changed from a horizontal-pattern position to the down-blow-pattern position, the surface effect maintains the horizontal-discharge pattern until the discharge-air stream is effectively deflected at the diffuser face, resulting in a vertical pattern. However, when adjustable-pattern

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ceiling diffusers are mounted on exposed ducts and no surface effect exists, the air may assume any pattern between horizontal and vertical discharge. Directional or segmented horizontal air patterns can usually be obtained by adjusting internal baffles or deflectors.

| Type | Characteristics | Applications |
|---|---|---|
| Fixed blade grille | Single set of vertical or horizontal blades | Long perimeter grille installations |
| Adjustable single deflection blade grille | Single set of vertical or horizontal adjustable blades | Sidewall installation where single plane air deflection is required |
| Adjustable double deflection blade grille | One set of vertical and one set of horizontal adjustable blades | Preferred grille for sidewall installation provides both horizontal and vertical air deflection |
| Stamped plate grilles | Stamped from single sheet of metal with square, round or ornamental designed openings | No adjustment of air deflection possible. Use for architectural design purposes only |
| Variable area grille | Similar to adjustable double deflection blade grille with means to effectively vary the discharge area | Use with variable volume system to minimize variation of throw with variable supply air volume |
| Curved blade grilles | Curved blades to provide horizontal air pattern | Ceiling installation<br>High sidewall installation<br>Perimeter installation |
| Perpendicular-flow slot diffuser | Generally 25 to 1 dimensional aspect ratio with maximum height of 3 in. (75 mm) | High sidewall installation<br>Perimeter installation in sills, curbs and floors |
| Parallel-flow slot diffuser | Generally 25 to 1 dimensional aspect ratio with maximum height of 3 in. (75 mm) | Ceiling installation |
| Air light fixture slot diffuser | Use in conjunction with recessed fluorescent light fixtures with fixed or adjustable air discharge patterns | Ceiling installation–<br>Order to match light fixture |
| Multi-passage round ceiling diffuser | Series of flaring rings or louvers forming series of concentric air passages | Install in center of area served |
| Multi-passage square and rectangular ceiling diffuser | Series of flaring rings or louvers forming series of concentric air passages | Install in center of area served |
| Adjustable pattern round ceiling diffuser | Series of flaring rings or louvers forming series of concentric air passages. Air discharge pattern adjustable from horizontal to vertical or down blow pattern | Install for control of diffuser discharge pattern or where specific requirement to direct airflow pattern either horizontal or vertical |
| Adjustable pattern square and rectangular ceiling diffuser | Series of flaring rings or louvers forming series of concentric air passages. Air discharge pattern adjustable from horizontal to vertical or down blow pattern | Install for control of diffuser discharge pattern or where specific requirement to direct airflow pattern either horizontal or vertical |
| Multi-pattern square and rectangular ceiling diffuser | Special louvers discharge air in one or more directions | Install in center of area served or adjacent to partitions. Set pattern according to flow requirements |
| Half round diffuser | Matches round diffuser | Install in ceiling adjacent to partition or high sidewall |
| Supply and return concentric diffuser | Combination diffuser with return grille in center of diffuser | Install in center of area served |
| Light fixture air diffuser combination | Combination diffuser-light fixture | Ceiling installation combined with light fixture pattern |

**Table 3–7 Supply Air Outlet Types**

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## 3.8 OUTLETS IN VARIABLE AIR VOLUME (VAV) SYSTEMS

The performance of a particular outlet or diffuser is generally independent of the terminal box that is upstream. For a given supply-air volume and temperature differential (to meet a particular load), a standard outlet does not recognize whether the terminal box is of a constant volume, variable volume, or induction type. However, any diffuser, or system of diffusers, gives optimum air diffusion at some particular load condition and air volume.

In a variable air volume system, the performance of outlets with regard to throw, room velocity, and noise levels will vary greatly with the discharge volume. A volume variation of 25 to 35 percent is generally effective in controlling the load, without substantial adverse effect on the performance of properly selected outlets. When areas are unoccupied, a volume variation of up to 50 percent is permissible. Specially designed outlets can be used that will perform with air volumes substantially below half of the design volume. This will still allow the desired space temperature to be maintained. Outlets should be selected that are designed to perform within the limits of the variable air volume system parameters.

## 3.9 INLET CRITERIA

### 3.9.1 General

Return-air inlets may either be connected to a duct or be simple vents that transfer air from one area to another. Exhaust-air inlets remove air directly from a building and are always connected to a duct. Whatever the arrangement, inlet size, and configuration determine velocity and pressure requirements for the required airflow.

In general, the same type of equipment, grilles, slot diffusers, and ceiling diffusers used for supplying air may also be used for air return and exhaust. Inlets do not require the deflection, flow equalizing, and turning devices necessary for supply outlets. However, volume dampers installed in the return branch ducts are necessary to balance the airflows in the return-air duct system.

### 3.9.2 Types of Inlets

a. Adjustable-Blade Grilles: The same grilles used for the air supply are used to match the deflection setting of the blades with that of the supply outlets.

b. Fixed-Blade Grills: This grille is the most common return-air inlet. Blades are straight or set at a certain angle, the latter being preferred when appearance is important.

c. V-Blade Grille: The V-blade grille, with blades in the shape of inverted V's stacked within the grille frame has the advantage of being sight-proof. It can be viewed from any angle without detracting from appearance. Door grilles are usually V-blade grilles. The capacity of the grille decreases with increased sight tightness.

d. Lightproof Grille: A lightproof grille is used to transfer air to or from darkrooms. The blades of this type of grille form a labyrinth.

e. Stamped Grille: Stamped grilles are frequently used as return and exhaust inlets, particularly in restrooms and utility areas.

f. Diffusers: Ceiling and slot diffusers may also be used as return and exhaust inlets.

g. Filter Return Grilles: Return grilles with an integral filter are typically used for residential HVAC as the primary filter but these can also be used in commercial HVAC as a prefilter. The primary benefit is to "catch" airborne dust before it enters the return system or to capture dust in areas where unusual amounts of dust are generated-like workshops. In commercial applications, filter return grilles would act as a supplement to the primary filters and should not be used instead of a high-efficiency filter system.

### 3.9.3 Selection Procedures

Select return-air and exhaust-air inlets to suit architectural design requirements including appearance, compatibility with supply outlets, and space available for installation of inlets and ductwork. Generally, inlets should be installed to return room air of the greatest temperature differential that collects in the stagnant air areas. The location of return and exhaust inlets will not compensate for ineffective supply air distribution.

The selection of return and exhaust inlets depends on the velocity in the occupied zone near the inlet, the permissible pressure drop through the inlet, and the noise level.

Velocity: Control of the room air motion to maintain a comfort condition depends on proper supply outlet selection. The effect of airflow through return inlets on

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air movement in the room is slight. Air handled by the inlet approaches the opening from all directions, and its velocity decreases rapidly as the distance from the opening increases. Table 3-9 shows recommended return air inlet face velocities.

Permissible Pressure Drop: Permissible pressure drop depends on the choice of the designer. Proper pressure drop allowances should be made for control or directive devices.

Noise: The problem of return-air inlet noise is the same as that for supply outlets. In computing resultant room noise levels from the operation of an air conditioning system, the return inlet must be included as a part of the total grille area. The major difference between supply outlets and return inlets are the frequent installation of the latter near ear level. When they are so located, the return-inlet velocity should not exceed 75 percent maximum of the permissible outlet velocity.

Application: Be careful not to locate a return-air inlet directly in the primary airstream from the supply outlet. To do so will short circuit the supply air back into the return without mixing with room air to obtain desired room-air temperature.

| Group | Type | Mounting | Discharge Direction | Characteristics | |
|---|---|---|---|---|---|
| | | | | Cooling | Heating |
| A | High Sidewall Grilles | Ceiling, High Sidewall | Horizontal | Good mixing with warm room air. Minimum temperature variation within room. Particularly suited to cooling applications | Large amount of stagnant air near floor. In interior zones where loading is not severe, stagnant air is practically non-existent |
| | Sidewall Diffusers | | | | |
| | Ceiling Diffusers | | | | |
| | Slot Diffusers (Parallel Flow) | | | | |
| | Variable Area Grille | Ceiling, High Sidewall | Horizontal Specially adapted for variable volume systems | Maintain design air distribution characteristics as air volume changes | Maintain design air distribution characteristics as air volume changes |
| | Variable Area Diffuser | | | | |
| B | Floor Grilles | Floor, Low Sidewall, Sill | Vertical Non spreading Air Jet | Small amount of stagnant air generally above occupied zone | Small amount of stagnant air than Group A outlets |
| | Baseboard Units | | | | |
| | Fixed Bar Grilles | | | | |
| | Linear Grilles | | | | |
| C | Floor Grilles | Floor, Low Sidewall, Sill | Vertical Spreading Air Jet | Large amount of stagnant air than Group B outlets | Smaller amount of stagnant air than Group B outlets - particularly suited to heating applications |
| | Adjustable Bar Grilles | | | | |
| | Linear Diffusers | | | | |
| D | Baseboard Units Grilles | Floor, Low Sidewall | Horizontal | Large amount of stagnant air above floor in occupied zone - recommended for comfort cooling | Uniform temperature throughout area. Recommended for process applications |
| E | Ceiling Diffusers | Ceiling, High Sidewall | Vertical | Small amount of stagnant air near ceiling. Select for cooling only applications | Good air distribution. Select for heating only applications |
| | Linear Grilles | | | | |
| | Grilles | | | | |
| | Slot Diffusers (Vertical Flow) | | | | |
| | Sidewall Diffusers | | | | |

**Table 3–8 Supply Air Outlet Performance**

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## 3.10 EXHAUST OUTLETS

Exhaust outlets located in walls and doors, depending on elevation, have the characteristics of either floor or ceiling returns. In large buildings with many small rooms, the return air may be brought through door grilles or door undercuts into the corridors and then to a common return or exhaust. However, most fire codes will not allow using a public corridor for the return-air path unless the building is sprinkled. The pressure drop through interior door returns should not be excessive, or the air diffusion to the rooms may be seriously unbalanced by opening or closing the doors. Outward leakage through interior doors or windows cannot be relied on for dependable results.

## 3.11 SPECIAL SITUATIONS

The designer should consider special situation requirements in locating return and exhaust inlets in special-use areas including bars, kitchens, lavatories, dining rooms, club rooms, and conference rooms. These normally should be located near or at the ceiling level to collect the warm air, odors, smoke, and fumes. Return-air inlets are typically prohibited in bathrooms and some kitchen areas.

## 3.12 AIR DISTRIBUTION SUMMARY

### 3.12.1 General

Approximate values of pressure-drop requirements for various types of air outlet and inlet devices may be found in Chapter 8. These values should be adequate for preliminary duct-design layout requirements.

The final duct design system calculations and layout must include a selection of each air-distribution outlet and inlet device using the following air-distribution product catalog data:

a. Pressure loss through outlet

b. Throw

c. Spread

d. Drop

e. Noise level

Refer to the engineering section of the air device catalog for an explanation of the proper use of the manufacturer's data for the devices to be used.

*CAUTION – All air outlet terminal devices located on each branch duct or duct run should be selected with similar pressure drops. Mixing outlets with different pressure drops on the same duct run may cause excessive airflow through the outlets with the lowest pressure drops. Using terminal device dampers to control the excessive air distribution may create unacceptable noise levels.*

Obstructions must be considered when selecting air outlet devices. As an example, outlets should be installed below the bottom of beams in beamed ceilings, or below surface-mounted or suspended light fixture to avoid deflection of the airstream. Outlets should be located to neutralize undesirable convection currents set up by concentrated cold air moving downward across a window or hot air moving upwards from a heat source.

Some outlet devices are of a unique, patented design and can only be furnished by one manufacturer. When a system is designed for competitive bidding, outlets should be chosen so that several manufacturers can furnish air-outlet devices acceptable to the designer.

### 3.12.2 Supply Outlets Summary

To summarize the procedure for supply outlet location and selection:

a. Determine room supply-air quantity from heating- and cooling-load calculations and design ventilation requirements.

b. Select type and quantity of outlets for each room evaluating:

   1. Outlet airflow
   2. Outlet throw pattern (performance)
   3. Building structural characteristics

c. Locate outlets to provide uniform room temperature using as uniform an air distribution pattern as possible.

d. Select proper outlet size from manufacturer's catalog data considering:

   1. Outlet airflow
   2. Discharge velocity and throw
   3. Distribution pattern
   4. Pressure loss
   5. Sound level

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| Inlet Location | Velocity Over Gross Inlet Area | |
|---|---|---|
| | Feet per Minute | Meters per Second |
| Above occupied zone | 800 Up | 4 Up |
| Within occupied zone, not near seats | 600 - 800 | 3 - 4 |
| Within occupied zone, near seats | 400 - 600 | 2 - 3 |
| Door or wall louvers | 200 - 300 | 1 - 1.5 |
| Undercut doors | 200 - 300 | 1 - 1.5 |

**Table 3–9 Recommended Return Air Inlet Face Velocities**

### 3.12.3 Accessories

Accessory devices should be chosen to obtain the desired design performance of air outlet and inlet devices, *see* Table 3-11.

### 3.12.4 Return and Exhaust Inlets Summary

To summarize the procedure for inlet location and selection:

a. Determine room return- and exhaust-air quantity from supply outlet selection requirements.

b. Select type and quantity of inlets for each room evaluating:

1. Inlet airflow
2. Inlet velocity
3. Architectural requirements/limitations

c. Locate inlets to enhance room-air circulation and to remove undesirable air considering air temperature and contamination.

d. Select proper inlet size from manufacturer's catalog data considering:

1. Inlet airflow
2. Inlet velocity
3. Pressure loss
4. Sound level, *see* Chapter 10

## 3.13 ROOM TERMINAL DEVICES

Air-distribution criteria will vary considerably in commercial and institutional buildings, as will temperature and humidity levels. People sitting with little activity require closer tolerances than those actively moving about, as discussed earlier. Spillover from open, refrigerated display equipment in supermarkets causes frequent complaints from customers.

| Type | Characteristics | Applications |
|---|---|---|
| Fixed blade grille | Fixed grille blades straight or set at certain angle for appearance to match supply outlets | Return, exhaust and transfer grilles |
| Adjustable blade grilles | Blade return to match supply outlets | Return, exhaust and transfer grilles |
| V-blade grille | Sight proof | Particulary suited for door louvers |
| Light proof grille | Light proof | Used for dark rooms |
| Stamped grilles | Match supply outlets | Return and exhaust grilles |
| Ceiling diffusers | Match supply outlets | Return and exhaust grilles |
| Slot diffusers | Match supply outlets | Return and exhaust grilles |
| Air light fixture | Match supply outlets | Return and exhaust grilles |
| Perforated face inlet | Match supply outlets | Return and exhaust grilles |
| Egg crate grille | Match supply outlets | Return and exhaust grilles |

**Table 3–10 Return and Exhaust Air Inlet Types**

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| Device | Characteristics | Comments |
|---|---|---|
| Opposed-blade volume damper | Volume adjustment to discharge air in series of jets without adversely deflecting airstream to one side of outlet | Behind grille (grille with damper called register) or diffuser to adjust air volume |
| Multi-shutter damper | Parallel blade damper will deflect airstream when damper partially open | Use to adjust air volume only when airstream deflection acceptable |
| Gang-operated turning vane (extractor) | Vanes pivot and remain parallel to duct airflow, creates turbulence in both branch duct and main duct | At branch duct connection equalize flow to grille or diffuser (Not recommended) |
| Individually adjusted turning vanes | 2 parallel sets of vanes-downstream set equalizes flow across collar-upstream set act as turning vanes | Designed to equalize flow but not serve as a damper |
| Slot diffuser damper | Integral equipment with slot diffuser | Minor volume adjustment |
| Slot diffuser flow equalizing vanes | Integral equipment with slot diffuser | Adjust discharge pattern of slot diffuser |
| Multi-louver round diffuser damper | Series of parallel blades | Adjust air volume |
| Opposed blade round diffuser damper | Series of pie shaped blades mounted in round frame | Adjust air volume |
| Diffused splitter damper | Single plate hinged at duct branch connection to outlet | Adjust air volume. Use only with equalizing device |
| Diffuser equalizing device | Individual adjusted blades | Use to provide uniform airflow to diffuser |
| Diffuser blank off baffle | Blank off section of diffuser | Use to prevent supply air from striking obstruction, such as a column, to reduce flow in given direction |
| Diffuser panel | Size to match ceiling tile size | Grid ceiling systems |
| Diffuser anti-smudge rings | Round, square or rectangular frame | To minimize ceiling smudging |
| Air-light fixture slot diffuser, plenum with damper, flex inlet collar | To attach to slot diffuser light fixture | For controlled air connection to light fixture slot diffuser |
| Linear grille blank-offs | Cap linear grille inlet | Inactivate sections of continuous linear grille |
| Linear grille plenum | Plenum attaches to linear grille section with collar for flex duct connection | To connect supply air to linear grille |
| Adapter | Square or rectangular connection to diffuser with round neck duct connection | To adapt square or rectangular diffuser neck to round duct connection |
| Plaster frames | Round, square or rectangular secondary plaster frame | Installed prior to plastering. Provides clean frame for easy installation of outlet or inlet device. |

**Table 3–11 Accessory devices**

The "Comfort Applications" section of the *ASHRAE Handbook - HVAC Applications* specifically addresses residential, commercial, and institutional HVAC systems.

### 3.13.1 Supply Air Outlets

The correct types of supply outlets, properly sized and located, control the air pattern within a given space to

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provide proper air motion and temperature equalization.

Supply air outlets can actually "pull" room air back into the supply air system due to the effect of negative pressure caused by air turbulence at the main to branch duct junction. This problem is common in exposed round ducts at outlet connections near the supply fan discharge and connections just after an elbow.

Accessories used with these outlets regulate the volume of supply air and control its flow pattern. For example, the outlet cannot discharge air properly and uniformly unless the air is conveyed to it in a uniform flow. Accessories may also be necessary for proper air distribution in a space, so they must be selected and used in accordance with the manufacturer's recommendations. Outlets should be sized to project air so its velocity and temperature are at acceptable levels in the occupied zone.

Outlets with high induction rates move air short distances but have rapid temperature equalization. Ceiling diffusers with radial patterns have shorter throws and obtain more rapid temperature equalization than slot diffusers. Grilles that have long throws have the lowest diffusion and induction rates. Therefore, in those cases, round or square ceiling diffusers deliver more air to a given space than grilles and slot diffuser outlets. In some spaces, higher room velocities can be tolerated or the ceilings may be high enough to permit a throw long enough to result in the recommended velocities within in occupied space.

Outlets with high induction characteristics can also be used advantageously in air-conditioning systems with low supply-air temperatures and consequently high-temperature differentials between room air temperature and supply-air temperatures. Therefore, ceiling diffusers may be used in systems with cooling temperature differentials of 30 to 35 degrees F (17 to 19 degrees C) and still provide satisfactory temperature equalization within the spaces. Slot diffusers may be used in systems with cooling temperature differentials as high as 25 degrees F (14 degrees C). Grilles may generally be used in well-designed systems with cooling temperature differentials up to 20 degrees F (11 degrees C).

## 3.14 SUPPLY AIR GRILLE AND REGISTER TYPES

### 3.14.1 Adjustable Bar Grille

The Adjustable Bar Grille is the most common type of grille used as a supply outlet. The single-deflection grille consists of a frame, enclosing a set of vertical or horizontal blades. Vertical blades deflect the airstream in the horizontal plane, while horizontal blades deflect the airstream in the vertical plane. The double-deflection grille has a second set of blades installed behind, and at right angles to, the face blades. This grille controls the airstream in both the horizontal and vertical planes.

### 3.14.2 Fixed Bar Grille

The Fixed Bar Grille is similar to the single-deflection grille except the blades are not adjustable. These blades may be straight or at an angle. The angle at which the air is discharged from this grille depends on the type of deflection blades and the set angle.

### 3.14.3 Stamped Grille

The Stamped Grille is stamped from a single sheet of metal to form a fretwork through which air can pass. Various designs are used varying from square or rectangular holes to intricate ornamental designs.

### 3.14.4 Variable Area Grille

A Variable Area Grille is similar to the adjustable, double-deflection grille but can vary the discharge area to achieve an air volume change (variable volume outlet) at constant pressure. This ensures the variation in throw is minimized for a given change in supply-air volume.

### 3.14.5 Under–Floor Diffusers

Diffusers designed for use with under-floor air distribution systems.

## 3.15 SUPPLY AIR CEILING DIFFUSER TYPES

### 3.15.1 Multi–Passage Ceiling Diffusers

A Multi-Passage Ceiling diffuser consists of a series of flaring rings or louvers which form a series of concentric air passages. They may be round, square, or rectangular. For easy installation, these diffusers are usually made in two parts; an outer shell with a duct collar and an easily-removable inner assembly.

### 3.15.2 Flush and Stepped–Down Diffusers

The Flush Diffusers, as the name implies, have all rings or louvers level with a plane surface. In a stepped-down diffuser, they project out, beyond the surface in steps.

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Common variations of this diffuser type are the *adjustable-pattern* and the *multi-pattern* diffuser. In the adjustable-pattern diffuser, the air-discharge pattern may be changed from a horizontal to a vertical or downblow pattern. Special construction of the diffuser or separate deflection devices allows adjustment. Multipattern diffusers are square or rectangular and have special louvers to discharge the air in one or more directions.

Other outlets available as standard equipment are half-round diffusers, supply and return diffusers, and light-fixture air diffuser combinations.

### 3.15.3 Perforated–Face Ceiling Diffusers

A Perforated-Face Ceiling Diffuser meets architectural demands for air outlets that blend into perforated ceilings. Each has a perforated-metal face with an open area of 10 to 50 percent that determines its capacity. Units are usually equipped with a deflection device to obtain multi-pattern, horizontal-air discharge.

### 3.15.4 Variable Area Ceiling Diffusers

Variable Area Ceiling Diffusers may be round, square or linear and have parallel or concentric passages, or a perforated face. In addition, they feature a means of varying the discharge area to achieve an air volume change at the outlet at a constant pressure so the variation in throw is minimized for a given change in supply-air volume.

### 3.15.5 Low–Temperature Air Diffusers

Low-temperature air diffusers are specially designed to maximize discharge and room air mixing when using a low-temperature air distribution system.

## 3.16 VAV AND THERMAL BOXES

In HVAC air distribution systems, the duct velocities and static pressures sometimes require special control and acoustical equipment for proper introduction of conditioned air into the space. Control equipment for high-velocity and low-velocity systems may be classified as *pressure reducing valves* or *terminal boxes*. Terminal boxes may be further classified as single-duct or dual-duct boxes, reheat boxes, variable air volume boxes, or ceiling-induction boxes. They provide all or some of the following functions: pressure reduction, volume control, temperature modulation, air mixing, and sound attenuation.

### 3.16.1 Pressure Reducing Valves

Pressure reducing or air valves consist of a series of gang-operated vane sections mounted within a rigid casing with a gasket to reduce air leakage as much as possible between the valve and duct. They are usually installed between a high-pressure trunk-duct and a low-pressure branch duct.

Pressure is reduced by partially closing the valve, which results in a high-pressure drop through the valve. This action generates noise, which must be attenuated in the low-pressure discharge duct. The length and type of duct lining depend on the level and frequency of noise to be attenuated.

Volume control is obtained by adjusting the valve manually, mechanically, or automatically. Automatic adjustment is achieved by a pneumatic or electric-mechanical motor actuated by a pressure regulator, thermostat, or some other element of an automation system.

Pressure-reducing values are generally equal in size to the low-pressure branch duct connected to the valve discharge. This arrangement provides minimum pressure drop with valves opened fully.

### 3.16.2 Terminal Boxes

The terminal box fulfills one or more of the following functions:

a. Controls the velocity, pressure, or temperature of the air.

b. Controls the rate of airflow.

c. Mixes airstreams of different temperatures or humidity.

d. Mixes, within the assembly, air at high velocity and high pressure with air from the treated space.

To achieve these functions, the terminal box assemblies may include a casing, mixing section, manual damper, heat exchanger, induction section, and flow-rate controller.

A terminal box commonly integrates a sound chamber to reduce noise generated by the damper or flow-rate controller reducing the inlet air to lower velocity airflow. The sound-attenuation chamber is typically lined with thermal and sound-insulating material and equipped with baffles. Special sound attenuation in the

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air discharge ducts is not usually required in smaller boxes.

Terminal boxes are typically identified by the function of their flow-rate controllers, which are generally categorized as *constant-flow-rate* or *variable-flow-rate* devices. They are further categorized as being *pressure-dependent*, where the airflow rate through the assembly varies in response to changes in system pressure or as *pressure-independent* where the airflow rate through the device does not vary in response to changes in system pressure.

Constant-flow-rate controllers may be of the *mechanical-volume-control* type, which are actuated by means of the static pressure in the primary duct system and require no outside source of power. Constant-flow-rate controllers may also be of the *pneumatic* or *electric-volume regulator* type, are actuated by an outside motive power, and typically require internal differential pressure sensing, selector devices, and pneumatic or electric motors for operation.

Variable-flow-rate controllers may be of the *mechanical pneumatic* or *electric-volume control* type, which incorporates a means to reset the constant-volume regulation automatically to a different control point within the range of the control device in response to an outside signal, such as a thermostat. Boxes with this feature are pressure-independent and may be used with reheat components. A modulating damper ahead of a constant-volume regulator may also obtain variable-flow-rate. This arrangement typically allows for variations in flow between high and low limits, or between a high limit and shutoff. These boxes are pressure dependent and volume limiting in function. Pneumatic-variable volume may be pressure independent, volume limiting, or pressure dependent, according to the equipment selected.

Terminal boxes can be further categorized as *system powered*, when the assembly derives all of the energy necessary for its operation from the supply air within the distribution system, or as *externally powered*, when the assembly derives part of all of the energy necessary for its operation from a pneumatic or electric outside source. In addition, assembles are *self-contained* with all necessary controls including actuators, regulators, motors, and thermostats, or as *non-self-contained* assemblies with part or all of the necessary controls for operation furnished by someone other than the assembly manufacturer. In this latter case, the controls may be mounted on the assembly by the manufacturer or may be mounted by others after delivery of the equipment.

## 3.17 TERMINAL BOX VARIATIONS

### 3.17.1 Reheat Boxes

Box reheat add sensible heat to the supply air. They cannot be used in some areas unless the added heat is "recovered heat" due to energy conservation codes. Water or steam coils, or electric resistance heaters, are located within or attached directly to the air-discharge end of the box. These boxes typically are single-duct and can be either constant or variable volume. However, if they are variable air volume (VAV), they must maintain some minimum airflow to accomplish the reheat function and provide code-required ventilation airflow. This type of equipment can accomplish localized individual reheat without a central equipment station or zone change.

### 3.17.2 Dual–Duct Boxes

Dual-Duct Boxes receive warm and cold air from separate air supply ducts in accordance with room requirements to provide room control without zoning. Pneumatic and electric-volume-regulated boxes often have individual modulating dampers and operators to regulate the amount of warm and cool air. When a single-modulating damper operator regulates the amount of both warm and cold air, a separate pressure-reducing damper or volume controller is needed to reduce pressure and limit airflow. Specially designed baffles may be required within the unit or discharge to mix varying amounts of warm and cold air and to provide uniform flow downstream. Dual-duct boxes can be equipped with constant-flow rate or variable-flow rate devices, to be either pressure-independent or pressure-dependent, to provide a number of volume and temperature control functions.

### 3.17.3 Ceiling–Induction Boxes: Air–to–Air

The Air-to-Air Ceiling-Induction Box provides either primary air, or a mixture of primary air, and relatively warm air to the conditioned space. It accomplishes this function by permitting the primary air to induce air from the ceiling plenum or by inducted return air from individual rooms. A single duct supplies primary air at a temperature cool enough to satisfy all zone cooling loads. The ceiling-plenum air inducted into the primary air is at a higher temperature than the room because heat from recessed lighting fixtures enters the plenum directly.

The induction box contains damper assemblies controlled by an actuator in response to the control device to vary the amount of cool primary air and induced air.

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As reduction in cooling is required, the primary airflow rate is gradually reduced and the induced-air rate is generally increased.

Reheat coils can be used in the primary air supply and in the induction opening to meet occasional interior and perimeter load requirements. Either hot water or electric reheat coils may be used.

### 3.17.4 Ceiling–Induction Boxes: Fan Assisted

A Variable Air Volume (VAV) System controls the dry-bulb temperature within a space by varying the volume of supply air rather than the supply-air temperature. VAV systems can be applied to interior or perimeter zones, with common or separate fan systems, common or separate air-temperature control, and with or without auxiliary heating devices. The variable air volume concept may apply to vary air volumes in the main system and to the control zones.

A space thermostat can control flow by varying a damper or volume-regulating device in the duct or a pressure-reducing box. Depending on the complexity of the air-distribution system, the throttling ratio, and the complexity of the part-load balancing, variable air volume may or may not be combined with fan or system static pressure controls.

Simple VAV systems typically are cooling only and have no requirement for simultaneous heating and cooling in various zones. Perimeter radiation, radiant heat, or an independent-constant-volume, variable-temperature air system normally handles heating requirements.

The fan system is designed to handle the largest simultaneous block load, not the sum of the individual peaks. As each zone peaks at a different time of day, it "borrows" the extra air from off-peak zones. This transfer of air from low-load to high-load zones occurs only in a true variable volume system. A system that runs excess air into a return air ceiling or bypasses it to the return duct may use variable volume temperature control principles; however, the total design air volume is the sum of the peaks since it is circulated on a constant-volume basis. These systems do not permit smaller fans and duct mains.

## 3.18 BASIC VAV SYSTEM DESIGN

### 3.18.1 Air Volume

If net room loads are not determined accurately, a variable volume system will have too much airflow, causing partial throttling at full load, and excessive throttling at part loads. Oversizing of fans, ducts, terminals, and control valves should be avoided.

Air movement below 10 fpm (0.05 m/s) is uncomfortable for some people so there has been concern over reduced airflow, particularly during the heating season. Minimum air circulation can be maintained by:

a. Varying supply-air temperatures on a system basis or on a reheated zone-by-zone basis.

b. Auxiliary heat in the room independent of the air system.

c. Individual zone fan recirculation and blending of varying amounts of supply with room air or supply with ceiling-plenum air.

d. Variable volume-induction unit recirculation.

When a variable air volume system is controlled by a constant-volume fan that returns surplus air to the return air ceiling plenum, the heat from lights and roof transmission becomes a room load. Constant-air-volume of this type requires each zone to satisfy the worst peak condition and the benefit of variable air volume is not realized.

### 3.18.2 Supply Air Conditions

When a common duct with temperature control supplies perimeter and interior spaces that require cooling during winter and include separate perimeter-radiation, it is often advisable to reset the supply air temperature higher as the ambient temperature decreases to avoid cold-draft sensations. However, the higher temperature increases the fan energy required to move the greater volume of air.

For maximum heating and cooling flexibility, the supply-air temperature must be low enough to handle the most severe winter cooling zone. The relationship between heater temperature and supply-air temperature schedules is important for flexibility and operating economy.

Full shutoff terminals used with either radiation or transmission-heating systems do not need any special

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provisions for variable volume terminals at the perimeter, since night setback or morning warm-up is performed by the perimeter-heating system independent of the variable volume terminal. The interior system VAV terminals will stay shut as long as room temperatures stay below the night setting so the VAV fan system should remain idle at night.

### 3.18.3 Humidification

Whenever the supply air is throttled, its capacity to dehumidify air in the cooled zone decreases. Normally, this decrease is not a problem at minimum air volumes. For normal occupancy concentrations with fairly constant internal lighting loads, a small reduction in evaporator temperature and supply air will compensate for the reduced air volume.

However, spaces with high latent loads must be examined carefully on the psychometric chart before variable air volume can be applied. Such spaces can often be treated economically in variable volume systems with a combination of variable volume and one of the several methods for maintaining minimum ventilation. In critical spaces, a humidistat may override the air-volume reduction controlled by thermostat and control a reheat coil to maintain proper zone temperatures.

## 3.19 VAV COMPONENTS AND CONTROLS

### 3.19.1 Variable Air Volume Units

Variable Air Volume Units can be controlled by duct-mounted units serving air outlets in a control zone or by control units integral to each supply air outlet.

### 3.19.2 Pressure Independent Units

Pressure Independent Units regulate the flow-rate in response to the thermostat's call for heating or cooling. The required flow-rate is maintained regardless of fluctuation of the VAV unit inlet or system pressure. These units can be field or factory adjusted for maximum and minimum (or shut off) air volume settings. They will operate at inlet static pressures as low as 0.2 in. wg (50 Pa) at maximum system design volumes.

### 3.19.3 Combination Pressure Dependent And Independent Units

Combination Pressure Dependent and Independent Units regulate maximum volume, but the flow-rate below maximum varies with the inlet pressure variation. Generally, airflow will oscillate when pressure varies. They are less expensive than pressure-independent units and can be used where pressure independence is required only at maximum volume, when system pressure variations are relatively minor and some degree of constantly changing discharge airflow is acceptable.

### 3.19.4 Pressure Dependent Units

Pressure Dependent Units do not regulate the flow-rate but position the volume-regulating device in response to the thermostat. They are the least expensive and should only be used where there is no need for limit control and the system pressure is stable.

### 3.19.5 Bypass (Dumping) Units

VAV room supply is accomplished in constant-volume systems by returning excess supply air into the return ceiling plenum or return air duct, thus bypassing the room. However, this reduction of system volume is not energy efficient. Use of a Bypass Unit is generally restricted to small systems where a simple method of temperature control is desired, initial cost is modest, and energy conservation is unimportant.

### 3.19.6 Supply Outlet Throttling Units

The area of the throat or the discharge openings of the Supply Outlet Throttling Units, which are usually linear diffusers, are thermostatically varied. The opening varies in approximate proportion to the air-volume to maintain throw-pattern stability, even with low-air quantities. Since these units are pressure-dependent, constant-pressure regulators are usually required in the duct system. Noise is a particular concern when selecting outlets.

### 3.19.7 VAV Controls

The types of controls for Variable Air Volume Units vary with the terminal device. Most use either pneumatic or electric-mechanical actuators. To conserve horsepower and limit system noise in larger systems, fan operating characteristics and system static pressure should be controlled. Many methods of VAV control are available including fan-speed control, variable-inlet-vane control, fan bypass, fan discharge damper, and variable-pitch fan control.

The pressure-sensing location usually depends on the type of variable volume terminal. Pressure-dependent units without controllers should be near the static pressure midpoint of the duct run to ensure minimum pressure variation in the system. Where pressure-indepen-

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dent units are installed, pressure controllers may be at the end of the duct run having the highest static pressure loss. This sensing point ensures maximum fan-power savings while maintaining the minimum required pressure at the longest end-of-run terminal.

However, as the flow through the various parts of a large system varies, so does the static pressure loss. Some field adjustment is usually required to find the best location for the pressure sensor. In many systems, the first location is two-thirds to three-fourths the distance from the supply fan to the end of the main trunk duct. As the pressure at the system control point increases due to terminal units closing, the pressure controller signals the fan controller to position the fan volume control, which reduces flow and maintains constant pressure.

The fan power equations show that the potential for fan energy savings with fan speed control is substantially greater than with variable-inlet vanes. Power is directly related to both total pressure and air volume and both methods affect the same energy saving when volume is reduced. However, variable-inlet-vane control only takes advantage of the portion of the system's pressure reduction that is not compensated for by throttling the fan while variable-speed control takes advantage of the full system-pressure reduction, which is a square-root function of flow.

A velocity controller installed in the outside air intake ensures that dampers are open sufficiently to provide the minimum outside airflow desired when total flow is reduced. A less costly method is to schedule a variable minimum position of the outside-air damper from the duct static-pressure sensor or the fan control operator, either of which reacts directly to system volume reduction.

## 3.20 VAV SYSTEM ADVANTAGES

Some advantages of Variable Air Volume Systems are:

a. Combined with one of the perimeter heating systems, a VAV system offers inexpensive temperature control for multiple-zoning and a high degree of simultaneous heating-cooling flexibility.

b. Changing loads from lights, occupancy, solar, and equipment can provide VAV system diversities of as much as 30 percent, compared with a system based on the sum of the peaks. Consequently, the cost is lower for fans, refrigeration, heating, and associated plant auxiliaries, as well as for duct mains, insulation, and piping systems.

c. Self-balancing is possible for a true VAV system (except pressure-dependent), limited only by inadequate static pressure control of volume regulation. The larger the network, the greater the need for these controls.

d. Sub-dividing an existing space or satisfying increased room loads is fairly simple, as long as the overall system has the reserve for the load increase or if this new load does not exceed the original design simultaneous peak load. Return-air systems do not have this advantage, except as the distribution hardware might lend itself to subdivisions and partition changes.

e. Operating cost savings are accrued from the following building characteristics.

- Fans run long hours at reduced volumes and, with fan control, they use less energy. In addition, the installed fan power is reduced.
- Refrigeration, heating, and pumping matches diversity of loads, so energy is saved.
- Outside-air economizer cooling, where applicable, provides better operating economy from lower mixing temperatures at constant minimum outdoor air with reduced supply air. With less parasitic reheat, part-load zones may first be reduced to minimum supply-air and the supply temperature may be raised before requiring reheat. This gives longer filter life, since less outside air is required for the same cold-supply temperature.
- Airflow to unoccupied areas may be fully blocked off to decrease both refrigeration and ventilation requirements.

## 3.21 VAV SYSTEM DESIGN PRECAUTIONS

In applying a VAV system, the designer should consider the following:

### 3.21.1 Air Distribution

a. Install high entrainment types of outlets to achieve higher air velocity at minimum flow.

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b. Consider duct lining to first outlet to minimize noise at maximum flow.

c. Evaluate system performance and effect on space-air movements at minimum volume.

d. Locate ceiling breaks and obstructions at the end of the design throw of the air jet. A single-plane ceiling allows air to hug the ceiling at reduced volumes (Coanda effect).

e. Note that air-troffer lighting may reduce design air volume to an undesirably low level, when using VAV terminals that only throttle the flow.

### 3.21.2 Fans and Controls

a. Use fan controls to save power and to operate at minimum system pressure for noise control.

b. Note that during cooling start-up with all variable volume controls wide open, system static pressure will be abnormally low and system volume abnormally high. Limiting the load of fans motors is desirable in this case.

c. Check supply and return fan operating characteristics at maximum and minimum flow.

d. Provide minimum outside-air volume to comply with ventilation codes.

### 3.21.3 System Operating Characteristics

a. Determine the warm-up requirements after prolonged shutdown, particularly for systems using cold air bypass to plenums or if terminals do not close tightly at no load. Consider the possibility of overcooling if minimum settings are used on VAV units.

b. Identify the need for heating or reheat to prevent increased space humidity in zones with highly fluctuating loads.

c. Remember that flow controllers require uniform entering air conditions to perform accurately, while volume regulators do not.

d. Consider the effect of interior-space cooling loads during winter in ceiling return-air systems if warm air is used on perimeter units. The higher ceiling temperature may increase the total ceiling load in the room and decrease the amount of ceiling plenum-air induced, requiring a greater air volume for winter than for summer.

e. Check system sound levels at maximum-flow conditions. Assuming adequate selection and static pressure control for fans and variable volume devices are given, the variable volume system usually operates at maximum noise level at full load. Inlet vanes and fan-discharge dampers must be checked for sound levels at minimum open position. Adjustable outlets for volume control must also be checked to determine sound level at minimum-flow position of the outlets. Except for low-pressure systems, throttling devices must be located 5 to 10 diameters from the outlet in the occupied space.

### 3.21.4 VAV Perimeter Systems

a. The fan energy savings listed under advantages apply to interior VAV systems of duplex designs having a separate-perimeter air handler. A separate-perimeter system requires more fan energy than the perimeter-radiation system because it supplies a constant volume of air and fan-powered heating during unoccupied periods. The fan energy required for common-fan, reheated-perimeter designs is even higher, especially if VAV terminals in interior areas do not shut off completely at night.

b. Fire and smoke control could be jeopardized with under-window perimeter supply outlets, since fire-rated floors may be pierced with substantial openings.

c. Recognize that the initial cost of a separate fan system is generally higher than VAV and radiation combinations for comparable perimeter zoning, unless expensive radiation enclosures are used. The margin of difference, however, is lower with overhead perimeter distribution, especially in milder climates than for under-window air distribution.

### 3.21.5 VAV Bypass Systems

Note that limited benefits result from Variable Volume Ceiling Bypass Systems, as compared with true system-throttling variable control and variable volume systems. Benefits are limited to simple temperature control and variable zone flexibility at moderate initial cost with low noise levels, and with the advantages at-

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tributed to air systems in general. Energy and operational economics possible with true variable air volume systems, and initial cost advantages resulting from the effects of load shifting and diversities on equipment and duct systems, are not applicable to VAV systems.

## 3.22 VAV TERMINAL DEVICES

### 3.22.1 VAV System Fundamentals

VAV systems deliver air to the conditioned spaces at a constant temperature. The space loads are satisfied by changing the HVAC system airflow quantities into the spaces. Interior zones may vary only 10 to 20 percent while perimeter zones vary extensively because of shifting solar and infiltration loads.

Each VAV box or zone is designed for the maximum-peak load of that space that will be reached sometime during the operation year. The central plant equipment, including the boiler, chiller, pumps, fans, and HVAC equipment is designed for the maximum peak of the total building at some point during the operation year. However, the duct system is designed to handle the total-peak load of all of the VAV boxes or zones.

### 3.22.2 VAV Systems Diversity

The total HVAC system load, divided by the total of the VAV box loads, equals the "diversity factor." For example, if the sum of all VAV box maximum capacities was 820,000 Btuh (240.2 kW) and the capacity of the central plant equipment was 660,000 Btuh (193.4 kW), the diversity factor would be:

**(I-P)**

$$\frac{660,000}{820,000} = 0.805 \text{ or } 80.5 \text{ percent}$$

**(SI)**

$$\frac{193.4}{240.3} = 0.805 \text{ or } 80.5 \text{ percent}$$

The same diversity factor will be obtained by using airflow totals instead of Btuh (kW) loads.

### 3.22.3 VAV System Diffusers

VAV system-diffuser selections should be based on maximum airflows and outlet velocities with round, square, or slot diffusers. Square and round outlets handling approximately 400 to 500 cfm (200 to 250 L/s) generally will have minimal changes in throw. Ceiling-slot diffusers have earned a reputation for handling larger reductions in airflow without creating perceived stagnant areas.

Special consideration to sound levels must occur when selecting diffusers at maximum flow rates. However, space airflow velocities should range between 20 to 50 fpm (0.10 to 0.25 m/s), depending on the season and activity level.

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CHAPTER 4

# GENERAL APPROACH TO DUCT DESIGN

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## 4.1 SCOPE

HVAC system duct design can be completed as *soon* as all room loads and air quantities have been determined. Actual design procedures will be examined in more detail in later chapters of this manual.

Consider the type of duct system needed, based on an economic analysis of the building design and use, unless the owner or architect specifies a preference. The specific type of system will affect the type of air handling equipment that is selected.

### 4.1.1 Duct System Selection

The primary purpose of the HVAC system is to provide comfort to the occupants of the conditioned space, or to provide a specific set of environmental conditions required within the conditioned space. Factors that affect comfort and indoor air quality include:

a. Air cleanliness

b. Odor

c. Space temperature

d. Means of temperature control

e. Air motion and distribution

f. Mean radiant temperature

g. Quality of ventilation

h. Humidity control

i. Noise level

Air duct systems can be separated into two main categories – single-duct and dual-duct systems. Single-duct systems are those with the main heating and cooling sources in a series-flow path, using a single path duct distribution system with a common (variable) air temperature to feed all terminal apparatus; or using a separate duct to each zone after blending the air from the hot and cold sources within the air handling unit (multi-zone unit).

A dual-duct system contains the main heating and cooling sources with a parallel airflow path, using a separate cold- and warm-air duct distribution system that blends the air at the terminal apparatus. Systems may also be constant or variable volume, have reheat capabilities at the room terminal device, or induce secondary air for controlling terminal air temperatures. This manual will provide fundamental design methods and procedures, without exploring special applications of these methods for the design of variable volume and dual-duct systems. These fundamental design methods may be used when designing special duct systems found in the SMACNA *HVAC Systems Applications* manual.

### 4.1.2 Air Distribution

Identify where the supply air outlets will be located, then select the size and type of outlet required for proper air distribution in each conditioned space, *see* Chapter 3. Air distribution in the conditioned space is highly important in influencing the comfort of the occupants. Good air distribution is achieved by proper consideration of the basic factors in the selection of the outlet terminal devices.

The outlet terminal devices need to provide the proper air velocities within the rooms occupied zone, an air layer between 3 and 72 in. (75 and 1800 mm) above the floor and more than 24 in. (600 mm) from the walls or fixed air-conditioning equipment, and the proper room temperature equalization. Entrainment of room air by this supply airstream to counteract the effects of natural room air convection is very important.

It is recommended that all air distribution terminal devices be selected from industry standard configurations so that they will be available from multiple sources. Most terminal device manufacturers' catalogs provide data on airflow, throw, drop, air pattern, terminal velocities, acoustics, and ceiling heights.

Supply outlets on the same branch duct should be chosen with approximately the same pressure loss; and no more than 0.05 in. wg (12.5 Pa) variation through to the outlet and associated air straightening and balancing devices. Mixing ceiling supply diffusers with sidewall supply grilles on the same branch should be avoided, unless there is no significant difference in pressure drops between the different types.

For a comprehensive review of considerations in the selection of air distribution equipment, refer to Chapter 3 and to air-distribution equipment manufacturer's engineering data. Some of the basic procedures used in the selection of air distribution equipment are:

a. Consider the ambient conditions that could affect occupant comfort.

b. Decide on the location of air supply outlets, such as in the floor, sill, sidewall, exposed

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duct, or ceiling, taking into account the system serving them.

c. Locate return and exhaust air devices.

d. Consider the special requirements affecting outlets when used with a variable air volume (VAV) system.

e. Select straightening vanes and dampers for outlet devices to provide uniform face velocity and minor balancing.

f. Refer to manufacturer's data regarding material, finish, throw, spread, drop, and noise level.

### 4.1.3 Zoning

After the outlet devices have been selected, but before duct layout and duct sizing can begin, the designer must identify how many zones of temperature control will be required for both perimeter and interior zones. In general, the exterior zone will be divided into zones determined by building exposure; North, East, South, or West.

These perimeter zones may be further subdivided into smaller control zones, depending on variations in internal load, or a requirement for individual occupant control. Typical situations would include private executive offices, where the owner may want individual control, or areas of high heat gain or loss, such as computer rooms, conference rooms, or corner rooms having two exposed walls.

Similarly, the interior zones may also be subdivided into control zones to satisfy individual room requirements or variations created by internal loads such as lights, meeting or conference areas, or large concentrations of equipment.

### 4.1.4 Preliminary Layout

The next step is to create a preliminary schematic diagram for the ductwork which will convey the design air quantity to each zone outlet by the most efficient and economical path. However, many HVAC designers use computer-aided drafting (CAD) much earlier in the design process, and can view floor-plan "layers" with their duct design layouts.

By combining the initial duct layout with the floor plans the designer will have a better feel for the final relationship of air terminals, branch ducts, main ducts, risers, and primary HVAC equipment. This procedure will help the designer coordinate the ductwork with the structural components of the building, and other building systems and services.

The designer should indicate the design airflows and pressures throughout the system. If a constant-volume system is chosen, it will be the arithmetic sum of the CFM (L/s) for each terminal (including branches), working back from the end of the longest run to the fan. However, if a variable air volume system is chosen, the designer must apply the proper diversity factors. This allows calculating the total of the peak design airflows to determine their impact on branch and main duct sizes coming from the supply fan.

The same procedure must also be followed for return and exhaust air systems. This not only helps size the ductwork properly, but also permits the designer to evaluate the total HVAC system design while balancing the proper proportions of supply air, return air, exhaust air, and outside makeup air.

### 4.1.5 Duct Sizing Procedure

Having completed the preliminary HVAC system duct layout, the designer will then proceed to use one of the methods for sizing the duct system discussed later in this chapter and in detail in Chapter 7. Generally, these methods will give the equivalent round duct sizes and the pressure losses for the various elements of the duct system. The designer will then incorporate this information on the preliminary duct layout.

If round ductwork is to be used throughout, the duct sizing is now complete, providing the ductwork will physically fit into the building. If rectangular or flat-oval ductwork is chosen, the proper conversions must be made from the equivalent round-duct sizes to rectangular or flat-oval sizes. Applying the appropriate duct friction loss correction factors and using the duct fitting loss coefficients, the duct system total-pressure loss can be calculated.

### 4.1.6 Automated Duct Sizing

More design firms are using computer programs developed by the HVAC industry to calculate building block loads, zone loads, system loads, coil sizes, terminal equipment sizes, and airflows required for each zone.

It is now possible to direct the output from these programs directly into a CAD system, which will translate these loads and airflows into duct and system layout drawings. Several of the more robust software pro-

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grams are excellent and have years of proven use and refinement. However, everyone involved with this process must realize that neat computerized drawings do not guarantee accuracy, as everything started with one individual entering data. If any initial data input was in error, there will be a "cascade" effect through the entire design, layout, and sizing processes that follows, making it very difficult to identify the error until "caught" during construction or, even worse, when system balancing objectives cannot be achieved.

### 4.1.7 Equipment Space Requirements

With HVAC system duct sizes now selected and the total pressure or static pressure losses calculated, the designer must determine if the ductwork will "fit" into the building in the spaces provided. At this point, the designer must consider the additional space required beyond the bare sheet metal sizes for reinforcing and circumferential joints. Consideration must be given to external insulation or internal duct liner which may be required, clearance for piping, conduit, light fixtures, and clearance for the removal of ceiling tiles.

### 4.1.8 Service Access

Many HVAC systems cannot be maintained because access doors to filters and other system components are totally blocked by equipment supports, ceiling grids, building structure, piping, and electrical conduit installed by others. When locating system components that will require servicing, the design professional must make sure adequate clearances have been provided and all potential interference conflicts have been resolved.

A further consideration in the sizing and routing of ductwork is the space and access requirements for air terminals, mixing boxes, VAV boxes, fire and smoke dampers, balancing dampers, access doors, reheat coils, and other accessories.

## 4.2 DESIGN METHODS – OVERVIEW

There is no single design method that will automatically provide the most economical duct system for all conditions. Duct systems are normally designed using one or more of the following methods. Please note that some of these methods are now considered obsolete:

a. Equal friction.

b. Static regain.

c. Extended plenum or semi-extended plenum.

d. T-Method.

e. Velocity reduction.

f. Total pressure.

g. Constant velocity.

h. Residential system design method.

A careful evaluation of all cost variables entering into a duct system should be made with each design method, or combination of methods. The cost variables to consider include: the cost of the duct material (the aspect ratios are a large factor), duct insulation or lining (duct heat gain or loss), type of fittings, space requirements, fan power, balancing requirements, sound attenuation, air distribution terminal devices, and heat recovery equipment.

Slightly different duct system pressure losses can be obtained using different design methods. Some require a broad background of design knowledge and experience. Therefore, combinations of the most widely used duct design methods will be used in Chapter 7 Duct Sizing Procedures, along with the "semi-extended plenum" modification. The careful use of these methods will allow the designer to efficiently size HVAC duct systems for larger residences, institutional, and commercial buildings, including some light industrial process ducts. Traditional duct design methods follow.

### 4.2.1 Equal Friction

The equal friction method of duct sizing, where the pressure loss per foot of duct is the same for the entire system, is probably the most universally used means of sizing lower-pressure supply, return, and exhaust duct systems. It is not normally used for higher-pressure systems. With supply duct systems, this design method "automatically" reduces air velocities in the direction of airflow.

The major disadvantage of the equal friction method is there is no provision for equalizing pressure drops in duct branches, except symmetrical layouts.

### 4.2.2 Static Regain

The static regain method of duct sizing may be used to design supply air systems of any velocity or pressure. It normally is not used for return systems where the airflow is toward the HVAC unit fan. This method is more complex to use than the equal friction method. But, it is a theoretically sound method that meets the require-

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ments of maintaining uniform static pressure at all branches and outlets. Duct velocities are systematically reduced, allowing a large portion of the velocity pressure to convert to static pressure that offsets the friction losses in the succeeding section of duct. This static regain, which is assumed at 75 percent for average duct systems, could be as high as 90 percent under ideal conditions.

Another advantage is the duct system will stay in balance, because the losses and gains are proportional to a function of the velocities. Therefore, it is an excellent method for designing variable air volume systems.

A disadvantage of the static regain method is that oversized ducts can occur at the ends of long branches, especially if a duct run is unusually long. Often, the resultant very-low velocities require the installation of thermal insulation on that portion of the duct system to prevent unreasonable duct heat gains or losses.

*Caution: The loss coefficients for duct fittings found in the Appendix of this manual, or the ASHRAE Handbook - Fundamentals, include static pressure regain or loss for the velocity condition changes that occur at divided-flow or change-of-size duct fittings. Additional duct static pressure regain (or loss) must not be calculated and added to (or subtracted from) the total duct system pressure losses when those fitting losses are used.*

### 4.2.3 Extended Plenums

An *extended plenum* is a trunk duct—usually at the discharge of a fan, fan coil unit, mixing box, or variable air volume box—extended as a plenum to serve multiple outlets or branch ducts.

A *semi-extended plenum* is a *trunk duct system* utilizing the concept of the extended plenum that has a minimum number of size reductions. This modification can be used with equal friction and static regain design methods. Some of the advantages include: lower first costs, lower operating costs, ease of balancing, and adaptability to branch duct or outlet changes.

However, low airflow velocities could result in excessive heat gain or loss to the airstream through the duct walls. This duct sizing method is explained in more detail in Chapter 7.

### 4.2.4 T–Method

The T-Method of duct sizing is a duct-design optimization procedure that includes system initial costs and operating costs, energy costs, hours of operation, annual escalation, and interest rates. Manual procedures and equations may be found in the *ASHRAE Handbook - Fundamentals*, but the method is best used with computer software.

## 4.3 SELDOM USED METHODS

### 4.3.1 Velocity Reduction

An experienced designer, who uses sound judgment in selecting arbitrary velocities, is qualified to design a relatively simple duct system using the velocity reduction method. Others should not attempt to use this method, except for estimating purposes, unless the system only has a few outlets and can be easily balanced.

A system velocity is selected at the section next to the fan and arbitrary reductions in velocity are made after each branch or outlet. The resulting pressure loss differences in the various sections of the duct system are not taken into account and balancing is attempted by the use of quality dampers at strategic locations.

### 4.3.2 Total Pressure

The total pressure method is a further refinement of the static regain method, which allows the designer to determine the actual friction and dynamic losses at each section of the duct system. The advantage is having the actual pressure losses of the duct sections and the fan total pressure requirements provided.

### 4.3.3 Constant Velocity

With adequate experience, many designers are able to select an optimum velocity that is used throughout the design of a duct system. This method is best adapted to the higher-pressure systems that use attenuated terminal boxes to reduce the velocity and noise before distribution of the air to the occupied spaces.

### 4.3.4 Residential System Design

The SMACNA *Installation Standards for Residential Heating and Air Conditioning Systems* contains a simplified duct design method for use in the residential heating and air conditioning segment of the industry. However, this manual provides a more accurate method of sizing larger residential duct systems when the equal friction method is used.

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## 4.4 DUCT HEAT GAIN OR LOSS

At the beginning of this chapter, it was stated that duct design follows building load calculations. An often-overlooked factor in load calculations is duct heat gain or loss. The method of calculating this load is well described in other texts, including the *ASHRAE Handbook - Fundamentals*. In this section, some of the practical considerations in duct design that affect duct heat gain or loss are reviewed.

First, consider a conditioned air supply system with the air handling equipment and ductwork in the conditioned space. No additional load is imposed on the system; however, if the ductwork is long and velocities low, the designer should check that airflows are proportioned properly. The air in the ductwork still gets warmer or cooler as it passes through the conditioned space, thus decreasing the temperature difference. As a result, less air is required to supply the outlets at the start of the supply run and more is required at the end.

Naturally, when a duct or plenum carrying conditioned air is located outside of the conditioned space, the heat gain or loss must be accounted for in both the design air quantity and total sensible load. This system load must be calculated by the designer when routing ductwork through boiler rooms, attics, outdoors, or other unconditioned spaces. Alternate routing might be more desirable than increasing the system load.

In several places in this manual life-cycle costing is discussed, and it is suggested that semi-extended plenums could reduce first and operating cost. However, the designer must also realize if the air velocities are reduced too much, duct wall heat transfer increases proportionately as a ratio of CFM to duct surface area decreases and additional duct insulation may be required.

The use of additional insulation on ductwork is becoming more universal with increased energy costs, as evidenced by the fact that ASHRAE energy standards require certain ducts and plenums to be insulated or lined. This greatly reduces the impact of duct heat gain or loss.

## 4.5 SOUND AND VIBRATION

With the duct system design approaching the final stages, an analysis must now be made to determine if acoustical treatment is necessary. The addition of sound traps, duct liner, or vibration isolation might be required due to conducted or generated noise and vibration in certain critical areas. Chapter 10 contains general guidance to make this determination. Some duct resizing may be required at this stage to incorporate the necessary acoustical treatment into the duct system design. For more detailed sound and vibration design guidance, refer to SMACNA's *Sound and Vibration Manual*.

## 4.6 PRESSURE CLASSIFICATION

The designer should indicate all ductwork static-pressure classification changes on the drawings. For clear interpretation of the requirements for ductwork fabrication, and economical attainment of performance objectives, it is essential that the contract documents define the portion of each duct system to be constructed for a particular static pressure classification, s*ee* Table 4-1. These static pressure rating changes are shown on the duct drawings by "flags" at each point where the duct static pressure classification changes, with the number on the "flag" indicating the pressure class of the ductwork on each side of the dividing line, *see* Figure 4-1. Also see the sample duct layouts for the duct design examples shown in Chapters 7.

| Static Pressure Class | | Operating Pressure | | Type of Pressure | Seal Class | Maximum Velocity | |
|---|---|---|---|---|---|---|---|
| U.S. Units in. wg | Metric Pa | U.S. Units in wg | Metric Pa | | | U.S. Units fpm | Metric m/s |
| ½ | 125 | Up to ½ | Up to 125 | Pos/Neg | C | 2000 | 10 |
| 1 | 250 | Over ½ to 1 | Over 125 to 250 | Pos/Neg | C | 2500 | 12.5 |
| 2 | 500 | Over 1 to 2 | Over 250 to 500 | Pos/Neg | C | 2500 | 12.5 |
| 3 | 750 | Over 2 to 3 | Over 500 to 750 | Pos/Neg | B | 4000 | 20 |
| 4 | 1000 | Over 3 to 4 | Over 750 to 1000 | Pos | A | 4000 | 20 |
| 6 | 1500 | Over 4 to 6 | Over 1000 to 1500 | Pos | A | As Specified | As Specified |
| 10 | 2500 | Over 6 to 10 | Over 1500 to 2500 | Pos | A | As Specified | As Specified |

**Table 4–1 HVAC Duct Pressure Velocity Classification**

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Special consideration must be given to the pressure classes of ductwork used for variable air volume (VAV) systems. It is possible for these supply-duct systems to experience the total fan pressure at the most distant VAV box at minimum airflow. Under these conditions, the maximum duct construction classification should remain the same throughout the supply duct system upstream of the VAV boxes.

Special consideration must also be given to emergency operations, such as when the smoke control system takes over fan operation, or fire dampers close against full system airflow. Select duct pressure classifications that will handle these sudden pressure changes without damage to the duct distribution system or include duct pressure relief devices in critical locations.

## 4.7 DUCT LEAKAGE

Some leakage is anticipated, but there is a point at which added sealing expense may not be justified. A large commercial duct system has hundreds of separate fittings, transitions, and straight segmented runs having hundreds of joints. Some are also required to remain flexible at HVAC equipment connections.

There are many ways to fabricate these joint connections, and even more ways to seal them after installation. It is important for the system specifier to determine the seal classification and allowable leakage rate that is appropriate for each duct installation and make this clear in the design documents. Reasonable allowances for duct leakage must be adopted, since no duct system is airtight unless it is of completely welded construction.

The anticipated amount of duct system leakage may be calculated after the duct pressure and seal classifications are determined. These leakage rates should be calculated in terms of cfm/ft$^2$ (L/s per m$^2$) of duct surface but they may also be converted into a percentage of total system airflow. Seal class and duct leakage class tables and charts, along with examples of use may be found in Chapter 5. The amount of system leakage must be added to the total airflow capacity of the HVAC system fan(s).

## 4.8 FAN SIZING

With the various elements of the HVAC duct system selected, and the duct system layout and sizing finalized, the designer now needs to calculate the total pressure of the systems. Chapter 7 has a detailed description on how to determine the friction losses in ductwork, and the dynamic losses through fittings. These in combination with the pressure loss data for duct system components and apparatus listed in the Appendix, enable the designer to determine the total pressure requirements for the fan(s). Estimated system air leakage must be added to the system airflow at this time.

## 4.9 TESTING, ADJUSTING AND BALANCING (TAB)

A very important step in HVAC duct system design is to provide the proper physical layout for testing, adjusting, and balancing the airflow in the system after the building is completed. It is essential that sufficient length of straight duct be provided in an accessible area so TAB personnel can perform their function properly, and to determine the actual total system airflow with a reasonable degree of accuracy. This also applies to critical branch ducts of the supply air system. This subject is discussed in more detail in Chapter 9; however, it is important that the design professional indicate all necessary balancing dampers and devices on the drawings, and where access doors are needed for TAB work and future system maintenance.

## 4.10 FINAL DESIGN DOCUMENTS

After all of the steps discussed in this chapter have been followed, the final plans can be drawn and the specifications completed. Using the preliminary design as a guide, a double-line duct system layout should be shown on the final mechanical drawings, employing symbols commonly used for ventilation and air conditioning design drafting, s*ee* Figures 4-1 to 4-3. Adequate detail must be employed to accurately convey to the installing HVAC contractor what types of fittings are required, and the locations of equipment, ductwork, fire and smoke dampers, balancing dampers, and access doors. This will make sure the installed system will function within the design parameters required and meet applicable codes.

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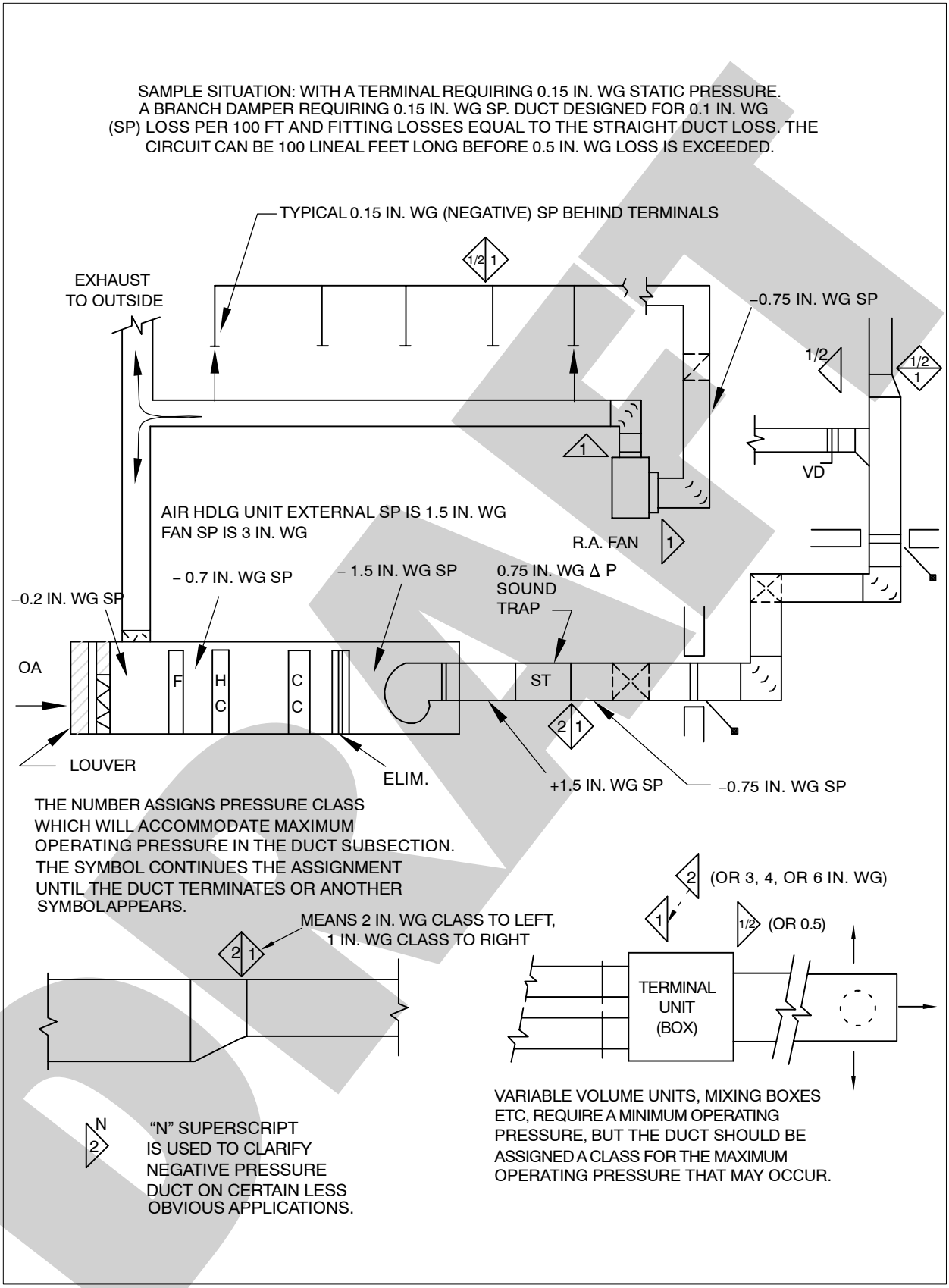


**FIGURE 4–1 DUCT PRESSURE CLASS DESIGNATION (I–P)**

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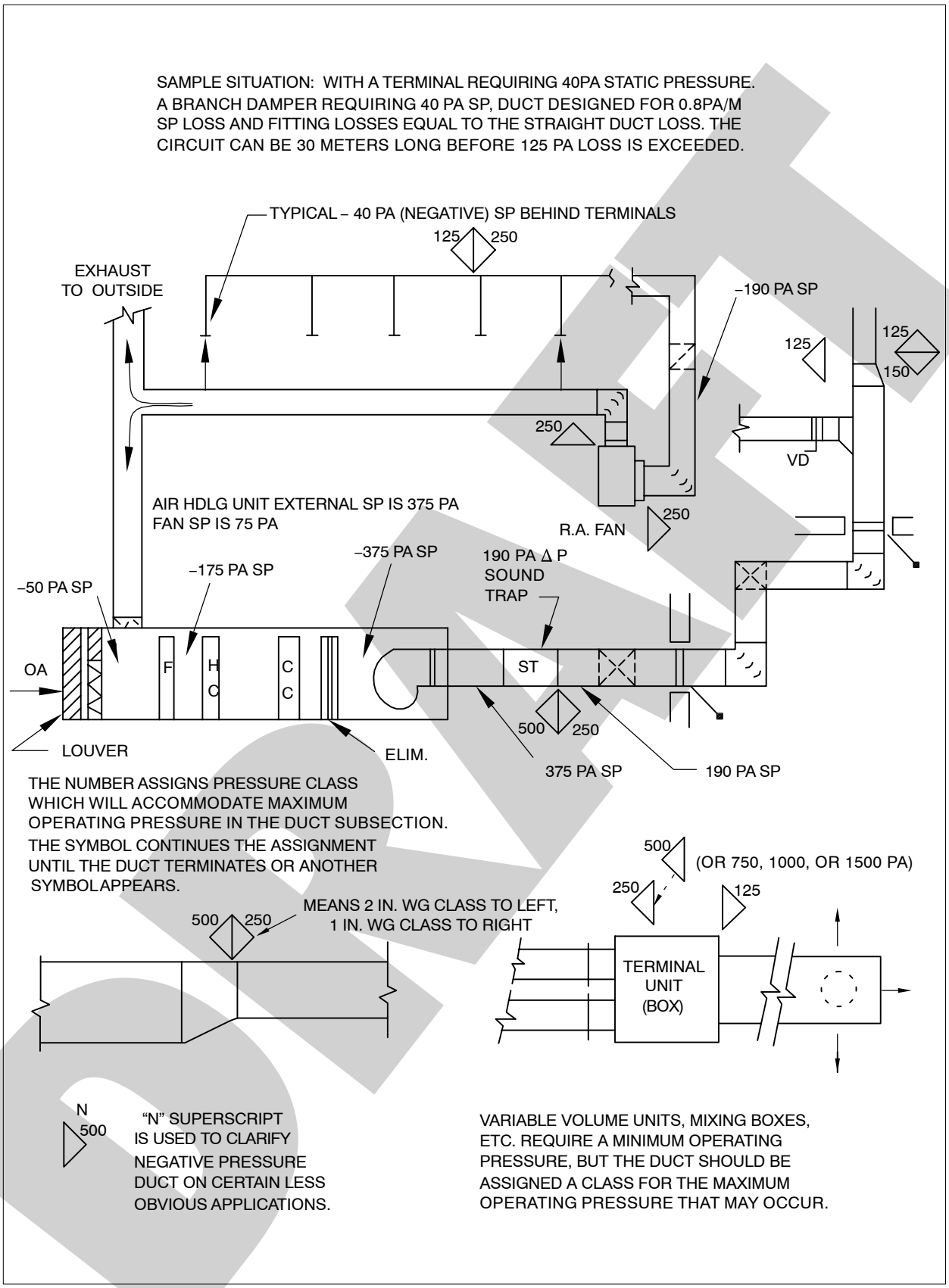


**FIGURE 4–1M DUCT PRESSURE CLASS DESIGNATION (SI)**

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| SYMBOL MEANING | SYMBOL |
|---|---|
| POINT OF CHANGE IN DUCT CONSTRUCTION (BY STATIC PRESSURE CLASS) | |
| DUCT (1ST FIGURE, SIDE SHOWN 2ND FIGURE, SIDE NOT SHOWN) | 20 x 12 |
| ACOUSTICAL LINING DUCT DIMENSIONS FOR NET FREE AREA | |
| DIRECTION OF FLOW | |
| DUCT SECTION (SUPPLY) | S 30 x 12 |
| DUCT SECTION (EXHAUST OR RETURN) | E OR R 20 x 12 |
| INCLINED RISE (R) OR DROP (D) ARROW IN DIRECTION OF AIR FLOW | R |
| TRANSITIONS: GIVE SIZES. NOTE F.O.T. FLAT ON TOP OR F.O.B. FLAT ON BOTTOM IF APPLICABLE | |
| STANDARD BRANCH FOR SUPPLY & RETURN (NO SPLITTER) 45° INLET | S R |
| WYE JUNCTION | |
| VOLUME DAMPER MANUAL OPERATION | VD |
| AUTOMATIC DAMPERS MOTOR OPERATED | SEC MOD |
| ACCESS DOOR (AD) ACCESS PANEL (AP) | AP AD |
| FIRE DAMPER: SHOW VERTICAL POS. SHOW HORIZ. POS. | FD AD |
| SMOKE DAMPER | S AD S |
| HEAT STOP – CEILING DAMPER – RADIATION DAMPER – | C R |
| TURNING VANES (TYPE AS SPECIFIED) | |
| FLEXIBLE DUCT FLEXIBLE CONNECTION | |
| GOOSENECK HOOD (COWL) | |
| BACK DRAFT DAMPER | BDD |

| SYMBOL MEANING | SYMBOL |
|---|---|
| SUPPLY GRILLE (SG) | 20 x 12 SG 700 CFM |
| RETURN (RG) OR EXHAUST (EG) GRILLE (NOTE AT FLR OR CLG) | 20 x 12 RG 700 CFM |
| SUPPLY REGISTER (SR) (A GRILLE + INTEGRAL VOL. CONTROL) | 20 x 12 SR 700 CFM |
| EXHAUST OR RETURN AIR INLET CEILING (INDICATE TYPE) | 20 x 12 GR 700 CFM |
| SUPPLY OUTLET. CEILING, ROUND (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION | 20 700 CFM |
| SUPPLY OUTLET. CEILING, SQUARE (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION | 12 x 12 700 CFM |
| TERMINAL UNIT. (GIVE TYPE AND OR SCHEDULE) | T.U. |
| COMBINATION DIFFUSER AND LIGHT FIXTURE | |
| DOOR GRILLE | DG 12 x 6 |
| SOUND TRAP | ST |
| FAN & MOTOR WITH BELT GUARD & FLEXIBLE CONNECTIONS | |
| VENTILATING UNIT (TYPE AS SPECIFIED) | |
| UNIT HEATER (DOWNBLAST) | |
| UNIT HEATER (HORIZONTAL) | |
| UNIT HEATER (CENTRIFUGAL FAN) PLAN | |
| THERMOSTAT | T |
| POWER OR GRAVITY ROOF VENTILATOR – EXHAUST (ERV) | |
| POWER OR GRAVITY ROOF VENTILATOR – INTAKE (SRV) | |
| POWER OR GRAVITY ROOF VENTILATOR – LOUVERED | |
| LOUVERS & SCREEN | 36 H x 24 L |

**FIGURE 4–2 SYMBOLS FOR VENTILATION AND AIR CONDITIONING (I–P)**

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| SYMBOL MEANING | SYMBOL |
|---|---|
| POINT OF CHANGE IN DUCT CONSTRUCTION (BY STATIC PRESSURE CLASS) | |
| DUCT (1ST FIGURE, SIDE SHOWN 2ND FIGURE, SIDE NOT SHOWN) | 508 x 305 |
| ACOUSTICAL LINING DUCT DIMENSIONS FOR NET FREE AREA | |
| DIRECTION OF FLOW | |
| DUCT SECTION (SUPPLY) | S 762 x 305 |
| DUCT SECTION (EXHAUST OR RETURN) | E OR R 762 x 305 |
| INCLINED RISE (R) OR DROP (D) ARROW IN DIRECTION OF AIR FLOW | R |
| TRANSITIONS: GIVE SIZES. NOTE F.O.T. FLAT ON TOP OR F.O.B. FLAT ON BOTTOM IF APPLICABLE | |
| STANDARD BRANCH FOR SUPPLY & RETURN (NO SPLITTER) 45° INLET | S R |
| WYE JUNCTION | |
| VOLUME DAMPER MANUAL OPERATION | VD |
| AUTOMATIC DAMPERS MOTOR OPERATED | SEC MOD |
| ACCESS DOOR (AD) ACCESS PANEL (AP) | AP AD |
| FIRE DAMPER: SHOW VERTICAL POS. SHOW HORIZ. POS. | FD AD |
| SMOKE DAMPER | S AD S |
| HEAT STOP – CEILING DAMPER – RADIATION DAMPER – | C R |
| TURNING VANES (TYPE AS SPECIFIED) | |
| FLEXIBLE DUCT FLEXIBLE CONNECTION | |
| GOOSENECK HOOD (COWL) | |
| BACK DRAFT DAMPER | BDD |

| SYMBOL MEANING | SYMBOL |
|---|---|
| SUPPLY GRILLE (SG) | 508 x 305 SG 330 LPS |
| RETURN (RG) OR EXHAUST (EG) GRILLE (NOTE AT FLR OR CLG) | 508 x 305 RG 330 LPS |
| SUPPLY REGISTER (SR) (A GRILLE + INTEGRAL VOL. CONTROL) | 508 x 305 SR 330 LPS |
| EXHAUST OR RETURN AIR INLET CEILING (INDICATE TYPE) | 508 x 305 GR 330 LPS |
| SUPPLY OUTLET. CEILING, ROUND (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION | 508 330 LPS |
| SUPPLY OUTLET. CEILING, SQUARE (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION | 508 330 LPS |
| TERMINAL UNIT. (GIVE TYPE AND OR SCHEDULE) | T.U. |
| COMBINATION DIFFUSER AND LIGHT FIXTURE | |
| DOOR GRILLE | DG 305 x 152 |
| SOUND TRAP | ST |
| FAN & MOTOR WITH BELT GUARD & FLEXIBLE CONNECTIONS | |
| VENTILATING UNIT (TYPE AS SPECIFIED) | |
| UNIT HEATER (DOWNBLAST) | |
| UNIT HEATER (HORIZONTAL) | |
| UNIT HEATER (CENTRIFUGAL FAN) PLAN | |
| THERMOSTAT | T |
| POWER OR GRAVITY ROOF VENTILATOR – EXHAUST (ERV) | |
| POWER OR GRAVITY ROOF VENTILATOR – INTAKE (SRV) | |
| POWER OR GRAVITY ROOF VENTILATOR – LOUVERED | |
| LOUVERS & SCREEN | 914 H x 610 L |

**FIGURE 4–2M SYMBOLS FOR VENTILATION AND AIR CONDITIONING (SI)**

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CHAPTER 5

# DUCT DESIGN FUNDAMENTALS

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## 5.1 SCOPE

When the word "fluid" is used in the HVAC industry, it can refer to the properties of air, water, steam, or refrigerants. In this chapter, the fluid properties of air will be addressed, as these will have a direct impact on the behavior of air in all flow and temperature calculations.

## 5.2 FLUID PROPERTIES

The basic categories of fluid properties are: state, compressibility, viscosity, weight or density, specific volume, volatility, vapor pressure, specific heat, and heat content. These properties will be discussed as they relate to HVAC systems.

### 5.2.1 State

The state of a fluid refers to it being either liquid or gas. Liquids used in environmental systems are water, thermal fluids, such as ethylene glycol, and refrigerants in the liquid state. Gases are steam, evaporated refrigerants, and the air-water vapor mixture found in the atmosphere. Some substances, including commonly used refrigerants, may exist in any of the three states. An example is water, which may be solid (ice), liquid (water), or gas (steam or water vapor).

### 5.2.2 Compressibility

Compressibility of a fluid, or the ease which it may be reduced in volume by the application of pressure, depends upon the state of the fluid as well as the type of fluid. Liquids are not impossible to compress but are considered in this text to be incompressible. Gases of all fluids and environmental air are relatively easy to compress and the compression is related to temperature and pressure by equations.

### 5.2.3 Viscosity

Viscosity refers to the liquid state, to the ease with which it flows, or the difficulty experienced in making it flow. The higher the viscosity of a fluid, the greater the pressure required to cause it to flow in piping, which requires more energy. Viscosity is not a problem for HVAC fans but it does create problems along duct walls in the form of friction loss.

### 5.2.4 Weight, Density, and Specific Volume

The weight of a substance is the amount of force it exerts under pull by the Earth's gravitational field and that force is measured in pounds (kilograms).

The density of a substance relates to the number of particles or molecules of the substance in a given volume. It is recognized that it is easier to walk in still air than in still water, so it can be assumed that the density of air is much less than that of water. Density is referred to in terms of units of weight per unit of volume. When used in HVAC systems, lbs/ft$^3$ (kg/m$^3$) is used.

The standard conditions for air referred to in HVAC system are: *dry* air at 70 degrees F (21 degrees C) and at an atmospheric pressure of 29.92 in. mercury (101.325 kPa). At these conditions, the *density* of air is 0.075 lbs/ft$^3$ (1.204 kg/m$^3$).

Specific volume is the reciprocal of density and is used to determine the volume if the weight is known. Both density and specific volume are affected by temperature and pressure. The specific volume of air under standard conditions is 13.33 ft$^3$/lb (0.83 m$^3$/kg).

### 5.2.5 Volatility, Capillary Action, Cohesion and Surface Tension

Volatility, surface tension, and capillary action of a fluid are incidental to environmental systems.

Volatility is the rate at which liquids evaporate. Temperature and pressure also affect volatility, an increase in temperature increases evaporation in an open system.

When the open end of a small tube is immersed into any liquid, the liquid will rise higher than the surrounding liquid surface if it "wets" the tube walls, and will remain below the surrounding liquid surface if it does not. This characteristic of a fluid is called capillary action.

"Cohesion" is the attraction of the liquid molecules to each other, and "adhesion" is the attraction of the liquid molecules to the surface of another material it contacts, as shown in Figure 5-1. The liquid will "wet" the tube's interior surface and rise higher in the tube if the liquid's adhesive forces are stronger than its cohesion forces.

Surface tension is the property that causes a liquid to form a "bead" on some surfaces, and is defined as the work done to extend the liquid's surface over a unit of

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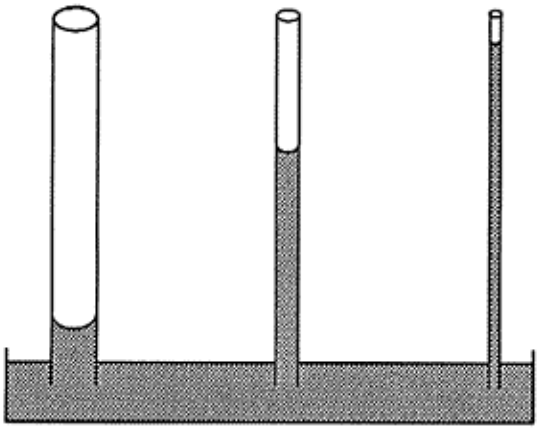

*NOTE: The force of capillary action can be illustrated by a group of small tubes of different diameters with one end submerged in water. The diameter of the tube directly affects the height the water will rise in each tube. The surface tension (upward force) of the water causes the water to rise—the narrower the tube, the higher the liquid will climb because a narrow column of liquid weighs less than a thicker one. A negative pressure at the upper end of the tube—even a very small one—will increase the amount of rise. This is part of the reason that controlling building pressure differentials is so important.*

**FIGURE 5–1 CAPILLARY ACTION**

area. Any fluid with a high surface tension will make it more difficult for the liquid's molecules located just below the surface to evaporate into the surrounding air. By increasing the temperature of the liquid, or increasing the speed of the airflow across the surface of the liquid, the rate of evaporation can be significantly increased.

### 5.2.6 Vapor Pressure

Vapor pressure denotes the lowest absolute pressure at which a given liquid at a given temperature will remain liquid, before evaporating into its gaseous form or state. The vapor pressure of a fluid can limit the suction lift of a hydronic pump. Vapor pressure is measured in lb/in.$^3$ (mm Hg).

### 5.2.7 Specific Heat

Specific heat ($C_p$) is the amount of heat energy required to raise the temperature of a one unit of a substance by one degree. The following are specific heat values at standard conditions:

Water $C_p$ = 1.00 Btu/lb degree F
(4.19 kJ/kg. degree C)

Air $C_p$ = 0.24 Btu/lb degree F
(l.005 kJ/kg. degree C)

## 5.3 FLUID STATICS

The term "static" as used in this text means "at rest" or "not moving". Although the "static" condition continues to exist when a fluid is being pumped through a system, the concept will be easier to discuss as if fans or pumps are not operating.

### 5.3.1 Standard Atmospheric Pressure

Weight is the gravitational force that a substance exerts, measured in pounds (grams or kilograms). A cubic foot of air under standard conditions weighs 0.075 pounds (1.204 kg/m$^3$).

The weight of a column of air of the Earth's atmosphere pressing on one square inch is 14.696 pounds (or 101,325 Newtons per one square meter.) 1 N/m$^2$ = 1 Pa so standard atmospheric pressure equals 101.325 kPa.

*NOTE: 1 kPa = 1000 Pa*

### 5.3.2 Absolute Pressure

Air at standard conditions (70 degrees F air at sea level with a barometric pressure of 29.92 in. Hg.) exerts a pressure of 14.696 psi. This is the pressure in a system when the pressure gage reads zero.

So the absolute pressure of a system is the gage pressure in pounds per square inch added to the atmosphere pressure of 14.696 psi. The symbol "psia" stands for pounds per square inch absolute.

In the metric system, 101.325 kPa is added to the gage pressure. Often a "g" for gage pressure is written behind the pressure terms (psig) when working with absolute pressures. Absolute pressures normally are not used for everyday HVAC system design work.

**Equation 5–1**

$$P_a = Pg + 14.7 \text{ psi } (101.3\text{kPa})$$

Where:

$P_a$ = Absolute Pressure (psia) (kPa a)

$P_g$ = Gage Pressure (psig) (kPa g)

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*Example 5-1 (I-P)*

Find the absolute pressure of a system at the point where an accurate pressure gage shows a reading of 10.0 psi.

$$P_a = P_g + 14.7 \text{ psi} = 10.0 + 14.7$$

$P_a = 24.7$ psia

*Example 5-1 (SI)*

Find the absolute pressure of a system where an accurate pressure gage shows a reading of 70.5 kPa.

$$P_a = P_g + 101.3\,kPa = 70.5 + 101.3$$

$P_a = 171.8$ kPa absolute

### 5.3.3 Static Head

Static head is the pressure developed by the weight of a fluid at rest in a system. Starting with only atmospheric pressure acting on the top surface of an open column of air or liquid, the static head pressure will increase up to a maximum at the lowest point in the column. Since water is comparatively incompressible, the density does not vary with the height of the column.

However, air is a compressible fluid. So the density of air will be reduced as the height or elevation increases.

In any system containing a liquid, such as water, the pressure or static head increases resulting from elevation are significant. This must be accounted for in the HVAC equipment selected to contain them and in the design of the systems in which they flow.

*Example 5-2 (I-P)*

Find the corrected barometric pressure from standard conditions for 100 degree F and 5000 ft elevation in psi.

Using Table A-17 in the Appendix:

correction factor = 0.78

corrected pressure = $0.78 \times 14.696 = 11.463$ psi

*Example 5-2 (SI)*

Find the corrected barometric pressure from standard conditions for 50 degree C and 1500 m elevation in Pa.

Using Table A-17M in the Appendix:

correction factor = 0.77

corrected pressure = $0.77 \times 101.325 = 78.020$ kPa

## 5.4 FLUID DYNAMICS

In fluid mechanics, the term "dynamics" is used to describe the condition of motion of a fluid in a system. With respect to HVAC systems, it refers to the system in operation with fluids flowing, as compared to the static condition when the system is shut down and the fluids are at rest.

### 5.4.1 Potential Energy

Potential energy is the ability to do work. Water behind a dam has the ability to do work, because of the difference in static heads between the highest water level behind the dam and the water level below the foot of the dam. However, until there is flow, there is no work and the stored energy is all potential.

If the water behind the dam is allowed to flow through a gate in the dam, the potential energy (at rest) is transformed into kinetic energy (of motion) and work is performed (the force of the water moving through a height).

### 5.4.2 Velocity

The velocity of a fluid is related to the cross-sectional area of the conduit (pipe or duct) through which it is flowing, and the volume of flow within the conduit. The velocity of a fluid varies within the conduit as shown in the diagram in Figure 5-2. This phenomenon, known as the velocity profile, is caused by friction between the conduit walls and the fluid.

When the flow is turbulent, the friction rate increases, heat transfer through the walls of an exchanger increases, and usually, so does system noise from higher turbulent fluid flow rates.

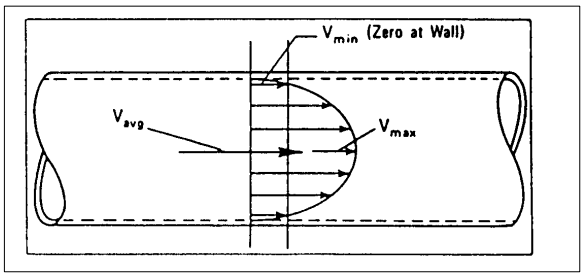


**FIGURE 5–2 VELOCITY PROFILE**

In HVAC duct systems the following equation is used to obtain the average air velocity:

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**Equation 5–2**

$$V = \frac{Q}{A}$$

Where:

$V$ = Velocity – fpm (m/s)

$Q$ = Airflow rate – cfm ($m^3/s$)

$A$ = Cross-sectional area – $ft^2$ ($m^2$)

Be sure that the correct units of measurement are used in Equation 5-2. Other units of measurement can be used with Equation 5-2, but constants may have to be applied. For example, 1 $m^3/s$ = 1000 L/s in the metric system where L/s is common used. So Equation 5-2 would then become:

$$V(m/s) = \frac{Q(l/s)}{1000 \times A(m^2)}$$

When referring to fluid flow, its velocity is considered to be moving linearly, although this straight-line flow will change direction as it moves through ductwork or piping fitting and turns. This straight-flow velocity is measured in feet per minute (fpm) or meters per second (m/s). However, the rotating velocity of rotating system components like pumps and fans is measured in revolutions per minute (rpm).

Fluids, however, pose a more difficult challenge. The particles or molecules of the fluid are somewhat independent, and may move at different velocities relative to each other. Consequently, velocities measured at different points in the same cross-section of a duct are not the same, since molecules may "slip" against one another.

Friction causes different velocities of a fluid in a duct cross-section. However, friction at the unmoving wall of the duct reduces the velocity of the fluid in contact to near zero. The next "layer" of fluid molecules is slowed down by the first, the third "layer" by the second and so on until the highest velocity is attained by the single center group of "layers". Without friction, all particles would move independently but at the same speed.

The calculated velocity of the fluid in the duct using Equation 5-2, is an average velocity, rather than a value that represents the speed of the fluid at any one point in the cross-section.

*Example 5-3 (I-P)*

Find the average velocity in a 48 × 24 in. duct handling 11,500 cfm.

$$V = \frac{Q}{A} = \frac{11,500\ (\text{cfm})}{48 \times 24/144\ (\text{ft}^2)}$$

$V$ = 1437.5 fpm

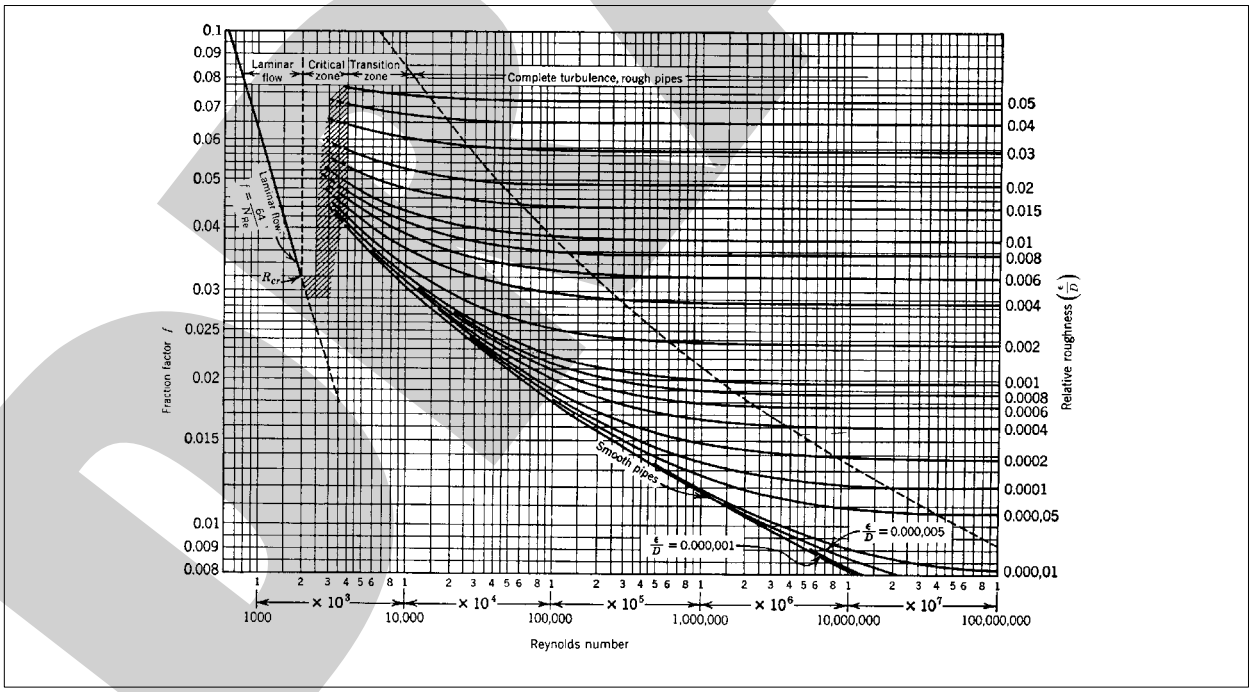


**FIGURE 5–3 RELATION BETWEEN FRICTION FACTOR AND REYNOLDS NUMBER**

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*Example 5-3 (SI)*

Find the average velocity in a 1200 × 600 mm duct handling 5800 L/s.

$$5800 \text{ L/s} = 5.8 \text{ m}^3/\text{s}$$

$1200 \times 600 \text{ mm} = 1.2 \times 0.6 \text{ m}$

$$V = \frac{Q}{A} = \frac{5.8\ (m^3/s)}{1.2 \times 0.6\ (m^2)}$$

$V = 8.06$ m/s

### 5.4.3 Flow In Ducts

The shape of the velocity profile in Figure 5-2, illustrates the phenomenon known as laminar flow. Velocity lines are shown straight for simplicity to illustrate the point. This velocity profile exists under special conditions depending on viscosity, friction, velocity, and other factors which, when combined, result in a value known as Reynolds number. When the Reynolds number is low, laminar flow occurs and all particles flow in a straight line.

When the velocity is increased to or above a critical value of Reynolds number, approximately 2000 to 3000 ($2.0$ to $3.0 \times 10^3$) the situation becomes more complicated. Molecules start to tumble about in a random or haphazard manner, but constantly move forward in the direction of flow. This condition is called turbulent flow. Note the laminar and turbulent areas in Figure 5-3.

Laminar flow produces lower friction losses, but such duct air velocities are impractically low. A special case of laminar flow is called stratification, which results in the air forming thermal layers due to differences in fluid density. This is a particular problem when very cold outside air tries to mix with warm return air at the inlet to some HVAC units. Laminar flow generates considerably less system noise, but heat transfer is poor.

Turbulent flow, conversely, produces higher friction losses, higher velocities, and reduced chance of stratification of mixed air, more noise, and excellent heat transfer.

In general, turbulent flow is easier to produce and more desirable to the system operation, but a hindrance to the technician attempting to take readings when balancing the system. For simplicity, the velocity average is indicated as a single arrow in the direction of flow.

### 5.4.4 Duct Friction Loss

Flow of fluid is resisted by friction; a natural resistance caused by a substance moving at a different rate than the substance with which it is in contact. One substance may be stationary and the other moving or both may be moving at different velocities.

In many cases, friction is necessary and useful. For example, if there were no friction, walking would be impossible. In oversimplified terms, without friction, nothing moving could be stopped by conventional means. In HVAC duct friction is an expensive disadvantage and, by selecting duct materials with smooth interior surfaces, the resulting low friction saves fan energy.

Frictional losses are due to fluid velocity and are a result of momentum exchange between molecules in laminar flow and between particles moving at different velocities in turbulent flow. Frictional losses occur along the entire duct length.

Table A-1 shows that normally used galvanized steel (Medium Smooth category) has a roughness factor ($\in$) of 0.0003 ft (0.09 mm). This is the value that ASHRAE used in developing the duct friction loss chart shown in Figures A-1 and A-1M. This data may be obtained from Table A-1. The velocities obtained from the duct friction loss charts are higher than the average velocities obtained from Equation 5-2 for the same volumes and duct sizes.

### 5.4.5 Reynolds Numbers

The Reynolds number ($R_e$) is the ratio of the inertia force, to the viscous force caused by changes in velocity within the fluid. The Reynolds number provides a convenient, non-dimensional means of comparing the two flows. The Moody Chart in Figure 5-3 shows the relationship of the friction factor ($f$), Reynolds number ($R_e$) and relative roughness ($\in/D$).

Within the region of laminar flow (Reynolds numbers less than 2000), the friction factor is a function of Reynolds numbers only. For turbulent flow, the friction factor depends on Reynolds number, duct surface roughness, and internal obstructions such as joints. The Moody Chart (Figure 5-3) depicts the behavior for round ducts. For hydraulically-smooth ducts, the friction factor again depends only on the Reynolds number, but the dependence is markedly different from laminar flow.

In general, for non-smooth surfaces the friction factor depends on roughness and Reynolds number. Howev-

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er, for a particular level of roughness beyond a sufficiently large Reynolds number, the friction factor becomes independent of the Reynolds number, a flow condition considered as fully rough. Between the bounding limits of hydraulically-smooth behavior and fully-rough behavior, is a transitional roughness zone where the friction factor depends on both roughness and Reynolds number. This transitionally rough turbulent zone is where most airflows occur in HVAC applications.

For round ducts, the Reynolds number may be calculated using Equation 5-3. For rectangular ducts, an equivalent diameter may be obtained by using Equation 5-4 for use in Equation 5-3 to obtain the Reynolds number.

**Equation 5–3 (I–P)**

$$R_e = 8.56\,DV$$

Where:

$R_e$ = Reynolds number

$D$ = Duct diameter, in.

$V$ = Velocity, fpm

**Equation 5–3 (SI)**

$$R_e = 66.4\,DV$$

Where:

$R_e$ = Reynolds number

$D$ = Duct diameter, mm

$V$ = Velocity, m/s

**Equation 5–4**

$$D = \frac{2HW}{H + W}$$

Where:

$D$ = Duct diameter, in. (mm)

$H$ = Duct height, in. (mm)

$W$ = Duct width, in. (mm)

*Example 5-4 (IP)*

Find the Reynolds number of a 60 × 15 in. duct with a velocity of 2200 fpm. Is the flow in the duct laminar or turbulent?

Using Equation 5-3:

$$D = \frac{2HW}{H + W} = \frac{2 \times 60 \times 15}{60 + 15}$$

$D$ = 24 in. diameter

Using Equation 5-3:

$$R_e = 8.56\,DV = 8.56 \times 24 \times 2200$$

$$R_e = 451,968 = 4.52 \times 10^5$$

Using Figure 5-3, the flow is found to be turbulent.

*Example 5-4 (SI)*

Find the Reynolds number of a 1500 × 375 mm duct with a velocity of 11 m/s. Is the flow laminar or turbulent?

Using Equation 5-4:

$$D = \frac{2HW}{H + W} = \frac{2 \times 1500 \times 375}{1500 + 375}$$

D = 600 mm diameter

Using Equation 5-3:

$$R_e = 66.4\,DV = 66.4 \times 600 \times 11$$

$$R_e = 438,240 = 4.38 \times 10^5$$

Using Figure 5-3, the flow is found to be turbulent.

## 5.5 FLUID FLOW PATTERNS

When air flows through a duct of constant diameter, the average velocity must be parallel to the centerline of the duct. The velocity at the duct wall is zero, and the average velocity can remain parallel to the duct by two methods:

a. *Laminar Flow:* All velocities are parallel to the duct wall.

b. *Turbulent Flow:* Velocities at various points across the duct can have components perpendicular to the wall, provided that the average of these components is zero.

For HVAC ducts, laminar flow does not normally exist and all airflow can be considered turbulent, except for extremely low air velocities. The velocity components perpendicular to the wall assure that the velocity parallel to the wall will be almost uniform over most of the duct, with a large change taking place close to the duct wall.

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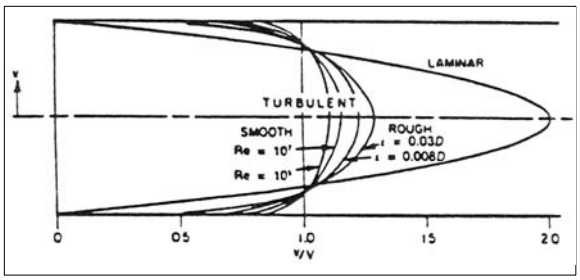


**FIGURE 5–4 VELOCITY PROFILES OF FLOW IN DUCTS**

The velocity profiles in a duct system for fully developed flow will vary depending on whether the flow is laminar or turbulent, and the degree of duct roughness. Velocity profiles of various flow conditions are shown in Figure 5-4.

The absolute velocity of the air stream will vary substantially over the face of the duct area. However, for HVAC duct systems, the velocity used for determining the velocity pressure is always the average velocity given by Equation 5-2:

$$V = \frac{Q}{A}$$

In Figure 5-4:

$\in$ = Duct Roughness Factor –ft (mm)

$R_e$ = Reynolds number – dimensionless

$v$ = Velocity at any point – fpm (m/s)

$V$ = Average velocity – fpm (m/s)

The duct velocity profiles shown in Figure 5-4 are uniform along the length of the duct and symmetrical around the centerline. Where there are disturbances in the ducts, such as turns, expansions, or contractions, the velocity profile across the duct can become asymmetrical as the airflow separates. Refer to the 90-degree mitered elbow in Figure 5-6.

The flow will return to a normal velocity profile after a disturbance, if there is sufficient length of straight duct (3 to 20 diameters) to allow the velocity distribution to regain uniformity, depending on duct size and air velocity.

### 5.5.1 Duct Fittings

The total pressure loss characteristics of duct fittings are determined by laboratory testing with a sufficient length of straight duct (normally 3 to 10 diameters) before and after the fitting to ensure that a uniform velocity profile is established. Where fittings are closely spaced, the pressure losses will be significantly higher than the sum of the losses for the individual fittings, due to the effect of the non-uniform velocity distribution entering the next fitting and the loss of any static pressure regain associated with the preceding fitting. This situation becomes a real problem when attempting to balance HVAC systems with closely-spaced duct fittings.

In addition to the separation of flow caused by turbulence at the heel of the mitered elbow in Figure 5-6, the airflow can even separate in a gradual expansion fitting as shown in Figure 5-5. This condition is very difficult to predict when designing a duct system.

### 5.5.2 Flow Devices

Behavior of flow devices such as a butterfly-balancing damper also interacts with other elements of a system.

Figure 5-7 shows a control damper in a duct, discharging air from a plenum held at constant pressure. With a longer duct, the damper position does not affect the flow rate until it is about one-quarter closed. Duct length has little effect when the damper is more than half closed. The damper has been taken to exactly close the duct for the 90 degree position (K = $\infty$); the K values for the damper settings are indicated above the horizontal scale.

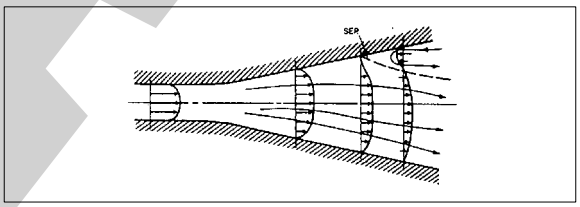

**FIGURE 5–5 SEPARATION IN FLOW IN A DIFFUSER**

## 5.6 DUCT SYSTEM PRESSURES

### 5.6.1 Pressure Relationships

Pressure is the force required to overcome the friction and dynamic losses of HVAC systems. Pressure is produced by a pumping device, which in HVAC systems is a circulating pump or fan.

The only purpose of this pressure-producing device is to furnish a pressure sufficient to overcome the resistance of the system to the flow of the fluid. The pressure produced is indicated by the pressure difference between the fan inlet and the fan discharge. This pressure difference is exactly equal to the system resistance to flow, and in the case of water, the elevation differences for the fluid being pumped. A measurement of the difference between the inlet and discharge pressures of a

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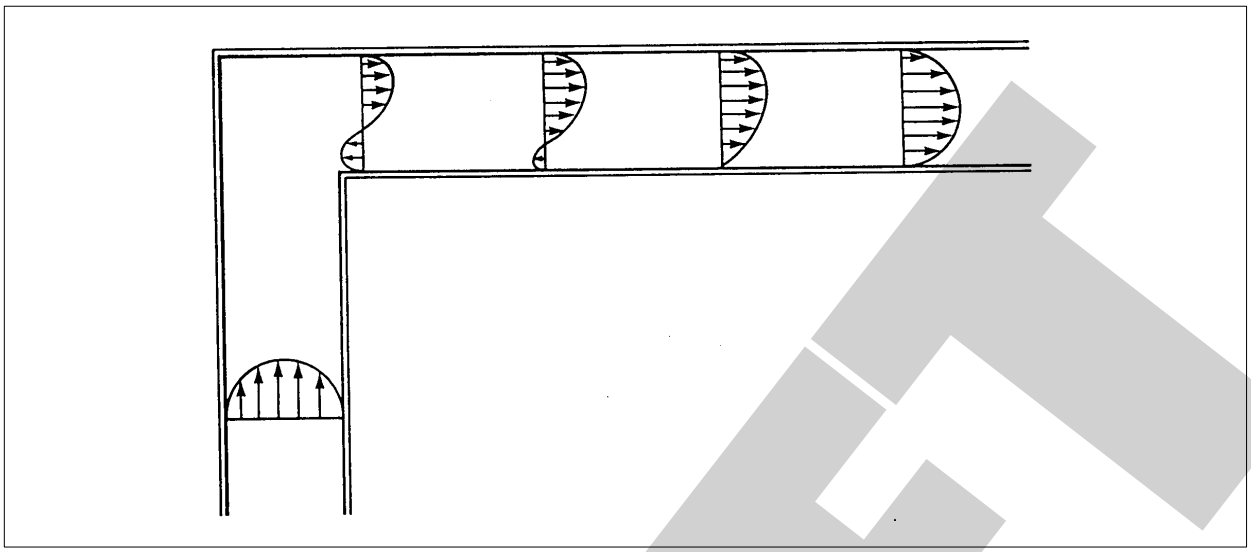

**FIGURE 5–6 CHANGING VELOCITY PROFILES AT A MITERED ELBOW**

pressure device of any kind is a measurement of the system resistance at a particular flow rate.

It is now possible to summarize the forces in a system that produce and resist the fluid flow. These forces (in terms of pressures) can be related to useful equations. They are:

Total pressure (TP) determines how much energy is in the fluid at any point in the system.

Static pressure (SP) in the system at any point is a measure of the potential energy available to produce flow and to maintain that flow against resistance. This pressure is exerted equally in all directions; for example, it is similar to the internal pressure of a balloon.

Velocity pressure ($V_p$) in the system at any point is a measure of kinetic energy resulting from the flow of fluid. Velocity pressure and velocity are directly related by an equation. This pressure is exerted in the direction of flow only.

The relationship between the three pressures at a specific point in a system can be defined by the following equation:

**Equation 5–5**

$$TP = SP + V_p$$

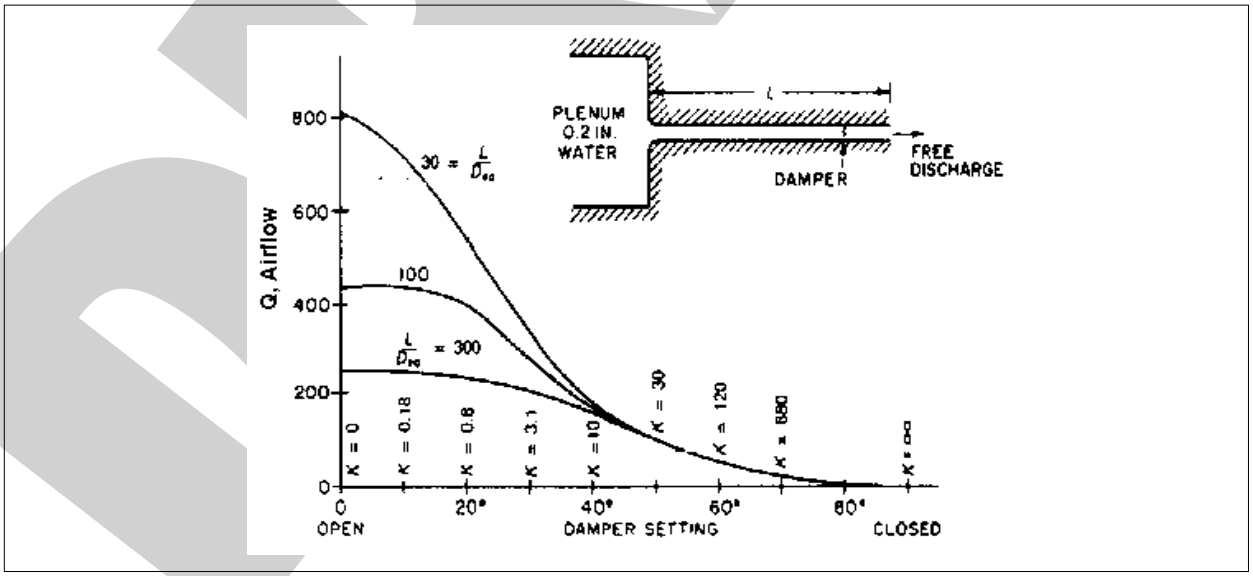


**FIGURE 5–7 EFFECT OF DUCT LENGTH ON DAMPER ACTION**

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Where:

$TP$ = Total pressure – in. wg (Pa)

$SP$ = Static pressure – in. wg (Pa)

$V_p$ = Velocity pressure – in. wg (Pa)

ASHRAE uses different symbols for the above three pressures. However, SMACNA has used the above symbols for years, so it is best to know both. ASHRAE symbols are:

$P_t$ = Total pressure

$P_s$ = Static pressure

$P_v$ = Velocity pressure

### 5.6.2 Pressure Changes

The pressures in HVAC air systems are simpler than those in hydronic systems, because the weight of air in the systems is ignored in most calculations. The resistance to airflow, imposed by a duct system, is overcome by the fan. It supplies the energy in the form of total pressure to overcome this resistance and maintain the necessary airflow. Figure 5-10 illustrates an example of the typical pressure changes in a duct system, with the total pressure and static pressure grade lines in reference to the atmospheric pressure datum line.

In HVAC work, the pressure differences are usually so small that incompressible flow is assumed. Air relationships are expressed at a standard density of 0.075 lbs/ft$^3$ (1.204 kg/m$^3$), and corrections are necessary for significant differences in density due to altitude or temperature. Static pressure ($S_p$) and velocity pressure (VP) are mutually convertible, and can either increase or decrease in the direction of flow. Total pressure (TP), however, always decreases in the direction of airflow.

Referring to Equation 5-5, at any cross-section, the total pressure (TP) is the sum of the static pressure (SP) and the velocity pressure ($V_p$). For all constant-area straight duct sections, the change in static-pressure losses is equivalent to the change in total pressure losses, because there is no change in velocity. If the change in velocity pressure equals zero because the velocity is constant, $\Delta TP = \Delta SP + 0$ or the change in total pressure ($\Delta TP$) = the change in static pressure ($\Delta SP$).

Therefore, for straight sections with constant flow and area, the mean velocity pressure is constant. The pressure losses in straight duct sections are only *friction losses*. Where the straight duct sections have smaller cross-sectional areas, such as duct sections BC and FG in Figure 5-10, the pressure lines fall more rapidly than those of the larger area ducts (pressure losses increase almost as the square of the velocity).

### 5.6.3 Static Regain

Referring to the diagram in Figure 5-10 increases in duct cross-sectional areas, such as at diverging sections C (gradual) and G (abrupt), cause a decrease in velocity and velocity pressure. A continuing decrease in total pressure and an increase in static pressure caused by the conversion of velocity pressure to static pressure. This increase in static pressure is commonly known as *static regain* and is expressed in terms of either the upstream or downstream average airstream velocity pressure.

Total pressure and velocity pressure can be measured with the proper instrumentation.

Velocity pressure cannot be measured directly, but it may be calculated using the following equations:

**Equation 5–6 (I–P)**

$$V_p = \left(\frac{V}{4005}\right)^2$$

Where:

$V_p$ = Velocity pressure – in. wg

$V$ = Velocity – fpm

**Equation 5–6 (SI)**

$$V_p = 0.602V^2$$

Where:

$V_p$ = Velocity pressure – Pa

$V$ = Velocity – m/s

*Example 5-5 (I-P)*

Assume that at the abrupt change G in Figure 5-10, the system total pressure change at that point is negligible. Then using the following pressures and new downstream velocity, calculate the change in static pressure (static regain) and the new system static pressure.

TP (0.40 in. wg) = SP (0.15 in. wg) + $V_p$ (0.25 in. wg), downstream velocity = 800 fpm.

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Using Equation 5-6:

$$V_p = \left(\frac{800}{4005}\right)^2 = 0.04 \text{ in. wg}$$

Using Equation 5-5 ($\Delta TP = 0$):

$$\Delta SP = VP = (0.25 - 0.04)$$

$$\Delta SP = 0.21 \text{ in. wg (regain)}$$

New $SP = 0.15 + 0.21 = 0.36$ in. wg

*Example 5-5 (SI)*

Assume that at the abrupt change G in Figure 5-10, the system total pressure change is negligible. Then using the following pressures and new downstream velocity, calculate the change in static pressure (static regain) and the new system static pressure.

TP (100 Pa) = SP (40 Pa) + $V_P$ (60 Pa) downstream velocity = 4 m/s

Using Equation 5-6:

$$V_p = 0.602(4)^2 = 9.63 \text{ Pa}$$

Using Equation 5-5 ($\Delta TP = 0$)

$$\Delta SP = \Delta VP = (60 - 9.63)$$

$$\Delta SP = 50.37 \text{ Pa (regain)}$$

$$\text{New SP} = 40 + 50.37 = 90.37 \text{ Pa}$$

Words of caution – The above examples were used to illustrate a process only. Actual system static regain is included in the fitting loss coefficient tables that will be explained in the "Dynamic Losses" section.

## 5.7 FRICTION LOSSES

### 5.7.1 Loss Data

The pressure drop in a straight duct is caused by surface friction and varies with the air velocity, the duct size and length, and the interior surface roughness. Friction loss data is most readily obtained from the Duct Friction Loss Charts, Figures A-1 and A-1M. They are based on standard air with a density of 0.075 lb/ft$^3$ (1.204 kg/m$^3$) flowing through average, clean, round, galvanized metal ducts with an absolute roughness factor of 0.0003 ft (0.09 mm). The values obtained from Figures A-1 and A-1M may be used with the correction factors discussed in the next paragraph, for temperatures between 50 and 90 degrees F (10 and 32 degrees C) at altitudes up to 2000 ft (600 m), and for any relative humidity. Beyond those limits, corrections should be made for other than standard air densities and when using other duct materials or flexible duct.

### 5.7.2 Correction Factors

The total friction loss of a duct section also may be obtained by using the *Darcy-Weisback Equation* (Equations 5-7 and 5-8) with the friction factor obtained from the Moody Chart in Figure 5-3.

**Equation 5–7 (I–P)**

$$\Delta TP = f\left(\frac{12L}{D}\right)V_p$$

Where:

$\Delta TP$ = Change in Total Pressure – in. wg

$f$ = Friction factor – dimensionless

$L$ = Duct length – ft

$D$ = Hydraulic diameter – in.

If the hydraulic diameter (D) is in ft, then,

**Equation 5–8 (I–P)**

$$\Delta TP = f\left(\frac{L}{D}\right)V_p$$

**Equation 5–7 (SI)**

$$\Delta TP = f\left(\frac{1000L}{D}\right)V_p$$

Where:

$\Delta TP$ = Change in Total Pressure – Pa

$f$ = Friction factor – dimensionless

$L$ = Duct length – m

$D$ = Hydraulic diameter – mm

*Example 5-6 (I-P)*

Find the friction loss of 50 ft of 36 × 12 in. rigid fibrous glass board duct at a velocity of 2000 fpm using Equation 5-7 or 5-8.

Check the answer using the correction factor from Figure A-2 and data from Figure A-1.

From Table A-1, the absolute roughness ($\in$) for fibrous glass duct, rigid is 0.003-ft using Equation 5-4:

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$$D = \frac{2HW}{H + W} = \frac{2 \times 36 \times 12}{36 + 12} = 18 \text{ in.}$$

$$\text{and } \frac{18}{12} = 1.5 \text{ ft}$$

Relative roughness = $\frac{\epsilon}{D} = \frac{0.003}{1.5} = 0.002$ (both values must be in terms of ft)

Using Equation 5-3:

$$R_e = 8.56DV = 8.56 \times 18 \times 2000 = 3.08 \times 10^5$$

Using the Moody Chart in Figure 5-3, the $\epsilon$/D=0.002 curve crosses the $R_e = 3.08 \times 10^5$ vertical line at a friction factor (f) = 0.024.

Using Equation 5-6:

$$V_p = \frac{V}{4005^2} = \frac{2000}{4005^2} = 0.25 \text{ in. wg}$$

Using Equation 5-8:

$$\Delta TP = f\left(\frac{L}{D}\right)V_p = 0.024 \times \left(\frac{50}{1.5}\right) \times 0.25$$

$$\Delta TP = 0.2 \text{ in. wg friction loss}$$

From Figure A-2 the correction factor for 18 in. diameter medium rough duct at 2000 fpm is 1.47. Using the chart in Figure A-1, the friction loss rate for 18 in. diameter duct at 2000 fpm reads 0.27 in. wg per 100 ft.

$$\Delta TP = \frac{50 \text{ ft} \times 0.27}{100 \text{ ft} \times 1.47} = 0.198 \text{ in. wg}$$

That accurately confirms the above answer.

*Example 5-6 (SI)*

Find the friction loss of 15 m of 900 × 300 mm rigid fibrous glass board duct at a velocity of 10 m/s using Equation 5-7.

Check the answer using the correction factor from Figure A-2 and data from Figure A-1.

From Table A-1, the absolute roughness ( $\epsilon$ ) for fibrous glass duct, rigid is 0.9 mm. Using Equation 5-4:

$$D = \frac{2HW}{H + W} = \frac{2 \times 900 \times 300}{900 + 300} = 450 \text{ mm.}$$

Relative Roughness = $\frac{\epsilon}{D} = \frac{0.9}{450} = = 0.002$

Using Equation 5-3 (SI):

$$R_e = 66.4 = 66.4 \times 450 \times 10 = 2.99 \times 10^5$$

Using the Moody Chart in Figure 5-3, the $\epsilon$/D = 0.002 curve crosses the $R_e = 2.99 \times 10^5$ vertical line at a friction factor (f) of 0.024.

Using Equation 5-6 (SI):

$$VP = 0.602\,V^2 = 0.602(10)^2 = 60.2 \text{ Pa}$$

Using Equation 5-7 (SI):

$$\Delta TP = f\left(\frac{1000L}{D}\right)$$

$$V_p = 0.024 \times \left(\frac{1000 \times 15}{450}\right) \times 60.2$$

$$\Delta TP = 48.16 \text{ Pa friction loss}$$

From Figure A-2, the correction factor for 450 mm diameter medium rough duct at 10 m/s is 1.47. Using the chart in Figure A-1, the friction loss rate for 450 mm diameter duct at 10 m/s reads 2.2 Pa/m.

$$\Delta TP = 15 \text{ m} \times 2.2 \text{ Pa/m} \times 1.4748.5 \text{ Pa friction loss}$$

This accurately confirms the above answer.

Fortunately, the charts in Figure A-1 and A-1M are easy to use, so that only the second step of the above examples needs to be used on a day-to-day basis. However, the examples do show how accurate the duct-friction loss charts are for obtaining pressure losses of straight HVAC duct sections without using more complicated and time-consuming procedures.

The most important thing to remember about straight-duct pressure losses is duct friction losses are in terms of total pressure and are based on the Darcy-Weisback Equation (Equations 5-7 and 5-8). Also, almost all HVAC duct systems fall in the turbulent zone of the Moody Chart (Figure 5-3). To verify this, a Reynolds number of $6.0 \times 10^4$ falls in the middle of the "transition region" in Figure 5-3. Using Equation 5-3 and a small duct diameter (10 in.) as would be found at the end of an HVAC duct system, solve for the velocity:

$$R_e = 8.56DV \text{ and } V = \frac{R_e}{8.56D}$$

$$V = \frac{60,000}{8.56 \times 10} = 701 \text{ fpm}$$

Therefore, only small ducts with low velocities have low Reynolds numbers.

## 5.8 DYNAMIC LOSSES

### 5.8.1 Loss Coefficients

Referring again to the diagram in Figure 5-10, when duct cross-sectional areas are reduced, such as at con-

verging sections B (abrupt) and F (gradual), both the velocity and velocity pressure increase in the direction of airflow, and the absolute value of both the total pressure and static pressure decreases. The pressure losses at point B and F are *dynamic pressure losses.*

Dynamic pressure losses are due to changes in direction or velocity of the air, and occur at transitions, elbows, and duct obstructions, such as dampers. Dynamic losses of a fitting can be expressed as a *loss coefficient.* This is a constant that produces the dynamic pressure losses when multiplied by the velocity pressure, based on the average velocity of the air stream at that point, or by terms of the *equivalent length of straight duct* that has the same loss magnitude. Equivalent lengths are no longer used by SMACNA and ASHRAE for HVAC duct design.

At exit fitting section H, the fitting loss coefficient may be greater than the upstream velocity pressure, equal to the velocity pressure, or less than the velocity pressure. The magnitude of the total pressure loss, as viewed in the local loss section, depends on the discharge Reynolds number and its shape. A simple duct discharge with turbulent flow has a total pressure loss coefficient of 1.0, while the same discharge with laminar flow can have a total pressure loss coefficient greater than 1.0. Thus, the static pressure just upstream of the discharge fitting can be calculated by subtracting the upstream velocity pressure from the total pressure upstream.

The entrance fitting at section A also may have total pressure-loss coefficients less than 1.0 or greater than 1.0. These coefficients are references to the downstream velocity pressure. Immediately downstream of the entrance, the total pressure is simply the sum of the static pressure and velocity pressure. Note that on the suction side of the fan, total pressure and static pressure are always negative with respect to atmospheric pressure. However, velocity pressure is always a positive value.

### 5.8.2 Using Fitting Loss Coefficients

The dynamic loss coefficient "C" is dimensionless and represents the percentage of the velocity head lost at duct transitions, branches, or bends. One *velocity head* is equal to the velocity pressure at the point. Values of the dynamic-loss coefficient for elbows and other duct elements have been determined from laboratory testing of the fittings or calculated, and can be found in the Appendix.

A fitting loss coefficient (C) of 0.75 means that the fitting loses 75 percent of the velocity head (or velocity pressure) at that point in the duct system. In laboratory testing, many sizes or combinations of sizes of fittings or ducts were tested in a set range of duct average velocities. The total pressure losses (corrected for duct friction loss and to standard air conditions) were graphed, and average values were established. Each set of fitting loss coefficients were then established using Equation 5-9. Any static regain or loss due to velocity changes by the velocity decrease in the fitting was greater than the dynamic pressure loss of the fitting.

Velocity pressures obtained by using Equation 5-6 also may be more easily selected from Tables A-4 and A-4M. These velocity pressures are used with the fitting loss coefficients in Equation 5-9 to obtain the total pressure loss of the selected fitting at the indicated conditions. Be sure to verify whether the velocity pressure ($V_P$) is from the upstream or downstream velocity.

**Equation 5–9**

$$TP = C \times V_p$$

Where:

$TP$ = Total pressure; in. wg (Pa)

$C$ = Fitting loss coefficient; dimensionless

$V_p$ = Velocity pressure; in. wg (Pa)

The velocity pressure ($V_P$) used for rectangular duct fittings must be obtained from the average velocity (V) obtained by using Equation 5-9: $V = \frac{Q}{A}$

Where different duct cross-sectional areas are involved, letters with or without subscripts are used to denote the area at which the mean velocity is to be calculated, such as A for inlet area, $A_1$ for outlet area, and $A_o$ for orifice area.

Note that any static regain caused by a duct fitting is included in the duct fitting loss coefficient. Do not add additional static regain that is found in some HVAC industry duct design programs.

*Example 5-7 (I-P)*

A 48 × 16 in. duct elbow handling 8600 cfm has a loss coefficient of 0.28. Find the total pressure loss of the elbow.

Using Equation 5-2:

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$$V = \frac{Q}{A} = \frac{8600}{48 \times 16/144} = 1613 \text{ fpm}$$

Using Equation 5-6 (or Table A-4)

$$V_p = \left(\frac{1613}{4005}\right)^2 = 0.16$$

Using Equation 5-9:

$$TP = C \times V_p = 0.28 \times 0.16$$

$TP$ = 0.045 in. wg pressure loss

*Example 5-7: (SI)*

A 1200 × 400 mm duct elbow handling 4300 L/s has a loss coefficient of 0.28. Find the total pressure loss of the elbow.

Using Equation 5-2 (SI):

$$V = \frac{Q}{1000A} = \frac{4300 l/s}{1000(1.2 \times 0.4 \text{ m})} = 8.96 \text{ m/s}$$

Using Equation 5-6 (SI) or Table A-4M:

$$V_p = 0.602\, V^2 = 0.602(8.96)^2 = 48.3 \text{ Pa}$$

Using Equation 5-9:

$$TP = C \times V_p = 0.28 \times 48.3$$

$TP$ = 13.52 Pa pressure loss

*Example 5-8 (I-P)*

A 24 in. diameter 5-pc 90 degree elbow (R/D = 1.5) has a flow of 7200 cfm. Find the total pressure loss of the elbow.

From Table A-7B the loss coefficient is 0.24. Using Equation 5-2:

$$V = \frac{Q}{A} = \frac{Q}{\pi R^2} = \frac{7200 \text{ cfm}}{\pi(1 \text{ ft})^2}$$

$V$ = 2292 fpm

Using Equation 5-6 (or Table A-4):

$$V_p = \left(\frac{2292}{4005}\right)^2 = 0.33$$

Using Equation 5-9:

$$TP = C \times V_p = 0.24 \times 0.33$$

$TP$ = 0.079 in. wg pressure loss

*Example 5-8 (SI)*

A 600 mm diameter 5-pc 90 degree elbow (R/D = 1.5) has a flow of 3600 L/s. Find the total pressure loss of the elbow.

From Table A-7B the loss coefficient is 0.24.

Using Equation 5-2 (SI):

$$V = \frac{Q}{A} = \frac{3.6\, m^3/s}{\pi(0.3m)^2} = 12.73 \text{ m/s}$$

Using Equation 5-6 (SI):

$$V_p = 0.602\, V^2 = 0.602 \times (12.73)^2$$

$V_p$ = 97.56 Pa

Using Equation 5-9:

$$TP = C \times V_p = 0.24 \times 97.56$$

$TP$ = 23.41 Pa pressure loss

## 5.9 BASIC DUCT SIZING

### 5.9.1 Loads

The amount of air flowing in HVAC supply air duct systems is based on the total heat gain/loss of the building or space to be heated, cooled, or ventilated. This text does not include building or space heating and cooling loads, but the sensible heat equation for air will allow some examples and a study of the airflow in HVAC ductwork and its relationship to space demands.

After the loads are calculated for all spaces, the total load is used to select the HVAC equipment. For simplicity, a heating-only air handler may be selected from an air equipment catalog, but the airflow volume (cfm or L/s), the temperature rise (ºF or ºC) and the heat output (Btuh or W) will be based on Equation 5-10.

**Equation 5–10 (I–P)**

$$Q = 1.08 \times cfm \times \Delta t$$

Where:

$Q$ = Heat flow (Btuh)

$cfm$ = Airflow (cubic ft per minute)

$\Delta t$ = Temperature difference (ºF)

**Equation 5–10 (SI)**

$$Q = 1.23 \times L/s \times \Delta t$$

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Where:

$Q$ = Heat flow (watts)

$L/s$ = Airflow (liters per second)

$\Delta t$ = Temperature difference (°C)

*Example 5-9 (I-P)*

A heating unit has an airflow rating of 4000 cfm and an output of 150,000 Btuh. Find the final discharge temperature with 75 degree F inlet air.

Using Equation 5-10:

$$Q = 1.08 \times cfm \times \Delta t$$

$$\Delta t = \frac{Q}{1.08 \times cfm} = \frac{150,000}{1.08 \times 4000}$$

$\Delta t$ = 34.7°F

Final temperature = 75° + 34.7° = 109.7°F

*Example 5-9 (SI)*

A heating unit has airflow of 1000 L/s and an output of 50 kW. Find the final discharge temperature with 24 degree C inlet air.

Using Equation 5-10:

$$Q = 1.23 \times l/s \times \Delta t$$

$$\Delta T = \frac{Q}{1.23 \times l/s} = \frac{50kW \times 1000W/kW}{1.23 \times 2000l/s}$$

$\Delta t$ = 20.3°C

Final temperature = 24° + 20.3°= 44.3°C

### 5.9.2 HVAC Unit/Duct Relationship

The selection of HVAC units is important to the design of the connected duct systems. A lower airflow will result in higher temperature rises. The air velocity and length of the duct system also affects the temperature losses through the duct walls as well as the straight duct and fitting pressure losses.

Once the HVAC air handling unit has been selected, the individual space loads are proportioned from the total airflow and the airflow for each section is marked on the duct layout. The range of the HVAC unit external static pressure may limit the velocities and pressure drops that may be used in the individual duct sections. For lower pressure systems, a pressure drop of 0.1 in. wg per 100 in. (0.8 Pa/m) is a good place to start.

Measure the longest duct run (an assumption is made that it has the highest pressure drop). For a rough estimate, double or triple the duct run length— this will aproximately account for the losses caused for typical fittings in a duct run— and multiply by 0.1 in. wg/100 ft (0.8 Pa/m) and compare with the available external static pressure of the HVAC unit.

Example 5-10 (I-P)

An air handling unit has an external static pressure range of 0.50 to 1.50 in. wg depending on the fan rpm and motor horsepower. Estimate if a 225 ft duct run with many fittings would be feasible.

225 ft × 4 (1 + 3) = 900 ft equivalent length

$$\frac{900 \text{ ft} \times 0.1 \text{ in.wg}}{100 \text{ ft}} = 0.9 \text{ in. wg}$$

The estimated system loss falls about half way within the HVAC unit pressure capacity.

Example 5-10 (SI)

An air handling unit has an external static pressure of 125 to 375 Pa depending on the fan rpm and motor power (kW). Estimate if an 80 m duct run with many fittings would be feasible.

80 m × 4 (1 + 3) = 320 m equivalent length

320 m × 0.8 Pa/m = 256 Pa

The estimated system loss falls about halfway within the HVAC unit pressure capacity.

### 5.9.3 Friction Loss Chart

Figures 5-8 and 5-9 are part charts from Figures A-1 and A-1M. If data for any two conditions of the duct are known, the other two unknowns may be obtained from the charts. The shaded area indicates the recommended design range.

*Example 5-11 (I-P)*

Using Figure A-1 find the duct diameter and airflow velocity for 700 cfm at 0.1 in. wg/100 ft duct friction loss.

The plotted solution can be found in Figure A-1. The duct velocity = 900 fpm and the duct size = 12 in. diameter.

*Example 5-11 (SI)*

Using Figure A-1M, find the duct diameter and airflow velocity for 200 L/s at 0.9 Pa/m duct friction loss.

The plotted solution can be found in Figure A-1M. The duct velocity = 4.2 m/s and the diameter = 250 mm.

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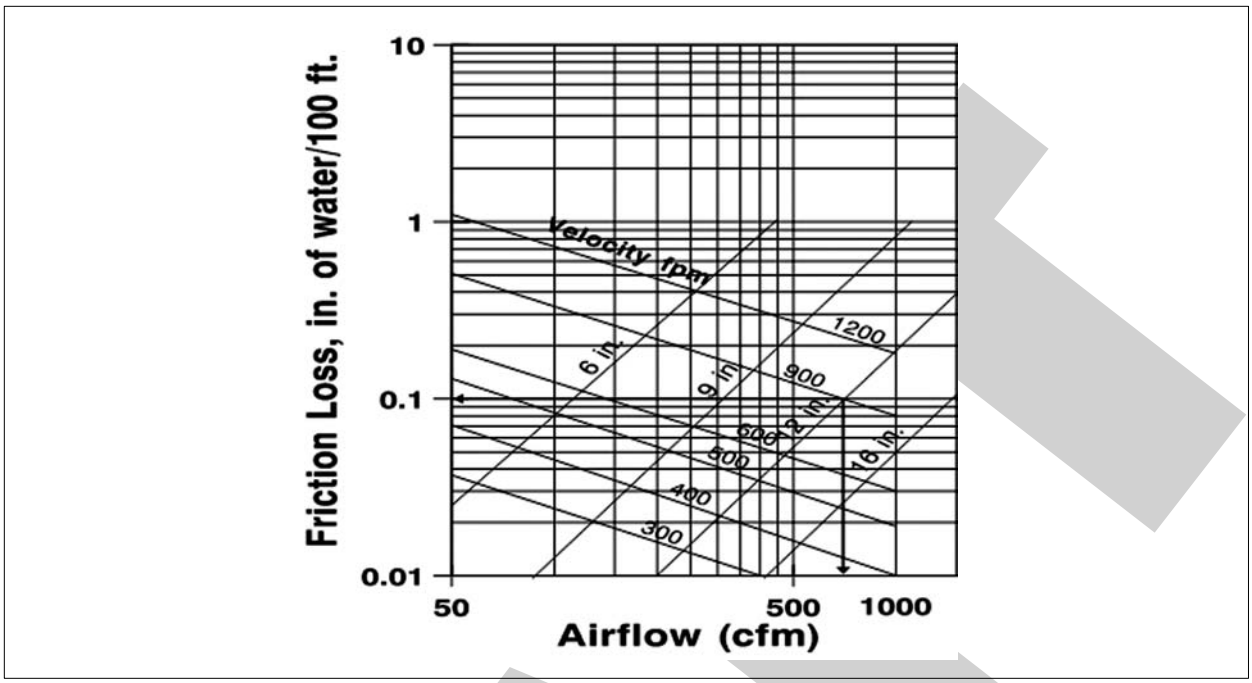


**FIGURE 5–8 PART DUCT FRICTION LOSS CHART (I–P)**

### 5.9.4 Correction Charts

A round duct section has a hydraulic diameter equal to its geometric diameter. However, the *hydraulic diameter* of a non-circular section (such as a rectangular duct or flat-oval duct) equals the geometric diameter of an *equivalent circular* duct as found in Tables A-2 and A-2M. Circular equivalents for flat-oval sizes may be found in Tables A-3 and A-3M.

The friction loss rates found in Figures A-1 and A-1M are based on an absolute roughness of 0.0003 ft (0.09 mm). By using Table A-1 and Figure A-2, a *duct friction loss correction factor* may be obtained for use with other duct construction methods or materials. Additional correction factors for altitude and temperature may be found in Figure A-4, Tables A-17, and A17M.

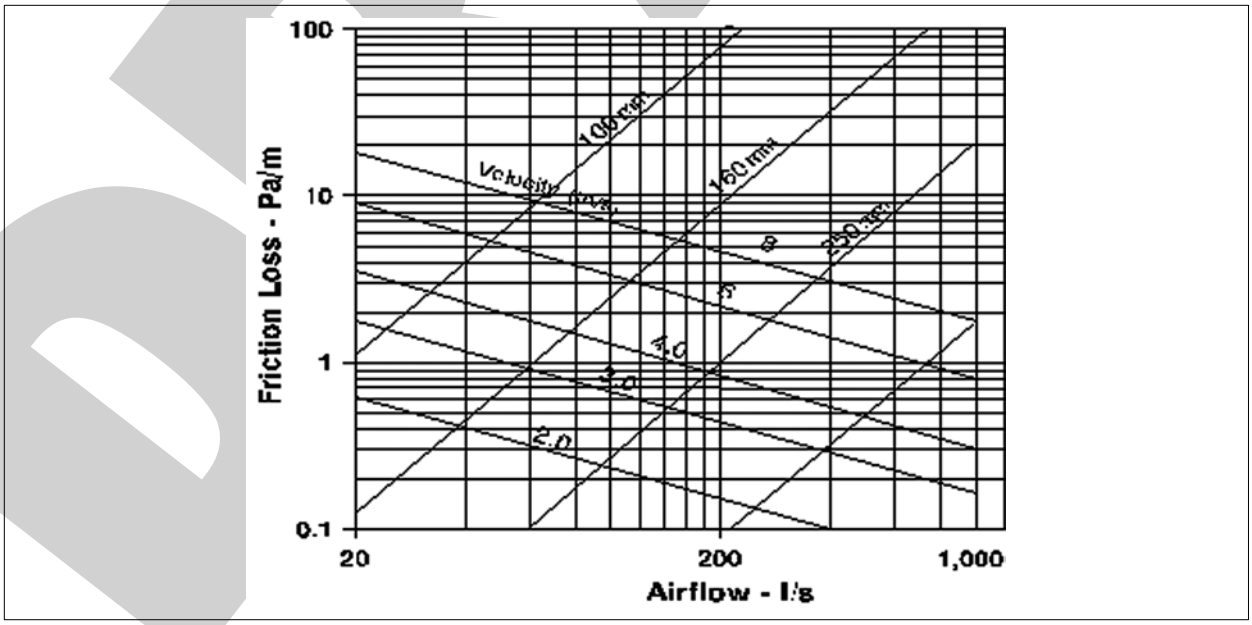


**FIGURE 5–9 PART DUCT FRICTION LOSS CHART (SI)**

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*Example 5-12 (I-P)*

The duct shown in Figure A-1 is 500 ft underground, made of concrete, and is at an altitude of 6000 ft. The airflow temperature is 70 degree F.

From Table A-1, the roughness category is "rough". From Figure A-2, the DFLCF for rough 12 in. diameter duct at 900 fpm is 1.92. From Table A-17, the ADCF for 6000 ft at 70 degree F is 0.80.

$$\text{Duct friction loss} = \frac{500 \text{ ft} \times 0.1 \text{ in.wg}}{100 \text{ ft} \times 1.92 \times 0.80}$$

Duct friction loss = 0.768 in. wg

*Example 5-12 (SI)*

The duct shown in Figure 5-9 is 150 m underground, made of concrete and is at an altitude of 1750 m. The airflow temperature is 20 degree C.

From Table A-1, the roughness category is "rough". From Figure A-2M, the DFLCF for rough 250 mm diameter duct at 4 m/s is 1.94. From Table A-17M, the ADCF for 1750 m at 20 degree C is 0.82.

Duct friction loss =
150 m × .08 Pa/m × 1.94 × 0.82

Duct friction loss = 190.9 Pa

## 5.10 DUCT CONFIGURATIONS

### 5.10.1 Round Ductwork

Round ducts have a smaller amount of duct surface than rectangular ducts carrying the same volume of air at the same velocity. However, there are many other factors that the designer must consider before sizing the duct system. Most often the duct height can be a major reason to use rectangular duct with a fairly high aspect ratio. Also, when there is a group of ducts running together, tight turns generally can be made more efficiently using rectangular elbows with turning vanes.

Another factor is the type of duct fabrication equipment used by the installing contractor. Many will use round duct, while others will prefer rectangular ductwork. If acoustic lining is required, round or rectangular fibrous glass, or lined sheet metal ducts must be used. Round ductwork generally has fewer joints and tighter seams; therefore it has lower overall leakage rates than rectangular ductwork.

Round ductwork may be sized directly from Figures A-1 and A-1M. Note that as the duct diameters and air volumes increase, the duct velocities also increase at a specific friction loss rate. For example, a friction loss rate of 0.2 in. wg/100 ft (1.7 Pa/m) is near the center of the shaded zone until 20,000 cfm (10,000 L/s). At a little over 100,000 cfm (50,000 L/s) the friction loss rate line leaves the shaded zone, and the duct velocity at that point has increased to 4000 fpm (20 m/s).

### 5.10.2 Rectangular Ductwork

Ductwork for HVAC duct systems is usually sized as round ductwork first. Then, if rectangular ducts are desired, duct sizes are selected to provide flow rates equivalent to those of the round ducts originally selected. Tables A-2 and A-2M give the circular equivalents of rectangular ducts for equal friction and flow rates. Note that the mean velocity in a rectangular duct will be less than in its circular equivalent.

Rectangular duct sizes should not be calculated directly from the actual duct cross-sectional areas: Tables A-2 and A-2M should be used. If the tables are not used, the resulting duct sizes will be smaller, with a greater mean velocity and friction loss for the given airflow.

*Example 5-13 (I-P)*

Using the duct size in Figure 5-8, select a rectangular duct size that will not exceed 8 in. in height.

Using the 12 in. diameter duct from Figure 5-8, Table A-2 indicates under the "8 in. column": 11.8 for 15 and 12.2 for 16. As most designers and fabricators use even numbers for standard duct sizes, the selected rectangular duct size would be 16 × 8 in.

*Example 5-13 (SI)*

Using the duct size in Figure 5-9, select a rectangular duct size that would not exceed 150 mm in height.

Using the 250 mm diameter duct from Figure 5-9, Table A-2M, indicates under the "150 mm column": 245 for 350 and 260 for 400. As most designers and fabricators use standard duct sizes, the selected rectangular duct size would be either 350 × 150 mm or 400 × 150 mm.

In the above examples, the duct sizes selected could be smaller (slightly increasing the pressure drop), or larger (slightly lowering the pressure drop). As the system designer sizes the duct sections and selects terminal outlet devices, other factors may affect the duct sec-

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tion sizes. Most often, an attempt is made to change only one dimension of each duct section as sizes progressively change from one end of a duct system to the other.

### 5.10.3 Flat–Oval Ductwork

There has been little independent research on flat-oval ductwork other than that done by the duct manufacturers. Tables A-3 and A-3M give the circular equivalents for many sizes of flat-oval ducts needed to obtain the duct friction losses.

Because little dynamic fitting loss data for flat-oval fittings are available, use data for rectangular fittings as an approximation. This assumes that the shape of the fitting section does not influence the flat-oval fitting pressure losses.

### 5.10.4 Flexible Ductwork

Flexible ducts are very versatile and are made from many different metallic, fabric, or composition materials. Table A-1 places all flexible ducts into the "rough" category (when fully extended) and Figure A-3 gives an additional correction factor for percent of compression length. Additional pressure losses must be added for bends (use the closest comparable round radius elbow).

Unfortunately, many workers installing flexible ductwork will compress long lengths (instead of cutting off the excess), make exceedingly sharp bends, and install long runs instead of using rigid duct and a short flexible connection. All of this creates so much pressure loss, that the airflow at the end often is insufficient.

*Example 5-14 (I-P)*

Four 25 ft lengths of 10 in. flexible duct are connected to a 10 in. diffuser rated at 440 cfm with a pressure drop of 0.12 in. wg. The flexible duct is compressed 30 percent and has the equivalent of five tight 90 degree bends. Determine the inlet static pressure to obtain full airflow from the diffuser.

a. If only 0.25 in. wg is available, calculate the approximate new duct size and velocity.

b. From Figure A-1, velocity = 800 fpm, friction = 0.1 in. wg/100 ft.

   From Table A-1 and Figure A-2, DFLCF = 1.94

   From Table A-3, the correction factor is 4.

   Duct pressure loss =
   $$\frac{100 \text{ ft} \times 0.1 \text{ in.wg}}{100 \text{ ft} \times 1.94 \times 4} = 0.776 \text{ in.wg.}$$

   $V_p$ for 800 fpm (Table A-4) = 0.04 in. wg

   TP = C × $V_P$ = 5 elbows (Table A-7A;

   RD = 0.5) × 0.71 × 0.04 = 0.142 in. wg

   Totals:

| | |
|---|---|
| Duct pressure loss | 0.776 in. wg |
| Elbow pressure loss | 0.142 in. wg |
| 10 in. diam. Diffuser (given) | 0.120 in. wg |
| Total inlet pressure required | 1.038 in. wg |

c. Pressure available =
   0.25 - 0.12 = 0.13 in. wg

   Equivalent length =
   $$100 \times \frac{0.776 + 0.142}{0.776} = 118.3 \text{ ft}$$
   $$\frac{\text{Pressure drop}}{100\text{ft}} = \frac{0.0142 \text{ in.wg}}{100 \text{ ft}}$$

   Using Figure A-1, duct size=15 in. at 360 fpm

*Example 5-14 (SI)*

Thirty meters of 250 mm flexible duct is connected to a 250 mm diffuser rated at 200 L/s with a pressure drop of 30 Pa. The flexible duct is compressed 30 percent and has the equivalent of five tight 90 degree bends. Determine the inlet static pressure to obtain full airflow from the diffuser.

a. If only 65 Pa is available, calculate the new duct size and velocity.

b. From Figure A-1M, velocity = 4 m/s, friction = 0.8 Pa/m

   From Table A-1M and Figure A-2M, DFLCF = 1.94

   From Figure A-3M, the correction factor is 4

   Duct pressure loss =
   30m × 0.8 Pa/m × 1.94 × 4 = 186.2 Pa

   $V_p$ for 4 m/s (Table A-4M) = 9.6 Pa

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TP = C × Vp = 5 elbows (Table A-7A; R/D= 0.5) × 0.71 × 9.6 = 34.1 Pa

Totals:

| | |
|---|---|
| Duct pressure loss | 186. 2 Pa |
| Elbow pressure loss | 34.1 Pa |
| 250 mm. diam. Diffuser (given) | 30.0 Pa |
| Total inlet pressure required | 250.3 Pa |

c. Pressure available = 65 Pa - 30 Pa = 35 Pa

Equivalent Length =

$$100 \times \frac{0.776 + 0.142}{0.776} = 118.3 \text{ ft}$$

$$\frac{\text{Pressure drop}}{100 \text{ ft}} = \frac{35 \text{ Pa}}{35.5 \times 1.94 \times 4} = 0.127 \text{ Pa/m}$$

Using Figure A-1M, duct size = 375 mm at 1.8 m/s

The above examples illustrate what often happens after a system is installed. When it is possible, a much larger flexible duct is used to achieve the needed airflow because of obstructions, etc. that were not shown on the plan.

## 5.11 DUCT FITTINGS

### 5.11.1 Fitting Loss Coefficients

Duct fitting loss coefficients are obtained from data developed by testing various duct fittings in laboratories, in different combinations of duct sizes, and under varying conditions. Pitot tubes or other measuring devices are placed upstream and downstream of the test fitting. The duct friction losses from the measuring devices to the centerline of the fitting being tested are deducted from the total measured pressure drop.

These net pressure drops are used with related velocity pressures in Equation 5-9 ($T_p = C \times V_p$) to obtain a series of loss coefficients for each condition. Most of the Tables A-7 to A-17 has been developed in this manner.

As this entire loss coefficient data are empirical, beyond the listed values, the table data cannot be extrapolated in any direction. Interpolation between loss coefficient values may be used, and should be reasonably accurate.

To review, duct fitting loss coefficients are in terms of total pressure, and they contain static regain or loss that might have occurred in the duct system at that fitting under the conditions that were present.

### 5.11.2 Elbows

There are many variations of turning a duct around a corner. Table A-7 has 12 configurations. The last tap at the end of a duct run or branch, as shown in Figure 5-27, also could be considered to be an elbow. Some system designers forget that the pressure loss of a fitting varies with the square of the velocity:

$$TP = C \times V_p$$

$$V_p = \left(\frac{V}{4005}\right)^2$$

$$TP = C \times \left(\frac{V}{4005}\right)^2$$

For metric:

$$TP = C \times V_p$$

$$V_p = 0.602\, V^2$$

$$TP = C \times 0.602\, V^2$$

For example, a 90 degree mitered elbow without turning vanes (C=1.2) near the HVAC unit would have a large pressure loss of 0.47 in. wg (113 Pa) with a velocity of 2500 fpm (12.5 m/s):

$$T_p = C \times V_p = 1.2 \times 0.39 = 0.47 \text{ in. wg}$$

$$T_p = C \times V_p = 1.2 \times 94 = 1,113 \text{ Pa}$$

However, at the end of the system where the velocity is 600 fpm (3 m/s), the pressure loss for that same elbow drops to 0.02 in. wg (6.5 Pa):

$TP = 1.2 \times 0.02 = 0.024$ in. wg ($1.2 \times 5.4 = 6.5$ Pa)

From 0.47 to 0.024 in. wg (113 Pa to 6.5 Pa) is a big difference in pressure drop for the same elbow in the same system.

Similar reasoning applies to the use of a tap similar to Figure 5-27. If a supply air register has a pressure loss of 0.15 in. wg (38 Pa), a pressure loss of 0.01 to 0.02 (2.5 to 5.0 Pa) for the rectangular tap is somewhat insignificant to the total picture. A much higher loss of a similar tap near the HVAC unit does not enter the picture. This is because only the pressure loss of the last outlet at the end of the longest run of the system is used to calculate the total pressure loss of the supply air duct

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system. Often, some system designers use higher-loss duct fittings in shorter duct runs to build-in "system balance."

When using smaller duct fittings at low velocities, a Reynolds number correction factor may need to be applied.

*Example 5-15 (I-P)*

An 18 × 12 in. elbow, H/W = 1.5, R/W = 0.75 (Table A-7A) handles 900 cfm. V = Q/A = 900/1.5 = 600 fpm.

a. Use the chart on page A-19 to determine if calculations need be made for a Reynolds number correction factor (RNCF).

b. Calculate the fitting pressure loss.

c. From Table A-2, the circular equivalent diameter of 18 × 12 in. = 16.0 in.. Using 16 in. diameter and 600 fpm, the elbow falls into the shaded area. Re = 8.56 DV = 8.56 × 16 × 600 = 82,176 = $8.22 \times 10^4$. Using the lower table of Table A-7H:
RNCF = 1.37 (by interpolation).

d. Fitting loss = C × $V_p$ = 0.40 × 0.02 = 0.008 in. wg

Fitting loss (corrected) = 0.008 × 1.37 = 0.011 in. wg

*Example 5-15 (SI)*

A 450 × 300 mm elbow, H/W = 1.5,
R/W = 0.75 (Table A-7H) handles 450 L/s.
V = L/s/1000A = 3.33 m/s.

a. Use the chart on page A-19 to determine if calculations need be made for a Reynolds number correction factor (RNCF).

b. Calculate the fitting pressure loss.

c. From Table A-2M, the circular equivalent diameter of 450 × 300 mm = 400 mm. Using 400 mm and 3.33 m/s, the elbow falls into the shaded area. $R_e$ = 66.4 DV = 66.4 × 400 × 3.33 = 88,445 = $8.84 \times 10^4$. Using the lower table of Table A-7H: RNCF = 1.35 (by interpolation).

d. Fitting loss = C × $V_P$ = 0.40 × 6.7 = 2.68 Pa

Fitting loss (corrected) = 2.68 × 1.35 = 3.62 Pa

From the above examples, one can see that small elbows with low velocities are used either at the end of a long duct run or at the end of small branches. If the above elbows were at the end of a long duct run, the Reynolds number correction only added 0.003 in. wg (0.94 Pa).

Under normal conditions, duct system fabrication and installation will change the system pressures up or down far greater than these Reynolds numbers corrections.

*Example 5-16 (I-P)*

A 5 × 5-in. elbow, R/W = 0.75 (Table A-7H) has a velocity of 3,000 fpm. The velocity can drop to 2000 fpm. Calculate the difference in pressures.

a. From Table A-2: 5 × 5 in. = 5.5 in. diam.

b. $R_e$ = 8.56 × 5.5 × 3000 = 141,240 = $14.12 \times 10^4$
RNCF = 1.15

c. $R_e = 8.56 \times 5.5 \times 2000 = 94{,}160 = 9.42 \times 10^4$

d. RNCF = 1.32 (by interpolation)

e. Fit. loss (3000) = C × $V_p$ = 0.44 × 0.56 = 0.246 in. wg

f. Fit. loss (3000) corrected = 0.446 × .15 = 0.283 in. wg

g. Fit. loss (2000) = C × $V_p$ = 0.44 × 0.25 = 0.110 in. wg

h. Fit. loss (2000) corrected = 0.11 × 1.32 = 0.145 in. wg

In each case, the corrected difference was 0.035 in. wg or greater.

*Example 5-16 (SI)*

A 125 × 125 mm elbow, (R/W = 0.75 (Table A-7H) has a velocity of 15 m/s. The velocity can drop to 10 m/s. Calculate the difference in pressures.

a. From Table A-2M: 125 × 125 mm = 137 mm

b. $R_e$ = 66.4 DV = 66.4 × 137 × 15 = 36,452 = $13.64 \times 10^4$ (15 m/s)

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c. RNCF = 1.16 (by interpolation)

d. $R_e = 66.4\ DV = 66.4 \times 137 \times 10 = 90{,}968 = 9.10 \times 10^4$ (10 m/s)

e. RNCF = 1.34 (by interpolation)

f. Fit. loss (15 m/s) = $C \times V_P = 0.44 \times 135 = 59.4$ Pa

g. Fit. loss (15 m/s) corrected = $59.4 \times 1.16 = 68.9$ Pa

h. Fit. loss (10 m/s)= $C \times V_P = 0.44 \times 60 = 26.4$ Pa

i. Fit. loss (10 m/s) corrected = $6.4 \times 1.34 = 35.4$ Pa

In each case the corrected difference was 9.0 Pa or greater.

From the above Examples 5-15 and 5-16, the RNCF correction of 0.035 in. wg (9 Pa) or greater in Example 5-16 was approximately three times greater than the total fitting losses in Example 5-15.

However, most HVAC system design will rarely involve the use of Reynolds number correction factors for duct elbows because most ductwork is too big and the duct velocities too high. Smooth radius and three- to five-piece round elbows (Table A-7A and B) do not require corrections, nor do most rectangular elbows with a proper throat radius.

### 5.11.3 Transitions

Transitions, which are changes of size of the duct with the airflow in the same direction, can be found in Table A-8 (diverging flow) and Table A-9 (converging flow). As with most duct fitting loss coefficients, care must be exercised to use the velocity pressure of the correct velocity, either upstream or downstream, as noted at the top of each page of fitting loss coefficients. Tables A-8C and D require the use of equations to find the angle θ for use in Table A-8B.

*Example 5-17 (I-P)*

Find the fitting pressure loss for a 24 in. diameter duct diverging to a 36 × 36 in. rectangular duct, "L" = 30 in., upstream velocity = 2000 fpm.

From Table A-8C:

$$\tan\frac{\theta}{2} = \frac{1.13\sqrt{HW} - D}{2L} = \frac{1.13\sqrt{36 \times 36} - 24}{2 \times 30} = 0.278$$

$\tan\left(\frac{\theta}{2}\right) = 0.278$ using the Natural Trigonometric Function Table in the Appendix.

$\frac{\theta}{2} = 15.5^\circ$ and $\theta = 31^\circ$

Using Table A-8B:

$$\frac{A_1}{A} = \frac{36 \times 36}{\pi\,(12)^2} = 2.86$$

C = 0.36 (by interpolation) and $V_p = 0.25$

Fitting Loss = $C \times V_p = 0.36 \times 0.25 = 0.09$ in. wg

*Example 5-17 (SI)*

Find the fitting pressure loss for a 600 mm diameter duct diverging to a 900 × 900 mm rectangular duct, L – 750 mm, upstream velocity = 10 m/s.

From Table A-8C:

$$\tan\left(\frac{\theta}{2}\right) = \frac{1.13\sqrt{900 \times 900} - 600}{2 \times 750} = 0.278$$

Using the Natural Trigonometric Function Table in the Appendix: $= \frac{\theta}{2} = 15.5^\circ$ and $\theta = 31^\circ$

Using Table A-8B:

$$\frac{A_1}{A} = \frac{900 \times 900}{\pi(300)^2} = 2.86$$

C = 0.36 (by interpolation) and $V_P = 60$ PA

Fitting Loss = $C \times V_p = 0.36 \times 60 = 21.6$ Pa

### 5.11.4 Branch Fittings

Chapter 3 discusses the use of less expensive tap fittings for branch ducts and the elimination of "robber" dampers and extractors. A 45 degree entry tap fitting (Table A-11N), when used within the proper range of velocity and airflow volume ratios, has lower fitting loss coefficients than rectangular taps with the dampers or extractors (Tables A-11Q and R). Not only did the "bubble" videotape *Destroying Duct Design Myths* show the turbulence created by these air extractor devices in the branch ducts, but also the turbulence created in the main supply air duct.

Improperly located supply air branch fittings can actually "pull" room air back into the supply air system from the supply outlets due to an area of negative pressure caused by air turbulence. This problem is common in exposed round ducts at outlet connections near

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the supply fan discharge and connections just after an elbow.

Caution should be used with Tables A-10 and A-11. Because of the way different fittings were tested in the laboratories, some use area ratios and velocity ratios, some use area ratios and volume ratios. The range of sizes of divided flow fittings that have been tested are limited.

In many instances, you can find the configuration and the size and volume ratio needed only in rectangular dimensions instead of round. The reverse is also true. If this is the case, you should use the indicated duct. The loss coefficient will be in the same range and will be much closer than attempting to extrapolate from some other limited table.

*Example 5-18 (I-P)*

Compare a 45 degree entry tap loss coefficient (Table A-11N) with a rectangular tap having a "robber" damper (Table A-11Q) or extractor (Table A-11R) where $\frac{Q_b}{Q_c} = 0.5$ and $\frac{V_b}{V_c} = 1.0$.

C = 0.74 (45° entry tap)

C = 1.29 (rectangular tap with damper)

C = 1.21 (rectangular tap with extractor)

### 5.11.5 Dampers

Simple butterfly dampers (Tables A-15A and B) do a good job as balancing dampers as long as they are not installed as 100 percent shutoff dampers. The amount of extra pressure drop required of the damper may be predicted including the angle at which it should be set.

*Example 5-19 (I-P)*

2000 cfm flows through a 10 ft long 24 × 12 in. branch duct with a damper to a 24 × 12 in. register ($\Delta P = 0.12$ in. wg) at design conditions. The SP available at the duct connection is 0.3 in. wg. Find the extra pressure drop needed and the angle of the damper to attain it.

$$V = \frac{Q}{A} = \frac{2000}{24 \times 12/144} = 1000 \text{ fpm}$$

$V_p = 0.06$ in. wg

From Table A-2:

24 × 12 in. duct = 18.3 in. diameter.

From Figure A-1:

Duct pressure drop = $\frac{0.09 \text{ in. wg}}{100 \times 10 \text{ ft}} = 0.009$ in. wg

Damper (open) pressure drop = $C \times V_P = 0.04 \times 0.06 = 0.002$ in. wg

Register (given) = 0.12 in. wg

Totals:

| | |
|---|---|
| Duct pressure loss | 0.009 in. wg |
| Damper loss (open) | 0.002 in. wg |
| Register loss (given) | 0.120 in. wg |
| Total loss (normal) | 0.131 in. wg |

Express pressure = 0.300 – 0.131 = 0.169 in. wg

$$\text{"}\times\text{"} = \frac{\Delta P}{V_p} = \frac{0.169 + 0.002}{0.06} = 2.85$$

From Table A-15B, Angle setting = 27.9 degrees

*Example 5-19 (SI)*

1000 L/s flows through a three meter long 600 × 300 mm branch duct with a damper to a 600 × 300 mm register ($\Delta$ = 30 Pa) at design conditions. The SP available at the duct connection is 75 Pa. Find the extra pressure drop needed and the angle of the damper setting to attain it.

$$V = \frac{l/s}{1000A} = \frac{1000}{(1000(0.6 \times 0.3)} = 5.56 \text{ m/s;}$$

$V_p = 18.6$ Pa

From Table A-2M:

600 × 300 mm duct = 457 mm diameter

From Figure A-1M:

Pressure drop = 0.85 Pa/m x 3 m = 2.55 Pa

Damper loss (open)
TP = C x $V_p$ = 0.04 x 18.6 = 0.74 Pa

Register loss (given) = 30.00 Pa

Total loss (normal) = 33.29 Pa

Excess pressure = 75 – 33.29 = 41.71 Pa

New Damper "C" $= \frac{\Delta P}{V_p} = \frac{41.71 + 0.74}{18.6} = 2.28$

From Table A-15B, Angle Setting = 25.1degrees

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## 5.12 SYSTEM PRESSURE CHANGES

### 5.12.1 Changes Caused by Flow

The resistance to airflow imposed by a duct system is overcome by the fan which supplies the energy, in the form of total pressure, to overcome this resistance and maintain the necessary airflow. Figure 5-10 illustrates an example of the typical pressure changes in a duct system with the total pressure and static pressure grade lines in reference to the atmospheric pressure.

At any cross-section, the total pressure (TP) is the sum of the static pressure (SP) and the velocity pressure ($V_p$).

**Equation 5–11**

$$TP = SP + V_p$$

Where:

$TP$ = Total Pressure – in. wg (Pa)

$SP$ = Static Pressure – in. wg (Pa)

$V_p$ = Velocity Pressure – in. wg (Pa)

In HVAC work, the pressure differences are ordinarily so small that incompressible flow is assumed. Relationships are expressed for air at standard density of 0.075 lb/ft$^3$ (1.204 kg/m$^3$), and corrections are necessary for significant differences in density due to attitude or temperature. Static pressure and velocity pressure are mutually convertible and can either increase or decrease in the direction of flow. Total pressure, however, always decreases in the direction of airflow.

### 5.12.2 Straight Duct Sections

For all constant-area straight duct sections, the static pressure losses are equivalent to the total pressure losses. Thus, for a section with constant flow and area, the mean velocity pressure is constant. These pressure losses in straight duct sections are termed *friction losses*. Where the straight duct sections have smaller cross-sectional areas, such as duct sections BC and FG, the pressure lines fall more rapidly than those of the larger area ducts (pressure losses increase almost as the square of the velocity).

### 5.12.3 Reducers

When duct cross-sectional areas are reduced, such as at converging sections B (abrupt) and F (gradual), both the velocity and velocity pressure increase in the direction of airflow and the absolute value of both the total pressure and static pressure decreases. The pressure losses are due to changes in direction or velocity of the air, and occur at transitions, elbows, and duct obstructions, such as dampers. Dynamic losses can be expressed as a loss coefficient (the constant which produces the dynamic pressure losses when multiplied by the velocity pressure), or by the equivalent length of straight duct which has the same loss magnitude.

### 5.12.4 Increasers

Increases in duct cross-sectional areas, such as at diverging sections C (gradual) and G (abrupt), cause a decrease in velocity and velocity pressure, a continuing decrease in total pressure, and an increase in static pressure caused by the conversion of velocity pressure to static pressure. This increase in static pressure is commonly known as *static regain* and is expressed in terms of either the upstream or downstream velocity pressure.

### 5.12.5 Exit Fittings

At the exit fitting, section H, the total pressure loss coefficient may be greater than one upstream velocity pressure, equal to one velocity pressure, or less than one velocity pressure. The magnitude of the total pressure loss may be seen in the local loss section, which depends on the discharge Reynolds number and its shape. A simple duct discharge with turbulent flow has a total pressure loss coefficient of 1.0, while the same discharge with laminar flow can have a total pressure loss coefficient greater than 1.0. Thus, the static pressure just upstream of the discharge fitting can be calculated by subtracting the upstream velocity pressure from the total pressure upstream.

### 5.12.6 Entrance Fittings

The entrance fitting at section A may also have total pressure loss coefficients less than 1.0 or greater than 1.0. These coefficients are referenced to the downstream velocity pressure. Immediately downstream of the entrance, the total pressure is the sum of the static pressure and velocity pressure. Note that on the suction side of the fan, the static pressure is negative with respect to the atmospheric pressure. However, velocity pressure is always a positive value.

### 5.12.7 System Pressures

It is important to distinguish between static pressure and total pressure. *Static pressure* is commonly used as the basic pressure for duct system design and construction, but *total pressure* determines the actual amount

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of energy that must be supplied to the system to maintain airflow. Total pressure always decreases in the direction of airflow. But static pressure may decrease, then increase in direction of airflow (as it does in Figure 5-10), and may go through several more increases and decreases in the course of the system. It can become negative (below atmospheric) on the discharge side of the fan, as demonstrated by Points G and H (in Figure 5-10). The distinction must be made between static pressure loss (sections BC or FG) and static pressure change as a result of conversion of velocity pressure (section C or G).

### 5.12.8 Fan Pressures

The total resistance to airflow is noted by $\Delta TP_{sys}$ in Figure 5-10. Since the prime mover is a vane-axial fan, the inlet and outlet velocity pressures are equivalent; for example, $\Delta TP_{sys} = \Delta SP_{sys}$. When the prime mover is a centrifugal fan, the inlet and outlet areas are usually not equal, thus the suction and discharge velocity pressures are not equal, and obviously $\Delta TP_{sys} \Delta SP_{sys}$. If you need to know the static pressure requirements of a centrifugal fan, and the total pressure requirements are known, the following relationship may be used:

**Equation 5–12**

$$Fan\ SP = TP_d - TP_s - V_{p_d}$$

$$\text{or } SP = TP - V_p$$

$$Fan\ SP = SP_d - TP_s$$

where: the subscripts "d" and "s" refer to the discharge and suction sections, respectively, of the fan. The inlet and outlet "System Effect," due to the interaction of the fan and duct system connections, are not shown in this illustrative example. Only actual system resistances are shown.

### 5.12.9 Return Air System Pressures

Some HVAC designers believe that return air in a duct system is "sucked back" by the fan; therefore, the ductwork and fittings do not need to follow good design practices. They do not see the need for turning vanes for mitered elbows, smooth airflow into the fan inlet, and use "panned" joists in residential systems. However, this attitude is very wrong.

A diagram is shown in Figure 5-11 of a simple return air system. Converting to absolute pressures, an atmo-

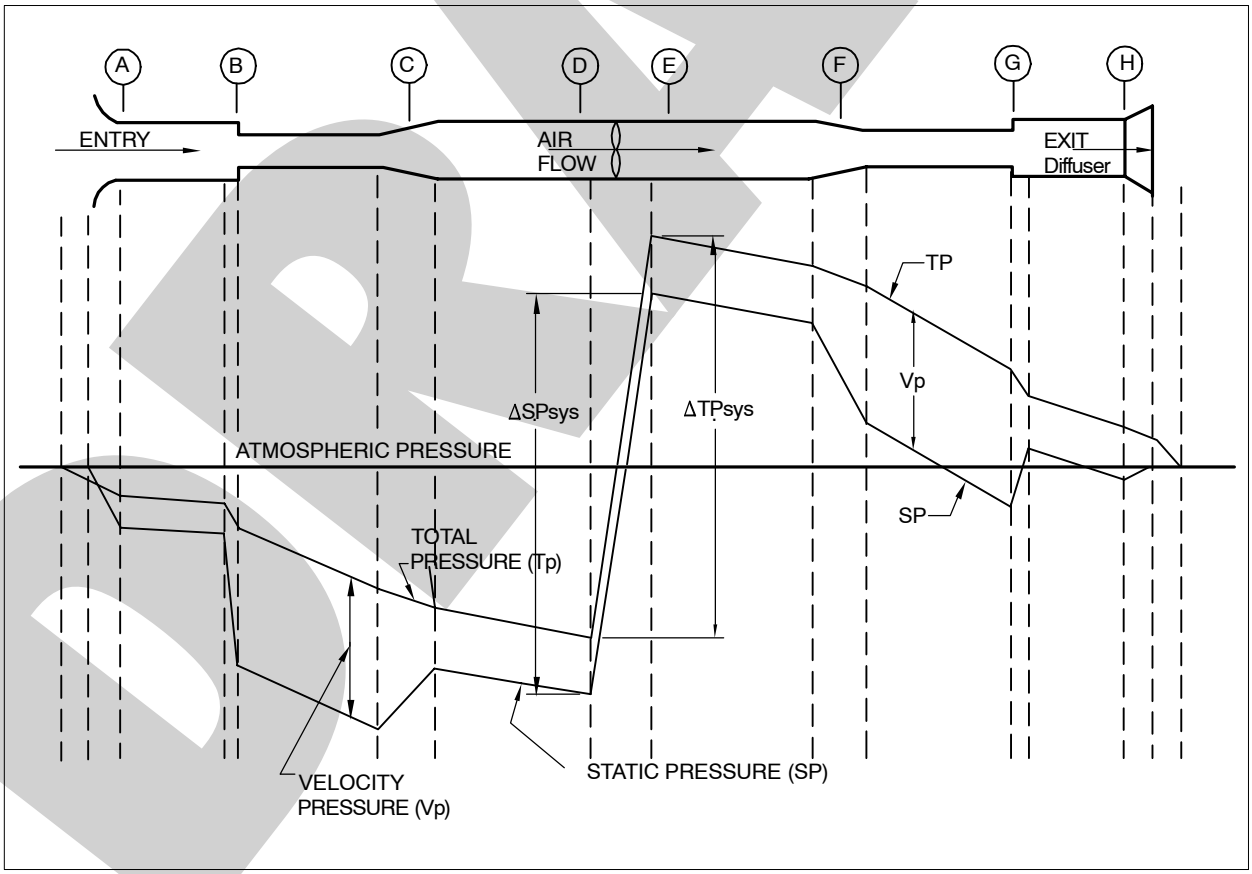


**FIGURE 5–10 PRESSURE CHANGES DURING FLOW–IN DUCTS**

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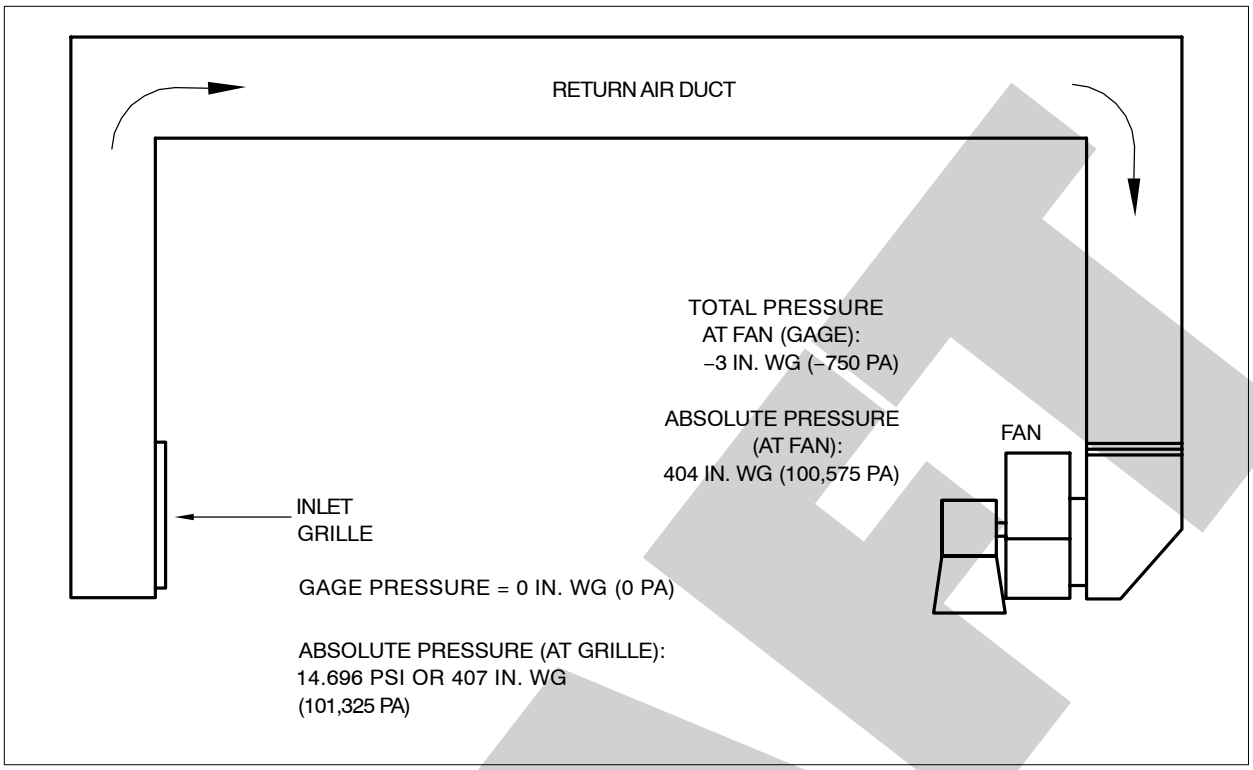


**FIGURE 5–11 RETURN AIR DUCT EXAMPLE**

spheric pressure of 14.7 psi or 407 in. wg (101,325 Pa) at the inlet grille acts as a pressure device (fan or pump) to PUSH the air through the duct to the lower pressure end (404 in. wg – 100,575 Pa) at the system fan inlet. The total pressure drop of 3 in. (750 Pa) could be reduced substantially, if the 90 degree mitered elbows had turning vanes and the fan inlet connection was better designed. In reality, a return or exhaust air duct behaves exactly as a supply duct with atmospheric pressure pushing the air to the lower pressure area created by the fans suction.

## 5.13 STRAIGHT DUCT LOSSES

Pressure drop in a straight duct section is caused by surface friction and varies with the velocity, the duct size and length, and the interior surface roughness. Friction loss is most readily determined from Air Duct Friction Charts, Figures A-1 and A-1M. These are based on standard air with a density of 0.075 lb/ft$^3$ (1.204 kg/m$^3$) flowing through average clean, round, galvanized metal ducts with beaded-slip couplings on 48 in. (1220 mm) centers, equivalent to an absolute roughness of 0.0003 ft (0.09 mm).

In HVAC work, the values from the friction loss charts and the SMACNA Duct Design Calculators may be used without correction for temperatures between 50 to 140 degree F (10 to 60ºC) and up to 2000 ft (600 m) altitude. Figure A-4 and Tables A-17 and A-17M may be used where air density is a significant factor, such as at higher altitudes or where high-temperature air is being handled to correct for temperature and altitude. The actual air volume (cfm or L/s) is used to find the duct friction loss using Figures A-1 and A-1M. This loss is multiplied by the correction factor(s) to obtain the adjusted duct friction loss.

### 5.13.1 Circular Equivalents

HVAC duct systems usually are first sized as round ducts. Then, if rectangular ducts are desired, duct sizes are selected to provide flow rates equivalent to those of the round ducts originally selected. Tables A-2 and A-2M give the circular equivalents of rectangular ducts for equal friction and airflow rates for aspect ratios not greater than 11.7:1. Note that the mean velocity in a rectangular duct will be less than the velocity for its circular equivalent.

Multiplying or dividing the length of each side of a duct by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 × 26 in. (2030 × 660 mm) duct is required, it will be twice that of a 40 ×13 in. (1015 × 330 mm) that has a circular equivalent of 24 in. (610 mm) diameter or 2 × 24 = 48 in. (1220 mm) diameter.

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Rectangular ducts should not be sized directly for actual duct cross-sectional areas. Instead, Tables A-2 and A-2M must be used or the resulting rectangular duct sizes will be smaller creating greater duct velocities for a given airflow.

## 5.14 DYNAMIC LOSSES

Wherever turbulent flow is present, due to sudden changes in the direction or the magnitude of the velocity of the airflow, a greater loss in total pressure takes place than would occur in a steady flow through a similar length of straight duct having a uniform cross-section. The amount of this loss in straight-duct friction is termed *dynamic loss*. Although dynamic losses may be assumed to be caused by changes in area actually occupied by the airflow, they are divided into two general classes for convenience: (1) those caused by *changes in direction* of the duct and (2) those caused by *changes in cross-sectional area* of the duct.

### 5.14.1 Duct Fitting Loss Coefficients

The dynamic loss coefficient "C" is dimensionless and represents the number of velocity heads lost at the duct transition or bend, in terms of velocity pressure. Values of the dynamic-loss coefficient for elbows and other duct elements have been determined by laboratory testing and can be found in the tables in the Appendix. It should be noted, however, that absolutely reliable dynamic-loss coefficients are not available for all duct elements and the information available for pressure losses due to area changes is generally restricted to symmetrical-area changes.

Tables A-4 and A-4M, which shows the relationship of velocity-to-velocity pressure for standard air, can be used to find the dynamic pressure loss for any duct element whose *dynamic loss coefficient* "C" is known.

**Equation 5–13**

$$TP = C \times V_p$$

Where:

$TP$ = Total pressure loss (in. wg or Pa)

$C$ = Fitting Loss coefficient

$V_p$ = Velocity Pressure (in. wg or Pa)

The velocity pressure ($V_p$) used for rectangular duct fittings must be obtained from the velocity (V) obtained by using the following equation:

**Equation 5–14 (I–P)**

$$V = \frac{Q}{A}$$

Where:

$V$ = Velocity (fpm)

$Q$ = Airflow (cfm)

$A$ = Cross-sectional Area (sq ft)

**Equation 5–14 (SI)**

$$V = \frac{Q}{A}$$

Where:

$V$ = Velocity (m/s)

$Q$ = Airflow ($m^3/s$) and 1000 L/s = 1 $m^3/s$

$A$ = Cross-sectional Area ($m^2$)

(or)

$$V = \frac{10,000\,Q}{A}$$

Where:

$V$ = Velocity (m/s)

$Q$ = Airflow (L/s)

$A$ = Area ($mm^2$)

In fittings, such as junctions where different areas are involved, letters with and without subscripts are used to denote the area at which the mean velocity is to be calculated, such as "A" for inlet area, "$A_c$" for upstream or "common" duct area, "$A_b$" for branch duct area, "$A_s$" for downstream or "system" duct area, "$A_0$" for orifice area.

Velocity pressure ($V_p$) may be calculated from Equation 5-15 or obtained from Tables A-4 and A-4M.

**Equation 5–15 (I–P)**

$$V_p = \left(\frac{V}{5004}\right)^2$$

**Equation 5–15 (SI)**

$$V_p = 0.602\,V^2$$

Where:

$V_p$ = Velocity Pressure (in. wg or Pa)

$V$ = Velocity (fpm or m/s)

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*Example 5-20 (I-P)*

An elbow in a 24 × 20 in. duct conveying 7000 cfm has a loss coefficient (C) of 0.40. Find the elbow pressure loss.

Using Equation 5-14:

$$V = \frac{Q}{A} = \frac{7000}{24 \times 20/144} = 2,11 \text{ fpm}$$

Using Equation 5-15:

$$V_p = \frac{(2100)^2}{4005} = 0.275 \text{ in. wg}$$

Using Equation 5-14:

$$TP = C \times V_p = 0.40 \times 0.275$$

$$TP = C \times V_p = 0.40 \times 0.275$$

(Elbow pressure loss)

*Example 5-20 (SI)*

An elbow in a 600 × 500 mm duct conveying 3500 L/s has a loss coefficient (C) of 0.40. Find the elbow pressure loss.

Using Equation 5-14:

3500 L/s = 3.5 $m^3/s$ and 600 mm = 0.6 m

500 mm = 0.5 m

(or)

$$V = \frac{1000Q}{A} = \frac{1000 \times 3500}{600 \times 500}$$

$V$ = 11.67 m/s

Using Equation 5-14:

$V_p = 0.602 \times (11.67)^2 = 81.99$ Pa (Use 82)

Using Equation 5-13:

$$TP = C \times V_p = 0.40 \times 82$$

$TP$ = 32.8 Pa

### 5.14.2 Pressure Losses in Elbows

Dynamic loss coefficients for elbows, *see* Table A-7, are nearly independent of the air velocity and are affected by the roughness of the duct walls only in the case of the bends. However, the elbow-loss coefficients in Table A-7 are used with the duct velocity pressure to calculate the "total pressure" loss of each fitting. The additional duct friction loss, if any, for the elbow is included in the calculations for the adjacent straight duct sections by measuring to the centerline of each fitting.

Data now available for losses in compound bends, where two or more elbows are close together, do not warrant refinement of design calculations beyond the sum of the losses for the individual elbows. In reality, the losses may be somewhat more or less than for two bends. Loss coefficients for some normally-used, double-elbow configurations may be obtained from Table A-7.

Loss coefficients for some elbows with angle bends other than 90 degrees may be computed from the table in Note 1 on page A-15. Loss coefficients for elbows discharging air directly into a large space are higher than those given for elbows within duct systems, *see* Table A-13 Figure E.

*NOTE: Splitters and turning vanes are beneficial at the top of stack heads to align airflow into the outlets. However, turning vanes are not normally used in small, low-velocity duct fittings-even square-throated elbows.*

## 5.15 SPLITTER VANES

Smooth-radius rectangular duct elbows (with radius throat and heel) have a reasonably low loss coefficient when the R/W ratio is equal to 1.0 or higher, *see* Table A-7, Figure F. However, most installations do not have room for this configuration and smaller R/W ratios are required. The use of splitter vanes drops the fitting loss coefficient values of these low R/W ratio radius elbows to a minimal amount. The splitter vane spacing may be calculated as shown in Figure 5-12.

*Example 5-20 (I-P)*

A 48 (H) × 24 in. (W) smooth radius elbow has a throat radius of 6 in.. Find the radius of each of two splitter vanes and the fitting loss coefficient.

Using Figure 5-12 and Table A-7, Figure G:

$$\frac{R}{W} = \frac{6}{24} = 0.25$$

From Table A-7. Figure G for two splitter vanes, CR = 0.585

$$R_1 = \frac{R}{CR} = \frac{6}{0.585} = 10.26 \text{ in.}$$

$$R_2 = \frac{R}{CR^2} = \frac{6}{(0.585)^2} = 17.53 \text{ in.}$$

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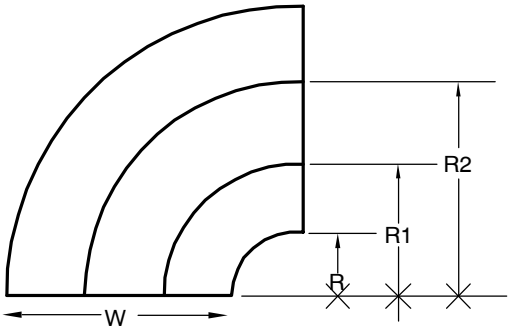
$$\frac{H}{W} = \frac{48}{24} = 2.0$$

From the fitting loss coefficient table for two splitter vanes (opposite R/W = 0.25) and C = 0.04




Elbow with two splitter vanes
(Section View)

**FIGURE 5–12 TO CALCULATE SPLITTER VANE SPACING FOR A SMOOTH RADIUS RECTANGULAR ELBOW**

*Example 5-20 (SI)*

A 1200 (H) × 600 mm (W) smooth radius elbow has a throat radius of 150 mm. Find the radius of each of two splitter vanes and the fitting loss coefficient.

Using Figure 5-12 and Table A-7, Figure G:

$$\frac{R}{w} = \frac{150}{600} = 0.25$$

From Table A-7 Figure G for two splitter vanes, CR = 0.585

$$R_1 = \frac{R}{CR} = \frac{150}{0.585} = 256.4 \text{ mm}$$

$$R_2 = \frac{R}{CR^2} = \frac{150}{(0.585)^2} = 438.3 \text{ mm}$$

$$\frac{H}{W} = \frac{1200}{600} = 2.0$$

From the fitting loss coefficient table for two splitter vanes (opposite R/W = 0.25, C=0.04)

## 5.16 TURNING VANES

### 5.16.1 Single Verses Double Thickness

Duct fitting loss coefficient tables for elbows with turning vanes have been in earlier editions of the SMACNA *HVAC Systems Duct Design* manual and the 2001 *ASHRAE Handbook - Fundamentals*. Research on duct-fitting turning vanes indicates that using double-thickness turning vanes instead of single-thickness vanes increases the pressure loss of elbows, *see* Table A-7H.

Single thickness vanes have a maximum unsupported length of 36 in. (914 mm) as outlined in the SMACNA *HVAC Duct Construction Standards*. Double-thickness vanes keep their curved shape with the higher airstream velocity found in some HVAC system ductwork and are less prone to vibration or fluttering. They are not more aerodynamic than single-blade vanes, as originally thought, as the loss coefficients in Table A-7H indicate.

There often are higher losses caused by the shape of short, single-thickness vanes because of the distortion created by some turning vane rails (runners). However, multiple, single-thickness turning vane sections with vanes 36 in. (914 mm) long or less can be installed unsupported in large elbows instead of using double-thickness vanes.

### 5.16.2 Trailing Edges

SMACNA research has shown that turning vanes with trailing edges actually have a higher loss than vanes without them. When the vanes are accidentally installed with the airflow reversed, even higher losses develop.

### 5.16.3 Vanes Missing

For many years contractors have eliminated every other turning vane from the vane runners installed in rectangular mitered duct elbows. Some contractors even believed that they would lower the pressure loss of the elbow by doing this. However, this practice more than doubles elbow pressure losses and is not recommended.

Figure 5-13 is a chart developed from SMACNA sponsored research tested single-thickness turning vanes. The distance between vanes was varied in increments of ¼ in. (6 mm). Airflow velocities varied from 1000 to 2,500 fpm (5 to 12.5 m/s). The pressure loss of the elbow with missing turning vanes was over 2.5 times the pressure loss of a properly fabricated elbow containing all of the vanes.

*Example 5-21 (I-P)*

In a 2 in. wg pressure HVAC duct system that has 6,90 degree elbows, an airflow velocity of 2200 fpm, the velocity pressure ($V_p$) for 2200 fpm is 0.30 in. wg. Calculate the pressure loss of the 6 elbows, a) using 4.5 in. turning vanes, single thickness, with all vanes present (Table A-7, Figure H), b) with every other vane miss-

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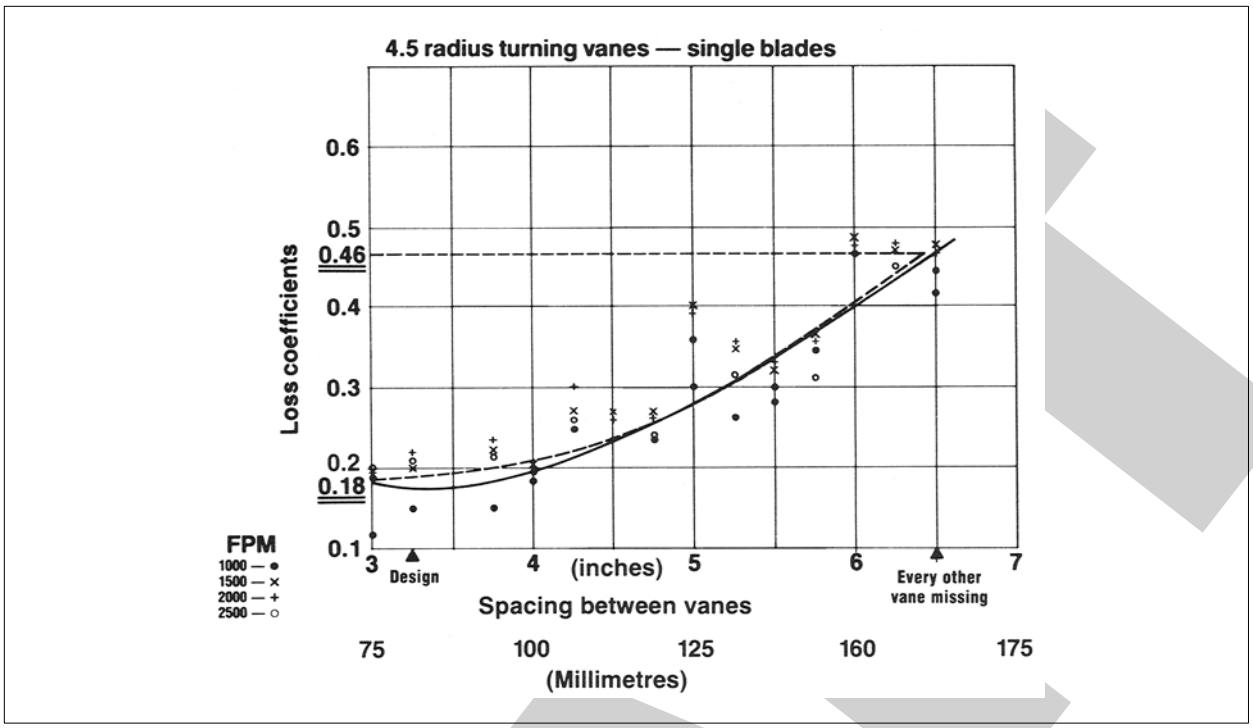


**FIGURE 5–13 TURNING VANES RESEARCH**

ing, *see* Figure 5-13, and c) with 2 in. double-thickness turning vanes on 2.25 in. centers, *see* Table A-7, Figure H.

a. Single, Standard Spacing:
The loss coefficient for a 90 degree elbow with 4.5 in. single thickness vanes is 0.23.

Using Equation 5-11:

$TP = C \times V_p = 0.23 \times 0.30$

$TP = 0.069$ in. wg

Loss for 6 elbows = 0.414 in. wg

b. Single, Every Other Vane Missing:

From Figure 5-13, C = 0.46

$TP = C \times V_p = 0.46 \times 0.30$

$TP = 0.138$ in. wg

Loss for 6 elbows = 0.828 in. wg

c. Double, Standard Spacing:

The loss coefficient for the 2 in. Double-thickness vane is 0.50 (2000 fpm).

$TP = C \times V_p = 0.50 \times 0.30$

$TP = 0.15$ in. wg

Loss for 6 elbows = 0.90 in. wg

*Example 5-21 (SI)*

In a 500 Pa pressure HVAC duct system that has 6,90 degree elbows, an airflow velocity of 11 m/s, the velocity pressure ($V_p$) is 71.6 Pa. Calculate the pressure loss of the 6 elbows, a) using 114 mm single-thickness turning vanes (Table A-7, Figure H); b) with every other vane missing, *see* Figure 5-13; and c) with 50 mm double-thickness turning vanes on 56 mm centers, *see* Table A-7, Figure H No. 3.

a. Single, Standard Spacing

The loss coefficient for a 90 degree elbow with 114 mm single-thickness vanes is 0.23. Using Equation 5-11:

$P = C \times V_p = 0.23 \times 71.6$

$TP = 16.47$ Pa

Loss for 6 elbows = 98.82 Pa

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b. Single, Every Other Vane Missing

From Figure 5-13, C = 0.46

$TP = C \times V_p = 0.46 \times 71.6$

$TP = 32.94$ Pa

Loss for 6 elbows = 197.6 Pa

The difference in losses, of the three different turning vanes in the same elbows, becomes very important to the energy conscious HVAC system designer who only has 2.0 in. wg (500 Pa) system static pressure with which to work. The single-thickness vane elbows in solution "a" used 0.414 in. wg (98.82 Pa) or 20.7 percent of the available pressure. The elbows in solution "b" with half of the turning vanes missing, consumed 0.828 in. wg (197.6 Pa) or 41.4 percent of the system pressure. The double-thickness vane elbows used in solution "c" used 0.90 in. wg (214.8 Pa) or 45.0 percent.

Another turning vane problem occurs when a rectangular duct mitered-elbow changes size from inlet to outlet. The pressure loss calculations should be based on the higher velocity pressure of the smaller size. The use of double-thickness vanes is not recommended, because they usually cannot be moved in many vane rails or runners to be tangent to the two airflows. However, a typical problem is turning vanes put into the vane rails, the same way as for a normal 90 degree elbow, as shown in Figure 5-14. Vanes that are not tangent to both airflow directions can cause a high-pressure loss. This "non-tangent to the airflow problem" also happens in normal 90 degree elbows with careless workmanship. A proper installation in a change-of-size elbow is shown in Figure 5-15 where the vanes have been installed so that they are tangent to the airflow.

### 5.16.4 Vanes in Small Duct

While splitters and turning vanes are beneficial at the top of stack heads to align airflow into the outlets, turning vanes are not normally used in small, low-velocity duct fittings-even square-throated elbows.

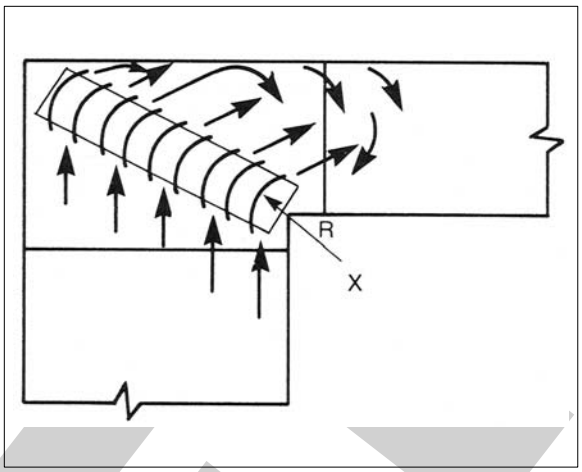


**FIGURE 5–14 TURBULENCE CAUSED BY IMPROPER MOUNTING AND USE OF TURNING VANES**

## 5.17 PRESSURE LOSSES IN DIVIDED–FLOW FITTINGS

### 5.17.1 Straight–Through Sections

Whenever air is diverted to a branch, there will be a velocity reduction in the straight-through section immediately following the branch. If no friction or dynamic losses occurred at the junction, there would be no loss in total pressure and the change in velocity pressure would be completely converted into a regain (rise) in static pressure.

Tests have shown that the regain coefficient across a takeoff can be as high as 0.90 for well-designed and properly constructed round ducts with no reducing section *immediately* after the takeoff.

Under less ideal conditions, such as in rectangular ducts with a high aspect ratio or takeoffs closely fol-

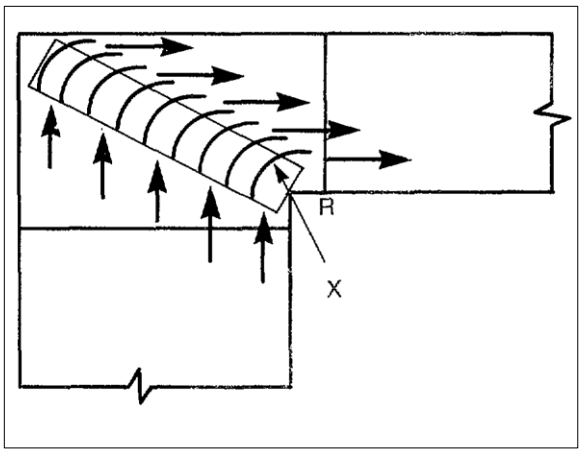


**FIGURE 5–15 PROPER INSTALLATION OF TURNING VANES**

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lowing an upstream disturbance, the regain coefficient can be as low as 0.50. A static pressure regain of 0.75 normally is used. Static regain (or loss) is included in the duct fitting loss coefficient tables, which have changes in cross-sectional areas of the main duct.

### 5.17.2 Diverted Flow Sections

The loss in a diverted flow section (tee or wye) depends on the ratio of the velocity of the diverted flow to the total flow, the areas of the inlets and exits, and the takeoff geometry. The total pressure loss coefficients for a variety of branch configurations for round and rectangular ductwork are shown in Table A-10 and A-11. These loss coefficient tables include static regain for converging and diverging flow patterns which can result in both positive and negative loss coefficients.

The junction of two parallel streams moving at different velocities is characterized by turbulent mixing of the streams, accompanied by pressure losses. In the course of this mixing, an exchange of momentum takes place between the particles moving at different velocities, finally resulting in the equalization of the velocity distributions into a common stream.

The total pressure loss of a "tee" or "wye" is a function of the branch velocity to the upstream (diverging) velocity or the downstream (converging) velocity using the nomenclature ($V_b/V_c$) shown in the figures in Tables A-10 and A-11. However, because of the different sources of the fitting loss coefficient data, the terms used to obtain the loss coefficient for different fittings will vary (such as $Q_b/Q_c$, $A_s/A_c$, $V_s/V_c$.)

For example, data for the SMACNA Duct Fitting Research Program shows that an inexpensive 45 degree entry branch from a rectangular main (Table A-11, Figure N) is a far more efficient fitting to use than a rectangular branch with an expensive extractor (Table A-11, Figure R). Using a $V_bV_c$ ratio of 1.0, the following can be extracted from the tables and compared.

If a commonly used plain round branch (Table A-11, figure S) is added to the comparison, one can see that the use of extractors should be eliminated as they can create other problems immediately downstream in the main duct. However, if a rectangular wye is used (Table A-11, Figure V) with the ratio $Q_b/Q_c = 0.4$, the branch loss coefficient will range from 0.30 to 0.41, depending on the fitting area ratios used with $A_b/A_c$ equals 0.5 This fabricated fitting is obviously more expensive to layout and make than a branch tap or takeoff, but the ongoing cost of operation of the system would be reduced – an important consideration with rising energy costs.

As part of the SMACNA Duct Fitting Research Program on diverted flow fittings, a video type entitled *Duct Research Destroys Design Myths* was produced, which demonstrates that turbulence is directly related to fitting loss coefficients. Soap bubbles introduced into the airstream of a lighted duct with one side of clear plastic dramatically shows the efficiency of the 45 degree entry fitting over the other types of branch duct tap fittings.

## 5.18 LOSSES DUE TO AREA CHANGES

Area changes in ducts are frequently necessitated by the building construction or changes in the volume of air carried. Experimental investigations of pressure changes and losses at changes of the area in duct cross sections indicates that the excess pressure loss over the normal friction loss is a dynamic one, due to a faster stream expanding into a slower stream, as determined by the actual areas occupied by the flow rather than by the areas of the duct. No perceptible dynamic loss is due to the converging of the airstream itself where the flow is contracted, but the airstream continues to converge beyond the edge of the contraction and reaches a minimum at the *vena contracta*. For contraction, therefore, the dynamic loss is caused by expansion from the vena contracta to the full area following the contraction. Abrupt contraction in area may, therefore, be considered as a special condition of abrupt enlargement.

Energy losses due to enlargement of the airstream are high relative to losses due to contraction. Typical loss coefficients, which include static regain or loss, are listed in Tables A-8 and A-9.

In determining the proportions of a specific transitional fitting, the designer should recognize that the total pressure loss is influenced far more by the velocity than by the loss coefficient of a particular geometry. The small losses associated with low velocity applications may not always justify the additional cost of fittings, which have low loss coefficients.

## 5.19 OTHER LOSS COEFFICIENTS

Loss coefficients for most commonly used entries; discharges, screens and plates, dampers and obstructions are found in Tables A-12 to A-15. Screens or perforated plates can also be added to many of the discharge or entry fittings by combining the loss coefficients, based on the use of the proper areas. Perforated plates may be used in plenum chambers to improve velocity

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profiles across filters and coils when irregular velocities are present due to approach angle or mixing conditions, and in front of fan discharge in blow-through units. They also may be used in branch ducts to dissipate excess static pressure in low-resistance runs.

Commonly used "shop" fabricated butterfly damper loss coefficients in Table A-15 are based on a constant velocity. Use of these coefficients will be found in the duct design examples in Chapter 7. However, AMCA tests have shown that there can be a dramatic increase in the pressure drop of small dampers as compared to large dampers of the same design, *see* Figure 5-16.

Attention is called to the large loss coefficients of a fan "free discharge," i.e. no ductwork on the discharge side of the fan. (Table A-13, Figures G and H.) When fans are tested and rated, discharge ductwork is attached. However, this "free discharge" installation has been used as an industry "standard" for roof mounted exhaust fans for many years. Example 5-22 using Table A-13G indicates why marginally-sized exhaust systems have suffered through the years.

*Example 5-22 (I-P)*

A small vent set has an outlet velocity of 1790 fpm at 0.25 in. wg static pressure. Calculate the capacity loss

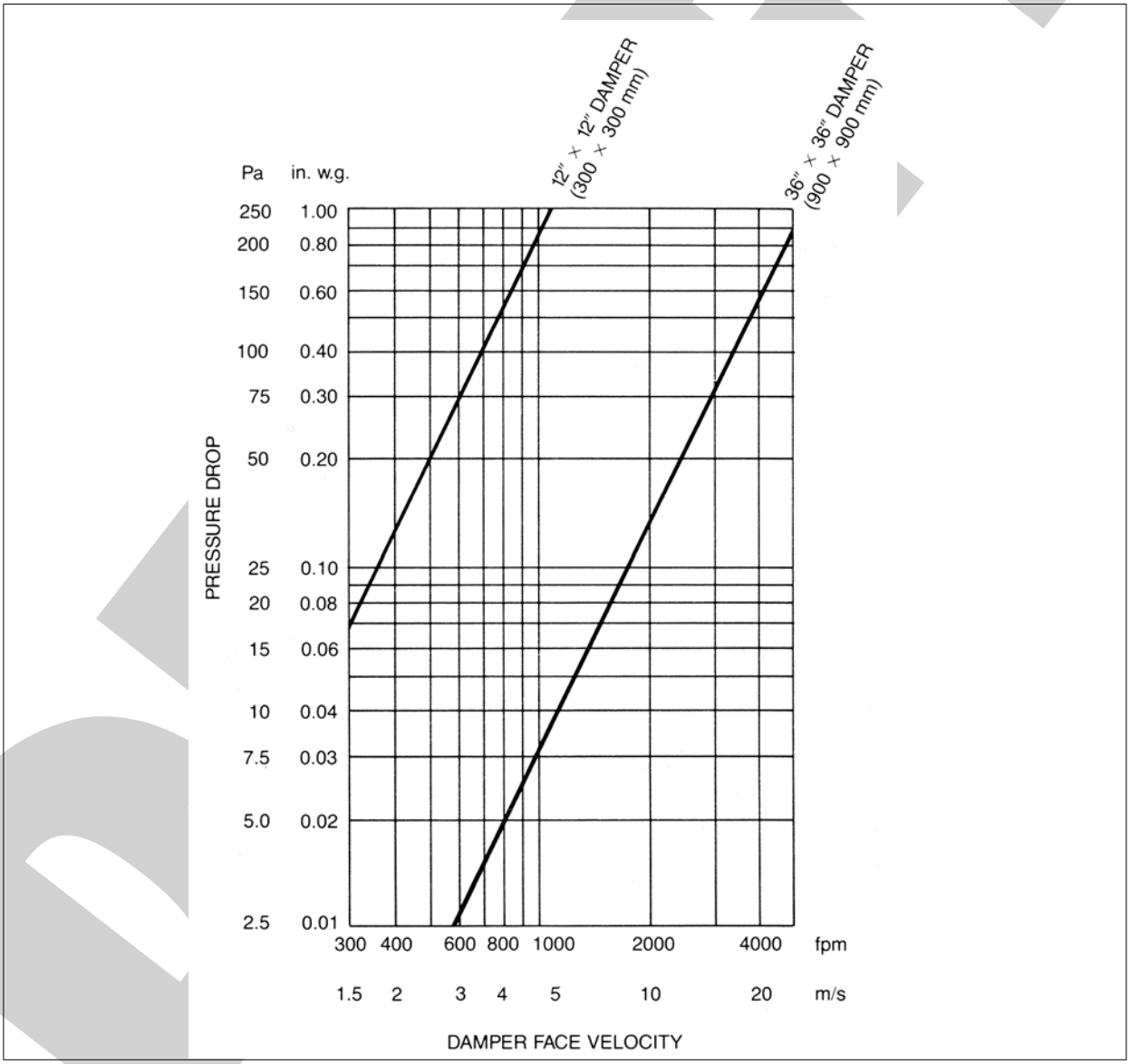


**FIGURE 5–16 AMCA DAMPER TESTS**

*Reprinted from AMCA (Standard or Publication, such as: AMCA Publication 201–90, Fans and Systems), with written permission from Air Movement and Control Association International, Inc.*

of the "free discharge" roof mounted fan. ($\theta = 30°$, $A_1/A = 1.5$.)

Find $V_p$ for 1,790 fpm (using Equation 5-15):

$$V_p = \left(\frac{1790}{4005}\right)^2 = 0.20 \text{ in. wg}$$

From Table A-13G, C= 0.63. Using Equation 5-13:

$$TP = C \times V_p = 0.63 \times 0.20$$

$$TP = 0.126 \text{ in. wg}$$

The "free discharge" consumes 50 percent of the rated 0.25 in. wg fan capacity.

*Example 5-22 (SI)*

A small vent set has an outlet velocity of 9 m/s at 60 Pa static pressure. Calculate the capacity loss of the "free discharge" roof mounted fan.

($\theta = 0.5$ rad and $\frac{A_1}{A} = 1.5$)

Find $V_p$ for 9 m/s (using Equation 5-15):

$$V_p = \left(\frac{9}{1.30}\right)^2 = 47.9 \text{ Pa}$$

From Table A-13G and C = 0.63

Using Equation 5-13:

$$TP = C \times V_p = 0.63 \times 47.9$$

$$TP = 30.2 \text{ Pa}$$

The "free discharge" consumes about 50 percent of the rated 60 Pa fan capacity.

## 5.20 OBSTRUCTION AVOIDANCE

One of the areas that the SMACNA Duct Fitting Research Program concentrated on was the problem of routing a duct under a beam or pipe where space was limited. Figure 5-17 shows one method to route a duct under a structured beam, but this design is more expensive to fabricate. Refer to Appendix Tables A-15, Figures "I" to "L" for a less expensive alternative design.

Depressing the height of a round or rectangular duct up to 30 percent without increasing the duct width can be done with duct fitting loss coefficients in the range of 0.24 to 0.35. Using a duct with a 2000 fpm (10 m/s) velocity ($V_p$ = 0.25 in. wg or 62 Pa) this type of fitting develops the following fitting pressure losses:

Round:
$C \times V_p = 0.24 \times 0.25\ (62) = 0.06$ in. wg (15 Pa) loss.

Rectangular:
$C \times V_p = 0.35 \times 0.25\ (62) = 0.09$ in. wg (22 Pa) loss.

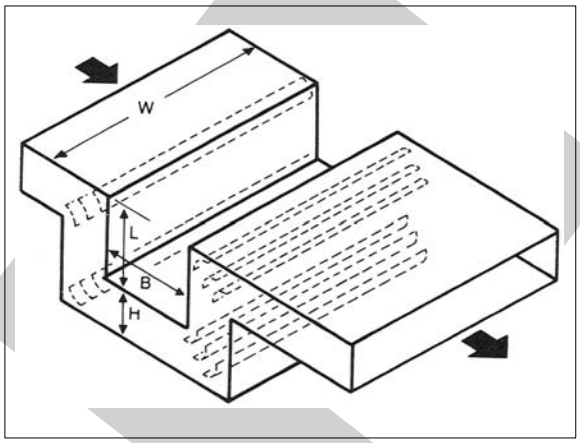


**FIGURE 5–17 DUCT OBSTRUCTIONS**

However, when there is a deep beam surrounded by many other types of pipes and conduits (such as above a dropped ceiling in a hospital), a fitting such as that found in Table A-15 Figure L can be used, but must have turning vanes.

*Example 5-23 (I-P)*

An average low-pressure duct system might be designed to develop a velocity of 2000 fpm at 2.5 in. wg total pressure in the main supply duct leaving the fan. What would be the pressure loss of the fitting found in Table A-15, Figure L if the beam/duct height ratio (L/H) was 2 with and without single thickness turning vanes?

From Table A-5, $V_p$ = 0.25 for 2000 fpm.

From Table A-15, figure L,

C = 0.77 for single blade turning vanes

C = 9.24 without turning vanes

With Turning Vanes:

Fitting loss = $C \times V_p = 0.77 \times 0.25 = 0.19$ in. wg

Without Turning Vanes:

Fitting loss = $C \times V_p = 9.24 \times 0.25 = 2.31$ in. wg

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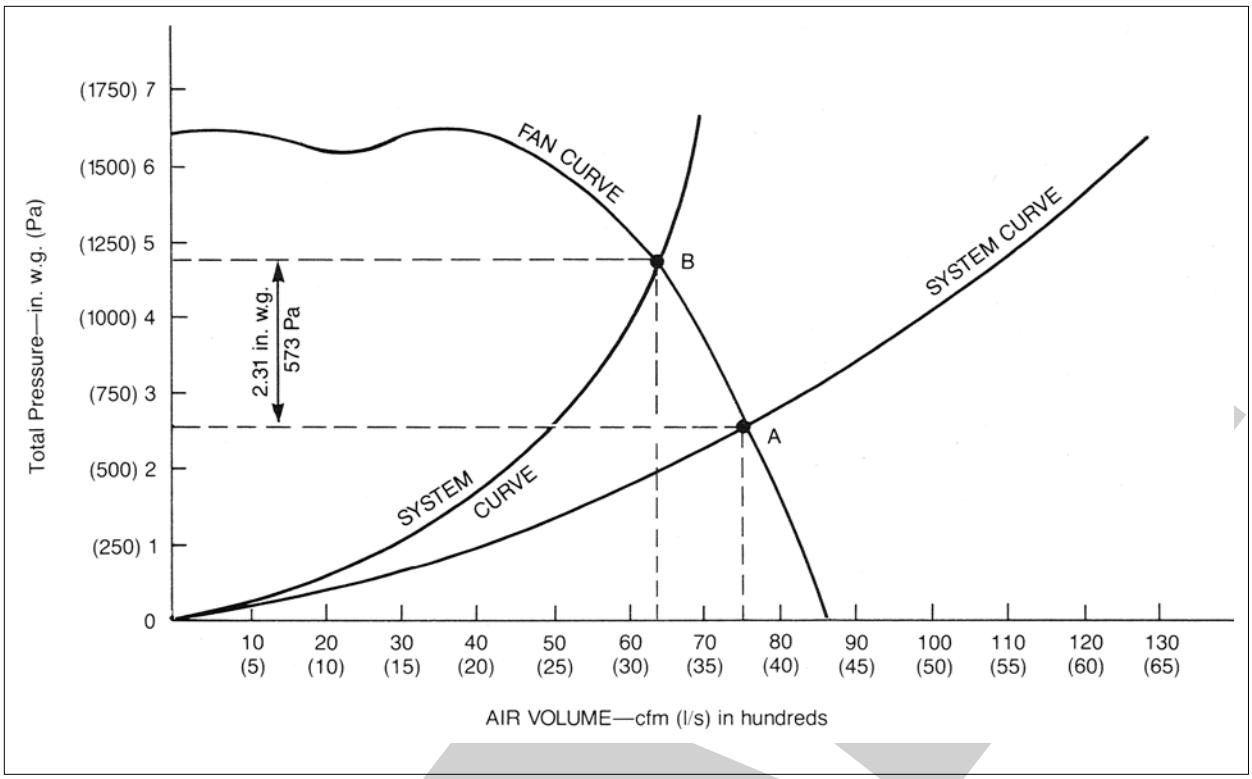


**FIGURE 5–18 EXAMPLE 5–5 FAN/SYSTEM CURVE**

| Rectangular | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Pressure (in. wg) | cfm/ft | | | | Pressure (Pa) | l/s/m | | | |
| | Pittsburg Lock | | Button Punch Snap Lock | | | Pittsburg Lock | | Button Punch Snap Lock | |
| | 26 ga | 22 ga | 26 ga | 22 ga | | 0.55 mm | 0.85 mm | 0.55 mm | 0.85 mm |
| ½ | 0.0117 | 0.0009 | 0.0156* | 0.0019 | 125 | 0.0183 | 0.0014 | 0.0244 | 0.0030 |
| 1 | 0.0198 | 0.0017 | 0.0266 | 0.0032 | 250 | 0.0309 | 0.0027 | 0.0416 | 0.0005 |
| 2 | 0.0322 | 0.0027 | 0.0344 | 0.0056 | 500 | 0.0503 | 0.0042 | 0.0538 | 0.0088 |
| 3 | 0.0428 | 0.0035 | 0.0598* | 0.0073* | 750 | 0.0067 | 0.0055 | 0.0934 | 0.0114 |

| Round[a] | | | | | |
|---|---|---|---|---|---|
| cfm/ft | | | l/s/m | | |
| | Snap Lock | Grooved | | Snap Lock | Grooved |
| Range | 0.04 to 0.14 | 0.11 to 0.18 | Range | 0.06 to 0.22 | 0.17 to 0.28 |
| Average | 0.11 | 0.12 | Average | 0.17 | 0.19 |

**Table 5–1 Unsealed Longitudinal Seam Leakage For Metal Ducts**

[a]Leakage rate is at 1 in. wg (250 Pa) static pressure
*See Figure 2-2 HVAC Duct Construction Standards for construction requirements.

One can see that the 0.19 in. wg pressure loss of the fitting with turning vanes is but 8 percent of the initial 2.5 in. wg in the duct system. The 2.31 in. wg pressure loss of the fitting without turning vanes theoretically destroys the system airflow by wiping out 92 percent of the 2.5 in. wg total system pressure!

Actually, the operating point of the system and fan curve interchange moves up and to the left on the fan curve, substantially reducing the system airflow, but not by 92 percent intimated above, *see* Figure 5-23.

*Example 5-23 (SI)*

An average low-pressure duct system might be designed to develop a velocity of 10 m/s at 625 Pa total pressure in the main supply duct leaving the fan. What would be the pressure loss of the fitting found in Table A-15, Figure L if the beam/duct height ration (L/H) was 2 with and without single thickness turning vanes?

Without Turning Vanes:

Fitting loss = $C \times V_p = 9.24 \times 62 = 573$ Pa

One can see that the 47.7 Pa pressure loss of the fitting with turning vanes is but 8 percent of the initial 625 Pa in the duct system. *The 573 Pa pressure loss of the fitting without turning vanes theoretically destroys the system airflow by wiping out 92 percent of the 625 Pa total system pressure!* Actually, the operating point of the system and fan curve interchange moves up and to the left on the fan curve, substantially reducing the system airflow, *see* Figure 5-23.

## 5.21 DUCT AIR LEAKAGE

The amount of duct leakage in an HVAC system may be determined in advance by the system designer using data extracted from the SMACNA *HVAC Duct Construction Standards – Metal and Flexible* and the SMACNA *HVAC Air Duct Leakage Test Manual*.

Leakage in all unsealed ducts varies considerably with the fabricating machinery used, the methods of assembly, and the workmanship. For sealed ducts, a wide variety of sealing methods and products exists.

Many sealants contain volatile solvents that evaporate and can introduce shrinkage and curing factors. Surface cleanliness and sealant application in relation to air pressure direction are other variables. With the exception of the UL 181 series of tested products, which is specific to fibrous glass and flexible round duct, no standard tests exist to evaluate performance and to grade sealing products. A variety of sealed and unsealed duct leakage tests have confirmed that longitudinal seam, transverse joint, and assembled duct leakage can be represented by:

**Equation 5–16**

$$F = C_L P^N$$

Where:

$F$ = Leak rate per unit of cut surface

$C_L$ = Constant

$P$ = Static Pressure

$N$ = Exponent relating turbulence

Joint SMACNA/ASHRAE/TIMA tests have shown that leakage for the same construction is not significantly different in the negative and positive modes. A range of leakage rates for seams commonly used in the

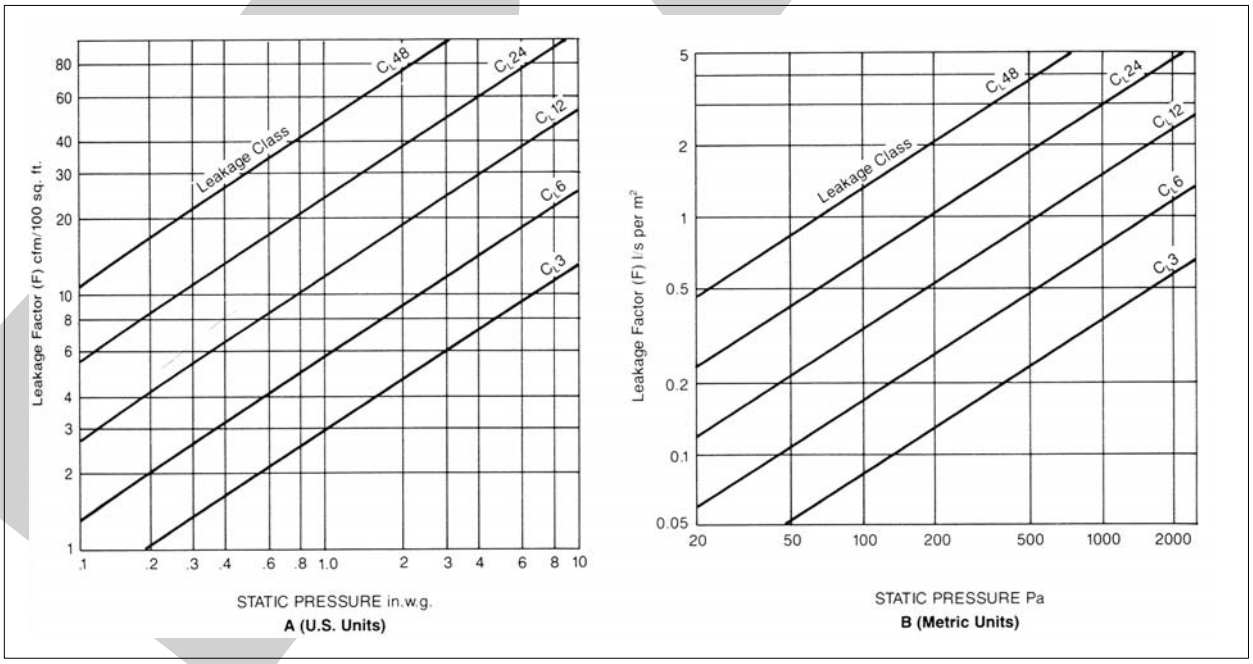


**FIGURE 5–19 DUCT LEAKAGE CLASSIFICATIONS**

construction of metal ducts is presented in Table 5-1. Longitudinal seam leakage for metal ducts is about 10 to 15 percent of TOTAL duct leakage. Total duct leakage ranges from 1 to 100 cfm/100 ft$^2$ of duct surface area depending on the operating pressure, sealing method, type of joint, etc., *see* Table 5-2.

Analysis of the SMACNA/ASHRAE/TIMA data resulted in the categorization of duct systems into a leakage class ($C_L$) based on Equation 5-17, where the exponent N is assumed to be 0.65. A selected series of leakage classes based on Equation 5-17 is shown in Figure 5-19.

**Equation 5–17 (I–P)**

$$C_L = \frac{F}{P^{0.65}}$$

**Equation 5–17 (SI)**

$$C_L = \frac{720F}{P^{0.65}}$$

Where:

$C_L$ = Leakage class at 1 in. wg (250 Pa) static Pressure – cfm/100 ft$^2$ (L/s per m$^2$)

$P$ = Static Pressure – in. wg (Pa)

$F$ = Leakage rate – cfm/100 ft$^2$ (L/s per m$^2$) duct surface

From Table A-5M, $V_p$ = 62 Pa for 10 m/s.

From Table A-15, Figure L,

C = 0.77 for single thickness turning vanes

C = 9.24 without turning vanes

With Turning Vanes:

Fitting loss = $C \times V_p = 0.77 \times 62 = 47.7$ Pa

Table 5-2 is a summary of the leakage class attainable for good duct construction and sealing practices. Connections of ducts to grilles, diffusers, and registers are not represented in the test data. The HVAC system designer is responsible for assigning acceptable leakage rates. Although leakage as a percentage of fan airflow rate is an important evaluation criterion, *see* Table 5-3, designers should first become familiar with the leakage rates from selected construction detail. This knowledge allows the designer to analyze both first- and life-cycle cost of a duct system so the owner benefits. In performing an analysis, the designer should independently account for air leakage in casings and frames of equipment attached to the duct system. Casings or volume-controlling air terminal units may leak 2 to 5 percent of their maximum flow. The effects of such leakage should be anticipated, if allowed, and the ductwork should not be expected to compensate for equipment leakage. Allowable leakage should be controlled consistent with airflow tolerances at the air terminals. A leakage class of 3 is attainable for all duct systems by careful selection of joints and sealing methods in combination with good workmanship.

Zero leakage is not a practical objective except in critical situations such as nuclear safety-related application because it is prohibitively expensive. One percent leakage also is difficult to attain in larger systems without using expensive sealing methods or welded joints. The shaded area in Table 5-3 predicts that one percent leakage in duct systems is only attainable up to 2 in. wg (500 Pa) static pressure, which eliminates all higher-pressure systems and all larger systems where the system airflow per square foot of duct surface is low. Additional discussions of leakage analysis may be found in the SMACNA *HVAC Air Duct Leakage Test Manual*.

*Example 5-7 (I-P)*

Using a typical duct system shown in Figure 7-2 of Chapter 7, find the total leakage of the supply ductwork in both cfm and percentage of airflow.

a. The average pressure from A to F is 2.5 in. wg [(3 + 2)/2]. From Table 5-2, the leakage class for a 3 in. wg duct class round metal duct is "6". Using Figure 5-19 (A), the leakage factor would be 10.6-cfm/100 ft$^2$. The 34 in. diameter duct from A to F has 800 ft$^2$ of duct surface.

   Leakage = 10.6/100 × 800 = 85 cfm

b. The average pressure from F to J and P is 1.5 in. wg [(2 + 1)/2]. From Table 5-2, the leakage class for a 2 in. wg duct class round metal duct is "12". Using Figure 5-19 (A), the leakage factor would be 15.5 cfm/100 ft$^2$. The total calculated duct surface is 900 ft$^2$.

   Leakage = 15.5/100 × 900 = 140 cfm (F to J and P).

   Leakage for similar ducts (F to W and X branches) would be the same, so the total would be 140 + 140 = 280 cfm of leakage.

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| Duct Class (*See* Table 4-1) | ½ 1, 2 in. wg (125, 250, 500 Pa) | | 3 in. wg (750 Pa) | 4, 6, 10 in. wg (1000, 1500, 2500 Pa) |
|---|---|---|---|---|
| Seal Class | None | C | B | A |
| Applicable Sealing | N/A | Transverse Joints Only | Transverse Joints and Seams | All Joints, Seams and Wall Penetrations |
| **Leakage Class (CL) cfm/100 sq. ft (l/s per $m^2$) at 1 in. wg (250 Pa)** | | | | |
| Rectangular Metal | 48 | 24 | 12 | 6 |
| Round and Oval Metal | 30 | 12 | 6 | 3 |
| Rectangular Fibrous Glass | N/A | 6 | N/A | N/A |
| Round Fibrous Glass | N/A | 3 | N/A | N/A |

**Table 5–2 Applicable Leakage Classes[a]**

*NOTE:*

[a]The leakage classes listed in this table are averages based on tests conducted by SMACNA/ASHRAE/TIMA. Leakage classes listed are not necessarily recommendations on allowable leakage. The designer should determine allowable leakage and specify acceptable duct leakage classifications.

| Leakage Class | System Airflow | | Static Pressure in. wg (Pa) | | | | | |
|---|---|---|---|---|---|---|---|---|
| | cfm/$ft^2$ | l/s per m2 | ½ (125) | 1 (250) | 2 (500) | 3 (750) | 4 (1000) | 6 (1500) |
| 48 | 2 | 10 | 15 | 24 | 38 | | | |
| | 2.5 | 12.7 | 12 | 19 | 30 | | | |
| | 3 | 15 | 10 | 16 | 25 | | | |
| | 4 | 20 | 7.7 | 12 | 19 | | | |
| | 5 | 25 | 6.1 | 9.6 | 15 | | | |
| 24 | 2 | 10 | 7.7 | 12 | 19 | | | |
| | 2.5 | 12.7 | 6.1 | 9.6 | 15 | | | |
| | 3 | 15 | 5.1 | 8.0 | 13 | | | |
| | 4 | 20 | 3.8 | 6.0 | 9.4 | | | |
| | 5 | 25 | 3.1 | 4.8 | 7.5 | | | |
| 12 | 2 | 10 | 3.8 | 6 | 9.4 | 12 | | |
| | 2.5 | 12.7 | 3.1 | 4.8 | 7.5 | 9.8 | | |
| | 3 | 15 | 2.6 | 4.0 | 6.3 | 8.2 | | |
| | 4 | 20 | 1.9 | 3.0 | 4.7 | 6.1 | | |
| | 5 | 25 | 1.5 | 2.4 | 3.8 | 4.9 | | |
| 6 | 2 | 10 | 1.9 | 3 | 4.7 | 6.1 | 7.4 | 9.6 |
| | 2.5 | 12.7 | 1.5 | 2.4 | 3.8 | 4.9 | 5.9 | 7.7 |
| | 3 | 15 | 1.3 | 2.0 | 3.1 | 4.1 | 4.9 | 6.4 |
| | 4 | 20 | 1.0 | 1.5 | 2.4 | 3.1 | 3.7 | 4.8 |
| | 5 | 25 | .8 | 1.2 | 1.9 | 2.4 | 3.0 | 3.8 |
| 3 | 2 | 10 | 1.0 | 1.5 | 2.4 | 3.1 | 3.7 | 4.8 |
| | 2.5 | 12.7 | .8 | 1.2 | 1.9 | 2.4 | 3.0 | 3.8 |
| | 3 | 15 | .6 | 1.0 | 1.6 | 2.0 | 2.5 | 3.2 |
| | 4 | 20 | .5 | .8 | 1.3 | 1.6 | 2.0 | 2.6 |
| | 5 | 25 | .4 | .6 | .9 | 1.2 | 1.5 | 1.9 |

**Table 5–3 Leakage As A Percentage of System Airflow**

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c. The average pressure from J to M is 1 in. wg because the VAV boxes require 1 in. wg inlet pressure. From Table 5-2, the leakage class will remain at "12", as it is based on 1 in. wg. The calculated duct surface for all four branches (M, S, W, and X) is 1320 $ft^2$.

Leakage = $12/100 \times 1320 = 159$ cfm

d. The total duct system leakage not counting the flexible connections is:

| | | | |
|---|---|---|---|
| a. | AF | = | 85 cfm |
| b. | FJ/P | = | 280 cfm |
| c. | JM/etc. | = | 159 cfm |
| | Total | = | 524 cfm duct leakage |

Percent leakage = $524 \times 100/20{,}000$ cfm
= 2.62 percent

*Example 5-24 (SI)*

Using a typical duct system shown in Figure 7-6 of Chapter 7, find the total leakage of the supply ductwork in both L/s and percentage of airflow.

a. The average pressure from A to F is 625 Pa [(750 + 500)/2]. From Table 5-2, the leakage class for a 750 Pa duct class round metal duct is "6". Using Figure 5-19(B), the leakage factor would be 0.6 L/s per square meter. The 900 mm diameter duct from A to F has 90 $m^2$ of surface.

Leakage = $0.6 \times 90 + 54$ L/s

b. The average pressure from F to l and 0 is 375 Pa [500 + 250)/2]. From Table 5-2, the leakage class for a 500 Pa duct class round metal duct is "12". Using Figure 5-19(B), the leakage factor would be 0.75 L/s per square meter. The total calculated duct surface (including all four branches) would be 146 $m^2$.

Leakage = $0.75 \times 146 + 110$ L/s

c. The average pressure from l to M is 250 Pa because the VAV boxes require 250 Pa inlet pressure. From Table 5-2, the leakage class will remain at "12", as it is based on 250 Pa. The calculated duct surface for all four branches (M, S, W and X) is 208 $m^2$. Using Figure 5-219B), the leakage factor is 0.65 L/s per $m^2$.

Leakage = $0.65 \times 208 = 135$ L/s

d. The total duct system leakage (not counting the flexible connections) is:

| | | | |
|---|---|---|---|
| 1. | AF | = | 54 L/s |
| 2. | F/10 | = | 110 L/s |
| 3. | IM/etc. | = | 135 L/s |
| | Total | = | 299 L/s duct leakage |

Percent leakage = $299 \times 100/10{,}000$ L/s
= 2.99 percent

*NOTE: Although the duct systems in Figures 7-2 and 7-6 are similar, the metric unit dimensions are <u>not</u> conversions from the I-P unit dimensions. Therefore, the percentage leakages from the two examples are not identical.*

If a VAV system was built to a duct class of 3 in. wg (750 Pa) throughout and 1 in. wg (250 Pa) was required at the VAV boxes, the average pressure would be 2 in. wg (500 Pa). From Table 5-2 and Figure 5-19 the following is obtained:

Round metal duct,
$C_L$ = 6 and F = 9 cfm/100 $ft^2$ (0.5 L/s per $m^2$)

Rectangular metal duct,
$C_L$ = 12 and F = 18 cfm/100 $ft^2$ (0.95 L/s per $m^2$)

To obtain a one percent leakage rate using a 10,000 cfm (5000 L/s) fan, the system would be limited in size to the following:

a. I-P
One percent of 10,000 cfm = 100 cfm
Round duct = 100 cfm/9 cfm/100 $ft^2$
= 1111 $ft^2$ (maximum)

Rectangular duct = 100 cfm/18 cfm/100 $ft^2$
= 556 $ft^2$ (maximum)

b. SI
One percent of 5000 L/s = 50 L/s
Round duct = 50 L/s / 0.5 L/s / $m^2$
= 100 $m^2$ (maximum)

Rectangular duct = 50 l s / 0.95 l/ s per $m^2$
= 53 $m^2$ (maximum)

It becomes obvious that to obtain a one percent leakage rate, the designer is limited to a very small duct distribution system. Yet, some designers insist that it can be done using normal duct sealing methods on normal-sized systems. If energy losses are critical or if the ducts must have zero leakage, as in nuclear power work, then the duct joints must be welded or soldered,

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with the resultant extreme increase in costs of fabrication and erection.

## 5.22 DUCT HEAT GAIN/LOSS

ANSI/ASHRAE/IES *Standard 90.1* requires thermal insulation of all duct systems and their components (i.e., ducts, plenums, and enclosures) installed in or on buildings. Adequate thermal insulation thickness is determined according to duct location, type of duct—heating only, cooling only, return—and climate zone.

R = Thermal resistance excluding film resistances, $ft^2$ °F · h/Btu ($m^2$·°/ C/W)

$\Delta t$ = design temperature differential between duct air and duct surface, degree F (degree C)

Since *Standard 90.1* does not consider condensation, additional insulation with vapor barriers may be required.

Duct heat gains or losses must be known to calculate supply air quantities, supply air temperatures, and coil loads. To estimate duct heat transfer and entering or leaving air temperatures, use Equations 5-18 and 5-19.

**Equation 5–18 (I–P)**

$$Q_1 = \frac{UPL}{12}\left[\left(\frac{t_e + t_1}{2}\right) - t_a\right]$$

**Equation 5–18 (SI)**

$$Q_1 = \frac{UPL}{1000}\left[\left(\frac{t_e + t_1}{2}\right) - t_a\right]$$

**Equation 5–19 (I–P & SI)**

$$t_e = \frac{t_1(y + 1) - 2t_a}{(y - 1)}$$

**Equation 5–19 (I–P & SI)**

$$t_e = \frac{t_1(y - 1) + 2t_a}{(y + 1)}$$

Where:

$y$ = $2.4AV_p/UPL$ for rectangular ducts ($2.01\ AV_p/UPL$

$y$ = $0.6DV_p/UL$ for round ducts ($0.5DV_p/UL$)

$A$ = cross-sectional area of duct, $in.^2$ ($mm^2$)

$V$ = average velocity, fpm (m/s)

$D$ = diameter of duct, in. (mm)

$L$ = duct length, ft (m)

$Q_1$ = heat loss/gain through duct walls, Btu/h (W) negative for heat gain

$U$ = overall heat transfer coefficient of duct wall, Btu/h·$ft^2$°F (W/($m^2$·°C)

$P$ = perimeter of bare or insulated duct, in. (mm)

$\rho$ = density, $lb_m/ft^3$ ($kg/m^3$)

$t_e$ = temperature of air entering duct, degree F (degree C)

$t_1$ = temperature of air leaving duct, degree F (degree C)

$t_a$ = temperature of air surrounding duct, degree F (degree C)

Use Figures A-5 and A-5M to determine the "U-values" for insulated and uninsulated ducts. For a 2 in. (50 mm) thick, 0.75 $lb/ft^3$ (12 $kg/m^3$) fibrous-glass blanket compressed 50 percent during installation; the heat transfer rate increases approximately 20 percent as shown in Figure A-5(a) (A-5M(a)). Previous flexible duct liners also influence heat transfer significantly as shown in Figure A-65b) (A-5(b)). At 2500 fpm (12.5 m/s), the pervious liner "U-value" is 0.33 Btuh/$ft^2$·°F (1.87 W/$m^2$·°C); for an impervious liner the "U-value" is 0.19 Btuh/$ft^2$·°F (1.08 W/$m^2$·°C).

*Example 5-25 (I-P)*

A 65 ft length of 24 × 36 in. uninsulated sheet metal duct, freely suspended, conveys heated air through a space maintained above freezing at 40 degrees F. Based on heat loss calculations for the heated zone, 17,200 cfm of standard air at a supply air temperature of 122 degrees F is required. The duct is connected directly to the heated zone. Determine the air temperature entering the duct and the duct heat loss.

a. Calculate the duct velocity using Equation 5-14:

$$A = \frac{Q}{A} = \frac{17,200}{24 \times 36/144} = 2900 \text{ fpm}$$

Select U = 0.73 Btuh/$ft^2$·°F from Figure A-5.

Calculate P = 2(24. + 36 in.) = 120 in.

$$y = 2.4A\ V_\rho/UPL$$

$$y = \frac{2.4 \times 24 \text{ in.} \times 36 \text{ in.} \times 2900 \times 0.075 \text{ lb/ft}^3}{0.73 \times 120 \text{ in.} \times 65 \text{ ft}}$$

$$y = 79.2$$

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b. Calculate the entering air temperature using Equation 5-20:

$$t_e = \frac{122^\circ F(79.2 + 1) - (2 \times 40^\circ F)}{(79.2 - 1)}$$

$$t_e = 124.1^\circ F$$

c. Calculate the duct heat loss using Equation 5-19:

$$Q_1 = \frac{0.73 \times 120 \times 65\,ft}{12} \times \left[\left(\frac{124.1^\circ F + 122^\circ F}{2}\right) - 40^\circ F\right]$$

$$Q_1 = 39,200 \text{ Btuh}$$

*Example 5-25(SI)*

A 20 m length of 600 x 900 mm uninsulated sheet metal duct, freely suspended, conveys heated air through a space maintained above freezing at 5 degrees C. Based on heat loss calculations for the heated zone, 8100 L/s of standard air at a supply-air temperature of 50 degrees C is required. The duct is connected directly to the heated zone. Determine the air temperature entering the duct and the duct heat loss.

a. Calculate the duct velocity using Equation 5-14:

$$V = \frac{Q}{A} = \frac{1000 \times 8100}{600 \times 900} = 15 \text{ m/s}$$

$$Q_1 = 39,200 \text{ Btuh}$$

Select U = 4.16 W/m$^2$· °C (from Figure A-7)

Calculate P = 2 (600 + 900) = 3000 mm

$$y = 2.01A\,V\rho/UPL$$

$$y = \frac{2.01 \times 600 \times 900 \times 15 \times 1.204 \text{ kg/m}^3}{4.16 \times 3000 \times 20 \text{ m}}$$

$$y = 78.5$$

b. Calculate the entering air temperature using Equation 5-20:

$$t_e = \frac{50^\circ C(78.5 + 1) - (2 \times 5^\circ C)}{78.5 - 1}$$

$$t_e = 51.2^\circ C$$

c. Calculate the duct heat loss using Equation 5-19:

$$Q_1 = \frac{4.16 \times 3000 \times 20 \text{ m}}{1000} \times \left[\left(\frac{51.2^\circ C + 50^\circ C}{2}\right) - 5^\circ C\right]$$

*Example 5-26 (I-P)*

Same as Example 5-8, except the duct is insulated externally with 2 in. thick fibrous glass with a density of 0.75 lb/ft$^3$. The insulation is wrapped with 0 percent compression.

All values, except U and P, remain the same as Example 5-8. From Figure 4-16(a), U = 0.15 btu/h·ft$^2$·°F at 2,900 fpm. P = 136-in. Therefore:

$y$ = 441

$t_e$ = 122.4°F

$Q_1$ = 9,083 Btuh

Insulating this duct reduces heat loss to 20 percent of the un-insulated duct.

*Example 5-26 (SI)*

Same as Example 5-25, except the duct is insulated externally with 50 mm thick fibrous glass with a density of 12 kg/m$^3$. The insulation is wrapped with 0 percent compression.

All values, except U and P, remain the same as Example 5-25. From Figure A-7(a), U = 0.83 W/(m$^2$·°C) at 15 m/s. P = 3400 mm. Therefore:

Insulating this duct reduces heat loss to 20 percent of the un-insulated duct.

## 5.23 SMACNA DUCT RESEARCH

SMACNA has worked independently with universities and testing laboratories, and jointly with ASHRAE in various duct system research projects. The fitting loss coefficient data in the Appendix is not totally reliable under all conditions. However, the data is accurate within the research test parameters listed.

### 5.23.1 Other Elbow Configurations

In addition to the various test projects on turning vane elbows previously discussed, other types of mitered rectangular duct elbows without turning vanes were tested. Loss coefficients were obtained for three sizes of 90 degrees mitered elbows; 12 × 12 in. (300 × 300 mm), 22 × 8 in. (550 × 200 mm), and 8 × 22 in. (200 × 550 mm). These are compared with existing data from Table A-10D and Figure 5-20. Note that the new data is reasonably consistent with older data being used. Test velocities ranged from 800 fpm (4 m/s) to 4400 fpm (22 m/s) in 200 fpm (1 m/s) increments.

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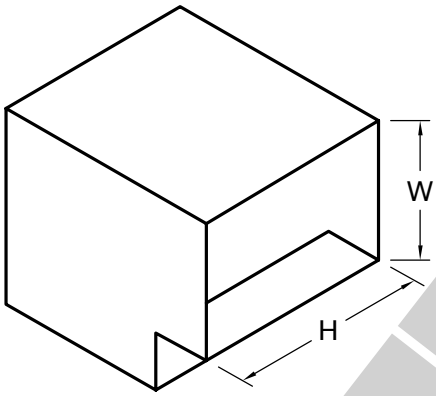
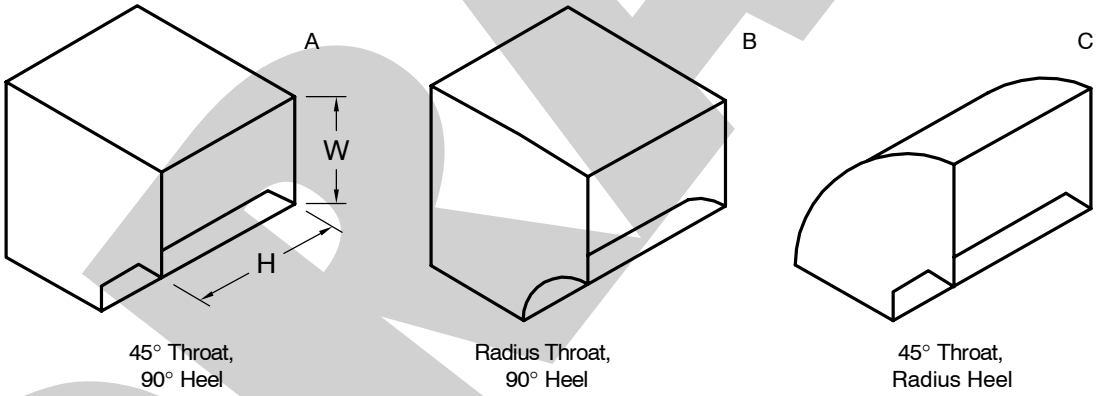



| | Fitting Loss Coefficient (C) | | | | | | |
|---|---|---|---|---|---|---|---|
| | H/W | | | | | | |
| | **0.25** | **0.36** | **0.50** | **1.0** | **2.0** | **2.75** | **3.0** |
| From Table 14-10D | 1.3 | – | 1.3 | 1.2 | 1.1 | – | 0.98 |
| From SMACNA Research | – | 1.20 | – | 1.26 | – | 1.19 | – |

**FIGURE 5–20 RECTANGULAR ELBOW 90 DEGREE THROAT, 90 DEGREE HEEL**




| Elbow | Fitting Loss Coefficient (C) | | |
|---|---|---|---|
| | H/W | | |
| | **0.36** | **1.00** | **2.75** |
| A | 0.41 | 0.60 | 0.53 |
| B | 0.36 | 0.58 | 0.45 |
| C | 0.34 | 0.33 | 0.32 |
| *R/W = 0.75 | 0.55 | 0.44 | 0.39 |
| *R/W = 1.0 | 0.26 | 0.21 | 0.18 |

*Smooth radius rectangular elbow w/o vanes, *see* Table 14-10F

**FIGURE 5–21 DIFFERENT CONFIGURATION ELBOW RESEARCH**

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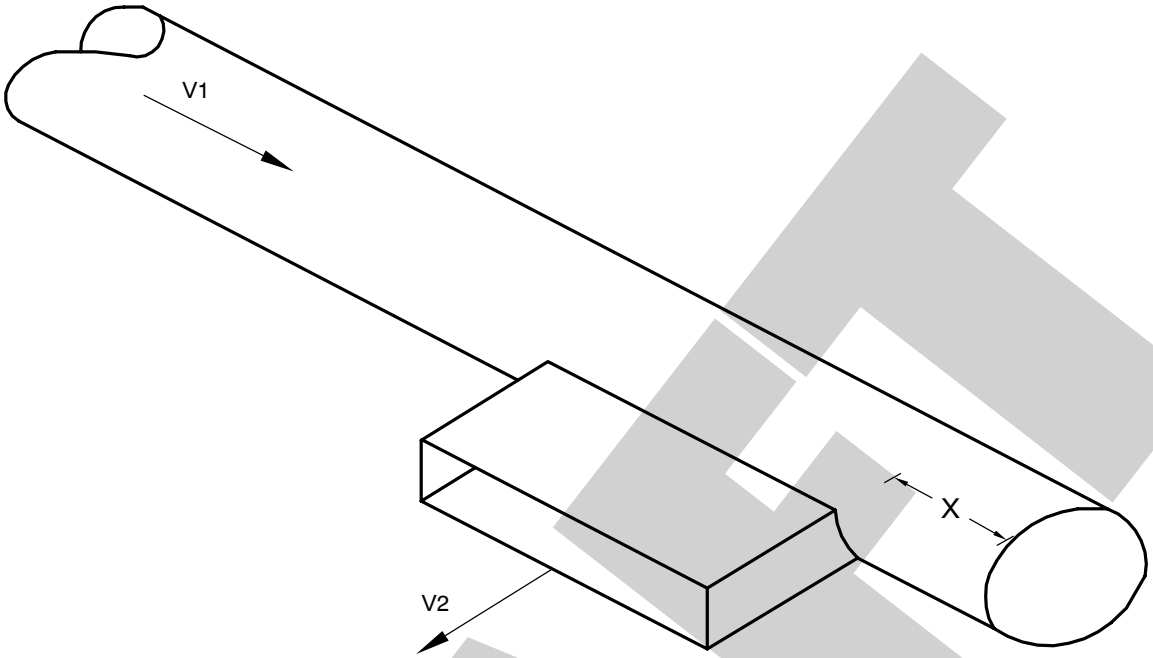


Rectangular tap near end of round duct

| V2/V1 | Fitting Loss Coefficient (C) | | |
|---|---|---|---|
| | X = 1 in. (25 mm) | X = 6 in. (150 mm) | X = 12 in. (300 mm) |
| 0.45 | 0.69 | 0.68 | 0.65 |
| 0.64 | 0.82 | 0.84 | 0.81 |
| 0.94 | 1.00 | 1.01 | 0.95 |
| 1.35 | 1.36 | 1.45 | 1.43 |

**FIGURE 5–22 END TAP RESEARCH**

Figure 5-21 shows three unusual configuration 90 degree elbows that were included in the above test project. Generally in all of the testing, the lowest value for a fitting loss coefficient was obtained at the highest test velocities, and the highest values were obtained at velocities below 1200 fpm (6 m/s).

However, other inaccuracies enter in as the velocity pressure is being reduced more rapidly because it is a function of the square of the reduced velocity. Existing data for a smooth radius 90 degree rectangular elbow, *see* Table A-10F with R/W = 0.75 and 1.0 is compared with that of Elbows A, B, and C in Figure 5-21. Note that when the throat of the 90 degree mitered elbow (Figure 5-20 is changed from 90 degree to 45 degree (Elbow A) or is made on a curved radius (Elbow B), the loss coefficient values are cut by 50 to 70 percent. This could amount to a substantial savings of pressure loss, i.e. energy.

### 5.23.2 Taps at End of Ducts

Many new duct systems are installed without supply outlets in place until the tenant space is leased and a floor plan submitted. If the last outlet in a duct is not at the very end, does the "cushion head" affect the loss coefficient? To answer this question, tests were made of 10 and 12 in. (250 and 300 mm) diameter ducts with $14 \times 6$ in. and $22 \times 8$ in. ($350 \times 150$ mm, and $550 \times 200$ mm) taps located 1, 6, and 12 in. (25, 150 and 300 mm) from the capped end of each round duct and tap.

The results were plotted and the fitting loss coefficients are shown in Figure 5-22. The surprise was that the distance from the tap to the end of the duct only changed the values by a small amount, but the velocity ratio between the tap and the main duct was somewhat proportional to the change in loss coefficient values. As this configuration also is essentially an elbow,

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compare the values with Tables A-10D and A-14Q in the Appendix.

## 5.24 FAN PRESSURES

In any HVAC system, a fan is the "pump" that causes air to flow within the duct system by creating a pressure difference between the fan intake and discharge. The two most common fan designs used in HVAC systems are centrifugal and axial flow.

### 5.24.1 Centrifugal Fan Pressure

The centrifugal fan design incorporates a rotating "cage" with parallel blades mounted around the perimeter. Return air enters the center of this rotating cage, and passes through the outer blades which impart kinetic energy to the now rotating air by centrifugal force.

### 5.24.2 Axial Flow Fan Pressure

The axial flow fan design incorporates a "straight through" flow with the air passing straight into the path of rotating blades. An axial flow fan produces a pressure difference from the change in velocity of the air passing through the rotating blades. Since the entering air column is not being rotated like the centrifugal fan, the axial flow fan does not impart a centrifugal force component to the airflow. There are three types of axial flow fans:

a. Propeller: Propeller fans are normally used at or near free air, and are mounted in a flat plate or inlet ring. Propeller fans are usually not attached to ductwork, and have a relatively slow rotational tip speed and large tip to inlet ring clearance. This allows moving a large volume of air with a low pressure difference.

b. Tubeaxial: Tubeaxial fans incorporate a close-fitting tube around the rotating blades with a smaller clearance between the blades and tube walls. Tubeaxial fans operate at much higher blade tip speeds which give them a much higher total pressure capability than a propeller fan.

c. Vaneaxial: Vaneaxial fans are essentially a tubeaxial fan, with closer fan tip clearances and includes fixed guide vanes. The guide vanes help to "straighten" the discharge airflow leaving the blades which improves the fan's pressure, efficiency, and noise characteristics.

## 5.25 FAN DEFINITIONS

### 5.25.1 Fan Air Volume

Fan air volume is the volume of air propelled by a fan in a given system. It is independent of the air density.

cfm: Volume in $ft^3$/min (L/s) of air handled by a fan at any air density.

scfm: Volume in standard $ft^3$/min (L/s) of standard air at a density 0.075 lb./$ft^3$ (1.204 kg/$m^3$) handled by a fan.

### 5.25.2 Fan Total Pressure (TP)

Fan total pressure is the difference between the total pressure at the fan outlet and the total pressure at the fan inlet. The fan total pressure is the measure of the total mechanical energy added to the air or gas by the fan. This is measured as illustrated in Figure 5-23.

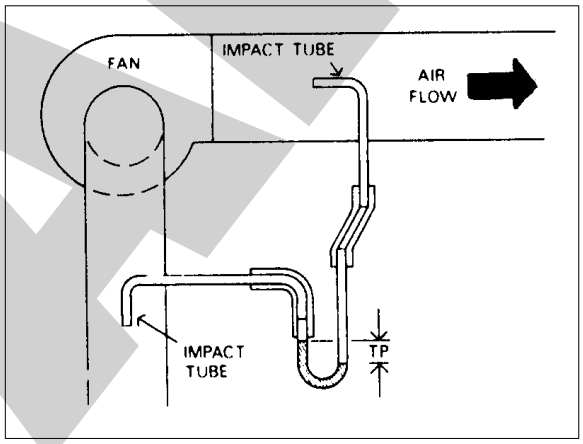


**FIGURE 5–23 FAN TOTAL PRESSURE (TP)**

### 5.25.3 Fan Static Pressure (SP)

Fan static pressure (Figure 5-24) is the fan total pressure, less the fan velocity pressure. It can be calculated by subtracting the total pressure at the fan inlet from the static pressure at the fan outlet. This is a source of some confusion within the industry, but by definition:

Fan SP = Fan TP(outlet)–TP (inlet) – $V_p$ (outlet)

Also,

TP (outlet) – SP (outlet) = $V_p$ (outlet)

and substituting, results in

Fan SP = SP (outlet) – TP (inlet)

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When taking field measurements, care must be taken when a duct inlet is used to measure the total pressure at the inlet, rather than just static pressure.

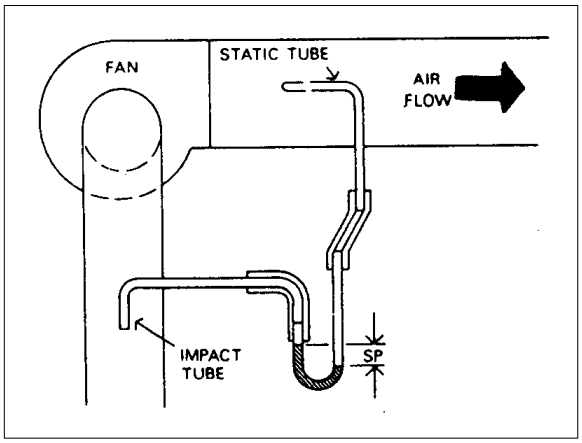


**FIGURE 5–24 FAN STATIC PRESSURE (SP)**

### 5.25.4 Fan Outlet Velocity

This is the theoretical velocity of the air as it leaves the fan outlet and is calculated by dividing the air volume by the fan outlet area. However, all fans have a non-uniform outlet velocity, that is, velocity varies over the cross-section of the fan outlet. Therefore, outlet velocity as calculated above is only a theoretical value that could occur at a point several duct diameters from the fan. Therefore all velocity (velocity pressure) readings, including total pressure and static pressure, should be taken farther along in a straight duct connected to the fan discharge where the airflow is more uniform.

### 5.25.5 Fan Velocity Pressure ($V_P$)

Fan velocity pressure is the pressure corresponding to the fan outlet velocity. It is the kinetic energy per unit volume of flowing air.

### 5.25.6 Brake Power

Brake power is the actual power required to drive the fan. It is greater than a theoretical "air horsepower" because it includes loss due to turbulence and other inefficiencies in the fan, plus bearing losses. Brake horsepower, or simply horsepower, is important because it is the power that the fan motor must furnish.

### 5.25.7 Tip Speed

Tip speed, also called peripheral velocity, equals the circumference of the fan wheel times the rpm of the fan and is expressed in ft per minute (meters per second)

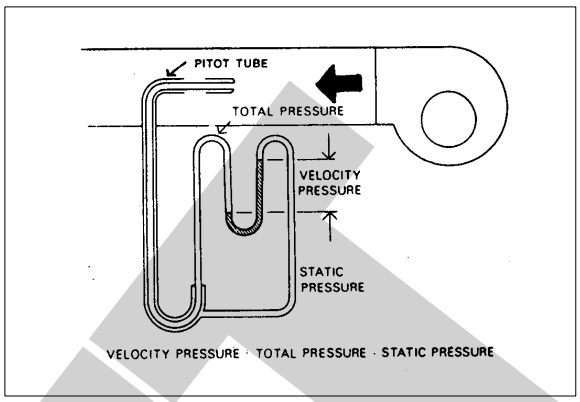


**FIGURE 5–25 FAN VELOCITY PRESSURE ($V_P$)**

as shown by Figure 5-30. Some countries use radians per second instead of revolutions per minute. There are $2\pi$ radians per revolution, so one radian per second (rad/s) equals 9.549 rpm.

**Equation 5–20 (I–P)**

$$Tip\ Speed = \frac{\pi \times D \times RPM}{12\ \text{in/ft}}$$

Where:

*Tip Speed* = Ft per minute

*D* = Wheel diameter – in.

*RPM* = Revolutions per minute

**Equation 5–20 (SI)**

$$Tip\ Speed = \frac{\pi \times D \times RPM}{60\ \text{sec/min}}$$

Where:

*Tip Speed* = Meters per second

*D* = Wheel diameter – meters

*RPM* = Revolutions per minute

**Example 5–27 (I–P)**

Find the tip speed of a 30-inch diameter fan wheel rotating at 954 rpm.

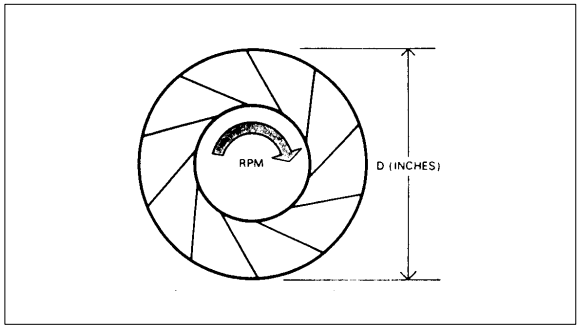


**FIGURE 5–26 TIP SPEED**

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$$Tip\ Speed = \frac{\pi \times D \times RPM}{12} = \frac{\pi \times 30 \times 954}{12}$$

*Tip Speed* = 7493 ft per minute (fpm)

**Example 5–27 (SI)**

Find the tip speed of a 750 mm diameter fan wheel rotating at 954 rpm.

$$Tip\ Speed = \frac{\pi \times D \times RPM}{60} = \frac{\pi \times 0.75 \times 954}{60}$$

*Tip Speed* = 37.46 meters/second (m/s)

## 5.26 FAN LAWS

The fan laws relate the performance variables for any dynamically-similar series of fans. The variables are fan size, rotational speed, air density, volume flow rate, static (or total) pressure, power, and mechanical efficiency. For HVAC duct systems, the following equations (fan laws) are used. $Q_2/Q_1$ may be substituted for $rpm_2/rpm_1$ in Equations 5-21 through 5-24.

**Equation 5–21**

$$\frac{Q_2}{Q_1} = \frac{rpm_2}{rpm_1}$$

**Equation 5–22**

$$\frac{P_2}{P_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$$

**Equation 5–23**

$$\frac{FP_2}{FP_1} = \left(\frac{rpm_2}{rpm_1}\right)^3$$

**Equation 5–24**

$$\frac{d_2}{d_1} = \frac{P_2}{P_1}\ when\ Q_1 = Q_2$$

Where:

$Q$ = Airflow – cfm (L/s)

$P$ = Pressure – in. wg (Pa)

$FP$ = Fan power = BHP (W)

$D$ = Density – lb/ft$^3$ (kg/m$^3$)

$rpm$ = Revolutions per minute

The fan laws apply only to a series of dynamically-similar fans at the same point of rating on the performance curve. They can be used to predict the performance of any fan, when test data are available for any fan of the same series. Fan laws may also be used with a particular fan to determine the effect of changing fan rpms. However, caution should be exercised in these cases, since they apply only when all flow conditions are similar. Changing the rpms of a given fan changes parameters that may invalidate the fan laws.

Unless otherwise identified, fan performance data is based on dry air at standard condition – 14.696 psi, 70 degree F and a density of 0.075 lb/ft$^3$ (101.325 kPa, 20 degree C and 1.204 kg/m$^3$). In actual applications, the fan may be required to handle air or gas at some other density. The change in density may be because of temperature, composition of the gas, or altitude. As indicated by the fan laws, the fan performance is affected by gas density. With constant size and speed, the horse-

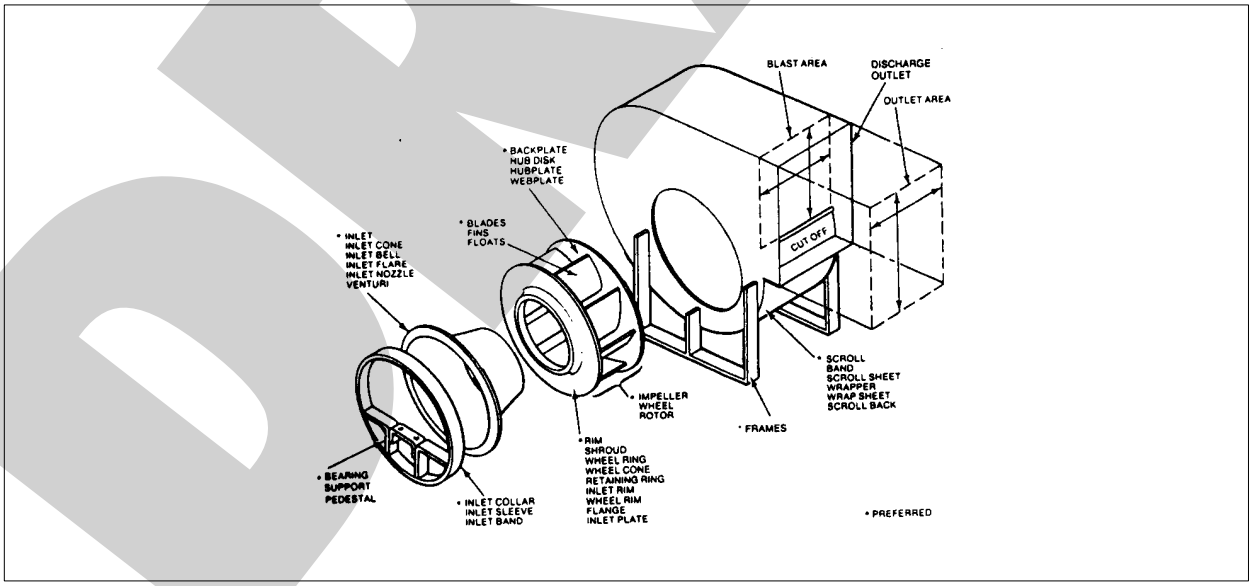


**FIGURE 5–27 CENTRIFUGAL FAN COMPONENTS**

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power and pressure vary in accordance with the ratio of gas density to the standard air density. The airflow (cfm or L/s) remains constant also.

Designers need to be cognizant of temperature and altitude correction factors and the physics behind them. Using the temperature correction factors to compensate for the in-operation higher temperatures of an HVAC system may save fan horsepower but that will only be acceptable if the fan has the power or duty rating to operate through the transition of a "cold start" until the higher "normal" operating temperature is achieved. Starting an HVAC system "outside" of design parameters can cause excessive fan motor overloads. Nearly double the operating horsepower has been reported when starting an HVAC system in cold temperatures; the same "physics" can also cause the return duct to "see" higher-than-design negative pressures. In special cases—especially where extreme measures are being pursued to save fan energy by using the smallest motor possible—it may be prudent to use temporary heat to assist the permanent HVAC system from a cold start to its normal operating temperature.

*Example 5-28*

A centrifugal fan has a brake horsepower of 4.6 HP (3.43 W) with airflow of 14,500 cfm (7250 L/s). Find the new fan power if the airflow is increased 10 percent.

Using Equation 5-24:

$$\frac{FP_2}{FP_1} = \left(\frac{rpm_2}{rpm_1}\right)^3 = \left(\frac{Q_2}{Q_1}\right)^3$$

$$FP_2 = FP_1 \times \left(\frac{Q_2}{Q_1}\right) = 4.6 \times \left(\frac{15,950}{14,500}\right)^3$$

$$FP_2 = 6.12\,HP\,(I - P)$$

$$FP_2 = 3.43 \times \left(\frac{7975}{7250}\right)^3$$

$$FP_2 = 4.57\,W\,(SI)$$

## 5.27 FAN TESTING

Figure 5-29 shows one of the most common procedures for developing the characteristics of a fan, in which it is tested from *shutoff* conditions to nearly *free delivery* conditions. At shutoff, the duct is completely blanked off; at free delivery, the outlet resistance is reduced to zero. Between these two conditions, various flow restrictions are placed on the end of the duct to simulate various conditions of the fan. Sufficient points are obtained to define the curve between the shutoff point and free delivery conditions. Pitot tube traverses of the test duct are performed with the fan operating at constant speed. The point of rating may be any point on the fan performance curve. For each case, the specific point on the curve must be defined by referring to the flow rate and the corresponding total pressure.

Fans designed for use with duct systems are tested with a length of duct between the fan and the measuring station. This length of duct smoothes the flow from the fan and provides stable, uniform-flow conditions at the plane of measurement. The measured pressures are corrected back to fan-outlet conditions. Fans designed for use without ducts, including propeller fans and power roof ventilators, are tested without ductwork.

Not all sizes are tested for rating. Test information may be used to calculate the performance of larger fans that are geometrically similar, such information should not be extrapolated to smaller fans. For the performance

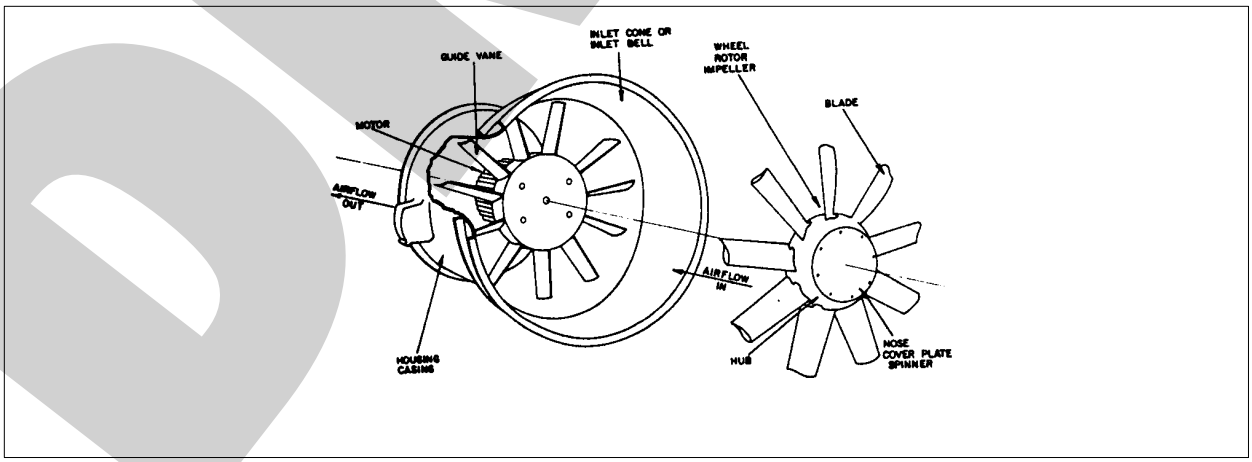


**FIGURE 5–28 AXIAL FAN COMPONENTS**

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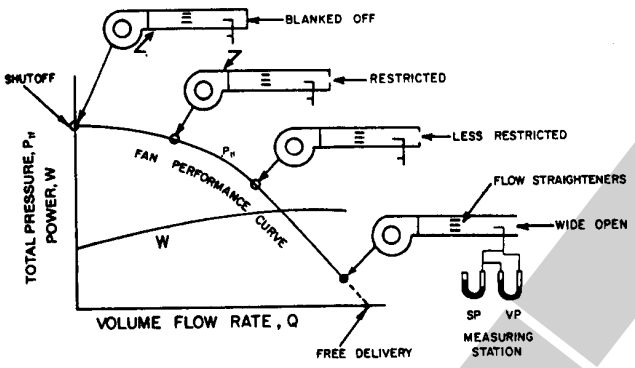


**FIGURE 5–29 METHOD OF OBTAINING FAN PERFORMANCE CURVES**

of one fan to be determined from the known performance of another, the two fans must be dynamically similar. Strict dynamic similarity requires that the important non-dimensional parameters vary in only insignificant ways. These non-dimensional parameters include those that affect aerodynamic characteristics such as Reynolds number, surface roughness, and gap size. For more specific information, the manufacturer's application manual or engineering data should be consulted.

## 5.28 FAN CLASSIFICATIONS

In a fan manufacturer's rating table, if the fan speeds and static pressures increase above certain given conditions, the "class" of the fan changes. Class again refers to an AMCA standard that has been developed to regulate actual structural limitations of the wheels, bearings, and housing of fans. Under the class standards, there are three classifications based on static pressure.

## 5.29 FAN TYPES

### 5.29.1 Centrifugal Fans

The first of the two basic types of fans normally encountered in HVAC systems is the centrifugal fan, with basic parts identified in Figure 5-27. Three variations of the centrifugal fan are commonly used.

### 5.29.2 Forward Curved Fans

The forward curve (FC) centrifugal fan operates at a relatively slow speed and generally is used for producing high volumes at low static pressures. The static pressure proportion of the total pressure discharge is 20 percent while the velocity pressure is 80 percent.

The typical operating range of this fan is from 30 to 80 percent of wide-open volume, *see* Figure 5-30. The maximum static efficiency of 60-68 percent generally occurs slightly to the right of the peak static pressure. The horsepower curve has an increasing slope, and is referred to as an "overloading type" fan.

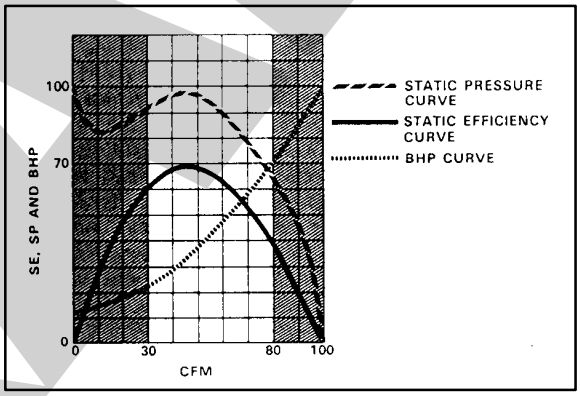


**FIGURE 5–30 CHARACTERISTIC CURVES FOR FC FANS**

The efficiency of a forward curved fan is somewhat less than airfoil and backward inclined fans. They usually are fabricated of lightweight and low-cost construction, and have shallow blades with both the heel and tip curved forward. For a given duty, the wheel is the smallest of all centrifugal types and operates at the lowest speed.

Advantages of the FC fan are: its low cost, the slow speed (that minimizes shaft and bearing size), and its wide operating range. The disadvantages are the unusual shape of its static pressure curve, which allows the possibility of overloading the motor if system static pressure decreases. It is also not suitable for material handling, because of its blade configuration. It has an inherently weak structure, so that forward curved fans

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generally are not capable of the high speeds necessary for developing higher static pressures.

### 5.29.3 Backward Inclined Fans

Backward inclined fans (BI) travel at about twice the revolutions per minute of the forward curved fan. The normal selection range of the backward inclined fan is approximately 40-85 percent of wide-open volume, *see* Figure 5-31. Maximum static efficiency of about 80 percent generally occurs close to the edge of its normal operating range. Generally, the larger the fan, the more efficient a given selection will be. The magnitude of surge for a BI fan is greater than for a FC fan.

The efficiency of the BI fan is only slightly less than that of airfoil fans. Backward inclined or backward curved blades are single thickness.

Advantages of the BI fan are its higher efficiency and non-overloading horsepower curve. The horsepower curve generally reaches a maximum in the middle of the normal operating range, thus overloading is normally not a problem. Inherently stronger design makes it suitable for the higher static pressure operation of 70 percent of the total pressure, measured at the fan discharge. This leaves the measured velocity pressure at only 30 percent.

Disadvantages include the higher speed that requires larger shaft and bearing sizes and places more importance on proper balance. Another disadvantage is the unstable operation that occurs as block-tight static pressure is approached.

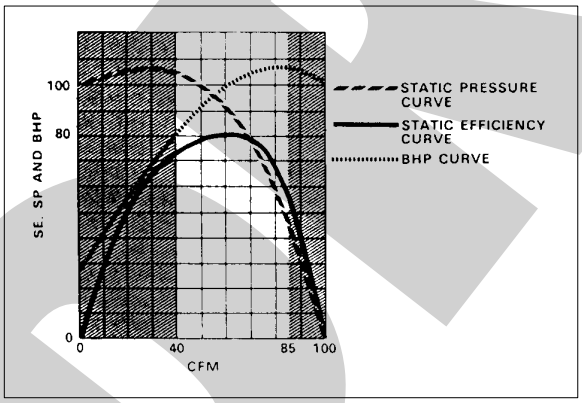


**FIGURE 5–31 CHARACTERISTIC CURVES FOR BI FANS**

### 5.29.4 Airfoil Fans

A refinement of the flat-bladed backward inclined fan is the use of airfoil-shaped blades. This improves the static efficiency to about 86 percent, the highest of all centrifugal fan types, and reduces the noise level slightly. The magnitude of surge also increases with the airfoil blades. Characteristic curves for airfoil fans are shown in Figure 5-32.

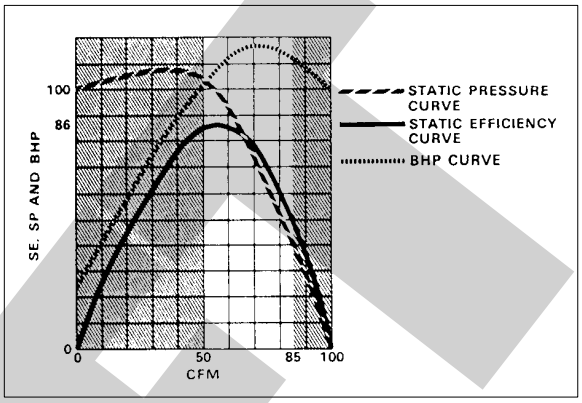


**FIGURE 5–32 CHARACTERISTIC CURVES FOR AIRFOIL FANS**

The airfoil fan has blades that are curved away from the direction of rotation. Air leaves the impeller at a velocity less than its tip speed. For given conditions, the airfoil fan will operate at the highest speed of all types of centrifugal fans.

### 5.29.5 Tubular Centrifugal Fans

Tubular centrifugal fans, as illustrated in Figure 5-33, generally consist of a single-width airfoil wheel arranged in a cylinder to discharge air radically against the inside of the cylinder. Air is then deflected parallel with the fan shaft to provide straight-through flow. Vanes are used to recover static pressure and to straighten airflow.

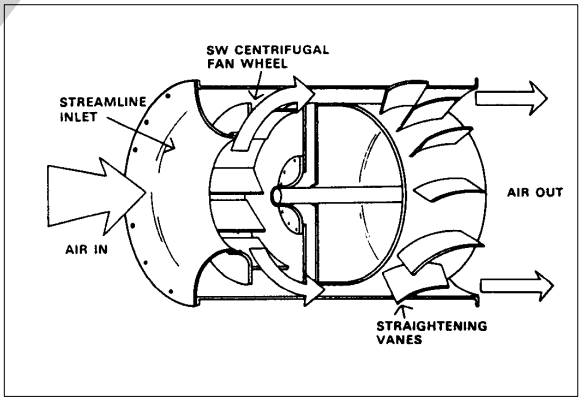


**FIGURE 5–33 TUBULAR CENTRIFUGAL FAN**

Characteristic curves are shown in Figure 5-34. The selection range is generally about the same as the scroll type BI or airfoil-bladed wheel, 50 to 85 percent of wide-open volume. However, because there is no housing of the turbulent airflow path through the fan,

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static efficiency is reduced to a maximum of about 72 percent and noise level is increased.

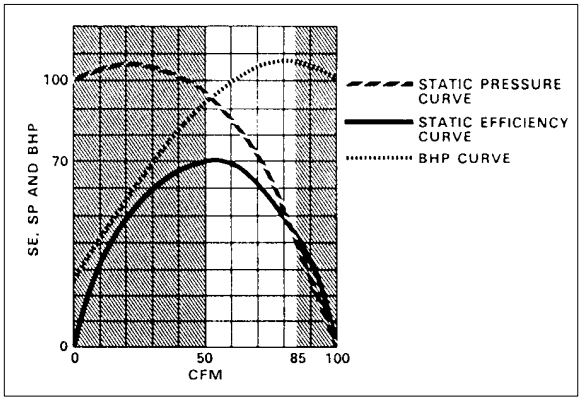


**FIGURE 5–34 CHARACTERISTIC CURVES FOR TUBULAR CENTRIFUGAL FANS**

Frequently, the straight-through flow results in significant space savings. This is the main advantage of tubular centrifugal fans.

### 5.29.6 Axial Fans

The second basic type of HVAC fan is the axial fan. Its parts are identified in Figure 5-28. Axial fans are divided into three groups – propeller, tubeaxial, and vaneaxial.

### 5.29.7 Propeller Fans

The propeller fan (Figure 5-35) is well suited for high volumes of air at little or no static pressure differential. The efficiency is low and the energy transfer primarily is in the form of velocity pressure. Usually of inexpensive construction, the impeller is made of two or more blades of single-thickness metal attached to a relatively small hub.

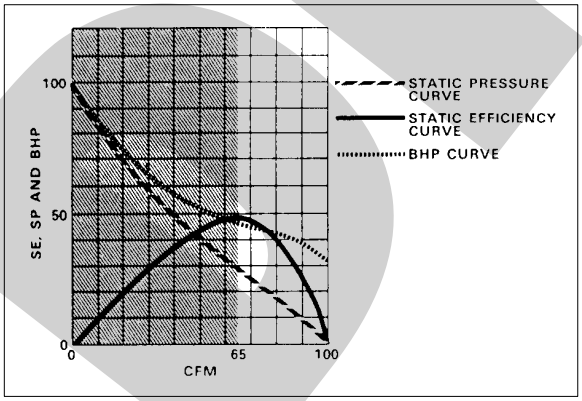


**FIGURE 5–35 CHARACTERISTIC CURVES FOR PROPELLER FANS**

### 5.29.8 Tubeaxial and Vaneaxial Fans

Tubeaxial and vaneaxial fans are simply propeller fans mounted in a cylinder, and are similar except for vane-type strengtheners on the vaneaxial. These vanes remove much of the swirl from the air and improve the efficiency. Thus, a vaneaxial fan is more efficient than a tubeaxial and can achieve higher static pressures. Note that with axial fans the fan power is maximum at the block-tight static pressure. With centrifugal fans, the fan power is minimum at block-tight static pressure.

Tubeaxial and vaneaxial fans are generally used for handling large volumes of air at low static pressures.

Advantages of tubeaxial and vaneaxial flow fans are the reduced size and weight, and the straight-through airflow that frequently eliminates elbows in the ductwork. The maximum static efficiency of an industrial vaneaxial fan is approximately 65 percent. The operating range for axial fans is from 65 to 90 percent.

Disadvantages of axial fans are the high noise level and lower efficiency.

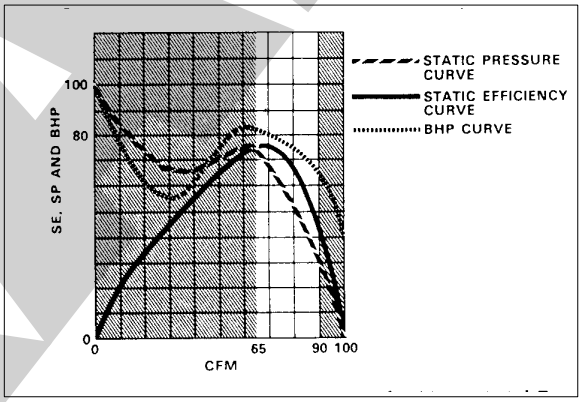


**FIGURE 5–36 CHARACTERISTIC CURVES FOR VANEAXIAL FANS**

## 5.30 FAN CURVES

Figure 5-37 illustrates the application of the fan laws for a change in fan speed, (N), for a specific size fan. The computed total pressure (TP) curve is derived from the base curve. For example, point E ($N_2$ = 650 rpm) is computed from point D ($N_1$ = rpm) as follows:

(I-P)

At point D:

$$Q_1 = 6000 \text{ cfm and } TP_1 = 1.13 \text{ in.wg}$$

Using Equation 5-22 at point E:

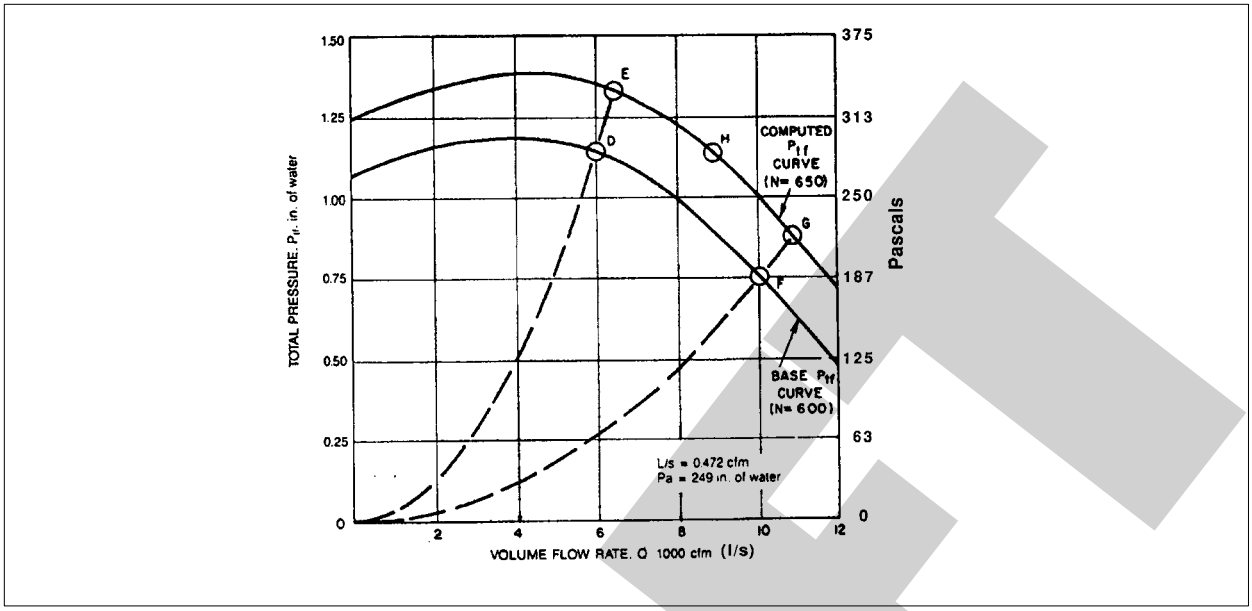


**FIGURE 5–37 APPLICATION OF THE FAN LAWS**

$$Q_2 = 6000 \times \left(\frac{650}{600}\right) = 6500 \text{ cfm}$$

Using Equation 5-23:

$$TP_2 = 1.13 \times \left(\frac{650}{600}\right)^2 = 1.33 \text{ in.wg}$$

The completed total pressure curve, $T_2$ at $N_2$ = 650 rpm curve thus may be generated by computing additional points from data on the base curve, such as point G from point F.

If equivalent points of rating are joined, as shown by the dotted lines in Figure 5-37, these points will form parabolas that are defined by the relationship expressed in Equation 5-23.

Each point on the base $TP_1$ curve determines only one point on the computed curve. For example, point H cannot be calculated from either point D or point F. Point H is, however, related to some point between these two points on the base curve and only that point can be used to locate point H. Furthermore, point D cannot be used to calculate point F on the base curve. The entire base curve must be defined by testing.

(SI)

The above example calculations follow in metric units.

At point D:

$$Q_1 = 3000 \text{ L/s and } TP_1 = 283 \text{ Pa}$$

Using Equation 5-22 at point E:

$$Q_2 = 3000 \times \left(\frac{650}{600}\right) = 3250 \text{ L/s}$$

Using Equation 5-23:

$$TP_2 = 283 \times \left(\frac{650}{600}\right) = 332 \text{ Pa}$$

A typical centrifugal fan performance table may be found in Figure 5-38. If the fan curve for this fan is not available, one can plot a curve for each rpm found in the table.

*Example 5-29*

Plot a fan curve using data from Figure 5-38 for approximately 510 rpm (Fan Curve A) and 400 rpm (Fan Curve B).

Draw a graph using 0 to 1.5 in. wg (0 to 375 Pa) static pressure vertically and from 7000 to 14,000 cfm (3500 to 7000 L/s) airflow horizontally, *see* Figure 5-39. Interpolating where necessary, plot the airflow opposite each 0.25 in. wg (63 Pa) increase in static pressure for 510 rpm (Fan Curve A) and draw a curve through the points. Do the same for 440 rpm (Fan Curve B).

Note that at the last points plotted, higher static pressures, the curve's rise was lessening. Compare the plotted curves to the fan curves in Figures 5-30 to 5-32.

Plot a metric unit (Fan Curve C) in a similar manner using metric data from the centrifugal fan performance table in Figure 5-39 for 545 rpm. The fan curve

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| | OV | VP | 100 Pa SP | | 150 Pa SP | | 200 Pa SP | | 250 Pa SP | | 300 Pa SP | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | m/s | Pa | rpm | kW | rpm | kW | rpm | kW | rpm | kW | rpm | kW |
| 4292 | 4.0 | 10 | 270 | .59 | 307 | .84 | 342 | 1.1 | 376 | 1.4 | 408 | 1.7 |
| 4828 | 4.5 | 12 | 286 | .70 | 319 | .97 | 352 | 1.3 | 383 | 1.6 | 413 | 1.9 |
| 5365 | 5.0 | 15 | 303 | .83 | 334 | 1.1 | 363 | 1.4 | 392 | 1.7 | 421 | 2.1 |
| 5901 | 5.5 | 18 | 321 | .98 | 349 | 1.3 | 377 | 1.6 | 404 | 1.9 | 430 | 2.3 |
| 6437 | 6.0 | 22 | 339 | 1.1 | 366 | 1.5 | 392 | 1.8 | 417 | 2.2 | 442 | 2.5 |
| 6974 | 6.5 | 25 | 358 | 1.3 | 384 | 1.7 | 408 | 2.1 | 432 | 2.4 | 455 | 2.8 |
| 7510 | 7.0 | 29 | 378 | 1.5 | 402 | 1.9 | 425 | 2.3 | 448 | 2.7 | 470 | 3.1 |
| 8047 | 7.5 | 34 | 398 | 1.7 | 421 | 2.2 | 443 | 2.6 | 464 | 3.0 | 485 | 3.4 |
| 8583 | 8.0 | 38 | 419 | 2.0 | 441 | 2.4 | 462 | 2.9 | 482 | 3.3 | 501 | 3.8 |
| 9120 | 8.5 | 43 | 440 | 2.3 | 460 | 2.7 | 480 | 3.2 | 500 | 3.7 | 519 | 4.2 |
| 9656 | 9.0 | 49 | 461 | 2.6 | 480 | 3.1 | 500 | 3.6 | 518 | 4.1 | 536 | 4.6 |
| 10193 | 9.5 | 54 | 482 | 2.9 | 501 | 3.4 | 519 | 4.0 | 537 | 4.5 | 554 | 5.0 |
| 10729 | 10.0 | 60 | 504 | 3.3 | 521 | 3.8 | 539 | 4.4 | 556 | 4.9 | 573 | 5.5 |
| 11802 | 11.0 | 73 | 547 | 4.1 | 563 | 4.7 | 580 | 5.3 | 595 | 5.9 | 611 | 6.6 |
| 12875 | 12.0 | 86 | | | 606 | 5.7 | 621 | 6.4 | 636 | 7.1 | 650 | 7.7 |
| 13948 | 13.0 | 101 | | | 649 | 6.9 | 663 | 7.6 | 677 | 8.4 | 690 | 9.1 |

**FIGURE 5–39 CENTRIFUGAL FAN PERFORMANCE TABLE (METRIC UNITS)**

IMPELLER DIAMETER 36-1/2 INCHES
OUTLET AREA 7.65 SQ FT
TIP SPEED IN FPM 9.56 X RPM
MAXIMUM BHP 18.3 x (RPM/100)$^3$

| Volume | Outlet | 1/4" SP | | 3/8" SP | | 1/2" SP | | 5/8" SP | | 3/4" SP | | 7/8" SP | | 1" SP | | 1-1/4" SP | | 1-1/2" SP | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| CFM | Velocity | RPM | BHP | RPM | BHP | RPM | BHP | RPM | BHP | RPM | BHP | RPM | BHP | RPM | BHP | RPM | BHP | RPM | BHP |
| 3825 | 500 | 222 | .185 | | | | | | | | | | | | | | | | |
| 4590 | 600 | 236 | .233 | 270 | .334 | | | | | | | | | | | | | | |
| 5355 | 700 | 253 | .292 | 284 | .400 | 313 | .519 | | | | | | | | | | | | |
| 6120 | 800 | 272 | .365 | 300 | .483 | 327 | .608 | 352 | .743 | | | | | | | | | | |
| 6885 | 900 | 292 | .450 | 317 | .579 | 343 | .716 | 366 | .856 | 389 | 1.01 | 411 | 1.17 | | | | | | |
| 7650 | 1000 | 314 | .560 | 337 | .695 | 360 | .840 | 382 | .992 | 403 | 1.15 | 424 | 1.31 | 443 | 1.48 | | | | |
| 8415 | 1100 | 338 | .682 | 358 | .832 | 378 | .981 | 399 | 1.144 | 419 | 1.31 | 438 | 1.48 | 458 | 1.60 | 494 | 2.04 | | |
| 9180 | 1200 | 361 | .826 | 379 | .988 | 398 | 1.149 | 417 | 1.314 | 436 | 1.49 | 455 | 1.68 | 472 | 1.86 | 507 | 2.25 | 540 | 2.67 |
| 9945 | 1300 | 385 | .989 | 402 | 1.163 | 419 | 1.340 | 437 | 1.514 | 454 | 1.69 | 472 | 1.89 | 489 | 2.09 | 522 | 2.49 | 554 | 2.92 |
| 10710 | 1400 | 409 | 1.175 | 426 | 1.360 | 441 | 1.553 | 457 | 1.741 | 473 | 1.93 | 489 | 2.12 | 506 | 2.34 | 538 | 2.76 | 568 | 3.20 |
| 11475 | 1500 | 434 | 1.387 | 449 | 1.587 | 464 | 1.78 | 479 | 1.993 | 494 | 2.19 | 509 | 2.40 | 524 | 2.61 | 555 | 3.06 | 584 | 3.52 |
| 12240 | 1600 | 458 | 1.626 | 473 | 1.837 | 488 | 2.048 | 501 | 2.269 | 515 | 2.49 | 529 | 2.70 | 543 | 2.92 | 572 | 3.39 | 600 | 3.87 |
| 13005 | 1700 | 483 | 1.895 | 498 | 2.115 | 511 | 2.346 | 525 | 2.570 | 537 | 2.80 | 550 | 3.03 | 564 | 3.26 | 590 | 3.73 | 617 | 4.24 |
| 13770 | 1800 | 508 | 2.191 | 522 | 2.424 | 535 | 2.665 | 538 | 2.901 | 560 | 3.15 | 572 | 3.40 | 585 | 3.64 | 610 | 4.12 | 635 | 4.63 |
| 14535 | 1900 | | | 547 | 2.767 | 559 | 3.017 | 571 | 3.276 | 584 | 3.52 | 595 | 3.78 | 606 | 4.04 | 630 | 4.55 | 654 | 5.07 |
| 15300 | 2000 | | | 571 | 3.144 | 584 | 3.403 | 595 | 3.672 | 607 | 3.93 | 618 | 4.21 | 629 | 4.48 | 651 | 5.02 | 674 | 5.56 |
| 16830 | 2200 | | | 621 | 4.003 | 633 | 4.289 | 644 | 4.577 | 654 | 4.87 | 665 | 5.16 | 675 | 5.46 | 695 | 6.06 | 715 | 6.65 |
| 18360 | 2400 | | | | | 682 | 5.335 | 693 | 5.632 | 703 | 5.96 | 712 | 6.28 | 721 | 6.61 | 741 | 7.24 | 759 | 7.90 |
| 19890 | 2600 | | | | | | | 742 | 6.885 | 752 | 7.22 | 761 | 7.56 | 769 | 7.91 | 788 | 8.60 | 805 | 9.30 |
| 21420 | 2800 | | | | | | | 791 | 8.308 | 801 | 8.67 | 810 | 9.03 | 818 | 9.40 | 834 | 10.15 | 852 | 10.88 |
| 22950 | 3000 | | | | | | | | | 850 | 10.32 | 859 | 10.71 | 867 | 11.09 | 883 | 11.89 | 898 | 12.70 |
| 24480 | 3200 | | | | | | | | | | | 908 | 12.60 | 916 | 13.01 | 932 | 13.84 | 946 | 14.70 |
| 26010 | 3400 | | | | | | | | | | | | | 965 | 15.16 | 981 | 16.03 | 995 | 16.92 |
| 27540 | 3600 | | | | | | | | | | | | | 1015 | 17.52 | 1030 | 18.47 | 1044 | 19.39 |
| 29070 | 3800 | | | | | | | | | | | | | | | 1072 | 21.16 | 1093 | 22.13 |
| 30600 | 4000 | | | | | | | | | | | | | | | 1129 | 24.11 | 1142 | 25.16 |

**FIGURE 5–38 CENTRIFUGAL FAN PERFORMANCE TABLE (I–P)**

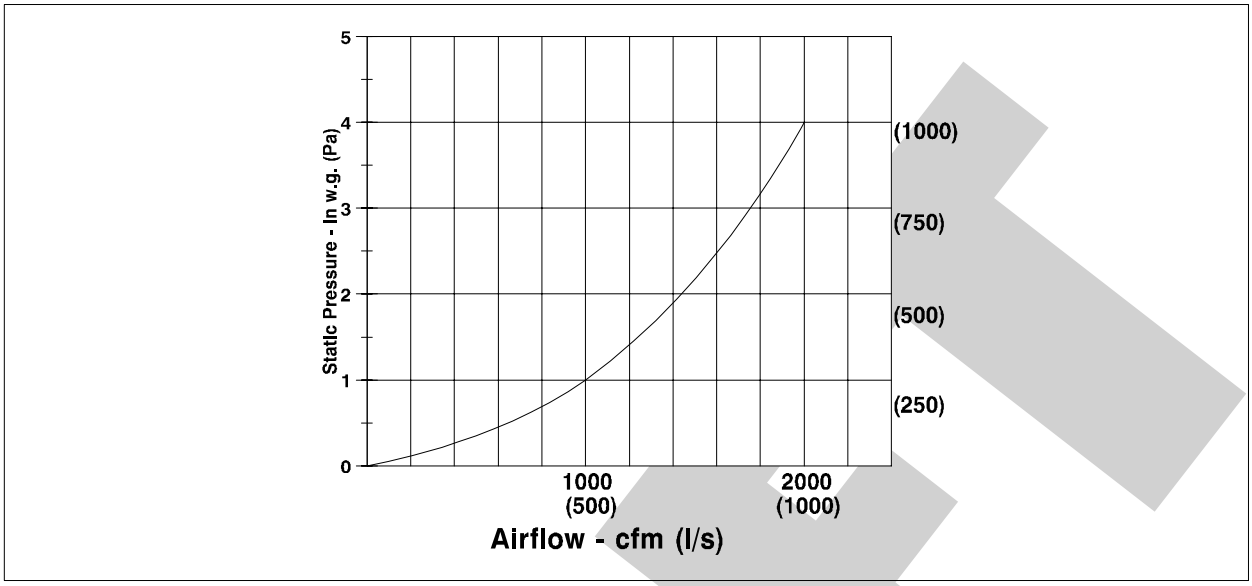


**FIGURE 5–40 SYSTEM RESISTANCE CURVE**

will have a different shape because it is a different size and capacity fan than used for fan curves "A" and "B".

### 5.31 DUCT SYSTEM AIRFLOW

An HVAC air distribution system may consist simply of a fan with ductwork connected to either the inlet or discharge or to both. A more complicated system may include a fan, ductwork, air control dampers, cooling coils, heating coils, filters, diffusers, sound attenuation, and turning vanes. The fan is the component or "air pump" in the system which provides energy to the airstream to overcome the resistance to flow of the other components. The discussion in this chapter and the accompanying tables and figures on fan and system curves were developed by the Air Moving and Conditioning Association, Inc. and reprinted with some minor editing with their permission. (AMCA Publication 201 – *Fans and Systems.*)

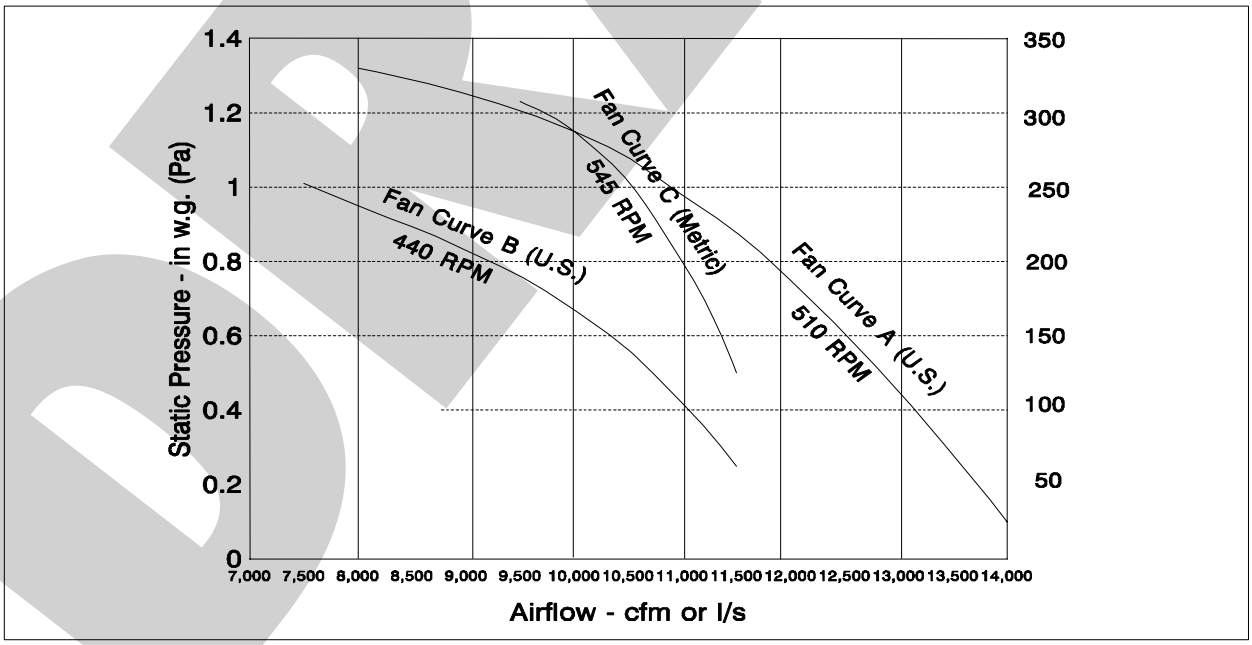


**FIGURE 5–41 FAN CURVE PLOTS**

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### 5.31.1 Component Losses

Each duct system has a combined set of pressure resistances to flow, which are usually different from every other system and are dependent upon individual duct system components.

The amount of the total pressure drop or resistance to flow for the individual duct system components can be obtained from the component manufacturer. For preliminary computations, some pressure data is available in Chapter 8.

## 5.32 SYSTEM CURVES

### 5.32.1 Establishing a System Curve

Duct system resistance is the sum total of all pressure losses in a duct system, including duct friction losses, fitting dynamic pressure losses, pressure drops through coils, filters, dampers, and terminal devices. For a system where all pressure losses are constant, with no changes in damper settings, the system resistance will vary as the square of the airflow as shown by Equation 5-26.

**Equation 5–25**

$$\frac{SP_2}{SP_1} = \left(\frac{Q_2}{Q_1}\right)^2$$

Where:

$SP$ = Static Pressure – in. wg (Pa)

$Q$ = Airflow – cfm (L/s)

The system resistance curve for any system is represented by a single curve that is a plot of the pressure required to move a specific amount of air through that system, *see* Figure 5-41.

For example, a system handling 1000 cfm (500 L/s) has a total measured resistance of 1 in. wg (250 Pa). If the airflow is doubled, the static pressure resistance will increase by that ratios squared to 4 in. wg (1000 Pa) as shown in Figure 5-41 by using Equation 5-26:

$$\frac{SP_2}{SP_1} = \left(\frac{Q_2}{Q_1}\right)^2$$

$$SP_2 = 1\left(\frac{2000}{1000}\right)^2 = 4 \text{ in.wg (U.S.)}$$

$$SP_2 = 250\left(\frac{1000}{500}\right)^2 = 1000 \text{ Pa (Metric)}$$

This plotted curve changes, however, as filters load with dirt, coils start condensing moisture, or when dampers change position.

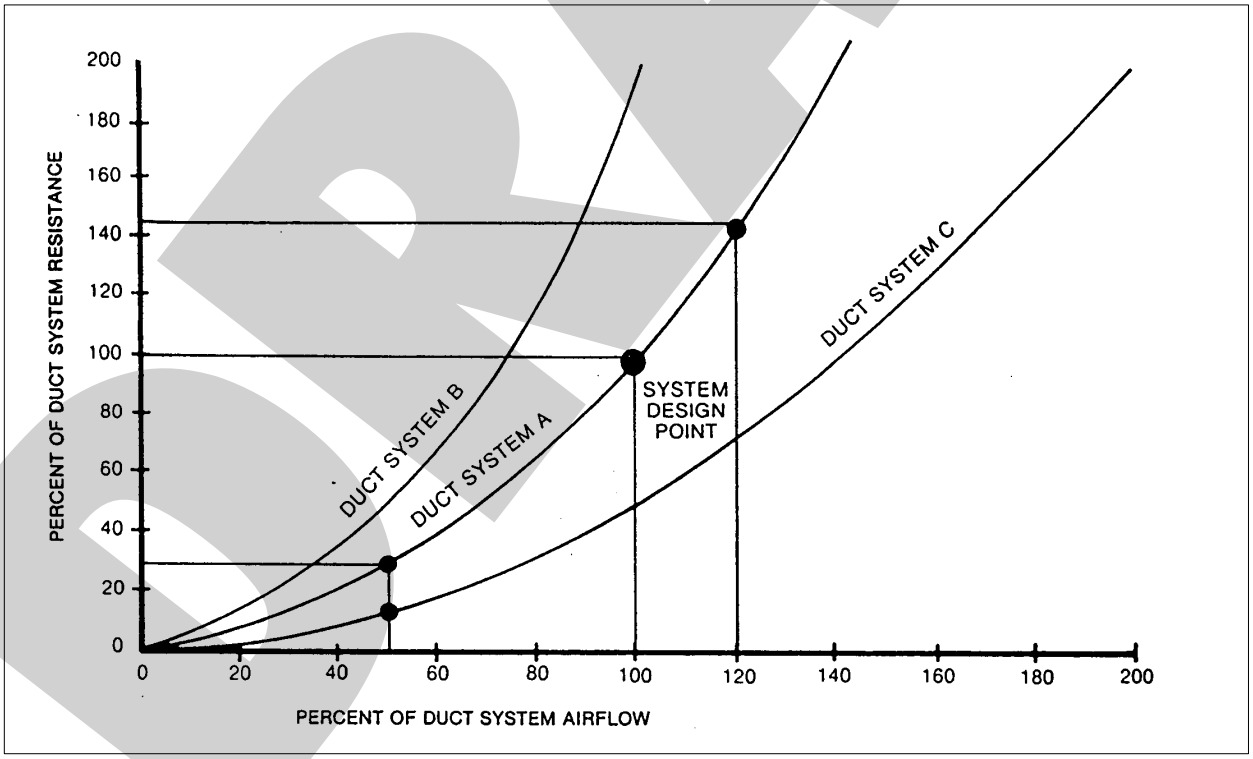


**FIGURE 5–42 NORMALIZED DUCT SYSTEM CURVES**

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Typical plots of the resistance to flow versus volume flow rate for three different and arbitrary fixed systems (A, B, and C) are illustrated in Figure 5-37. For a fixed system, an increase or decrease in the system airflow volume rate will increase or decrease the system resistance along the given system curve only.

Referring to System Curve A in Figure 5-37, assume a system design point at 100 percent volume and 100 percent resistance. If the airflow rate is increased to 120 percent of design volume, the system resistance will increase to 144 percent of the design resistance in accordance with Equation 5-26. A further increase in volume results in a corresponding increase in system pressure. A decrease in airflow to 50 percent of design volume would result in a decrease to 25 percent of the design resistance.

Notice that on a percentage basis, the same relationships also hold for the System Curves B and C. These relationships are characteristic of typical fixed systems.

If the system characteristic curve, composed of the resistance to flow of the system and the appropriate "System Effect Factors" has been accurately determined, then the fan selected will develop the necessary pressure to meet the system requirements at the designed airflow rate.

### 5.32.2 System Operating Point

When a system curve is plotted on the fan curve of the fan used (Figure 5-37) for that system, the operating point at which the fan and system will perform is determined by the intersection of the system curve and the fan performance curve. Every fan operates only along its performance curve for a given rpm. If the designed system resistance is not the same as the resistance in the installed system, the operating point will change and the static pressure and volume of air delivered will not be as calculated.

In Figure 5-39 the actual system pressure drop is greater than predicted by the system designer, so a new system curve indicates the lower airflow volume. The point of operation would move along the fan curve to the new system curve. A typical fan power curve (shown in Figure 5-38) would indicate a decrease in fan power.

At any given constant speed and horsepower, as the fan curve moves to the right, the airflow increases. But in order for it to increase, the static pressure must decrease. When a duct system curve is drawn on the same fan

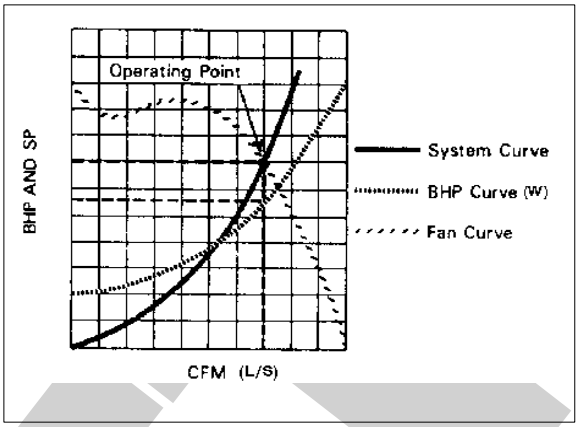


**FIGURE 5–43 OPERATING POINT**

curve, it moves up as it progresses to the right as shown in Figure 5-45. The duct system curve also predicts performance, as the airflow increases the same as the fan curve, at some point, the two curves will intersect. This intersection point (operating point) will predict the performance of this fan on this system. The duct system curve does enforce the fundamental statement on duct systems, that is, as the airflow increases, the system static pressure also increases.

What this means is each duct system has its own unique system curve, and each fan has a fan curve for each speed or rpm. Where the two curves intersect will be the operating point of both the fan and duct system. It now can be seen that both theories of static pressure versus airflow are correct. In the system, the static pressure does rise as the airflow increases. However, the static pressure usually decreases as the airflow increases for the fan, at least in the right hand portion of the fan curves.

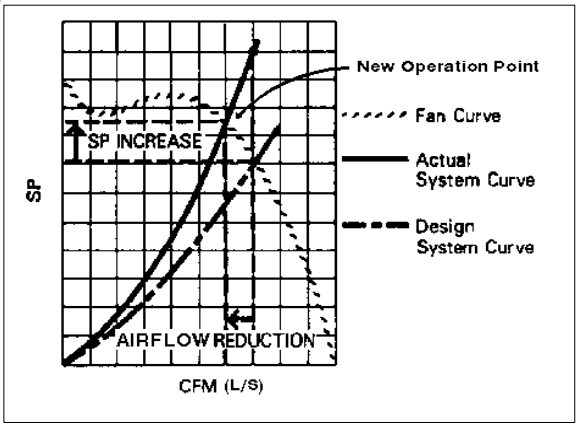


**FIGURE 5–44 VARIATIONS FROM DESIGN – AIR SHORTAGE**

Using fan laws (Equations 5-21 to 5-24), fan curves, and system curves, any change to any component can be calculated and graphically portrayed. Looking at

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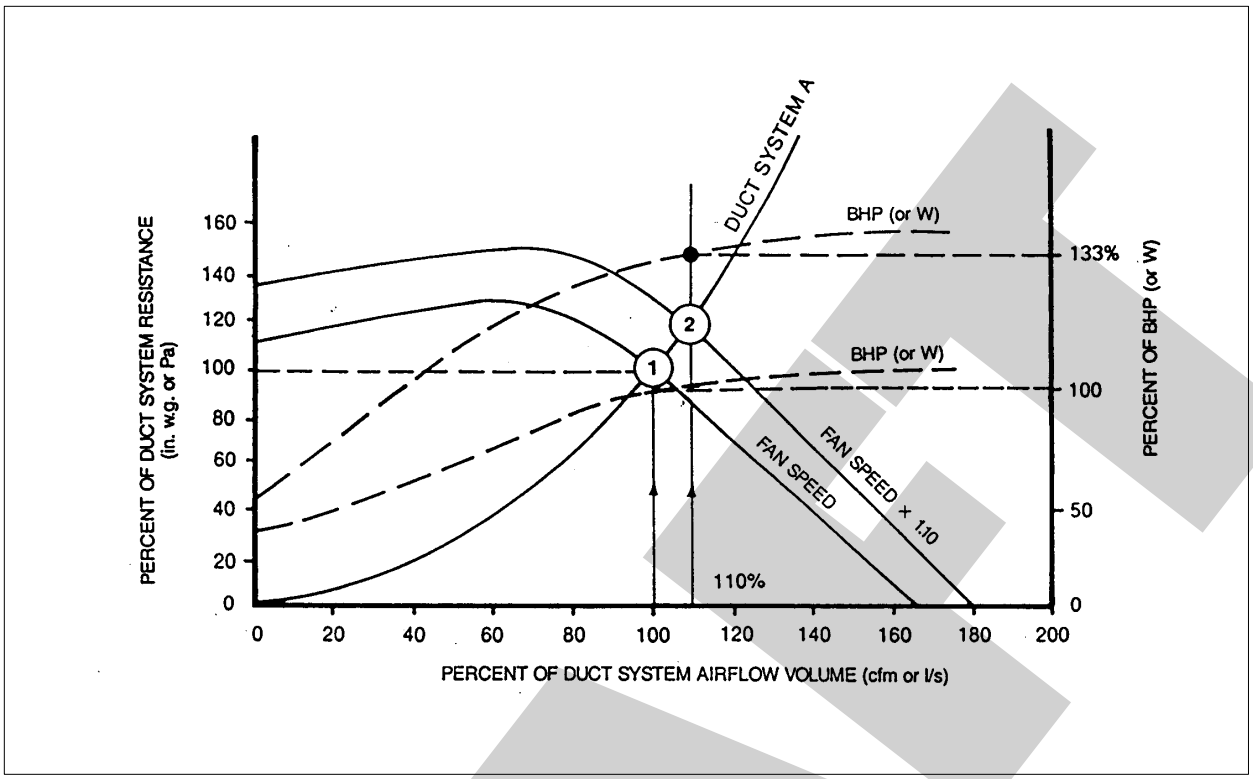


**FIGURE 5–45 EFFECT OF 10 PERCENT INCREASE IN FAN SPEED**

operating point 1 of the system curve and fan curve in Figure 5-45, if the airflow is increased, the operating point moves to the right. Would the static pressure increase or decrease? Either answer may be correct, because it depends on which curve the operating point is moved to the right.

To move the operating point, a physical change must be made to the fan or to the system. It could be to the fan, for example, with a change in pulley sheave size to speed up or slow down the fan. An increase in speed will increase the airflow and increase the static pressure, since the system curve will remain unchanged. The new fan curve will become a higher but parallel curve. The operating point will travel along the system curve to the new fan speed curve.

Increases or decreases in fan speed will alter the airflow rate through a system. Figure 5-45 illustrates the increase in airflow when the speed of a fan is increased 10 percent to operating point 2. The 10 percent increase in airflow, however, extracts a severe fan power penalty. According to the fan laws, the brake fan power increase is 33 percent. This is often startling to the system designer who finds that the system airflow is too low. Only 10 percent more air is needed, but the selected motor power is not capable of a 33 percent increase in load and the fan may need to be moved into another pressure classification.

The increase in power requirements is the result of an increase in work, a greater volume of air is moved by the fan against the resulting higher system resistance to the airflow. In the same system, the brake fan power increases, as the "cube" of the speed ratio changes, and the fan efficiency remains the same at all points on the same system curve.

The airflow rate through the system in a given installation may be varied by changes in the system resistance from fan inlet-vane dampers, resetting balancing dampers and the operation of mixing boxes or terminal units. Referring to Figure 5-46, the airflow rate may be varied from 100 percent of the design airflow volume (Point 1, Curve A) to approximately 80 percent of the design volume by increasing the system resistance (system Curve B). This results in fan operation at Point 2, the intersection of the fan curve and the new system Curve B. Similarly, the flow rate can be increased to approximately 120 percent of the design volume by decreasing the system resistance, thus changing the system curve characteristic to Curve C. This results in fan operation at Point 3, the intersection of the fan curve and the new System Curve C.

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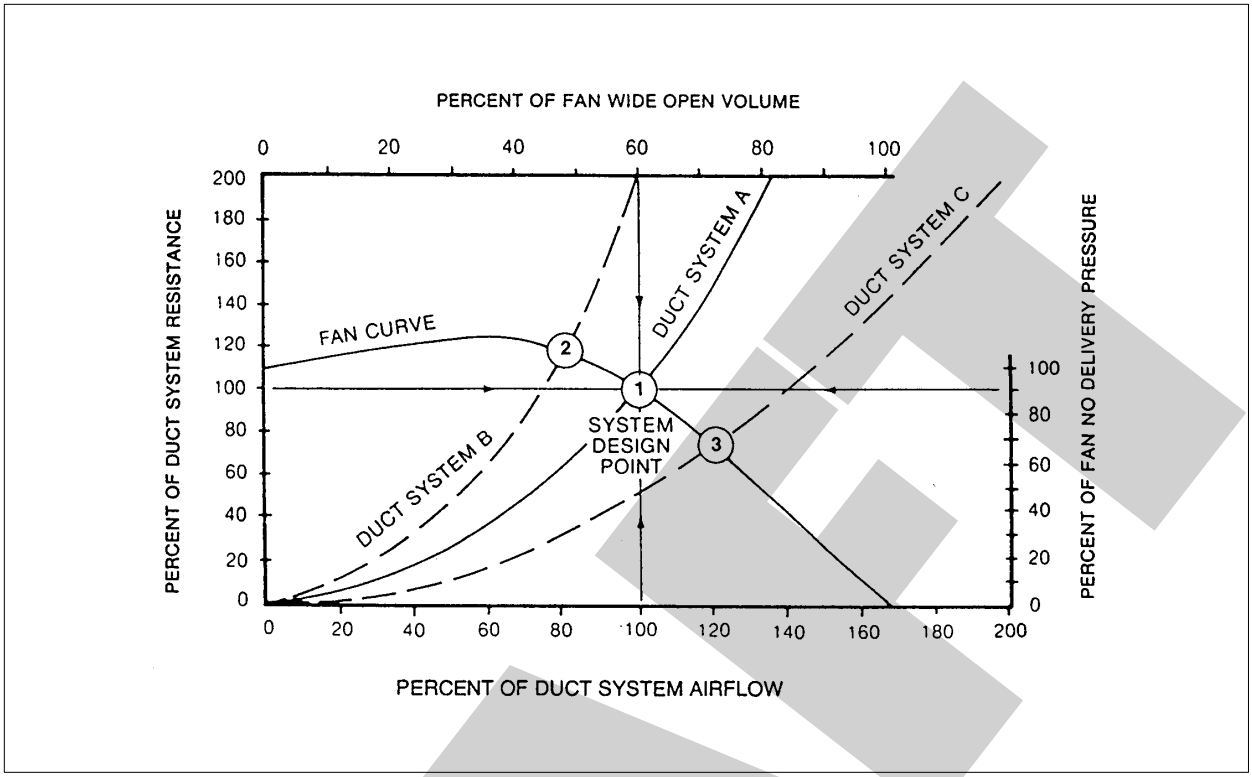


**FIGURE 5–46 INTERACTIONS OF SYSTEM CURVES AND FAN CURVE**

## 5.33 AIR DENSITY EFFECTS

### 5.33.1 Constant Air Volume

The resistance of an HVAC duct system is dependent on the density of the air flowing through the system. Air density at standard condition of 0.075 lbs/ft$^3$ (1.2041 kg/m$^3$) is used for rating fans in the HVAC industry. A fan is a constant volume machine and will produce the same volume of airflow regardless of the air density being handled, *see* Figure 5-47. The fan static pressure and fan power, however, will vary directly as the air density increases or decreases.

**Equation 5–26**

$$\frac{SP_2}{SP_1} = \frac{d_2}{d_1}$$

**Equation 5–27**

$$\frac{FP_2}{FP_1} = \frac{d_2}{d_1}$$

Where (airflow$_1$ = airflow$_2$):

$SP$ = Static pressure – in. wg (Pa)

$d$ = Density – lb/ft$^3$ (kg/m$^3$)

$FP$ = Fan power = Bhp (W)

In other words, the heavier or denser the air, the greater the fan power of static pressure will be.

*Example 5-30 (I-P)*

A 15,000 cfm fan is delivering 250 degree F air from an oven through an air-to-air heat exchanger against 2.6 in. wg SP. The fan is drawing 9.22 Bhp from a 10 HP motor. If the fan is run with the oven off (70 degree

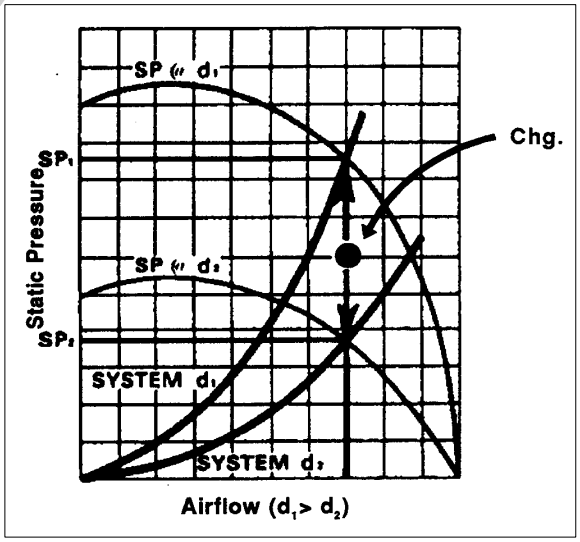


**FIGURE 5–47 EFFECT OF DENSITY CHANGE (CONSTANT VOLUME)**

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F ambient), calculate the new SP and Bhp. (Air density at 250 degrees F = 0.0563 lb/ft$^3$).

Using Equations 5-47 and 5-48:

$$SP_2 = SP_1 \times \frac{d_2}{d_1} = 2.6 \times \frac{0.075}{0.0653}$$

$$SP_2 = 3.46 \text{ in. wg}$$

$$FP_2 = FP_1 \times \frac{d_2}{d_1} = 9.22 \times \frac{0.075}{0.0563}$$

$$FP_2 = 12.28 \text{ Bhp}$$

With only a 10 HP motor, a 23 percent motor overload occurs.

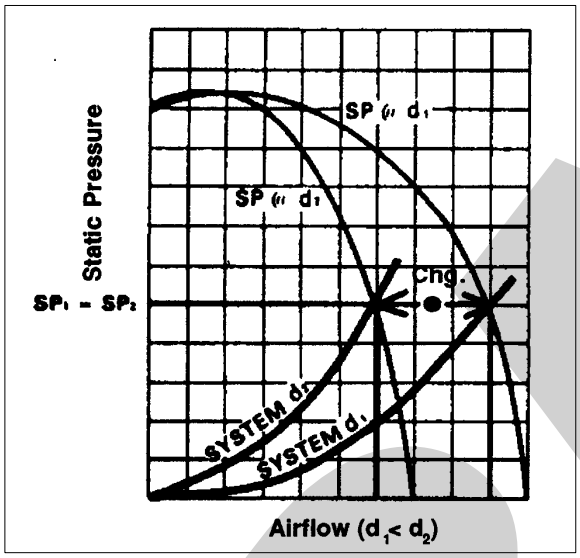


**FIGURE 5–48 EFFECT OF DENSITY CHANGE (CONSTANT STATIC PRESSURE)**

*Example 5-30 (SI)*

A 7500 L/s fan is delivering 125 degree C air from an oven through an air-to-air heat exchanger against 650 Pa SP. The fan is drawing 6.88 kW from a 7.5 kW motor. If the fan is run with the oven off (20 degree C ambient) calculate the new SP and kW. (Air density at 125 degrees C = 0.89 kg/m$^3$.)

Using Equations 5-27 and 5-28:

$$SP_2 = SP_1 \times \frac{d_2}{d_1} = 650 \times \frac{1.20415}{0.891}$$

$$SP_2 = 878 \text{ Pa}$$

$$FP_2 = FP_1 \times \frac{d_2}{d_1} = 6.88 \times \frac{1.2041}{0.891}$$

$$FP_2 = 9.308 \text{ kW}$$

With only a 7.5 kW motor, a 24 percent motor overload occurs.

### 5.33.2 Constant Static Pressure

If the system static pressure remains constant, the airflow volume, fan speed, and fan power will vary inversely as the square root of the density, *see* Figure 5-48.

**Equation 5–28**

$$\frac{Q_1}{Q_2} = \sqrt{\frac{d_2}{d_1}}$$

**Equation 5–29**

$$\frac{RPM_1}{RPM_2} = \sqrt{\frac{d_2}{d_1}}$$

**Equation 5–30**

$$\frac{FP_1}{FP_2} = \sqrt{\frac{d_2}{d_1}}$$

Where ($SP_1 = SP_2$):

$Q$ = Airflow – cfm (L/s)

$D$ = Density – lb/ft$^3$ (kg/m$^3$)

$RPM$ = Fan Speed

$FP$ = Fan power – Bhp (W)

### 5.33.3 Constant Mass Flow

With a constant mass flow rate in a system that remains constant without any changes and using the same fan with a variable drive, the airflow rate, RPM, and static pressure will vary inversely with the air density. The fan brake power will vary inversely with the square of the density, *see* Figure 5-49.

**Equation 5–31**

$$\frac{Q_1}{Q_2} = \frac{d_2}{d_1}$$

**Equation 5–32**

$$\frac{RPM_1}{RPM_2} = \frac{d_2}{d_1}$$

**Equation 5–33**

$$\frac{SP_1}{SP_2} = \frac{d_2}{d_1}$$

**Equation 5–34**

$$\frac{FP_1}{FP_2} = \frac{d_2}{d_1}$$

Where:

$Q$ = Airflow – cfm (/s)

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$D$ = Density – lb/ft$^3$ (kg/m)$^3$

$RPM$ = Fan speed

$SP$ = Static pressure – in. wg (Pa)

$FP$ = Fan power – Bhp (W)

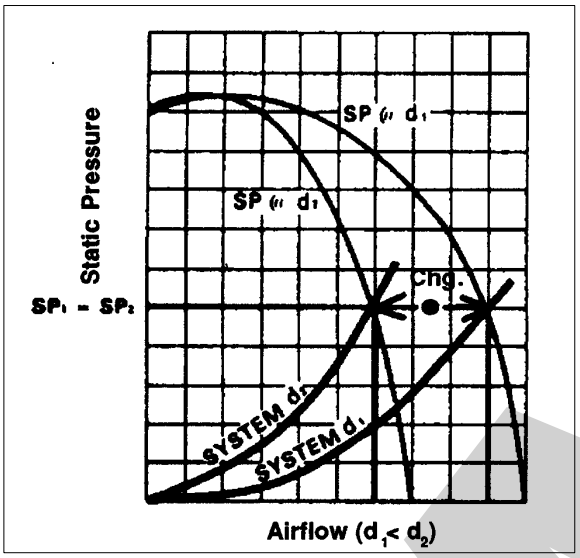


**FIGURE 5–49 EFFECT OF DENSITY CHANGE (CONSTANT MASS FLOW)**

*Example 5-31 (I-P)*

A fan is required to handle 15,000 cfm at 2 in. wg, 150 degree F, 950 RPM, 2000 ft altitude and 5.6 Bhp. Find the fan cfm, SP, Bhp, and RPM that must be selected from the fan table.

Using Table A-26 the correction factor to Standard conditions is 0.81 or:

$$\frac{d_2(Act.)}{d_1(Std.)} = 0.81$$

Using Equations 5-32 to 5-35:

$$Q_1\,(Std.) = Q_2\,(Act.)\,\frac{d_2}{d_1}$$

$$Q_1 = 15,000 \times 0.81 = 12,150 \text{ cfm}$$

$$SP_1\,(Std.) = 2.0 \times 0.81 = 1.62 \text{ in. wg}$$

$$RPM_2\,(Act.) = \frac{RPM_1\,(Std.)}{d_2/d_1}$$

$$RPM_2 = \frac{950}{0.81} = 1173 \text{ rpm}$$

$$FP_2\,(Act.) = \frac{FP_1\,(Std.)}{(d_2/d_1)^2}$$

$$FP_2 = \frac{5.6}{(0.81)^2} = 8.54 \text{ Bhp}$$

*Example 5-31 (SI)*

A fan is required to handle 7500 L/s at 500 Pa, 75 degree C, 950 rpm, 750 meters altitude and 4.2 kW. Find the fan airflow, SP, RPM, and fan brake power that must be selected from the fan table.

Using Table A-32 the correction factor to standard conditions is 0.78 or:

$$\frac{d_2(Act.)}{d_1(Std.)} = 0.78$$

Using Equations 5-32 to 5-35:

$$Q_1\,(Std.) = Q_2\,(Act.)\,\frac{d_2}{d_1}$$

$$Q_1 = 7,500 \times 0.78 = 5,850 \text{ L/s}$$

$$SP_1\,(Std.) = 500 \times 0.78 = 390 \text{ Pa}$$

$$RPM_2\,(Act.) = \frac{RPM_1\,(Std.)}{d_2/d_1}$$

$$RPM_2 = \frac{950}{0.78} = 1218 \text{ rpm}$$

$$FP_2\,(Act.) = \frac{FP_1\,(Std.)}{(d_2/d_1)^2}$$

$$FP_2 = \frac{4.2}{(0.78)^2} = 6.90 \text{ kW}$$

## 5.34 ESTIMATING SYSTEM RESISTANCE

Figure 5-50, Curve B, shows a situation where an actual duct system has more resistance to flow than was expected. The fan is operating as tested. This condition is generally the result of an inaccurate estimate of system resistance to flow. All losses must be considered when calculating system pressure losses or the final system will be more restrictive than designed and actual flow rate will be deficient, Point 2.

If the actual system pressure loss is greater than design, as in System B, Figure 5-50, an increase in fan speed may be necessary to achieve the design volume flow rate at Point 5. Before attempting to increase the speed of any fan, check with the fan manufacturer to determine if the speed can be safely increased and also determine the expected increase in fan power. The connected motor may not carry the increased fan brake power.

## 5.35 SAFETY FACTORS

System designers sometimes add "Safety Factors" to their estimate of the system resistance to compensate

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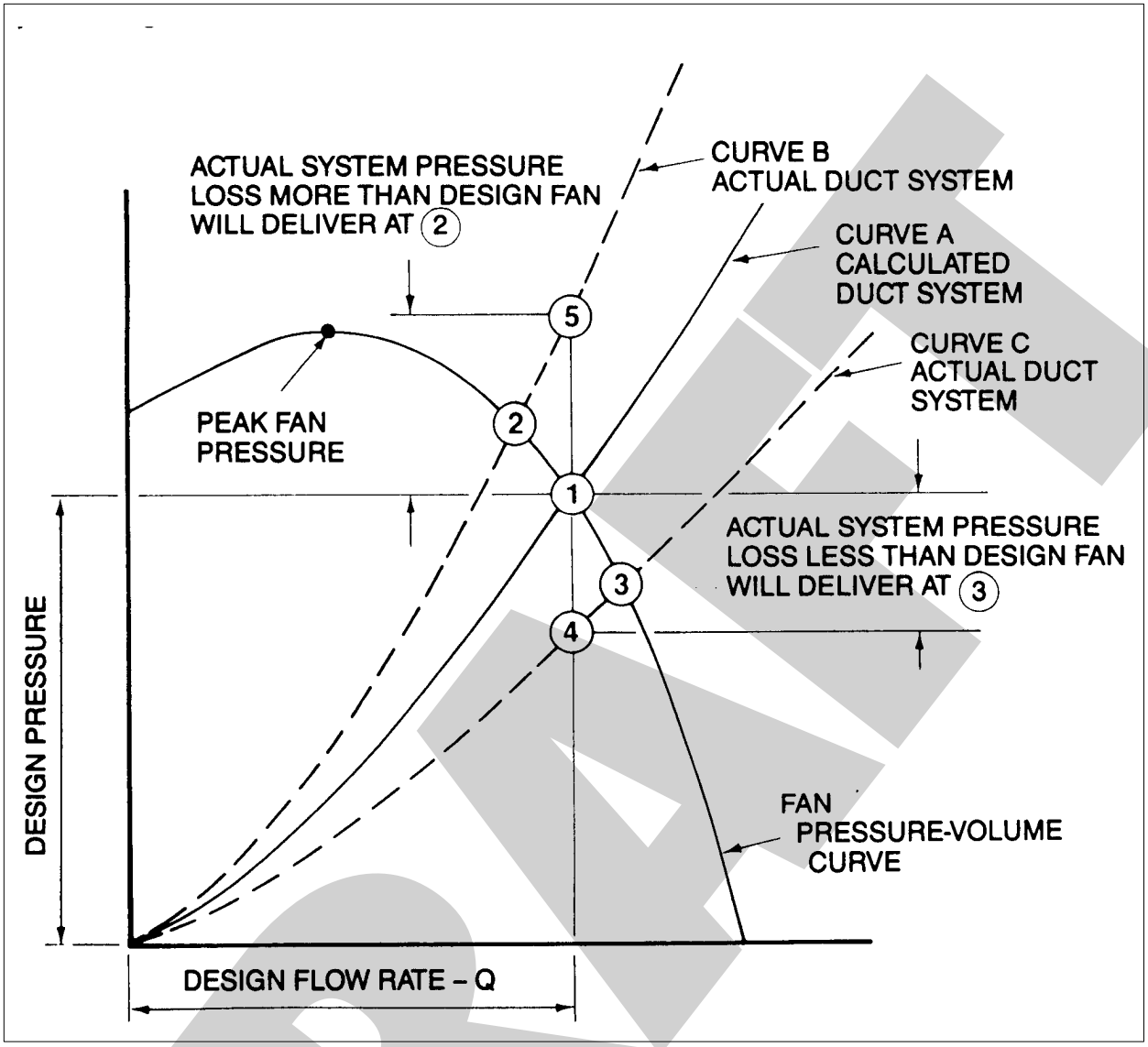


**FIGURE 5–50 DUCT SYSTEM CURVE NOT AT DESIGN POINT**

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for unknown field conditions. These "Safety Factors" may compensate for resistance losses that were overlooked and the actual system will deliver the design airflow (Point 1, Figure 5-50). Occasionally, however, the estimated system resistance, including the "Safety Factors", is well in excess of the actual installed system conditions. Since the fan has been selected for design conditions (Point 1), it will deliver more air (Point 3), because the actual system resistance at the design airflow rate is less than expected (Point 4). This result may not necessarily be an advantage because the fan usually will be operating at a less efficient point on the performance curve and may require more fan power than at the design airflow. Under these conditions, it may be necessary to reduce the fan speed. Often when a system is balanced, system volume dampers will increase the actual system resistance (Curve C) to the original design system resistance. This is not usually the most energy efficient way to solve the problem.

## 5.36 THE FAN OUTLET

### 5.36.1 Fan Testing

Most fan manufacturers rate the performance of their products from tests made in accordance with AMCA Standard 210, *Laboratory Methods of Testing Fans for Rating*. The purpose of Standard 210 is to establish uniform methods for laboratory testing of fans and other air moving devices to determine performance in terms of flow rate, pressure, power, air density, speed

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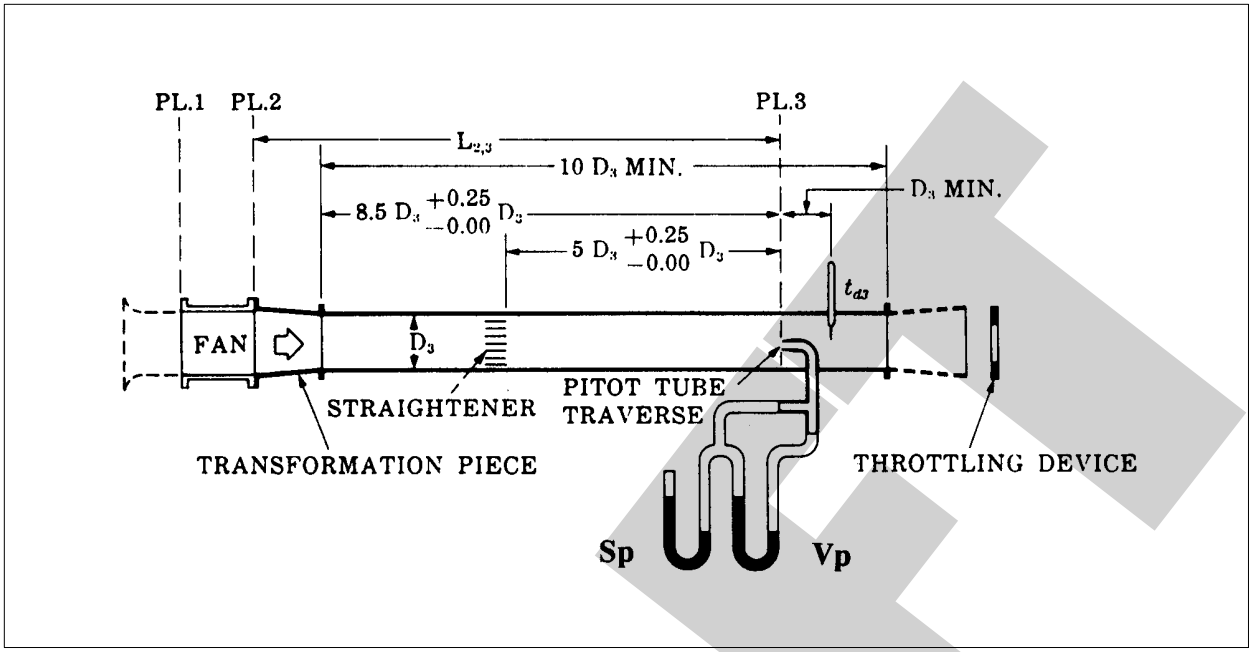


**FIGURE 5–51 AMCA FAN TEST – PITOT TUBE**

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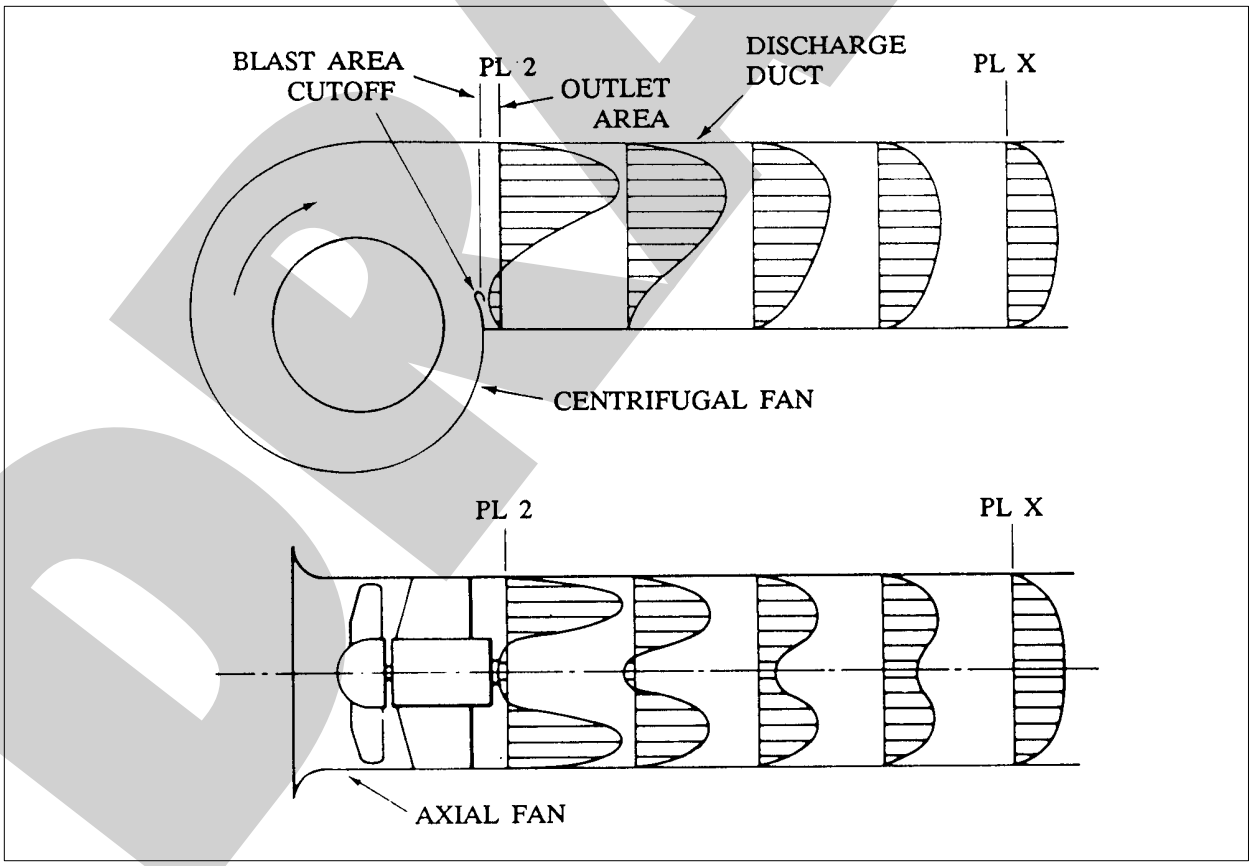


**FIGURE 5–52 ESTABLISHMENT OF A UNIFORM VELOCITY PROFILE**

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of rotation, and efficiency, for rating or guarantee purposes.

Two basic methods of measuring airflow are the Pitot tube and the long radius flow nozzle. These are incorporated into a number of different "setups" or "figures". In general, a fan is tested on the setup that most closely simulates the way in which it will be installed in an HVAC system. Centrifugal, tubeaxial, and vaneaxial fans are usually tested with only an outlet duct.

Figure 5-51 is a reproduction of a test setup from AMCA Standard 210. Note that this particular setup includes a long straight duct connected to the outlet of the fan. A straightener is located upstream of the Pitot tube traverse to remove swirl and rotational components from the airflow and to ensure that the flow at the plane of measurement is as uniform as possible.

The angle of the transition between the test duct and the fan outlet is limited to ensure that uniform flow will be maintained. A steep transition or abrupt change of cross-section would cause turbulence and eddies, and lead to non-uniform flow. The effect of this type of flow disturbance at the fan outlet is discussed later.

Uniform flow conditions ensure consistency and reproducibility of test results and permits the fan to develop its maximum performance. In any installation where uniform flow conditions do not exist, the fan's performance will be reduced.

The velocity profile at the outlet of a fan is not uniform. This is illustrated in Figure 5-52 Plane 2. The section of straight duct attached to the fan outlet controls the diffusion of the non-uniform outlet flow and establishes a more uniform velocity profile before discharging into a plenum chamber or to the atmosphere as shown in Figure 5-52 Plane X.

The energy loss when a gas passes through a sudden enlargement is related to the square of the velocity. Thus, the duct outlet with its more uniform velocity significantly reduces the loss at the point of discharge to atmosphere.

A manufacturer may test a fan with or without an outlet or inlet duct. Catalog ratings should state whether ducts were used during the rating tests. If the fans do not have similar duct configurations as used in the test setup, an allowance should be made for the difference in the resulting performance.

### 5.36.2 Fan Outlet Velocity

Centrifugal fans with cutoff plates, *see* Figure 5-52, generate a higher velocity through the smaller "blast area" than that found in the fan outlet duct. This difference in areas is known as the "blast area ratio" which is the blast-area is divided by the fan outlet duct area. The higher blast area velocity pressure is partially converted to static pressure when fan-outlet-sized ducts are used because the outlet duct velocity is lower. The fan airflow volume is determined by the outlet duct velocity that is measured by the Pitot tube traverse (Figure 5-51). The static pressure measured by the Pitot tube includes this static regain, less the duct friction loss from PL 2 to PL X and is part of the static pressure rating of the fan.

When centrifugal fans with cutoff plates are discharged directly into the atmosphere or into plenums, the velocity pressure energy conversion to static regain is lost. In some fan applications, this velocity pressure loss and the System Effect may exceed the fan outlet velocity pressure listed in the fan catalog, as defined in terms of the fan outlet area.

### 5.36.3 Fan Outlet System Effects

For 100 percent energy recovery, the duct and an acceptable transition usually must be as long as the full "effective duct length" before a duct fitting is used.

A "System Effect Factor" is a pressure loss to the fan capacity caused by fan inlet or outlet restrictions or other conditions influencing fan performance when the fan is installed in an HVAC system.

Any changes in the discharge duct within the "effective duct length", which differ from the duct configuration used when the fan was tested and rated, may cause the fan to perform less efficiently.

Fan outlet velocities for HVAC systems usually are in a range from 900 to 3000 fpm (4.5 to 15 m/s), but fans often are selected with velocities near 2000 fpm (10 m/s). Blast area/outlet area ratios usually are in a range of 0.7 to 0.8, use 0.7 if data is not available.

*Example 5-10*

A toilet exhaust utility centrifugal fan, installed on a roof without a discharge duct, is rated at 4000 cfm (2000 L/s) with 0.5 in. wg (125 Pa) total static pressure and an outlet velocity of 2,000 fpm (10 m/s). If system effect is involved, calculate the fan capacity loss.

Using Table 6-1 in Chapter 6, no discharge duct and a blast area of 0.7 indicates an "S" System Effect Curve.

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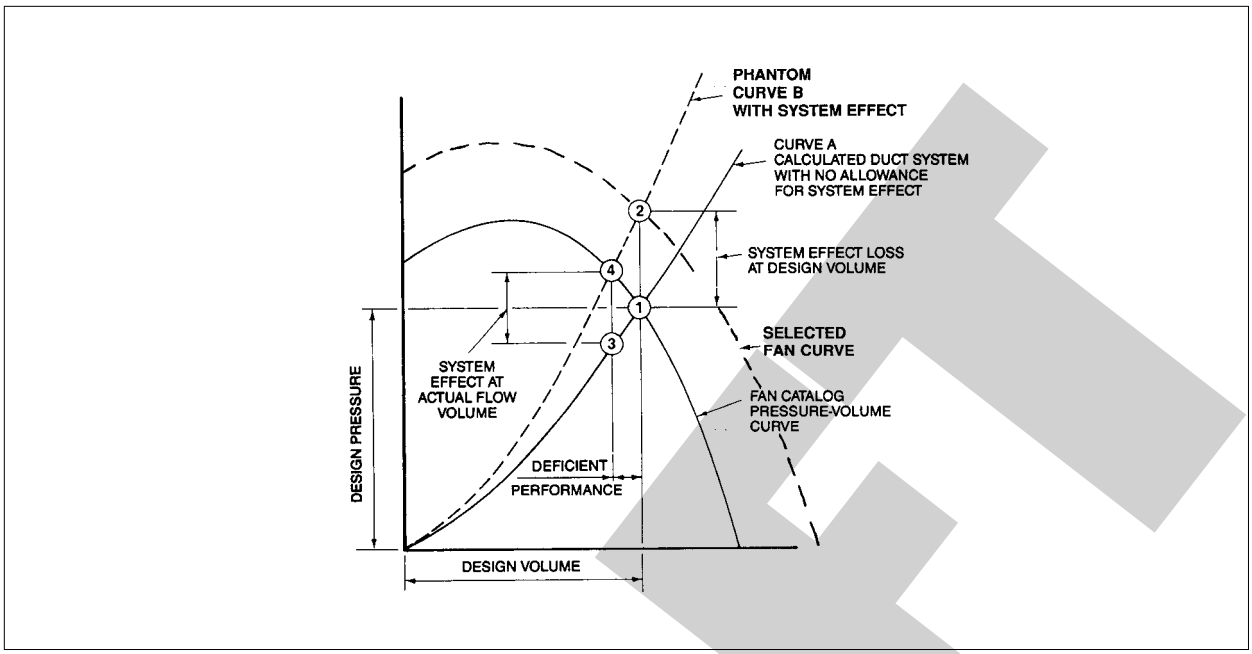


**FIGURE 5–53 EFFECTS OF SYSTEM EFFECT**

Figure 6-1 in the text shows a System Effect loss of 0.2 in. wg (50 Pa) at 2000 fpm (10 m/s) with an "S" curve. The fan capacity is reduced by 0.2 in. wg (50 Pa) to 0.3 in. wg (75 Pa). The fan inlet will be discussed later in this lesson.

Figure 5-53 illustrates deficient fan/system performance caused by System Effect. The HVAC system pressure losses have been determined accurately and the fan selected to operate at Point 1. However, no allowance has been made for the effect of the system connections on the fan performance. To compensate, a System Effect Factor must be added to the calculated system pressure losses to determine a system curve that is used to select the fan.

The point of intersection between the fan performance curve and the actual system curve is Point 4. The actual flow volume is deficient by the difference from 1 to 4. To achieve the design airflow volume, a System Effect Factor equal to the pressure difference between Point 1 and 2, where the new corrected fan RPM curve crosses "phantom system curve B," must be used. A higher brake horsepower is also required.

However, when the testing and balancing technician measures the actual HVAC system operating conditions, the airflow volume and static pressure will be established as Point 1, because that is where the system is actually operating. The system is not operating on the "phantom system curve", which was used only to select the de-rated capacity fan. System Effect generally *cannot* be measured in the field, only calculated after a visual inspection of the fan/duct system connections.

Because System Effect Factors are velocity related, the difference represented between Points 1 and 2 is greater than the difference between Points 3 and 4. The System Effect Factor includes only the effect of the system configuration on the fan's performance. All duct fitting pressure losses are calculated as part of the HVAC system pressure losses.

### 5.36.4 Fan Outlet Conditions

When elbows must be installed near a fan discharge, they are often at the flexible connection, as shown by Figure 6-3. If they are installed at a distance less than 100 percent of the effective duct length, the installation requires a calculation for System Effect. Table 6-2 is used for the calculations.

Fan outlets are sometimes connected directly to a larger duct or a large plenum, without an intermediate transition. This connection causes a pressure loss of up to one velocity head using the highest velocity in the fan outlet. This highest velocity is in the blast area and in the swept area of a vaneaxial fan. If the average outlet velocity is used based on the outlet flange area, the pressure loss may exceed one velocity head. A duct the same size as the fan outlet between the fan outlet and plenum, and one duct diameter or more in length, will reduce this loss considerably.

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*NOTE: Velocity head can be expressed as $h = V^2/2g$ in which V represents velocity and g the acceleration due to gravity (32.2 ft/sec$^2$)*

A diffuser with the same inlet size as the fan outlet may be used to connect the fan to the plenum. In this case, the losses must be calculated according to the fan manufacturer's recommendations, since they depend substantially on fan design. An arrangement in which a fan discharges through a coil having a face area considerably greater than the fan outlet area, presents an especially difficult problem. To improve the velocity distribution across the coil face, a perforated baffle plate is sometimes located between the fan outlet and coil. This type of baffle causes an increase in system resistance that is almost impossible to predict.

*Example 5-11*

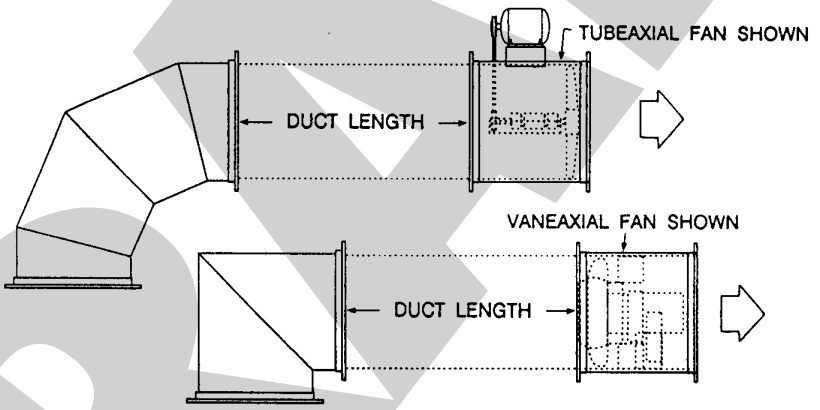
An air handling unit has a single width fan discharging horizontally at 2500 fpm (12.5 m/s). An elbow (position C in Figure 6-3) is attached to the fan outlet with a flexible connection. Find the amount of System Effect.

From Table 6-2 and using a blast area ratio of 0.7, under "No Outlet Duct", a "Q-R" System Effect Factor Curve is to be used. From Figure 6-1, a "Q-R" curve at 2500 fpm (12.5 m/s) gives about 0.54 in. wg (135 Pa) static pressure de-rating of the fan.

## 5.37 THE FAN INLET

### 5.37.1 Inlet Ducts

Power roof exhausters are tested and rated when mounted on a roof curb through which the exhaust air duct passes, so System Effect is not a problem. The problem with HVAC centrifugal and axial flow fans is they are tested without any inlet obstructions or duct connections, *see* Figure 5-52 or Figure 6-2. For rated performance, the air must enter the fan uniformly over the inlet area in an axial direction without pre-rotation. Non-uniform flow into the inlet is the most common cause of reduced fan performance.




| | H/T | 90° Elbow | No Duct [1][2] | 0.5 D [1][2] | 1.0 D [1][2] | 3.0 D |
|---|---|---|---|---|---|---|
| Tubeaxial Fan | .25 | 2 piece | U | V | W | --- |
| Tubeaxial Fan | .25 | 4 piece | X | --- | --- | --- |
| Tubeaxial Fan | .35 | 2 piece | V | W | X | --- |
| Vaneaxial Fan | .61 | 2 piece | Q-R | Q-R | S-T | T-U |
| Vaneaxial Fan | .61 | 4 piece | W | W-X | --- | --- |

[1] Instability in fan operation may occur as evidenced by an increase in pressure fluctuations and sound level. Fan instability, for any reason, may result in serious structural damage to the fan.

[2] The data presented in Figure 7–4 is representative of commercial type tubeaxial and vaneaxia fans, i.e. 60% to 70% fan static efficiency.

**FIGURE 5–54 SYSTEM EFFECT CURVES FOR INLET DUCT ELBOWS – AXIAL FANS**

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Such inlet conditions are not equivalent to a simple increase in the system resistance; therefore, they cannot be treated as a percentage decrease in the flow and pressure from the fan. A poor inlet condition results in an entirely new fan performance. Many poor inlet conditions affect the fan more at near-free delivery conditions than at peak pressure, so there is a continually varying difference between these two points.

For example, an elbow at the fan inlet causes turbulence, uneven flow into the fan impeller. Figures 6-9, 6-11, and 6-12 diagram several elbow arrangements and the corresponding System Effect Curves for calculating the System Effect Factors. The System Effect Factor can be eliminated by including an adequate length of straight duct between the elbow and the fan inlet.

### 5.37.2 Inlet Duct Elbows (Axial Fans)

A word of caution is required with the use of inlet elbows close to fan inlets. Other than the incurred System Effect Factor, instability in fan operation may occur as evidenced by an increase in pressure fluctuations and sound power levels. Fan instability, for any reason, can result in serious structural damage to the fan. Axial fan instabilities were experienced in some configurations tested with inlet elbows close to the fan inlet. Pressure fluctuations approached ten times the magnitude of fluctuations of the same fan with good inlet and outlet designs. Sound power levels increased in excess of ten decibels in some octave bands under the same operating conditions. It is strongly advised that inlet elbows be installed a minimum of three diameters away from any axial or centrifugal fan inlet.

The System Effect Curves shown in Figure 5-52 for tubeaxial and vaneaxial fans are the result of tests run with two- and four-piece mitered inlet elbows at or close to the fan inlets. Other variables tested included hub to tip (H/T) ratio and blade rigidity. The number of blades did not have a significant affect on the inlet elbow System Effect Factor.

### 5.37.3 Inlet Duct Elbows Centrifugal Fans

Non-uniform flow into a fan inlet (Figure 6-9) is the most common cause of deficient fan performance. The System Effect Curves for mitered 90 degree round elbows of given radius/diameter ratios are listed in Figure 6-11, and the system effect curves for various square duct elbows of given radius/diameter ratios are listed on Figure 6-12. The System Effect Factor for a particular elbow is found in Figure 6-1 at the intersection of the average fan inlet velocity and the tabulated System Effect Curve.

This pressure loss is in addition to the friction and dynamic losses already determined for the particular elbow. Note that when duct splitter vanes or a suitable length of duct are used (three to eight diameters, depending on velocities) between the fan inlet and the elbow, the System Effect Factor is not as large. These improvements help maintain uniform flow into the fan inlet and thereby approach the flow conditions of the laboratory test setup. Also, the inlet velocity for a centrifugal fan is normally about 0.75 to 0.85 of the discharge velocity.

A very common connection where space is at a premium is shown in Figure 6-10. The reduction in capacity and pressure for this type of inlet condition is impossible to calculate. The many possible variations in width and depth of the duct influence the reduction in performance to varying degrees; therefore, this inlet should be avoided. Capacity losses as high as 45 percent have been observed in this poorly designed inlet.

### 5.37.4 Inlet Boxes

Inlet boxes are used with centrifugal and axial fans instead of elbows in order to provide predictable inlet conditions and maintain stable fan performance. The fan manufacturer should include the effect of any recommended inlet box on the fan performance and, when evaluating a proposal, it should be established that an appropriate loss has been incorporated in the fan rating. Should this information not be available from the manufacturer, refer to Figure 6-20 for information on a field-fabricated fan inlet box.

*Example 5-12*

Using the toilet exhaust utility centrifugal fan of Example 7-A: 4000 cfm (2000 L/s) at 0.5 in. wg (125 Pa) and an outlet velocity of 2000 fpm (10 m/s), calculate the total losses from System Effect if the exhaust air duct is connected as shown in Figure 6-10.

| | in. wg (Pa) |
|---|---|
| Discharge loss (from Example 5-10) | 0.20 ( 50) |
| Inlet: 45% × 0.50 (125) | 0.23 ( 56) |
| Total losses (System Effect) | 0.43 (106) |
| Effective fan capacity remaining | 0.07 (19) |

This example illustrates why many toilet exhaust vent sets installed on the roof are inadequate.

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## 5.38 ASHRAE METHODS

*ASHRAE Handbook - HVAC Systems and Equipment*, provides additional information on the use of System Effect Factors.

**Equation 5–35**

$$SEF = K \times V_p$$

Where:

$SEF$ = System Effect Factor – in. wg (Pa)

$K$ = Value from ASHRAE (Tables 5-4 and 5-5)

$V_p$ = Fan Velocity Pressure – in. wg (Pa)

The velocity pressure (VP) is based on the following velocities (V):

Centrifugal Fans:

$V$ = fan inlet based on area at the inlet collar fpm (m/s)

$V$ = fan outlet velocity base on outlet area – fpm (m/s)

Axial Fans:

$V$ = fan inlet or outlet velocity based on area calculated from fan diameter, D – fpm (m/s)

If more than one configuration is included in a system, the System Effect Factor for each must be determined separately and the total of these System Effects must be added to the total system pressure losses in order to select the fan.

However, the field-measurable total pressure losses of the HVAC system will be without the System Effect losses added.

Velocity pressures may be obtained from Tables A-5 and A-5M.

*Example 5-32 (I-P)*

A 12,600 cfm fan discharges into a 24 × 36 in. 90 degree elbow (position C) located 21 in. from the fan. Determine the System Effect pressure loss.

For Double-Width, Double-Inlet (DWDI) fans, determine the K value from the above table or SWSI fans and calculate the System Effect Factor (SEF) using Equation 5-56, then apply the multiplier from the table below:

Multipliers for DWDI Fans
Elbow position A = 1.00 SEF
Elbow position B = 1.25 SEF
Elbow position C = 1.00 SEF
Elbow position D = 0.85 SEF

Using Equation 5-2,

$$V = \frac{Q}{A} = \frac{12,600}{24 \times 36/144} = 2100 \text{ fpm}$$

Using Equation 5-6,

$$V_p = \left(\frac{2100}{4005}\right)^2 = 0.275 \text{ in.wg}$$

From Figure 6-2,

$$\textit{Equiv Duct Diam.} = \sqrt{\frac{4\,ab}{\pi}} = 33.2 \text{ in.}$$

$$\textit{Eff. Duct Length} = 2.5 \times 33.2 = 83 \text{ in.}$$

21 in./83 in. = 25.3 percent Eff. Duct

From Table 5-5 (blast ratio = 0.7):

| | No Duct | 12% Effective Duct | 25% Effective Duct | 50% Effective Duct | 100% Effective Duct |
|---|---|---|---|---|---|
| Pressure Recovery | 0% | 50% | 80% | 90% | 100% |
| Blast Area / Outlet Area | K Values | | | | |
| 0.4 | 2.0 | 1.00 | 0.40 | 0.18 | No System Effect Factor |
| 0.5 | 2.0 | 1.00 | 0.40 | 0.18 | |
| 0.6 | 1.00 | 0.66 | 0.33 | 0.14 | |
| 0.7 | 0.80 | 0.40 | 0.14 | — | |
| 0.8 | 0.47 | 0.22 | 0.10 | — | |
| 0.9 | 0.22 | 0.14 | — | — | |
| 1.0 | — | — | — | — | |

**Table 5–4 K Values for Outlet Ducts**

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| Blast Area Outlet Area | Outlet Elbow Position | No Outlet Duct | 12% Effective Duct | 25% Effective Duct | 50% Effective Duct | 100% Effective Duct |
|---|---|---|---|---|---|---|
| 0.4 | A | 3.20 | 2.68 | 1.76 | 0.84 | |
| | B | 4.00 | 3.32 | 2.24 | 1.04 | |
| | C | 5.80 | 4.84 | 3.20 | 1.52 | |
| | D | 5.80 | 4.84 | 3.20 | 1.52 | |
| 0.5 | A | 2.28 | 1.88 | 1.28 | 0.60 | |
| | B | 2.84 | 2.36 | 1.56 | 0.72 | |
| | C | 4.00 | 3.32 | 2.24 | 1.04 | |
| | D | 4.00 | 3.32 | 2.24 | 1.04 | |
| 0.6 | A | 1.60 | 1.32 | 0.88 | 0.40 | |
| | B | 2.00 | 1.68 | 1.12 | 0.52 | |
| | C | 2.92 | 2.44 | 1.64 | 0.76 | |
| | D | 2.92 | 2.44 | 1.64 | 0.76 | |
| 0.7 | A | 1.08 | 0.88 | 0.60 | 0.28 | |
| | B | 1.32 | 1.08 | 0.72 | 0.36 | |
| | C | 1.96 | 1.64 | 1.08 | 0.52 | |
| | D | 1.96 | 1.64 | 1.08 | 0.52 | |
| 0.8 | A | 0.76 | 0.64 | 0.44 | 0.20 | |
| | B | 0.96 | 0.80 | 0.52 | 0.24 | |
| | C | 1.40 | 1.16 | 0.76 | 0.36 | |
| | D | 1.40 | 1.16 | 0.76 | 0.36 | |
| 0.9 | A | 0.60 | 0.48 | 0.32 | 0.16 | |
| | B | 0.76 | 0.64 | 0.44 | 0.20 | |
| | C | 1.12 | 0.92 | 0.64 | 0.28 | |
| | D | 1.12 | 0.92 | 0.64 | 0.28 | |
| 1.0 | A | 0.56 | 0.48 | 0.32 | 0.16 | |
| | B | 0.68 | 0.56 | 0.36 | 0.16 | |
| | C | 1.00 | 0.84 | 0.56 | 0.28 | |
| | D | 1.00 | 0.84 | 0.56 | 0.28 | |

No System Effect Factor

**Table 5–5 K Values for Single Width, Single Inlet Fans (SWSI)**

$K = 1.08$

Using Equation 5-56,

$$SEF = K \times V_P = 1.08 \times 0.275$$

$SEF = 0.297$ in. wg

*Example 5-32 (SI)*

A 6300 L/s fan discharges into a 600 × 900 mm 90 degree elbow (position C) located 525 mm from the fan. Determine the System Effect pressure loss.

Using Equation 5-2,

$$V = \frac{L/s}{1000 \times m^2} = \frac{6300}{1000 \times 0.6 \times 0.9}$$

$V = 11.67$ m/s

Using Equation 5-6,

$$V_P = 0.602\,V^2 = 0.602(11.67)^2 = 82.0 \text{ Pa}$$

From Figure 6-2,

$$\textit{Equiv Duct Diam.} = \sqrt{\frac{4\,ab}{\pi}} = 829.2 \text{ mm.}$$

*Eff. Duct Length* = 2.5 × 829.2 = 2073 mm

525 mm/2,073 mm = 25.3 percent Eff. Duct

From Table 5-5 (blast ratio = 0.7):

$K = 1.08$

Using Equation 5-36,

$$SEF = K \times V_p = 1.08 \times 82$$

$SEF = 88.6$ Pa

## 5.39 FUNDAMENTALS HANDBOOK

*ASHRAE Handbook - Fundamentals* provides additional information on the use of System Effect Factors. Equation 5-36 is used in this chapter which is the same as Equation 5-36 in different terms because $V_P = d\,(V/1097)^2$, ($V_P = d\,V^2/2$).

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**Equation 5–36 (I–P)**

$$FSE = C_0 d \left(\frac{V}{1097}\right)^2$$

**Equation 5–36 (SI)**

$$FSE = \frac{C_0 d V_0^2}{2}$$

Where:

$FSE$ = Fan System Effect Pressure Loss in. wg (Pa)

$C_0$ = Fan System Effect Loss Coefficient

$d$ = Density - lb/ft$^3$ (kg/m$^3$)

$V_0$ = Fan Outlet Velocity - fpm (m/s)

ASHRAE text states, "To compensate, a fan system effect must be added to the calculated system pressure losses to determine the actual system curve." This is a clear, concise statement. However, by combining System Effect Factor tables with duct fitting loss coefficient tables and using the same designation ($C_0$), the result adds to the confusion already present in the HVAC industry about System Effect. Inexperienced designers are using these System Effect "$C_0$" values, in lieu of the actual fitting loss coefficients when calculating system pressure losses.

All System Effect pressure losses are in addition to the calculated or measured HVAC system pressure losses.

## 5.40 DEFICIENT FAN PERFORMANCE

### 5.40.1 System Effect

It is unfortunate that many HVAC system designers, fan company sales engineers, and HVAC contractors have not applied System Effect to fan capacity problems. As System Effect values are only approximate, and because all fans react a little differently to the system to which they are connected, a judgment must be applied. This judgment can be verified by the test reports from the testing, adjusting, and balancing (TAB) contractor. Then future judgmental decisions can build on these experiences.

Many times, System Effect is not anticipated by the designer and is only detected after the HVAC system is installed. "One-line" drawings of HVAC system cannot provide a clear understanding of how the duct system must be installed and connected to the fans. Also, when the HVAC system designer is reviewing shop drawings for duct fittings attached to or near the fan, the designer is responsible for analyzing the potential interactions of the fittings and fans for the consequences of undesirable System Effects. Reviews at the design stage will identify needed changes to the fan size, fan drive, or fan motor electrical service when changes are least costly to the owner.

## 5.41 SYSTEM EFFECT FACTORS

The system designer must take System Effect into account if a realistic estimate of fan/system performance is to be made. It must be appreciated that the System Effect Factors given in Chapter 6 of this manual are intended as guidelines and are, in general, approximations. Some have been obtained from research studies, others have been published previously by individual fan manufacturers, and many represent the consensus of engineers with considerable experience in the application of fans.

Fans of different types, and even fans of the same type but supplied by different manufacturers, will not necessarily react with the system in exactly the same way. Therefore, it will be necessary to apply judgment based on actual experience in applying the System Effect Factors.

Figure 5-61 illustrates deficient fan/system performance resulting from undesirable flow conditions. It is assumed that the system pressure losses have been accurately determined (Point 1, Curve A) and a suitable fan selected for operation at that point. However, no allowance has been made for the *effect of the system connections on the fan's performance*. To compensate for this System Effect and to explain how it works, it will be necessary to add a System Effect Factor to the calculated system pressure losses to determine the actual system curve. The System Effect Factor for any given configuration is dependent on the airflow velocity at that point.

In the example illustrated on Figure 5-61, the point of intersection between the fan performance curve and the actual system curve is Point 4. The actual airflow volume will, therefore, be deficient by the difference from 1 to 4. To achieve the design airflow volume, a System Effect Factor equal to the pressure difference between Points 1 and 2 should have been added to the calculated system pressure losses and the fan selected to operate at Point 2. Note, that because the System Effect is velocity related, the difference represented between Points 1 and 2 is greater than the difference between points 3 and 4.

Chapter 6-1, "Fan – Duct Connection Pressure Losses," contains the necessary data, charts and tables to determine the System Effect Factors required by

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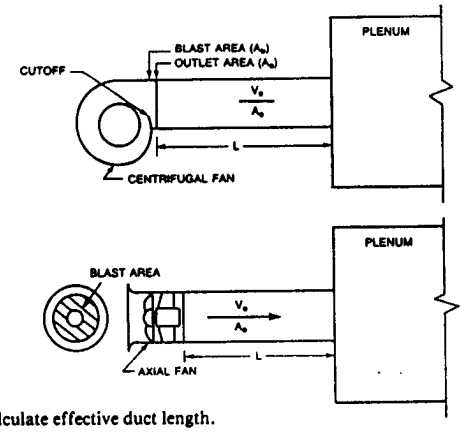
## FAN-SYSTEM CONNECTIONS

**7-1 Fans Discharging into a Plenum (ACMA 1973, Fig. 19)**




Calculate effective duct length.

$V_o > 2500$ fpm: $L_e = V\sqrt{A_o}/10{,}600$

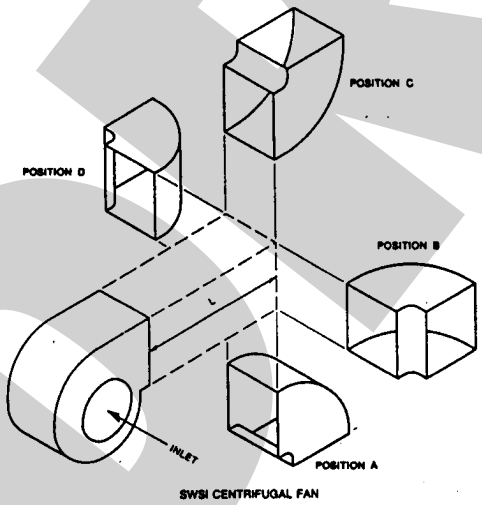
$V_o \leq 2500$ fpm: $L_e = \sqrt{A_o}/4.3$

*where*

$V_o$ = duct velocity, fpm
$L_e$ = effective duct length, ft
$A_o$ = duct area, in²

| | $C_o$ | | | | |
|---|---|---|---|---|---|
| | $L/L_e$ | | | | |
| $A_b/A_o$ | 0 | 0.12 | 0.25 | 0.5 | ≥ 1.0 |
| 0.4 | 2.0 | 1.0 | 0.40 | 0.18 | 0 |
| 0.5 | 2.0 | 1.0 | 0.40 | 0.18 | 0 |
| 0.6 | 1.0 | 0.66 | 0.33 | 0.14 | 0 |
| 0.7 | 0.8 | 0.40 | 0.14 | 0 | 0 |
| 0.8 | 0.47 | 0.22 | 0.10 | 0 | 0 |
| 0.9 | 0.22 | 0.14 | 0 | 0 | 0 |
| 1.0 | 0 | 0 | 0 | 0 | 0 |

**7-2 Single Width Single Inlet (SWSI) Fan with an Outlet Duct Elbow (AMCA 1973, Fig. 22)**

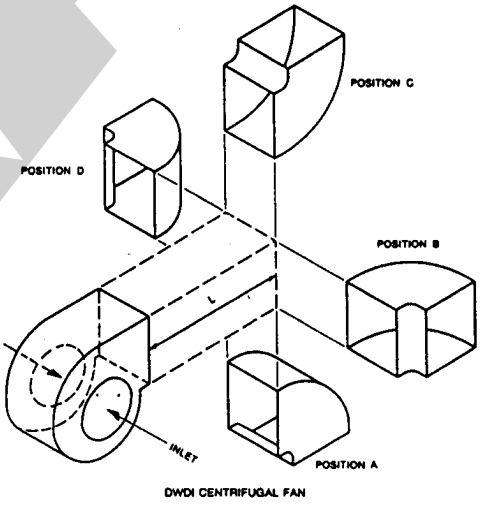



$A_b$ = centrifugal fan blast area (see Fitting 7-1)
$A_o$ = duct/outlet area (see Fitting 7-1)

To calculate effective duct length $L_e$, see Fitting 7-1.

| | | $C_o$ | | | | |
|---|---|---|---|---|---|---|
| | | $L/L_e$ | | | | |
| $A_b/A_o$ | Outlet Elbow Position | 0 | 0.12 | 0.25 | 0.5 | ≥ 1.0 |
| 0.4 | A | 3.2 | 2.7 | 1.8 | 0.84 | 0 |
| | B | 4.0 | 3.3 | 2.2 | 1.0 | 0 |
| | C | 5.8 | 4.8 | 3.2 | 1.5 | 0 |
| | D | 5.8 | 4.8 | 3.2 | 1.5 | 0 |
| 0.5 | A | 2.3 | 1.9 | 1.3 | 0.60 | 0 |
| | B | 2.8 | 2.4 | 1.6 | 0.72 | 0 |
| | C | 4.0 | 3.3 | 2.2 | 1.0 | 0 |
| | D | 4.0 | 3.3 | 2.2 | 1.0 | 0 |
| 0.6 | A | 1.6 | 1.3 | 0.88 | 0.40 | 0 |
| | B | 2.0 | 1.7 | 1.1 | 0.52 | 0 |
| | C | 2.9 | 2.4 | 1.6 | 0.76 | 0 |
| | D | 2.9 | 2.4 | 1.6 | 0.76 | 0 |
| 0.7 | A | 1.1 | 0.88 | 0.60 | 0.28 | 0 |
| | B | 1.3 | 1.1 | 0.72 | 0.36 | 0 |
| | C | 2.0 | 1.6 | 1.1 | 0.52 | 0 |
| | D | 2.0 | 1.6 | 1.1 | 0.52 | 0 |
| 0.8 | A | 0.76 | 0.64 | 0.44 | 0.20 | 0 |
| | B | 0.96 | 0.80 | 0.52 | 0.24 | 0 |
| | C | 1.4 | 1.2 | 0.76 | 0.36 | 0 |
| | D | 1.4 | 1.2 | 0.76 | 0.36 | 0 |
| 0.9 | A | 0.60 | 0.48 | 0.32 | 0.16 | 0 |
| | B | 0.76 | 0.64 | 0.44 | 0.20 | 0 |
| | C | 1.1 | 0.92 | 0.64 | 0.28 | 0 |
| | D | 1.1 | 0.92 | 0.64 | 0.28 | 0 |
| 1.0 | A | 0.56 | 0.48 | 0.32 | 0.16 | 0 |
| | B | 0.68 | 0.56 | 0.36 | 0.16 | 0 |
| | C | 1.0 | 0.84 | 0.56 | 0.26 | 0 |
| | D | 1.0 | 0.84 | 0.56 | 0.28 | 0 |

**7-3 Double Width Double Inlet (DWDI) Fan with an Outlet Duct Elbow (ACMA 1973, Fig. 22)**




$A_b$ = centrifugal fan blast area (see Fitting 7-1)
$A_o$ = duct/outlet area (see Fitting 7-1)

To calculate *effective* duct length $L_e$, see Fittings 7-1.

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**FIGURE 5–55 SAMPLE ASHRAE FAN SYSTEM EFFECT "LOSS COEFFICIENTS"**

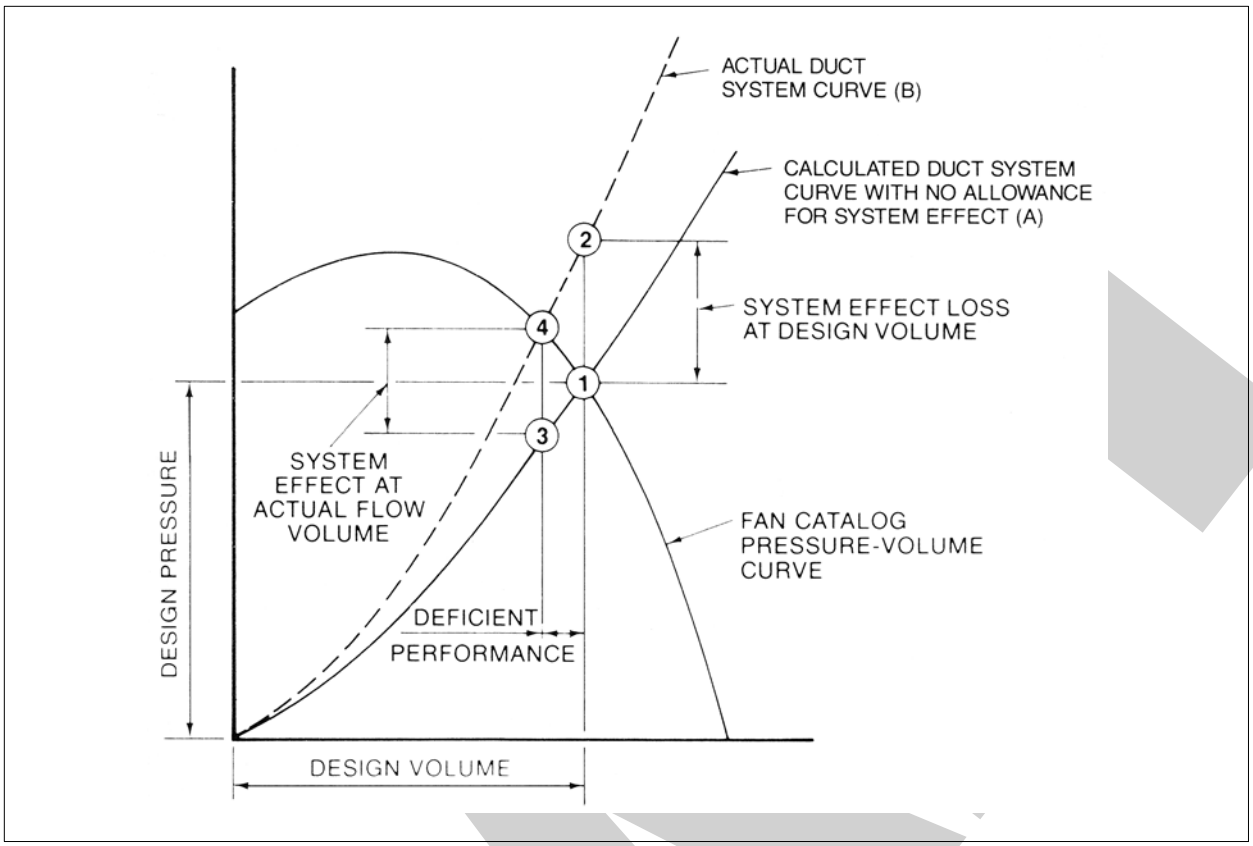


**FIGURE 5–56 CHANGES FROM SYSTEM EFFECT**

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duct connections to HVAC system fans. The System Effect Factor is given in in. wg (Pa) and may be added to the total system pressure losses as shown on Figure 5-35. However, System Effect cannot be measured in the field when the system is being tested and balanced. It can only be calculated using the data in Chapter 6. Therefore, the HVAC system designer must de-rate the HVAC system supply fan by deducting the System Effect Factor from the fan's rated capacity.

The velocity figure used in entering the chart will be either the inlet or the outlet velocity of the fan. This will be dependent on whether the configuration in question is related to the fan inlet or the outlet. Most catalog ratings include outlet velocity figures, for centrifugal fans it may be necessary to calculate the inlet velocity. The inlet requirements are usually included in the fan catalog.

### 5.41.1 Deficient System Performance

The System Effect pressure loss de-rates the performance of the fan but this pressure loss cannot be measured in the field. However, there are many other major causes of deficient system performance that can be field measured including:

a. Air performance characteristics of the installed system that are significantly different from the system designer's specifications and intent.

b. System design calculations that did not include adequate allowances for the effect of accessories and appurtenances; or the fan selection was made without allowing for the effect of appurtenances on the fan's performance.

c. Dirty filters, dirty terminal devices, and dirty coils that will increase the system resistance and consequently reduce the airflow rate, often significantly.

d. Duct system air leakage.

### 5.41.2 Objectives of System Effect Methods

The use of System Effect tables and charts in this SMACNA manual and AMCA's publication *Fans and*

*Systems* minimizes mathematical calculations. Regardless of the methods used, the answers should be similar.

It must be emphasized that System Effect pressure loss values are approximate and that TAB Contractors cannot isolate and measure them in the field.

ASHRAE research has determined that accurate duct velocity measurements cannot be made until a near uniform velocity profile has developed, *see* Figures 5-6 and 5-52. This point may vary from 3 to 20 duct diameters downstream from the object causing the turbulence. So any accurate field measurements on the discharge side of a fan would be well away from where System Effect occurs.

Non-uniform airflow, "spin" in the airflow at the fan inlet, or a duct condition that produces a vortex contributes to System Effect. Reviewing the various duct-fan inlet conditions in the Figures should confirm that accurate field measurements are impossible. However, on the fan inlet side, the System Effect capacity loss usually occurs within the entry to the fan wheel, which also is not field measurable.

Finally, System Effect is a real and an often-occurring problem. It may be avoided by using better fan to system duct connections. It may also be avoided by ordering HVAC fans and equipment with the proper inlet and discharge-connection configurations so elbows would not immediately change airflow direction near the fan outlet. Larger fans operating at slower speeds also help minimize System Effect. Being aware of System Effect may help to avoid or solve many HVAC fan and system problems.

## 5.42 BUILDING PRESSURES

The pressures within the conditional spaces of a building compared to atmospheric pressure outside the building are determined by many factors including the relationship of supply air volumes to exhaust air/outside air volumes, wind infiltration/exfiltration, building stack effect, and internal air paths.

### 5.42.1 HVAC Air Volumes

It is obvious that the pressure in a space will build up when more air is being supplied to the space than is being exhausted or being returned to the HVAC unit. This method of positive space pressure control is used effectively in buildings with clean rooms, where often there are different pressure gradients between the clean spaces and surrounding non-sterile spaces. HVAC systems using this type of space-to-space pressure control must have effective volume control, as well as careful sizing of the devices and openings between spaces.

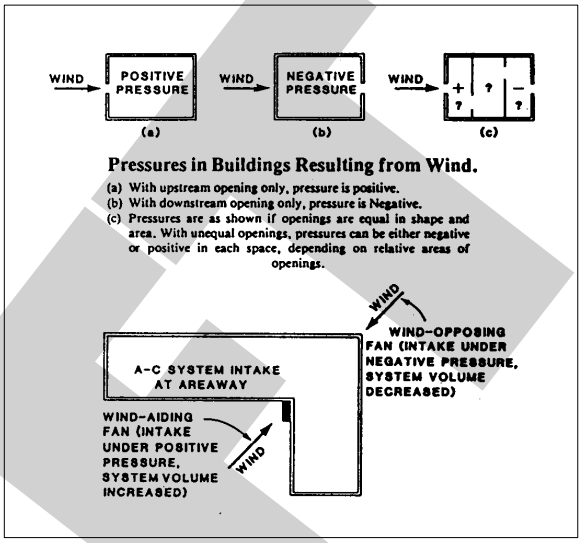


**FIGURE 5–57 SENSITIVITY OF SYSTEM VOLUME TO LOCATIONS OF BUILDING OPENINGS, INTAKES, AND EXHAUSTS**

*NOTE: Damper leakage classes are tested under UL 5555 with Class I being the lowest leakage class.*

### 5.42.2 Locating Exterior Air Intakes and Exhausts

Ventilating system components, including outdoor air intakes and exhaust air outlets, must be capable of performing adequately under all conditions. With the increased risk of biological and chemical terrorist activity, the location of large outside air intake and exhaust ducts is no longer arbitrary. In addition to being an entry point for anyone trying to purposely contaminate building ventilation air, these large exterior wall and roof duct penetrations can also serve as a means of vandal entry to areas storing expensive merchandise or valuable financial records. Using strong, vandal-resistant exterior louvers and low-leak age Class I dampers will reduce both of these security risks.

When security and air contamination is a design concern, outside air intakes should not be located near the ground level, or in the vicinity of exterior loading docks, vehicular exhausts, cooling towers, emergency generators, bathroom exhausts, or trash dumpsters.

In addition to these security concerns, outdoor intake and exhaust locations are also affected by the prevailing winds. A building with only upwind-side openings is under a positive pressure (Figure 5-62). Building pressures are negative when there are only downwind-

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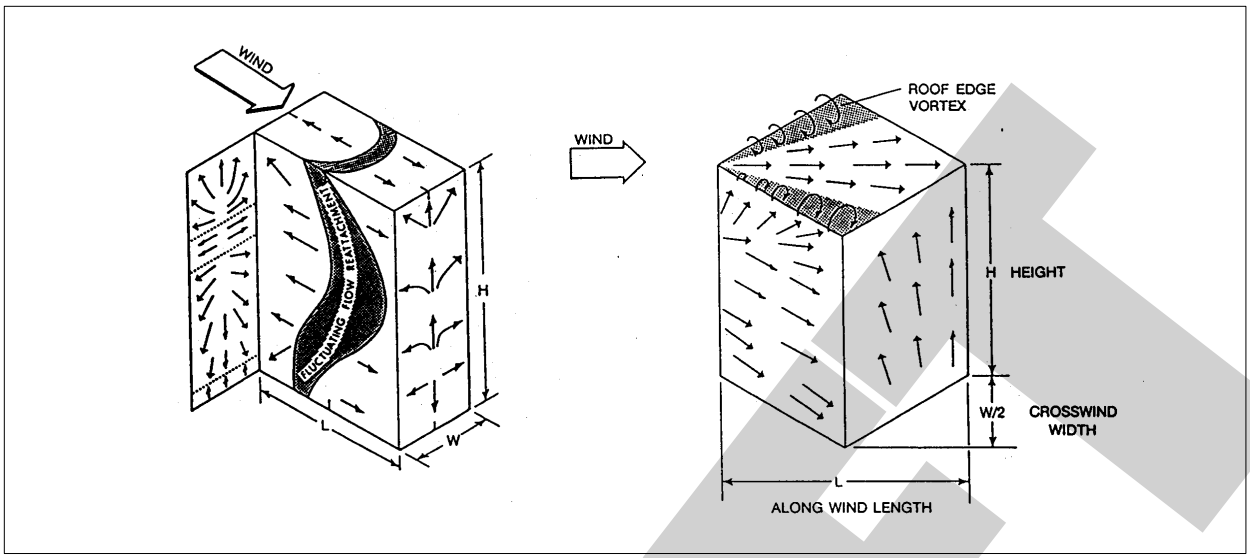


**FIGURE 5–58 BUILDING SURFACE FLOW PATTERNS**

side openings. A building with internal partitions and openings is under various pressures depending on the relative sizes of the openings and the wind direction. With larger openings on the windward face, the building interior tends to remain under positive pressure; the reverse is true when the larger openings are on the leeward face.

With natural ventilation, wind may augment, impede, or sometimes reverse the airflow through a building. For large roof areas (Figure 5-63), the wind can reattach to the roof downstream from the cavity and thus reverse natural ventilation discharging out of monitor or similar windows. These reversals can be avoided by using stacks, continuous roof ventilators, or other exhaust devices in which the flow is augmented by the wind.

To offset changing building pressures, HVAC duct systems must be designed for prevailing conditions, but with provisions like additional zoning, to take care of the unusual conditions. Higher system pressures may be used where system stability is essential, but this method can substantially increase long-term operating costs and energy use.

Wind effect can be reduced by careful selection of inlet and outlet locations. Because wall surfaces are subject to a wide variety of positive and negative pressures, wall openings should be avoided. Mechanical ventilation systems should operate at a high enough pressure to minimize wind effect. Low-pressure systems and propeller exhaust fans should not be used with wall openings unless their ventilation rates are small or used in non-critical services such as storage areas.

Although roof air intakes within flow re-circulation zones are the most effective to minimize wind effect on system flow, current and future air quality in these zones must be considered. Locations should be avoided if a source of contamination exists. The best area for roof intakes is near the middle of the roof, as the negative pressure there is least affected by changes in wind direction. Avoid edges of the roof and walls, where large, wind-driven pressure fluctuations occur. Either vertical or horizontal (mushroom) openings can be used. On large roof areas where the intake may be outside the roof re-circulation zone, mushroom or 180 degree gooseneck designs minimize impact pressure from wind. The frequently used 135 degree gooseneck or vertical louvered openings are undesirable for this purpose.

Exhausts for heated air or industrial contaminants should discharge vertically through stacks, above the roof re-circulation zone. Horizontal, louvered (45 degree down), and 135 degree gooseneck discharges are undesirable even for heat removal systems because of their sensitivity to wind effect. A 180 degree gooseneck for systems handling hot air may be undesirable, because of air impingement on tar and felt roofs. Vertically-discharging stacks located in a re-circulation region (except near a wall) have the advantage of being subjected only to negative pressure created by wind flow over the tip of the stack.

### 5.42.3 Wind Effect

With few exceptions, building intakes and exhausts cannot be located or oriented for prevailing wind. Wind can assist or hinder supply and exhaust air fans, depending on their position on the building, but even

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in locations with a predominant wind direction, the ventilating system must perform adequately for all wind directions.

Airflow through a wall opening results from positive or negative external and internal pressures. During high winds, such differential pressures may exceed 0.5 in. wg (125 Pa). Supply and exhaust systems, and openings, dampers, louvers, doors, and windows make the building flow conditions too complex for calculation. The opening and closing of doors and windows by building occupants add further complications.

Mechanical HVAC systems are affected by wind conditions. A low-pressure wall exhaust fan, 0.05 to 0.1 in. wg. (12 to 25 Pa) can suffer a drastic reduction in capacity. Flow can be reversed by wind pressures on windward walls, or its rate can be increased substantially when subjected to negative pressures on the lee. Clarke (1967) when measuring HVAC Systems operating at 1 to 1.5 in. wg (250 to 375 Pa), found flow rate changes of 25 percent for wind blowing into intakes on an L-shaped building compared to the reverse condition.

For mechanical systems, the wind can be thought of as producing a pressure in series with a system fan, either assisting or opposing. Where system stability is essential, the supply and exhaust air systems must be designed for pressures in the 3 to 4 in. wg (750 to 1,000 Pa) range to minimize unacceptable variations in airflow. To conserve energy, the system pressure selected should be consistent with system needs.

Where building balance and minimum infiltration are important, consider the following:

a. Fan system design with operating pressure adequate to minimize wind effects.

b. Controls to regulate airflow or pressure or both.

c. Separate supply and exhaust systems to serve each building area requiring control or balance.

d. Self-closing doors or double-door air locks to non-controlled adjacent areas, particularly outside doors.

e. Sealing windows and other leakage sources and closing natural vent openings.

### 5.42.4 Stack Effect

When the outside air is colder than the inside air, an upward movement of air often occurs within building shafts such as stairwells, elevator shafts, dumbwaiter shafts, mechanical shafts, or mail chutes. This phenomenon, referred to as stack effect, is caused by the air in the building being warmer and less dense than the outside air. Stack effect is greater when outside temperatures are low and in taller buildings. However, stack effect can exist even in a one-story building.

When the outside air is warmer than the building air, a downward airflow frequently exists in shafts. This downward airflow is called "reverse stack effect." At standard atmospheric pressure, the pressure difference due to either normal or reverse stack effect is expressed as:

**Equation 5–37**

$$\Delta p = k_s\left(\frac{1}{T_0} - \frac{1}{T_1}\right)h$$

(or)

$$\frac{\Delta p}{h} = k_s\left(\frac{1}{T_0} - \frac{1}{T_1}\right)$$

Where:

$\Delta p$ = pressure difference, in. wg (Pa)

$T_0$ = absolute temperature or outside air, °R (K)

$T_1$ = absolute temperature of air inside shaft, °R(K)

$h$ = distance above neutral plane, ft (m)

$k_s$ = coefficient, 7.64 (3460)

For a building 200 ft (60 m) tall with a neutral plane at the mid-height, an outside temperature of 0 degree F (-18 degree C) and an inside temperature of 70 degree F (21 degree C), the maximum pressure difference due to stack effect would be 0.22 in. wg (55 Pa).

This means that at the top of the building, a shaft would have a pressure of 0.22 in. wg (55 Pa) greater than the outside pressure. At the bottom of the shaft, the shaft would have a pressure of 0.22 in. wg (55 Pa) less than the outside pressure.

Figure 5-64 diagrams the pressure difference between a building shaft and the outside. In the diagram, a positive pressure difference indicates that the shaft pressure is higher than the outside pressure and a negative pressure difference indicates the opposite. These pres-

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sures would affect all HVAC systems operating throughout the spaces.

Stack effect usually exists between a building and the outside. The air movement in buildings caused by stack effect is illustrated in Figure 5-65. In this case, the pressure difference expressed in Equation 5-37 refers to the pressure difference between the shaft and the outside of the building.

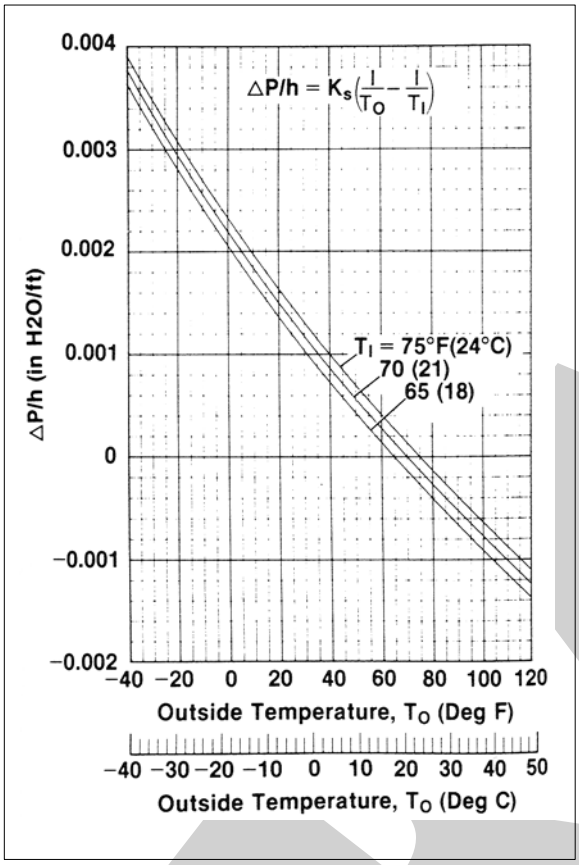


**FIGURE 5–59 PRESSURE DIFFERENCE DUE TO STACK EFFECT**

Figure 5-64 can be used to determine the pressure difference due to stack effect. For stack effect, $\Delta p/h$ is positive and the pressure difference is positive, above the neutral plane and negative below it. For reverse stack effect, $\Delta p/h$ is negative and the pressure difference is negative above the neutral plane and positive below it.

In unusually tight buildings with exterior stairwells, reverse stack effect has been observed even with low outside air temperatures (Klote 1980). In this situation, the exterior stairwell temperature was considerably lower than the building temperature. The stairwell held a cold column of air and other shafts within the building provided warm columns of air.

If the building's leakage paths are uniform with height, the neutral plane is near the mid-height of the building. However, when the leakage paths are not uniform, the location of the neutral plane can vary considerably, as in the case of vented shafts.

## 5.43 BUILDING AIRLOW CONTROL

For buildings that contain airborne contaminants, the correct airflow for HVAC systems is toward the contaminated areas. Airflow direction is maintained by controlling pressure differentials between spaces. In a laboratory building, for example, peripheral rooms such as offices and conference rooms are maintained at a positive pressure and laboratories at a negative pressure, with reference to corridor pressure. Pressure differentials between spaces are normally obtained by balancing the HVAC supply system airflows in the spaces in conjunction with the exhaust systems in the laboratories with differential pressure instrumentation to control the airflow.

An outdoor reference probe that senses static pressure at doorways and air intakes can be used to control airflow and pressure in corridors. The differential pressure measured between the corridor and the outside may then signal a controller to increase or decrease airflow to the corridor. Unfortunately, it is difficult to locate an external probe where it will sense the proper external static pressure. High wind velocity and resulting pressure changes around entrances can cause variations in pressure. Care must be taken to ensure that the probe is unaffected by wind pressure.

Using the corridor pressure as the reference can control the pressure differential for a room adjacent to a corridor. Outdoor pressure cannot control pressure differentials within rooms, even during periods of relatively constant wind velocity and pressure. A single pressure sensor can measure the outside pressure at one point only and will not be representative of pressures elsewhere.

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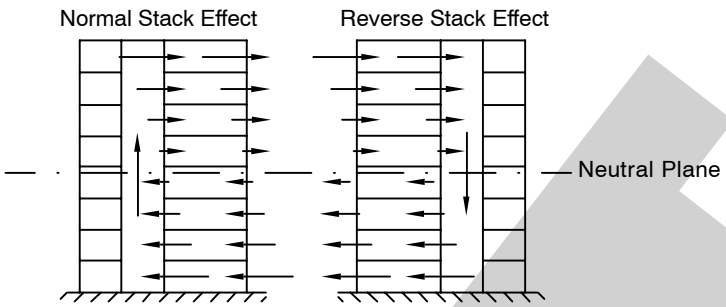


NOTE: ARROWS INDICATE DIRECTION OF AIR MOVEMENT

**FIGURE 5–60 AIR MOVEMENTS DUE TO NORMAL AND REVERSE STACK EFFECT**

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CHAPTER 6

# FAN–DUCT CONNECTION PRESSURE LOSSES

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## 6.1 SCOPE

Most of the text and accompanying tables and figures in this chapter were developed by the Air Movement and Control Association International, Inc. and reprinted with their permission AMCA Publication 201 – *Fans and Systems*.

System Effect Curves were discussed in Chapter 5 but the basics will be repeated as they relate to fan equipment connections. Figure 6-1 shows a series of 24 System Effect Curves. By entering the chart at the appropriate air velocity, it is possible to read across from any curve to find the System Effect Factor for a particular configuration. System Effect Curve "letter designations" may be obtained from Tables 6-1 through 6-4 and Figures 6-9, 6-11, 6-12, and 6-17 in this section. The System Effect Factor is given in in., wg or Pa, it must be added to the total system pressure losses or subtracted from the fan performance pressure rating.

The velocity rate used for the chart will be the inlet or outlet velocity of the fan, depending on whether the configuration is related to the fan inlet or outlet. Most catalog ratings include outlet velocity figures, but for centrifugal fans, it may be necessary to calculate the inlet velocity, *see* Figures 6-20 and 6-21. Dimensioned drawings are usually included in the fan catalog.

If more than one configuration is included in a system, the System Effect Factor for each must be determined separately, and the total of these System Effects must be added to the total system pressure losses, or subtracted from the fan pressure rating.

## 6.2 FAN OUTLET CONDITIONS

### 6.2.1 Fan Outlet Ducts

Fans intended primarily for use with duct systems are usually tested with an outlet duct in place. The system designer should examine catalog ratings carefully for statements defining whether the published ratings are based on tests made with outlet ducts, inlet ducts, both, or no ducts. If information is not available, assume that the tests were made only with an outlet duct.

AMCA Standard 210 specifies an outlet duct that is not greater than 105 percent, nor less than 95 of the fan outlet area. It also requires that the included angle of the transition elements should not be greater than 15 degree for converging elements, nor greater than 7 degree for diverging elements.

Figure 6-2 shows the change in velocity profiles at various distances from the fan outlet. For 100 percent recovery, the duct, including the transition, should extend at least two and one half equivalent duct diameters, and will need to be as long as six equivalent duct diameters at outlet velocities of 6000 fpm (30 m/s) and higher. If it is not possible to use a full-length outlet duct, a System Effect Factor must be added to the system resistance losses.

To determine the applicable System Effect Factor, calculate the average velocity in the outlet duct and enter the System Effect Curves (Figure 6-1) at this velocity. Select the appropriate System Effect Curve from Table 6-1. The ratio of blast area to outlet area is not usually included in fan catalog data and it will be necessary to obtain this from the fan manufacturer.

The System Effect Factor includes only the effect of the system configuration on the fan's performance. Any additional friction losses due to additional ductwork should be added to the calculated system pressure loss. Also, System Effect cannot be field measured, only calculated.

### 6.2.2 Fan Outlet Diffusers

The process taking place in the outlet duct is often referred to as "static regain." The relatively high-velocity airstream leaving the blast area of the fan gradually expands to fill the duct. The kinetic energy—velocity pressure—decreases and the potential energy—static pressure—increases.

In many systems, it may be feasible to use an outlet duct that is considerably larger than the fan outlet. In these cases, converting some of the fan outlet velocity pressure to static pressure can increase the static pressure available to overcome system resistance.

To achieve this conversion efficiently, it is necessary to use a connection piece between the fan outlet and the duct that allows the airstream to expand gradually. This is called a diffuser.

The efficiency of conversion will depend upon the angle of expansion, the length of the diffuser section, and the blast area/outlet area ratio of the fan.

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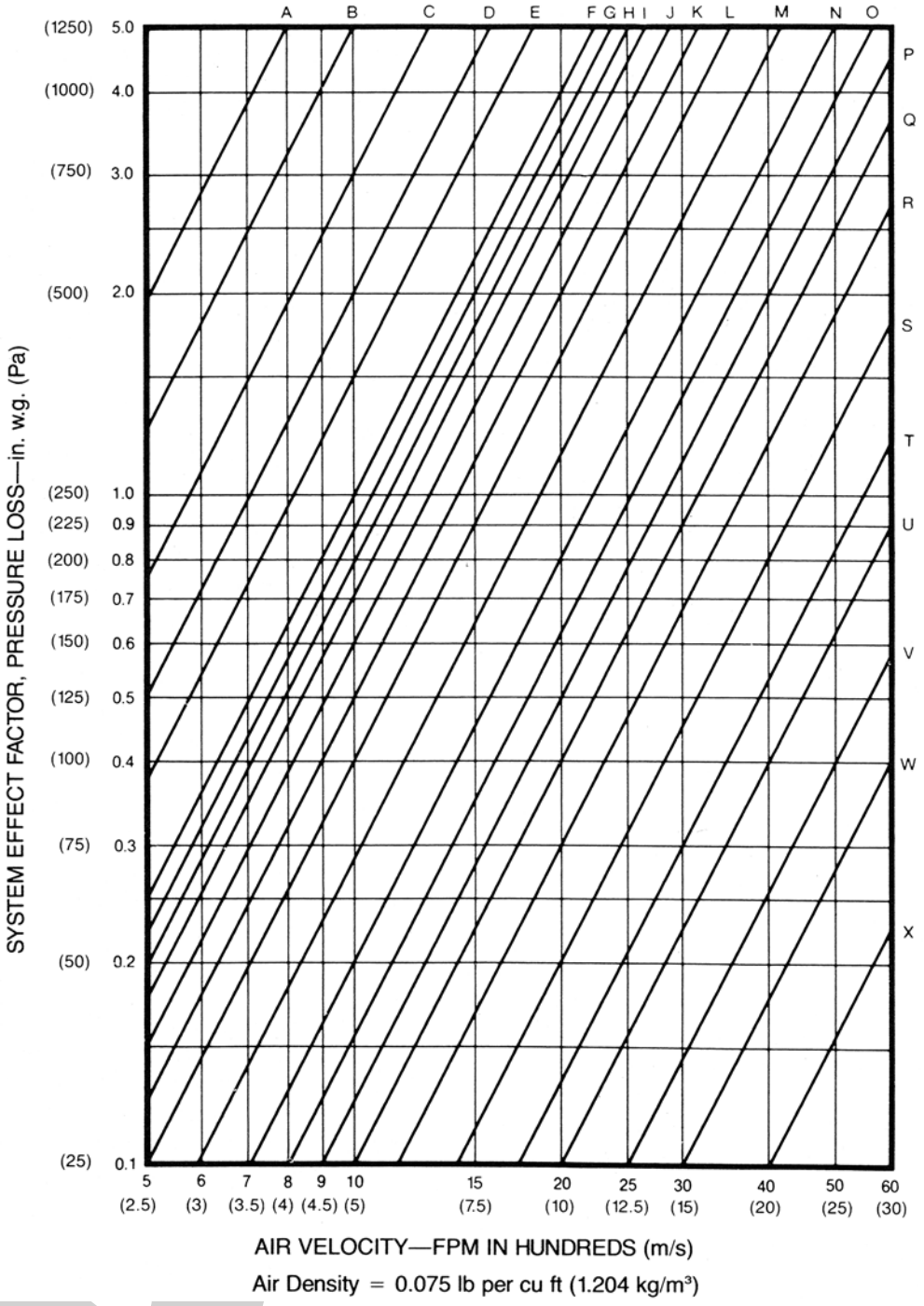
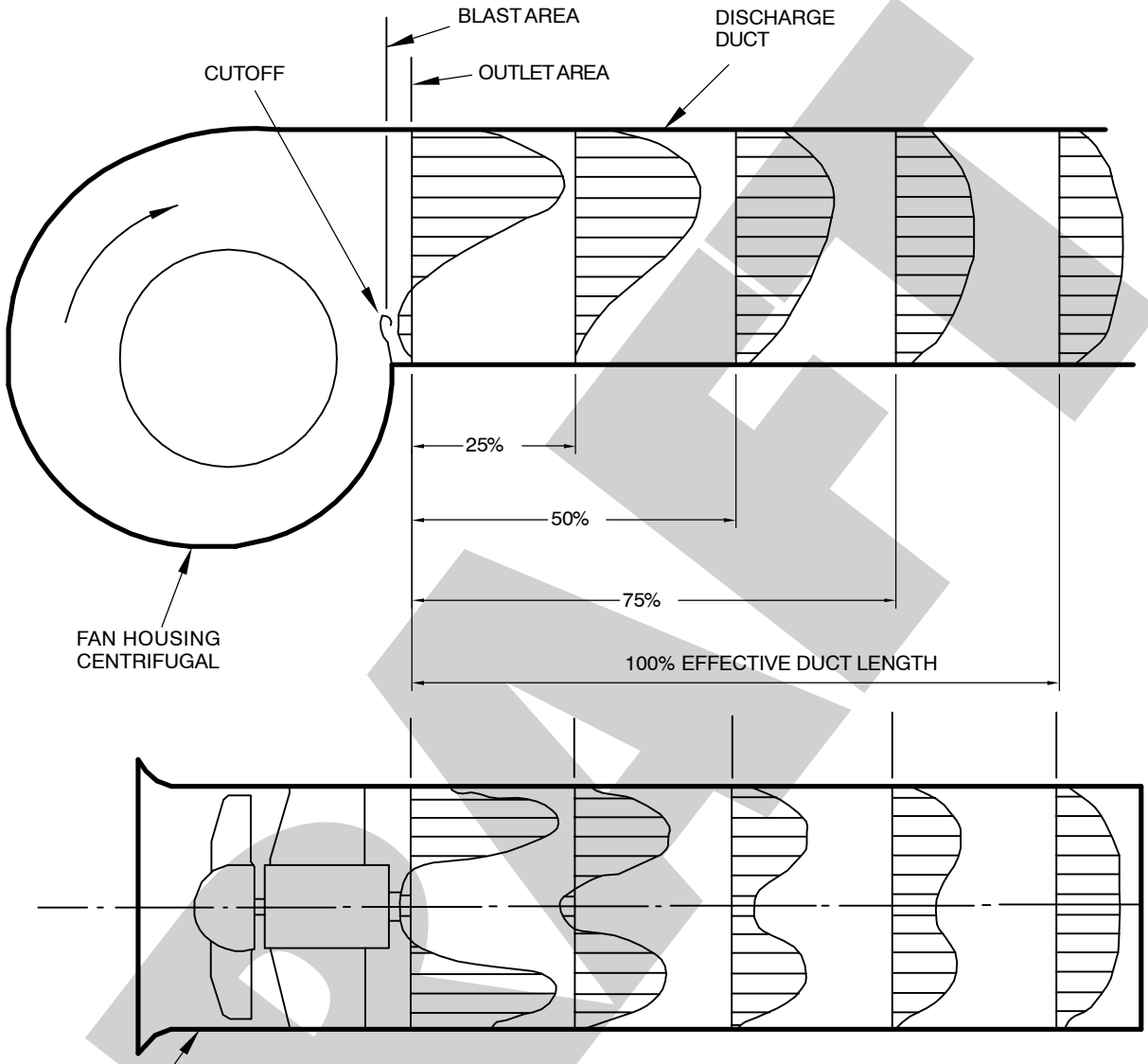



**FIGURE 6–1 SYSTEM EFFECT CURVES**




TO CALCULATE 100 PERCENT EFFECTIVE DUCT LENGTH, ASSUME A MINIMUM OF 21/2 DUCT DIAMETERS FOR 2500 FPM (12.5 M/S) OR LESS. ADD 1 DUCT DIAMETER FOR EACH ADDITIONAL 1000 FPM (5 M/S)

EXAMPLE: 5000 FPM (25 M/S) = 5 EQUIVALENT DUCT DIAMETERS.

IF DUCT IS RECTANGULAR WITH SIDE DIMENSIONS A AND B, THE EQUIVALENT DUCT DIAMETER IS EQUAL TO $\sqrt{\frac{4ab}{\pi}}$

TRANSITIONS (SLOPE OF ELEMENTS NOT GREATER THAN 15 DEGREES FOR CONVERGING AND 7 DEGREES FOR DIVERGING) INCLUDED IN EFFECTIVE DUCT LENGTHS.

**FIGURE 6–2 CONTROLLED DIFFUSION AND ESTABLISHMENT OF A UNIFORM VELOCITY PROFILE IN A STRAIGHT LENGTH OF OUTLET DUCT**

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| | No Duct | 12% Effective Duct | 25% Effective Duct | 50% Effective Duct | 100% Effective Duct |
|---|---|---|---|---|---|
| **Pressure Recovery** | **0%** | **50%** | **80%** | **90%** | **100%** |
| **Blast Area Outlet Area** | **System Effect Curve** | | | | |
| 0.4 | P | R-S | U | W | – |
| 0.5 | P | R-S | U | W | – |
| 0.6 | R-S | S-T | U-V | W-X | – |
| 0.7 | S | U | W-X | – | – |
| 0.8 | T-U | V-W | X | – | – |
| 0.9 | V-W | W-X | – | – | – |
| 1.0 | – | – | – | – | – |

**Table 6–1 System Effect Curves for Outlet Ducts**

### 6.2.3 Fan Outlet Duct Elbows

Values for pressure losses through elbows are based upon a uniform velocity profile approaching the elbow. Any non-uniformity in the velocity profile ahead of the elbow will result in a pressure loss greater than the published value.

The amount of increased loss will depend upon the location and orientation of the elbow, relative to the fan outlet. In some cases, the effect of the elbow will further distort the outlet-velocity profile of the fan. This will increase the losses and can result in so much uneven flow that branch takeoffs near the elbow will not deliver their designated airflow.

Where possible, a length of straight duct should be installed at the fan outlet to permit diffusion and development of a uniform-flow profile, before an elbow is inserted in the duct. If an elbow must be located near the fan outlet, it should have a minimum center-line radius to duct-diameter ratio of 1.5 and should be arranged to give the most uniform airflow possible, as shown in Figure 6-3.

Table 6-2 lists System Effect Factor Curves that can be used to estimate the effect of an elbow at the fan outlet. It also shows the reduction in losses resulting from use of a straight-outlet duct.

### 6.2.4 Turning Vanes

Turning vanes will usually reduce the pressure loss through an elbow, but where a non-uniform approach velocity profile exists, such as a fan outlet, the vanes may actually serve to continue the non-uniform profile beyond the elbow. This may result in increased losses in other system components downstream of the elbow.

### 6.2.5 Fan Volume Control Dampers

The fan manufacturer can furnish dampers as accessory equipment. However, in many systems a volume control damper will be located by the designer in the ductwork at or near the fan outlet, *see* Figure 6-24.

Volume control dampers are manufactured with either "opposed" or "parallel" blades. When partially closed, the parallel-bladed damper diverts the airstream to the side of the duct. This results in a non-uniform velocity profile beyond the damper and flow to close downstream branch ducts may be seriously affected, *see* Figure 6-4.

The use of an opposed-blade damper is recommended when volume control is required at the fan outlet and there are other system components, such as coils or branch takeoffs, downstream of the fan. When the fan discharges into a large plenum or free space, a parallel-blade damper may be satisfactory.

For a centrifugal fan, the best air performance usually will be achieved by installing the damper with its blades perpendicular to the fan shaft. However, other considerations may require installation of the damper with its blades parallel to the fan shaft.

Published pressure losses for control dampers are based upon uniform approach velocity profiles. When a damper is installed close to the outlet of a fan, the approach velocity profile is non-uniform and much higher pressure losses through the damper can result. Figure 6-5 lists multipliers that should be applied to the damper manufacturer's catalog pressure loss when the damper is installed at the outlet of a centrifugal fan.

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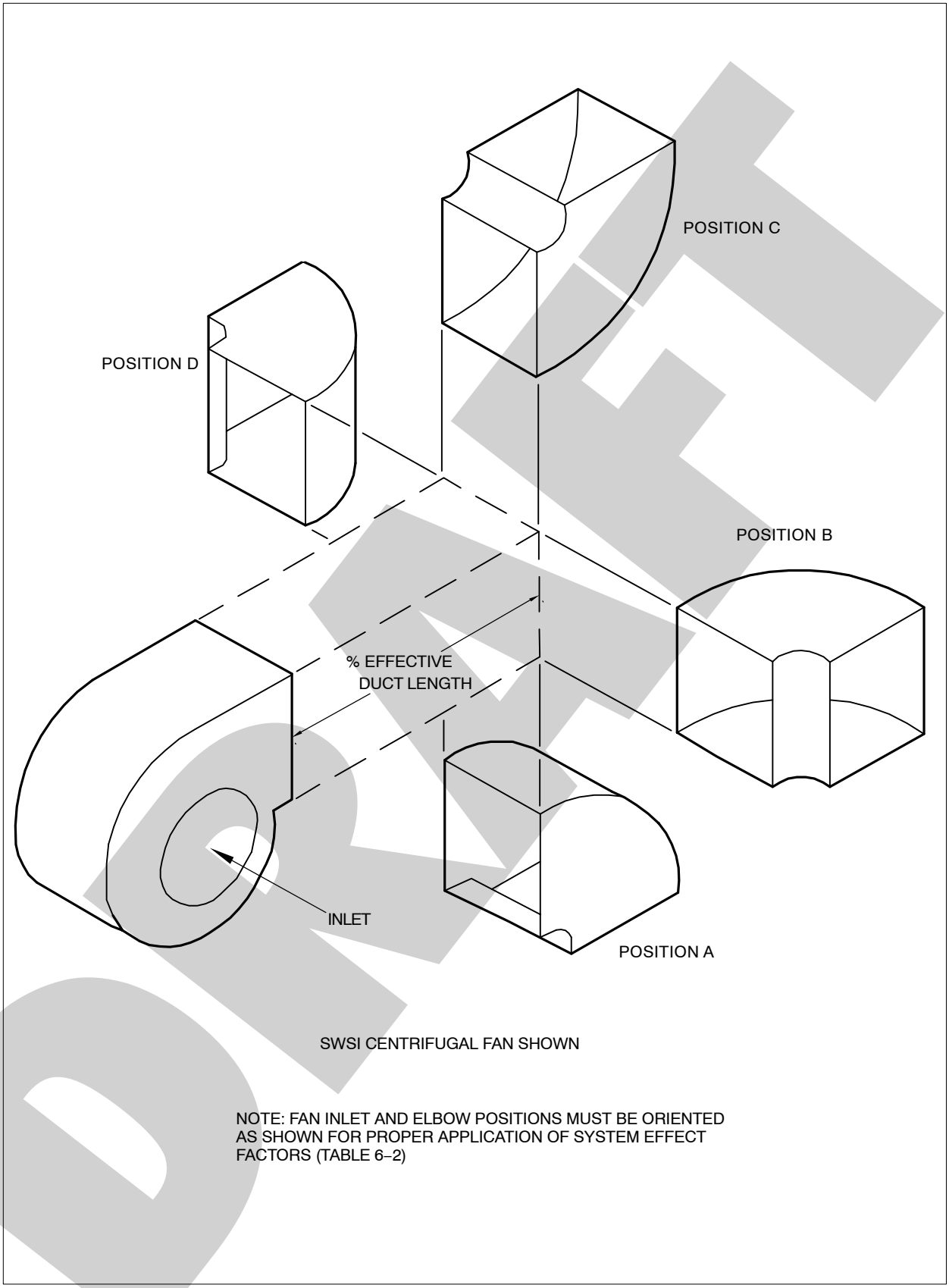


**FIGURE 6–3 OUTLET DUCT ELBOWS**

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| Blast Area Outlet Area | Outlet Elbow Position | No Outlet Duct | 12% Effective Duct | 25% Effective Duct | 50% Effective Duct | 100% Effective Duct |
|---|---|---|---|---|---|---|
| 0.4 | A | N | O | P-Q | S | NO SYSTEM EFFECT FACTOR |
| | B | M | M-N | O | R | |
| | C | L-M | M | N | Q | |
| | D | L-M | M | N | Q | |
| 0.5 | A | P | Q | R | T | |
| | B | N-O | O-P | P-Q | S | |
| | C | M-N | N-O | O-P | R-S | |
| | D | M-N | N-O | O-P | R-S | |
| 0.6 | A | Q | Q-R | R-S | U | |
| | B | P | Q | R | T | |
| | C | N-O | O-P | P-Q | S | |
| | D | O | P | Q-R | S-T | |
| 0.7 | A | S-T | T | U | W | |
| | B | R-S | S | T | V | |
| | C | Q-R | R | S | U-V | |
| | D | R | R-S | S-T | U-V | |
| 0.8 | A | S | S-T | T-U | V-W | |
| | B | R | R-S | S-T | U-V | |
| | C | Q | Q-R | R-S | U | |
| | D | Q-R | R | S | U-V | |
| 0.9 | A | S-T | T | U | W | |
| | B | R-S | S | T | V | |
| | C | R | R-S | S-T | U-V | |
| | D | R-S | S | T | V | |
| 1.0 | A | R-S | S | T | V | |
| | B | S-T | T | U | W | |
| | C | R-S | S | T | V | |
| | D | R-S | S | T | V | |

**Table 6–2 System Effect Factor Curves for Outlet Elbows**

**System Effect Factor Curves for SWSI Fans**

For DWDI fans determine effect factor curve using the above tabulation for SWSI fans. Next determine system effect factor ($\Delta$ P) by using Figure 6-1. The apply appropriate multiplier from tabulation below:

**Multipliers from DWDI fans**

Elbow Position B = $\Delta \times 1.25$
Elbow Position D = $\Delta$ P $\times 0.85$
Elbow Position A and C = $\Delta$ P $\times 1.00$

Refer to Figure 6-3 for elbow position designation curves

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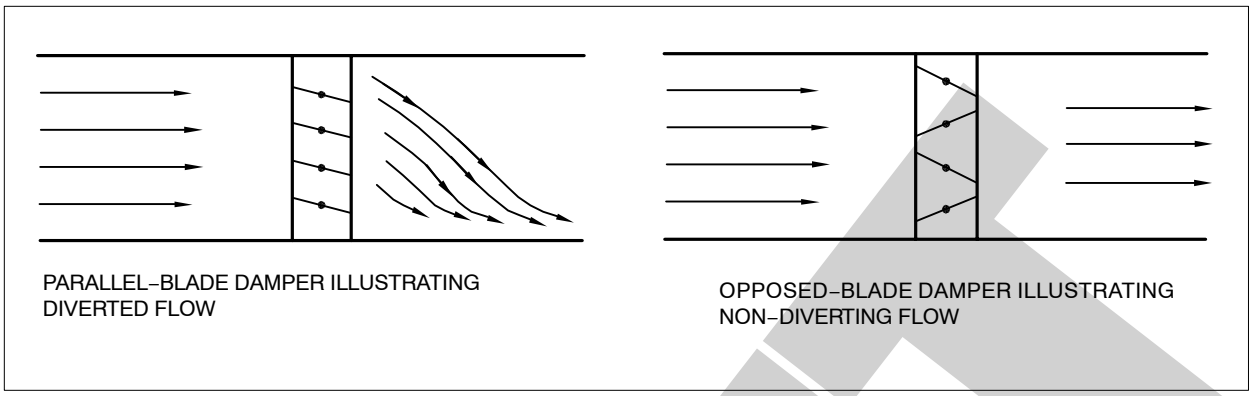


**FIGURE 6–4 PARALLEL VERSES OPPOSED DAMPERS**

### 6.2.6 Duct Branches

Standard procedures for the design of duct systems are all based on the assumption of uniform flow profiles in the system, *see* Figure 6-6.

If branch takeoffs or splits are located close to the fan outlet, non-uniform flow conditions will exist and pressure loss and airflow may vary widely from design intent. Where possible, a length of straight duct should be installed between the fan outlet and any split or branch takeoff.

## 6.3 FAN INLET CONDITIONS

Fan inlet swirl and non-uniform inlet flow can often be corrected by inlet straightening vanes or guide vanes. Restricted fan inlets located too close to walls or obstructions or restrictions caused by a plenum or cabinet will decrease the performance of a fan. Cabinet-clearance effect (plenum-effect) is considered a component part of the entire system. The pressure losses through the cabinet plenum must be considered as a System Effect when determining system characteristics.

### 6.3.1 Inlet Ducts

Some fans intended primarily for use as "exhausters" may be tested with an inlet duct in place, or with a special bell-mouth inlet to stimulate the effect of a duct. Figure 6-6 illustrates the variations in inlet flow that will occur. A ducted inlet condition is shown as (a), the un-ducted condition as (d), and the effect of a bell-mouth inlet as (f).

Flow into a sharp-edged duct as shown in (c), or into an inlet without a smooth entry as shown in (d), is similar to flow through a sharp-edged orifice in that a "vena contracta" is formed. The reduction in flow area caused by the vena contracta and the following rapid expansion causes a loss that should be considered as a System Effect. This loss can be largely eliminated by providing the duct or fan inlet with a rounded entry, as shown in (e) and (f). If it is not practical to include such a smooth entry, a converging taper will substantially diminish the loss of energy. Even a simple flat flange on the end of a duct will reduce the loss to about one-half of the loss through an un-flanged entry.

AMCA Standard 201 limits an inlet duct to a cross-sectional area not greater than 112.5 percent nor less than 92.5 percent of the fan inlet area. The included angle of transition elements is limited to 15 degrees converging and 7 degrees diverging.

### 6.3.2 Inlet Elbows

Non-uniform flow into the inlet is the most common cause of deficient fan performance. An elbow or a 90 degree duct turn located at the fan inlet will not allow the air to enter uniformly and will result in turbulent and uneven flow distribution at the fan impeller. Air has weight and a moving airstream has momentum; therefore, the airstream resists a change in direction within an elbow as illustrated in Figures 6-9 and 6-10.

The System Effect Curves for round-section elbows with given Radius/Diameter (R/D) ratios are listed on Figure 6-9. The System Effect Factor for a particular elbow can be obtained from Figure 6-1 using the average fan inlet velocity and the tabulated System Effect Curve. This pressure loss must be added to the friction and dynamic losses already determined for that particular elbow, unless they are deducted from the fan capacity. This System Effect Factor loss only applies when the elbow is located at the fan inlet as shown in Figure 6-9.

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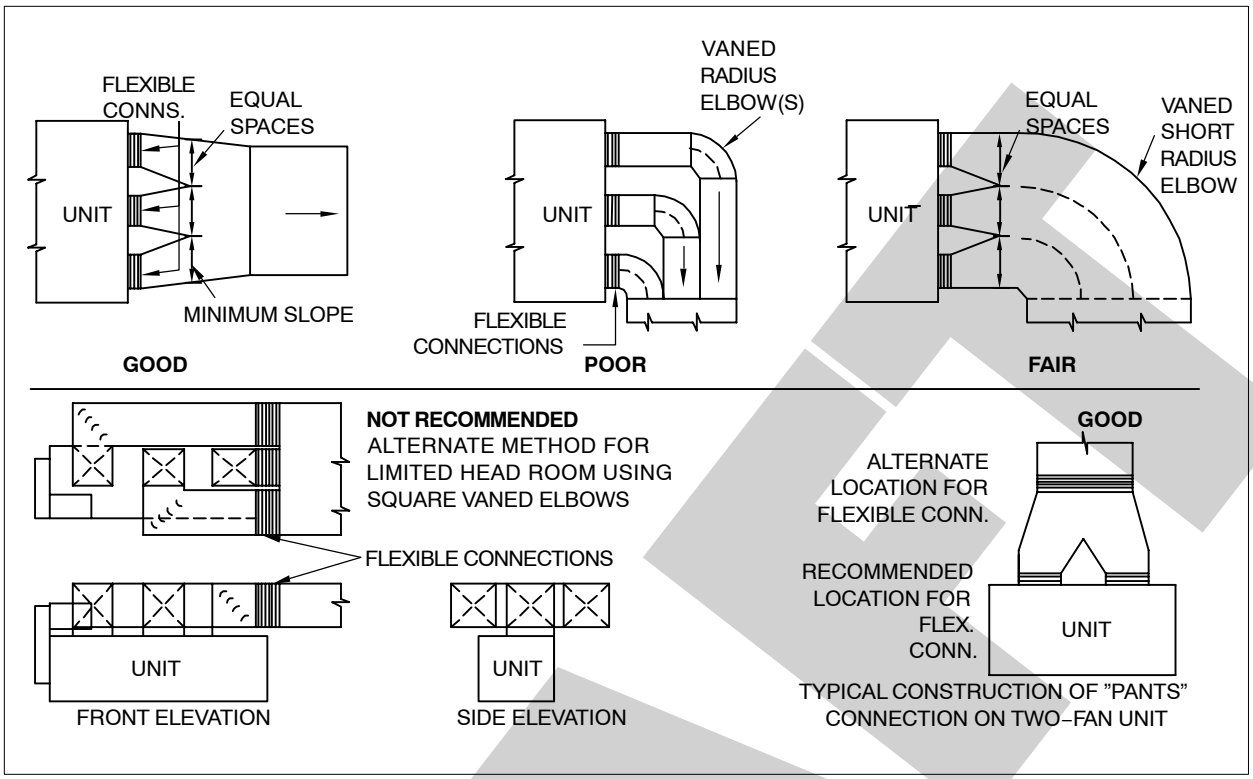


**FIGURE 6–5 TYPICAL HVAC UNIT CONNECTIONS**

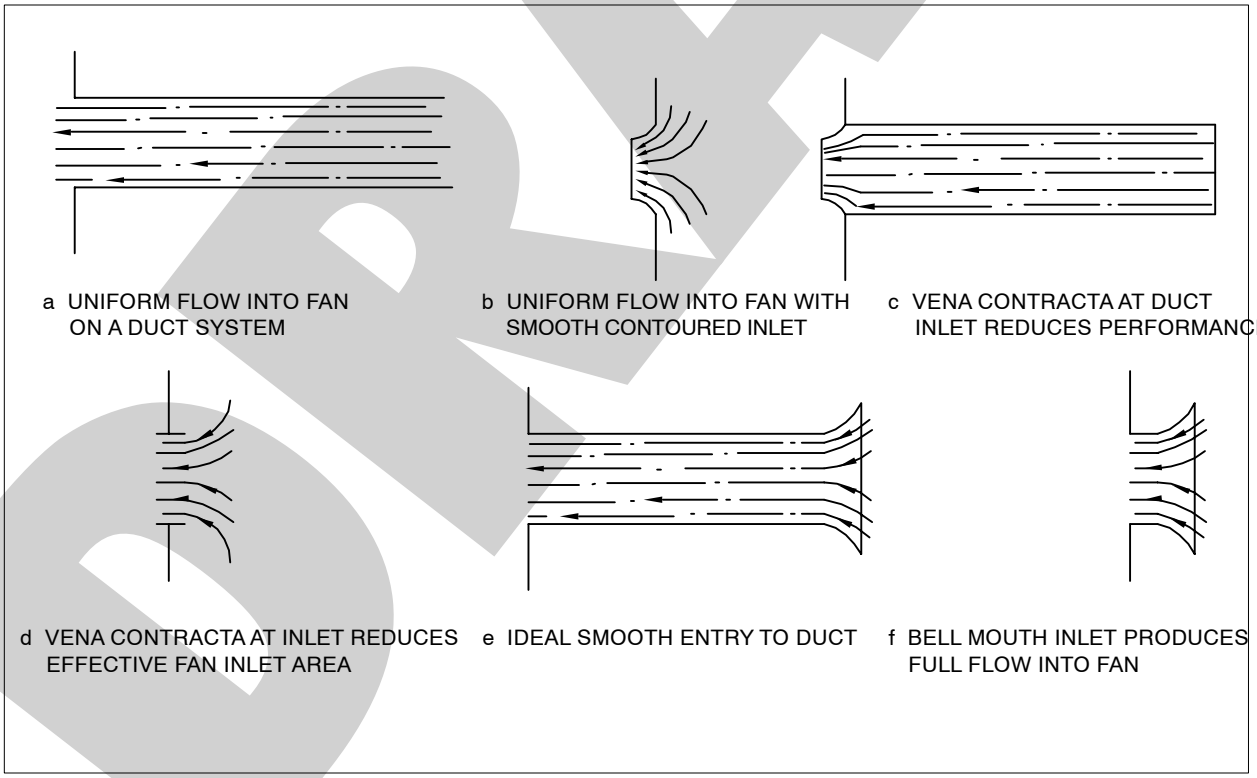


**FIGURE 6–6 TYPICAL INLET CONNECTIONS FOR CENTRIFUGAL AND AXIAL FANS**

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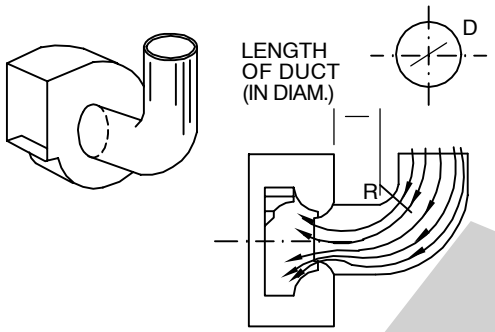
Refer to Figures 6-11 and 6-12 for the System Effect Curves for other inlet elbows and 90 degree duct turns which produce non-uniform inlet flow. Note that when duct turning vanes and a suitable length of duct is used (three to eight diameters long, depending on velocities) between the fan inlet and elbow, the System Effect Factor is not as great or is off of the chart. These improvements help maintain uniform flow into the fan inlet, and approach the flow conditions of the laboratory test setup. Most fan manufacturers can furnish design and System Effect information for special inlet boxes for particular flow and entry conditions, *see* Figure 6-20.




**SYSTEM EFFECT CURVES**

| R/D | NO DUCT | 2D DUCT | 5D DUCT |
|---|---|---|---|
| 0.75 | Q–R | S | U |
| 1.0 | R | S–T | U–V |
| 2.0 | R–S | T | U–V |
| 3.0 | S–T | U | V–W |

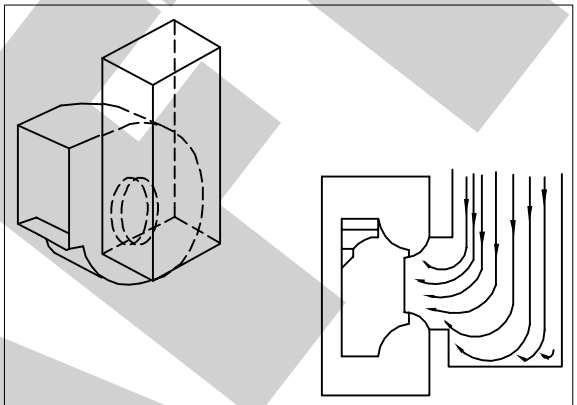
**FIGURE 6–7 NON–UNIFORM FLOW INTO A FAN INLET INDUCED BY A 90 DEGREE ROUND SECTION ELBOW – NO TURNING VANES**

### 6.3.3 Inlet Vortex

Another major cause of reduced performance is an inlet-duct condition that produces a vortex or spin in the airstream entering a fan inlet. An example of this condition is illustrated in Figure 6-13.

The ideal inlet condition allows the air to enter axially and uniformly, without spin in either direction. A spin in the same direction as the impeller rotation reduces the pressure-volume curve by an amount dependent upon the intensity of the vortex. The effect is similar to the change in the pressure-volume curve achieved by inlet vanes installed in a fan inlet, which induce a controlled spin and vary the volume flow rate of the system. A counter-rotating vortex at the inlet will result in a slight increase in the pressure-volume curve but the horsepower will increase substantially.

Inlet spin may arise from a variety of approach conditions, sometimes the cause is not obvious. Some common duct connections that cause inlet spin are illustrated in Figure 6-14 but since there are many variations, no System Effect Factors are tabulated. It is recommended that these duct connections should be avoided, but when required, inlet conditions can usually be improved by the use of vanes to break the spinning vortex, *see* Figure 6-15.



THE REDUCTION IN CAPACITY AND PRESSURE FOR THIS TYPE OF INLET CONDITION IS IMPOSSIBLE TO TABULATE. THE MANY POSSIBLE VARIATIONS IN WIDTH AND DEPTH OF THE DUCT INFLUENCE THE REDUCTION IN PERFORMANCE TO VARYING DE–GREES AND THEREFORE THIS INLET SHOULD BE AVOIDED. CAPACITY LOSSES AS HIGH AS 45 PER–CENT HAVE BEEN OBSERVED. EXISTING INSTALLA–TIONS CAN BE IMPROVED WITH VANES OR THE CONVERSION TO SQUARE OR MITERED ELBOWS WITH VANES.

**FIGURE 6–8 NON–UNIFORM FLOW INDUCED INTO FAN INLET BY A RECTANGULAR INLET DUCT**

### 6.3.4 Inlet Duct Vanes

Where space limitations prevent the use of optimum fan inlet connections, more uniform flow can be achieved by the use of vanes in the inlet elbow.

Numerous variations of vanes are available ranging from a single-curved, sheet-metal vane, to multi-bladed “airfoil” vanes.

The pressure drop through elbows with these devices are part of the system pressure losses. The cataloged pressure loss of proprietary vanes will be based upon uniform airflow at the entry to the elbow. If the airflow approaching the elbow is non-uniform, because of the disturbance further upstream in the system, the pres-

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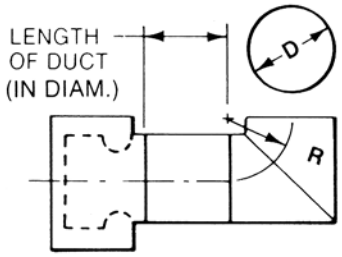
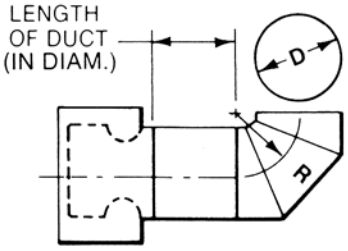



**SYSTEM EFFECT CURVES**

| R/D | NO DUCT | 2D DUCT | 5D DUCT |
|---|---|---|---|
| — | N | P | R-S |

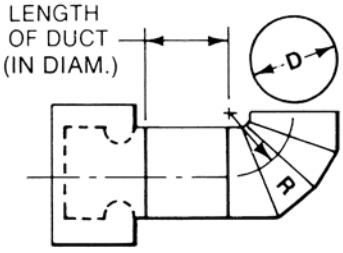
**a** TWO PIECE MITERED 90° ROUND SECTION ELBOW – NOT VANED




**SYSTEM EFFECT CURVES**

| R/D | NO DUCT | 2D DUCT | 5D DUCT |
|---|---|---|---|
| 0.5 | O | Q | S |
| 0.75 | Q | R-S | T-U |
| 1.0 | R | S-T | U-V |
| 2.0 | R-S | T | U-V |
| 3.0 | S | T-U | V |

**b** THREE PIECE MITERED 90° ROUND SECTION ELBOW – NOT VANED




**SYSTEM EFFECT CURVES**

| R/D | NO DUCT | 2D DUCT | 5D DUCT |
|---|---|---|---|
| 0.5 | P-Q | R-S | T |
| 0.75 | Q-R | S | U |
| 1.0 | R | S-T | U-V |
| 2.0 | R-S | T | U-V |
| 3.0 | S-T | U | V-W |

**c** FOUR OR MORE PIECE MITERED 90° ROUND SECTION ELBOW – NOT VANED

**FIGURE 6–9 SYSTEM EFFECTS FOR VARIOUS MITERED ELBOWS WITHOUT VANES**

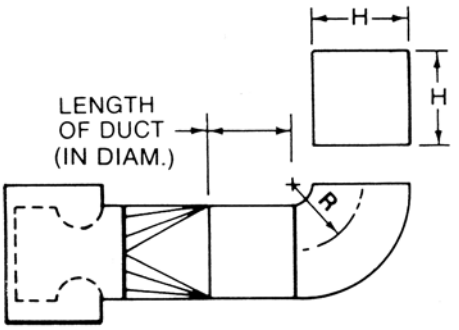
sure loss through the elbow will be higher than the published or calculated figure. The effectiveness of the vanes in the elbow will also be reduced.

### 6.3.5 Straighteners

Airflow straighteners (egg-crates) are often used to eliminate or reduce swirl or vortex flow in a duct. An example of an egg-crate straightener (Figure 6-14) is reproduced from AMCA.

### 6.3.6 Enclosures

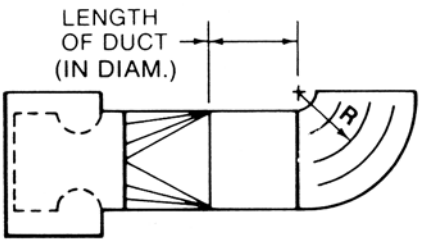
Fans within plenums and cabinets, or next to walls, should be located so air may flow unobstructed into the inlets. Fan performance is reduced if the space between the fan inlet and the enclosure is too restrictive. It is common practice to allow at least one-half impeller diameter between an enclosure wall and the fan inlet. The inlets of multiple, double-width centrifugal fans located in a common enclosure should be at least one impeller diameter apart, if optimum performance is to be expected. Figure 6-15 illustrates fans located




**SYSTEM EFFECT CURVES**

| R/H | NO DUCT | 2D DUCT | 5D DUCT |
|---|---|---|---|
| 0.5 | O | Q | S |
| 0.75 | P | R | S-T |
| 1.0 | R | S-T | U-V |
| 2.0 | S | T-U | V |

a. SQUARE RADIUS ELBOW WITH INLET TRANSITION—NO VANES




**SYSTEM EFFECT CURVES**

| R/H | NO DUCT | 2D DUCT | 5D DUCT |
|---|---|---|---|
| 0.5 | S | T-U | V |
| 1.0 | T | U-V | W |
| 2.0 | V | V-W | W-X |

b. SQUARE RADIUS ELBOW WITH INET TRANSITION—3 LONG SPLITTER VANES

THE INSIDE AREA OF THE SQUARE DUCT (H × H) IS EQUAL TO THE INSIDE AREA CIRCUMSCRIBED BY THE FAN INLET COLLAR. THE MAXIMUM PERMISSIBLE ANGLE OF ANY CONVERGING ELEMENT OF THE TRANSITION IS 15 DEGREE, AND FOR A DIVERGING ELEMENT 7½. $D = \frac{2H}{\sqrt{\pi}}$

**FIGURE 6–10 SYSTEM EFFECTS FOR SQUARE DUCT ELBOWS**

in an enclosure and lists the System Effect Curve for restricted inlets.

The manner in which the airstream enters an enclosure in relation to the fan inlets also affects fan performance. Plenum or enclosure inlets or walls which are not symmetrical with the fan inlets will cause uneven flow and possible inlet spin. Figure 6-16 illustrates this condition which must be avoided to achieve maximum performance from a fan. If this is not possible, inlet conditions can usually be improved with a splitter sheet to break up the inlet vortex, as illustrated in Figure 6-17.

### 6.3.7 Obstructed Inlets

A reduction in fan performance can be expected when an obstruction to airflow is located in the plane of the fan inlet. Structural members, columns, butterfly valves, blast gates, and pipes are examples of common inlet obstructions.

Some accessories, such as fan bearings, bearing pedestals, inlet vanes, inlet dampers, drive guards, and motors, may also cause inlet obstruction.

Obstruction at the fan inlet may be classified in terms of the unobstructed percentage of the inlet area. Because of the shape of the inlet cones for many fans, it is sometimes difficult to establish the area of the fan inlet. Figures 6-19 and 6-20 illustrate the convention adopted for this purpose. Where an inlet collar is provided, (Figure 6-19) the inlet area is calculated from the inside diameter of this collar. Where no collar is provided, the inlet plane is defined by the points of tangent of the fan housing with the inlet cone radius, *see* Figure 6-20.

The unobstructed percentage of the inlet area is calculated by projecting the profile of the obstruction onto the profile of the inlet. The adjusted inlet velocity obtained is then used to enter the System Effect Curve Chart. The System Effect Factor can be determined

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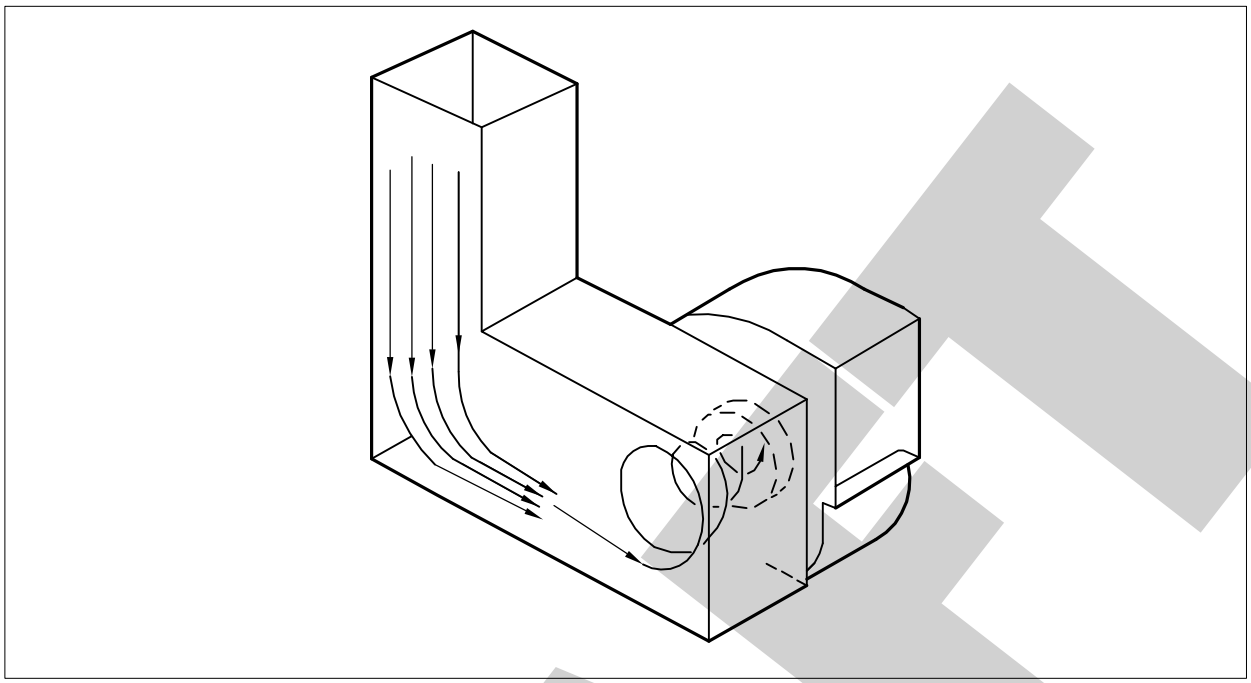

**FIGURE 6–11 EXAMPLE OF A FORCED INLET VORTEX (SPIN–SWIRL)**

from the curve listed for that unobstructed percentage of the inlet area.

### 6.3.8 Field Fabricated Fan Inlet Box

Inlet boxes have been used for years on industrial centrifugal-fan applications with predictable results. The dimensions of the inlet boxes have been established by extensive field testing. Figure 6-18 shows the inlet box configuration and dimensions based on the size of the fan wheel of the centrifugal fan. The inlet box allows a 90 degree connection to the fan with almost no horizontal duct.

The inlet box should be made of a metal gage equal to that of the fan scrolls and it should be bolted tightly to the fan inlet ring with the flexible connection at the return air duct on the inlet of the box. This requires the box to be adequately supported by the fan base and the

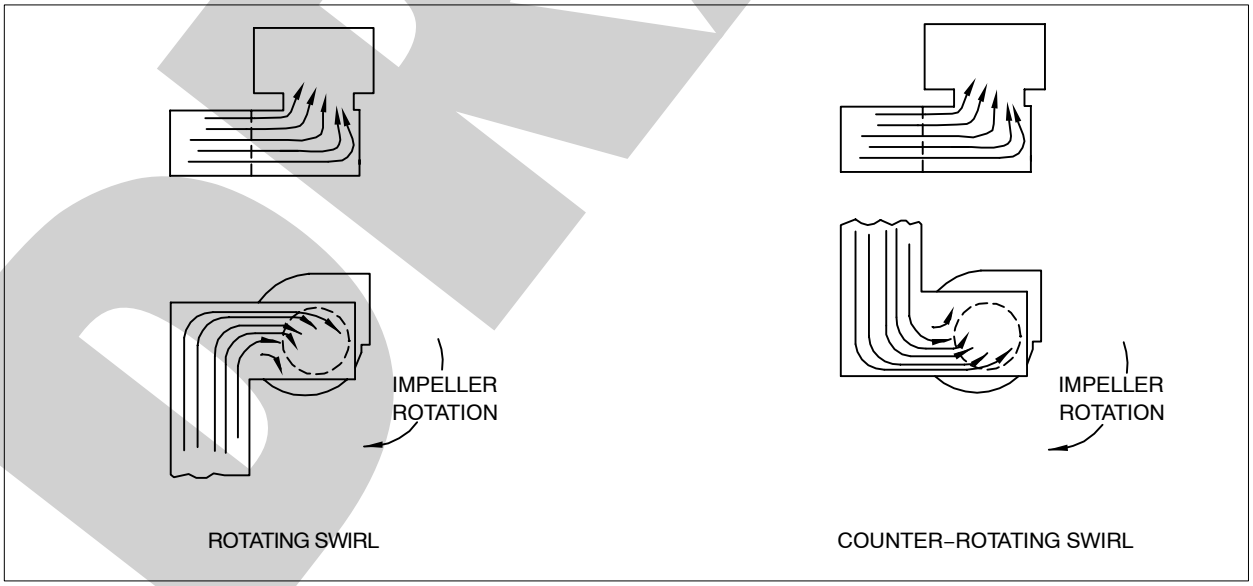


**FIGURE 6–12 INLET DUCT CONNECTIONS CAUSING INLET SPIN**

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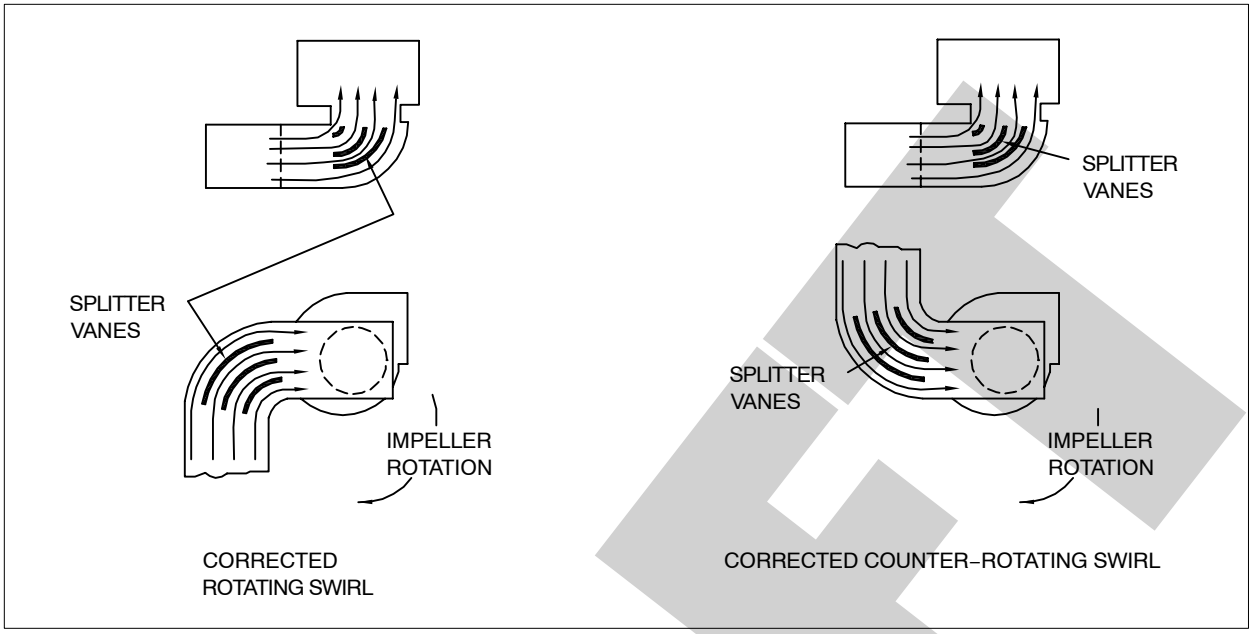


**FIGURE 6–13 CORRECTIONS FOR INLET SPIN**

vibration-isolation pad or mountings to be designed to include the weight of the inlet box.

When an inlet box is used, a duct fitting loss coefficient (c) of 1.0 should be included for the inlet box. This is multiplied by the velocity pressure ($V_p$) based on the return-air duct velocity. No additional System Effect Factor should be calculated.

## 6.4 EFFECTS OF FACTORY SUPPLIED ACCESSORIES

Unless the manufacturer's catalog clearly states to the contrary, it should be assumed that published fan performance data does not include the effects of any accessories supplied with the fan.

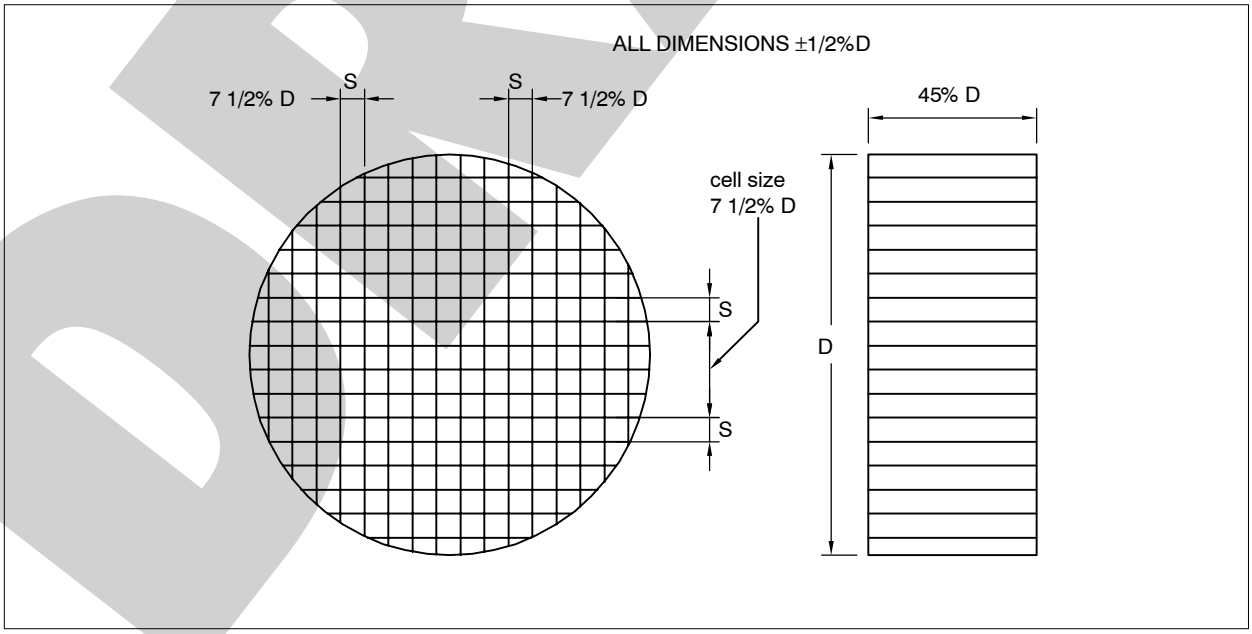


**FIGURE 6–14 AMCA STANDARD 210 FLOW STRAIGHTENER**

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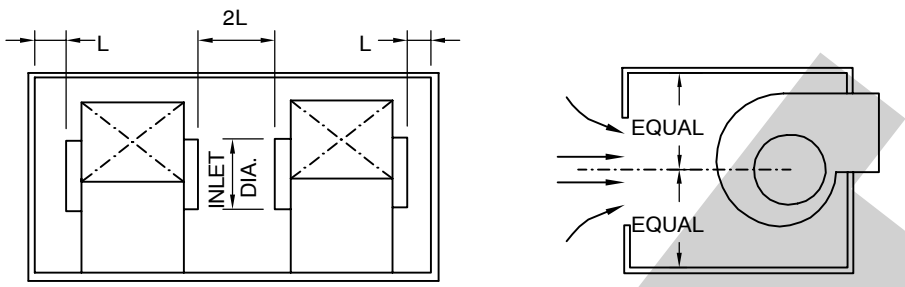


| L DISTANCE INLET TO WALL | SYSTEM EFFECT CURVES |
|---|---|
| 0.75 × DIA OF INLET | V–W |
| 0.5 × DIA OF INLET | U |
| 0.4 × DIA OF INLET | T |
| 0.3 × DIA OF INLET | S |
| 0.2 × DIA OF INLET | R |

**FIGURE 6–15 SYSTEM EFFECT CURVES FOR FANS LOCATED IN PLENUMS AND CABINET ENCLOSURES AND FOR VARIOUS WALL TO INLET DIMENSIONS**

All necessary information should be obtained directly from the fan manufacturer, when possible. The data presented in this chapter are offered only as a guide, in the absence of specific data from the fan manufacturer.

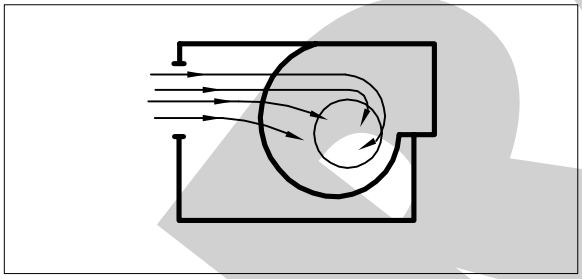

**FIGURE 6–16 ENCLOSURE INLET NOT SYMMETRICAL WITH FAN INLET, PREROTATIONAL VORTEX INDUCED**

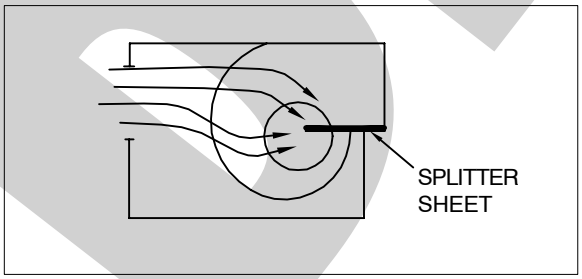


**FIGURE 6–17 FLOW CONDITION OF FIGURE 6–16 IMPROVED WITH A SPLITTER SHEET**

### 6.4.1 Bearing Supports

Some fans require the fan shaft to be supported by a bearing and bearing support in the fan inlet, or adjacent to it.

These components can have an effect on the airflow to the fan inlet and on fan performance, depending on the size of the bearings and supports in relation to the fan inlet opening. Whether the bearing support is located in the actual inlet sleeve, or "stepped out" from the inlet will also have an effect.

In cases where manufacturer's performance ratings do not include the effect of the bearings and supports, it will be necessary to compensate for this inlet restriction by use of the fan manufacturer's allowance for bearings in the fan inlet.

If more accurate data is not available, an approximation may be made as described under 6.3.7 Obstructed Inlets.

### 6.4.2 Drive Guards

Most fans require a belt-drive guard in the area of the fan inlet. Depending on design, the guard may be located at the plane of the inlet, along the casing side sheet, or it may be "stepped out" due to the bearing pedestals.

Depending on the location of the guard and on the inlet velocity, the fan performance may be significantly affected by this obstruction.

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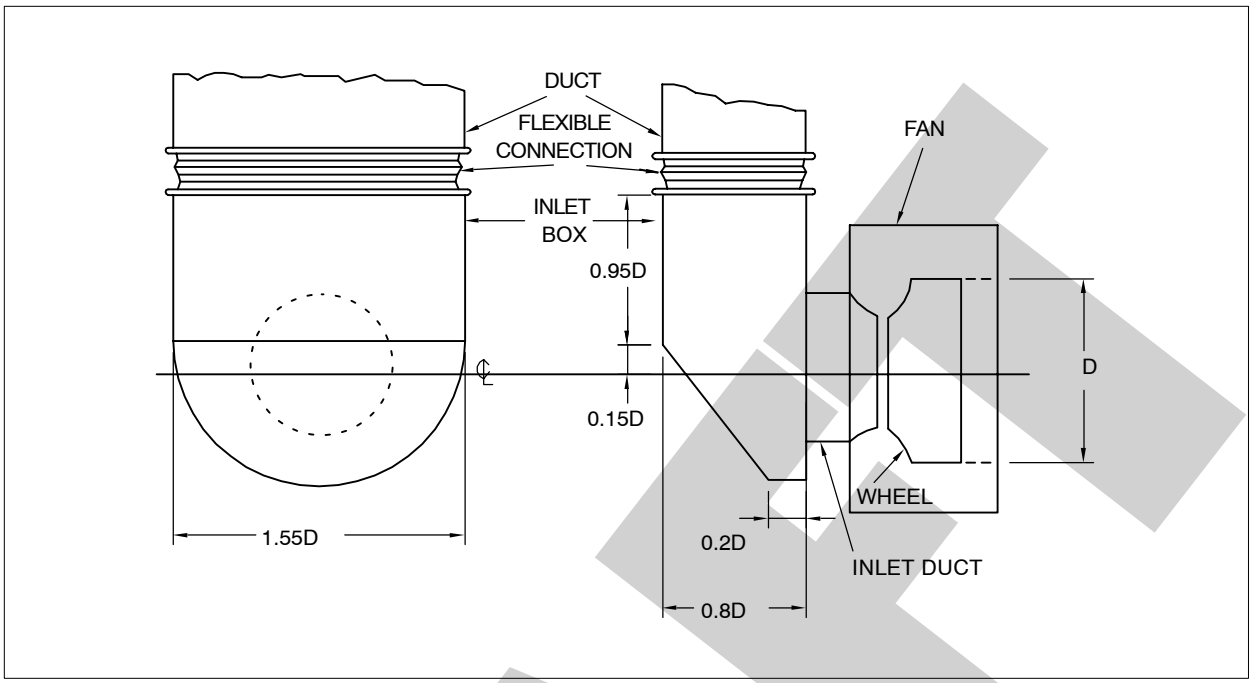


**FIGURE 6–18 CENTRIFUGAL FAN INLET BOX**

It is desirable that a drive guard located in this position should be furnished with as much opening as possible, to allow maximum airflow to the fan inlet. However, the guard design must comply with any Occupational Health and Safety Act requirements or any other applicable codes.

If available, use the fan manufacturer's allowance for drive guards obstructing the fan inlet. System Effect Curves for drive-guard obstructions situated at the inlet of a fan may be approximated using Figures 6-19, 6-20, and Table 6-3.

Where possible, open construction guards are recommended to allow free air passage to the inlet. Guards and sheaves should be designed to obstruct as little of the inlet as possible, in no case should the obstruction be more than ⅓ of the inlet area.

### 6.4.3 Belt Tube in Axial Fans

With a belt-driven axial-flow fan it is usually necessary for the fan motor to be mounted outside the fan housing.

To protect the belts from the airstream and to also prevent any leakage from the fan housing, manufacturers usually provide a belt tube.

Most manufacturers include the effects of this belt tube in their rating tables. However, where this is not reflected, the appropriate System Effect Curves obtained from Table 6-3 may be used.

### 6.4.4 Factory–Made Inlet Boxes

The System Effect of fan inlet boxes can vary widely, depending upon the design. This data should be available from the fan manufacturer. In the absence of fan manufacturer's data, a well-designed inlet box should approximate System Effect Curves "S" or "T" of Figure 6-1.

Inlet box dampers may be used to control the airflow volume through the system. Either parallel- or opposed-blade types may be used.

The parallel-blade type is installed with the blades parallel to the fan shaft, so when they are in a partially-closed position, a forced inlet vortex will be generated. The effect on the fan characteristics will be similar to that of inlet vane control.

The opposed-blade type is used to control airflow volume by using the addition of the pressure loss created by the damper in a partially-closed position.

If possible, complete data should be obtained from the fan manufacturer giving the System Effect or pressure loss of the inlet box and damper over the range of application. If data is not available, System Effect Curves "S" and "T" from Figure 6-1 should be applied in making the fan selection.

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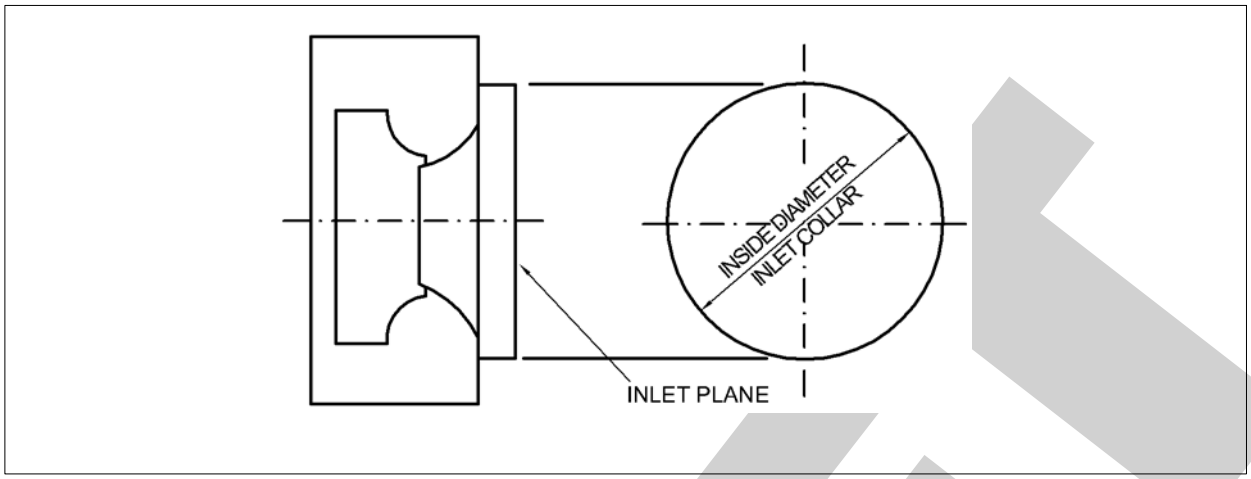


**FIGURE 6–19 FREE INLET AREA PLANE – FAN WITH INLET COLLAR**

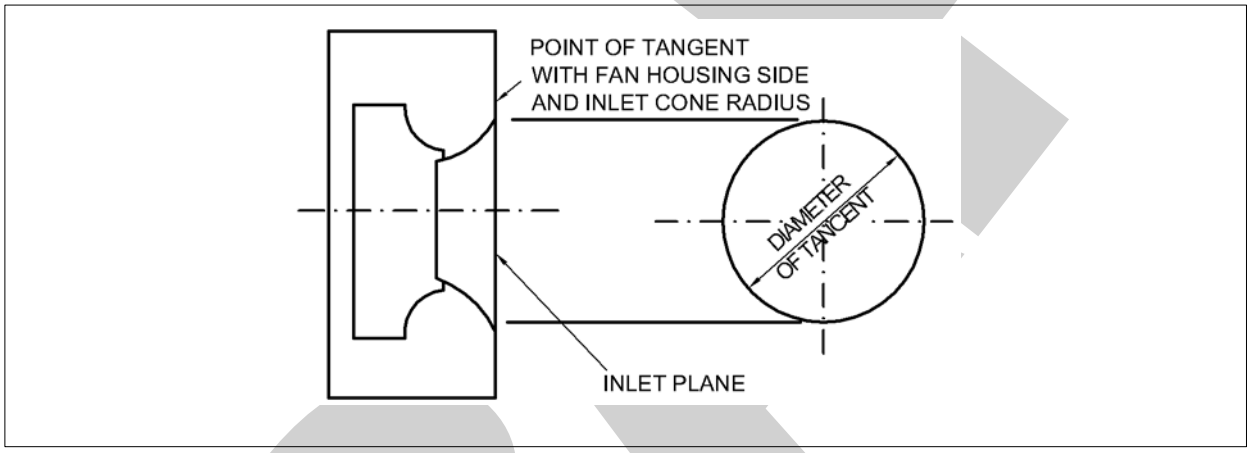


**FIGURE 6–20 FREE INLET AREA PLANE – FAN WITHOUT INLET COLLAR**

| Percentage of Unobstructed Inlet Area | System Effect Curve (Figure 6-1) |
|---|---|
| 100 | No Loss |
| 95 | V |
| 90 | U |
| 85 | T |
| 75 | S |
| 50 | Q |
| 25 | P |

**Table 6–3 System Effect Curves for Inlet Obstructions**

### 6.4.5 Inlet Vane Control

To maintain fan efficiency at reduced-flow conditions, the airflow quantity is often controlled by variable vanes mounted in the fan inlet, *see* Figure 6-21.

These are arranged to generate a forced-inlet vortex that rotates in the same direction as the fan impeller.

Inlet vanes may be two different basic types:

a. Integral (built-in)

b. Cylindrical (add on)

The System Effect of a wide-open inlet vane must be accounted for in the original fan selection. This data should be available from the fan manufacturer. If not available, the System Effect Curves of Table 6-4 can be applied in making the fan selection using Figure 6-23.

## 6.5 CALCULATING SYSTEM EFFECT

The HVAC system designer is responsible for the layout of the equipment room and the equipment/duct-connection configuration. Therefore, System Effect

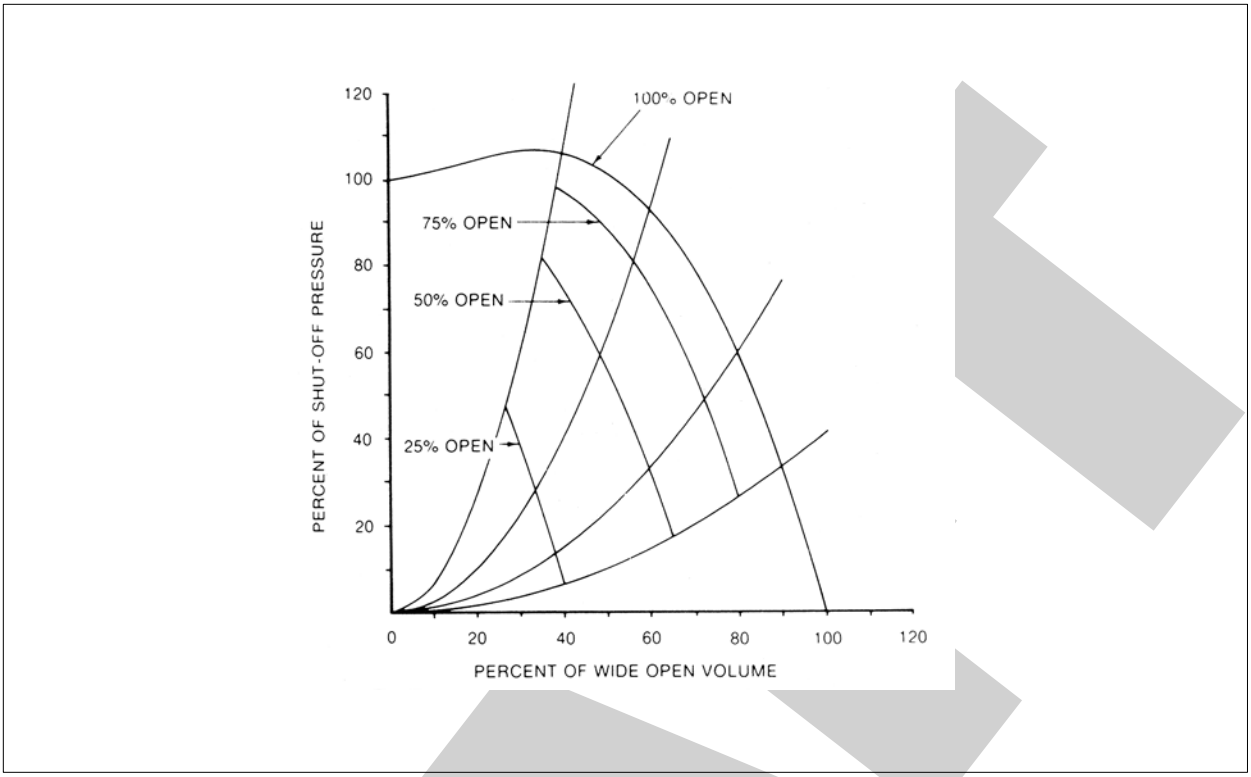


**FIGURE 6–21 TYPICAL NORMALIZED INLET VALVE CONTROL PRESSURE – VOLUME CURVE**

Factors can be noted and included in the system total pressure loss and fan capacity calculations.

Using a fan similar to that in the duct system example in Figures 7-2 or 7-7, the fan is in a plenum having adequate clearance for air entry to the fan inlet. However, the fan contains integral inlet vanes. With the blades wide open (Table 6-4), System Effect Curve "Q" will be used in Figure 6-1 to determine the static pressure loss.

The manufacturer's literature indicates that the selected 48 in. (1220 mm) SWSI fan has an inlet and outlet area of 13.1 $ft^2$ (1.22 $m^2$) each. At 20,000 cfm (10,000 L/s) and 2.4 in. wg (600 Pa) static pressure, the velocities are 1527 fpm (7.76 m/s). From Figure 6-1, reading up from 1527 fpm (7.76 m/s) to the "Q" curve gives a System Effect Factor of 0.23 in. wg (57 Pa) for the inlet side of the fan. This becomes part of the static pressure de-rating of the fan.

| Vane Type | System Effect Curve |
|---|---|
| 1. Integral (built-in) | "Q" or "R" |
| 2. Cylindrical (add-on) | "S" |

**Table 6–4**

The fan discharge size for this example is 43 (1092 mm) wide × 44 in. (1,118 mm) high, and the blast area ratio is 0.8. The 1.5 R/W elbow (the duct size is the same as the fan discharge size) is located 30 in. (760 mm) from the fan discharge, which would result in an approximately "25 percent effective duct" in position A, *see* Figures 6-2 and 6-3. From Table 6-2 the system Effect Factor Curve "T" or "U" is selected to be used in Figure 6-1. At 1527 fpm (7.76 m/s), both curves are off the graph, so no System Effect Factor would be added for the discharge side of the fan. Therefore, the fan would be rated at 2.17 in. wg (2.4 – 0.23) or 543 Pa (600 – 57) static pressure.

In many cases, a duct transition is used at the fan discharge connection, normally made with a flexible connection. The velocity in the duct has no relationship with the fan discharge velocity, unless it falls within the parameters discussed earlier in 6.2.1 Outlet Ducts.

It is important to note that System Effect cannot be measured in the field by testing and balancing technicians. Therefore, the system designer should deduct System Effect from the fan capacity, rather than adding it to the total pressure loss of the HVAC system.

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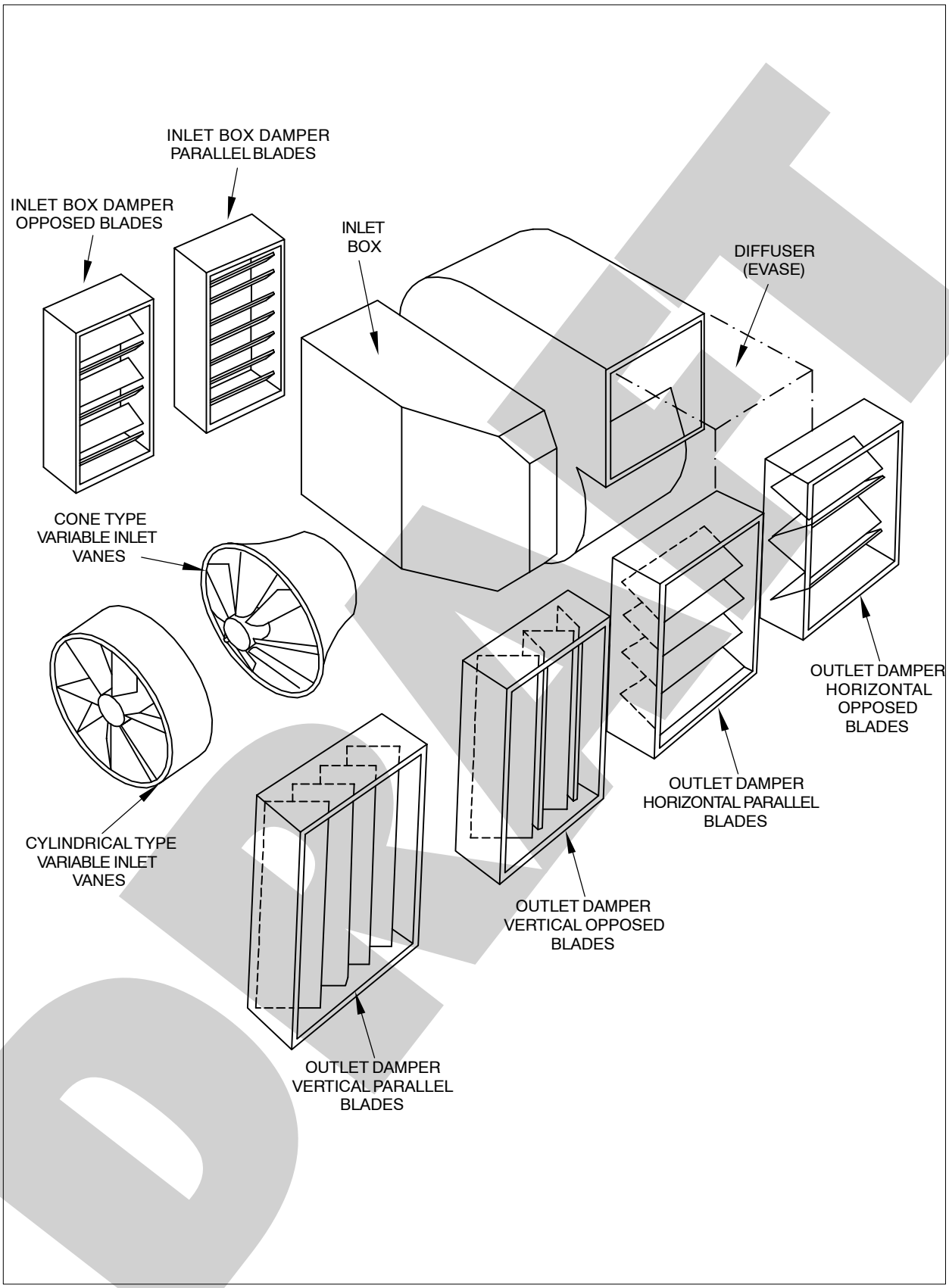


**FIGURE 6–22 COMMON TERMINOLOGY FOR CENTRIFUGAL FAN APPURTRENANCES**

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CHAPTER 7

# DUCT SIZING PROCEDURES

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## 7.1 SCOPE

The total pressure (TP) at any location within a system is the sum of the static pressure (SP) and the velocity pressure ($V_p$). Total pressure always decreases algebraically in the direction of airflow. Negative values of return or exhaust air systems increase in the direction of airflow, positive values of supply air systems decrease in the direction of airflow. However, the losses in total pressure between the fan and the end of each branch of a system are the same.

It should be noted that static pressure and velocity pressure are mutually convertible, either can increase or decrease in the direction of flow.

## 7.2 DESIGN OBJECTIVES

The following design objectives should be the basic goals of the duct system designer:

a. Design the duct system to convey the design airflow from the fan to the terminal devices in the most efficient manner, as allowed by the building structure.

b. Consider energy conservation in the fan selection, duct configuration, and duct wall heat gain or loss.

c. Special consideration should be given to sound attenuation and breakout noise.

d. Testing, adjusting, and balancing devices and dampers should be clearly indicated on the drawings.

e. Locations, size, and type of all life-safety devices, such as fire and smoke dampers, should be shown on the mechanical drawings.

f. The designer should consider the pressure losses that occur from all duct obstructions.

g. If the ductwork is well designed and constructed, at least 75 to 90 percent of the original velocity pressure can be regained.

h. Round ducts generally are preferred for higher-pressure systems.

i. Branch takeoffs, and fittings with low loss coefficients, should be used. Both 90 degree and 45 degree duct takeoffs can be used. However, the use of conical tees or angular takeoffs can reduce pressure losses.

j. Use of the SMACNA Duct Design Calculators can aid the duct design process, especially when considering aspect ratio options or making changes in the field.

## 7.3 DUCT SYSTEM SIZING PROCEDURES

### 7.3.1 Equal–Friction Method

The "equal-friction" method of duct sizing has probably been the most universally used means of sizing low-pressure supply air, return air, and exhaust air systems.

It is not normally used for sizing high-pressure systems. This design method "automatically" reduces air velocities in the direction of airflow. By using a reasonable initial velocity, the chances of introducing airflow-generated noise from high velocities are reduced or eliminated. When noise is an important consideration, the system velocity can be checked at any point. There is the opportunity to reduce velocity-created noise, by increasing duct size or adding sound-attenuation duct lining.

The major disadvantages of the equal-friction method are: (1) no natural provision for equalizing pressure drops in the branches, except a symmetrical layout; and (2) no provision for providing the same static pressure behind each supply or return terminal device. Balancing can be difficult in short duct runs, even with a considerable amount of damper adjustment. However, the equal-friction method can be modified by designing portions of the longest run with different friction rates than used for the short or long branches.

Static regain (or loss) due to velocity changes has been added to the equal-friction design procedure by using fitting loss coefficient tables in the Appendix. The omission of system static regain when using older tables could cause the calculated system fan static pressure to be greater than actual field conditions, particularly in larger, more complicated systems. Therefore, the "modified-equal-friction" low-pressure duct design procedure presented in this subsection will combine the advantages of several design methods, when used with the loss coefficient tables in the Appendix.

### 7.3.2 Modified Equal Friction Design Procedure

"Equal friction" does *not* mean that total friction remains constant throughout the system. It means that a *specific* friction loss, or static pressure loss per 100

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equivalent ft (m) of duct, is selected before the ductwork is laid out. That loss per 100 ft (Pa/m) must be used constantly throughout the design. The figure used for this "constant" is entirely dependent upon the experience and desire of the designer but there are practical limits based on economy and the allowable velocity range required to maintain the low-pressure system status.

To size the main supply duct leaving the fan, the usual procedure is to select an initial velocity from the chart in Figure A-1. *NOTE: The shaded area represents the recommended velocity range.* This velocity can be selected above the shaded section of Figure A-1, if higher sound levels and fan energy conservation are not limiting factors. The chart in Figure A-1 is used to determine the friction loss by using the design air quantity (cfm)(L/s), and the selected velocity (fpm)(m/s). A friction loss value commonly used for lower pressure duct sizing is 0.1 in. (in. wg) per 100 equivalent ft of ductwork or 0.8 to 1.0 Pa/m. However, other values both lower and higher are used by some designers as their "standard" or for special applications. This same friction loss "value" generally is maintained throughout the design and the respective round duct diameters are obtained from the chart in Figure A-1.

The friction losses for each duct section should be corrected for other materials and construction methods by use of Table A-1 and Figure A-2. The correction factor from Figure A-2 is applied to the duct friction loss for the straight sections of the duct, prior to determining the round duct diameters. The round duct diameters thus determined can then be used to select the equivalent rectangular duct sizes from Table A-1M, unless round ductwork is to be used.

The flow rate in the second section of the main supply duct, after the first branch takeoff, is the original airflow volume supplied by the fan, reduced by the amount of volume into the first branch. Using Figure A-1, the new flow rate value using the recommended friction rate of 0.1 in. wg / 100 ft (0.8 to 1.0 Pa/m) will determine the duct velocity and diameter for that section. The equivalent rectangular size of that duct section again is obtained from Table A-1M, if needed. All subsequent sections of the main supply duct, and all branch ducts, can be sized from Figure A-1 using the same friction loss rate and procedures.

The total pressure drop measured at each terminal device, air outlet or inlet, of a small duct system, or of branch ducts of a larger system, should not differ more than 0.05 in. wg (12.0 Pa.) If the pressure difference between the terminals exceeds this amount, dampering would be required, which can create objectionable air-noise levels.

The modified equal friction method is used for sizing duct systems that are not symmetrical, or have both long and short runs. Instead of depending upon volume dampers to artificially increase the pressure drop of short branch runs, the branch ducts are sized to dissipate (bleed-off) the available pressure, by using higher duct-friction loss values. Only the main duct, which usually is the longest run, is sized by the original duct friction loss value.

Care should be exercised to prevent excessively high velocities in the short branches, with the higher friction rates. If calculated velocities are found to be too high, then duct sizes must be recalculated to yield lower velocities and opposed-blade volume dampers or static-pressure plates must be installed in the branch duct, at or near, the main duct to dissipate the excess pressure. Regardless, it is good design practice to include balancing dampers in HVAC duct systems to balance the airflow to each branch.

## 7.4 FITTING PRESSURE LOSS TABLES

Tables A-7 to A-15 contains the loss coefficients for elbows, fittings, and duct components. The "loss coefficient," represents the ratio of the total pressure loss to the dynamic pressure in terms of velocity pressure. It does not include duct friction loss, which is determined by measuring the duct sections to fitting center lines. However, the loss coefficient does include static regain (or loss), where there is a change in velocity.

**Equation 7–1(I–P)**

$$TP = C \times V_p$$

Where:

$TP$ = Total Pressure (in. wg)

$C$ = Dimensionless Loss Coefficient

$V_p$ = Velocity Pressure (in. wg)

**Equation 7–1(SI)**

$$TP = C \times V_p$$

Where:

$TP$ = Total Pressure (Pa)

$C$ = Dimensionless Loss Coefficient

$V_p$ = Velocity Pressure (Pa)

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By using the duct fitting loss coefficients in the Appendix, which include static pressure, regain, or loss, accurate duct-fitting pressure losses are obtained. When combined with the static- pressure friction losses sized by the *modified equal friction method*, the result will be the closest possible approximation of the actual system total pressure requirements for the fan.

To demonstrate the use of the loss coefficient tables, several fittings are selected from a sample duct system that has a velocity of 2550 fpm (12.7m/s). Using Table A-4, the velocity pressure ($V_p$) is 0.41 in. wg (100 Pa). The total pressure (TP) loss of each fitting is determined as follows:

*Example 7-1 (I-P)*

36 (H) × 12 in. (W), 90 degree Radius Elbow ($\frac{R}{W}$ = 1.5), no vanes. From Table A-7, Figure F, the loss coefficient of 0.14 is obtained using $\frac{H}{W}$ = 3.0.

The loss coefficient should not be used without checking to see if a correction is required for the Reynolds number:

$$D = \frac{2HW}{H + W} = \frac{2 \times 36 \text{ in.} \times 12 \text{ in.}}{36 \text{ in.} + 12 \text{ in.}} = 18 \text{ in.}$$

$$R_e = 8.56 \times DV$$

$$R_e = 8.56 \times 18 \times 2550 = 392,904$$

$$R_e 10^{-4} = \frac{392,904}{10^4} = 39.29$$

The correction factor of 1.0 is found where $\frac{R}{W} > 0.75$ and $R_e\ 10^{-4} > 20$. So the loss coefficient remains at 0.14. Then:

$$TP = C \times V_p = 0.14 \times 0.41 = 0.057 \text{ in. wg}$$

All of the above calculations for $R_e 10^{-4}$ could have been avoided, if the graph in the *Reynolds Number Correction-Factor Chart* on Page A.15 had been checked. This plotted point is outside the shaded area requiring correction, when using the duct diameter and velocity to plot the point.

If the elbow were 45 degree instead of 90 degree, another correction factor of 0.60 *(see* the reference to Note 1 on page A.15) would be used:

$$0.60 \times 0.057 = 0.034 \text{ in.wg}$$

*Example 7-1 (SI)*

900 (H) × 300 mm (W), 90 degree Radius Elbow ($\frac{R}{W}$ = 1.5), no vanes. From Table A-7, Figure F, the loss coefficient of 0.14 is obtained using $\frac{H}{W}$ = 3.0.

The loss coefficient should not be used without checking to see if a correction is required for the Reynolds number:

$$D = \frac{2HW}{H + W} = \frac{2 \times 900 \text{ mm} \times 300 \text{ mm}}{900 \text{ mm} + 300 \text{ mm}} = 450 \text{ mm}$$

$$R_e = 66.4 \times DV$$

$$R_e = 66.4 \times 450 \times 13 = 388,440$$

$$R_e 10^{-4} = \frac{388,440}{10^4} = 38.84$$

The correction factor of 1.0 is found where $\frac{R}{W} > 0.75$ and $R_e\ 10^{-4} > 20$. So the loss coefficient remains at 0.14. Then:

$$TP = C \times V_p = 0.14 \times 100 = 14 \text{ Pa}$$

All of the above calculations for $R_e 10^{-4}$ could have been avoided, if the graph in the *Reynolds Number Correction-Factor Chart* on Page A-15 had been checked. This plotted point is outside the shaded area requiring correction using the duct diameter and velocity to plot the point.

If the elbow was 45 degrees instead of 90 degrees, another correction factor of 0.60 *(see* the reference to Note 1 on page A-15) would be used: 0.60  14 = 8.4 Pa.

*Example 7-2 (I-P)*

45 degree Round Wye, 20 in. diameter main duct, (2500 fpm); 10 in. diameter branch duct, branch velocity of 1550 fpm. Determine the fitting pressure losses. (Figure A of Table A-11.)

$$A_b = \pi r^2 = \pi(5)^2 = \pi\ 25$$

$$A_c = \pi^2 = \pi 10^2 = \pi\ 100$$

$$\frac{A_b}{A_c} = \frac{25}{100} = 0.25$$

From Figure A-1:

For 10 in. diameter, 1550 fpm; $Q_b$ = 850 cfm

For 20 in. diameter, 2500 fpm; $Q_c$ = 5,500 cfm

$$\frac{Q_b}{Q_c} = \frac{850}{5500} = 0.155$$

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Interpolating in the table between $\frac{A_b}{A_c} = 0.2$ and 0.3; and $\frac{Q_b}{Q_c} = 0.1$ and 0.2; indicates 0.56 is the branch fitting loss coefficient. The branch pressure loss is calculated.

Obtain $V_p$ of 0.39 for 2500 fpm from Table A-5.

$$TP = C \times Vp = 0.56 \times 0.39 = 0.218 \text{ in. wg}$$

The main pressure loss is calculated by first establishing $V_s$:

$$Q_s = Q_c - Q_b = 5500 - 850 = 4650 \text{ cfm}$$

Using Figure A-11, 20 in. diameter:

$$V_s = 2120 \text{ fpm}$$

$$\frac{V_s}{V_c} = \frac{2120}{2500} = 0.85$$

From the Table A-11, Figure A, C = 0.02

$$TP = C \times V_p = 0.02 \times 0.39 = 0.008 \text{ in. wg}$$

*Example 7-2 (SI)*

45 degree Round Wye, 500 mm diameter main duct, (12.5 m/s); 250 mm diameter branch duct, branch velocity 0f 7.7 m/s. Determine the fitting pressure losses. (Figure A of Table A-11.)

$$A_b = \pi r^2 = \pi(125)^2 = \pi\ 15,625$$

$$A_c = \pi r^2 = \pi(250)^2 = \pi\ 62,500$$

$$\frac{A_b}{A_c} = \frac{15,625}{62,500} = 0.25$$

From Figure A-1M:

For 250 mm diameter, 7.7 m/s; $Q_b$ =370 L/s

For 500 mm diameter, 12.5 m/s; $Q_c$ = 2400 L/s

$$\frac{Q_b}{Q_c} = \frac{370}{2400} = 0.154$$

Interpolating in the table between $\frac{A_b}{A_c} = 0.2$ and 0.3; and $\frac{Q_b}{Q_c} = 0.1$ and 0.2; indicates 0.56 is the branch fitting loss coefficient. The branch pressure loss can now be calculated.

Obtain $V_p$ of 92.5 for 12.5 m/s from Table A-5M or by using Equation 5-8 (SI).

$$TP = C \times V_p = 0.56 \times 92.5 = 51.8 \text{ Pa}$$
(Equation 5-6.)

The main pressure loss is calculated by first establishing $V_s$:

$$Q_s = Q_c - Q_b = 2400 - 370 = 2030 \text{ L/s}$$
(downstream airflow.)

From Figure A-1M, 500 mm diameter and 2030 L/s:

$$V_s = 10.5 \text{ m/s}$$

$$\frac{V_s}{V_c} = \frac{10.5}{12.5} = 0.84$$

From the Table A-11, Figure A, C = 0.02

$$TP = C \times V_p = 0.02 \times 92.5 = 1.85 \text{ Pa}$$

*Example 7-3 (I-P)*

36 × 12 in. rectangular to 20 in. diameter round transition where θ = 30 degree (Table A-9, Figure A), $V_p$ = 0.4.

$$A_1 = 36 \times 12 = 432 \text{ in.}^2$$

$$A = \pi r^2 = \pi 10^2 = 314 \text{ in.}^2$$

$$A_1 = A = 1.38 \text{ (use 2)}$$

0.05 is selected as the loss coefficient.

$$TP = C \times V_p = 0.05 \times 0.4 = 0.02 \text{ in. wg}$$

*Example 7-3 (SI)*

900 × 300 mm rectangular to 500 mm diameter round transition where θ = 30 degrees (Table A-9, Figure A), $V_p$ = 100 Pa.

$$A_1 = 900 \times 300 = 270,000 \text{ m}^2$$

$$A = \pi r^2 = \pi(250)^2 = 196,350 \text{ m}^2$$

$$\frac{A_1}{A} = \frac{270,000}{196,350} = 1.37 \text{ (use 2)}$$

0.05 is selected as the loss coefficient.

$$TP = C \times V_p = 0.05 \times 100 = 5 \text{ Pa}$$

## 7.5 SUPPLY AIR DUCT SYSTEM–SIZING EXAMPLE 1 (I–P)

The following duct sizing example and worksheet use only I-P Units. Please skip to Section 7-9 for the same example and worksheet using SI Units.

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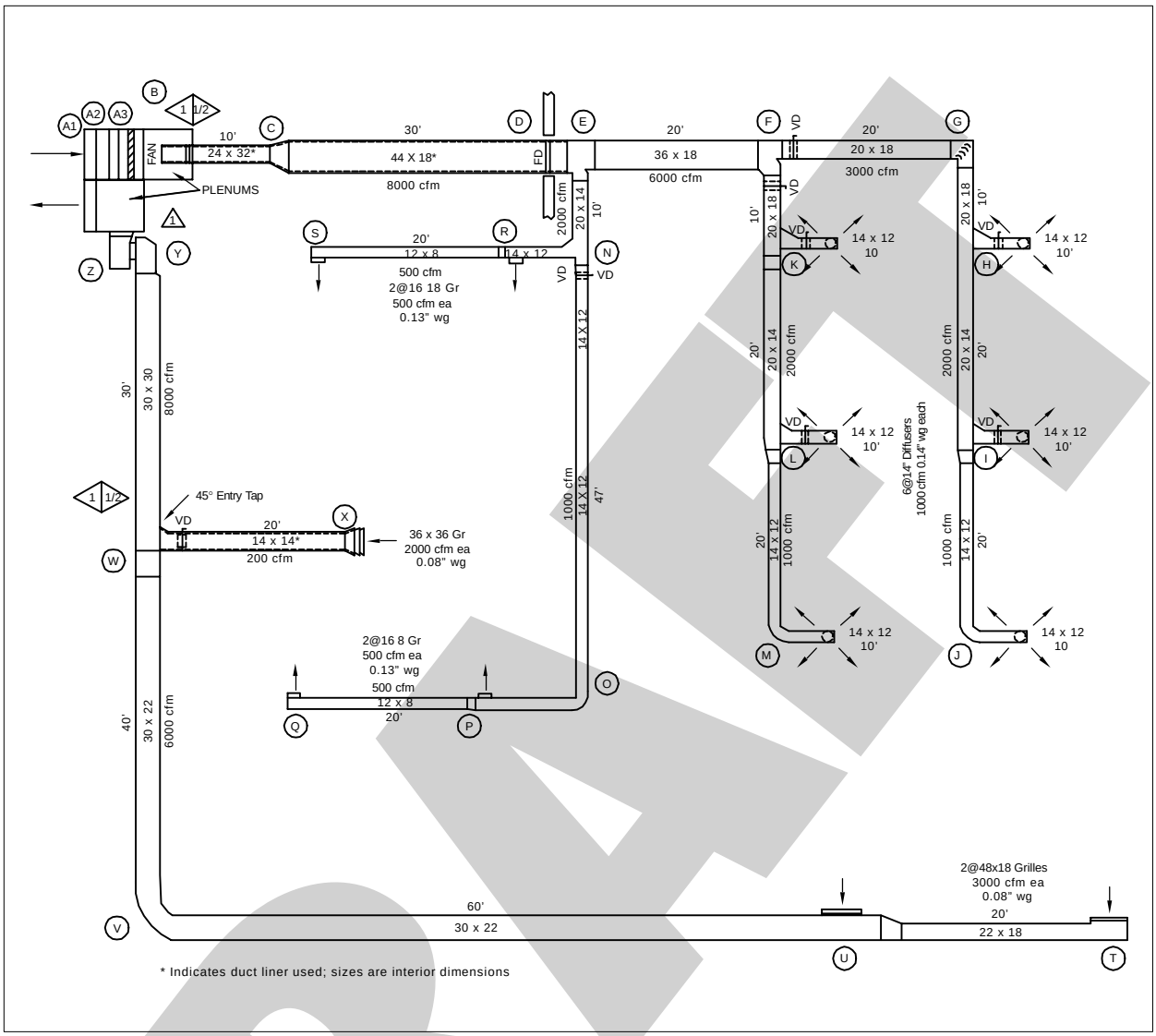


**FIGURE 7–1 DUCT SYSTEMS FOR DUCT SIZING EXAMPLES 1 AND 2**

A plan of a sample building HVAC duct system is shown in Figure 7-1 and the tabulation of the computations can be found in Table 7-1. A full size "Duct Sizing Work Sheet" may be found in Figure 7-5. It may be photocopied for "in-house" use only. The conditioned area is assumed to be at zero pressure and the two fans have been sized to deliver 8000 cfm each. The grilles and diffusers have been tentatively sized to provide the required flow, throw, and noise level with the sizes and pressure drops are indicated on the plan. To size the ductwork and determine the supply-fan total pressure requirement, a suggested step-by-step procedure follows.

### 7.5.1 Supply Fan Plenum

From manufacturer's data sheets or from the Figures or Tables in Chapter 8, the static pressure losses of the energy recovery device, filter bank, and heating-cooling coil are entered in Table 7-1 in column L. Velocities, if available, are entered in column F for reference information only. With 10 ft of duct discharging directly from fan "B" (duct is fan outlet size), no System-Effect Factor needs to be added for either side of the fan. As the plenum static pressure (SP) loss is negligible, the losses for the inlet-air portion of the fan system entered in column L are added, the loss of 0.90 in. wg is entered in column M on line 3.

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DUCT SIZING WORK SHEET
(U.S. UNITS)

DATE _____ PAGE 1 OF 2

PROJECT SAMPLE BUILDING LOCATION FIRST FLOOR SYSTEM SUPPLY AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW CFM | FRIC-TION PER 100′ | VELOC-ITY FPM | $V_p$ | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMU-LATIVE LOSS | TOTAL LOSS |
| 1 | PLENUM B | $A_1$ | – | H.R. DEVICE | 8000 | – | 500 | – | – | – | – | – | 0.30 | | 1.382 | 1.382 |
| 2 | ” | $A_2$ | – | FILTERS | 8000 | – | 400 | – | – | – | – | – | 0.15 | | | ($A_1$J) |
| 3 | ” | $A_3$ | – | COIL | 8000 | – | 550 | – | – | – | – | – | 0.45 | 0.90 | | |
| 4 | RUN BJ | BC | 10′ | DUCT | 8000 | 0.095 | 1600 | – | – | 30.2 | * 24 × 32 | 1.40 | 0.013 | 0.013 | 0.492 | |
| 5 | ” | CE | 30′ | DUCT | 8000 | 0.095 | 1600 | – | – | 30.2 | * 44 × 18 | 1.40 | 0.040 | | 0.479 | |
| 6 | ” | C | – | TRANS. | 8000 | – | 1500 | 0.14 | 0.25 | – | 24 × 32 / 44 × 18 | – | 0.035 | | | |
| 7 | ” | D | – | F.DAMP. | 8000 | – | – | – | – | – | 44 × 18 | – | 0.06 | 0.135 | | |
| 8 | ” | EF | 20′ | DUCT | 6000 | 0.095 | 1500 | – | – | 27.4 | 36 × 18 | – | 0.019 | | 0.344 | |
| 9 | ” | E | 90° | WYE | 8000 / 6000 | – | 1455 | 0.13 | –0.01 | – | 44 × 18 / 36 × 18 | – | –0.001 | 0.018 | | |
| 10 | ” | FH | 30′ | DUCT | 3000 | 0.095 | 1260 | – | – | 20.7 | 20 × 18 | – | 0.029 | | 0.326 | |
| 11 | ” | F | 90° | WYE | 6000 / 3000 | – | 1333 | 0.11 | 0.05 | – | 36 × 18 | – | 0.006 | | | |
| 12 | ” | F | – | VOL DAMP | 3000 | – | 1200 | 0.09 | 0.04 | – | 20 × 18 | – | 0.004 | | | |
| 13 | ” | G | 90° | ELBOW | 3000 | – | 1200 | 0.09 | 0.24 | – | 20 × 18 | – | 0.022 | 0.061 | | |
| 14 | ” | HI | 20′ | DUCT | 2000 | 0.095 | 1140 | – | – | 18.2 | 20 × 14 | – | 0.019 | | 0.265 | |
| 15 | ” | H | – | TRANS. | 3000 / 2000 | – | 1200 | 0.09 | 0.05 | – | 20 × 18 / 20 × 14 | – | 0.005 | 0.024 | | |
| 16 | ” | IJ | 30′ | DUCT | 1000 | 0.080 | 900 | – | – | 14.2 | 14 × 12 | – | 0.024 | | 0.241 | |
| 17 | ” | I | – | TRANS | 2000 / 1000 | – | 1029 | 0.07 | 0.05 | – | 20 × 14 / 14 × 12 | – | 0.004 | | | |
| 18 | ” | J | 90° | ELBOW | 1000 | – | 857 | 0.05 | 0.16 | 14.2 | 14 × 12 | 1.32 | 0.011 | | | |
| 19 | ” | J | – | VOL DAMP | 1000 | – | 857 | 0.05 | 0.04 | – | 14 × 12 | – | 0.002 | | | |
| 20 | ” | J | – | TAP FIT. | 1000 | – | 857 | 0.05 | 1.20 | – | 14 × 12 / 140 | – | 0.060 | | | |
| 21 | ” | J | – | DIFF. | 1000 | – | – | – | – | – | 140 | – | 0.140 | 0.241 | | |
| 22 | | | | | | | | | | | | | | | | |
| 23 | | | | | | | | | | | | | | | | |
| 24 | | | | | | | | | | | | | | | | |
| 25 | | | | | | | | | | | | | | | | |

**Table 7–1 Duct Sizing, Supply Air System – Example 1**

NOTE: * Indicates duct lining used. Sizes are interior dimensions.



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DUCT SIZING WORK SHEET
(U.S. UNITS)

DATE _____ PAGE 2 OF 2

PROJECT SAMPLE BUILDING LOCATION FIRST FLOOR SYSTEM SUPPLY AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW CFM | FRIC-TION PER 100′ | VELOC-ITY FPM | $V_p$ | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMU-LATIVE LOSS | TOTAL LOSS |
| 1 | $A_1F$ | – | – | TOTAL OF LINES 1–9 (SHEET 1 OF 2) | | | | | | | | – | – | 1.067 | 1.403 | 1.403 |
| 2 | RUN FM | F | 90° | WYE | 6000 / 3000 | – | 1333 | 0.11 | 0.52 | – | 36 × 18 / 20 × 18 | – | 0.057 | | 00336 | ($A_1M$) |
| 3 | ” | F | | VOL DAMP | 3000 | – | 1200 | 0.09 | 0.04 | – | 20 × 18 | – | 0.004 | | | |
| 4 | ” | FK | 10′ | DUCT | 3000 | 0.095 | 1260 | – | – | 20.7 | 20 × 18 | – | 0.010 | 0.071 | | |
| 5 | ” | KM | – | TOTAL OF LINES 1– 7 (SHEET 1 OF 2) | | | | | | | | – | – | 0.265 | 0.265 | |
| 6 | | | | | | | | | | | | – | | | | |
| 7 | $A_1E$ | – | – | TOTAL OF LINES 1– 7 (SHEET 1 OF 2) | | | | | | | | – | – | 1.049 | 1.367 | 1.367 |
| 8 | RUN EQ | EN | 10′ | DUCT | 2000 | 0.095 | 1140 | – | – | 18.2 | 20 × 14 | – | 0.010 | | 0.318 | ($A_1Q$) |
| 9 | ” | E | 90° | WYE | 8000 / 2000 | – | 1455 | 0.13 | 0.43 | – | 44 × 18 / 20 × 14 | – | 0.056 | 0.066 | | |
| 10 | ” | NP | 55′ | DUCT | 1000 | 0.080 | 900 | – | – | 14.2 | 14 × 12 | – | 0.044 | | 0.252 | |
| 11 | ” | N | – | TRANS | 2000 / 1000 | – | 1029 | 0.07 | 0.05 | – | 20 × 14 / 14 × 12 | – | 0.004 | | | |
| 12 | ” | N | – | VOL DAMP | 1000 | – | 857 | 0.05 | 0.04 | – | 14 × 12 | – | 0.002 | | | |
| 13 | ” | O | 90° | ELBOW | 1000 | – | 857 | 0.05 | 0.12 | – | 14 × 12 | – | 0.006 | 0.056 | | |
| 14 | ” | PQ | 20′ | DUCT | 500 | 0.095 | 810 | – | – | 10.7 | 12 × 8 | – | 0.019 | | 0.196 | |
| 15 | ” | P | – | TRANS | 1000 / 500 | – | 857 | 0.05 | 0.06 | – | 14 × 12 / 12 × 8 | – | 0.003 | | | |
| 16 | ” | Q | 90° | ELBOW | 500 | – | 750 | 0.04 | 1.00 | – | 12 × 8 | 1.09 | 0.044 | | | |
| 17 | ” | Q | – | GRILLE | 500 | – | – | – | – | – | 16 × 8 | – | 0.130 | 0.196 | | |
| 18 | $A_1N$ | – | – | TOTAL OF LINES 7–9 (ABOVE) | | | | | | | | – | – | 1.115 | 1.371 | 1.371 |
| 19 | RUN NS | NR | 8′ | DUCT | 1000 | 0.080 | 900 | – | – | 14.2 | 14 × 12 | – | 0.006 | | 0.256 | ($A_1S$) |
| 20 | ” | N | 45° | ENT TAP | 2000 / 1000 | – | 1029 | 0.07 | 0.07 | – | 20 × 14 / 14 × 12 | – | 0.052 | | | |
| 21 | ” | N | – | VOL DAMP | 1000 | – | 857 | 0.05 | 0.05 | – | 14 × 12 | – | 0.002 | 0.060 | | |
| 22 | ” | RS | 20′ | DUCT | 500 | 0.095 | 810 | – | – | 10.7 | 12 × 8 | – | 0.019 | | 0.196 | |
| 23 | ” | R | – | TRANS | 1000 / 500 | – | 857 | 0.05 | 0.05 | – | 14 × 12 / 12 × 8 | – | 0.003 | | | |
| 24 | ” | S | 90° | ELBOW | 500 | – | 750 | 0.04 | 0.04 | – | 12 × 8 | 1.09 | 0.044 | | | |
| 25 | ” | S | – | GRILLE | 500 | – | – | – | – | – | 16 × 8 | – | 0.130 | 0.196 | | |

**Table 7–1 (a) Duct Sizing, Supply Air System – Example 1 (Continued)**

NOTE: * Indicates duct lining used. Sizes are interior dimensions.

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### 7.5.2 Supply Air System

#### 7.5.2.1 Duct Section BC

The 24 × 32 in. fan discharge size has a circular equivalent of 30.2 in., *see* Table A-2. Using the chart in Figure A-1, a velocity of 1600 fpm and a friction loss of 0.095 in. wg per 100 ft of duct are established within the recommended velocity range (shaded area) using the 8000 cfm system airflow. The data is entered on line 4 in the appropriate columns. Without any changes in direction to reduce fan noise, and with the duct located in an unconditioned space up to the first branch (at point E), internal fibrous-glass lining can be used to satisfy both the acoustic and thermal requirements. Therefore, the duct size entered in column J is marked with an asterisk and the fibrous-glass liner "medium rough" correction factor 1.40 is obtained from Table A-1 and Figure A-3 and entered in column K. Duct sections BC static pressure (SP) loss I computed as follows:

$$SP\ (\text{duct section}) = 10\ \text{ft (duct)} \times \frac{0.095\ \text{in. wg}}{100\ \text{ft}} \times 1.40$$

$$(\text{corr. factor}) = 0.013\ \text{in. wg}$$

The duct section BC static pressure loss is entered in column L and, as it is the only loss for that section, the loss also is entered in column M.

#### 7.5.2.2 Duct Section CE

At point C, building construction conditions require that the duct aspect ratio changes so a duct transition is needed. Using the same 0.095 in. wg per 100 ft duct friction loss and 30.2 inches duct diameter for the 8000 cfm airflow, a 44 × 18 in. duct is selected from Table A-2 and entered in column J on line 5. This section of duct continues to require acoustical and thermal treatment so the section friction loss is computed:

$$SP = 30\ \text{ft} \times \frac{0.095\ \text{in. wg}}{100\ \text{ft}} \times 1.40 = 0.040\ \text{in. wg}$$

(enter on line 5 in column L).

The transition loss coefficient can be obtained after determining if the fitting is diverging or converging.

$$A = 24 \times 32 = 768$$

$$A_1 = 44 \times 18 = 792$$

$$\frac{A_1}{A} = \frac{792}{768} = 1.03$$

The average velocity of the entering airstream (Equation 5-7) = $\frac{Q}{A}$

$$\text{cfm/Area (ft)} = \frac{8000}{24} \times \frac{32}{144} = 1500\ \text{fpm}$$

From Table A-11, Figure B, using $\theta = 30$ degree and $\frac{A_1}{A} = 2$ (smallest number for $\frac{A_1}{A}$), the loss coefficient of 0.25 is entered on line 6 in column H. The velocity pressure ($V_p$) of 0.14 in. wg is obtained from Table A-6 for 1500 fpm, and entered in column G. The transition fitting pressure loss of 0.035 in. wg ($C \times V_p = 0.25 \times 0.14$) is entered in column L. As this is a dynamic-pressure loss, the correction factor for the duct lining does not apply.

The static pressure loss of 0.06 in. wg for the fire damper at D is obtained from Chapter 8 or manufacturer's data sheets and entered in column L on line 7. The three static pressure losses in column L on lines 5, 6, and 7, are totaled (0.133 in. wg) and entered in column M on line 7. This is the total pressure loss of the 44 × 18 in. duct section CD (inside dimensions) and its components.

#### 7.5.2.3 Duct Section EF

An assumption must now be made as to which duct run has the greatest friction loss. As the duct run to the "J" air-supply diffuser is apparently the longest with the most fittings, this run will be the assumed path for further computations. Branch-duct run EQ will be compared with duct run EJ after calculations are completed.

Applying the 6000 cfm (for duct section EF) and 0.095 in. wg per 100 ft to the chart in Figure A-1, a duct diameter of 27.1 in. and 1500 fpm velocity are obtained and entered on line 8. Table A-2 is used to select a 36 × 18 in. rectangular duct size needed by keeping the duct height 18 in. (equivalent duct diameter = 27.4 in.).

Normally, duct size changes are made changing only one dimension for ease and economy of fabrication. The use of 27.4 in. instead of 27.1 in. does not change the velocity, velocity pressure, or duct friction losses significantly to require the use of different values. A review of the chart in Figure A-1 in the Appendix will verify this, so 1500 fpm and 0.095 in. wg will continue to be used.

As the galvanized duct system is being fabricated in 4 ft sections, the degree of roughness (Table A-1) indicates "medium smooth". No correction factor is needed; as the chart in Figure A-1 is based on an Absolute Roughness of 0.0003 ft.

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The static-pressure loss for duct section EF is:

$$SP = \frac{20 \text{ ft} \times 0.095}{100 \text{ ft}} = 0.019 \text{ in. wg}$$

(enter on line 8 column L).

The diverging 90 degree wye fitting used at E can be found in Table A-14, Figure W. In order to obtain the proper loss coefficient "C" to calculate the fitting pressure loss, preliminary calculations to obtain $A_b$ must be made. If a different friction loss rate is used later when computing the branch losses, subsequent recalculation might be necessary.

$A_b$ (Preliminary) for 2000 cfm @ 0.095 in. wg = 254 in.$^2$, area of 18.0 in. diameter duct obtained from Figure A-1. Then:

$$\frac{A_b}{A_s} = (9.0)^2\pi(13.7)^2\pi = \frac{254}{590} = 0.43$$

$$\frac{A_b}{A_c} = \frac{254}{707} = 0.36$$

$$\frac{Q_b}{Q_c} = \frac{2000}{8000} = 0.25$$

Using $\frac{A_b}{A_s} = 0.33$; and $\frac{A_b}{A_s} = 0.25$ (the closest figures), C (Main) = – 0.01 obtained by interpolation. The $V_p$ for 1455 fpm $\left(\frac{8000}{44} \times \frac{18}{144}\right)$ is 0.13 in. wg.

The fitting "loss" thus has a negative value (– 0.01 × 0.13 = – 0.001) and is entered on line 9 in column L with a minus sign. The static regain is actually greater than the dynamic-pressure loss of the fitting. The pressure losses on lines 8 and 9 in column L are added (– 0.001 + 0.019 = 0.018 in. wg) and entered on line 9 in column M.

### 7.5.2.4 Duct Section FH

The wye fitting at F and duct section FH are computed in the same way as above and the values entered on lines 10 and 11. By using 0.095 in. wg and 3000 cfm in Figure A-1, 1260 fpm and 20.7 in. diameter are obtained from Figure A-1; 20 × 18 in. equivalent duct size from Table A-2:

FH duct section loss =

$$\frac{30 \text{ ft} \times 0.095}{100 \text{ ft}} = 0.029 \text{ in. wg}$$

(enter on line 10).

For the wye fitting at F, Table A-14, Figure W is again used. With the 6000 cfm airflow dividing equally into two 3000 cfm airstream ducts, $A_b = A_s$. Therefore,

$$\frac{A_b}{A_s} = 1.0$$

$$\frac{A_b}{A_c} = \frac{(10.5)2\pi}{(13.7)2\pi} = \frac{346}{590} = 0.59$$

$$\frac{Q_b}{Q_c} = \frac{3000}{6000} = 0.5$$

Using $\frac{A_b}{A_s} = 1.0$ and $\frac{A_b}{A_c} = 0.5$

$C$ (Main) = 0.05

velocity = 1333 fpm $\left(\frac{6000}{36} \times \frac{18}{144}\right)$

$V_p$ = 0.11 (From Table A-6)

Fitting loss =

$$C \times V_p = 0.05 \times 0.11 = 0.006 \text{ in. wg (line11)}$$

The loss coefficient for the thin-plate volume damper near F can be obtained from Table A-18, Figure B, set wide open or 0 degree. The velocity pressure ($V_p$) of 0.09 in. wg for 1200 fpm $\left(\frac{3000}{20} \times \frac{18}{144}\right)$ is obtained from Table A-6.

Damper Loss =

$$C \times V_p = 0.04 \times 0.09 = 0.004 \text{ in. wg (line 12)}$$

Elbow G in the FH duct run is a square elbow with 4.5 in. single-thickness turning varies on 3.25 in. centers. The loss coefficient of 0.24 is obtained from Table A-10, Figure H for the 20 × 18 in. elbow, and entered on line 13 along with the other data (cfm, fpm, $V_p$)

G fitting loss =

$$C \times V_p = 0.24 \times 0.09 = 0.022 \text{ in. wg (line 13)}$$

The total pressure loss for duct section FH from lines 10, 11, 12, and 13 in column L (0.029 + 0.006 + 0.004 + 0.022) of 0.061 in. wg, is entered on line 13 in column M.

### 7.5.2.5 Duct Section HI

Data for duct section HI is developed as other duct sections above. Starting with 2000 cfm, the values of 1140 fpm, 18.0 in. diameter, and the duct size of 20 × 14 in. is obtained by changing only one duct dimension.

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Then, HI duct section loss =

$$\frac{20 \text{ ft} \times 0.095}{100} = 0.019 \text{ in. wg (line 14)}$$

The loss coefficient for transition H (converging flow) is obtained from Table A-12, Figure A using θ = 30 degrees (use the upstream velocity based on 3000 cfm) to compute the $V_p$, assuming that there is not an instant change in the upstream airflow velocity. This will hold true for each similar fitting in this example.

$$\text{Velocity} = \frac{3000}{20} \times \frac{18}{144} = 1200 \text{ fpm}$$

$V_p = 0.09$

$$\frac{A_1}{A} = \frac{20 \text{ in.} \times 18 \text{ in.}}{20 \text{ in.} \times 14 \text{ in.}} = 1.29 \text{ and } C = 0.05$$

H fitting loss =

$$C \times V_p = 0.05 \times 0.09 = 0.005 \text{ in. wg (line 15)}$$

The loss values in column L (0.019 and 0.005) are again totaled and entered on line 15 in column M (0.024 in. wg).

#### 7.5.2.6 Duct Section IJ

Duct section IJ is calculated as the above duct sections. The same type of transition is used (1000 cfm, 970 fpm, 13.9 in. diameter, with a 14 × 12 in. duct size selected at a 14.2 in. diameter equivalent):

IJ duct loss =

$$\frac{30 \text{ ft} \times 0.095}{100 \text{ ft}} = 0.029 \text{ in. wg}$$

If the 14.2 in. circular equivalent of the 14 ×12 in. duct is re-plotted on the chart in Figure A-1 for 1000 cfm, a velocity of 900 fpm and a friction loss of 0.080 will be obtained. A recalculation for the IJ duct loss is:

IJ duct loss =

$$\frac{30 \text{ ft} \times 0.080}{100 \text{ ft}} = 0.024 \text{ in. wg}$$

Since the new value is 0.003 in. wg, a rounded value of 0.024 in. wg is entered on line 16. However, if this were calculated on a computer, the larger (safer) amount would be used.

Transition at I (Table A-12, Figure A):

$$\frac{A_1}{A} = \frac{20 \text{ in.} \times 14 \text{ in.}}{14 \text{ in.} \times 12 \text{ in.}} = 1.67 \text{ and } C = 0.05$$

$$\text{Velocity} = \frac{2000}{20} \times \frac{14}{144} = 1029 \text{ fpm}$$

$V_p = 0.07$,

I fitting loss =

$$C \times V_p = 0.05 \times 0.07 = 0.004 \text{ in. wg (line 17)}$$

The "J" elbow is smooth, long radius without vanes (Table A-10, Figure F) having a $\frac{R}{W}$ ratio of 2.0. As $\frac{H}{W} = \frac{12}{14} = 0.86$, the loss coefficient of 0.16 is used.

By applying values of the 14.2 in. equivalent duct diameter and the duct velocity of 900 fpm to the *Reynolds Number Correction-Factor Chart*, it is found that a correction factor must be used. The actual average velocity is:

$$V = \frac{1000}{14} \times \frac{12}{144} = 857 \text{ fpm}$$

The equations under Note 3 for A-20 are solved to allow the correction factor to be obtained.

$$D = \frac{2HW}{H + W} = \frac{2 \times 12 \text{ in.} \times 14 \text{ in.}}{12 \text{ in.} + 14 \text{ in.}} = 12.92 \text{ in.}$$

$$R_e = 8.56\,DV = 8.56 \times 12.92 \times 857$$

$R_e = 94{,}780$

$R_e 10^{-4} = 9.48$

From the table (Note 3), the correction factor of 1.32 is obtained and the $V_p$ of 0.05 for 857 fpm is used.

Fitting loss =

$$C \times V_p \times K_{Re} = 0.16 \times 0.05 \times 1.32 = 0.011 \text{ in. wg}$$

(enter on line 18).

If the $K_{Re}$ correction factor was not used, the calculated loss of 0.008 in. wg (0.16 × 0.05) is 0.003 in. wg lower than the value used. On a long, winding duct run with many elbows, this could become significant.

The volume damper at J has the same coefficient as that used at F. Using the $V_p$ for 857 fpm:

Damper Loss =

$$C \times V_p = 0.04 \times 0.05 = 0.002 \text{ in. wg (line 19)}$$

Figure T of Table A-14 (Tee, Rectangular Main to Round Branch) should not be used for a round tap at the end of a duct run, nor should Figure Q for a square tap under the same conditions, as the total airflow is going through the tap. The closest duct configurations

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in the Appendix would be the mitered elbows in Table A-10, Figures C, D, or E. The average loss coefficient value for a 90 degree turn from these figures is 1.2, which is the recommended value to use.

Obviously, if there was ample room in the ceiling, the use of a vaned- or long-radius elbow and a rectangular to round transition would be the most energy efficient and have the lowest combined pressure loss. Therefore, the loss of the fitting at the diffuser should be calculated:

Fitting loss =

$$C \times V_p = 1.2 \times 0.05 = 0.060 \text{ in. wg.}$$

(enter on line 20).

The diffuser pressure loss from Figure 7-1 for the diffuser at J includes the pressure losses for the damper with the diffuser. The 0.14 in. wg is entered on line 21 in column L.

In Table 7-1, the pressure losses on lines 16 through 21 in column L are totaled (0.241 in. wg) and the value entered on line 21 in column M and on line 16 in column N.

Starting from the bottom (line 16), the pressure losses of each section in column M are accumulated in column N, resulting in a total pressure loss of 0.492 in. wg (line 4) for the duct run B to J (the assumed longest duct run). This total is added to the 0.90 in. wg on line 3 of column M (Fan Plenum B) for the total pressure loss of 1.382 in. wg. This is the design total pressure at which supply fan B must operate for 8000 cfm. The value of 1.382 in. wg is entered on line 1 in columns N and O. The numbers in column N and O are calculated after columns A to M.

Attention is called to the progressively lower value of the velocity pressure as the velocity continues to be reduced; velocity pressure is proportional to the square of the velocity. By carefully selecting fittings with low-loss coefficients, actual dynamic pressure loss values become very low. However, straight-duct loss values per 100 ft remain constant, as these losses depend only on the friction-loss rate selected. The minor modification at the last duct section was made because of the rectangular duct size selected.

The last section of duct (IJ), with all of its fittings and the terminal device had over half the pressure loss generated by the complete duct run (BJ). This is because all of the fittings in the main run had a static regain (included in the loss coefficients), with each lowering of the airstream velocity that reduced the actual pressure loss of each section.

#### 7.5.2.7 Duct Section FM

As the branch duct run F to M is similar to duct run G to J, one would assume the duct sizes would be the same. This assumes the branch pressure loss of the wye at F had approximately the same pressure loss as the 20 ft of duct from F to G (0.019 in. wg) and the elbow at G (0.014 in. wg for a total loss of 0.033 in. wg). However, to compute the complete duct run from $A_1$ to M, lines 1 to 9 ($A_1$ to F) in column M must be totaled (1.067 in. wg), and the result entered on line 1 (column M) of the table in Figure 7-1 (a) using a new duct sizing form.

Referring again to Table A-14, Figure W (used before for the wye at F), and using the same ratios as before, $\left(\frac{A_b}{A_s} = 1.0;\ \frac{A_b}{A_c} = 0.5 \text{ and } \frac{Q_b}{Q_c} = 0.5\right)$, the *branch* loss coefficient C = 0.52.

F fitting loss =

$$C \times V_p = 0.52 \times 0.11 = 0.057 \text{ in. wg (line 2)}$$

It should be noted that the fitting *entering* velocity of 1333 fpm is used to determine the velocity pressure for the computations. The branch loss of 0.057 in. wg for fitting F is compared to the 0.033 in. wg computed above for duct EG and elbow G. As the difference between them of 0.024 in. wg is within the 0.05 in. wg allowable design difference, the fitting used at F was a good selection.

However, the $A_1$M duct run will have a 0.024 in. wg greater pressure loss than the $A_1$J duct run. It appears the assumed "longest run" did not have the greatest pressure loss although, again, the difference was within 0.05 in. wg. This also confirms the need for the use of balancing dampers in each of the 20 ×18 in. ducts at F.

The information for the "branch" volume damper at F can be copied from line 12 of Table 7-1, since all conditions are the same. This will be entered on line 3 of Table 7-1 (a). The calculations then are made for the 10 ft of 20 ×18 in. duct (FK):

FK duct loss =

$$\frac{10 \text{ ft} \times 0.095}{100} = 0.010 \text{ in. wg}$$

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(enter on line 4).

The pressure losses on lines 2, 3, and 4 in column L are totaled and entered on line 4 in column M (0.071 in. wg) of Table 7-1 (a).

The pressure loss of the K to M duct section is identical to the H to J duct section, including the diffusers. Therefore, lines 15 and 21 in column M of Table 7-1 are totaled (0.265 in. wg) and entered on line 5 in columns M and N of Table 7-1 (a).

Finally, the figures in column M are entered in column N, starting from the bottom, to obtain the new total pressure loss of 1.403 in. wg for the fan B duct system (line 1, column 0). This loss is only 0.021 in. wg higher than the $A_1$J duct system pressure loss (Table 7-1), but it is the higher total pressure loss value to be used in the selection of Fan B.

#### 7.5.2.8 Duct Section EN

Using the balance of the duct sizing form Table 7-1 (a), the next duct run to be sized is the branch duct EQ. The pressure loss for the duct system from $A_1$ to E is obtained by totaling lines 1 to 7 of Table 7-1 and entering the 1.049 in. wg value on line 7 in column M.

Data for duct section EN is obtained (2000 cfm, 1140 fpm, and 18.2 in. diameter, with 20 × 14 in. being the selected rectangular size) using the same 0.095 in. wg friction-loss rate. This rate has changed only once in this example up to this point:

EN duct loss =

$$\frac{10 \text{ ft} \times 0.095}{100 \text{ ft}} = 0.010 \text{ in. wg}$$

(enter on line 8).

The data used before for computing the "main" loss coefficient for Wye E (Table A-14, figure W) is again used to obtain the "branch" loss coefficient, *see* Duct Section EF.

$\frac{A_b}{A_s} = 0.33$

$\frac{A_b}{A_c} = 0.25$

$\frac{Q_b}{Q_c} = 0.25$

(the preliminary calculations to branch EN are verified).

$C$ (branch) = 0.43 (by interpolation)

E fitting loss =

$$C \times V_p = 0.43 \times 0.13 = 0.056 \text{ in. wg (line 9)}$$

The loss values in column L (0.010 + 0.056) are totaled and entered on line 9 in column M (0.066 in. wg).

#### 7.5.2.9 Duct Section NP

Data for the 55 ft duct run from N to P is computed using the lower friction loss rate from duct section IJ, and the 14 × 12 in. rectangular size again is selected using 14.2 in. diameter, 0.08 in. wg per 100 ft friction loss rate, and a 900 fpm velocity.

NP duct loss =

$$\frac{55 \text{ ft} \times 0.08}{100 \text{ ft}} = 0.044 \text{ in. wg}$$

(enter on line 10).

At N, a 45 degree entry tap is used for branch duct NS and a 30 degree transition is used to reduce the duct size for the run to P. From Table A-12, Figure A:

$\frac{A_1}{A} = \frac{20 \times 14}{14 \times 12} = 1.67$

$C = 0.05$ for $\theta = 30$ degree,

Velocity = $\frac{2000}{20} \times \frac{14}{144} = 1029$ fpm,

$V_p = 0.07$

N fitting loss =

$$C \times V_p = 0.05 \times 0.07 = 0.004 \text{ in. wg (line 11)}$$

The volume damper at N has the same numbers as used above for the damper at J:

Damper loss =

$$C \times V_p = 0.04 \times 0.05 = 0.002 \text{ in. wg (line 12)}$$

At 0, a smooth radius elbow with one splitter vane is selected (Table A-10, Figure G):

$\frac{R}{W} = 0.25$, $\frac{H}{W} = \frac{12}{14} = 0.86$ and $C = 0.12$
(by interpolation)

0 fitting loss =

$$C \times V_p = 0.12 \times 0.05 = 0.006 \text{ in. wg (line 13)}$$

The cumulative loss of 0.056 in. wg (0.044 + 0.004 + 0.002 + 0.006) is entered on line 13 in column M.

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### 7.5.2.10 Duct Section PQ

Data for the last 20 ft of duct is obtained from Figure A-1 and Table A-2 (500 cfm, 810 fpm, 10.7 in. diameter), which is the equivalent of a 12 × 8 in. rectangular size:

PQ duct loss =

$$\frac{20 \text{ ft} \times 0.095}{100 \text{ ft}} = 0.019 \text{ in. wg}$$

(enter on line 14).

The loss coefficient for transition P is obtained from Table A-12, Figure A (converging flow) using θ = 45 degree:

$$\frac{A_1}{A} = \frac{14 \times 12}{12 \times 8} = 1.75$$

$C = 0.06$

Velocity = 857 (from the 14 × 12 in. duct)

P fitting loss =

$$C \times V_p = 0.06 \times 0.05 = 0.003 \text{ in. wg (line 15)}$$

The fitting at Q is a mitered 90 degree change-of-size elbow (Table A-10, Figure E).

$\frac{H}{W}$ = 8/12 = 0.67; $\frac{W_1}{W} = \frac{16}{12} = 1.33$

Velocity = $\frac{500}{12} \times \frac{8}{144} = 750$ fpm and $V_p = 0.04$

A fitting loss coefficient of 1.0 is selected. Plotting the data on the *Reynolds Number Correction-Factor Chart* indicates that a correction factor will be required.

$$D = \frac{2 \times 8 \text{ in.} \times 12 \text{ in.}}{8 \text{ in.} + 12 \text{ in.}} = 9.6 \text{ in.}$$

$R_e = 8.56$

$DV = 8.56 \times 9.6 \times 750 = 61{,}632$

$R_e 10^{-4} = 6.16$ and $K_{Re} = 1.09$

Q fitting loss =

$$1.0 \times 0.04 \times 1.09 = 0.044 \text{ in. wg}$$

(enter on line 16).

The pressure loss of 0.13 in. wg on the drawing (Figure 7-1) for the 16 × 8 in. grille is entered on line 17.

The pressure losses on lines 14-17 in column L are totaled (0.196 in. wg) and the value entered on line 17 in column M and on line 14 in column N. Starting from the bottom (line 14), the pressure losses of each section in column M are accumulated in column N, resulting in the total pressure loss of 1.367 in. wg. This is entered on line 7 in columns N and O.

The $A_1$M duct-run pressure loss of 1.430 in. wg is 0.063 in. wg higher than the 1.367 in. wg pressure loss of the $A_1$Q duct run. This gives a result that is slightly above the 0.05 in. wg suggested good design difference. Nevertheless, adding balancing dampers in the branch ducts at N, should allow the TAB technician to properly balance the system.

### 7.5.2.11 Duct Section NS

The pressure losses from $A_1$ to N (lines 7 to 9) are totaled (1.115 in. wg) and entered on line 18 in column M. The last section of the supply-duct system is sized using the same procedures and data from above:

NR duct loss =

$$\frac{8 \text{ ft} \times 0.080}{100 \text{ ft}} = 0.006 \text{ in. wg}$$

(enter on line 19).

A 45 degree entry rectangular tap is used for the branch duct at N. From Table A-14, Figure N:

$\frac{V_b}{V_c} = \frac{857}{1029} = 0.83$ (Use 1.0)

$\frac{Q_b}{Q_c} = \frac{1000}{2000} = 0.5$ and $C = 0.74$

Velocity = 1029 fpm and $Vp = 0.07$

N fitting loss =

$$C \times V_p = 0.74 \times 0.07 = 0.052 \text{ in. wg}$$

(enter on line 20).

The data for the volume damper in the branch duct at N, is the same as on line 12, which can be copied and entered on line 21. The total of lines 19-21 in column L of 0.060 can be entered on line 21 in column M.

Using the data from line 14:

RS duct loss =

$$\frac{20 \text{ ft} \times 0.095}{100 \text{ ft}} = 0.019 \text{ in. wg}$$

(enter on line 22).

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R Transition loss =

$C \times V_p = 0.06 \times 0.05$ (from line 15) $= 0.003$ in. wg

enter on line 23).

S Elbow loss =

$$C \times V_p \times K_{Re} \text{ (from line 16)} = 1.0 \times 0.04 \times 1.09 = 0.044 \text{ in. wg}$$

(enter on line 25).

The losses for Run RS in column L are totaled and 0.196 in. wg value is placed in column M on line 25 and in column N on line 22.

The section losses in column M are again added from the bottom in column N, and the total system loss from $A_1$ to S (1.371 in. wg) is entered on line 18 in columns N and O. This loss again is almost equal to that of the other portions of the duct system.

#### 7.5.2.12 Balancing Damper Setting Calculations

If the NS branch loss had been substantially lower, reasonable differences could have been compensated for by adjustments of the balancing damper. The damper loss coefficient used in each case was based on $\theta = 0$ degree (wide open). The preliminary damper setting angle $\theta$ can be calculated in this situation as follows, assuming a total system loss difference of 0.038 in. wg between points S and Q for this example:

System loss difference = 0.038 in. wg

N damper loss (set at 0 degree) = 0.002 in. wg

N damper loss (set at ? degrees) = 0.040 in. wg (0.038 + 0.002)

Damper loss =

$$C \times V_p \text{ or } C = \frac{\text{Damper loss}}{V_p} = \frac{0.040}{0.50} = 0.80$$

Referring back to Table A-18, Figure B, the loss coefficient when $C = 0.80$ would require a damper angle $\theta$ of about 15 degrees (by interpolation). The duct airflow and velocity at the damper would still remain at the design values. Points S and Q of the duct system would then have the same total pressure loss relative to point $A_1$ or fan B.

### 7.5.3 Additional Discussion

Other advantages of the above duct sizing procedures are when using columns M and N, the designer can observe the places in the duct system that have the greatest total pressure losses and locate where the duct construction pressure classifications change, *see* Table 4-1 and Figure 4-1 in Chapter 4. After the duct system is sized, these static pressure "flags" should be noted on the drawings as shown on Figure 7-1, to obtain the most economical duct fabrication and installation bids.

Building-pressure allowance for supply air duct systems should be determined from building ventilation requirements considering normal building infiltration. Allowance in the range of 0.02 to 0.1 in. wg for building-pressurization is normally used. The designer should determine the proper building-pressurization value based upon individual system requirements and location. Consideration should also include elevator shaft ventilation requirements, tightness of building construction, building stack effect, and fire and smoke code requirements.

Finally, the system pressure loss checklist in of Chapter 8 should be used to verify that all system component pressure losses have been included in the fan total-pressure requirements. Be sure some allowance has been added for possible changes in the field. These additional items should be shown on the duct-sizing work sheets.

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DUCT SIZING WORK SHEET
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PROJECT SAMPLE BUILDING LOCATION FIRST FLOOR SYSTEM EXHAUST AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW CFM | FRIC-TION PER 100′ | VELOC-ITY FPM | $V_p$ | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMULA-TIVE LOSS | TOTAL LOSS |
| 1 | PLENUM Z | $A_1Z$ | – | H.R DEVICE | 8000 | – | 500 | – | – | – | – | – | 0.30 | | 1.272 | 1.272 |
| | ” | Z | – | FAN | 8000 | – | 1595 | 0.16 | – | – | – | – | 0.24 | 0.54 | | ($A_1T$) |
| 3 | RUN YT | YW | 30′ | DUCT | 8000 | 0.08 | 1500 | – | – | 32.8 | 30 × 30 | – | 0.024 | | 0.732 | |
| 4 | ” | Y | – | SYSTEM EFFECT | 8000 | – | 1500 | – | – | – | – | P | 0.280 | | | |
| 5 | ” | Y | – | TRANS | 8000 | – | 1280 | 0.10 | 0.24 | – | 30 × 30 / 330 | – | 0.024 | | | |
| 6 | ” | Y | 90° | ELBOW | 8000 | – | 1280 | 0.10 | 0.44 | – | 30 × 30 | – | 0.044 | 0.372 | | |
| 7 | ” | WU | 100′ | DUCT | 6000 | 0.08 | 1400 | – | – | 28.0 | 30 × 22 | – | 0.080 | | 0.360 | |
| 8 | ” | W | 45° | ENT. TAP | 8000 / 2000 | – | 1280 | 0.10 | 0.33 | – | 30 × 30 / 14 × 14 | – | 0.033 | | | |
| 9 | ” | W | – | TRANS | 8000 / 6000 | – | 1309 | 0.11 | 0.20 | – | 30 × 30 / 30 × 22 | – | 0.022 | | | |
| 10 | ” | V | 90° | ELBOW | 6000 | – | 1309 | 0.11 | 0.16 | – | 30 × 22 | – | 0.018 | 0.153 | | |
| 11 | ” | UT | 20′ | DUCT | 3000 | 0.08 | 1180 | – | – | 21.7 | 22 × 18 | – | 0.016 | | 0.207 | |
| 12 | ” | U | 90° | ENT. TAP | 6000 / 3000 | – | 1309 | 0.11 | 0.53 | – | 30 × 22 / 48 × 18 | – | 0.058 | | | |
| 13 | ” | U | – | TRANS | 6000 / 3000 | – | 1091 | 0.07 | 0.25 | – | 30 × 22 / 22 × 18 | – | 0.018 | | | |
| 14 | ” | T | 90° | ELBOW | 3000 | – | 500 | 0.02 | 1.75 | – | 22 × 18 / 48 × 18 | – | 0.035 | | | |
| 15 | ” | T | – | GRILLE | 3000 | – | – | – | – | – | 48 × 18 | – | 0.080 | 0.207 | | |
| 16 | ” | | | | | | | | | | | | | | | |
| 17 | RUN WX | WX | 20′ | DUCT | 2000 | 0.20 | 1550 | – | – | 15.3 | * 14 × 14 | 1.93 | 0.077 | | 0.360 | |
| 18 | ” | W | 45° | ENT. TAP | 8000 / 2000 | – | 1280 | 0.10 | –0.37 | – | 30 × 30 / 14 × 14 | – | –0.037 | | | |
| 19 | ” | X | 180° | TRAMS | 2000 | – | 1469 | 0.13 | 0.33 | – | 14 × 14 / 36 × 16 | – | 0.043 | | | |
| 20 | ” | X | – | GRILLE | 2000 | – | – | – | – | – | 36 × 16 | – | 0.080 | | | |
| 21 | ” | W | SET 22° | VOL DAMP | 2000 | – | 1469 | 0.13 | 1.52 | – | 14 × 14 | – | 0.197 | 0.360 | | |
| 22 | | | | | | | | | | | | | | | | |
| 23 | | | | | | | | | | | | | | | | |
| 24 | | | | | | | | | | | | | | | | |
| 25 | | | | | | | | | | | | | | | | |

**Table 7–2 Duct Sizing, Exhaust Air System – Example 2 (I–P)**

NOTE: * Indicates duct lining used. Sizes are interior dimensions.

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DUCT SIZING WORK SHEET
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PROJECT SAMPLE BUILDING LOCATION CONF AREA SYSTEM SUPPLY AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW CFM | FRIC-TION PER 100′ | VELOC-ITY FPM | $V_P$ | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMULA-TIVE LOSS | TOTAL LOSS |
| 1 | PLE-NUM A | A | – | SYSTEM EFFECT | 20,000 | – | 2190 | – | – | 40.9 | 44 × 32 | RS | 0.290 | | 2.875 | 2.875 |
| 2 | ” | B | 90° | ELBOW | 20,000 | – | 2045 | 0.26 | 0.15 | 40.9 | 44 × 32 | – | 0.039 | 0.329 | | (AS) |
| 3 | RUN CS | CF | 80′ | DUCT | 20,000 | 0.30 | 3200 | – | – | 34.0 | – | – | 0.240 | | 2.546 | |
| 4 | ” | C | 20° | TRANS | 20,000 | – | 2045 / 3200 | 0.64 | 0.05 | – | 44 × 32 / 340 | – | 0.032 | | | |
| 5 | ” | D | – | SOUND ATTEN. | 20,000 | – | – | – | – | 34.0 | – | – | 0.260 | | | |
| 6 | ” | E | 90° | ELBOW | 20,000 | – | 3200 | 0.64 | 0.15 | 34.0 | – | – | 0.096 | 0.628 | | |
| 7 | ” | FH | 50′ | DUCT | 10,000 | 0.30 | 2700 | – | – | 26.0 | – | – | 0.150 | | 1.918 | |
| 8 | ” | F | 45° | WYE | 20,000 / 10,000 | – | 3200 | 0.64 | 0.28 | 34 / 26 | – | – | 0.179 | | | |
| 9 | ” | F | 45° | ELBOW | 10,000 | – | 2700 | 0.45 | 0.15 | 26.0 | – | 0.6 | 0.041 | | | |
| 10 | ” | G | 90° | ELBOW | 10,000 | – | 2700 | 0.45 | 0.15 | 26.0 | – | – | 0.068 | 0.438 | | |
| 11 | ” | HO | 40′ | DUCT | 5000 | 0.30 | 2300 | – | – | 20.0 | – | – | 0.120 | | 1.480 | |
| 12 | ” | H | 45° | WYE | 10,000 / 5000 | – | 2700 | 0.45 | 0.51 | 26 / 20 | – | – | 0.230 | | | |
| 13 | ” | H | 0° | VOL DAMP | 5000 | – | 2300 | 0.33 | 0.20 | 20.0 | – | – | 0.066 | | | |
| 14 | ” | N | 90° | ELBOW | 5000 | | 2300 | 0.33 | 0.15 | 20.0 | – | – | 0.050 | 0.466 | | |
| 15 | ” | OP | 30′ | DUCT | 4000 | 0.34 | 2250 | – | – | 18.0 | – | – | 0.102 | | 1.014 | |
| 16 | ” | O | 45° | WUE | 5000 / 4000 | – | 2300 | 0.33 | 0.01 | 20 | – | – | 0.003 | | | |
| 17 | ” | O | 60° | TRANS | 4000 | – | 2250 | 0.32 | 0.06 | 20 / 18 | – | – | 0.019 | 0.124 | | |
| 18 | ” | PQ | 30′ | DUCT | 3000 | 0.36 | 2150 | – | – | 16.0 | – | – | 0.108 | | 0.890 | |
| 19 | ” | P | 45° | WYE | 4000 / 3000 | – | 2250 | 0.32 | 0.01 | 18.0 | – | – | 0.003 | | | |
| 20 | ” | P | 60° | TRANS | 3000 | – | 2150 | 0.29 | 0.06 | 18 / 16 | – | – | 0.017 | 0.128 | | |
| 21 | ” | QR | 30′ | DUCT | 2000 | 0.32 | 1880 | – | – | 14.0 | – | – | 0.096 | | 0.762 | |
| 22 | ” | Q | 45° | WYE | 3000 / 2000 | – | 2150 | 0.29 | 0.01 | 16.0 | – | – | 0.003 | | | |
| 23 | ” | Q | 60° | TRANS | 2000 | – | 1880 | 0.22 | 0.06 | 16 / 14 | – | – | 0.013 | 0.112 | | |
| 24 | ” | – | – | SUB–TOTAL FROM PAGE 2 0F 2 | | | | | | | | | – | 0.650 | 0.650 | |
| 25 | ” | | | | | | | | | | | | | | | |

**Table 7–3 Duct Sizing, Exhaust Air System – Example 3 (I–P)**

NOTE: * Indicates duct lining used. Sizes are interior dimensions.



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DUCT SIZING WORK SHEET
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PROJECT SAMPLE BUILDING LOCATION CONF AREA SYSTEM SUPPLY AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW CFM | FRIC-TION PER 100′ | VELOC-ITY FPM | $V_p$ | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMULA-TIVE LOSS | TOTAL LOSS |
| 1 | RUN CS | RS | 30′ | DUCT | 1000 | 0.19 | 1290 | – | – | 12.0 | – | – | 0.057 | | 0.650 | |
| 2 | (CONT.) ” | R | 45° | WYE | 2000 / 1000 | – | 1880 | 0.22 | 0.04 | 14.0 | – | – | 0.009 | | | |
| 3 | ” | R | 60° | TRANS | 1000 | – | 1290 | 0.10 | 0.06 | 14 / 12 | – | – | 0.006 | | | |
| 4 | ” | S | 45° | ELBOW | 1000 | – | 1290 | 0.10 | 0.15 | 12.0 | – | 0.6 | 0.009 | | | |
| 5 | ” | S | 5′ | FLEX DUCT | 1000 | 0.19 | 1290 | 0.10 | – | 12.0 | – | 1.95 | 0.019 | | | |
| 6 | ” | S | – | VAV BOX | 1000 | – | – | – | – | – | – | – | 0.550 | 0.650 | | |
| 7 | | | | | | | | | | | | | | | | |
| 8 | | | | | | | | | | | | | | | | |
| 9 | | | | | | | | | | | | | | | | |
| 10 | | | | | | | | | | | | | | | | |
| 11 | | | | | | | | | | | | | | | | |
| 12 | | | | | | | | | | | | | | | | |
| 13 | | | | | | | | | | | | | | | | |
| 14 | | | | | | | | | | | | | | | | |
| 15 | | | | | | | | | | | | | | | | |
| 16 | | | | | | | | | | | | | | | | |
| 17 | | | | | | | | | | | | | | | | |
| 18 | | | | | | | | | | | | | | | | |
| 19 | | | | | | | | | | | | | | | | |
| 20 | | | | | | | | | | | | | | | | |
| 21 | | | | | | | | | | | | | | | | |
| 22 | | | | | | | | | | | | | | | | |
| 23 | | | | | | | | | | | | | | | | |
| 24 | | | | | | | | | | | | | | | | |
| 25 | | | | | | | | | | | | | | | | |

**Table 3(a) Duct Sizing, Exhaust Air System – Example 3 (I–P) (Continued)**

NOTE: * Indicates duct lining used. Sizes are interior dimensions.

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## 7.6 RETURN AIR (EXHAUST AIR) DUCT SYSTEM–SIZING EXAMPLE 2

The exhaust air duct system of fan "Y" shown in Figure 7-1 will be sized using lower main duct velocities to reduce the fan power requirements. This will conserve energy and, therefore, lower the operating costs. However, the duct sizes will be larger, which could increase the initial cost of the duct system.

Attention is called to the discussion in Section 5.42. All of the static pressure and total pressure values are negative, with respect to atmospheric pressure on the suction side of the fan. Applying this concept to Equation 5-5:

Fan $SP = TP_d - TP_s - V_{pd}$ (Equation 5-4)

Fan $SP = TP_d - (-TP_s) - V_{pd}$

Fan $SP = TP_d + TP_s - V_{pd}$

as $TP = SP + V_p$, then:

**Equation 7–2**

$$\text{Fan } TP = TP_d + TP_s$$

Where:

$TP_d = TP$ of fan discharge

$TP_s = TP$ of fan suction

Using the suction side of Equation 7-2, all of the system pressure-loss values for the exhaust system (suction side of the fan) will be entered on the worksheet as positive numbers.

### 7.6.1 Exhaust Air Plenum Z

Pressure-loss data for the discharge side of the heat-recovery device A1Z is entered on line 1 of Table 7-2 in column L (0.30 in. wg). As the backwardly-curved blade fan Z free discharges into the plenum, a tentative fan selection must be made in order to obtain a velocity or velocity pressure to use to calculate the pressure loss. Most centrifugal fans are rated with duct connections on the discharge side. A loss due to "no static regain" must be added for the free discharge into the plenum.

From manufacturer's data, $Vp = 0.16$ and $C = 1.5$ from Table A-13, Figure I:

Z Fan pressure loss =

$$C \times V_p = 1.5 \times 0.16 = 0.24 \text{ in. wg}$$

(enter on line 2).

The plenum loss total of 0.54 in. wg is entered on line 2 in column M.

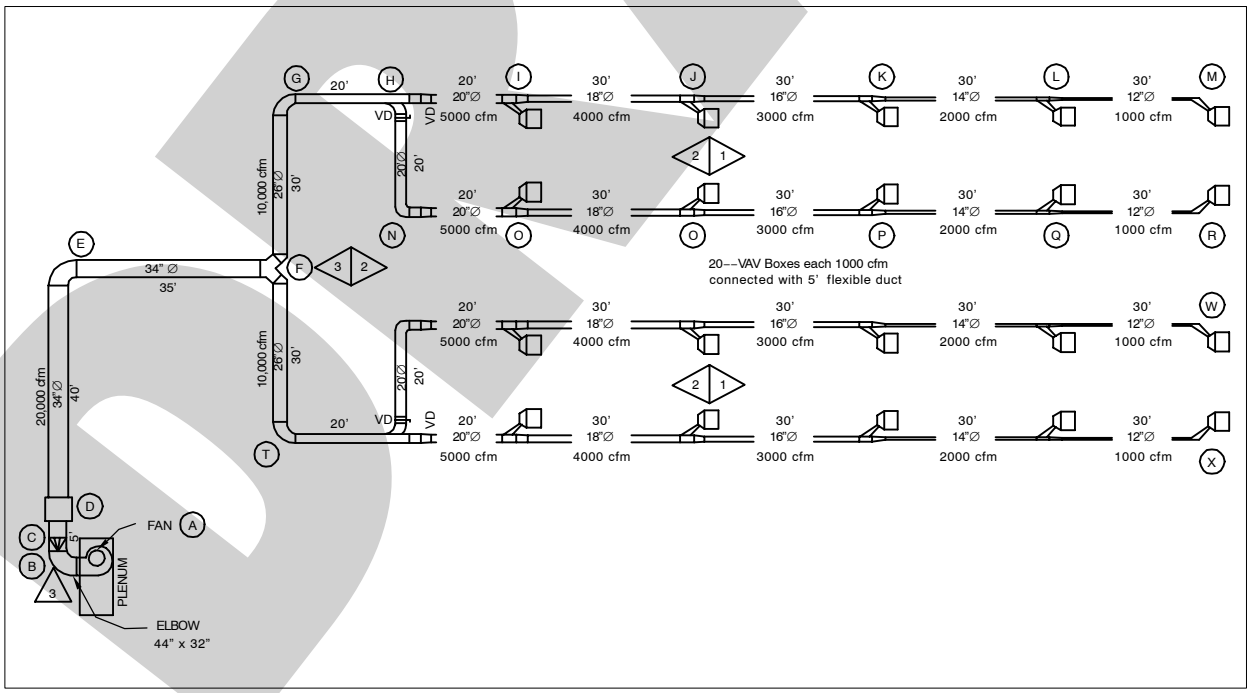


**FIGURE 7–2 SUPPLY AIR DUCT SYSTEM FOR SIZING EXAMPLE 3**

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### 7.6.2 Exhaust Air System

#### 7.6.2.1 Duct Section YW

Using 8000 cfm, 1500 fpm is selected from the chart in Figure A-1, which establishes the duct-friction loss at 0.08 in. wg per 100 ft of duct and the diameter at 32.8 in. From Table A-2, a 30 × 30 in. rectangular duct can be selected for the YW duct section and the computed friction loss value entered in column L.

YW duct loss =

$$\frac{30 \text{ ft} \times 0.08}{100} = 0.024 \text{ in. wg}$$

(enter on line 3).

The fan-intake connection must be examined for a possible System-Effect Factor, which can be added to the system losses, or deducted from the fan rating. For this example, it will be added to the system losses. Using a radius elbow with an inlet transition, *see* Figure 6-12a and no duct between, R/H = 0.75 indicates the use of the "P" System Effect Curve. Using the chart in Figure 6-1, a velocity of 1500 fpm indicates a System Effect Factor of 0.28 in. wg (entered on line 4).

The use of an inlet box, *see* Figure 6-20, would reduce the loss.

The dynamic friction loss of the elbow and transition must also be computed. Table A-7, Figure F can be used for the elbow, and Table A-8, Figure D for the transition.

Transition Y:

$$\tan\left(\frac{\theta}{2}\right) = \frac{D - 1.13\sqrt{HW}}{2L} = \frac{33 - 1.13\sqrt{30 \times 30}}{2 \times 2}$$

$$\tan\left(\frac{\theta}{2}\right) = \frac{33 - 33.9}{4} = 0.225$$

$$\frac{\theta}{2} = 12.68 \text{ degrees and } \theta = 25.36 \text{ degrees}$$

From Table A-8, Figure B:

$\frac{A_1}{A < 2}$ and $C = 0.24$ (by interpolation)

Velocity = $\frac{8000}{30} \times \frac{30}{144} = 1280$ fpm

From Table A-4 or by calculation,

$V_p = 0.10$ in. wg

Y Transition loss =

$$C \times V_p = 0.24 \times 0.10 = 0.024 \text{ in. wg}$$

(enter on line 5).

Elbow Y:

$\frac{H}{W} = 1.0$, $\frac{R}{W} = 0.75$, $C = 0.44$

Using the equivalent diameter, a quick check of the *Reynolds Number Correction-Factor Chart* on page A.15 indicates that no correction is needed.

Y Elbow loss =

$$C \times V_p = 0.44 \times 0.10 = 0.044 \text{ in. wg}$$

(enter on line 6).

Note that the combined pressure loss of 0.348 in. wg (0.280 + 0.024 + 0.044) for the system effect, transition and elbow are far greater than the loss when using an inlet box (loss coefficient of 1.0):

Inlet box loss =

$$C \times V_p = 1.0 \times 0.10 = 0.10 \text{ in. wg}$$

The total for YW (0.372) is entered on line 6 in column M.

#### 7.6.2.2 Duct Section WU

Using 6000 cfm and 0.8 in. wg per 100 ft, 1400 fpm is established with 28.0 in. diameter. Using Table A-2 rectangular size of 30 × 20 in. is selected (keeping one side the same size).

WU duct loss =

$$\frac{100\,ft \times 0.08}{100} = 0.08 \text{ in. wg}$$

(enter on line 7).

A converging 45 degree entry fitting will be used at W., *see* Table A-10, Figure F.

To obtain the "main" loss coefficient, the note in Fitting 14-13F refers to Fitting 14-13B:

Using Table A-10B (Main Coefficient):

$\frac{Q_b}{Q_c} = \frac{2000}{8000} = 0.25$ and $C = 0.33$

(by interpolation)

W fitting loss =

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$$C \times V_p = 0.33 \times 0.10 = 0.033 \text{ in. wg}$$

(enter on line 8).

Note that 0.10 is the velocity pressure of the 30 × 30 in. downstream section (note direction of flow).

The diverging flow transition at W with an included angle of 30 degree uses Table A-8, Figure E because of the change of only one duct dimension.

$\frac{A_1}{A} = \frac{30 \times 30}{30 \times 22} = 1.36$ (use 2) and $C = 0.20$

Upstream section velocity =
$\frac{6000}{30} \times \frac{22}{144} = 1309$ fpm.

$V_p = 0.11$ (from Table A-4 or by calculation)

W transition fitting loss =

$$C \times V_p = 0.20 \times 0.11 = 0.022 \text{ in. wg}$$

(enter on line 9).

Using a radius elbow without vanes (Table A-7, Figure F) at V, the following data is used:

$\frac{H}{W} = \frac{22}{30} = 0.73$ and $\frac{R}{W} = 2.0$

$C = 0.16$

Again, using the equivalent diameter of 28.0 and the velocity of 1309 fpm, a check of the *Reynolds Number Correction-Factor Chart* indicates that no correction is needed.

V fitting loss =

$$C \times V_p = 0.16 \times 0.11 = 0.018 \text{ in. wg}$$

(enter on line 10).

As before, the total section loss of 0.145 in. wg is entered in column M.

#### 7.6.2.3 Duct Section UT

The total-pressure loss is always the same as the static pressure when there is no velocity change. For the duct section UT this equals:

UT duct loss =

$$\frac{20 \text{ ft} \times 0.08}{100} = 0.016 \text{ in. wg} = 0.016$$

(enter on line 11).

From Figure A-1 where a 21.7 in. diameter duct and 1180 fpm was obtained for 3000 cfm. A 22 × 18 in. rectangular duct is selected from Table A-2. A converging 90 degree tee fitting (Table A-10, Figure D) will be used at U, but the "main" loss coefficient is obtained from Figure A-10B.

$\frac{Q_b}{Q_c} = \frac{3000}{6000} = 0.5$ and $C = 0.53$

U fitting loss =

$$0.53 \times 0.11 \text{ (downstream } V_p) = 0.058 \text{ in. wg}$$

(enter on line 12).

The U transition loss coefficient is found in Table A-8, Figure B and the following data computed:

$\frac{A_1}{A} = \frac{30 \times 22}{22 \times 18} = 1.67$ (use 2)

$\theta = 30$ degree

$C = 0.25$

Velocity = $\frac{3000}{22} \times \frac{18}{144} = 1091$ fpm

$V_p = 0.07$ (upstream duct)

U fitting loss =

$$C \times V_p = 0.25 \times 0.07 = 0.018 \text{ in. wg}$$

(enter on line 13).

The pressure loss for the change-of-size elbow at T will again be computed using Table A-7, Figure E. Caution should be used to determine airflow direction:

$\frac{H}{W} = \frac{18}{48} = 0.38$, $\frac{W_1}{W} = \frac{24}{48} = 0.5$

$C = 1.75$ (by interpolation and extrapolation)

Velocity of the upstream section (grille size) =
$\frac{3000}{48} \times \frac{18}{144} = 500$ fpm

$V_p = 0.02$

T fitting loss =

$$C \times V_p = 1.75 \times 0.02 = 0.035 \text{ in. wg}$$

(enter on line 14).

Turning vanes could be added to the change-of-size mitered elbow, but no loss coefficient tables are avail-

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able. If single-blade turning vanes reduce the C = 1.2 of a standard 90 degree mitered elbow to about C = 0.15, the C – 1.75 used above could be reduced to approximately C = 0.22 (using the same ratio).

The pressure loss of 0.08 in. wg for the exhaust grille at T is taken from Figure 7-1 and entered on line 15.

The section losses in column M are again added from the bottom in column N, and the Y fan duct system total of 1.272 in. wg entered on line 1 in columns N and O.

#### 7.6.2.4 Duct Section WX (Modified Design Method)

Branch WX must now be sized, but a visual inspection indicates that the pressure drop from W to X would be much less than that of the long run from W to T. The cumulative loss of 0.360 in. wg for duct run W to T (line 7, column N), is also the total pressure loss requirement for the short 20 ft duct run. A 0.05 in. wg value is an acceptable pressure difference between outlets or inlets on the same duct run.

In an attempt to dissipate this pressure, a velocity of 1550 fpm and a duct friction loss rate of 0.2 in. wg per 100 ft (15.3 in. diameter) are selected for the 2,000 cfm flow rate (Figure A-1). One inch thick duct lining (correction factor = 1.93 from Figure A-2 and Table A-1) also can be added for noise control and increased friction. A balancing damper is to be used for final adjustments. The computations using this modification of the design method are:

WX duct loss =

$$\frac{20 \text{ ft} \times 0.2}{100 \text{ ft}} \times 1.93 = 0.077 \text{ in. wg}$$

(enter on line 17).

Select the rectangular size of 14 × 14 in. from Table A-2.

The converging 45 degree entry fitting used at W (Table A-10, Figure F) is reviewed again to determine the branch loss coefficient.

$\frac{Q_b}{Q_c} = 0.25$,

Velocity ($V_c$) = 1280 fpm

$C = -0.37$

W fitting loss =

$$-0.37 \times 0.10 = -0.037 \text{ in. wg}$$

Since there is a negative branch pressure loss for this fitting because of static regain (data is entered on line 18), additional losses must be provided by a balancing damper or a perforated plate in the branch duct.

If a straight rectangular tap were used (Table A-10, Figure D) instead of the 45 degree entry tap, the loss coefficient would then become 0.01, a more appropriate selection.

An inefficient transition at X also will help build up the loss. Figure A is a rectangular converging transition in Table A-9.

$$\frac{A_1}{A} = \frac{36 \text{ in.} \times 36 \text{ in.}}{14 \text{ in.} \times 14 \text{ in.}} = 2.94$$

$\theta$ = 180 degree (abrupt)

$C$ = 0.33 by interpolation. The downstream velocity must be used to determine the $V_p$ used in the computations:

Velocity = $\frac{2000}{14} \times \frac{14}{144} = 1469$ fpm

$V_p$ = 0.13;

X fitting loss =

$$C \times V_p = 0.33 \times 0.13 = 0.043 \text{ in. wg}$$

(enter on line 19).

X grille loss (from Figure 7-1) = 0.08 in. wg (line 20)

Subtracting 0.163 in. wg (the total of lines 17 to 20) from the 0.360 in. wg duct run WT pressure loss shown on line 7 in. column N, leaves 0.197 in. wg of pressure for the balancing damper to dissipate.

Damper loss coefficient $C = \frac{TP}{V_p} = \frac{0.197}{0.13} = 1.52$

From Table A-15, Figure B, a damper set at 22 degree (by interpolation) has the loss coefficient of 1.53 that will balance the branch duct WX. The total of 0.360 in. wg (adding lines 17-21) is entered on line 21 in column M and on line 17 in column N.

A perforated plate (Table A-14, Figure B) is a non-adjustable alternate solution. If a ⅛ in. thick perforated plate were used instead of the balancing damper, the calculation procedure would be as follows, *see* Table A-14, Figure B:

Assuming ⅝ in. diameter holes

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$$\frac{T}{d} = \frac{0.125}{0.625} = 0.2$$

With $C = 1.53$ (from above)

$n = 0.64$ (by interpolation)

$$n = \frac{A_p}{A} \text{ and } A_p = n \times A$$

$A_p$ (flow area of perforated plate)
$= 130.64 \times 16 \times 13$
$= 133.12$ in.$^2$

$$\text{No. of holes} = \frac{A_p}{\text{area of a } 5/8 \text{ in. hole}} = \frac{133.12}{0.307}$$

= 434 (⅝ in. diameter)

## 7.7 SUPPLY AIR DUCT SYSTEM SIZING EXAMPLE 3 (I–P)

Higher-pressure supply air systems (over 3 in. wg) usually are required for most large central station HVAC supply-air duct distribution systems. Because of higher fan power requirements, ASHRAE provisions will cause the designer to analyze lower pressure duct systems, against the constantly increasing costs of building operation. The choice of duct system pressure is becoming more dependent on energy costs, the application, and the ingenuity of the designer.

The "Static Regain Method" and the "Total Pressure Method" have traditionally been used to design the higher pressure supply air systems.

### 7.7.1 Design Procedures

After analyzing the duct system layout, the chart in Figure A-1 is used to select an "approximate" initial velocity and a pressure loss per 100 ft that will be used for most duct sections throughout the system. This selected velocity should be within the shaded sections of the chart. Using the design airflow quantities (cfm) for the duct sections, and the selected velocity (fpm), the duct diameters and friction loss rates also may be obtained from Figure A-1. When rectangular duct sizes are to be used, selection may be made from the chart in Table A-2, based on circular equivalents.

The use of higher velocities normally increases duct system noise levels. The designer must consider that acoustical treatment might be required for the duct system and an allowance must be made for increased duct dimensions if lined or for additional space requirements if sound attenuators are used.

If acoustic treatment or insulation liner is required, the designer must clearly indicate if the duct dimensions shown on the duct layout drawings have been adjusted for this added thickness.

The designer must inspect the duct layout and make an assumption as to which duct run has the highest pressure loss. This is the path for the first series of calculations. The average velocity of the initial duct section based on the cross-sectional area is used to obtain the velocity pressure ($V_p$) from Table A-4, it may also be calculated using Equation 5-8 in Chapter 5. The velocity pressure is used with fitting loss coefficients from the tables in the Appendix to determine the dynamic-pressure loss of each fitting. The pressure losses of system components usually are obtained from equipment data sheets, but approximate data can be selected from the tables and charts in Chapter 8. The total pressure loss is then computed for the initial duct section by totaling the individual losses of the straight duct sections and duct fittings.

Each succeeding duct section is computed in the same manner, with careful consideration being given to the type of fitting selected. Compare each loss coefficient to obtain the most efficient fitting.

After the calculations are made, and each duct section properly sized, the pressure loss must be added for the terminal outlet device at the end of the last duct section. Adding from the bottom of the form to the top, the section losses are totaled in column N, to obtain the supply fan pressure requirements for the supply-air duct system (if the original "duct run with the highest pressure loss" assumption was correct).

Using the cumulative pressure subtotal of the main duct at the point of each branch, calculate the cumulative pressure total for each branch run as outlined above. If a duct run, other than the assumed duct run, has a higher cumulative pressure loss total then the higher amount now becomes the pressure that the fan must provide to the supply air duct system. The return air duct system, which is calculated separately, is also part of the fan load. Velocities and friction loss rates for the shorter runs may fall into a "higher velocity range" as long as the noise potential is considered.

Caution must be used in the above sizing procedure for the "longest duct run." The use of smaller duct sizes creates higher velocities and pressures and can increase the fan power and cost of operation. This is becoming more critical with rising energy costs.

A life-cycle cost analysis will probably dictate that lower operating costs should be considered more im-

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portant than lower first costs and space-saving requirements.

### 7.7.2 Supply Air System

Table 7-3 is the tabulation of design and computational data obtained when sizing the 20,000 cfm supply-duct system shown in Figure 7-2. The 290 ft duct run from C to S appears to be the path with the greatest resistance, although the duct run from C to W appears to have about the same resistance. All of the VAV terminal units have the same capacity (1000 cfm each). The airflow (cfm) of the duct sections varies from 20,000 to 1000 cfm. Selecting an initial velocity of approximately 3200 fpm and a friction rate of 0.30 in. wg per 100 ft would indicate (by following the 0.30 in. wg line horizontally to 1000 cfm) that the duct velocities would gradually be reduced to almost 1500 fpm at 1000 cfm.

#### 7.7.2.1 Plenum

Before the duct system is sized, the losses within the plenum must be calculated. Data from the manufacturer's catalog for the DWDI fan A, indicates a discharge outlet size of 43 × 32 in., a discharge velocity of 2190 fpm (velocity pressure = 0.30 in. wg), and a blast area/outlet area ratio of 0.6.

Elbow BN is sized 44 × 32 in. (so that it is similar to the outlet size) and a radius elbow $\left(\frac{R}{W} = 1.5\right)$ is selected. It is located 26 in. above the fan discharge opening.

Using the directions in Figure 6-2, Figure 6-3, and Table 6-2 for a DWDI fan, the pressure loss is calculated for the "System Effect" created by the discharge elbow at B:

Equiv. Diam. = $\sqrt{\frac{4 \times 44 \times 32}{\pi}} = 42.3$ in.

(or use 40.9 in. from Table A-2),

Percent Effective duct

$$\frac{\text{straight duct length} \times 100}{\text{Vel.}/1000(2.5\text{ min.}) \times \text{Equiv.Diam.}}$$

Percent Effective duct = $\frac{26 \times 100}{2.5 \times 42.3} = 24.6$ percent

From Table 6-2, System-Effect Curve R-S for a 0.6 blast area ratio and 25 percent Effective Duct is used with Figure 6-1 to find the System-Effect pressure loss of 0.29 in. wg (based on 2190 fpm). As the elbow is in position "A" (Figure 6-3), the multiplier for the DWDI fan from Table 6-2 of 1.00 does not change the value, which is entered on line 1 in column L of the duct sizing work sheet in Table 7-3. Again it is noted that the 0.29 in. wg could be subtracted from the total pressure output of the fan instead of being added to the total system loss.

The loss coefficient of 0.15 for elbow B is obtained (using Table A-7 Figure F) with $\frac{R}{W} = 1.5$ and $\frac{H}{W} = \frac{44}{32}$ = 1.38.

Average Velocity = $\frac{20,000}{44} \times \frac{32}{144} = 2,045$ fpm

The velocity pressure ($V_p$) of 0.26 in. wg is obtained from Table A-4 for a velocity of 2045 fpm. A quick check of the *Reynolds Number Correction-Factor Chart* shows that no correction is needed.

B fitting loss =

$$C \times Vp = 0.15 \times 0.26 = 0.039 \text{ (line 2)}$$

The total pressure loss of 0.329 in. wg for the plenum is entered on line 2 in column M.

#### 7.7.2.2 Duct Section CF

Round spiral duct with an absolute roughness of 0.0003 in. will be used in this supply duct system. For the 80 in. of duct in section CF, and using an assumed velocity of 3200 fpm, it falls right on the closest standard size duct diameter of 34 in. (in the chart of Figure A-1). The selected velocity of 3200 fpm has a friction loss rate of 0.30 in. wg per 100 ft. A duct friction-correction factor is not required, as the chart in Figure A-1 is based on the same absolute roughness.

CF duct loss =

$$\frac{80 \text{ ft} \times 0.30}{100 \text{ ft}} = 0.240 \text{ in. wg (line 3)}$$

The transition at C will be converging, rectangular to round (Table A-9, Figure A) with $\frac{A_1}{A} = 44 \times \frac{32}{(17)^2\pi}$ = 1.55 and $\theta = 20$ degree C = 0.05. The velocity pressure used is the same as the downstream section: 0.64 in. wg for 3200 fpm (Table A-4).

C transition loss =

$$C \times V_p = 0.05 \times 0.64 \text{ (leaving } V_p) = 0.032 \text{ in. wg (line 4)}$$

The pressure loss for a medium attenuation 34 in. diameter sound trap of 0.26 in. wg is obtained from

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Chapter 8. A preliminary loss also can be obtained from manufacturer's data sheets. The data is entered on line 5.

The smooth radius, 90 degree round elbow at E has an $\frac{R}{D}$ ratio = 1.5; $C$ = 0.15 (Table A-7, Figure A).

E elbow loss =

$$C \times V_p = 0.15 \times 0.64 = 0.96 \text{ in. wg (line 6)}$$

The pressure losses of the four items in duct section CF are added and the 0.628 in. wg total is entered in column M on line 6.

#### 7.7.2.3 Duct Section FH

Using the same procedure as above, the closest standard size for 10,000 cfm at 0.30 in. wg friction loss per 100 ft is 26 in. diameter (Figure A-1). A velocity of 2700 fpm and the related $V_p$ of 0.45 are used for further calculations.

FH duct loss =

$$\frac{50 \text{ ft} \times 0.30}{100 \text{ ft}} = 0.150 \text{ in. wg (line 7)}$$

Using a 45 degree round wye fitting (Table A-11, Figure Y) with 45 degree elbows at F,

$\frac{V_{lb}}{V_c} = \frac{2700}{3200} = 0.84$;

$C$ = 0.28 (by interpolation).

F wye fitting loss =

$$C \times V_p = 0.28 \times 0.64 = 0.179 \text{ in. wg (line 8)}$$

The 45 degree round elbow (RD = 1.5) at F will use the same loss coefficient as the 90 degree elbow above (Table A-7, Figure A) multiplied by the 0.6 correction factor for 45 degree (Note 1). $V_p$ for 2700 fpm = 0.45 in. wg.

F elbow fitting loss =

$$0.15 \times 0.45 \times 0.6 = 0.041 \text{ in. wg (line 9)}$$

The 90 degree round elbow at G uses the same values without the correction factor.

G elbow fitting loss =

$$0.15 \times 0.45 = 0.068 \text{ in. wg (line 10)}$$

The losses in column L again are totaled in column M (0.438 in. wg)

#### 7.7.2.4 Duct Section HO

The following values are obtained using the same procedures as above (5000 cfm at 0.30 in. wg per 100 ft friction loss): 20 in. diameter duct size, 2300 fpm velocity. Note that the velocity continues to decrease as the volume (cfm) becomes lower using the same duct friction loss rate (0.30 in. wg per 100 ft).

HO duct loss =

$$\frac{40 \text{ ft} \times 0.30}{100 \text{ ft}} = 0.120 \text{ in. wg (line 11)}$$

At point H in the duct system, the branch coefficient is obtained for the diverging 45 degree round wye with a conical main and branch with a 45 degree elbow (Table A-11, Figure M):

$\frac{V_b}{V_c} = \frac{2300}{2700} = 0.85$ and $C$ = 0.51

$V_p$ = 0.45 (2700 fpm)

H wye (branch) loss =

$$C \times V_p = 0.51 \times 0.45 = 0.230 \text{ in. wg (line 12)}$$

The 90 degree round elbow is calculated as the above 90 degree ell and the loss coefficient for the balancing damper is obtained from Table A-15, Figure A ($\theta$ = 0 degree); $C$ = 0.20.

H damper loss =

$$C \times V_p = 0.20 \times 0.33 = 0.066 \text{ in. wg (line 13)}$$

N elbow fitting loss =

$$C \times Vp = 0.15 \times 0.33 = 0.050 \text{ in. wg (line 14)}$$

The total for the HO duct section (0.466 in. wg) is entered in column M.

#### 7.7.2.5 Duct Section OP

For 4000 cfm at 0.30 in. wg, the closest standard size duct is 18 in. diameter. Using the 18 in. duct, the friction rate then becomes 0.34 in. wg per 100 ft and the velocity is 2250 fpm.

OP duct loss =

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$$\frac{30 \text{ ft} \times 0.34}{100\,ft} = 0.102 \text{ in. wg (line 15)}$$

$V_p$ for 2250 fpm = 0.32 (Table A-4)

The 45 degree round diverging conical wye at point 0 (Table A-11, Figure C) requires that the "main" coefficient C be obtained from Table A-11A.

$\frac{V_s}{V_c} = \frac{2250}{2300} = 0.98$; but when there is no change in velocity, the table indicates that there is no dynamic loss. For example, C = 0 for $\frac{V_s}{V_c} = 1.0$. Interpolating gives a questionable loss coefficient of 0.004, which multiplied by the $V_p$ of 0.33 to give a loss of 0.001. However, a minimum loss coefficient of 0.01 is used.

0 Wye (main) loss =

$0.01 \times 0.33 = 0.003$ in. wg (enter on line 16).

The 60 degree transition from 20 in. diameter to 18 in. diameter does have a dynamic pressure loss and the fitting loss coefficient is obtained from Table A-9, Figure A.

$\frac{A_1}{A} = \frac{10^2\pi}{9^2\pi} = 1.23$, $C = 0.06$

0 transition loss =

$C \times W_p$ (downstream) = $0.06 \times 0.32 = 0.019$ in. wg (line 17)

The section loss of 0.124 in. wg is entered in column M on line 17.

### 7.7.2.6 Duct Section PQ

The same calculations as used in duct section OP are repeated using 3000 cfm and a 0.30 in. wg per 100 ft friction loss rate to obtain the closest standard duct size of 16 in. diameter (Figure A-1). Using the exact 16 in. duct size, the new velocity is 2150 fpm and the friction loss rate is 0.36 in. wg per 100 ft:

PQ duct loss =

$$\frac{30 \text{ ft} \times 0.36}{100 \text{ ft}} = 0.108 \text{ in. wg (line 18)}$$

$V_p$ for 2,150 fpm = 0.29 (Table A-4)

For the 45 degree round conical wye at P (Table A-11 A):

$\frac{V_s}{V_c} = \frac{2150}{2250} = 0.96$

$C$ = 0.01 (0.004 by interpolation),

P wye (main) loss =

$C \times V_p = 0.01 \times 0.32 = 0.003$ in. wg (line 19).

60 degree transition at P: $\frac{A_1}{A} = \frac{9^2\pi}{8^2\pi} = 1.27$ and

$C$ = 0.06

P transition loss =

$C \times V_p = 0.60 \times 0.29 = 0.017$ in. wg (line 20)

This section loss of 0.128 is entered in column M.

### 7.7.2.7 Duct Section QR

The selection of a standard 14 in. diameter duct for 2000 cfm (Figure 15-1) indicates a 0.32 in. wg per 100 ft friction loss rate and a velocity of 1880 fpm.

QR duct loss =

$$\frac{30 \text{ ft} \times 0.32}{100 \text{ ft}} = 0.096 \text{ in. wg (line 21)}$$

$V_p$ for 1,880 fpm = 0.22 (by calculation)

Again using the same type of wye at Q:

$\frac{V_s}{V_c} = \frac{1880}{2150} = 0.87$

$C$ = 0.01 (0.013 by interpolation)

Q wye (main) loss =

$C \times V_p = 0.01 \times 0.29 = 0.003$ in. wg (line 22)

Q Transition: $\frac{A_1}{A} = \frac{8^2\pi}{7^2\pi} = 1.31$, $C = 0.06$

Q Tran. Fitting loss =

$C \times V_p = 0.06 \times 0.22 = 0.013$ in. wg (line 23)

The section loss of 0.112 is entered in column M.

### 7.7.2.8 Duct Section RS

Using an additional duct sizing form to record the data in Table 7-3a, the 1000 cfm duct size at 0.30 in. wg would be half way between the 10 in. and 12 in. standard duct sizes. The 10 in. diameter duct would be in a much higher pressure-loss category, so the 12 in. duct at 0.19 in. wg per 100 ft and 1290 fpm velocity, would be the better selection.

RS duct loss =

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$$\frac{30 \text{ ft} \times 0.19}{100 \text{ ft}} = 0.057 \text{ in. wg}$$

Enter on line 1 of Table 7-3a

$V_p$ for 1290 fpm = 0.10 in. wg (by calculation)

R wye fitting: $\frac{V_s}{V_c} = \frac{1290}{1880} = 0.69$ $C = 0.04$

R wye (main) loss =

$C \times V_p = 0.04 \times 0.22 = 0.009$ in. wg (line 2)

R transition: $\frac{A_1}{A} = \frac{7^2\pi}{6^2\pi} = 1.36$ and $C = 0.06$

R transition loss =

$C \times V_p = 0.06 \times 0.10 = 0.006$ in. wg (line 3)

A 45 degree elbow at the end of the duct is connected to the VAV box by a 5 ft piece of 12 in. diameter flexible duct. The correction factor for the flexible duct is obtained from the chart in Figure A-2, using Table A-1 as a guide. Bends of 30 degree or more would also add additional resistance. Verified data is not available, so a radius elbow loss coefficient could be used to obtain the additional loss. Figure A-3 also contains a correction factor for unextended (compressed) flexible duct.

S 45 degree elbow fitting loss =

$0.15 \times 0.10 \times 0.60 = 0.009$ in. wg (line 4)

S flex. duct loss =

$$\frac{5 \text{ ft} \times 0.19}{100\,ft} \times 1.95 \text{ (rough)} = 0.019 \text{ in. wg (line 5)}$$

An estimate of 0.25 in. wg is made for the downstream-side ductwork and diffuser from the VAV box. This is added to the VAV box pressure loss of 0.30 in. wg and the total (0.55 in. wg) is entered on line 6. The total pressure loss of 0.650 for the RS section is entered in columns M and N, and also in Column N on line 24 on page 1 (Table 7-3) of the duct-sizing work sheet. Working from the bottom to the top of the form, the section pressure losses are totaled in column N, with the total pressure loss for the supply duct system of 2.875 in. wg being entered on line 1 in column 0.

#### 7.7.2.9 Review

The same procedure is used to size the other segments of the supply duct system; or, if the layout is symmetrical, the same sizes can be used for similar segments of the system. However, as was found in the supply air duct system sizing Example 1, several fittings with higher pressure losses or "high loss" VAV boxes can allow a duct run that was not the originally selected for design computations to be the duct run with the greatest pressure loss.

Assuming that the return air duct system of Example 3 (not shown) had an approximate total-pressure loss of 2.0 in. wg, the output of the system supply fan would need to be 20,000 cfm at 4.88 in. wg (2.875 + 2.0). Attention is called to the fact that although the fan total pressure requirements are in the upper portion of the duct-pressure classification range, all of the supply air duct system beyond the wye fitting at F, is in the low pressure range (under 2 in. wg), even though there are velocities up to 2700 fpm.

It is extremely important to indicate static pressure "flags" on the drawings after the duct system is sized (as in indicated in Figure 7-2). Table 2-5 indicates the relative costs of fabrication and installation of the different pressure classes of ductwork for the same size duct. The initial installation cost savings become quite apparent by this simple procedure, especially when the system designer specifies a higher-pressure duct construction classification for duct systems, when a lower classification would be more than adequate.

### 7.8 EXTENDED PLENUM DUCT SIZING

In the design of air-distribution duct layouts, a design variation commonly referred to as "extended plenum" or "semi-extended plenum" is often incorporated into the duct sizing method being employed.

An *extended plenum* is a trunk duct of constant size, usually at the discharge of a fan, fan-coil unit, mixing box or variable air volume (VAV) box extended as a plenum to serve multiple outlets and branch ducts.

A *semi-extended plenum* is a trunk design system utilizing the concept of extended plenum that incorporates a minimum number of size reductions to compensate for the decreasing air volume.

#### 7.8.1 Properties

Some of the advantages realized through the use of the semi-extended plenum system concept are:

a. Lower first-cost due to an improved ratio of straight duct to fittings.

b. Lower operating cost due to savings in fan horsepower by elimination of high energy loss transition fittings.

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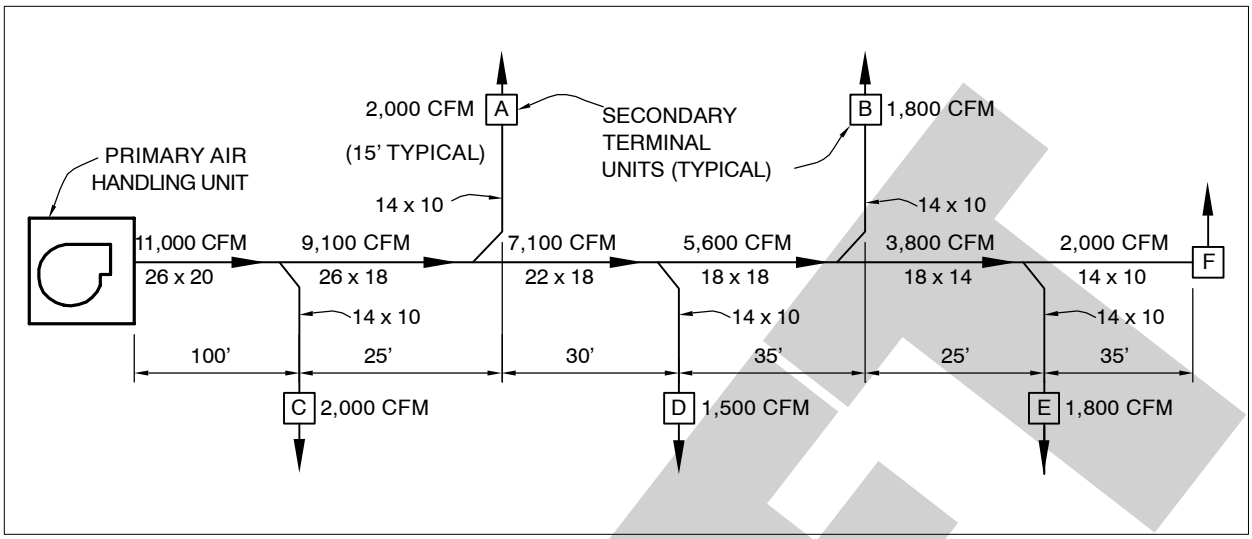


**FIGURE 7–3 SYSTEM "A" – SIZED BY EQUAL FRICTION METHOD**

c. Ease of balancing due to low branch take-off pressure losses and fewer trunk duct pressure changes.

As long as design-air volume is not exceeded, branch ducts can be added, removed, and re-located at any convenient point along the trunk duct, between size reductions, without affecting performance. This is particularly useful in "tenant development" work.

A limiting factor to be considered when using the extended plenum method is that undesirably low velocities can develop. This can result in excessive heat gain or loss through the duct walls compared to the volume of air handled.

### 7.8.2 Design Criteria

Actual installations and tests indicate that semi-extended plenum design is acceptable for use with system static pressures that range from 1 in. wg through 6 in. wg, and duct velocities up to 3000 fpm. Other specific design considerations include:

a. Branch takeoffs from the trunk duct should preferably be round duct connecting at a 45 degree angle. If rectangular branches are used, a 45 degree entry tap should be used.

b. Branch takeoff velocities should range between 55 and 90 percent of the trunk duct velocity to minimize static pressure loss across the takeoff.

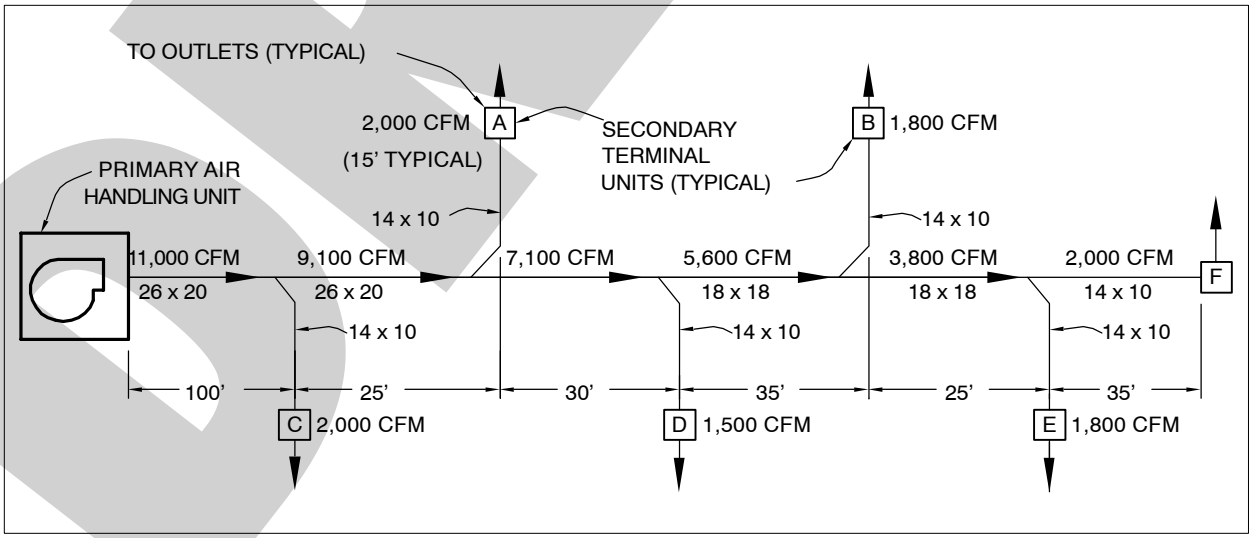


**FIGURE 7–4 SYSTEM "B" – MODIFIED BY SEMI–EXTENDED PLENUM CONCEPT**

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| System | Duct Losses (in. wg) | Fitting Losses (in. wg) | Total Losses (in. wg) | Fan Bhp Required |
|---|---|---|---|---|
| "A" - equal friction method | 1.27 | 0.66 | 1.93 | 6.3 |
| "B" semi-extended plenum method | 1.08 | 0.22 | 1.30 | 4.6 |

**Table 7–4 Semi–Extended Plenum Comparison**

| Description | System "A" Equal Friction Method | | | System "B" - Semi-Extended Plenum Method | | |
|---|---|---|---|---|---|---|
| | lbs. | Shop Labor | Field Labor | lbs. | Shop Labor | Field Labor |
| Straight Duct | 2394 | 12 | 61 | 2640 | 13 | 64 |
| Fittings | 314 | 16 | 27 | 68 | 3 | 4 |
| Totals | 2708 | 28 hours | 88 | 2758 | 26 hours | 70 |

**Table 7–5 Semi–Extended Plenum Installation Cost Comparison**

c. Branch velocities should not exceed the trunk duct velocity.

d. Balancing dampers should be installed in each branch duct.

### 7.8.3 Comparison of Design Methods

Figures 7-3 and 7-4 illustrate identical medium-pressure systems differing only in the trunk duct sizing techniques used. The trunk duct system shown in Figure 7-3 has been sized by the equal- friction method at a pressure loss of approximately 0.5 in. wg per 100 ft. Note that reducing fittings have been used at each branch takeoff.

The semi-extended plenum "concept" in Figure 7-4, has been used to keep duct reductions at a minimum. Note that System "A" utilizes six trunk-duct sizes and five reducing fittings, while System "B" has only three duct sizes and two reducing fittings. Assuming that the duct between the primary air handling unit and secondary terminal unit "F" has the highest supply pressure loss, and using friction-loss data from the Appendix, the results are tabulated in Table 7-4. Ignoring branch duct and outlet losses, which are identical for both systems, the semi-extended plenum system has a 0.63 in. wg with a 1.93 – 1.30 lower pressure loss than the system sized by the equal-friction method.

The fan power necessary to satisfy the supply pressure requirements selected from a typical manufacturer's catalog is also shown in Table 7-4. It can be seen that the semi-extended plenum design results in reduced fan power and lower operating costs. The cost savings, both first and operating, could be even greater with a return air duct system utilizing the semi-extended plenum concept.

### 7.8.4 Cost Comparison

Although energy conservation is extremely important, installation costs are still the primary concern to the designer, the contractor, and the owner. Table 7-5 illustrates the estimated installation cost comparison between the two systems analyzed. It shows how the overall installed cost for the semi-extended plenum system is appreciably less.

The utilization of an extended or semi-extended plenum is not actually a different method of duct or system sizing. It is merely the combination of good design and cost savings ideas using conventional duct sizing techniques.

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DUCT SIZING WORK SHEET
(U.S. UNITS)

DATE _____ PAGE _____

PROJECT _____ LOCATION _____ SYSTEM

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW CFM | FRIC-TION PER 100′ | VELOC-ITY FPM | $V_P$ | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMULA-TIVE LOSS | TOTAL LOSS |
| 1 | | | | | | | | | | | | | | | | |
| 2 | | | | | | | | | | | | | | | | |
| 3 | | | | | | | | | | | | | | | | |
| 4 | | | | | | | | | | | | | | | | |
| 5 | | | | | | | | | | | | | | | | |
| 6 | | | | | | | | | | | | | | | | |
| 7 | | | | | | | | | | | | | | | | |
| 8 | | | | | | | | | | | | | | | | |
| 9 | | | | | | | | | | | | | | | | |
| 10 | | | | | | | | | | | | | | | | |
| 11 | | | | | | | | | | | | | | | | |
| 12 | | | | | | | | | | | | | | | | |
| 13 | | | | | | | | | | | | | | | | |
| 14 | | | | | | | | | | | | | | | | |
| 15 | | | | | | | | | | | | | | | | |
| 16 | | | | | | | | | | | | | | | | |
| 17 | | | | | | | | | | | | | | | | |
| 18 | | | | | | | | | | | | | | | | |
| 19 | | | | | | | | | | | | | | | | |
| 20 | | | | | | | | | | | | | | | | |
| 21 | | | | | | | | | | | | | | | | |
| 22 | | | | | | | | | | | | | | | | |
| 23 | | | | | | | | | | | | | | | | |
| 24 | | | | | | | | | | | | | | | | |
| 25 | | | | | | | | | | | | | | | | |

**FIGURE 7–5 DUCT SIZING WORK SHEET (I–P)**

Note: * Indicates duct lining used. Sizes are interior dimensions.

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DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE _____ PAGE 1 OF 2

PROJECT SAMPLE PROJECT LOCATION FIRST FLOOR SYSTEM SUPPLY AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW (L/S) | FRICTION PER METER | VELOC-ITY (M/S) | $V_p$ (PA) | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMU-LATIVE LOSS | TOTAL LOSS |
| 1 | PLENUM B | $A_1$ | – | E.R. DEVICE | 4000 | – | 2.5 | – | – | – | – | – | 75 | | 362 | 362 |
| 2 | ” | $A_2$ | – | FILTERS | 4000 | – | 2.0 | – | – | – | – | – | 40 | | | |
| 3 | ” | $A_3$ | – | COIL | 4000 | – | 2.8 | – | – | – | – | – | 110 | 225 | | |
| 4 | RUN BJ | BC | 3m | DUCT | 4000 | 0.95 | 9.0 | – | – | 755 | *600 × 800 | 1.35 | 4 | 4 | 137 | |
| 5 | ” | CE | 10m | DUCT | 4000 | 0.95 | 9.0 | – | – | 755 | *1300 × 400 | 1.35 | 13 | | 133 | |
| 6 | ” | C | – | TRANS | 4000 | – | 7.7 | 35.7 | 0.25 | – | 600 × 800 / 1300 × 400 | – | 9 | | | |
| 7 | ” | D | – | FIRE DAMP | 4000 | – | – | – | – | – | 1300 × 400 | – | 15 | 37 | | |
| 8 | ” | EF | 6m | DUCT | 3000 | 0.95 | 8.4 | – | – | 676 | 1000 × 400 | – | 6 | | 96 | |
| 9 | ” | E | 90° | WYE | 4000 / 3000 | – | 7.7 | 35.7 | –0.05 | – | 1300 × 400 / 1000 × 400 | – | –2 | 4 | | |
| 10 | ” | FH | 10m | DUCT | 1500 | 0.95 | 7.0 | – | – | 510 | 550 × 400 | – | 10 | | 92 | |
| 11 | ” | F | 90° | WYE | 3000 / 1500 | – | 7.5 | 33.9 | 0.05 | – | 1000 × 400 / 550 × 400 | – | 2 | | | |
| 12 | ” | F | – | VOL DAMP | 1500 | – | 6.8 | 27.8 | 0.04 | – | 550 × 400 | – | 1 | | | |
| 13 | ” | G | 90° | ELBOW | 1500 | – | 6.8 | 27.8 | 0.15 | – | 550 × 400 | – | 4 | 17 | | |
| 14 | ” | HI | 6m | DUCT | 1000 | 0.95 | 6.4 | – | – | 456 | 450 × 400 | – | 6 | | 75 | |
| 15 | ” | H | – | TRANS | 1500 / 1000 | – | 6.8 | 27.8 | 0.05 | – | 550 × 450 / 450 × 400 | – | 1 | 7 | | |
| 16 | ” | IJ | 10m | DUCT | 500 | 0.95 | 5.4 | – | – | 340 | 400 × 250 | – | 10 | | 68 | |
| 17 | ” | I | – | TRANS | 1000 / 500 | – | 5.6 | 18.9 | 0.05 | – | 450 × 400 / 400 × 250 | – | 1 | | | |
| 18 | ” | J | 90° | ELBOW | 500 | – | 5.0 | 15.1 | 0.17 | – | 400 × 250 | 1.29 | 3 | | | |
| 19 | ” | J | – | VOL DAMP | 500 | – | 5.0 | 15.1 | 0.04 | – | 400 × 250 | – | 1 | | | |
| 20 | ” | J | – | TAP FIT | 500 | – | 5.0 | 15.1 | 1.2 | – | 400 × 250 / 350 | – | 18 | | | |
| 21 | ” | J | – | DIFF. | 500 | – | – | – | – | – | 350 | – | 35 | 68 | | |
| 22 | | | | | | | | | | | | | | | | |
| 23 | | | | | | | | | | | | | | | | |
| 24 | | | | | | | | | | | | | | | | |
| 25 | | | | | | | | | | | | | | | | |

**Table 7–6 Duct Sizing, Supply Air System – Example 1**

Note: * Indicates duct lining used. Sizes are interior dimensions.

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DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE _____ PAGE 2 OF 2

PROJECT SAMPLE PROJECT LOCATION FIRST FLOOR SYSTEM SUPPLY AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW (L/S) | FRICTION PER METER | VELOC-ITY (M/S) | $V_p$ (PA) | LOSS COEFF | EQUIV DIAM | RECTANGULAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMU-LATIVE LOSS | TOTAL LOSS |
| 1 | $A_1F$ | – | – | TOTAL OF LINES 1 TO 9 IN TABLE 7–6 | | | | | | | | | – | 270 | 367 | 367 |
| 2 | RUN FM | F | 90° | WYE | 3000 / 1500 | – | 7.5 | 33.9 | 0.52 | – | 1000 × 400 / 550 × 400 | – | 18 | | 97 | |
| 3 | ” | F | – | VOL DAMP | 1500 | – | 6.8 | 27.8 | 0.04 | – | 550 × 400 | – | 1 | | | |
| 4 | ” | FK | 3m | DUCT | 1500 | 0.95 | 7.0 | – | – | 510 | 550 × 400 | – | 3 | 22 | | |
| 5 | ” | KM | – | TOTAL OF LINES 14 TO 21 IN TABLE 7–6 | | | | | | | | | – | 75 | 75 | |
| 6 | | | | | | | | | | | | | | | | |
| 7 | $A_1E$ | – | – | TOTAL OF LINES 1 TO 7 IN TABLE 7–6 | | | | | | | | | – | 266 | 348 | 348 |
| 8 | RUN EQ | EN | 3m | DUCT | 1000 | 0.95 | 6.4 | – | – | 456 | 450 × 400 | – | 3 | | 82 | |
| 9 | ” | E | 90° | WYE | 4000 / 1000 | – | 7.7 | 35.7 | 0.44 | – | 1300 × 400 / 450 × 400 | – | 16 | 19 | | |
| 10 | ” | NP | 17m | DUCT | 500 | 0.95 | 5.4 | – | – | 340 | 400 × 250 | – | 16 | | 63 | |
| 11 | ” | N | – | TRANS | 1000 / 500 | – | 5.6 | 18.9 | 0.05 | – | 450 × 400 / 400 × 250 | – | 1 | | | |
| 12 | ” | N | – | VOL DAMP | 500 | – | 5.0 | 15.1 | 0.04 | – | 400 × 250 | – | 1 | | | |
| 13 | ” | O | 90° | ELBOW | 500 | – | 5.0 | 15.1 | 0.12 | – | 400 × 250 | – | 2 | 20 | | |
| 14 | ” | PQ | 6m | DUCT | 250 | 0.95 | 4.6 | – | – | 270 | 250 × 250 | – | 6 | | 43 | |
| 15 | ” | P | – | TRANS | 500 / 250 | – | 5.0 | 15.1 | 0.06 | – | 400 × 250 / 250 × 250 | – | 1 | | | |
| 16 | ” | Q | 90° | ELBOW | 250 | – | 4.0 | 9.6 | 0.90 | – | 250 × 250 / 400 × 250 | 1.08 | 4 | | | |
| 17 | ” | Q | – | GRILLE | 250 | – | – | – | – | – | 400 × 250 | – | 32 | 43 | | |
| 18 | A,N | – | – | TOTAL OF LINES 7 TO 9 ABOVE | | | | | | | | | – | 285 | 346 | 346 |
| 19 | RUN NS | NR | 3m | DUCT | 500 | 0.95 | 5.4 | – | – | 340 | 400 × 250 | – | 3 | | 61 | |
| 20 | ” | N | 45° | ENT. TAP | 1000 / 500 | – | 5.6 | 18.9 | 0.74 | – | 450 × 400 / 400 × 250 | – | 14 | | | |
| 21 | ” | N | – | VOL DAMP | 500 | – | 5.0 | 15.1 | 0.04 | – | 400 × 250 | – | 1 | 18 | | |
| 22 | ” | RS | 6m | DUCT | 250 | 0.95 | 4.6 | – | – | 270 | 250 × 250 | – | 6 | | 43 | |
| 23 | ” | R | – | TRANS | 500 / 250 | – | 5.0 | 15.1 | 0.06 | – | 400 × 250 / 250 × 250 | – | 1 | | | |
| 24 | ” | S | 90° | ELBOW | 250 | – | 4.0 | 9.6 | 0.90 | – | 250 × 250 / 400 × 250 | 1.08 | 4 | | | |
| 25 | ” | S | – | GRILLE | 250 | – | – | – | – | – | 400 × 250 | – | 32 | 32 | | |

**Table 7–6(a) Duct Sizing, Supply Air System – Example 1 (Continued)**

Note: *Indicates duct lining used. Sizes are interior dimensions.

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## 7.9 DESIGN FUNDAMENTALS (SI)

The easiest way the HVAC system designer can size a duct system using the metric system is to "think metric" throughout the complete design procedure. To make matters easier, the duct fitting loss coefficient "C" is dimensionless,. Therefore, it is applicable to both the inch-pound (I-P) and the metric (SI) measurement systems. Correction factors also are dimensionless, but sometimes they must be adjusted to the measuring system being used because of the "constant" number in the equation.

The following examples are in the same general range of values as the examples given for I-P Units. However, they are not "soft conversions", like the numbers multiplied by the conversion factors found in the Appendix, "Metric Units and Equivalents". For example, dividing 3.5 × 0.004022 in. wg (1 Pa), the answer would equal 870 (875 Pa), a rounded off number.

Some easy to remember "round number" conversions generally used to check calculations when exact conversions are not required are:

2 cfm = 1 liter per second (1 L/s)

200 fpm = 1 meter per second (1 m/s)

1 in. wg = 250 Pascals (250 Pa)

1 in. = 25 millimeters (25 mm)

Some duct friction loss charts being used by HVAC industry use "mm wg/m" (millimeters water gage per meter) instead of "Pa/m". One Pa equals 0.1022 mm water gage, so for practical purposes: 1 Pa = 0.1 mm wg, which is an easy conversion. Also, 1000 L/s equals 1 $m^3/s$, a unit used for airflow volume in some parts of the world. All other needed metric tables, conversions, and equations can be found in the Appendix.

## 7.10 SUPPLY AIR DUCT SYSTEM – SIZING EXAMPLE. 1 (SI)

A plan of a sample building HVAC duct system is shown in Figure 7-6 and the tabulation of the computations can be found in Table 7-6. A full size Duct Sizing Work Sheet may be found in Figure 7-9 at the end of the Chapter. It may be photocopied for "in-house" use only. The conditioned area is assumed to be at zero pressure and the two fans have been sized to deliver 4000 L/s each. The grilles and diffusers have been tentatively sized to provide the required flow, throw, and noise level and the sizes and pressure drops are indicated on the plan. To size the ductwork and determine the supply fan total pressure requirement, a suggested step-by-step procedure follows.

### 7.10.1 Supply Fan Plenum (SI)

From manufacturer's data sheets or the Figures or Tables in the Appendix, the static-pressure losses of the energy recovery device, filter bank, and heating-cooling coil are entered in Table 7-6 in column L. Velocities, if available, are entered in column F for reference information only. With 3 meters of duct discharging directly from fan "B," duct is fan outlet size, no System-Effect Factor, needs to be added for either side of the fan. As the plenum static-pressure (SP) loss is negligible, the losses for the inlet-air portion of the fan system entered in column L are added, the loss of 225 Pascals (Pa) is entered in column M on line 3.

### 7.10.2 Supply Air System (SI)

#### 7.10.2.1 Duct Section BC

The 600 × 800 mm fan discharge size has a circular equivalent of 755 mm, *see* Table A-2M. Using the chart in Figure A-1M, a velocity of 9.0 m/s and a friction loss of 0.95 Pa/m is established within the recommended velocity range (shaded area) using the 4000 L/s system airflow. The data is entered on line 4 in the appropriate columns. Without any changes in direction to reduce the fan noise, and with the duct located in an unconditioned space up to the first branch (at point E), internal fibrous-glass lining can be used to satisfy both the acoustic and thermal requirements. Therefore, the duct size of 600 × 800 mm entered in column J is marked with an asterisk and the fibrous-glass liner "medium rough" correction factor of 1.35 is obtained from Table A-1 and Figure A-2M and entered in column K. Duct section BC static pressure (SP) loss is computed as follows:

SP (duct section) = 3m (duct) × 0.95 Pa/m × 1.35 (correction factor) = 3.8 (use 4).

The duct section BC static pressure loss is entered in column L and, as it is the only loss for that section, the loss also is entered in column M.

#### 7.10.2.2 Duct Section CE

At point C, the building construction conditions require the duct aspect ratio to change, a duct transition is needed. Using the same 0.95 Pa/m duct friction loss and 755 mm duct diameter for the 4000 L/s or 4.0 $m^3/s$ airflow, a 1300 × 400 mm duct is selected from Table A-2M and entered in column J on line 5. This section

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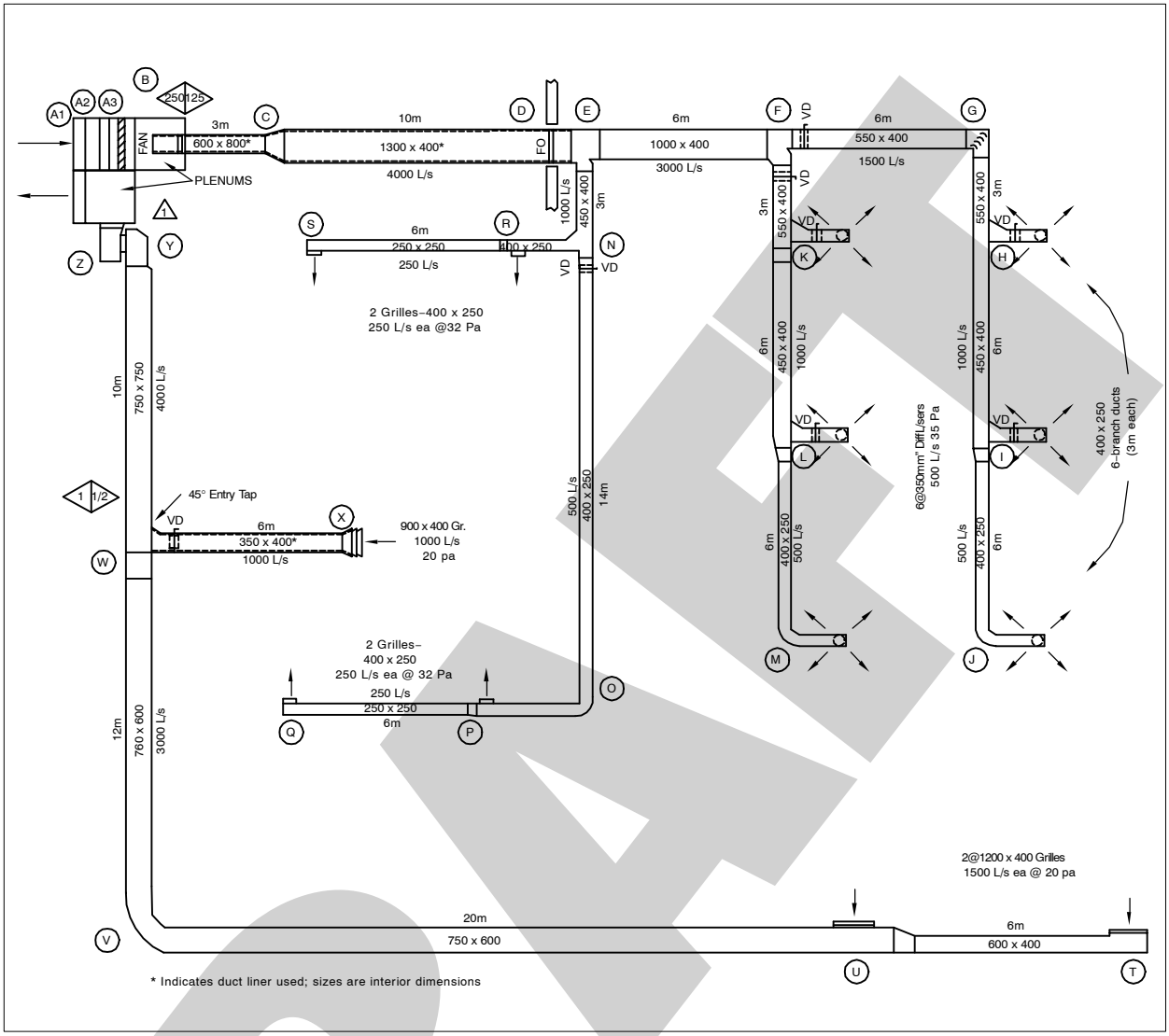


**FIGURE 7–6M DUCT SYSTEMS FOR DUCT SIZING EXAMPLES 1 AND 2 (SI)**

of duct continues to require acoustical and thermal treatment so the section friction loss is computed:

SP = 10 × 0.95 × 1.35 = 12.8 Pa

(enter 13 Pa on line 5 in column L).

The transition loss coefficient can be obtained after determining if the fitting is diverging or converging.

A = 600 × 800 = 480,000

$A_1$ = 1300 × 400 = 520,000

$\frac{A_1}{A} = \frac{520,000}{480,000} = 1.08$, so it is diverging (greater than 1.0).

The average velocity of the entering airstream (Equation 5-9)

$$V = \frac{Q}{A} = \frac{4.0\ \text{m}^3/\text{s}}{1.3\ \text{m} \times 0.4\ \text{m}} = 7.7\ \text{m/s}$$

From Table A-8, Figure B, using θ = 30 degree and $A_1/A$ = 2 (smallest number for $A_1A$), the loss coefficient of 0.25 is entered on line 6 in column H. The velocity pressure ($V_p$) of 35.7 Pa is calculated using Equation 5-10 ($V_p = 0.602\ V^2$) or is obtained from Table A-4M for 7.7 m/s and entered in column G. The transition fitting pressure loss of 9 Pa ($C \times V_p = 0.25 \times 35.7 = 8.9$) is entered in column L. As this is a dynamic pressure loss, the correction factor for the duct lining does not apply.

The static pressure loss of 158 Pa for the fire damper at D is obtained from Chapter 8 or manufacturer's data

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sheets, and entered in column L on line 7. The three static pressure losses in column L on lines 5, 6, and 7 are totaled (37 Pa) and entered in column M on line 7. This is the total pressure loss for the 1300 × 400 mm duct section CE (inside dimensions) and its components.

#### 7.10.2.3 Duct Section EF

An assumption must now be made as to which duct run has the greatest friction loss. As the duct run to the "J" air supply diffuser is apparently the longest with the most fittings, this run will be the assumed path for further computations. Branch duct run EQ will be compared with duct run EJ after calculations are completed.

Applying 3000 L/s (for duct section EF) and 0.95 Pa/m to the chart in Figure A-1M, a duct diameter of 676 mm and 8.4 m/s velocity are obtained and entered on line 8. Table A-2M is used to select a 1000 × 400 mm rectangular duct size, by keeping the duct height 400 mm. Normally, duct size adjustments are made changing only one dimension for ease and economy of fabrication.

As the galvanized duct system is being fabricated in 1,200 mm sections, the degree of roughness (Table A-1) indicates "medium smooth". No correction factor is needed; as the chart in Figure A-1 based on an Absolute Roughness of 0.09 mm.

The static pressure loss for duct section EF is:

SP = 6 m × 0.95 = 5.7 Pa (use 6)

(enter on line 8 column L).

The diverging 90 degree wye fitting used at E can be found in Table A-11, Figure W. In order to obtain the proper loss coefficient "C" to calculate the fitting pressure loss, preliminary calculations to obtain $A_b$ must be made. If different friction loss rate is used later when computing the branch losses, subsequent recalculation might be necessary.

$A_b$ (Preliminary) for 1000 L/s at 0.95 Pa/m = 196,350 $mm^2$, area of 450 mm diameter duct obtained from Figure A-1M. Then:

$$\frac{A_b}{A_s} = \frac{(225)^2\pi}{(338)^2\pi} = \frac{159,043}{358,908} = 0.44$$

$$\frac{A_b}{A_s} = \frac{(225)^2\pi}{(378)^2\pi} = \frac{159,043}{448,883} = 0.35$$

$$\frac{Q_b}{Q_c} = \frac{1000}{4000} = 0.25$$

$$\text{Velocity} = \frac{Q}{A} = \frac{4.0\ \text{m}^3/\text{s}}{1.3\ \text{m} \times 0.4\ \text{m}} = 7.7\ \text{m/s}$$

(Equation 5-9).

Using $\frac{A_b}{A_s} = 0.5$

$\frac{A_b}{A_c} = 0.5$ (the closest figures), $C$ (Main) = – 0.05

The $V_p$ for 7.7 m/s is 35.7 Pa (Equation 5-10)

The fitting "loss" has a negative value (TP = C × $V_p$ = – 0.05 × 35.7 = – 1.79). Minus 2 Pa is entered on line 9 in column L with a minus sign. The static regain is actually greater than the dynamic pressure loss of the fitting. The pressure losses on lines 8 and 9 in column L are added (– 2 + 6 = 4 Pa) and entered on line 9 in column M.

#### 7.10.2.4 Duct Section FH

The wye fitting at F and duct section FH are computed in the same way as above and the values entered on lines 10 and 11. By using 0.95 Pa/m and 1500 L/s in Figure A-1M, 7 m/s and 510 mm diameter are obtained from Figure A-1M; 550 × 400 mm equivalent duct size from Table A-2M: FH duct section loss = 10 × 0.95 = 9.5 Pa, (enter 10 Pa on line 10).

For the wye fitting at F, Table A-11, Figure W is again used. With the 3000 L/s airflow dividing equally into two 1,500 L/s airstream ducts, $A_b = A_s$. Therefore,

$$\frac{A_b}{A_c} = \frac{(225)^2\pi}{(338)^2\pi} = \frac{204,282}{358,908} = 0.57$$

$$\frac{Q_b}{Q_c} = \frac{1500}{3000} = 0.5$$

Using $\frac{A_b}{A_s} = 1.0$

$\frac{A_b}{A_c} = 0.5$

$C$ (Main) = 0.05

$$\text{Velocity} = \frac{3}{1.0 \times 0.4} = 7.5\ \text{m/s}$$

$Vp$ = 33.9 Pa (from Table A-4M)

Fitting loss =

$$C \times V_p = 0.05 \times 33.9 = 1.70\ \text{Pa}$$

(enter 2 Pa on line 11).

The loss coefficient for the thin-plate volume damper near F can be obtained from Table A-15, Figure B, set

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wide open or 0 degree. The velocity pressure ($V_p$) of 27.8 Pa or 6.8 m/s $\left(\frac{1.5}{0.55} \times 0.4\right)$ is obtained from Table A-4M or calculated.

Damper Loss =

$C \times V_p = 0.04 \times 27.8 = 1.1$ Pa (use 1 Pa on line 12).

Elbow G in the FH duct run is a square elbow with 114 mm single-thickness turning vanes on 57 mm centers. The loss coefficient of 0.15 is obtained from Table A-7, Figure H for the 550 × 400 mm elbow, and entered on line 13 along with the other data (cfm, fpm, $V_p$).

G fitting loss =

$C \times V_p = 0.15 \times 27.8 = 4.2$ Pa (use 4 Pa on line 13).

The total pressure loss for duct section FH from lines 10, 11, 12, and 13 in column L (10 + 2 + 1 + 4 = 17 Pa) is entered on line 13 in column M.

#### 7.10.2.5 Duct Section HI

Data for duct section HI is developed as other duct sections above. Starting with 1000 L/s, the values of 6.4 m/s, 456 mm diameter, and the duct size of 450 × 400 mm are obtained again changing only one duct dimension.

HI duct section loss =

6 m × 0.95 Pa/m = 5.7 Pa (use 6 Pa on line 14).

The loss coefficient for transition H (converging flow) is obtained from Table A-9, Figure A using θ = 30 degree. Use the upstream velocity based on 1500 L/s to compute the $V_p$, assuming that there is not an instant change in the upstream airflow velocity. This will hold true for each similar fitting in this example.

Velocity = $\frac{1.5}{0.55 \times 0.40} = 6.8$ m/s

Velocity pressure ($V_p$) = 27.8 Pa

$\frac{A_1}{A} = \frac{500 \text{ mm} \times 400 \text{ mm}}{450 \text{ mm} \times 400 \text{ mm}} = 1.22$ (use 2)

$C = 0.05$

H fitting loss =

$C \times V_p = 0.05 \times 27.8 = 1.4$ Pa (use 1 Pa on line 15)

The loss values in column L (6 + 1) are again totaled and entered on line 15 in column M (7 Pa).

#### 7.10.2.6 Duct Section IJ

Duct section IJ is calculated as the above duct sections, and the same type of transition is used (500 L/s, 5.4 m/s, 340 mm diameter, with a 400 × 250 mm duct size being selected):

IJ duct loss =

10 m × 0.95 Pa/m = 9.5 Pa

(enter 10 Pa on line 16).

It is obvious by now that using a duct friction loss of 0.95 to 1.0 Pa/m, makes the calculations quite simple, since there is 1 Pa pressure loss for each meter of duct!

Transition at I (Table A-9, Figure A):

$\frac{A_1}{A} = \frac{450 \times 400}{400 \times 250} = 1.8$ (use 2) and $C = 0.05$

Velocity = $\frac{1.0}{0.45 \times 0.4} = 5.6$ m/s

$V_p = 18.9$

I fitting loss =

$C \times V_p = 0.05 \times 18.9 = 0.95$ Pa (use 1 Pa on line 17)

The "J" elbow is a smooth long-radius type without vanes (Table A-7, Figure F) having a R/W ratio of 2.0. As $\frac{H}{W} = \frac{250}{400} = 0.63$, the loss coefficient of 0.17 (by interpolation) is used.

By applying the values of the 340 mm duct diameter and the duct velocity of 5.6 j/s to the *Reynolds Number Correction-Factor Chart*, we find a correction factor must be used. The actual average velocity is:

$V = \frac{0.5}{0.4 \times 0.25} = 5.0$ m/s (Equation 5-9)

The equations are solved to allow the correction factor to be obtained.

$D = \frac{2HW}{H + W} = \frac{2 \times 400 \times 250}{400 + 250} = 307.7$ mm

$R_e = 66.4$ DV = $66.4 \times 307.7 \times 5.0$

$R_e = 102{,}156$

$R_e 10^{-4} = 10.22$

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From the table when $\frac{R}{W} > 0.75$, the correction factor of 1.29 is obtained and a $V_p$ of 15.1 Pa for 5 m/s is used.

Fitting loss =

$$C \times V_p \times K_{Re} = 0.17 \times 15.1 \times 1.29 = 3.31 \text{ Pa}$$

(3 Pa is entered on line 18)

If the $K_{Re}$ correction factor was not used, the calculated loss of 2.57 Pa ($0.17 \times 15.1$) is 0.74 Pa lower than the value used. On a long, winding duct run with many elbows, this difference could become significant. However, when both are rounded to 3 Pa, the difference would not be noted.

The volume damper at J has the same coefficient as that used at F. Using the $V_p$ for 5 m/s:

Damper Loss =

$$C \times V_p = 0.04 \times 15.1 = 0.60 \text{ Pa}$$

(enter 1 Pa on line 19).

Figure T of Table A-11 (Tee, Rectangular Main to Round Branch) should not be used for a round tap at the end of a duct run, nor should Figure Q for a square tap under the same conditions, as the total system airflow is going through the tap. The closest duct configurations found in the Appendix would be the mitered elbows in Table A-7, Figures C, D, or E. The average loss coefficient value for a 90 degree turn from these figures is 1.2.

Obviously, if there was ample room in the ceiling, the use of a vaned- or a long-radius elbow and a rectangular-to-round transition, would be the most energy efficient with the lowest combined pressure loss. Therefore, the loss of the fitting at the diffuser should be calculated using the loss coefficient of 1.2:

Fitting Loss =

$$C \times V_p = 1.2 \times 15.1 = 18.1 \text{ Pa}$$

(enter 18 Pa on line 20).

The pressure loss of the 350 mm diameter diffuser in Figure 7-6 at J includes the pressure losses for the damper behind the diffuser. The 35 Pa loss is entered on line 21 in column L.

In Table 7-6, the pressure losses on lines 16 through 21 in column L are totaled (68 Pa) and the value entered on line 21 in column M and on line 16 in column N.

Starting from the bottom (line 16), the pressure losses of each section in column M are accumulated in column N on line 4, resulting in a total pressure loss of 137 Pa for the duct run B to J (the assumed longest duct run) being entered on line 4. This total is added to the 225 Pa on line 3 of column M (Fan Plenum B) for the total pressure loss of 362 Pa, the design total pressure at which supply fan B must supply 4000 L/s. The value of 362 Pa is entered on line 1 in column N and O. The numbers in columns N and O are calculated after columns A to M.

Attention is called to the progressively lower value of the velocity pressure, as the velocity continues to be reduced (velocity pressure is proportional to the square of the velocity). By carefully selecting fittings with low-loss coefficients, actual dynamic pressure-loss values become quite low. However, straight duct loss values remain constant, as these losses are dependent only on the friction loss rate selected.

The last section of duct (IJ), with all of its fittings and the terminal device, had half of the pressure loss generated by the complete duct run (BJ). The primary reason for this is all of the fittings in the main run had a static regain, included in the loss coefficients. Each lowering of the airstream velocity reduced the actual pressure loss of each section.

### 7.10.2.7 Duct Section FM

As the branch duct run F to M is similar to duct run G to J, one would assume that the duct sizes would be the same, provided that the branch pressure loss of the wye at F had approximately the same pressure loss as the 6 m of duct from F to G (6 Pa) and the elbow at G (4 Pa), for a total loss of 10 Pa. However, to compute the complete duct run from $A_1$ to $M_1$, lines 1 to 9 ($A_1$ to F) in column M must be totaled (270 Pa) and the result entered on line 1 (column M) of the table in Figure 7-6(a) using a new duct sizing form.

Referring again to Table A-11, Figure W (used before for the wye at F), and using the same ratios as before, $\left(\frac{A_b}{A_s} = 1.0;\ \frac{A_b}{A_c} = 0.5;\ \frac{Q_b}{Qc} = 0.5\right)$, the branch loss coefficient C = 0.52.

F fitting loss =

$$C \times V_p = 0.52 \times 33.9 = 17.6 \text{ Pa}$$

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(enter 18 Pa on line 2)

It should be noted that the fitting entering velocity of 7.5 m/s is used to determine the velocity pressure for the computations. The branch loss of 18 Pa for fitting F is compared to the 10 Pa computed above for duct EG and elbow G. As the difference of 8 Pa is within the 12 Pa allowable design difference, the fitting used at F was a good selection. However, the $A_1$M duct run will have a pressure loss of 8 Pa greater than the $A_1$J duct run. The assumed "longest run" did not have the greatest pressure loss, although again the difference was within 12 Pa. This also confirms the need for using balancing dampers in each of the 550 × 400 mm ducts at F.

The information for the "branch" volume damper at F can be copied from line 12 of Table 7-6 since all conditions are the same, and entered on line 3 of Table 7-6a. The calculations are then made for the 3 meters of 550 × 400 mm duct from F to K:

FK duct loss = 3 × 0.95 = 2.9 Pa

(enter 3 Pa on line 4).

The pressure losses on lines 2, 3, and 4, in column L are totaled (22 Pa) and entered on line 4 in column M of Table 7-6(a).

The pressure loss of the K to M duct section is identical to the H to J duct section (including the diffusers), so lines 15 and 21 in column M of Table 7-6 are totaled (75 Pa) and entered on line 5 in columns M and N of Table 7-6(a).

Finally, the figures in column M are accumulated in column N, starting from the bottom, to obtain the new total pressure loss of 367 Pa for the fan B duct system (line 1, column O). This loss is only 5 Pa higher than the $A_1$J duct system pressure loss (Table 7-6), but it is the total pressure loss value to be used in the selection of Fan B.

### 7.10.2.8 Duct Section EN

Using the balance of the duct sizing form Table 7-6(a), the next duct run to be sized is the branch duct EQ. The pressure loss for the duct system from $A_1$ to E is obtained by totaling lines 1 to 7 of Table 7-6 and entering the 266 Pa loss on line 7 in column M.

Data for duct section EN is obtained (1000 L/s, 6.4 m/s, 456 mm diameter with 450 × 400 mm being the selected rectangular size) using the same 0.95 Pa/m friction loss rate.

EN duct loss =

$$3 \times 0.95 = 2.9 \text{ Pa}$$

(enter 3 Pa on line 8).

The data used before for computing the "main" loss coefficient for wye E (Table A-11, Figure W) is again used to obtain the "branch" loss coefficient, *see* Duct Section EF.

$$\frac{A_b}{A_s} = 0.5;\ \frac{A_b}{A_c} = 0.5;\ \frac{Q_b}{Qc} = 0.25$$

The preliminary calculations to branch EN are verified, *see* text of Duct Section EF.

$C$ (branch) = 0.44 (by interpolation)

E fitting loss =

$$C \times V_p = 0.44 \times 35.7 = 15.7 \text{ Pa}$$

(enter 16 Pa on line 9).

The loss values in column L (3 + 16 = 19) are totaled and entered on line 9 in column M.

### 7.10.2.9 Duct Section NP

Data for the 17 m duct run from N to P is computed with the friction loss rate of 0.95 Pa/m obtaining the following: 5.4 m/s velocity, 340 mm diameter and 400 × 250 mm equivalent rectangular size.

NP duct loss =

$$17 \times 0.95 = 16.2 \text{ Pa}$$

(enter 16 Pa on line 10).

At N, a 45 degree entry tap is used for branch duct NS, and a 30 degree transition is used to reduce the duct size for the run to P. From Table A-9, Figure A:

$$\frac{A_1}{A} = \frac{450 \times 400}{400 \times 250} = 1.8 \text{ (use 2)}$$

$C$ = 0.05 for $\theta$ = 30 degree

$$\text{Velocity} = \frac{1.0}{0.45 \times 0.4} = 5.6 \text{ m/s}$$

$V_p$ (Table A-4M) = 18.9 Pa

N fitting loss =

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$$C \times V_p = 0.05 \times 18.9 = 0.95 \text{ Pa}$$

(enter 1 Pal on line 11).

The volume damper at N has the same numbers as used above for the damper at J:

Damper loss =

$$C \times V_p = 0.04 \times 15.1 = 0.6 \text{ Pa}$$

(enter 1 Pa on line 12).

At 0, a smooth radius elbow with one splitter vane is selected (Table A-7, Figure G):

$$\frac{R}{W} = 0.25, \ \frac{H}{W} = \frac{250}{400} = 0.63$$

$C = 0.12$ (by interpolation)

0 fitting loss =

$$C \times V_p = 0.12 \times 15.1 = 1.8 \text{ Pa}$$

(enter 2 Pa on line 13).

The cumulative loss of 20 Pa (16+1+1+2) is entered on line 13 in column M.

### 7.10.2.10 Duct Section PQ

Data for the last 6 m of duct is obtained from Figure A-1M and Table A-2M: 250 L/s, 4.6 m/s, 270 mm diameter, and 250 × 250 mm equivalent rectangular size.

PQ duct loss =

$$6 \times 0.95 = 5.7 \text{ Pa}$$

(enter 6 Pa on line 14).

The loss coefficient for transition P is obtained from Table A-9, Figure A (converging flow) using θ = 45 degree:

$$\frac{A_1}{A} = \frac{400 \times 250}{250 \times 250} = 1.6 \text{ (use 2)}$$

$C = 0.06$

Velocity = 5.0 m/s (from the 400 × 250 Duct)

$V_p = 15.1$ Pa

P fitting loss =

$$C \times V_p = 0.06 \times 15.1 = 0.9 \text{ Pa}$$

(enter 1 Pa on line 15).

The fitting at Q is a mitered 90 degree change of size elbow (Table A-7, Figure E):

$$\frac{H}{W} = \frac{250}{250} = 1.0$$

$$\frac{W_1}{W} = \frac{400}{250} = 1.6$$

$$\text{Velocity} = \frac{0.25}{0.25 \times 0.25} = 4.0 \text{ m/s}$$

$V_p = 9.6$ Pa.

A fitting loss coefficient of 0.90 is selected. Plotting the data on the *Reynolds Number Correction-Factor Chart* indicates that a correction factor will be required.

$$D = \frac{2 \times 250 \text{ mm} \times 250 \text{ mm}}{250 \text{ mm} + 250 \text{ mm}} = 250 \text{ mm}$$

$R_e = 66.4\ DV = 66.4 \times 250 \times 4.0 = 66{,}400$

$R_e 10^{-4} = 6.64$

$K_{Re} = 1.08$

Q fitting loss =

$$0.90 \times 4 \times 1.08 = 3.9 \text{ Pa}$$

(enter 4 Pa on line 16).

The pressure loss of 32 Pa on the drawing (Figure 7-6) for the 400 × 250 mm grille is entered on line 17.

The pressure losses on lines 14-17 in column L are totaled (43 Pa), and the value entered on line 17 in column M and on line 14 in column N. Starting from line 14, the pressure losses of each section in column M are accumulated in column N, resulting in the total pressure loss of 348 Pa. This is entered on line 7 in columns N and O.

The $A_1$M duct run pressure loss of 367 Pa is 19 Pa higher than the 348 Pa pressure loss of the $A_1$Q duct run. This indicates a system that is above 12 Pa, suggested a good design difference for branch ducts. Nevertheless, adding balancing dampers in the branch ducts at N should allow the TAB technician to properly balance the system.

### 7.10.2.11 Duct Section NS

The pressure loss from $A_1$ to N (lines 7 to 9) are totaled (285 Pa) and entered on line 18 in column M. The last

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section of the supply duct system is sized using the same procedures and data from above:

NR duct loss =

$$3 \times 0.95 = 2.9 \text{ Pa}$$

(enter 3 Pa on line 19).

A 45 degree entry rectangular tap is used for the branch duct at N. From Table A-11, Figure N:

$$V_b = \frac{0.5}{0.4 \times 0.25} = 5.0 \text{ m/s}$$

$$V_c = \frac{1.0}{0.45 \times 0.4} = 5.6 \text{ m/s}$$

$$\frac{V_b}{V_c} = \frac{5.0}{5.6} = 0.89 \text{ (use 1.0)}$$

$$\frac{Q_b}{Q_c} = \frac{500}{1000} = 0.5$$

$C = 0.74$

$V_p$ (upstream) of 5.6 m/s = 18.9 Pa

N fitting loss =

$$C \times V_p = 0.74 \times 18.9 = 14 \text{ Pa}$$

(enter on line 20).

The data for the volume damper in the branch duct at N is the same as on line 12, which can be copied and entered on line 21. The total of lines 19-21 in column L of 18 Pa can be entered on line 21 in column M.

Using similar data from line 14:

RS Duct loss =

$$6 \times 0.95 = 5.9 \text{ Pa}$$

(enter 6 Pa on line 22).

Using similar data from line 15:

R Transition loss =

$$C \times Vp = 0.06 \times 15.1 = 0.91 \text{ Pa}$$

(enter 1 Pa on line 23).

S Elbow loss =

$C \times V_p \times K_{Re}$ (from line 16) = $0.90 \times 9.6 \times 1.08 = 3.9$ Pa

(enter 4 Pa on line 24).

S Grille loss (from Figure 7-6) = 32 Pa

(enter on line 25).

The losses for Run RS in column L are totaled and the 43 Pa loss is placed in column M on line 25 and in column N on line 22.

The section losses in column M are again added from the bottom in column N and the total system loss from A1 to S of 346 Pa is placed on line 18 in columns N and O. This loss again is almost equal to that of the other portions of the duct system.

#### 7.10.2.12 Balancing Damper

If the NS branch loss had been substantially lower, reasonable differences could have been compensated for by adjustments of the balancing damper. The damper loss coefficient used in each case was based on θ = 0 degree, wide open. The preliminary damper setting angle θ can be calculated in this situation as follows, assuming a total system loss difference of 20 Pa between points S and Q for this example.

System loss difference = 20 Pa

N Damper loss (set at 0°) = 1 Pa

N Damper loss (set at ?) = 21 Pa (20 + 1)

Damper loss = $C \times V_p$ or $C = \frac{\text{Damper loss}}{V_p}$

$$C = \frac{21 \text{ Pa}}{15.1} = 1.39$$

Referring back to Table A-15, Figure B, the loss coefficient of C = 1.39 would require a damper angle θ of about 21 degrees (by interpolation). The duct airflow and velocity at the damper still would remain at the design values. Points S and Q of the duct system would then have the same total pressure loss relative to point $A_1$ or fan B.

### 7.10.3 Additional Discussion

Other advantages of the above duct sizing procedures are: using columns M and N, the designer can observe the places in the duct system that have the greatest total pressure losses and where the duct construction pressure classifications change, *see* Table 4-1 and Figure 4-1. After the duct system is sized, these static pressure

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"flags" should be noted on the drawings as shown on Figure 7-6 to obtain the most economical duct fabrication and installation bids.

Building pressure allowance for supply air duct systems should be determined from building ventilation requirements taking into consideration normal building infiltration. Allowance in the range of 5 to 25 Pa for building pressurization normally is used. The designer should determine the proper building pressurization value based upon individual system requirements and location. Consideration should also include elevator shaft ventilation requirements, tightness of building construction, building-stack effect, and fire and smoke code requirements.

Finally, the system pressure loss checklist in Chapter 8 should be used to verify that all system component-pressure losses have been included in the fan total-pressure requirements. Be sure that some allowance has been added for possible changes in the field. These additional items should be shown on the duct sizing work sheets.

## 7.11 RETURN AIR (EXHAUST AIR) DUCT SYSTEM–SIZING EXAMPLE 2 (SI)

The exhaust air duct system of fan "Y" shown in Figure 7-6 will be sized using lower main duct velocities to reduce the fan power requirements. This will conserve energy and lower the operating costs. However, the duct sizes will be larger, which could increase the initial cost of the duct system.

Attention is called to the discussion in Section 5.42. All of the static pressure and total-pressure values are negative with respect to atmospheric pressure on the suction side of the fan. Applying this concept to Equation 5-5:

Fan $SP = TP_d - TP_s - V_{pd}$ (Equation 5-4)

Fan $SP = TP_d - (-TP_s) - V_{pd}$

Fan $SP = TP_d + TP_s - V_{pd}$

as $TP = SP + V_p$, then:

**Equation 7–3**

$$\text{Fan } TP = TP_d + TP_s$$

Where:

$TP_d = TP$ of fan discharge

$TP_s = TP$ of fan suction

Using the section side of Equation 7-3, all of the system pressure loss values for the exhaust system (suction side of the fan) will be entered on the work sheet as positive numbers.

### 7.11.1 Exhaust Air Plenum Z

Pressure loss data for the discharge side of the heat recovery device $A_1Z$ is entered on line 1 of Table 7-7 in column L (75 Pa). As the backwardly-curved blade fan Z free discharges into the plenum, a tentative fan selection must be made in order to obtain a velocity or velocity pressure to use to calculate the pressure loss. Most centrifugal fans are rated with duct connections on the discharge, so the loss due to "no static regain" must be added for the free discharge into the plenum. From manufacturer's data, $V_p$ = 40 Pa and C = 1.5 from Table A-13, Figure l:

Z Fan pressure loss =

$$C \times V_p = 1.5 \times 40 = 60 \text{ Pa}$$

(enter in column L on line 2).

The plenum loss total of 135 Pa is entered on line 2 in column M.

### 7.11.2 Exhaust Air System

#### 7.11.2.1 Duct Section YW

The 750 mm inlet duct to the fan is connected to an inlet box, *see* Figure 6-20. The inlet box exhaust duct connection size is 600 (0.8 × 750) × 1160 mm (1.55 × 750). The given loss coefficient C is 1.0. The return air duct connection velocity is:

$$\text{Velocity} = \frac{4}{0.6 \times 1.16} = 5.8 \text{ m/s (Equation 5-9)}$$

From Table A-4M, $V_p$ = 20.3 Pa

Inlet box loss =

$$C \times V_p = 1.0 \times 20.3 = 2.03 \text{ Pa}$$

(enter 20 Pa on line 3).

Using 4000 L/s and 8 m/s from the chart in Figure A-1M, the following is established: duct friction loss of 0.7 Pa/m, 800 mm duct diameter; and from Table A-2M, a 750 × 750 mm rectangular equivalent size duct.

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YW Duct loss =

$$10\ \text{m} \times 0.7\ \text{Pa/m} = 7\ \text{Pa};$$

(enter on line 4).

The transition at Y (Table A-8, Figure B) has a 30 degree total slope.

$\frac{A_1}{A} = \frac{600 \times 1160}{750 \times 750} = 1.24$ (use 2);

Loss coefficient $C = 0.25$ for 30 degree

Velocity = $\frac{4.0}{0.75 \times 0.75} = 7.1$ m/s

$V_p = 30.4$ Pa

Y Transition loss =

$$C \times V_p = 0.25 \times 30.4 = 7.6\ \text{Pa}$$

(enter 8 Pa on line 5).

The total for section YW (20 + 7 + 8 = 35 Pa) is entered on line 5 in column M.

#### 7.11.2.2 Duct Section WU

Using 3000 L/s and 0.7 Pa/m, 7.4 m/s is established along with a duct diameter of 730 mm. Using Table A-2M, a rectangular size of 750 × 600 mm is selected, keeping one side the same size.

WU Duct loss =

$$32\text{m} \times 0.7\ \text{Pa/m} = 22.4\ \text{Pa}$$

(enter 22 Pa on line 6).

A converging 45 degree entry tee will be used at W, *see* Table A-10, Figure F, with the velocity pressure of the downstream airflow velocity (4,000 m/s).

To obtain the "main" loss coefficient, the note in Fitting A-10F refers to Fitting A-10B:

Using Table A-10B (Main Coefficient):

$\frac{Q_b}{Q_c} = \frac{1000}{4000} = 0.25$ and $C = 0.33$

(by interpolation);

W Fitting loss =

$$C \times V_p = 0.33 \times 30.4 = 10.0\ \text{Pa}$$

(enter on line 7)

The diverging flow transition at W with an included angle of 30 degrees uses Table A-8, Figure E, because of the change of only one duct dimension.

$\frac{A_1}{A} = \frac{750 \times 750}{750 \times 600} = 1.25$ (use 2) and

$C = 0.20$

Upstream section velocity = $\frac{3.0}{0.75 \times 0.6} = 6.7$ m/s

$V_p = 27.0$ for 6.7 m/s

(from Table A-4M or by calculation);

W Transition fitting loss =

$C \times V_p = 0.20 \times 27.0 = 5.4$ Pa (enter 5 Pa on line 8).

Using a radius elbow without vanes (Table A-7, Figure F) at V, the following data is used:

$\frac{H}{W} = \frac{600}{750} = 0.8,\ \frac{R}{W} = 2.0,\ C = 0.16$

Again, using the equivalent diameter of 730 mm and the velocity of 6.7 m/s from the 3000 L/s duct for a quick check, the *Reynolds Number Correction-Factor Chart* indicates that no correction is needed.

V Fitting loss =

$$C \times V_p = 0.16 \times 27.0 = = 4.3\ \text{Pa}$$

(enter 4 Pa on line 9).

As before, the total section loss of 41 Pa is entered in column M.

#### 7.11.2.3 Duct Section UT

The total pressure loss is always the same as the static pressure when there is no velocity change. For the duct section UT, this equals:

UT Duct loss =

$$6 \times 0.7 = 4.2\ \text{Pa}$$

(enter 4 Pa on line 10).

From Figure A-1M where a 540 mm diameter duct and 6.2 m/s were obtained for 1500 L/s, a 600 × 400 mm rectangular duct is selected from Table A-2M. A converging 90 degree tee fitting (Table A-10, Figure D) will be used at U, but again the "main" loss coefficient is obtained from Figure A-10B.

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$\frac{Q_b}{Q_c} = \frac{1500}{3000} = 0.5$

$C = 0.53$

U Fitting loss =

$$0.53 \times 27.0 \text{ (downstream } V_p) = 14.3 \text{ Pa}$$

(enter 14 Pa on line 11).

The U transition loss coefficient is found in Table A-8, Figure B, and the following data computed:

$\frac{A_1}{A} = \frac{750 \times 600}{600 \times 400} = 1.88$ (use 2) and

$\theta = 30$ degree with $C = 0.25$

Velocity = $\frac{1.5}{0.6 \times 0.4} = 6.3$ m/s

$V_p = 23.9$ Pa (upstream duct)

U Fitting loss =

$$C \times V_p = 0.25 \times 23.9 = 6.0 \text{ Pa}$$

(enter on line 12).

The pressure loss for the change-of-size elbow at T will again be computed using Table A-7, Figure E (caution should be used to determine airflow direction):

$\frac{H}{W} = \frac{400}{1200} = 0.33$

$\frac{W_1}{W} = \frac{600}{1200} = 0.5$

$C = 1.8$

Velocity of the upstream section (grille size)
$= 1.5/1.2 \times 0.4 = 3.1$ m/s

$V_p = 5.8$ Pa

T Fitting loss =

$$C \times V_p = 1.8 \times 5.8 = 10.4 \text{ Pa}$$

(enter 10 Pa on line 13).

Turning vanes could be added to the change-of-size mitered elbow, but no loss coefficient tables are available. We could speculate that if single-blade turning vanes reduce the C = 1.2 of a standard 90 degree mitered elbow to about C = 0.15, the C = 1.8 used above could be reduced to approximately C = 0.23 (using the same ratio).

The pressure loss of 20 Pa for the exhaust grille at T is taken from Figure 7-6 and entered on line 14.

The section losses in column M are again added from the bottom in column N, and the Y fan duct system total of 265 Pa is entered on line 1 in columns N and O.

#### 7.11.2.4 Duct Section WX (Modified Design Method)

Branch WX must now be sized, but a visual inspection indicates that the pressure drop from W to X would be much less than that of the long run from W to T. The cumulative loss of 95 Pa for duct run W to T (line 6, column N) is also the total pressure loss requirement for the short 6 m duct run. The 12 Pa is the acceptable pressure difference between outlets or inlets on the same duct run.

In an attempt to dissipate this pressure, a velocity of 8 m/s, a duct friction loss rate of 1.7 Pa/m, and 400 mm diameter, are selected for the 1000 L/s flow rate, *see* Figure A-1M. Duct lining of 25 mm thickness (correction factor = 1.93 from Figure A-2M and Table A-1) also can be added for noise control and increased friction. A balancing damper should be used for final adjustments. The computations using this modification of the design method are:

WX Duct loss =

$$6 \text{ m} \times 1.7 \text{ Pa/m} \times 1.93 = 19.7 \text{ Pa}$$

(enter 20 Pa on line 17).

Select the rectangular size of 350 × 400 mm from Table A-2M.

The converging 45 degree entry fitting used at W (Table A-10, Figure F) is reviewed again to determine the branch loss coefficient.

$\frac{Q_b}{Q_c} = 0.25$ and Velocity ($V_c$) = 7.1 m/s

$V_p = 30.4$ Pa with $C = -0.37$

W Fitting loss =

$$C \times V_p = 0.37 \times 30.4 = -11.2 \text{ Pa}$$

Since there is a negative branch pressure loss for this fitting because of static regain (data is entered on line 18), additional losses must be provided by a balancing damper or a perforated plate in the branch duct.

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If a straight rectangular tap were used (Table A-10, Figure D) instead of the 45 degree entry tap, the loss coefficient would then become 0.01, a more appropriate selection.

Inefficient transitions at grille X also will help build up the loss (note airflow direction). Figure A is a rectangular converging transition in Table A-9.

$$\frac{A_1}{A} = \frac{900 \text{ mm} \times 400 \text{ mm}}{350 \text{ mm} \times 400 \text{ mm}} = 2.57$$

$\theta$ = 180 degree (abrupt)

$C$ = 0.30. The downstream velocity must be used to determine the $V_p$ used in the computations:

$$\text{Velocity (downstream)} = \frac{1.0}{0.35 \times 0.4} = 7.1 \text{ m/s}$$

$V_p$ = 30.4 Pa

X Fitting loss =

$$C \times V_p = 0.30 \times 30.4 = 9.1 \text{ Pa}$$

(enter 9 Pa on line 20).

X Grille loss (from Figure 7-6) = 20 Pa

(enter on line 20).

Subtracting 38 Pa (the total of lines 17 to 20) from the 95 Pa duct run WT pressure loss shown on line 6 in column N, leaves 57 Pa of pressure for the balancing damper to dissipate.

Damper loss coefficient $C$ =

$$\frac{TP}{V_p} = \frac{57 \text{ Pa}}{30.4 \text{ Pa}} = 1.88$$

From Table A-15, Figure B, a damper set to 23 degree (by interpolation) has a loss coefficient of 1.88 will balance the branch duct WX. The total of 95 Pa (adding lines 17-21) is shown on line 21 in column M, and on line 17 in column N.

A perforated plate (Table A-14, Figure B) is a non-adjustable alternate solution. If a 3 mm thick perforated plate were used instead of the balancing damper, the calculation procedure would be as follows, *see* Table A-14, Figure B:

Assuming 16 mm diameter holes,

$$\frac{T}{d} = \frac{3}{16} = 0.19$$

With $C$ = 1.88 (from above)

$N$ = 0.60

$$n = \frac{A_p}{A}$$

$$A_p = n \times A$$

$A_p$ (flow area of perforated plate) = $0.60 \times 350 \times 400$ = 84,000 mm$^2$

$$\text{No. of holes} = \frac{A_p}{\text{area of a 16 mm diameter hole}}$$

$$\text{No. of holes} = \frac{84,000}{8^2\pi} = 418$$

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DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE _____ PAGE 1 OF 2

PROJECT SAMPLE BUILDING LOCATION FIRST FLOOR SYSTEM EXHAUST AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW (L/S) | FRICTION PER METER | VELOC-ITY (M/S) | $V_p$ (PA) | LOSS COEFF | EQUIV DIAM | RECTANGULAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMU-LATIVE LOSS | TOTAL LOSS |
| 1 | PLENUM Z | $A_1$ Z | – | E.R. DEVICE | 4000 | – | 2.5 | – | – | – | | – | 75 | | 265 | 265 |
| 2 | ” | Z | – | FAN | 4000 | – | 8.0 | 38.5 | – | – | | – | 60 | 135 | | |
| 3 | RUN YT | Y | – | INLET BOX | 4000 | – | 5.8 | 20.3 | 1.0 | – | 600 × 1160 | – | 20 | | 130 | |
| 4 | ” | YW | 10m | DUCT | 4000 | 0.7 | 8.0 | – | – | 800 | 750 × 750 | – | 7 | | | |
| 5 | ” | Y | – | TRANS | 4000 | – | 7.1 | 30.4 | 0.25 | – | 750 × 750 / 600 × 1160 | – | 8 | 35 | | |
| 6 | ” | WU | 52m | DUCT | 3000 | 0.7 | 7.4 | – | – | 730 | 750 × 600 | – | 22 | | 95 | |
| 7 | ” | W | 45° | ENT. TAP | 4000 / 1000 | – | 7.1 | 30.4 | 0.33 | – | 750 × 750 | – | 10 | | | |
| 8 | ” | W | – | TRANS | 4000 / 3000 | – | 6.7 | 27.0 | 0.20 | – | 750 × 750 / 750 × 600 | – | 5 | | | |
| 9 | ” | V | 90° | ELBOW | 3000 | – | 6.7 | 27.0 | 0.16 | – | 750 × 600 | – | 4 | 41 | | |
| 10 | ” | UT | – | DUCT | 1500 | 0.7 | 6.2 | – | – | 540 | 600 × 400 | – | 4 | | 54 | |
| 11 | ” | U | 90° | ENT. TAP | 3000 / 1500 | – | 6.7 | 27.0 | 0.53 | – | 750 × 600 | – | 14 | | | |
| 12 | ” | U | 6m | TRANS | 3000 / 1500 | – | 6.3 | 23.9 | 0.25 | – | 750 × 600 / 600 × 400 | – | 6 | | | |
| 13 | ” | T | 90° | ELBOW | 1500 | – | 3.1 | 5.8 | 1.8 | – | 600 × 400 / 1200 × 400 | – | 10 | | | |
| 14 | ” | T | – | GRILLE | 1500 | – | – | – | – | – | 1200 × 400 | – | 20 | 54 | | |
| 15 | | | | | | | | | | | | | | | | |
| 16 | | | | | | | | | | | | | | | | |
| 17 | RUN WX | WX | 6m | DUCT | 1000 | 1.7 | 8.0 | – | – | 400 | *350 × 400 | 1.93 | 20 | | 95 | |
| 18 | ” | W | 45° | ENT. TAP | 4000 / 1000 | – | 7.1 | 30.4 | –0.37 | – | 750 × 750 / 350 × 400 | – | –11 | | | |
| 19 | ” | X | 18° | TRANS | 1000 | – | 7.1 | 30.4 | 0.30 | – | 350 × 400 / 900 × 400 | – | 9 | | | |
| 20 | ” | X | – | GRILLE | 1000 | – | – | – | – | – | 900 × 400 | – | 20 | | | |
| 21 | ” | W | SET 23° | VOL. DAMP | 1000 | – | 7.1 | 30.4 | 1.88 | – | 350 × 400 | – | 57 | 95 | | |
| 22 | | | | | | | | | | | | | | | | |
| 23 | | | | | | | | | | | | | | | | |
| 24 | | | | | | | | | | | | | | | | |
| 25 | | | | | | | | | | | | | | | | |

**Table 7–7 Duct Sizing, Exhaust Air System – Example 2 (SI)**

Note: * Indicates duct lining used. Sizes are interior dimensions.

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DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE _____ PAGE 1 OF 2

PROJECT SAMPLE BUILDING LOCATION CONF. AREA SYSTEM SUPPLY AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW (L/S) | FRICTION PER METER | VELOC-ITY (M/S) | $V_p$ (PA) | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMULA-TIVE LOSS | TOTAL LOSS |
| 1 | PLENUM A | A | – | SYSTEM EFFECT | 10,000 | – | 11.0 | – | – | 1065 | 1100 × 810 | RS | 72 | | 683 | 683 |
| 2 | ” | B | 90° | ELBOW | 10,000 | – | 11.2 | 76 | 0.15 | 1065 | 1100 × 810 | – | 11 | 83 | | |
| 3 | RUN CS | CF | 27m | DUCT | 10,000 | 2.4 | 16.0 | – | – | 900 | – | – | 65 | | 600 | |
| 4 | ” | C | 20° | TRANS | 10,000 | – | 16.0 | 154 | 0.05 | – | 1100 × 810 / 900 | – | 8 | | | |
| 5 | ” | D | – | SOUND ATTN. | 10,000 | – | – | – | – | 900 | – | – | 65 | | | |
| 6 | ” | E | 90° | ELBOW | 10,000 | – | 16.0 | 154 | 0.15 | 900 | – | – | 23 | 161 | | |
| 7 | ” | FH | 16m | DUCT | 5000 | 2.0 | 13.0 | 102 | – | 700 | – | – | 32 | | 439 | |
| 8 | ” | F | 45° | WYE | 10,000 / 5000 | – | 16.0 | 154 | 0.27 | 900 / 700 | – | – | 45 | | | |
| 9 | ” | F | 45° | ELBOW | 5000 | – | 13.0 | 102 | 0.15 | 700 | – | 0.6 | 9 | | | |
| 10 | ” | G | 90° | ELBOW | 5000 | – | 13.0 | 102 | 0.15 | 700 | – | – | 15 | 101 | | |
| 11 | ” | HO | 12m | DUCT | 2500 | 1.9 | 10.6 | 68 | – | 550 | – | – | 23 | | 338 | |
| 12 | ” | H | 45° | WYE | 5000 / 2500 | – | 13.0 | 102 | 0.51 | 700 / 550 | – | – | 52 | | | |
| 13 | ” | H | 0° | VOL DAMP | 2500 | – | 10.6 | 68 | 0.20 | 550 | – | – | 14 | | | |
| 14 | ” | N | 90° | ELBOW | 2500 | – | 10.6 | 68 | 0.15 | 550 | – | – | 10 | 99 | | |
| 15 | ” | OP | 10m | DUCT | 2000 | 2.0 | 10.2 | 63 | – | 500 | – | – | 20 | | 239 | |
| 16 | ” | O | 45° | WYE | 2500 / 2000 | – | 10.6 | 68 | 0.01 | 550 | – | – | 1 | | | |
| 17 | ” | O | 60° | TRANS | 2000 | – | 10.2 | 63 | 0.06 | 550 / 500 | – | – | 4 | 25 | | |
| 18 | ” | PQ | 10m | DUCT | 1500 | 1.9 | 9.0 | 49 | – | 450 | – | – | 19 | | 214 | |
| 19 | ” | P | 45° | WYE | 2000 / 1500 | – | 10.2 | 63 | 0.01 | 500 | – | – | 1 | | | |
| 20 | ” | P | 60 | TRANS | 1500 | – | 9.0 | 49 | 0.06 | 500 / 450 | – | – | 3 | 23 | | |
| 21 | ” | QR | 10m | DUCT | 1000 | 1.8 | 8.0 | 39 | – | 400 | – | – | 18 | | 191 | |
| 22 | ” | Q | 45 | WYE | 1000 / 1000 | – | 9.0 | 49 | 0.01 | 450 | – | – | 1 | | | |
| 23 | ” | Q | 60 | TRANS | 1000 | – | 8.0 | 39 | 0.06 | 450 / 400 | – | – | 2 | 21 | | |
| 24 | ” | – | – | SUB TOTAL FROM TABLE 8–3 (a) | | | | | | | | – | – | | 170 | |
| 25 | | | | | | | | | | | | | | | | |

## Table 7–8 Duct Sizing, Supply Air System – Example 3

Note: *Indicates duct lining used. Sizes are interior dimensions.

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DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE _____ PAGE 2 OF 2

PROJECT SAMPLE BUILDING LOCATION CONF. AREA SYSTEM SUPPLY AIR

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW (L/S) | FRICTION PER METER | VELOC-ITY (M/S) | $V_p$ (PA) | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMULA-TIVE LOSS | TOTAL LOSS |
| 1 | RUN CS | RS | 10m | DUCT | 500 | 1.7 | 6.8 | 28 | – | 300 | – | – | 17 | | 170 | |
| 2 | (CONT) ” | R | 45 | WYE | 1000 / 500 | – | 8.0 | 39 | 0.01 | 400 | – | – | 1 | | | |
| 3 | ” | R | 60 | TRANS | 500 | – | 6.8 | 28 | 0.06 | 400 / 300 | – | – | 2 | | | |
| 4 | ” | S | 45 | ELBOW | 500 | – | 6.8 | 28 | 0.15 | 300 | – | 0.6 | 3 | | | |
| 5 | ” | S | 2m | FLEX DUCT | 500 | 1.7 | 6.8 | 28 | – | 300 | – | 1.95 | 7 | 170 | | |
| 6 | ” | S | – | VAV BOX | 500 | – | – | – | – | – | – | – | | | | |
| 7 | | | | | | | | | | | | | | | | |
| 8 | | | | | | | | | | | | | | | | |
| 9 | | | | | | | | | | | | | | | | |
| 10 | | | | | | | | | | | | | | | | |
| 11 | | | | | | | | | | | | | | | | |
| 12 | | | | | | | | | | | | | | | | |
| 13 | | | | | | | | | | | | | | | | |
| 14 | | | | | | | | | | | | | | | | |
| 15 | | | | | | | | | | | | | | | | |
| 16 | | | | | | | | | | | | | | | | |
| 17 | | | | | | | | | | | | | | | | |
| 18 | | | | | | | | | | | | | | | | |
| 19 | | | | | | | | | | | | | | | | |
| 20 | | | | | | | | | | | | | | | | |
| 21 | | | | | | | | | | | | | | | | |
| 22 | | | | | | | | | | | | | | | | |
| 23 | | | | | | | | | | | | | | | | |
| 24 | | | | | | | | | | | | | | | | |
| 25 | | | | | | | | | | | | | | | | |

**Table 7–8(a) Duct Sizing, Supply Air System – Example 3 (Continued)**

Note: *Indicates duct lining used. Sizes are interior dimensions.

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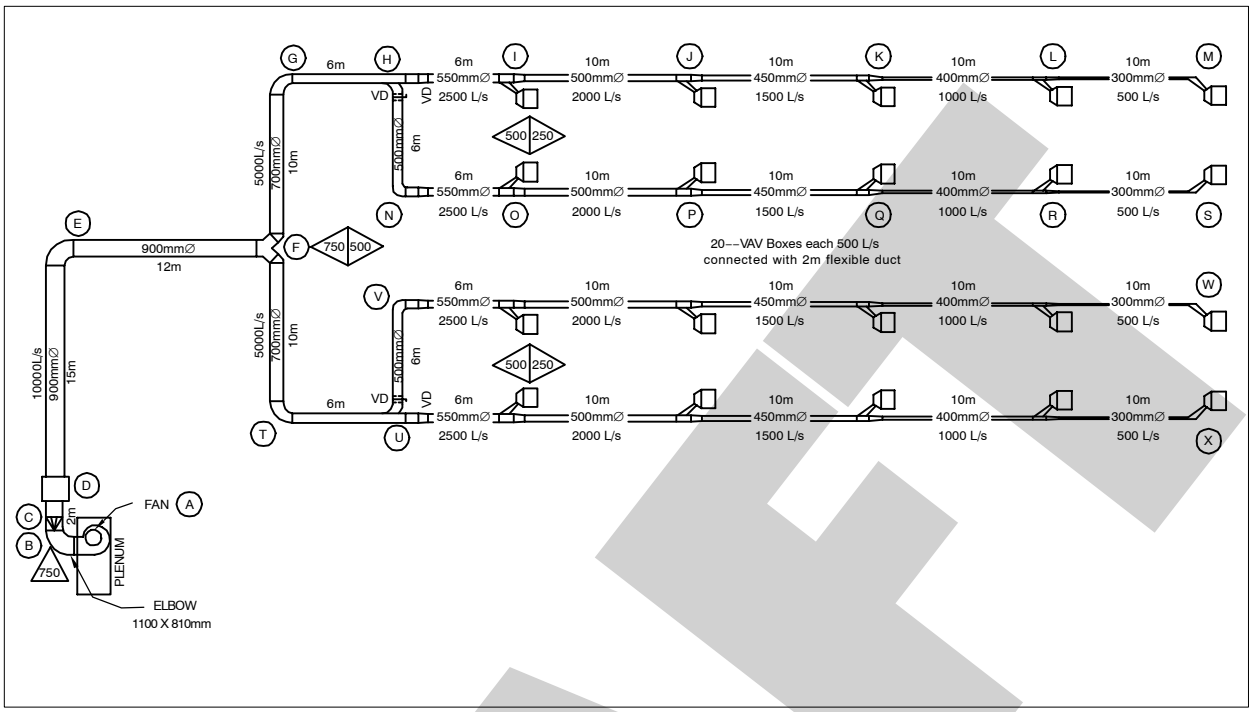


**FIGURE 7–7M SUPPLY AIR DUCT SYSTEM FOR SIZING EXAMPLE 3**

## 7.12 SUPPLY AIR DUCT SYSTEM SIZING EXAMPLE 3 (SI)

Higher-pressure supply air systems (over 750 Pa) usually are required for the large central-station HVAC supply air duct distribution systems. Because of higher fan power requirements, ASHRAE provisions will require the designer to analyze lower pressure duct systems, against the constantly increasing costs of building operation. The choice of duct system pressure is now more than ever dependent on energy costs, the application, and the ingenuity of the designer.

The "Static Regain Method" and the "Total Pressure Method" have traditionally been used to design the higher pressure supply air systems.

### 7.12.1 Design Procedures

After analyzing a duct system layout, the chart in Figure A-1M is used to select an "approximate" initial velocity and a pressure loss (Pa/m) that will be used for most duct sections throughout the system. This selected velocity should be within the shaded sections of the chart. Using the design airflow quantities (L/s) of the duct sections and the selected velocity (m/s), the duct diameters and friction loss rates may also be obtained from Figure A-1M. When rectangular-duct sizes are to be used, selection may be made from Table A-2M, based on circular equivalents.

The use of higher velocities normally increases duct system noise levels. The designer must consider that acoustical treatment might be required for the duct system and an allowance must be made for increased duct dimensions if lined or for additional space requirements if sound attenuators are used.

If acoustic treatment or insulation liner is required, the designer must clearly indicate if the duct dimensions shown on the duct layout drawings have been adjusted for this added thickness.

The designer must inspect the duct layout and make an assumption as to which duct run has the highest pressure loss. This is the path for the first series of calculations. The average velocity of the initial duct section (based on the cross-sectional area) is used to obtain the velocity pressure ($V_p$) from Table A-4M, or it may be calculated using Equation 5-8 in Chapter 5. The velocity pressure is used with fitting loss coefficients from the tables in the Appendix to determine the dynamic-pressure loss of each fitting. The pressure losses of system components usually are obtained from equipment data sheets, but approximate data can be selected from the tables and charts in Chapter 8. The total pressure loss is then computed for the initial duct section by totaling the individual losses of the straight duct sections and duct fittings.

Each succeeding duct section is computed in the same manner, with careful consideration being given to the

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type of fitting selected and comparing loss coefficients to obtain the most efficient fitting.

After the calculations are made and each duct section properly sized, the pressure loss must be added for the terminal outlet device at the end of the last duct section. Adding from the bottom of the form to the top, the section losses are totaled in column N to obtain the supply fan pressure requirements for the supply air duct system. This assumes that the originally selected duct run was correct and that it has the highest pressure loss.

Using the cumulative pressure subtotal of the main duct at the point of each branch, calculate the cumulative pressure total for each branch run as outlined above. If a duct run other than the assumed duct run has a higher cumulative pressure loss total, then the higher amount now becomes the pressure that the fan must provide to the supply air duct system. The return air duct system, which is calculated separately, also is part of the fan load. Velocities and friction loss rates for the shorter runs may fall into a "higher velocity range" as long as the noise potential is considered.

Caution must be used in the above sizing procedure for the "longest duct run," as the use of smaller duct sizes created by higher velocities and higher pressures can increase the fan power and cost of operation. This is becoming more critical with rising energy costs.

A life cycle cost analysis will probably dictate that lower op erating costs should be considered more important than lower first costs and space saving requirements.

### 7.12.2 Supply Air System

Table 7-8 is the tabulation of design and computation data obtained when sizing the 10,000 L/s supply duct system shown in Figure 7-7. The 95 m duct run from C to S appears to be the path with the greatest resistance, although the duct run from C to W appears to have about the same resistance. All of the VAV terminal units have the same capacity (500 L/s each.) The airflow of the duct sections varies from 10,000 to 500 L/s. Selecting an initial velocity of approximately 16 m/s and a friction rate of 2.4 Pa/m would indicate (by following the 2.4 Pa/m line horizontally to 500 L/s) that the duct velocities would gradually be reduced to less than 8 m/s at airflow of 500 L/s.

#### 7.12.2.1 Plenum

Before the duct system is sized, the losses within the plenum must be calculated. Data from the manufacturer's catalog for the DWDI fan A, which must be tentatively selected, indicates a discharge outlet size of $1100 \times 810$ mm, a discharge velocity of 11 m/s (velocity pressure = 75 Pa), and a blast area/outlet area ratio of 0.6.

Elbow B is sized $1100 \times 810$ mm (so that it is similar to the outlet size) and a radius elbow $\left(\frac{R}{W} = 1.5\right)$ is selected. It is located 660 mm above the fan discharge opening.

Using the directions in Figure 6-2, Figure 6-3, and Table 6-2 for a DWDI fan, the pressure loss is calculated for the "System Effect" created by the discharge elbow at B:

$$\text{Equiv. Diam.} = \sqrt{\frac{4 \times 1100 \times 810}{\pi}} = 1065 \text{ mm}$$

Percent Effective Duct =

$$\frac{\text{Straight duct length} \times 100}{\text{Vel./s(2.5 min)} \times \text{Equiv. Diam}}$$

Percent Effective Duct =

$$\frac{600 \times 100}{2.5 \times 1065} = 24.8 \text{ percent}$$

From Table 6-2, System Effect Curve R-S for a 0.6 blast area ratio and 25 percent Effective Duct is used with Figure 6-1 to find the System Effect pressure loss of 72 Pa (based on 11 m/s). Since the elbow is in position "A" (Figure 6-3), the multiplier for the DWDI fan from Table 6-2 of 1.00 does not change the value, which is entered on line 1 in column L of the duct sizing work sheet in Table 7-8. Note that the 72 Pa of system effect could be subtracted from the total pressure output of the fan, instead of being added to the total system loss.

The loss coefficient of 0.15 for elbow B is obtained (using Table A-7, Figure F) with $\frac{R}{W} = 1.5$ and $\frac{H}{W} = \frac{1100}{810} = 1.36$.

Average Velocity =

$$\frac{Q}{A}(\text{Equation } 5-9) = \frac{10}{1.1 \times 0.81} = 11.2 \text{ m/s}$$

The velocity pressure ($V_p$) of 76 Pa is obtained from Table A-4M for a velocity of 11.2 m/s. A quick check of the *Reynolds Number Correction-Factor Chart* indicates that no correction is needed.

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B Fitting loss =

$$C \times V_p = 0.15 \times 76 = 11.4 \text{ Pa (use 11 Pa)}$$

The total pressure loss of 83 Pa for the plenum is entered on line 2 in column M.

#### 7.12.2.2 Duct Section CF

Round spiral duct with an absolute roughness of 0.0003 ft will be used in this supply duct system. For the 27 m of duct in section CF using an assumed velocity of 16.0 m/s, it falls right on the closest standard size duct diameter of 900 mm (from the chart of Figure A-1M). The selected velocity of 16.0 m/s has a friction loss rate of 2.4 Pa/m. A duct-friction correction factor is not required, as the chart in Figure A-1M is based on the same absolute roughness.

CF Duct loss =

$$27 \times 2.4 = 64.8 \text{ Pa}$$

(enter 65 Pa on line 3).

The transition at C will be converging, rectangular to round (Table A-9, Figure A) with $A_1/A = 1100 \times 810/(450)^2 \pi = 1.40$ and $\theta = 20$ degree; C = 0.05. The velocity pressure used is that of the downstream section: 154 Pa for 16 m/s, *see* Table A-4M.

C Transition loss =

$$C \times V_p = 0.05 \times 154 \text{ (leaving } V_p) = 7.7 \text{ Pa}$$

(enter 8 Pa on line 4).

The pressure loss for a medium attenuation 900 mm diameter sound trap of 65 Pa is obtained from Chapter 8. A preliminary loss also can be obtained from manufacturer's data sheets. The data is entered on line 5.

The smooth radius, 90 degree round elbow at E has an $\frac{R}{D}$ ration = 1.5; *C* = 0.15 (Table A-7, Figure A).

E Elbow loss =

$$C \times V_p = 0.15 \times 154 = 23.1 \text{ Pa (line 6)}$$

The pressure losses of the four items in duct section CF are added and the 161 Pa total is entered in column M on line 6.

#### 7.12.2.3 Duct Section FH

Using the same procedure as above, the closest standard size for 5000 L/s at 2.4 Pa/m friction loss is 680 mm (use 700) diameter, *see* Figure A-1M. A velocity of 13.0 m/s, 2.0 Pa/m and the related $V_p$ of 102 Pa is used for further calculations based on the 700 mm diameter standard size duct.

FH Duct loss =

$$16 \times 2.0 = 32 \text{ Pa}$$

(enter 32 Pa on line 7).

Using a 45 degree round wye fitting (Table A-1M, Figure Y) with 45 degree elbows at F, $V_{ib}/B_c = 13.0/16 = 0.81$; C = 0.29 (by interpolation).

F Wye fitting loss =

$$C \times V_p = 0.29 \times 154 = 44.7 \text{ Pa}$$

(enter 45 Pa on line 8).

The 45 degree round elbow (RD = 1.5) at F will use the same loss coefficient as the 90 degree elbow above (Table A-7, Figure A) multiplied by the 0.6 correction factor for 45 degree (Note 1). $V_p$ for 13.0 m/s = 102 Pa.

F Elbow fitting loss =

$$0.15 \times 102 \times 0.6 = 9.2 \text{ Pa (line 9)}$$

The 90 degree round elbow at G uses the same values without the correction factor.

G Elbow fitting loss =

$$0.15 \times 102 = 15.3 \text{ Pa (line 10)}$$

The losses in column L again are totaled and 101 Pa is entered in column M.

#### 7.12.2.4 Duct Section HO

The following values are obtained using the same procedures as above:

2500 L/s at 2.4 Pa/m friction loss gives a 525 mm diameter duct size. Using a standard size of 550 mm, velocity = 10.6 m/s; $V_p$ = 68 Pa, and the friction loss rate = 1.9 m/s.

HO Duct loss =

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$$12 \times 1.9 = 22.8 \text{ Pa}$$

(enter 23 Pa on line 11).

At point H in the duct system, the branch coefficient is obtained for the diverging 45 degree round wye with a conical main and branch with a 45 degree elbow (Table A-11, Figure M):

$\frac{V_b}{V_c} = \frac{10.6}{13.0} = 0.82$; $C = 0.51$

$V_p = 102$ Pa (13.0 m/s)

H Wye (branch) loss =

$$C \times V_p = 0.51 \times 102 = 52.0 \text{ Pa (line 12)}$$

The 90 degree round elbow is calculated as the above 90 degree ell and the loss coefficient for the balancing damper is obtained from Table A-15, Figure A ($\theta = 0$ degree); C = 0.20; $V_p$ for 10.6 m/s =68 Pa.

H Damper loss =

$$C \times V_p = 0.20 \times 68 = 13.6 \text{ Pa (line 13)}$$

N Elbow fitting loss =

$$C \times V_p = 0.15 \times 68 = 10.2 \text{ Pa (line 14)}$$

The total for the HO duct section (99 Pa) is entered in column M.

### 7.12.2.5 Duct Section OP

For 2000 L/s at 2.4 Pa/m, the closest standard size duct is 500 mm diameter. Using the 500 mm duct, the friction rate then becomes 2.0 Pa/m, and the duct velocity is 10.2 m/s.

OP Duct loss =

$10 \times 2.0 = 20$ Pa (line 15)

$V_p$ for 10.2 m/s = 63 Pa (Table A-4M)

The 45 degree round diverging conical wye at point 0 (Table A-11, Figure C) require that the "main" coefficient C be obtained from Table A-11A.

$\frac{V_s}{V_c} = \frac{10.2}{10.6} = 0.96$; but when there is little or no change in velocity, the table indicates there is no dynamic loss. $C = 0$ for $\frac{V_s}{V_c} = 1.0$. Interpolating gives a questionable loss coefficient of 0.004, which multiplied by the $V_p$ of 68 Pa gives a loss of 0.3 Pa. However, a minimum loss coefficient of 0.01 is used to be on the safe side.

0 Wye (main) loss =

$$0.01 \times 68 = 0.7 \text{ Pa}$$

(enter 1 Pa on line 16).

The 60 degree transition from 550 mm diameter to 500 mm diameter has a dynamic pressure loss, and the fitting loss coefficient is obtained from Table A-9, Figure A.

$\frac{A_1}{A} = \frac{(275)^2\pi}{(250)^2\pi} = 1.21$ (use 2)

$C = 0.06$;

0 Transition loss =

$C \times V_p$ (downstream) $= 0.06 \times 63 = 3.8$ Pa (line 17)

The section loss of 25 Pa is entered in column M.

### 7.12.2.6 Duct Section PQ

The same calculations as used in duct section OP are repeated using 1500 L/s and a 2.4 Pa/m friction loss rate to obtain the closest standard duct size of 450 mm diameter, *see* Figure A-1M. Using the 450 mm duct size, the new velocity is 9.0 m/s and the friction loss rate is 1.9 Pa/m:

PQ Duct loss = $10 \times 1.9 = 19$ Pa (line 18)

$V_p$ for 9.0 m/s = 49 Pa (Table A-4M)

For the 45 degree round conical wye at P (Table 14A):

$\frac{V_s}{V_c} = \frac{9.0}{10.2} = 0.88$

$C = 0.01$

P Wye (main) loss =

$C \times V_p = 0.01 \times 63$ (upstream $V_p$) $= 0.6$ Pa

(enter 1 Pa on line 19).

60 degree transition at P: $\frac{A_1}{A} = \frac{(250)^2\pi}{(225)^2\pi} = 1.23$

(Table A-9A), C = 0.06;

P Transition loss =

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$$C \times V = 0.60 \times 49 = 2.9 \text{ Pa}$$

(enter 3 Pa on line 20).

This section loss of 23 Pa is entered in column M.

#### 7.12.2.7 Duct Section QR

The selection of a standard 400 mm diameter duct for 1000 L/s (Figure A-1M indicates a 1.8 Pa/m friction loss rate and a velocity of 8.0 m/s).

QR Duct loss =

$$10 \times 1.8 = 18 \text{ Pa (line 21)}$$

$V_p$ for 8.0 m/s = 39 Pa (Table A-4M)

Again using the same type of wye at Q:

$$\frac{V_s}{V_c} = \frac{8.0}{9.0} = 0.89$$

$C = 0.01$

Q Wye (main) loss =

$$C \times V_p = 0.01 \times 49 \text{ (upstream } V_p) = 0.5 \text{ Pa}$$

(enter 1 Pa on line 22).

Q Transition: $\frac{A_1}{A} = \frac{(225)^2\pi}{(200)^2\pi} = 1.27$ (Table 14-12A)

$C = 0.06$

Q Trans. Fitting loss =

$$C \times V_p = 0.06 \times 39 = 2.3 \text{ (line 23)}$$

The section loss of 21 Pa is entered in column M.

#### 7.12.2.8 Duct Section RS

Using an additional duct sizing form to record the data Table 7-8(a), the 500 mm duct size at 2.4 Pa/m would be between the 250 mm and 300 mm standard duct sizes. The 250 mm diameter duct would have a much higher-pressure loss, so the 300 mm duct at 1.7 Pa/m friction loss and 6.8 m/s velocity would be the better selection.

RS Duct loss =

$$10 \times 1.7 = 17 \text{ Pa (line 1)}$$

$V_p$ for 6.8 m/s = 28 Pa (Table 14-7)

R wye fitting: $\frac{V_x}{V_c} = \frac{6.8}{8.0} = 0.85$

$C = 0.01$ (Figure A-11A)

R Wye (main) loss =

$$C \times V_p = 0.01 \times 39 = 0.04 \text{ Pa}$$

(enter 1 Pal on line 2).

R transition: $\frac{A_1}{A} = \frac{(200)^2\pi}{(150)^2\pi} = 1.78$

$C = 0.06$ (Table A-9A)

R Transition loss =

$$C \times V_p = 0.06 \times 28 = 1.7 \text{ Pa (line 3)}$$

A 45 degree elbow at the duct end is connected to the VAV box by a 2 m piece of 300 mm diameter flexible duct. The correction factor for the flexible duct is obtained from the chart in Figure A-2, using Table A-2 as a guide. Bends of 30 degree or more would also add additional resistance. Verified data is not available, so a radius elbow loss coefficient could be used to obtain the additional loss. Figure A-3 also contains a correction factor for unextended (compressed) flexible duct.

S 45 degree Elbow fitting loss =

$$0.15 \times 28 \times 0.60 = 2.5 \text{ Pa (line 4)}$$

S Flex. Duct loss =

$$2 \text{ m} \times 1.7 \text{ Pa/m} \times 1.95 = 6.6 \text{ Pa (line 5)}$$

An estimate of 65 Pa is made for the downstream side ductwork and diffuser from the VAV box. This is added to the VAV box pressure loss of 75 Pa, and the total (140 Pa) is entered on line 6. The total pressure loss of 170 Pa for the RS section is entered in columns M and N, and also in column N on line 24 of page 1 (Table 7-8) of the duct sizing work sheet. Working from the bottom to the top of the form, the section pressure losses are totaled in column N, with the *total pressure* loss for the supply duct system of 683 Pa being entered on line 1 in columns N and O.

#### 7.12.2.9 Review

The same procedure is used to size the other segments of the supply duct system; or, if the layout is symmetrical, the same sizes can be used for similar segments of the system. However, as was found in the supply air duct system sizing Example 1, several fittings with higher-pressure losses or "high loss" VAV boxes can

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allow a duct run that was not the originally selected run for design computations to be the duct run with the greatest pressure loss.

Assuming that the return air duct system of Example 3 had an approximate total pressure loss of 500 Pa, the output of the system supply fan would need to be 10,000 L/s at 1182 Pa (500 + 683). Attention is called to the fact that although the fan total pressure requirements are in the upper portion of the duct pressure classification range, all of the supply air duct system beyond the wye fitting at F, is in the low pressure range (under 500 Pa), even though there are velocities up to 13 m/s.

This is the reason it is extremely important to indicate static-pressure "flags" on the drawings after the duct system is sized as indicated in Figure 7-7. Table 2-5 indicates the relative costs of fabrication and installation of different pressure classes of ductwork for the same size duct. So the initial installation cost savings become quite apparent by this simple procedure, especially when the system designer specifies a higher-pressure duct construction classification for the duct systems when a lower classification would be more than adequate.

## 7.13 EXTENDED PLENUM DUCT SIZING

In the design of air distribution duct layouts, a design variation commonly referred to as "extended plenum" or "semi-extended plenum," often is incorporated into the particular duct sizing method being employed.

An *extended plenum* is a trunk duct of constant size, usually at the discharge of a fan, fan-coil unit, mixing box or variable air volume (VAV) box extended as a plenum to serve multiple outlets and branch ducts.

A *semi-extended plenum* is a trunk design system utilizing the concept of extended plenum that incorporates a minimum number of size reductions to compensate for the decreasing air volume.

### 7.13.1 Properties

Some of the advantages realized through the use of the semi-extended plenum system concept are:

a. Lower first cost due to an improved ratio of straight duct to fittings.

b. Lower operating cost due to savings in fan horsepower through elimination of high-energy loss transition fittings.

c. Ease of balancing due to low branch take-off pressure losses and fewer trunk duct pressure changes.

As long as design air volume is not exceeded, branch ducts can be added, removed, and relocated at any convenient point along the trunk duct (between size reductions) without affecting performance. This is particularly useful in "tenant development" work.

A limiting factor to be considered when using the extended-plenum method is the low velocities which could develop. This can result in excessive heat gain or loss through the duct walls compared to the volume of air.

### 7.13.2 Design Criteria

Actual installations and tests indicate that semi-extended plenum design is acceptable for use with system static pressures that range from 250 to 1500 Pa, and duct velocities up through 15 m/s. Other specific design considerations include:

a. Branch takeoffs from the trunk duct should preferably be round duct connecting at a 45 degree angle. If rectangular branches are used, a 45 degree entry tap should be used.

b. Velocities in branch takeoffs should range between 55 and 90 percent of the trunk duct velocity to minimize static pressure loss across the takeoff.

c. Branch velocities should not exceed the trunk duct velocity.

d. Balancing dampers should be installed in each branch duct.

### 7.13.3 Comparison of Design Methods

Figures 7-8 and 7-9 illustrate identical medium pressure systems differing only in the trunk duct sizing techniques used. The trunk duct system shown in Figure 7-8 has been sized by the equal-friction method, at a pressure loss of approximately 4 Pa/m. Note that reducing fittings have been used at each branch takeoff.

In Figure 7-9, the semi-extended plenum "concept" has been used to keep duct reductions at a minimum. Note that System "A" utilizes six trunk duct sizes and five reducing fittings, while System "B" has only three duct sizes and two reducing fittings. Assuming that the duct between the primary air-handling unit and secondary terminal unit "F" has the highest supply-pressure loss, using friction loss data from the Appendix, the re-

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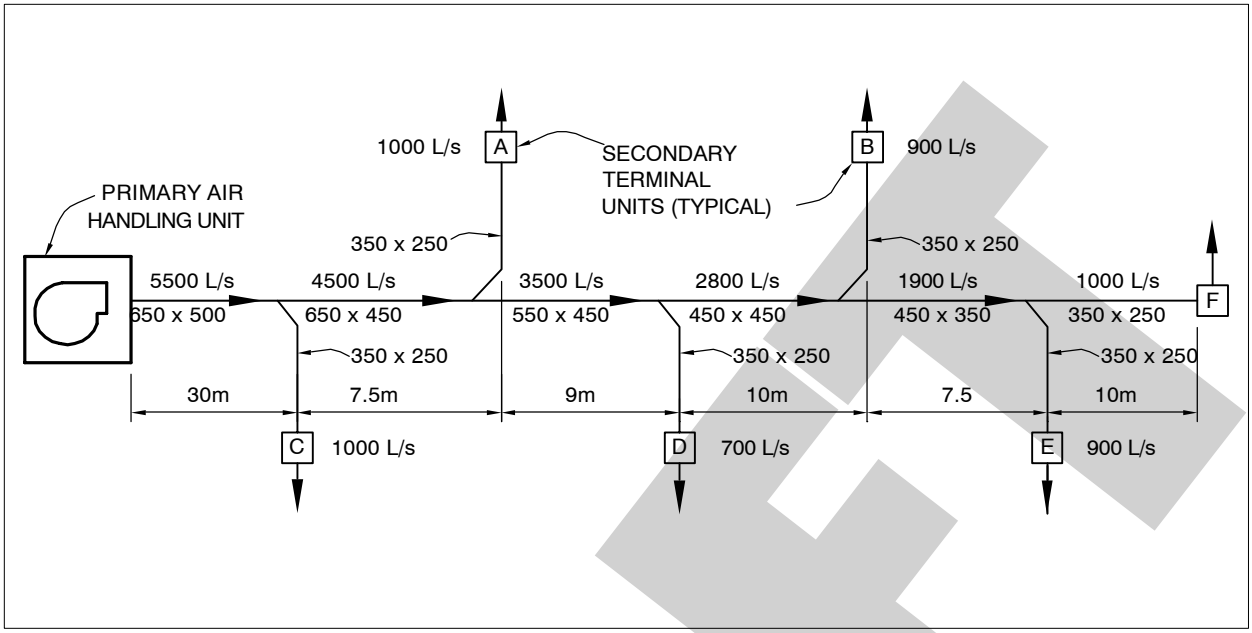


**FIGURE 7–8M SYSTEMS "A" SIZED BY EQUAL FRICTION METHOD**

sults are tabulated in Table 7-9. Ignoring branch duct and outlet losses, which are identical for both systems, the semi-extended plenum system has a 157 Pa (481-324) lower pressure loss than the system sized by the equal-friction method.

The fan power necessary to satisfy the supply pressure requirements selected from a typical manufacturer's catalog, are also shown in Table 7-9. Notice that the semi-extended plenum design results in reduced fan power and lower operating costs. The cost savings, both first and operating, could be even greater with a return air duct system utilizing the semi-extended plenum concept.

### 7.13.4 Cost Comparison

Although energy conservation is extremely important, installation costs are still of primary concern to the designer, the contractor and the owner. Table 7-10 illustrates the estimated installation cost comparison between the two systems analyzed. It shows how the overall installed cost for the semi-extended plenum system is appreciably less.

The utilization of an extended or semi-extended plenum is not actually a different method of duct or system sizing. It is merely the combination of good design and cost savings ideas, using conventional duct sizing techniques.

| System | Duct Losses (Pascals) | Fitting Losses (Pascals) | Total Losses (Pascals) | Fan W Required |
|---|---|---|---|---|
| "A" - equal friction method | 316 | 165 | 481 | 4.7 |
| "B" semi-extended plenum method | 269 | 55 | 321 | 3.4 |

**Table 7–9 Semi–Extended Plenum Comparison**

| Description | System "A" Equal Friction Method | | | System "B" - Semi-Extended Plenum Method | | |
|---|---|---|---|---|---|---|
| | lbs. | Shop Labor | Field Labor | lbs. | Shop Labor | Field Labor |
| Straight Duct | 2323 | 7 | 50 | 2553 | 8 | 55 |
| Fittings | 243 | 5 | 11 | 90 | 2 | 4 |
| Totals | 2566 | 73 hours | | 2643 | 69 hours | |

**Table 7–10 Semi–Extended Plenum Installation Cost Comparison**

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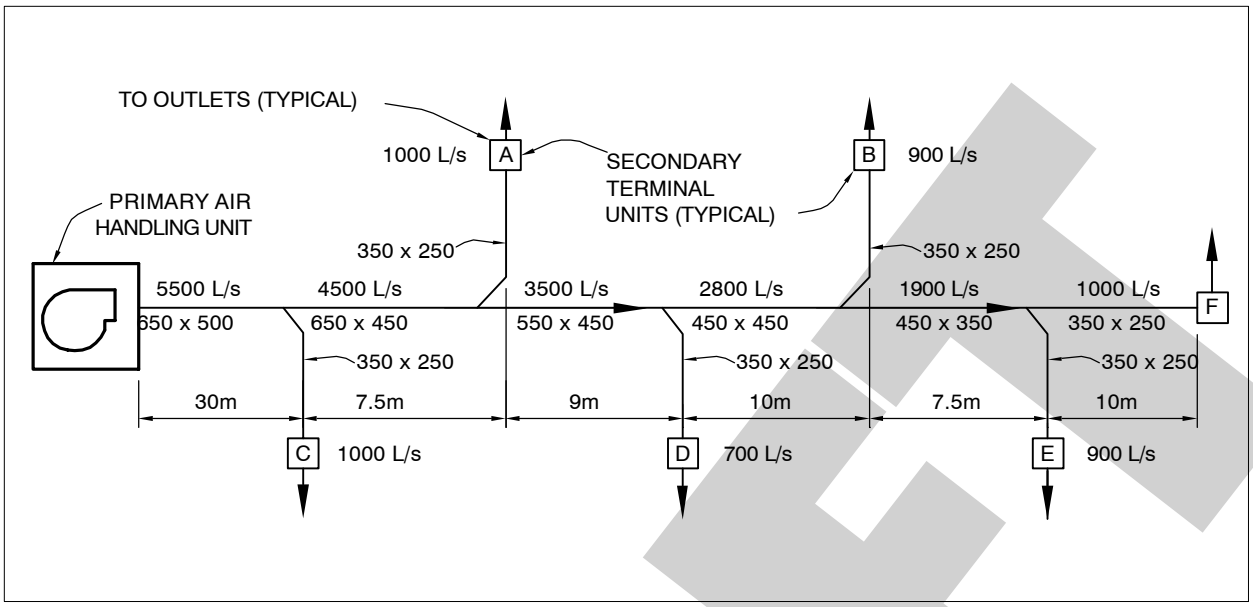


**FIGURE 7–9M "B" MODIFIED BY SEMI–EXTENDED PLENUM CONCEPT**

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DUCT SIZING WORK SHEET

DATE _____ PAGE_____

PROJECT _____ LOCATION _____ SYSTEM

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW (CFM) | FRICTION PER METER | VELOC-ITY (M/S) | $V_p$ (IN.WG) | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMU-LATIVE LOSS | TOTAL LOSS |
| 1 | | | | | | | | | | | | | | | | |
| 2 | | | | | | | | | | | | | | | | |
| 3 | | | | | | | | | | | | | | | | |
| 4 | | | | | | | | | | | | | | | | |
| 5 | | | | | | | | | | | | | | | | |
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| 25 | | | | | | | | | | | | | | | | |

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**FIGURE 7–10 DUCT SIZING WORK SHEET (I–P)**

Note: * Indicates duct lining used. Sizes are interior dimensions.

DUCT SIZING WORK SHEET

DATE _____ PAGE_____

PROJECT _____ LOCATION _____ SYSTEM

| | A | B | C | | D | E | F | G | H | I | J | K | L | M | N | O |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | DUCT RUN | SEC–TION | ITEM | | FLOW (L/S) | FRICTION PER METER | VELOC-ITY (M/S) | $V_p$ (PA) | LOSS COEFF | EQUIV DIAM | RECTANGU-LAR SIZE | CORR FACT. | LOSS PER ITEM | LOSS PER SECTION | CUMULA-TIVE LOSS | TOTAL LOSS |
| 1 | | | | | | | | | | | | | | | | |
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| 25 | | | | | | | | | | | | | | | | |

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**FIGURE 7–10M DUCT SIZING WORK SHEET (SI)**

Note: * Indicates duct lining used. Sizes are interior dimensions.

CHAPTER 8

# PRESSURE LOSS OF SYSTEMS COMPONENTS

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## 8.1 SCOPE

The pressure loss data provided in this chapter for each major duct component represents reasonable pressure loss allowance for each component to be installed. Where a pressure loss range is provided, the designer may select the high or low figure shown, or some representative average. A range has been provided, rather than a specific value, when data indicates that the pressure drop will vary at a given point (depending on the manufacturer selected). It is possible to select a component that will provide a pressure loss outside the range shown. However, for preliminary duct system design, the pressure loss figures shown should be adequate.

To find the pressure loss for each component, first determine the appropriate velocity. If a free-area velocity is indicated, it must be used for all calculations. *Please note the type of area used on each individual chart*. Representative free area can be selected from the free-area tables for louvers. This should be adequate for preliminary duct system design. All system component pressure losses should be entered on the SMACNA Duct Sizing Worksheet during preliminary design, *see* Figures 7-10 and 7-10M.

### 8.1.1 Final Design Data

After completing the preliminary duct system design, actual equipment selection can be made from manufacturer's data. This includes selection of heating coil capacity, size, flow rate of heating media, and static pressure loss. Preliminary pressure loss values must be replaced with actual manufacturer's pressure values for the selected components. A system static pressure loss checklist (Figure 8-1) has been provided to aid the designer in the final review of the system design to assure that all system pressure losses have been considered.

### 8.1.2 Submittal Review

The designer must review all equipment submittal drawings and data to be sure the purchased equipment has pressure losses compatible with values allowed in the duct system design. Any significant pressure loss changes that are allowed must be noted, and the total system pressure adjusted accordingly. If the pressure loss changes are not within the capabilities of the fans, then different equipment must be selected or the sections of the duct system must be redesigned.

The fan pressure must be capable of efficiently overcoming the system total pressure loss and the proper economic balance must be brought about between the first cost and the overall operating costs (life-cycle costs).

## 8.2 USE OF TABLES AND CHARTS

### 8.2.1 Filters

Table 8-1 contains static pressure loss ranges for a sampling of the most common types of air filters used in HVAC systems. Note how the static pressure loss will increase, the loss coefficient (c) more than doubles, significantly from clean to dirty. Actual system design should be based on manufacture's data for the specific filter types and sizes being considered.

**I-P UNITS**

| fpm | $V_p$ | C = 0.20 | C = 0.52 | Figure 8-2 |
|---|---|---|---|---|
| 1,000 | 0.06 | 0.012 in. wg | 0.031 in. wg | 0.032 in. wg |
| 1,500 | 0.14 | 0.028 in. wg | 0.073 in. wg | 0.065 in. wg |
| 2,000 | 0.25 | 0.050 in. wg | 0.130 in. wg | 0.14 in. wg |
| 3,000 | 0.56 | 0.112 in. wg | 0.292 in. wg | 0.32 in. wg |

**SI UNITS**

| m/s | $V_p$ | C = 0.20 | C = 0.52 | Figure 8-2 |
|---|---|---|---|---|
| 5.0 | 15.1 | 3.0 Pa | 7.8 Pa | 8.0 Pa |
| 7.5 | 33.9 | 6.8 Pa | 17.6 Pa | 16.2 Pa |
| 10.0 | 60.2 | 12.0 Pa | 31.3 Pa | 34.9 Pa |
| 15.0 | 135.0 | 27.1 Pa | 70.2 Pa | 79.7 Pa |

**Table 8–1 Filter Pressure Loss Data**

Since the previous edition of this manual was published ASHRAE has developed a new method of rating filters— Minimum Efficiency Reporting Value (MERV). This has led to other ASHRAE standards requiring minimum MERV filters in certain applications. Also, new types of filters and certain filter designs—particularly pleated filters— have become much more widely available and cost competitive. Pleated filters are an especially attractive filter option because they offer the combination of high dirt-holding capacity and efficient filtration. Designers should recognize that the life cycle cost of filters include the labor to change them, the increasing air resistance created as filters load, and cost of the filter. Designers should look at filters as an investment in clean coils, lower energy use-due to higher airflows over the life of a premium filter, reduced housekeeping.

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| MERV Rating Range | Particle Sizes Range | Examples of Particles |
|---|---|---|
| MERV 1-4 | 10 microns and larger | Pollen, carpet fibers, dust mites |
| MERV 5-8 | 3 to 10 microns | Mold spores, cement dust |
| MERV 9-12 | 1 to 3 microns | Coal dust, lead dust, welding fumes |
| MERV 13-16 | 0.3 to 1 microns | Tobacco smoke, bacteria |
| MERV 17-20 | 0.3 microns and smaller | viruses, carbon dust |

The last, most efficient MERV category includes the High-Efficiency Particle Arrestance (HEPA) filter class. For more details regarding filters and the MERV method refer to ASHRAE Standard 52.2.

In addition to media filters, an entire range of air-filtration and treatment technology is available to help designers meet indoor air quality requirements. Some of the more widely-used technologies include:

- Gaseous phase filters use activated carbon or charcoal to remove odors and gases.
- Ultraviolet light can be used to destroy biological and chemical contaminates.
- Electric static filters impart a static charge to particles and then "attract" the charged particles to a plate in the air stream that has the opposite charge.

The effectiveness of these technologies is dependent on an entire range of technical and maintenance requirements. Superior filtration requires examining the entire HVAC system and its entire life cycle.

### 8.2.2 Temporary Heat

One particularly important issue is the operation of the permanent HVAC system as a temporary construction air condition or heater. It is virtually never in the building owner's best interest to use the permanent HVAC system before the building itself is closed and ready for occupancy. Permanent HVAC systems, and the filter system, are designed for an air tight building that is kept clean. For more information regarding the issues associated with using permanent HVAC for temporary heating refer to SMACNA's *Indoor Air Quality in Occupied Buildings Under Construction*.

### 8.2.3 Dampers

Shop-fabricated butterfly dampers without frames have a "wide open" coefficient of C = 0.20 for round and C = 0.04 for rectangular. Parallel- or opposed-blade, crimped-leaf, shop-fabricated dampers with .25 in. metal frames have a loss coefficient of C = 0.52, *see* Table A-15, Figures E and F. These values can be compared with manufactured 36 × 36 in. (900 × 900 mm) volume dampers with frames (Figure 8-2) by selecting some duct velocities:

Shop-fabricated dampers with frames would have loss values similar to the *volume dampers* in Figure 8-2.

Volume dampers are needed to balance even the most carefully designed system. But excessive use, particularly dampers with high-pressure losses where tight shut-off is not essential, can quickly build up the duct pressure losses and become a source of noise.

For outside-air and mixed-air dampers, use a minimum velocity of 1500 fpm (7.5 m/s). Include a pressure drop of at least 0.25 in. wg (63 Pa) in the mixed-air plenum for an "air blender" to prevent stratification and coil freezing in northern climates.

Fire dampers and other special dampers, such as relief dampers, must comply with code requirements and unusual job conditions. Therefore, the designer must add the pressure losses of these required items to the system totals, *see* Figures 8-3 and 8-4.

### 8.2.4 Duct System Apparatus

Figures 8-5 to 8-11 contain pressure loss data for commonly used HEATING AND COOLING COILS.

Data needed for determining the free area of Louvers is furnished in Tables 8-2 to 8-4, as the "free area" velocity must be used to obtain louver pressure losses in Figure 8-13. Tables for other types of entries and exits can be found in the Appendix.

The pressure loss data for several sizes and types of Sound Traps should be used for preliminary calculations only! Manufacturer's data should be used for the final design calculations (Figures 8-14 to 8-18).

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Fan System _____ Project _____

Duct System Pressure Loss From Duct Sizing Form _____

*Allowance for Offsets, Etc., Required in Field _____

**Building Pressure Allowance (Supply Only) _____

TOTAL SYSTEM PRESSURE _____

**All Duct Accessories & Equipment Pressure Losses Must be Included in the Calculations entered in the SMACNA Duct Sizing Form**

- ☐ Air Monitor Devices
- ☐ Air Terminal Devices
- ☐ Air Washers
- ☐ Air Washer, Sprayed Coil
- ☐ Boxes, Constant Volume Mixing
- ☐ Boxes, Dual Duct Mixing
- ☐ Boxes, Induction Mixing
- ☐ Boxes, Variable Volume
- ☐ Coils, Cooling (Wet Surface)
- ☐ Coils, Heating (Dry Surface)
- ☐ Dampers, Backdraft
- ☐ Dampers, Fire
- ☐ Dampers, Opposed Blade Volume
- ☐ Dampers, Single Blade Volume
- ☐ Dampers, Smoke
- ☐ Diffusers
- ☐ Duct Heaters, Direct Fired
- ☐ Duct Heaters, Electric
- ☐ Eliminators, Moisture
- ☐ Energy Recovery Equipment
- ☐ Extractors
- ☐ Filters
- ☐ Flexible Duct
- ☐ Grilles
- ☐ Heat Exchangers, Air–to–Air
- ☐ Heat Exchangers, Direct Fired
- ☐ Heat Exchangers, Water–to–Air
- ☐ Humidifiers
- ☐ Louvers
- ☐ Obstructions
- ☐ Orifices
- ☐ Registers
- ☐ Screens
- ☐ Sound Traps
- ☐ Static Plates, Orifice, Eggcrate, etc.
- ☐ Surface Correction Factor
- ☐ System Effect Factors
- ☐ Temperature & Altitude Correction Factor
- ☐ Turning Vanes

* Some allowance must be made for offsets, etc., required in the field to avoid conflicts with plumbing, piping, electric, sprinklers, etc. A reasonable estimate should be made to provide a pressure loss for anticipated additional fittings; however, this allowance should not be over estimated. Over estimation will provide an oversized fan selection which will waste energy during operation of the system.

** Building pressure allowance for supply systems should be determined from building requirements considering acceptable building infiltration. Normally, 0.05 to 0.1 in. wg ( 12 to 25 Pa) static pressure allowance for building pressurization value based upon individual system requirements. Consideration should include elevator shaft ventilation requirements, tightness of building construction, stack effect, etc.

**FIGURE 8–1 SYSTEM PRESSURE LOSS CHECK LIST**

DRAFT

| Square Feet (Effective) | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Width (in.) | | | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Height (in.) | 4 | 8 | 12 | 16 | 20 | 24 | 28 | 32 | 36 | 40 | 44 | 48 | 52 | 56 | 60 | 64 | 68 | 72 | 76 | 80 | 84 | 88 | 92 | 96 | 100 | 104 | 108 | 112 | 116 |
| 6 | 0.01 | 0.04 | 0.06 | 0.09 | 0.1 | 0.1 | 0.2 | 0.2 | 0.2 | 0.2 | 0.3 | 0.3 | 0.3 | 0.3 | 0.4 | 0.4 | 0.4 | 0.4 | 0.5 | 0.5 | 0.5 | 0.5 | 0.5 | 0.6 | 0.6 | 0.6 | 0.6 | 0.7 | 0.6 |
| 8 | 0.03 | 0.09 | 0.2 | 0.2 | 0.3 | 0.3 | 0.4 | 0.4 | 0.5 | 0.6 | 0.6 | 0.7 | 0.7 | 0.8 | 0.9 | 0.9 | 1.0 | 1.0 | 1.1 | 1.2 | 1.2 | 1.3 | 1.3 | 1.4 | 1.5 | 1.5 | 1.6 | 1.6 | 1.7 |
| 12 | 0.05 | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.6 | 1.7 | 1.8 | 1.9 | 2.0 | 2.1 | 2.2 | 2.3 | 2.4 | 2.4 | 2.0 |
| 16 | 0.08 | 0.2 | 0.4 | 0.5 | 0.6 | 0.8 | 0.9 | 1.1 | 1.2 | 1.4 | 1.5 | 1.6 | 1.8 | 1.9 | 2.1 | 2.2 | 2.3 | 2.5 | 2.6 | 2.8 | 2.9 | 3.1 | 3.2 | 3.3 | 3.5 | 3.6 | 3.8 | 3.9 | 4.0 |
| 20 | 0.1 | 0.3 | 0.5 | 0.6 | 0.8 | 1.0 | 1.2 | 1.3 | 1.5 | 1.7 | 1.9 | 2.1 | 2.2 | 2.4 | 2.6 | 2.8 | 2.9 | 3.1 | 3.3 | 3.5 | 3.6 | 3.8 | 4.0 | 4.2 | 4.4 | 4.5 | 4.7 | 4.9 | 5.1 |
| 24 | 0.1 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.5 | 1.7 | 1.9 | 2.1 | 2.4 | 2.6 | 2.8 | 3.0 | 3.3 | 3.5 | 3.7 | 3.9 | 4.1 | 4.4 | 4.6 | 4.8 | 5.0 | 5.3 | 5.5 | 5.7 | 5.9 | 6.1 | 6.4 |
| 28 | 0.2 | 0.4 | 0.7 | 1.0 | 1.2 | 1.5 | 1.7 | 2.0 | 2.3 | 2.5 | 2.8 | 3.1 | 3.3 | 3.6 | 3.9 | 4.1 | 4.4 | 4.7 | 4.9 | 5.2 | 5.5 | 5.6 | 6.0 | 6.3 | 6.5 | 6.8 | 7.1 | 7.3 | 7.6 |
| 32 | 0.2 | 0.5 | 0.8 | 1.1 | 1.4 | 1.7 | 2.0 | 2.3 | 2.6 | 2.9 | 3.2 | 3.5 | 3.8 | 4.1 | 4.4 | 4.7 | 5.1 | 5.4 | 5.7 | 6.0 | 6.3 | 6.6 | 6.9 | 7.2 | 7.5 | 7.8 | 8.1 | 8.4 | 8.7 |
| 36 | 0.2 | 0.6 | 0.9 | 1.3 | 1.6 | 2.0 | 2.3 | 2.7 | 3.0 | 3.4 | 3.7 | 4.1 | 4.5 | 4.8 | 5.2 | 5.5 | 5.9 | 6.2 | 6.6 | 6.9 | 7.3 | 7.6 | 8.0 | 8.4 | 8.7 | 9.1 | 9.4 | 9.8 | 10.1 |
| 40 | 0.2 | 0.6 | 1.0 | 1.4 | 1.8 | 2.2 | 2.5 | 2.9 | 3.3 | 3.7 | 4.1 | 4.5 | 4.9 | 5.2 | 5.6 | 6.0 | 6.4 | 6.8 | 7.2 | 7.6 | 7.9 | 8.3 | 8.7 | 9.1 | 9.5 | 9.9 | 10.3 | 10.7 | 11.0 |
| 44 | 0.2 | 0.7 | 1.1 | 1.5 | 1.9 | 2.3 | 2.7 | 3.1 | 3.5 | 4.0 | 4.4 | 4.8 | 5.2 | 5.6 | 6.0 | 6.4 | 6.9 | 7.3 | 7.7 | 8.1 | 8.5 | 8.9 | 9.3 | 9.7 | 10.2 | 10.6 | 11.0 | 11.4 | 11.8 |
| 48 | 0.3 | 0.7 | 1.2 | 1.7 | 2.2 | 2.6 | 3.1 | 3.6 | 4.1 | 4.5 | 5.0 | 5.5 | 5.9 | 6.4 | 6.9 | 7.4 | 7.8 | 8.3 | 8.8 | 9.2 | 9.7 | 10.2 | 10.7 | 11.1 | 11.6 | 12.1 | 12.6 | 13.0 | 13.5 |
| 52 | 0.3 | 0.8 | 1.4 | 1.9 | 2.4 | 3.0 | 3.5 | 4.0 | 4.6 | 5.1 | 5.6 | 6.2 | 6.7 | 7.2 | 7.7 | 8.3 | 8.8 | 9.3 | 9.9 | 10.4 | 10.9 | 11.5 | 12.0 | 12.5 | 13.1 | 13.6 | 14.1 | 14.7 | 15.2 |
| 56 | 0.3 | 0.9 | 1.4 | 2.0 | 2.6 | 3.1 | 3.7 | 4.3 | 4.8 | 5.4 | 5.9 | 6.5 | 7.1 | 7.6 | 8.2 | 8.7 | 9.3 | 9.9 | 10.4 | 11.0 | 11.5 | 12.1 | 12.7 | 13.2 | 13.8 | 14.3 | 14.9 | 15.5 | 16.0 |
| 60 | 0.4 | 1.0 | 1.6 | 2.2 | 2.8 | 3.4 | 4.0 | 4.7 | 5.3 | 5.9 | 6.5 | 7.1 | 7.7 | 8.3 | 8.9 | 9.6 | 10.2 | 10.8 | 11.4 | 12.0 | 12.6 | 13.2 | 13.9 | 14.5 | 15.1 | 15.7 | 16.3 | 16.9 | 17.5 |
| 64 | 0.4 | 1.0 | 1.7 | 2.3 | 3.0 | 3.6 | 4.3 | 4.9 | 5.6 | 6.2 | 6.9 | 7.5 | 8.2 | 8.8 | 9.5 | 10.1 | 10.8 | 11.4 | 12.1 | 12.7 | 13.4 | 14.0 | 14.7 | 15.3 | 16.0 | 16.6 | 17.3 | 17.9 | 18.6 |
| 68 | 0.4 | 1.1 | 1.8 | 2.5 | 3.2 | 3.9 | 4.6 | 5.3 | 6.0 | 6.7 | 7.4 | 8.1 | 8.7 | 9.4 | 10.1 | 10.8 | 11.5 | 12.2 | 12.9 | 13.6 | 14.3 | 15.0 | 15.7 | 16.4 | 17.1 | 17.8 | 18.5 | 19.2 | 19.9 |
| 72 | 0.4 | 1.2 | 1.9 | 2.6 | 3.4 | 4.1 | 4.9 | 5.6 | 6.3 | 7.1 | 7.8 | 8.5 | 9.3 | 10.0 | 10.8 | 11.5 | 12.2 | 13.0 | 13.7 | 14.5 | 15.2 | 15.9 | 16.7 | 17.4 | 18.1 | 18.9 | 19.6 | 20.4 | 21.1 |
| 76 | 0.5 | 1.2 | 2.0 | 2.8 | 3.6 | 4.3 | 5.1 | 5.9 | 6.7 | 7.4 | 8.2 | 9.0 | 9.8 | 10.6 | 11.3 | 12.1 | 12.9 | 13.7 | 14.4 | 15.2 | 16.0 | 16.8 | 17.5 | 18.3 | 19.1 | 19.9 | 20.6 | 21.4 | 22.2 |
| 80 | 0.5 | 1.3 | 2.1 | 3.0 | 3.8 | 4.6 | 5.4 | 6.3 | 7.1 | 7.9 | 8.7 | 9.6 | 10.4 | 11.2 | 12.1 | 12.9 | 13.7 | 14.5 | 15.4 | 16.2 | 17.0 | 17.8 | 18.7 | 19.5 | 20.3 | 21.1 | 22.0 | 22.8 | 23.6 |
| 84 | 0.5 | 1.4 | 2.2 | 3.1 | 3.9 | 4.8 | 5.7 | 6.5 | 7.4 | 8.2 | 9.1 | 9.9 | 10.8 | 11.7 | 12.5 | 13.4 | 14.2 | 15.1 | 16.0 | 16.8 | 17.7 | 18.5 | 19.5 | 20.2 | 21.1 | 22.0 | 22.8 | 23.7 | 24.5 |
| 88 | 0.5 | 1.4 | 2.4 | 3.3 | 4.2 | 5.1 | 6.0 | 6.9 | 7.9 | 8.8 | 9.7 | 10.6 | 11.5 | 12.4 | 13.3 | 14.3 | 15.2 | 16.1 | 17.0 | 17.9 | 18.8 | 19.7 | 20.7 | 21.6 | 22.5 | 23.4 | 24.3 | 25.2 | 26.2 |
| 92 | 0.6 | 1.5 | 2.4 | 3.4 | 4.3 | 5.3 | 6.2 | 7.2 | 8.1 | 9.1 | 10.0 | 10.9 | 11.9 | 12.8 | 13.8 | 14.7 | 15.7 | 16.6 | 17.6 | 18.5 | 19.4 | 20.4 | 21.3 | 22.3 | 23.2 | 24.2 | 25.1 | 26.1 | 27.0 |
| 96 | 0.6 | 1.6 | 2.6 | 3.6 | 4.6 | 5.6 | 6.6 | 7.6 | 8.6 | 9.6 | 10.6 | 11.6 | 12.6 | 13.6 | 14.6 | 15.6 | 16.6 | 17.6 | 18.6 | 19.7 | 20.7 | 21.7 | 22.7 | 23.7 | 24.7 | 25.7 | 26.7 | 27.7 | 28.7 |

**Table 8–2 Louver Free Area Chart 2 in. Blades at 45 Degree Angle**

*Conversions:* 1 mm = 0.03937 in.; 1 $m^2$ = 10.76 $ft^2$
1 in. = 25.4 mm; 1 $ft^2$ = 0.929 $m^2$

DRAFT

| Square Feet (Effective) | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Width (in.) | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Height (in.) | 4 | 8 | 12 | 16 | 20 | 24 | 28 | 32 | 36 | 40 | 44 | 48 | 52 | 56 | 60 | 64 | 68 | 72 | 76 | 80 | 84 | 88 | 92 | 96 | 100 | 104 | 108 |
| 10 | 0.03 | 0.09 | 0.2 | 0.2 | 0.3 | 0.3 | 0.4 | 0.5 | 0.5 | 0.6 | 0.7 | 0.7 | 0.8 | 0.8 | 0.9 | 1.0 | 1.0 | 1.1 | 1.2 | 1.2 | 1.3 | 1.3 | 1.4 | 1.5 | 1.5 | 1.6 | 1.7 |
| 12 | 0.03 | 0.1 | 0.2 | 0.3 | 0.3 | 0.4 | 0.5 | 0.5 | 0.6 | 0.7 | 0.8 | 0.8 | 0.9 | 1.0 | 1.1 | 1.1 | 1.2 | 1.3 | 1.3 | 1.4 | 1.5 | 1.6 | 1.6 | 1.7 | 1.8 | 1.9 | 1.9 |
| 16 | 0.08 | 0.3 | 0.4 | 0.6 | 0.8 | 0.9 | 1.1 | 1.3 | 1.5 | 1.6 | 1.8 | 2.0 | 2.1 | 2.3 | 2.5 | 2.7 | 2.8 | 3.0 | 3.2 | 3.3 | 3.5 | 3.7 | 3.9 | 4.0 | 4.2 | 4.4 | 4.6 |
| 20 | 0.09 | 0.3 | 0.5 | 0.7 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 | 2.2 | 2.3 | 2.5 | 2.7 | 2.9 | 3.1 | 3.3 | 3.5 | 3.7 | 3.8 | 4.0 | 4.2 | 4.4 | 4.6 | 4.8 | 5.0 |
| 24 | 0.1 | 0.4 | 0.6 | 0.9 | 1.1 | 1.4 | 1.6 | 1.9 | 2.1 | 2.4 | 2.6 | 2.9 | 3.1 | 3.4 | 3.6 | 3.9 | 4.1 | 4.4 | 4.6 | 4.9 | 5.1 | 5.4 | 5.6 | 5.9 | 6.1 | 6.4 | 6.6 |
| 28 | 0.1 | 0.5 | 0.8 | 1.1 | 1.4 | 1.7 | 2.0 | 2.3 | 2.7 | 4.0 | 3.3 | 3.6 | 3.9 | 4.2 | 4.5 | 4.8 | 5.2 | 5.5 | 5.8 | 6.1 | 6.4 | 6.7 | 7.0 | 7.3 | 7.7 | 8.0 | 8.3 |
| 32 | 0.1 | 0.5 | 0.9 | 1.3 | 1.7 | 2.0 | 2.4 | 2.8 | 3.1 | 3.5 | 3.9 | 4.2 | 4.6 | 5.0 | 5.3 | 5.7 | 6.1 | 6.4 | 6.8 | 7.2 | 7.5 | 7.9 | 8.3 | 8.6 | 9.0 | 9.4 | 9.7 |
| 36 | 0.2 | 0.6 | 1.1 | 1.5 | 1.9 | 2.3 | 2.7 | 3.1 | 3.6 | 4.0 | 4.4 | 4.8 | 5.2 | 5.7 | 6.1 | 6.5 | 6.9 | 7.3 | 7.8 | 8.2 | 8.6 | 9.0 | 9.4 | 9.9 | 10.3 | 10.7 | 11.1 |
| 40 | 0.2 | 0.7 | 1.2 | 1.7 | 2.1 | 2.6 | 3.1 | 3.5 | 4.0 | 4.5 | 5.0 | 5.4 | 5.9 | 6.4 | 6.9 | 7.3 | 7.8 | 8.3 | 8.7 | 9.2 | 9.7 | 10.2 | 10.6 | 11.1 | 11.6 | 12.0 | 12.5 |
| 44 | 0.2 | 0.8 | 1.3 | 1.8 | 2.3 | 2.9 | 3.4 | 3.9 | 4.4 | 4.9 | 5.5 | 6.0 | 6.5 | 7.0 | 7.6 | 8.1 | 8.6 | 9.1 | 9.6 | 10.1 | 10.7 | 11.2 | 11.7 | 12.2 | 12.8 | 13.3 | 13.8 |
| 48 | 0.3 | 0.8 | 1.4 | 2.0 | 2.6 | 3.2 | 3.7 | 4.3 | 4.9 | 5.4 | 6.0 | 6.6 | 7.2 | 7.7 | 8.3 | 8.9 | 9.5 | 10.0 | 10.6 | 11.2 | 11.8 | 12.3 | 12.9 | 13.5 | 14.0 | 14.6 | 15.7 |
| 52 | 0.3 | 1.0 | 1.7 | 2.3 | 3.0 | 3.7 | 4.4 | 5.0 | 5.7 | 6.4 | 7.0 | 7.7 | 8.4 | 9.1 | 9.7 | 10.4 | 11.1 | 11.7 | 12.4 | 13.1 | 13.7 | 14.4 | 15.1 | 15.8 | 16.4 | 17.1 | 17.7 |
| 56 | 0.3 | 1.1 | 1.8 | 2.5 | 3.3 | 4.0 | 4.7 | 5.4 | 6.1 | 6.9 | 7.6 | 8.3 | 9.0 | 9.8 | 10.5 | 11.2 | 11.9 | 12.6 | 13.4 | 14.1 | 14.8 | 15.5 | 16.3 | 17.0 | 17.7 | 18.4 | 19.2 |
| 60 | 0.3 | 1.1 | 1.9 | 2.6 | 3.4 | 4.1 | 4.9 | 5.6 | 6.4 | 7.1 | 7.9 | 8.6 | 9.4 | 10.1 | 10.9 | 11.6 | 12.4 | 13.1 | 13.9 | 14.6 | 15.4 | 16.1 | 16.9 | 17.6 | 18.4 | 19.1 | 19.9 |
| 64 | 0.4 | 1.2 | 2.0 | 2.8 | 3.7 | 4.5 | 5.3 | 6.1 | 6.9 | 7.7 | 8.5 | 9.3 | 10.1 | 11.0 | 11.8 | 12.6 | 13.4 | 14.2 | 15.0 | 15.8 | 16.7 | 17.5 | 18.3 | 19.1 | 19.9 | 20.7 | 21.5 |
| 68 | 0.4 | 1.3 | 2.2 | 3.0 | 3.9 | 4.8 | 5.6 | 6.5 | 7.4 | 8.3 | 9.1 | 10.0 | 10.9 | 11.7 | 12.6 | 13.5 | 14.3 | 15.2 | 16.1 | 16.9 | 17.8 | 18.7 | 19.5 | 20.4 | 21.3 | 22.2 | 23.0 |
| 72 | 0.4 | 1.4 | 2.3 | 3.3 | 4.2 | 5.2 | 6.1 | 7.0 | 8.0 | 8.9 | 9.8 | 10.8 | 11.7 | 12.6 | 13.6 | 14.5 | 15.4 | 16.4 | 17.3 | 18.3 | 19.2 | 20.2 | 21.1 | 22.0 | 23.0 | 23.9 | 24.8 |
| 76 | 0.4 | 1.4 | 2.4 | 3.4 | 4.4 | 5.3 | 6.3 | 7.3 | 8.3 | 9.2 | 10.2 | 11.2 | 12.2 | 13.1 | 14.1 | 15.1 | 16.0 | 17.0 | 18.0 | 19.0 | 19.9 | 20.9 | 21.8 | 22.8 | 23.8 | 24.8 | 25.8 |
| 80 | 0.5 | 1.5 | 2.6 | 2.6 | 4.6 | 5.6 | 6.6 | 7.7 | 8.7 | 9.7 | 10.7 | 11.7 | 12.7 | 13.8 | 14.8 | 15.8 | 16.8 | 17.8 | 18.9 | 19.9 | 20.9 | 21.9 | 23.0 | 24.0 | 25.0 | 26.0 | 27.0 |
| 84 | 0.5 | 1.6 | 2.7 | 3.8 | 4.8 | 5.9 | 7.0 | 8.0 | 9.1 | 10.2 | 11.3 | 12.3 | 13.4 | 14.5 | 15.5 | 16.6 | 17.7 | 18.8 | 19.8 | 20.9 | 22.0 | 23.0 | 241 | 25.2 | 26.3 | 27.4 | 28.4 |
| 88 | 0.5 | 1.7 | 2.9 | 4.0 | 5.3 | 6.4 | 7.6 | 8.8 | 10.0 | 11.1 | 12.3 | 13.5 | 14.6 | 15.8 | 17.0 | 18.2 | 19.3 | 20.5 | 21.6 | 22.8 | 24.0 | 25.2 | 26.4 | 27.6 | 28.7 | 29.9 | 31.0 |
| 92 | 0.6 | 1.8 | 3.1 | 4.3 | 5.5 | 6.7 | 7.9 | 9.2 | 10.4 | 11.6 | 12.8 | 14.1 | 15.3 | 16.5 | 17.7 | 18.9 | 20.2 | 21.4 | 22.6 | 23.8 | 25.0 | 26.2 | 27.5 | 28.7 | 30.0 | 31.2 | 32.4 |
| 96 | 0.6 | 1.8 | 3.1 | 4.4 | 5.6 | 6.9 | 8.1 | 9.4 | 10.6 | 11.9 | 13.1 | 14.4 | 15.6 | 16.9 | 18.1 | 19.4 | 20.6 | 21.9 | 23.2 | 24.4 | 25.6 | 26.9 | 28.1 | 29.4 | 30.6 | 31.9 | 33.1 |
| 100 | 0.6 | 1.9 | 3.3 | 4.6 | 5.9 | 7.2 | 8.5 | 9.8 | 11.1 | 12.5 | 13.8 | 15.1 | 16.4 | 17.7 | 19.0 | 20.4 | 21.6 | 23.0 | 24.2 | 25.6 | 26.9 | 28.2 | 29.5 | 30.8 | 32.2 | 33.4 | 34.8 |
| 104 | 0.6 | 2.0 | 3.4 | 4.8 | 6.2 | 7.5 | 8.9 | 10.3 | 11.6 | 13.0 | 14.3 | 15.7 | 17.1 | 18.4 | 19.8 | 21.2 | 22.6 | 23.9 | 25.2 | 26.6 | 28.0 | 29.4 | 30.7 | 32.1 | 33.4 | 34.9 | 36.2 |
| 108 | 0.7 | 2.1 | 3.6 | 5.0 | 6.4 | 7.8 | 9.3 | 10.7 | 12.1 | 13.5 | 15.0 | 16.4 | 17.8 | 19.2 | 20.6 | 22.1 | 23.5 | 25.0 | 26.4 | 27.8 | 29.2 | 30.6 | 32.0 | 33.5 | 35.0 | 36.4 | 37.8 |

**Table 8–3 Louver Free Area Chart 4 in. Blades at 45 Degree Angle**

*Conversions:* 1 mm = 0.03937 in.; 1 $m^2$ = 10.76 $ft^2$
1 in. = 25.4 mm; 1 $ft^2$ = 0.0929 $m^2$

DRAFT

| Square Feet (Effective) | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Width (in.) | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| Height (in.) | 4 | 8 | 12 | 16 | 20 | 24 | 28 | 32 | 36 | 40 | 44 | 48 | 52 | 56 | 60 | 64 | 68 | 72 | 76 | 80 | 84 | 88 | 92 | 96 | 100 | 104 | 108 |
| 14 | 0.05 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 | 1.7 | 1.8 | 1.9 | 2.0 | 2.1 | 2.2 | 2.3 | 2.4 | 2.5 | 2.7 | 2.8 |
| 16 | 0.06 | 0.2 | 0.3 | 0.5 | 0.6 | 0.7 | 0.9 | 1.0 | 1.1 | 1.3 | 1.4 | 1.5 | 1.7 | 1.8 | 1.9 | 2.1 | 2.2 | 2.3 | 2.5 | 2.6 | 2.7 | 2.9 | 3.0 | 3.2 | 3.3 | 3.4 | 3.6 |
| 20 | 0.1 | 0.3 | 0.5 | 0.7 | 0.9 | 1.1 | 1.3 | 1.6 | 1.8 | 2.0 | 2.2 | 2.4 | 2.6 | 2.1 | 3.0 | 3.2 | 3.4 | 3.6 | 3.8 | 4.1 | 4.3 | 4.5 | 4.7 | 4.9 | 5.1 | 5.3 | 5.5 |
| 24 | 0.1 | 0.4 | 0.7 | 1.0 | 1.2 | 1.5 | 1.8 | 2.1 | 2.4 | 2.7 | 2.9 | 3.2 | 3.5 | 3.8 | 4.1 | 4.3 | 4.6 | 4.9 | 5.2 | 5.5 | 5.7 | 6.0 | 6.3 | 6.6 | 6.9 | 7.1 | 7.4 |
| 28 | 0.1 | 0.5 | 0.8 | 1.1 | 1.4 | 1.7 | 2.0 | 2.3 | 2.7 | 3.0 | 3.3 | 3.6 | 3.9 | 4.2 | 4.6 | 4.9 | 5.2 | 5.5 | 5.8 | 6.1 | 6.4 | 6.8 | 7.1 | 7.4 | 7.7 | 8.0 | 8.3 |
| 32 | 0.2 | 0.6 | 1.0 | 1.4 | 1.9 | 2.3 | 2.7 | 3.1 | 3.5 | 3.9 | 4.3 | 4.8 | 5.2 | 5.6 | 6.0 | 6.4 | 6.9 | 7.3 | 7.7 | 8.1 | 8.5 | 8.9 | 9.4 | 9.8 | 10.2 | 10.6 | 11.0 |
| 36 | 0.2 | 0.7 | 1.1 | 1.6 | 2.1 | 2.5 | 3.0 | 3.4 | 3.9 | 4.4 | 4.8 | 5.3 | 5.7 | 6.2 | 6.7 | 7.1 | 7.6 | 8.0 | 8.5 | 9.0 | 9.4 | 9.9 | 10.3 | 10.8 | 11.3 | 11.7 | 12.2 |
| 40 | 0.2 | 0.8 | 1.3 | 1.8 | 2.3 | 2.8 | 3.4 | 3.9 | 4.4 | 4.9 | 5.4 | 6.0 | 6.5 | 7.0 | 7.5 | 8.1 | 8.6 | 9.1 | 9.6 | 10.1 | 10.7 | 11.2 | 11.7 | 12.2 | 12.7 | 13.3 | 13.8 |
| 44 | 0.3 | 0.9 | 1.5 | 2.1 | 2.7 | 3.3 | 3.9 | 4.5 | 5.1 | 5.7 | 6.3 | 7.0 | 7.6 | 8.2 | 8.8 | 9.4 | 10.0 | 10.6 | 11.2 | 11.8 | 12.4 | 13.0 | 13.6 | 14.2 | 14.8 | 15.4 | 16.1 |
| 48 | 0.3 | 0.9 | 1.6 | 2.2 | 2.9 | 3.5 | 4.1 | 4.8 | 5.4 | 6.1 | 6.7 | 7.3 | 8.0 | 8.6 | 9.3 | 9.9 | 10.6 | 11.2 | 11.8 | 12.5 | 13.1 | 13.8 | 14.4 | 15.0 | 15.7 | 16.3 | 17.0 |
| 52 | 0.3 | 1.1 | 1.8 | 2.5 | 3.3 | 4.0 | 4.7 | 5.4 | 6.2 | 6.9 | 7.6 | 8.4 | 9.1 | 9.8 | 10.5 | 11.3 | 12.0 | 12.7 | 13.5 | 14.2 | 14.9 | 15.6 | 16.4 | 17.1 | 17.8 | 18.6 | 19.3 |
| 56 | 0.4 | 1.1 | 1.9 | 2.7 | 3.5 | 4.3 | 5.1 | 5.9 | 6.7 | 7.4 | 8.2 | 9.0 | 9.8 | 10.6 | 11.4 | 12.2 | 13.0 | 13.7 | 14.5 | 15.3 | 16.1 | 16.9 | 17.7 | 18.5 | 19.3 | 20.0 | 20.8 |
| 60 | 0.4 | 1.2 | 2.0 | 2.9 | 3.7 | 4.5 | 5.4 | 6.2 | 7.0 | 7.9 | 8.7 | 9.5 | 10.4 | 11.2 | 12.0 | 12.9 | 13.7 | 14.5 | 15.4 | 16.2 | 17.0 | 17.9 | 18.7 | 19.5 | 20.4 | 21.2 | 22.0 |
| 64 | 0.4 | 1.4 | 2.3 | 3.2 | 4.2 | 5.1 | 6.0 | 7.0 | 7.9 | 8.8 | 9.8 | 10.7 | 11.6 | 12.6 | 13.5 | 14.4 | 15.4 | 16.3 | 17.2 | 18.2 | 19.1 | 20.0 | 21.0 | 21.9 | 22.8 | 23.8 | 24.7 |
| 68 | 0.4 | 1.4 | 2.4 | 3.3 | 4.3 | 5.3 | 6.3 | 7.2 | 8.2 | 9.2 | 10.1 | 11.1 | 12.1 | 13.0 | 14.0 | 15.0 | 15.9 | 16.9 | 17.9 | 18.8 | 19.8 | 20.8 | 21.7 | 22.7 | 23.7 | 24.6 | 25.6 |
| 72 | 0.5 | 1.5 | 2.6 | 3.6 | 4.6 | 5.7 | 6.7 | 7.8 | 8.8 | 9.9 | 10.9 | 11.9 | 13.0 | 14.0 | 15.1 | 16.1 | 17.1 | 18.2 | 19.2 | 20.3 | 21.3 | 22.4 | 23.4 | 24.4 | 25.5 | 26.5 | 27.6 |
| 76 | 0.5 | 1.6 | 2.7 | 3.9 | 5.0 | 6.1 | 7.2 | 8.3 | 9.4 | 10.5 | 11.6 | 12.8 | 13.9 | 15.0 | 16.1 | 17.2 | 18.3 | 19.4 | 20.6 | 21.7 | 22.8 | 23.9 | 25.0 | 26.1 | 27.2 | 28.4 | 29.5 |
| 80 | 0.5 | 1.7 | 2.8 | 4.0 | 5.1 | 6.3 | 7.4 | 8.6 | 9.7 | 10.9 | 12.0 | 13.2 | 14.3 | 15.5 | 16.6 | 17.8 | 18.9 | 20.0 | 21.2 | 22.3 | 23.5 | 24.6 | 25.8 | 26.9 | 28.1 | 29.2 | 30.4 |
| 84 | 0.6 | 1.8 | 3.1 | 4.3 | 5.6 | 6.8 | 8.1 | 9.3 | 10.6 | 11.8 | 13.1 | 14.3 | 15.6 | 16.8 | 18.1 | 19.3 | 20.6 | 21.8 | 23.1 | 24.3 | 25.5 | 26.8 | 28.1 | 29.3 | 30.6 | 31.8 | 33.1 |
| 88 | 0.6 | 1.9 | 3.2 | 4.5 | 5.8 | 7.1 | 8.4 | 9.7 | 10.9 | 12.2 | 13.5 | 14.8 | 16.1 | 17.4 | 18.7 | 20.0 | 21.3 | 22.6 | 23.9 | 25.2 | 26.5 | 27.8 | 29.1 | 30.4 | 31.7 | 32.9 | 34.2 |
| 92 | 0.6 | 2.0 | 3.3 | 4.7 | 6.0 | 7.4 | 8.7 | 10.1 | 11.5 | 12.8 | 14.2 | 15.5 | 16.9 | 18.2 | 19.6 | 20.9 | 22.3 | 23.6 | 25.0 | 26.4 | 27.7 | 29.1 | 30.4 | 31.8 | 33.1 | 34.5 | 35.8 |
| 96 | 0.7 | 2.1 | 3.5 | 5.0 | 6.4 | 7.9 | 9.3 | 10.7 | 12.2 | 13.6 | 15.1 | 16.5 | 17.9 | 19.4 | 20.8 | 22.3 | 23.7 | 25.1 | 26.6 | 28.0 | 29.5 | 30.9 | 32.3 | 33.8 | 35.2 | 36.7 | 38.1 |
| 100 | 0.7 | 2.2 | 3.6 | 5.1 | 6.6 | 8.1 | 9.5 | 11.0 | 12.4 | 13.9 | 15.4 | 16.9 | 18.4 | 19.8 | 21.3 | 22.8 | 24.3 | 25.7 | 27.2 | 28.7 | 30.2 | 31.6 | 33.1 | 34.6 | 36.1 | 37.5 | 39.0 |
| 104 | 0.7 | 2.3 | 3.8 | 5.4 | 7.0 | 8.5 | 10.1 | 11.7 | 13.2 | 14.8 | 16.3 | 17.9 | 19.5 | 21.0 | 22.6 | 24.2 | 25.7 | 27.3 | 28.8 | 30.4 | 32.0 | 33.5 | 35.1 | 36.7 | 38.2 | 39.8 | 41.3 |
| 108 | 0.7 | 2.4 | 4.0 | 5.6 | 7.2 | 8.8 | 10.5 | 12.1 | 13.7 | 15.3 | 16.9 | 18.6 | 20.2 | 21.8 | 23.4 | 25.1 | 26.7 | 28.3 | 29.9 | 31.5 | 33.2 | 34.8 | 36.4 | 38.0 | 39.6 | 41.3 | 42.9 |

**Table 8–4 Louver Free Area Chart 6 in. Blades at 45 Degree Angle**

*Conversions:* 1 mm = 0.03937 in.; 1 $m^2$ = 10.76 $ft^2$
1 in. = 25.4 mm; 1 $ft^2$ = 0.0929 $m^2$

DRAFT

Pressure loss for a sprayed coil air washer can be obtained by combining the loss of an appropriate cooling coil (Figures 8-9 to 8-11) with the pressure loss of the three-bend moisture eliminator (Figure 8-19).

Duct mounted humidifiers normally offer minimal resistance to the duct airflow. Should the humidifier manifold be installed in a narrow duct where it would serve as an obstruction, Tables A-15G or H could be used to calculate the pressure loss or the duct could be expanded around the manifold, use the transition pressure losses.

Examples of pressure losses for energy recovery equipments: Air-to-air Plate Exchangers, Single-Tube Exchangers (Head Pipe Banks), Rotary Wheel, Runaround-Coil Systems, and Multiple-Tower Systems are shown in Figures 8-22 to 8-25, but must be used for *rough* estimates only. Heating and cooling coil data can be used for runaround-coil systems. The dry air evaporative cooler pressure loss data (Figure 8-26) is for the type of unit where the two air streams are totally separated, with no moisture interchange.

### 8.2.5 Room Air Terminal Devices

A word of caution is needed concerning air terminal device selection. Total pressure loss for room air terminals should be used to compute system total pressure loss. The fan is required to provide sufficient static pressure to overcome the static pressure loss through the air terminal and to overcome the velocity pressure loss as a result of delivering air at a given velocity through the air terminal opening into the room.

Air terminal pressure losses shown in Table 8-5 to 8-7 are total pressure losses. The values shown demonstrate comparative total pressure losses encountered using different types of air terminals. All air terminal devices included in each duct system should be chosen with similar total pressure drops, within 0.05 in. wg (12 Pa) If air terminals requiring substantially different total pressure loss values are included in the same duct system balancing will be difficult, if not impossible.

The designer should select air terminal devices from manufacturer's catalog data prior to completing the system design. Air terminal device total pressure requirements will vary with the terminal velocity selected, *see* Chapter 3.

### 8.2.6 Operating Conditions

The pressure drop data found in the following sub-sections C (Damper Charts) and D (Duct System Apparatus Charts) is for standard air (0.075 lb/ft$^3$., 70 degrees F, 29.92 in. Hg at sea level or 1.2041 kg/m$^3$, 20 degrees C, 101.325 kPa at sea level), and needs to be corrected where necessary to operating conditions by the following equation:

**Equation 8–1**

$$P_a = P_s\left(\frac{d_a}{d_s}\right)$$

Where:

$P_a$ = Actual pressure drop, in. wg (Pa)

$P_s$ = Pressure drop from tables, in. wg (Pa)

$D_a$ = Actual air density, lb/ft$^3$ (kg/m$^3$)

$D_s$ = Standard air density,

0.075 lb/ft$^3$ (1.204 kg/m$^3$)

DRAFT

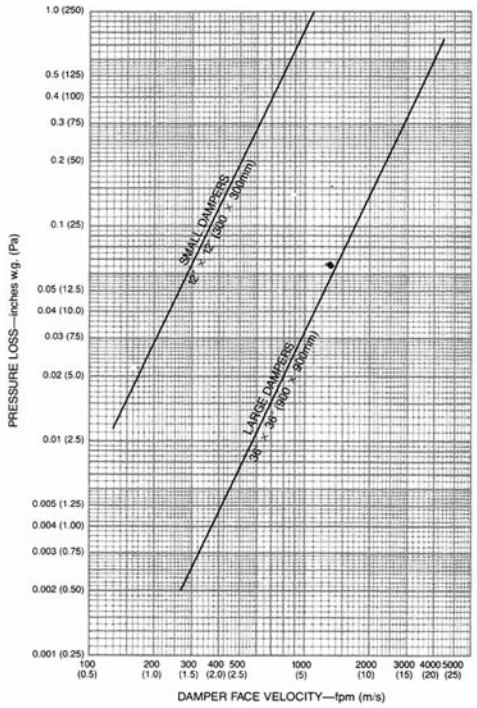


**FIGURE 8–2 VOLUME DAMPERS (*BASED UPON AMCA CERTIFIED VOLUME DAMPERS)**

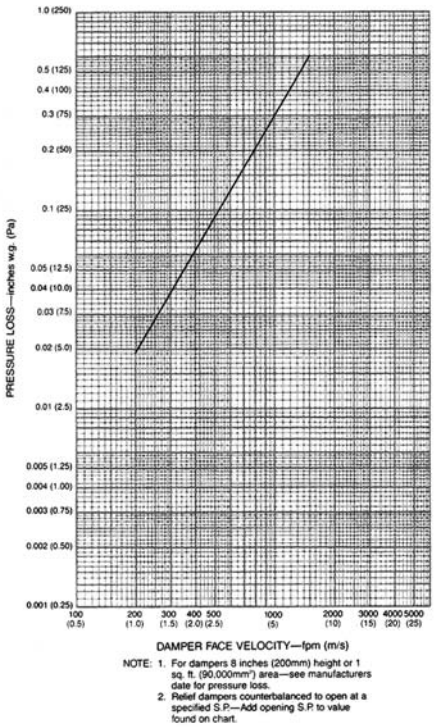


**FIGURE 8–3 BACKDRAFT OR RELIEF DAMPERS**

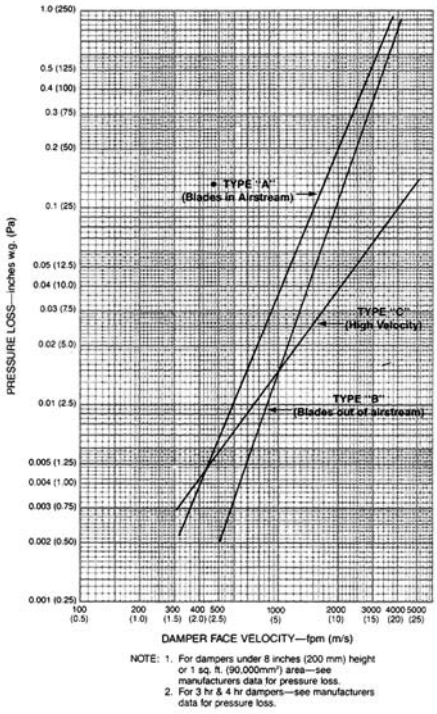


**FIGURE 8–4 2–HOUR FIRE & SMOKE DAMPERS (BASED ON AMCA CERTIFIED FIRE DAMPERS)**

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## 8.4 DUCT SYSTEM APPARATUS CHARTS

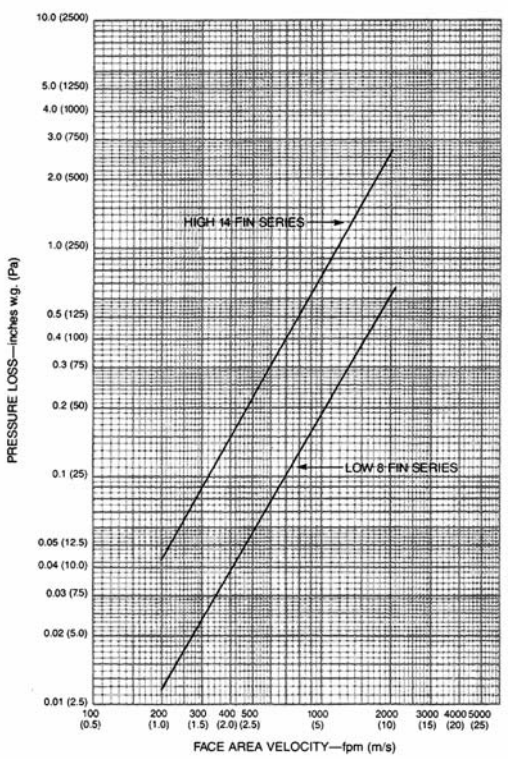


**FIGURE 8–5 HEATING COILS WITH 1 ROW**

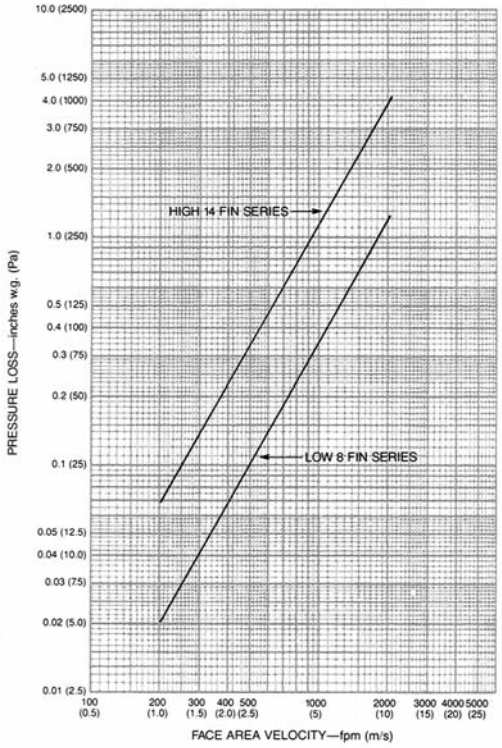


**FIGURE 8–6 HEATING COILS WITH 2 ROWS**

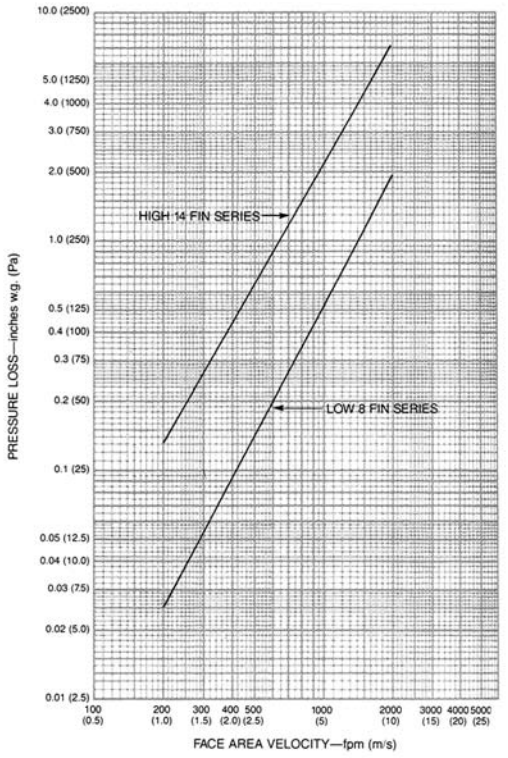


**FIGURE 8–7 HEATING COILS WITH 3 ROWS**

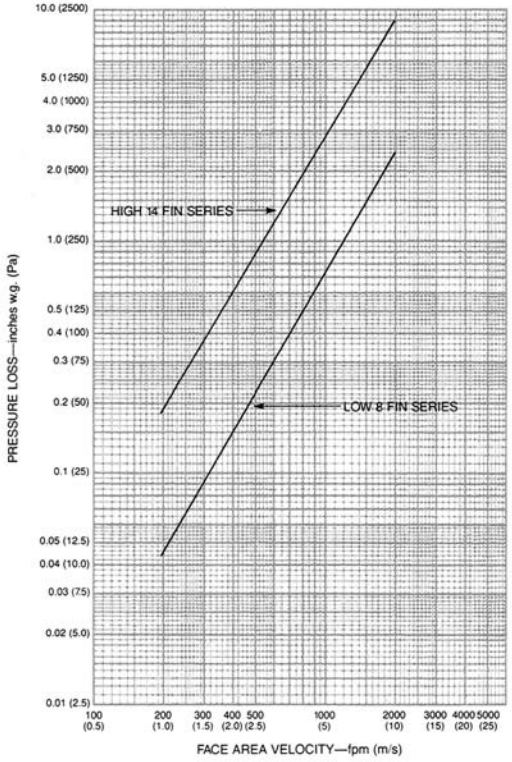


**FIGURE 8–8 HEATING COILS WITH 4 ROWS**

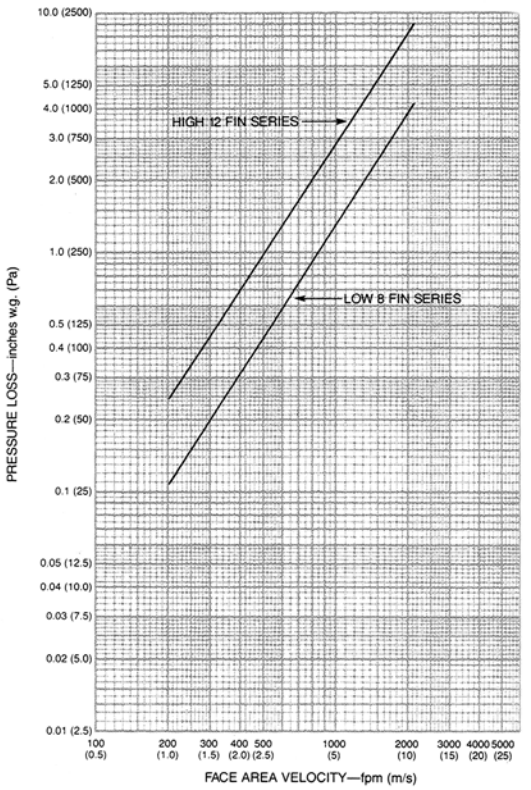


**FIGURE 8–9 COOLING COILS (WET) 4 ROW**

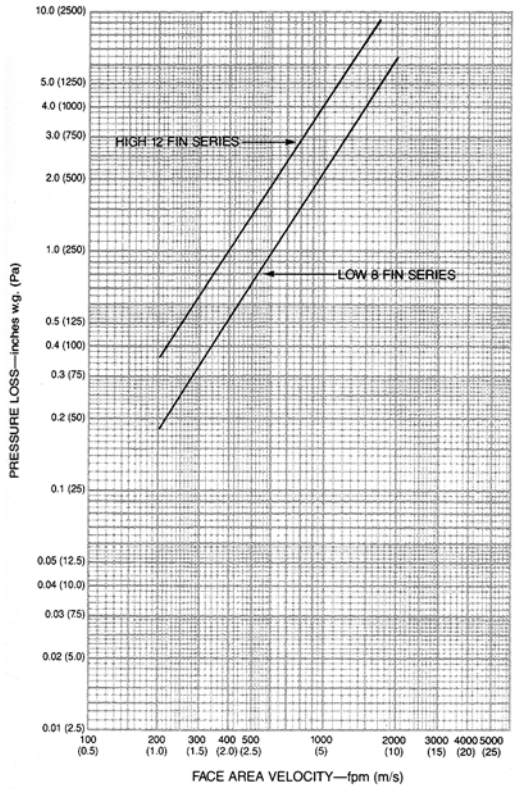


**FIGURE 8–10 COOLING COILS (WET) 6 ROW**

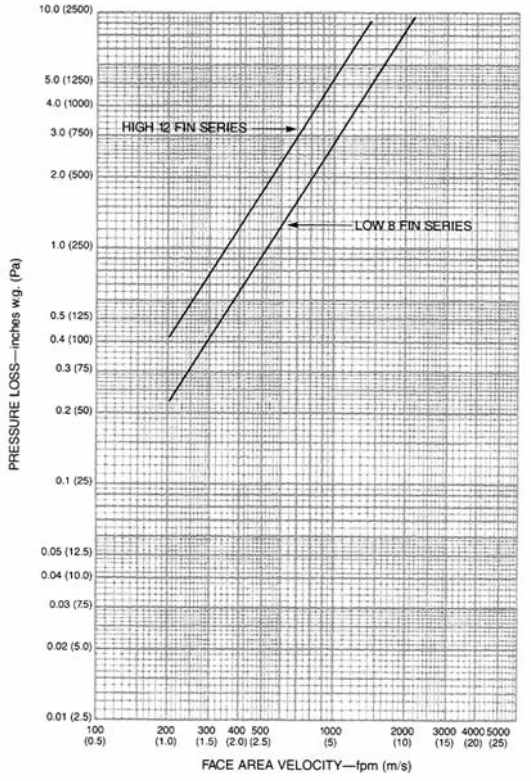


**FIGURE 8–11 COOLING COILS (WET) 8 ROW**

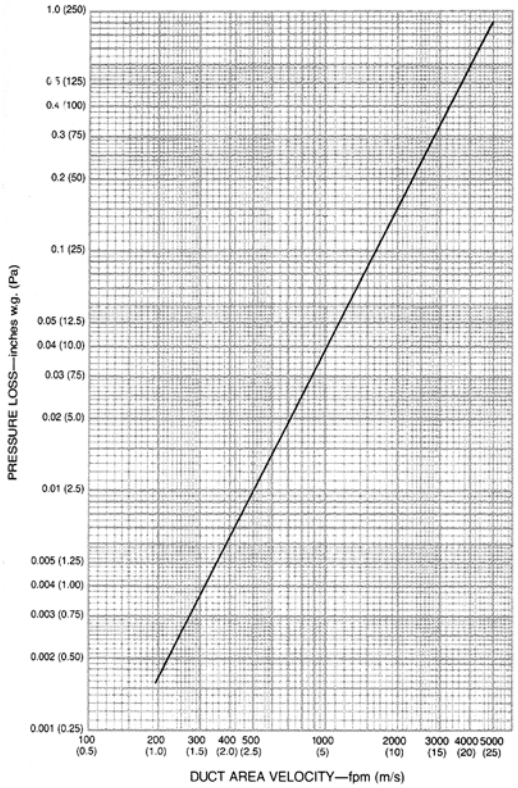


**FIGURE 8–12 AIR MONITOR DEVICE**

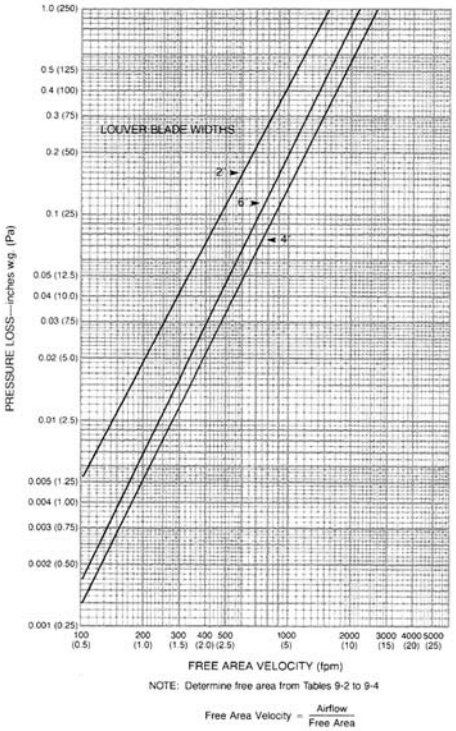


**FIGURE 8–13 LOUVERS WITH 45° BLADE ANGLE (*BASED ON AMCA CERTIFIED LOUVERS)**

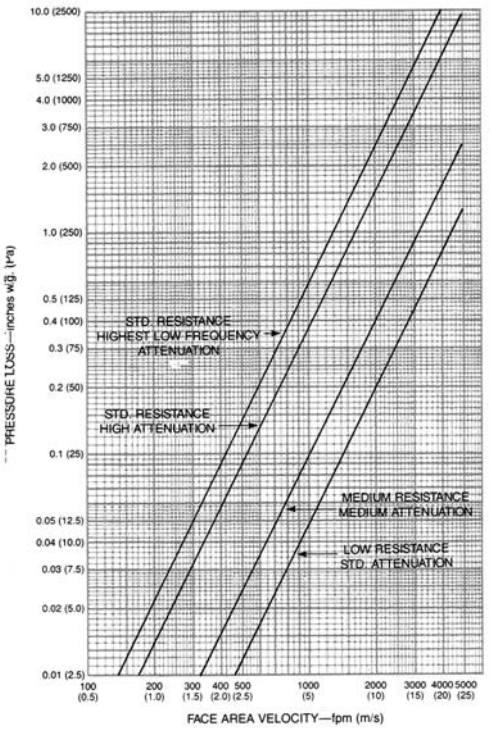


**FIGURE 8–14 3 RECTANGULAR SOUND TRAPS – 3 FOOT (1M)**

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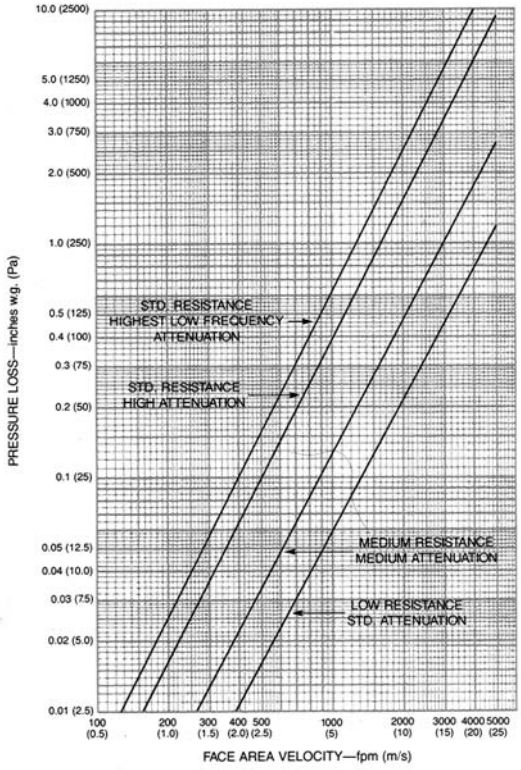



**FIGURE 8–15 RECTANGULAR SOUND TRAPS – 5 FOOT (1.5M)**

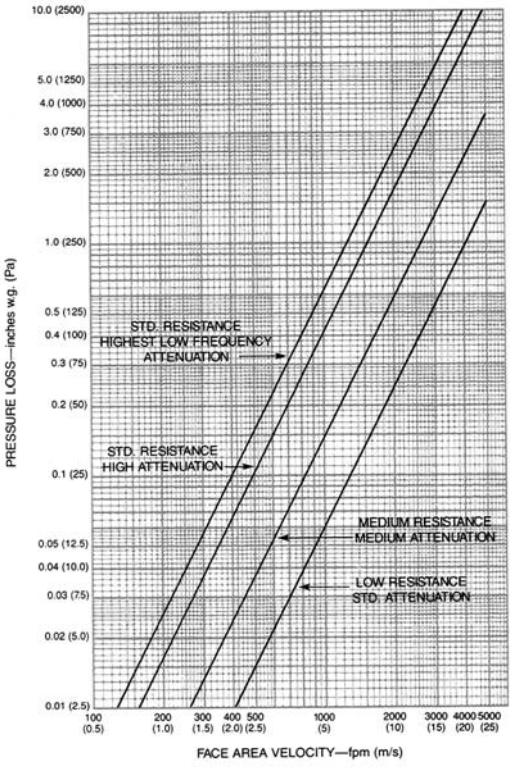



**FIGURE 8–16 RECTANGULAR SOUND TRAPS – 7 FOOT (2M)**

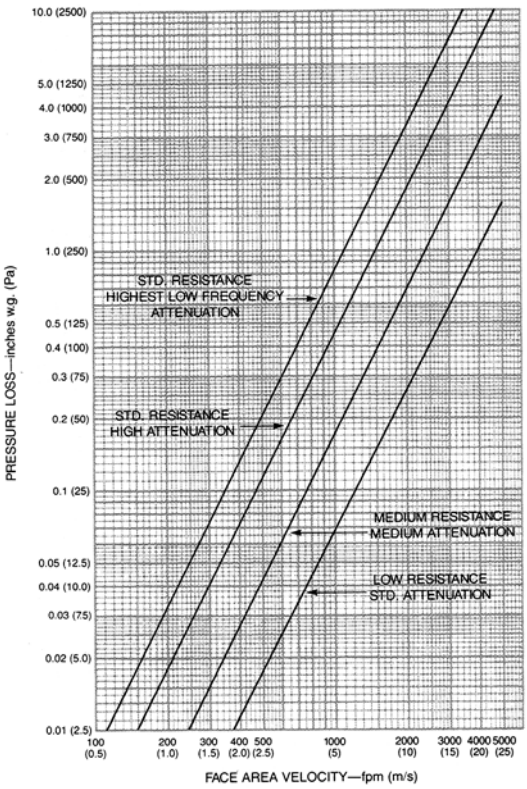



**FIGURE 8–17 RECTANGULAR SOUND TRAPS – 10 FOOT (3M)**

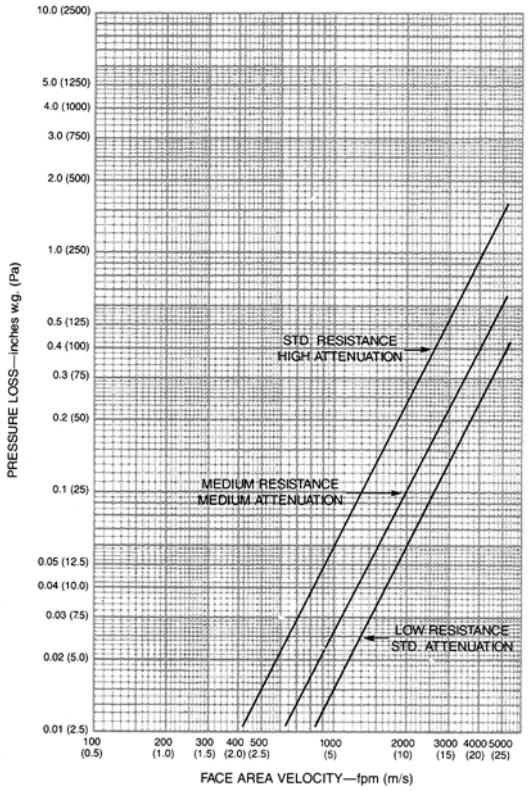



**FIGURE 8–18 ROUND SOUND TRAPS**

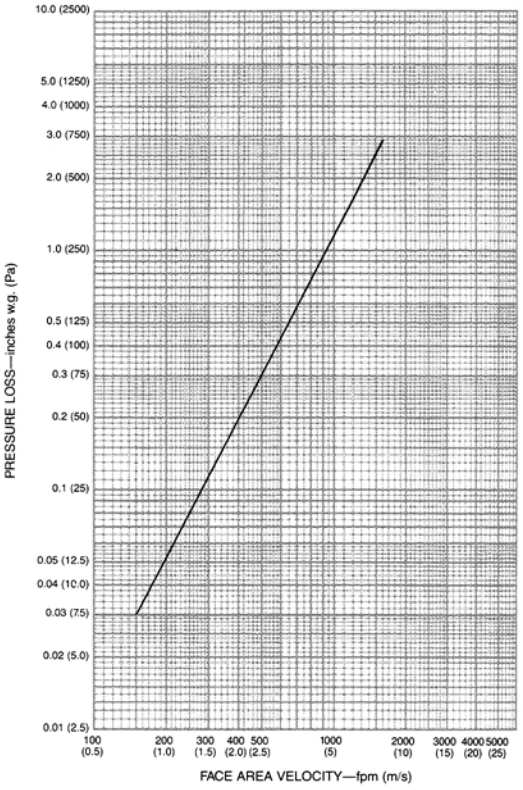



**FIGURE 8–19 ELIMINATORS THREE BEND**

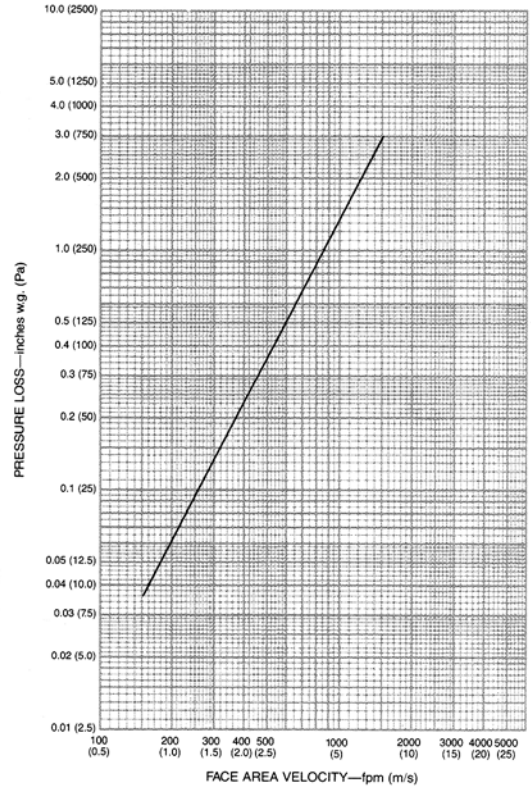
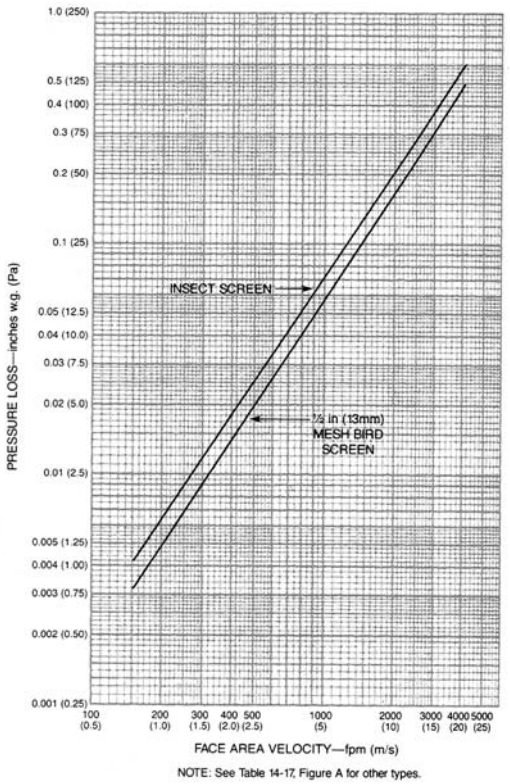



**FIGURE 8–20 AIR WASHER**




NOTE: See Table 14-17, Figure A for other types.

**FIGURE 8–21 SCREENS**

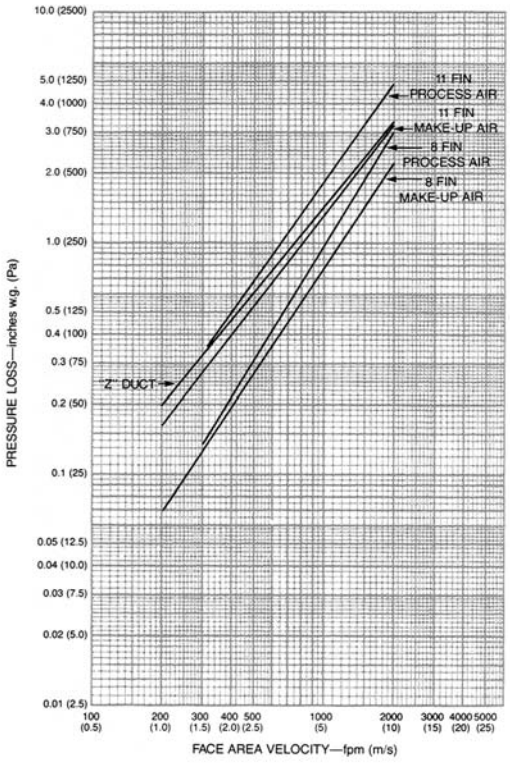



**FIGURE 8–22 AIR–TO–AIR PLATE EXCHANGERS (MODULAR)**

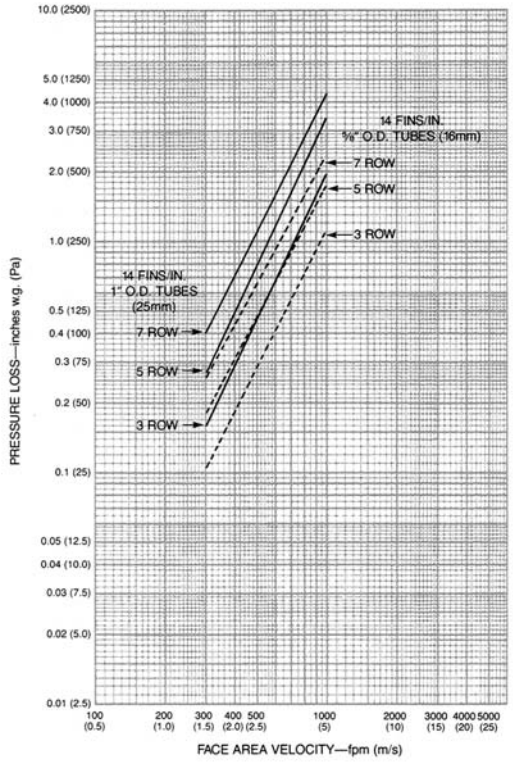



**FIGURE 8–23 AIR–TO–AIR SINGLE TUBE EXCHANGERS**

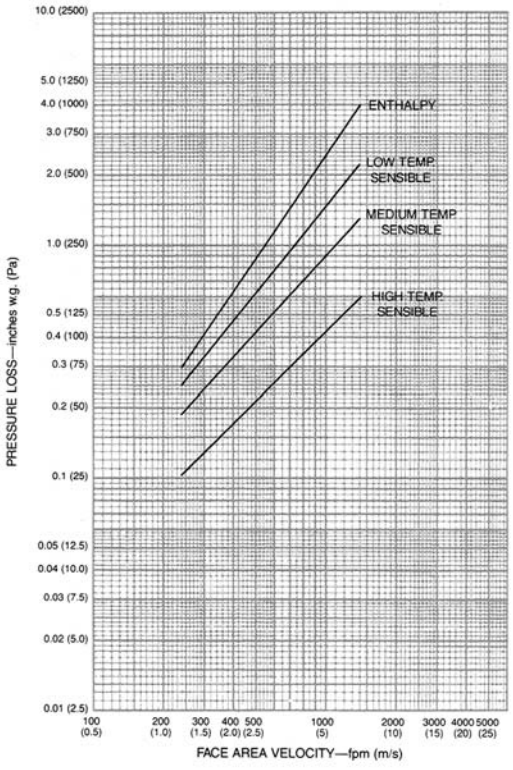



**FIGURE 8–24 ROTARY WHEEL EXCHANGER**

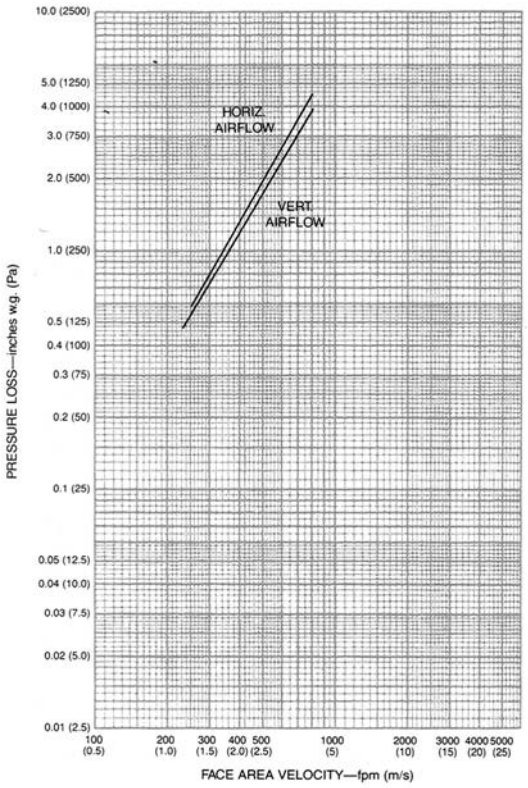



**FIGURE 8–25 MULTIPLE TOWER ENERGY EXCHANGERS**

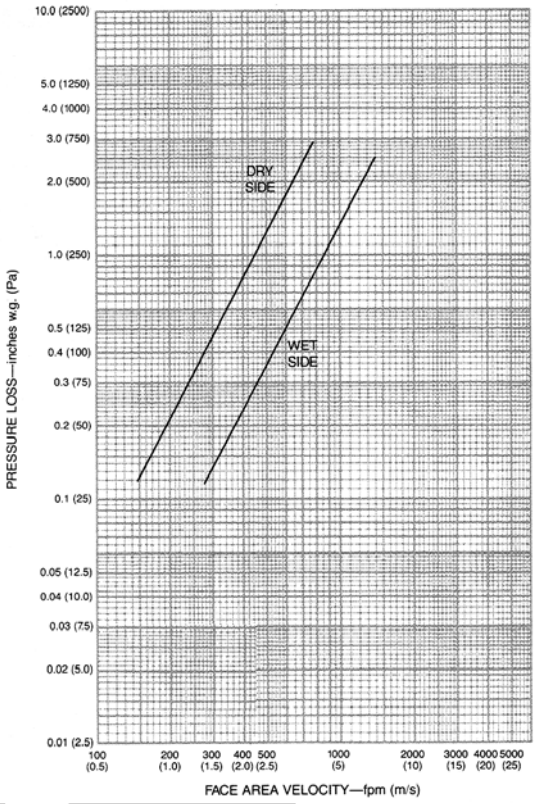



**FIGURE 8–26 DRY AIR EVAPORATIVE COOLER**

DRAFT

| Neck Velocity fpm m/s | 400 2.0 | 500 2.5 | 600 3.0 | 700 3.5 | 800 4.0 | 1000 5.0 |
|---|---|---|---|---|---|---|
| Round Diffuser | .024 | .039 | .056 | .075 | .096 | .152 |
| Half Round Diffuser | .035 | .054 | .080 | .107 | .141 | .219 |
| Half Round Diffuser, Flush | .046 | .074 | .106 | .143 | .184 | .290 |
| Square Diffuser | .021 | .033 | .048 | .064 | .083 | .130 |
| Square Diffuser, Adjustable | .036 | .057 | .080 | .112 | .144 | .226 |
| Rectangular Diffuser | .043 | .066 | .096 | .131 | .170 | – |
| Curved Blade Diffuser | .056 | .090 | .131 | .175 | .225 | .355 |
| Perforated Diffuser | .037 | .058 | .083 | – | .148 | .230 |
| High Capacity Diffuser | – | – | – | .050 | .060 | .100 |
| Slimline Diffuser, 2 Way* | .010 | .015 | .022 | .028 | .040 | .063 |
| Extruded Fineline Diffuser ½" (6 mm) Bar Spacing* | .011 | .015 | .024 | .030 | .044 | .069 |
| Linear Slot Diffuser | .051 | .079 | .110 | .150 | .200 | – |

**Table 8–5 Air Outlets & Diffusers – Total Pressure Loss Average**

*Velocity Thru Face Open Area

| Velocity fpm m/s | 300 1.5 | 400 2.0 | 500 2.5 | 600 3.00 | 800 4.0 | 1000 5.0 |
|---|---|---|---|---|---|---|
| 0 Degree Deflection | .010 | .017 | .028 | .038 | .069 | .107 |
| 22½ Degree Deflection | .011 | .019 | .031 | .043 | .078 | .120 |
| 45 Degree Deflection | .016 | .029 | .047 | .064 | .117 | .181 |

**Table 8–6 Supply Registers – Total Pressure Loss Average**

| Velocity fpm m/s | 300 1.5 | 400 2.0 | 500 2.5 | 600 3.00 | 800 4.0 | 900 4.5 |
|---|---|---|---|---|---|---|
| 12" Square Diffuser (300 mm) | .033 | .060 | .092 | .134 | .238 | .302 |
| 24" Square Diffuser (600 mm) | .068 | .122 | .187 | .272 | .483 | .614 |
| 21" × 12" Rectangular Diffuser (300 × 525 mm) | .055 | .098 | .152 | .222 | .390 | .496 |
| Perforated Return Diffuser (Neck Velocity) | .025 | .060 | .080 | .100 | .180 | .230 |
| Register, 0 Degree Deflection | .012 | .020 | .032 | .046 | .080 | .102 |
| Register, 30 Degree Deflection | .033 | .055 | .088 | .126 | .220 | .275 |
| Register, 45 Degree Deflection | .054 | .090 | .144 | .207 | .360 | – |
| Register, Perforated Face | .042 | .070 | .122 | .161 | .280 | .350 |

**Table 8–7 Return Registers – Total Pressure Loss Average**

DRAFT

| Duct Element | Face Velocity | |
|---|---|---|
| | fpm | m/s |
| Air Washers | | |
| A. Spray-Type | 300-700 | 1.5-3.5 |
| B. Cell-Type | Refer to Mfg. Data | |
| C. High Velocity Spray-Type | Refer to Mfg. Data | |
| Cooling Or Dehumidifying Coils | | |
| A. Without Eliminators | 500-600 | 2.5-3.0 |
| B. With Eliminators | 600-800 | 3.0-4.0 |
| Heating Coils | | |
| A. Steam and Hot Water | 500-600 | 2.5-3.0 |
| | Most Common | |
| | 200 min. | 1.0 min. |
| | 1500 max. | 7.5 max. |
| B. Electric: | | |
| 1. Open Wire | Refer to Mfg. Data | |
| 2. Finned Tubular | Refer to Mfg. Data | |
| Filters | | |
| A. Fibrous Media Unit Filters: | | |
| 1. Viscous Impingement | 250-700 | 1.3-3.5 |
| 2. Dry Type | Upto 750 | Upto 3.8 |
| 3. HEPA | 250 | 1.3 |
| B. Renewable Media Filters: | | |
| 1. Moving Curtain Viscous Impingement | 500 | 2.5 |
| 2. Moving Curtain Dry-Media | 200 | 1.0 |
| C. Electronic Air Cleaners: | | |
| 1. Ionizing Plate-Type | 300-500 | 1.5-2.5 |
| Louvers | | |
| A. Intake: | | |
| 1. 7000 cfm and greater | 400 | 2.0 |
| 2. Less than 7000 cfm | *See* Fig. 8-27 | |
| B. Exhaust | | |
| 1. 5000 cfm and greater | 500 | 2.5 |
| 2. Less than 5000 cfm | *See* Fig. 8-27 | |
| Sound Traps | | |
| A. Rectangular: | | |
| 1. High attenuation | 400-700 | 2.0-3.5 |
| 2. Medium attenuation | 700-1750 | 3.5-8.8 |
| 3. Low attenuation | 1750-2500 | 8.8-12.5 |
| B. Round | | |
| 1. High attenuation | 1000-2000 | 5.0-10.0 |
| 2. Medium attenuation | 1500-3000 | 7.5-15.0 |
| 3. Low attenuation | 2500-4000 | 12.5-20.0 |

**Table 8–8 Typical Design Velocities**

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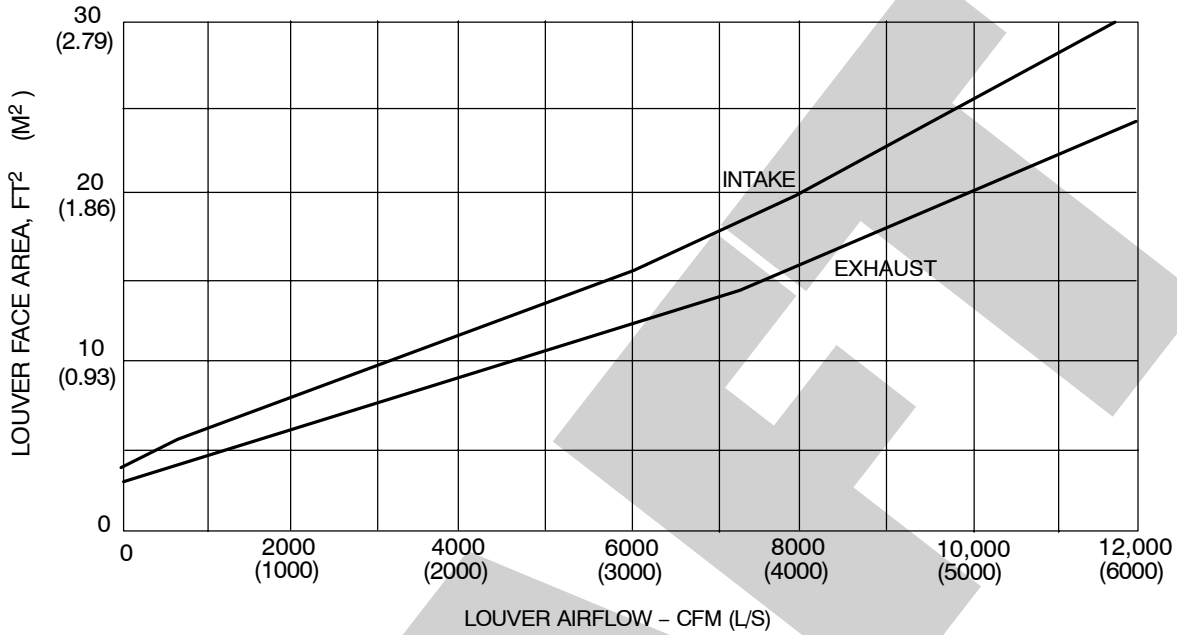


| Parameters Used Above | Intake Louver | Exhaust Louver |
|---|---|---|
| Minimum Free Area - 48 in. Square Test Section | 45% | 45% |
| Water Penetration, Oz ($ft^2$ –15 min.) | Negligible (Less than 0.2) | Not Applicable |
| Maximum Static Pressure Drop, in. wg (Pa) | 0.15 (38) | 0.25 (63) |

**FIGURE 8–27 RECOMMENDED CRITERIA FOR LOUVER SIZING**

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CHAPTER 9

# PROVISIONS FOR TESTING, ADJUSTING, AND BALANCING

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## 9.1 SCOPE

The need for accurate balancing of air and water systems is essential in today's construction industry. With continuing emphasis on energy conservation, the HVAC engineer is being called upon to design more efficient systems.

Systems must now be designed to vary ventilation air, supply air, and fluid-flow rates based on changing room occupancy and time of day. The proper distribution of air and water to meet the design loads is mandatory, yet excess must be avoided to maintain energy efficiency. Comfort is attained by balancing the systems to these design criteria.

Today's variable-volume systems, which, adjust flows to satisfy changing space loads, are more difficult to balance than a constant-flow system, since both high- and low-flow-limit balancing must be provided.

While systems will vary considerably in design, size, or extent of duct distribution, the same general procedures for testing, adjusting, and balancing will be employed on most projects.

## 9.2 TAB DESIGN CONSIDERATIONS

The system air must flow to the occupied space with the least practical losses from leakage and resistance. It must provide proper mixing of tempered air, with a minimal temperature change from heat gain or loss. After reaching the space, the air must be distributed in the most efficient, draftless, noiseless manner available for each site's requirements.

The means to accomplish these requirements are with the ductwork and outlets. Because of aesthetics and available space within the building, the selection of the type and size of ducts and outlets is often difficult. Compromises must sometimes be worked out which make the design process and installation require ingenuity.

The designer should give special consideration to the balancing and adjusting requirements during the initial design. The TAB technician must be able to test and analyze the particular installation, to properly balance the system with the least effort, yet obtain the greatest system efficiency and comfort level. The balancing capability needs to be initially designed into the system. The following are TAB-related considerations to apply when designing duct systems.

a. Ductwork to and from air conditioning equipment should be designed so any stratified air will mix properly before entering branch ducts or HVAC equipment.

b. Splitter-type dampers offer little or no control of air volume in ducts. They should only be regarded as air diverters. The damper/diverters are effective when used with duct systems having a low resistance to airflow.

c. Manually-operated, opposed- or single-blade, quadrant-type volume dampers should be installed in each branch duct takeoff after it leaves the main duct to control the amount of air entering or leaving the branch, *see* Figure 9-1.

d. Turning vanes should be installed oriented so that the air leaving the vanes is parallel to the downstream duct walls. Turning vanes should be utilized in all rectangular elbows. This includes return as well as supply and exhaust systems, *see* Figures 5-14.

e. Manual volume dampers should be provided in branch duct takeoffs, to control the total air to the face dampers of the registers and diffusers. The use of extractors is not recommended. Extractors cause turbulence in the main trunk duct, thereby increasing the total system pressure loss and affecting the performance of other branch outlets downstream.

f. Extractors should not be used at branch or main duct takeoffs to provide volume control. Branch duct tap-in fittings with a 45 degree entry throat provide efficient airflow from main into branch ducts.

g. Single-blade quadrant volume dampers immediately behind diffusers and grilles may tend to throw air to one side of the outlet, preventing uniform airflow across the outlet face or cones.

h. Register or diffuser dampers cannot be used to reduce high air volumes without inducing objectionable air-noise levels, *see* Figure 9-2. A slight opening of an opposed-blade volume damper will generate a relatively high noise level as the air passes through the damper opening under system pressure.

i. Dampers should be located at least two diameters from a fitting and as far as possible from an outlet to minimize duct noise.

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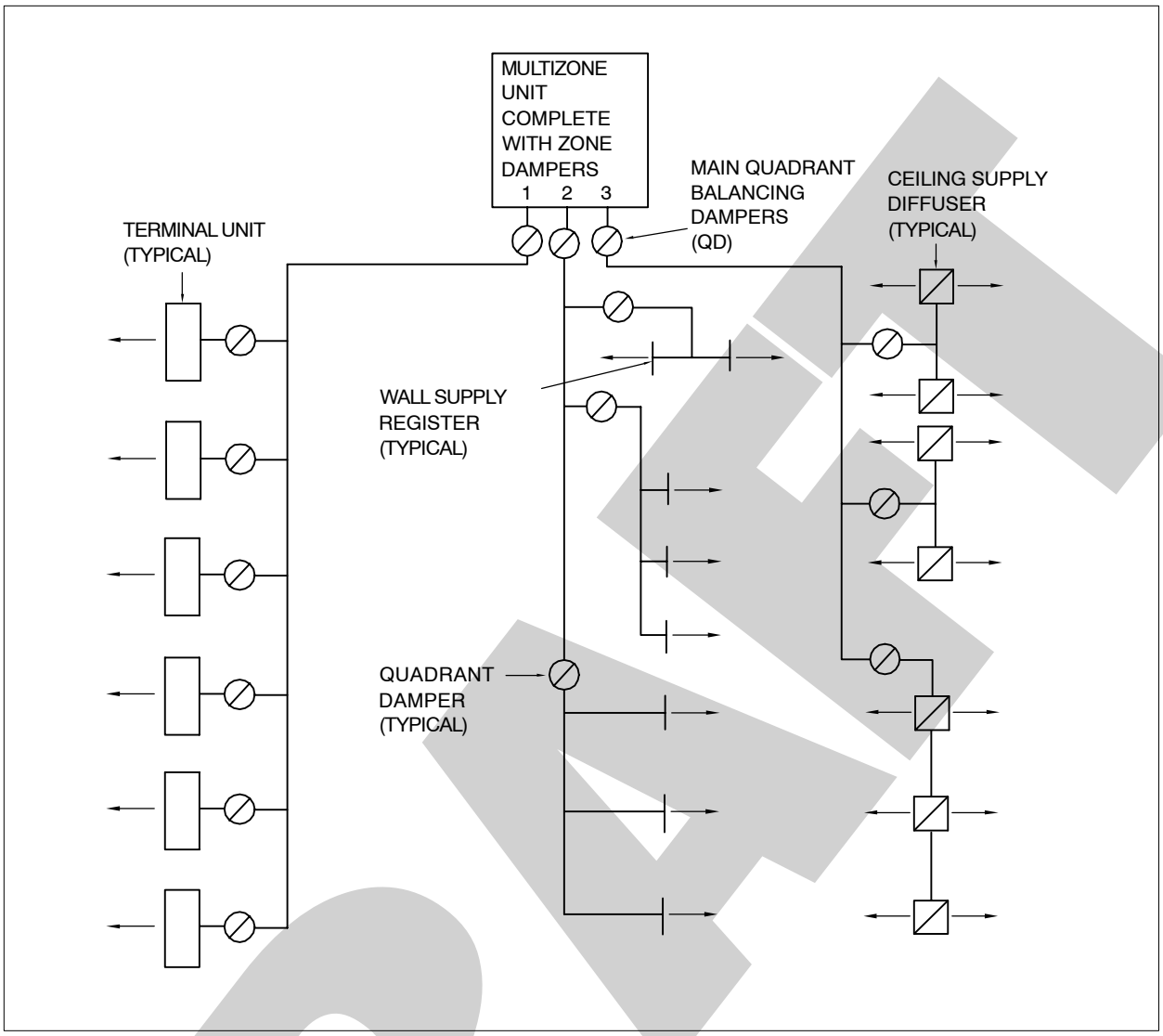


**FIGURE 9–1 DESIGN CONSIDERATIONS FOR DIFFUSER LAYOUTS AND BALANCING DAMPER LOCATIONS**

j. All portions of the main return-air duct system require manual balancing dampers at each branch duct inlet.

k. Avoid placing a return-air opening directly in, or adjacent to, the return-air plenum. Lining the duct behind the opening is unlikely to reduce the transmitted noise to acceptable levels.

l. Terminal boxes or volume control assemblies should be located so the discharge ductwork will minimize air turbulence and stratification, *see* Figure 9-3.

m. Provide the necessary space around components of the duct system, to allow the TAB technician to take proper readings. Allow straight duct sections of six to ten duct diameters from fan outlets, elbows, or open duct ends, for accurate traverse readings, *see* Figure 6-2, for velocity profiles at fan discharges.

n. Adequately-sized access doors should be installed within a practical working distance of all in-duct devices. This should include volume dampers, fire dampers, pressure reducing valves, reheat coils, volume control assemblies (boxes), blenders, and constant volume regulators. The designer must coordinate locations with the architectural and

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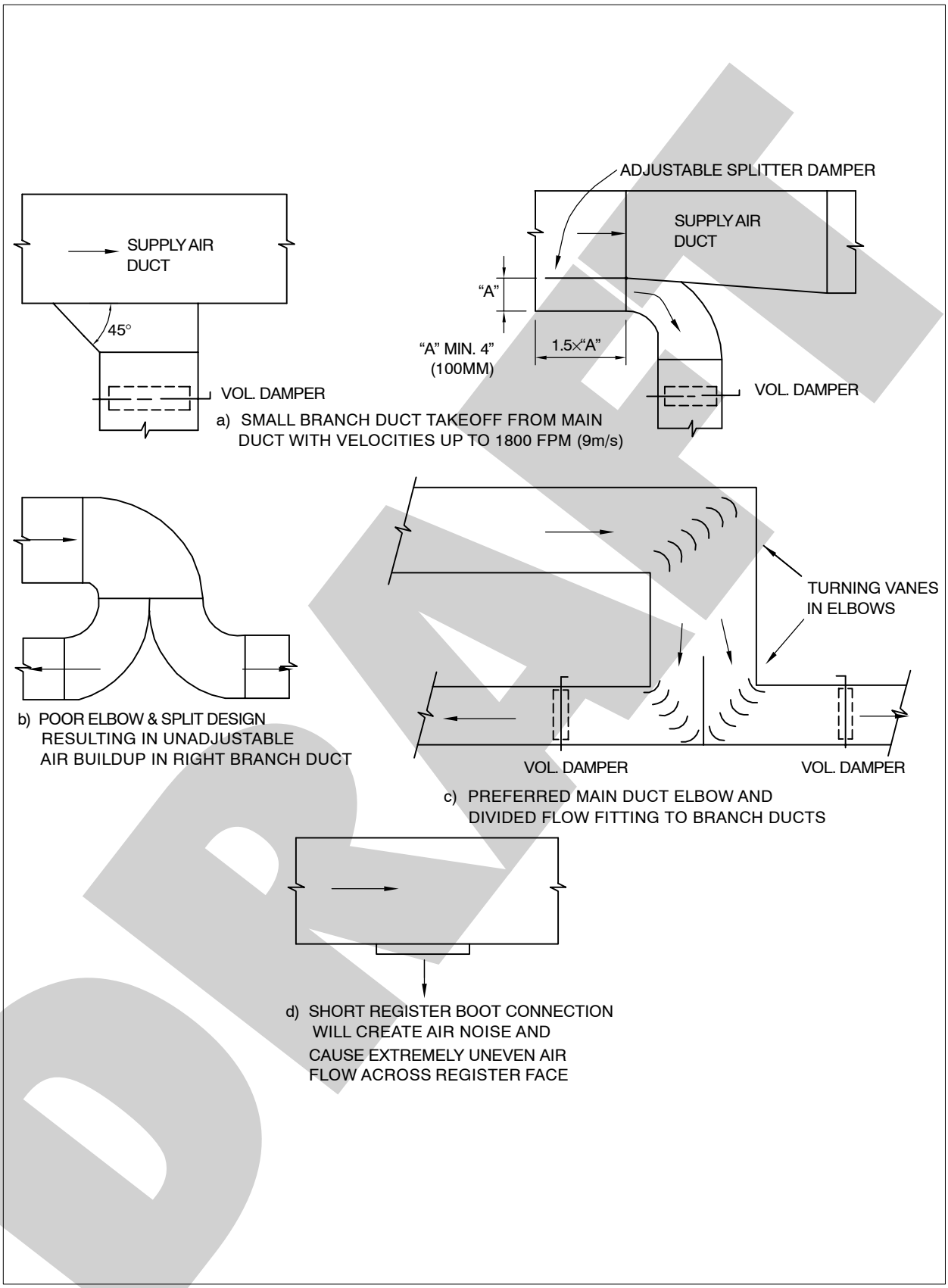


**FIGURE 9–2 DUCT DESIGN CONSIDERATIONS FOR SUGGESTED BALANCING DAMPER LOCATIONS**

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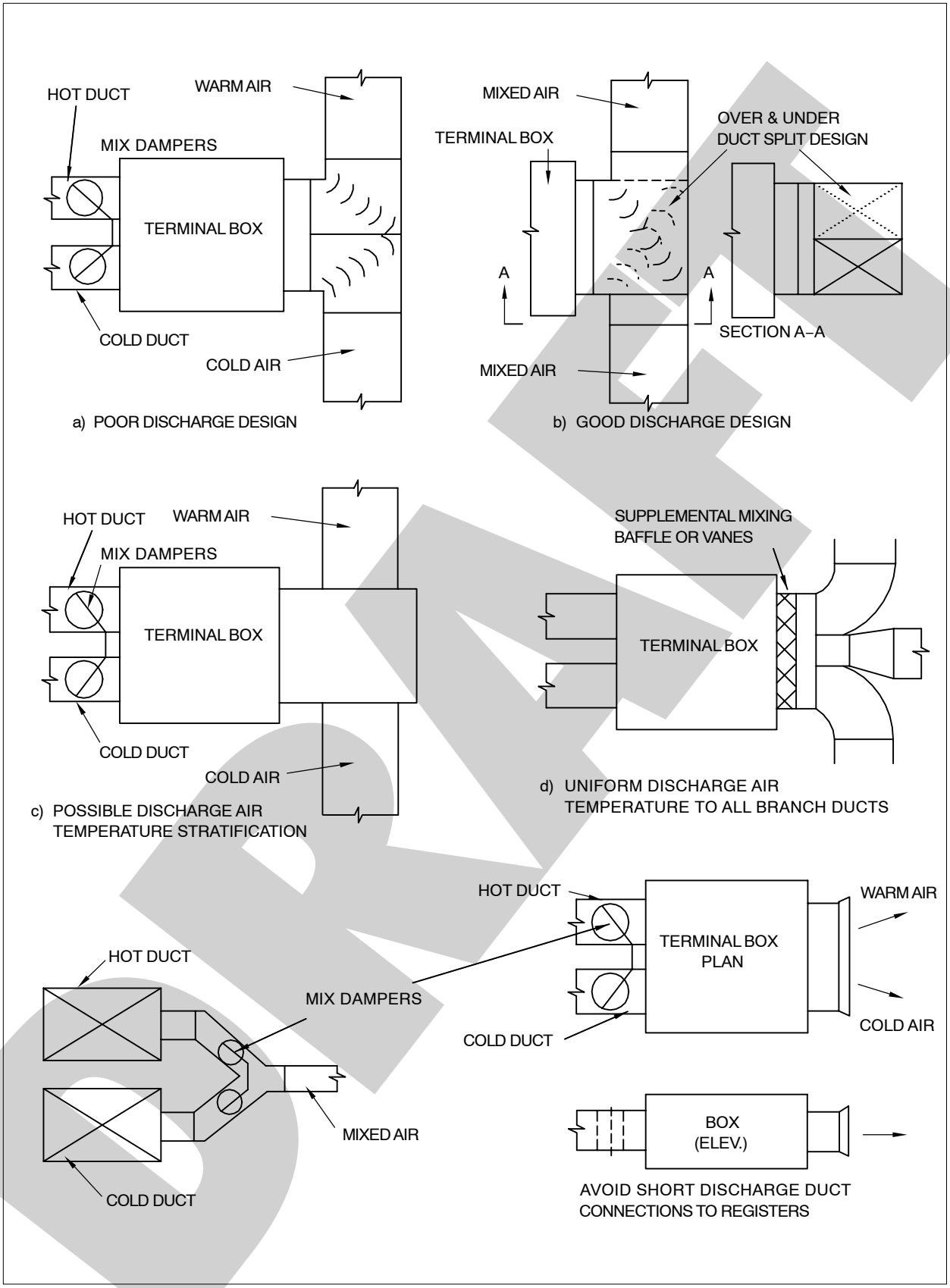


**FIGURE 9–3 DESIGN CONSIDERATIONS TO MINIMIZE AIRFLOW TURBULENCE AND STRATIFICATION FROM TERMINAL BOXES**

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construction plans and indicate both the access doors and all in-duct devices that are expected to be installed or serviced by the HVAC contractor or TAB technician.

o. Provide for test wells, plugged openings, and other testing points normally used in TAB procedures.

## 9.3 AIR MEASUREMENT DEVICES

Before 1960, there was no established procedure and few attempts were made to measure airflow in HVAC systems. Total volumetric airflow measurements were attempted by making traverses with anemometers at the central-station equipment. Airflow volumes at air terminals were determined by using instruments that measured jet velocities of the discharge or intake-air pattern, and then applying laboratory developed empirical area factors published by the terminal manufacturer.

Now, a test procedure involving Pitot tube traverses is the primary means of determining volumetric flows through air distribution systems. Consequently, HVAC systems can now be balanced and adjusted so that they closely approach design performance. Today, we have factory-fabricated volumetric airflow measuring and control devices that may be used in areas requiring critical air control. A complete list of these instruments, their accuracy, and use, may be found in the SMACNA *HVAC Systems – Testing, Adjusting & Balancing* manual.

## 9.4 BALANCING WITH ORIFICES

The use of sharp-edged orifice plates to balance airflow to outlets or branches, provides a high level of accuracy, but does not provide the flexibility inherent in dampers. Where the flow can be determined in advance, procedures can be used to accurately determine the airflow and the total pressure loss. For duct design purposes, Table A-14B may be used.

The sharp-edged orifice has more resistance to flow, but is easily constructed. It can also be made readily interchangeable for several orifice sizes. The orifice can be mounted between two flanged sections, sealed with rubber gaskets. Three orifice sizes, 1400 in., 2625 in., and 4900 in. (36.6 mm, 66.7 mm, and 124.5 mm) diameters, can be used to meter velocities from 50 to 8000 fpm (0.25 to 40 m/s).

If the orifice and pipe taps are made to exact dimensions, the calculated air volume will be within one percent of actual flow for standard air. The necessary equations and charts may be found in the SMACNA *HVAC Systems – Testing, Adjusting & Balancing* manual. Orifices for larger ducts can be sized using data found in the *Fan Engineering* handbook published by the Buffalo Forge Company.

The orifice can be calibrated with a standard Pitot tube. A micro-manometer is needed to read velocities below 600 fpm (3 m/s). At 1000 to 3000 fpm (5 to 15 m/s), with a 10:1 inclined manometer, an accuracy of ± 0.3 to 1.0 percent can be expected; at 3000 to 4000 fpm (15 to 20 m/s), an accuracy of ± 0.25 to 0.3 percent can be expected. If the orifice is made to precise dimensions, no calibration is needed, and the tabulated calculation can be used.

## 9.5 PROVISIONS FOR TAB IN SYSTEM DESIGN

### 9.5.1 General Procedures

Chapter 4 discussed how a schematic diagram of each duct system should be prepared. This is used as a guide to test and balance the systems after the installation work has been completed. It is also helpful to the system designer to develop these schematic diagrams when designing the systems to determine if all devices that will be necessary for testing and balancing have been included.

Where there is more than one system, make a separate diagram for each. All dampers, regulating devices, terminal units, outlets, and inlets should be indicated. Show the sizes, velocities, and airflow for main and branch ducts. Include the sizes and airflow ratings of all terminal outlets and inlets, including outside-air intakes, return and relief-air ducts, louvers etc. For rapid identification and reporting purposes, number all outlets. Add general notes indicating thermostat locations, room thermostat number, mounting device or box controlled, and other descriptive information.

### 9.5.2 HVAC System Testing, Adjusting & Balancing Manual

The SMACNA *HVAC System Testing, Adjusting and Balancing* manual presents the basic fundamentals, methods, and procedures that a TAB contractor with a reasonable technical background in HVAC systems, should use to adequately balance most HVAC systems.

In addition to the fundamentals and procedures for balancing air systems, this manual includes hydronic-piping system balancing fundamentals and procedures.

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## 9.6 LABORATORY TESTING

### 9.6.1 Fan Tests

Most fan manufacturers rate the performance of their products from tests made in accordance with the Air Movement and Control Association, Inc. (AMCA) Standard 210, *Laboratory Methods of Testing Fans for Rating*. The purpose of Standard 210 is to establish uniform methods for laboratory testing of fans and other air-moving devices, to determine performance in terms of flow rate, pressure, power, air density, speed of rotation, and efficiency, for rating or guarantee purposes.

Two basic methods of measuring airflow are included – the Pitot tube, and the long radius flow nozzle. These are incorporated into a number of different setups or figures. In general, a fan is tested on the setup which most closely simulates the way in which it will be installed in an HVAC system.

Centrifugal, tubeaxial, and vaneaxial fans are normally tested with an outlet duct. Propeller fans are normally tested in the wall of a chamber or plenum. Power roof exhausters are tested while mounted on a curb exhausting from the test chamber.

It is very important to realize that each setup in AMCA Standard 210 is a standardized arrangement, which is not intended to reproduce exactly every installation likely to be found in the field. The infinite variety of possible arrangements of actual air systems makes it impossible to duplicate all of these in the fan test laboratory.

Figure 9-4 is a reproduction of a test setup from AMCA Standard 210. Note that this particular setup includes a long straight duct connected to the outlet of the fan. A straightener is located upstream from the Pitot tube traverse to remove swirl and rotational components from the airflow. This ensures that the flow at the plane of measurement is as near to uniform as possible.

The angle of the transition between the test duct and the fan outlet is limited to ensure that uniform flow will be maintained. A steep transition, or abrupt change of cross-section, would cause turbulence and eddies and lead to non-uniform flow. The effect of this type of flow disturbance at the fan outlet was discussed in Chapter 6.

Uniform-flow conditions ensure consistency and reproducibility of test results. In any installation where uniform flow conditions do not exist at the fan inlet and discharge, the System Effect will reduce the fan's performance.

### 9.6.2 Duct and Fitting Tests

The following is a brief description of laboratory testing procedures. ASHRAE Standard – *Laboratory Methods of Determining Flow Resistance of Air Ducts*

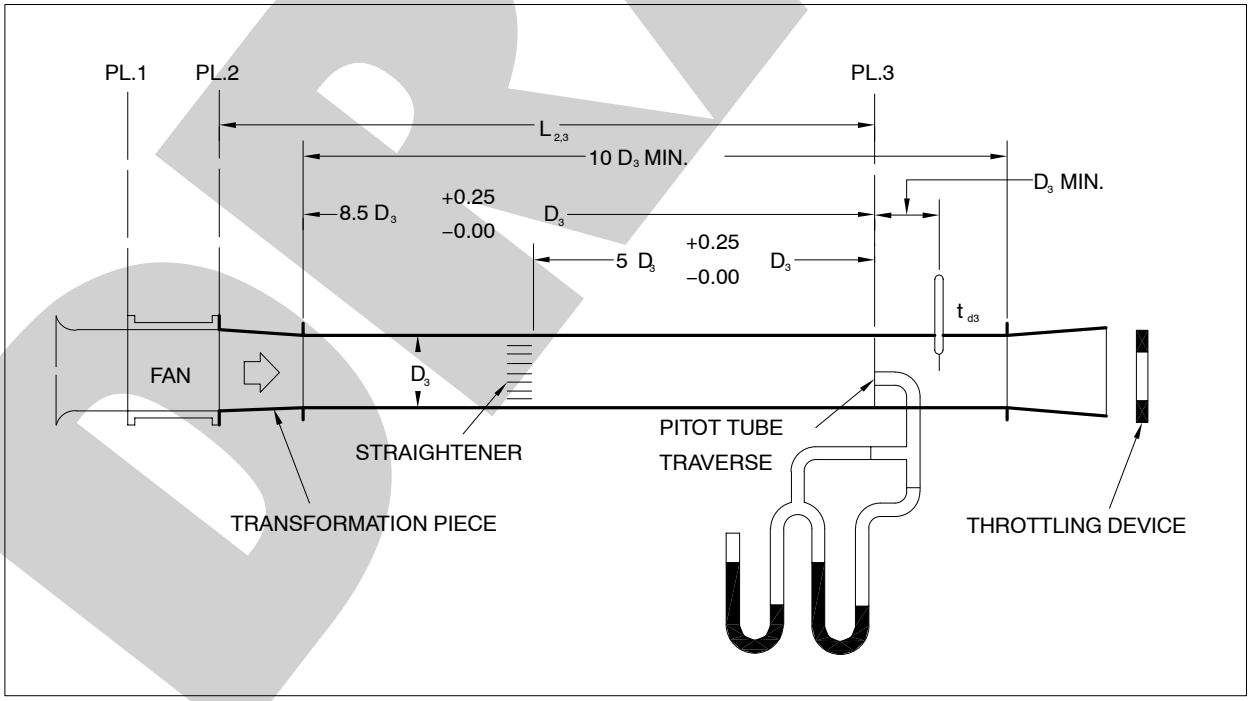


**FIGURE 9–4 FAN RATING TEST**

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*and Fittings* sets the following criteria for testing ductwork and duct fittings:

a. Size: 2 in. (50 mm) or larger

b. Velocities (Main): 1800 to 7000 fpm (9 to 36 m/s)

c. Velocities (Branch): 1200 fpm (9 m/s) minimum

d. Measured pressure differentials (minimum):

- Airflow rate or pressure drop – 0.30 in. wg (75 Pa)
- Main fitting – mean velocity of 1800 fpm (9 m/s) or 0.30 in. wg (75 Pa), whichever is smaller
- Branch fitting – mean velocity of 1200 fpm (6 m/s)

e. Loss coefficient: minimum of 0.20

#### 9.6.2.1 Fan and Airflow Control

A fan to move the air through the test system and a means of varying the airflow rate shall be provided for the test system. Fans used to obtain the desired test flow rate shall be selected to produce sufficient pressure at the desired flow rate. Acceptable flow adjustment means may be dampers, blade or vane pitch control, and speed control. Fans shall be selected so that they will not surge or pulsate during tests.

#### 9.6.2.2 Test Procedures

The test system shall be set up as shown in Figures 9-5 and 9-6. The ductwork and fittings shall be made and reinforced in accordance with the SMACNA *HVAC Duct Construction Standards* for the maximum test pressure. Totally seal the duct, so that leakage is less than 0.5 percent of the minimum test airflow rates. Prescribed methods shall be used for leak testing, *see* SMACNA *HVAC Air Duct Leakage Test Manual.*

Establish the minimum test volumetric airflow rate and leak test all instrument pressure measuring lines. Adjust the flow control device to the maximum velocity rate needed and establish equilibrium conditions.

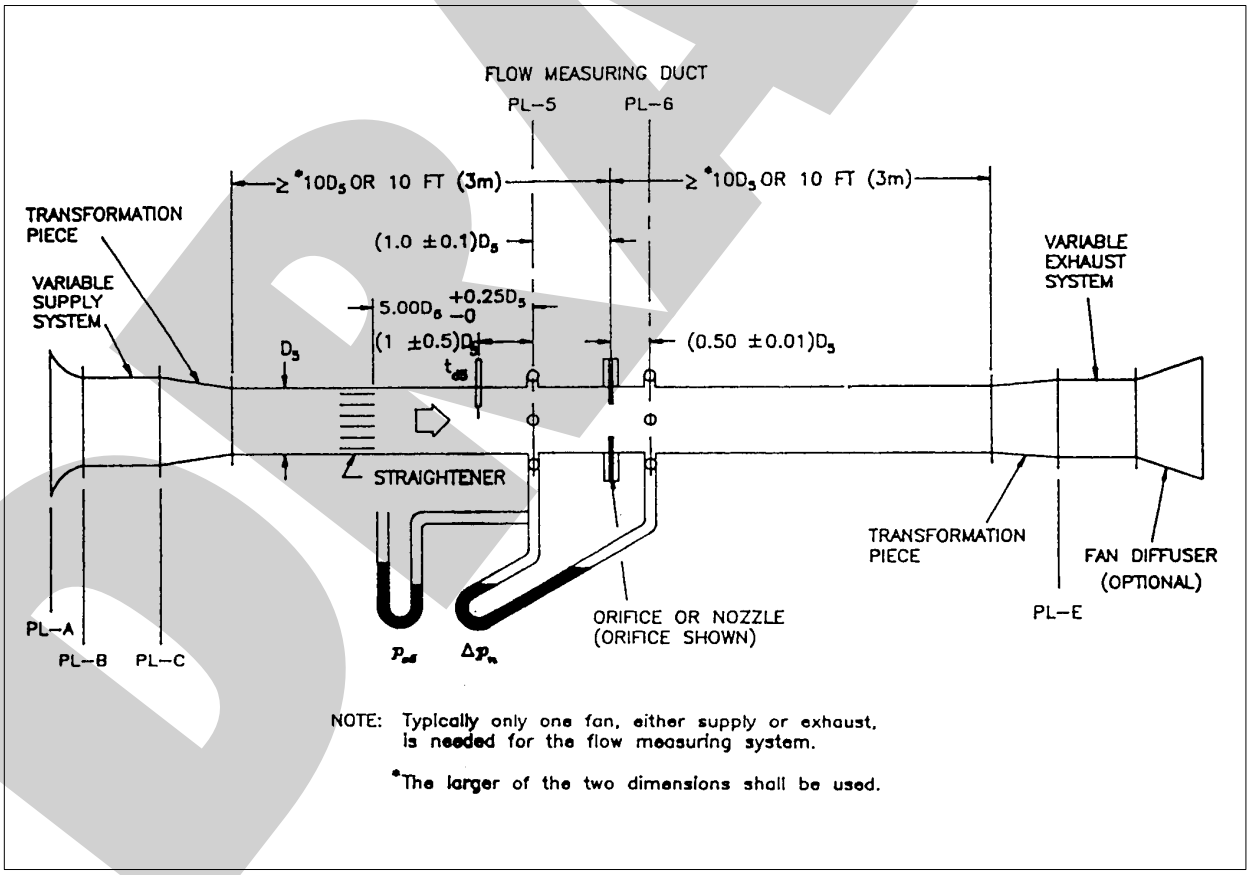


**FIGURE 9–5 LABORATORY DUCT FLOW MEASURING SYSTEM**

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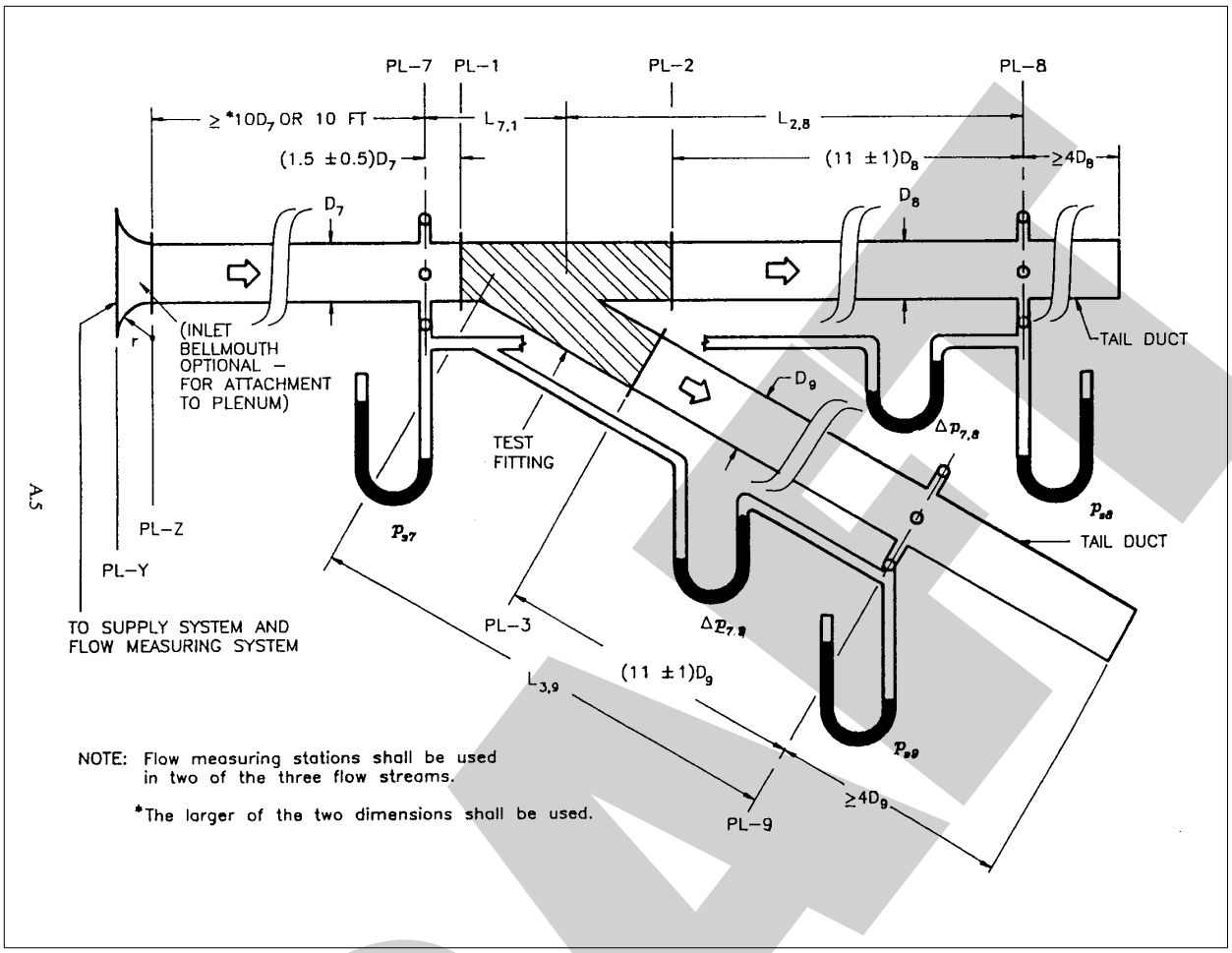


**FIGURE 9–6 LABORATORY DUCT FITTING TEST SETUP**

At the start of the tests record the time, ambient dry bulb, wet bulb, barometric pressure, and leaving air-dry bulb temperature from the test duct exit. Record pressure and flow rate measurements required to establish the test duct or duct fitting performance at each measured flow rate.

#### 9.6.2.3 General Test Requirements

The number of required test measurements that establishes the airflow loss of a duct or fitting over a test velocity range depends on the loss characteristics of the test device used. At least eight test measurements, evenly spaced over the desired range of test velocities shall be used for single-flow fittings, such as an elbow. At least five airflow rates in the main duct and at least eight velocity ratios for each airflow rate shall be used for branch-type fittings, either converging or diverging. If a low loss coefficient is expected, it is recommended that pressured-drop measurements be made first at the highest test velocity. If the fitting loss coefficient for this point is less than 0.02, then loss measurements for that fitting are not needed and the loss coefficient shall be reported only as <0.02 for all test velocities.

The complete test data procedures may be obtained from ASHRAE.

### 9.7 FIELD TESTING AND BALANCING

The testing, adjusting, and balancing (TAB) of environmental or HVAC system in a new building is needed to complete the installation of the system and to make the system perform as the designer intended.

Assuming the design of the system meets the comfort needs of the building occupants, good testing, adjusting, and balancing of each HVAC system allows it to meet space conditioning requirements as efficiently as possible.

It is rare to find an HVAC system of any size which will perform satisfactorily without the benefit of TAB work. It is important for the designer to require all TAB

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work to be done by a qualified firm. The TAB work can be part of the HVAC system installation contract, if the installing contractor has qualified TAB personnel or a related qualified TAB firm. Past performance and being a certified firm in an international TAB organization are the best indicators of qualifications.

### 9.7.1 Design Considerations

Testing, adjusting, and balancing starts out as design functions with most of the devices required for adjustments being integral parts of the design and initial installation. To ensure that proper balance can be achieved, the HVAC system designer needs to locate and specify a sufficient number of dampers, valves, flow measuring locations, and flow balancing devices. These must be properly located in straight lengths of pipe or duct for accurate measurement. The testing procedure depends on the system's characteristics and layout. Where possible, modifications should be made by the designer to provide "Pitot tube traverse planes" where accurate readings can be made. The interaction between individual terminals and devices that vary with system pressures and flows also affect the TAB work's accuracy.

### 9.7.2 Location of Traverse Plane

The qualifications for a Pitot tube traverse plan that is considered suitable for the measurements used in determining the airflow rate are as follows:

a. The velocity distribution should be uniform throughout the traverse plane. An ideal distribution would have 80 to 90 percent of the measurements greater than 10 percent of the maximum velocity. The poorest distribution is when no less than 75 percent of the velocity pressure measurements are greater than 10 percent of the maximum velocity, *see* Figure 9-7. However, when less than ideal distribution is encountered, traverse points must be increased even through accuracy will still be adversely affected.

b. The flows should be at right angles to the traverse plane. It is recommended that variations from these flow conditions, as a result of swirl or other mass turbulence, should be considered acceptable when the angle between the flow stream and the traverse plane is within 10 degrees of this right angle.

c. The cross-sectional shape and area of the duct should be uniform throughout the length of the duct in the vicinity of the traverse plane. In instances where the divergence or convergence of the duct is irregular, or more than moderate in degree, significantly non-uniform flow conditions may exist.

d. The traverse plane should be located to minimize the effects of leaks in the portion of the system that is located between the traverse plane and the fan. A location in a long, straight run of duct of uniform cross section will usually provide acceptable conditions for the Pitot-traverse plane. In some installations, more than one traverse plane may be required in order to account for the total flow. More than one traverse location per system may be used to substantiate accuracy of the system performance. Provisions should be included in the HVAC system design for suitable traverse plane locations.

e. In any instance in which the fan is ducted on the outlet side, and the traverse plane is to be located downstream from the fan. The traverse plane should be situated at a sufficient distance downstream to allow the flow to diffuse to a uniform velocity distribution and to allow the conversion of velocity pressure to static pressure.

f. The location for the traverse plane on the inlet side of the fan should be more than 0.5 equivalent diameters from the fan inlet. When the traverse plane must be located within an inlet box, the plane should be located a minimum of 12 in. (300 mm) downstream from the leaving edges of the damper blades and more than 0.5 equivalent diameters upstream from the edge of the inlet cone. In the case of double-inlet fans, traverses must be conducted in both inlet boxes in order to determine the total flow rate.

g. Regions immediately downstream from elbows, obstructions, and abrupt changes in airway area are not suitable traverse plane locations. Regions where unacceptable levels of swirl are usually present should be avoided, such as the region downstream from an axial-flow fan that is not equipped with straightening vanes.

### 9.7.3 Accuracy

Certification of the airflow rates to the specifications is the most difficult field measurement that the TAB technician has to perform. Most TAB procedures require measurements in the ducts as the most accurate

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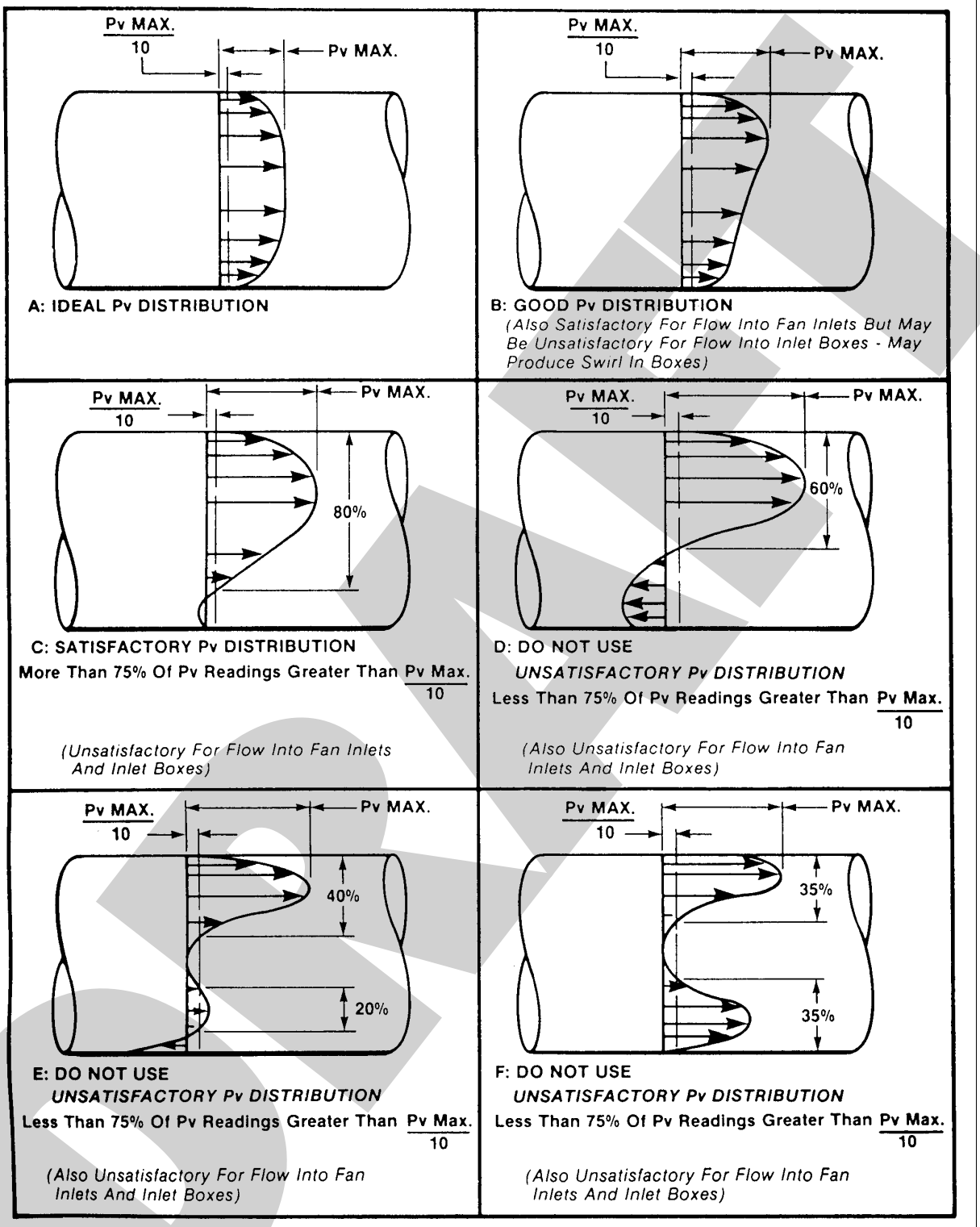


**FIGURE 9–7 TYPICAL VELOCITY PROFILES ENCOUNTERED IN VELOCITY PRESSURE MEASUREMENT PLANES**

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documentation of system performance. These measurements are at one plane; therefore, proper analysis of losses (duct leakage) must be considered. The duct traverse, as outlined in this section, is the only accepted method for field-testing flow-rate performance of fans by the AMCA 210 publications.

Rigorous error analysis shows that flow rate determinations by this method can range from 5 to 10 percent error. Experience shows that qualified TAB technicians can obtain measurements that range within 5 to 10 percent accuracy of actual flow, under good field conditions. It can also be determined by experienced technicians that suitable traverse conditions do not then exist and these measurement can exceed ± 10 percent error.

When the HVAC system designer requires a high degree of accuracy in the field measurement phase of the TAB work, the location and installation of orifice plates or venturi tubes must be included in the duct design and specifications.

## 9.8 TEST INSTRUMENTATION

Instruments for the measurement of airflow, water flow, rotation, temperature, and electricity are the "tools of the trade" of the TAB technician and the TAB firm. Many instruments are available to accomplish TAB tasks and tabulate TAB data and information. The instruments used by certified TAB firms are proven by experience to be reliable and accurate. In addition, electronic multi-purpose instruments are capable of providing more than one reading, such as airflow and temperature.

### 9.8.1 Tab Instrument Specification

The designer should specify that instruments selected or employed by the TAB firm, must meet the indicated range and minimum accuracy requirements of the SMACNA *HVAC Systems Testing, Adjusting & Balancing* manual. The accuracy and range reported by the instrument manufacturer shall be verified by an independent testing laboratory and be traceable to the National Institute of Standards and Technology or an equivalent institute in other countries.

Calibration requirements for each function should be specified and met. Some instruments, such as manometers, Pitot tubes, and thermometers, do not require calibration. However, if a "mechanical/electrical" device is substituted or employed in place of these types of instruments, the indicated calibration requirement should apply.

### 9.8.2 Application of Instruments

Instruments submitted by the TAB firm shall be suitable for their contemplated use. It is important to use the specific instrument that will produce the best results, with the least expenditure of time, effort, and money. However, in all situations, precise measurement must be the overriding factor. The following apply:

a. Instruments should be operated in accordance with the manufacturers recommendations.

b. The most suitable instrument, or combination of instruments, should be employed for a particular measurement or reading. For example, a traverse will be accomplished with a Pitot tube and manometer; it is unacceptable to use a Pitot tube with another device that does not provide the same overall accuracy.

Additional information is provided in Table 9-1 regarding recommended instrument use and instrument limitations.

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| INSTRUMENT | RECOMMENDED USES | LIMITATIONS |
|---|---|---|
| U-TUBE MANOMETER | Measuring pressure of air and gas above 1.0 in. wg (250 Pa)<br>Measuring low manifold gas pressures | Manometer should be clean and used with correct fluid.<br>Should not be used for readings under one inch of differential pressure. |
| VERTICAL INCLINED MANOMETER | Measuring pressure of air and gas above .02 in. wg (5 Pa)<br>Normally used with pitot tube or static probe for determination of static, total, and velocity pressures in duct systems. | Field calibration and leveling is required before each use.<br>For extremely low pressures, a micromanometer or some other sensitive instrument should be used for maximum accuracy. |
| MICRO-MANOMETER (ELECTRONIC) | Measuring very low pressures or velocities.<br>Used for calibration of other instrumentation. | Because some instruments utilize a time weighted average for each reading, it is difficult to measure pressures with pulsations. |
| PITOT TUBE | Used with manometer for determination of total, static and velocity pressures. | Accuracy depends on uniformity of flow and completeness of duct traverse.<br>Pitot tube and tubing must be dry, clean and free of leaks and sharp bends or obstructions. |
| PRESSURE GAUGE (MAGNEHELIC) | Used with static probes for determination of static pressure or static pressure differential. | Readings should be made in midrange of scale.<br>Should be "zeroed" and held in same position.<br>Should be checked against known pressure source with each use. |
| ANEMOMETER ROTATING VANE (MECHANICAL AND ELECTRONIC) | Measurement of velocities at air terminals, air inlets, and filter or coil banks. | Total inlet area of rotating vane must be in measured airflow.<br>Correction factors may apply, refer to manufacturer data. |
| ANEMOMETER DEFLECTING VANE | Measurement of velocities at air terminals and air inlets. | Instruments should not be used in extreme temperature or contaminated conditions. |
| ANEMOMETER THERMAL | Measurement of low velocities such as room air currents and airflow at hoods, troffers, and other low velocity apparatus. | Care should be taken for proper use of instrument probe.<br>Probes are subject to fouling by dust and corrosive air.<br>Should not be used in flammable or explosive atmosphere.<br>Temperature corrections may apply. |
| FLOW MEASURING HOOD | Measurement of air distribution devices directly in CFM (L/s) | Flow measuring hoods should not be used where the discharge velocities of the terminal devices are excessive.<br>Flow measuring hoods redirect the normal pattern of air diffusion which creates a slight, artificially imposed, pressure drop in the duct branch.<br>Capture hood used should provide a uniform velocity profile at sensing grid or device. |

**Table 9–1 Airflow Measuring Instruments**

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CHAPTER 10

# DESIGNING FOR SOUND AND VIBRATION

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## 10.1 SCOPE

Sound and vibration must be an integral part of the initial design for HVAC systems to perform at sound and vibration levels that are acceptable to building occupants. Knowledgeable acoustic professionals typically estimate a one to two percent investment to incorporate effective sound and vibration hardware and practices into the HVAC system design. This investment should be viewed by designers as an important attribute that should not be discounted by value engineering. Correcting acoustical problems after an HVAC system is installed can be many times more expensive, and in some situations, simply unachievable.

Even in cost-conscious construction for leasing, a quiet HVAC system can be the differentiator in satisfying clients and locating new tenants.

From a design engineering standpoint, HVAC system energy efficiency and "quiet" operation share many common technical elements.

Larger fan coils are quieter due to lower air velocities. This typically translates into lower energy use because of higher energy transfer rates.

Lower air velocities in duct systems may permit the use of lower horsepower fan motors, or at a minimum, less energy usage to distribute the conditioned air.

Smooth, long radius fittings and transitions are not only quieter than abrupt changes in direction, but also provide the desirable benefit of lower fan losses.

Since each HVAC system is unique, designers must have an understanding of the hardware and practices that can be employed to create a quiet, efficient HVAC system. However, the benefits and intent of even the best sound attenuation hardware can easily be erased by improper application, inattention to detail, or misunderstanding installation methods. When commissioning is part of the construction process, verifying that sound attenuation hardware is correctly installed assures the full benefit of the hardware.

This chapter only provides a general overview of sound and vibration as it applies to HVAC system design. Sound and vibration is a complex technical issue that requires specialized knowledge and training to address the initial design phase. When sound and vibration becomes a problem in a completed building, specialized diagnostic equipment and technicians with specific knowledge and training are required to characterize and address the problem. A more extensive treatment of this complex discipline is provided in SMACNA's *Sound and Vibration Manual* and other industry resources such as ASHRAE.

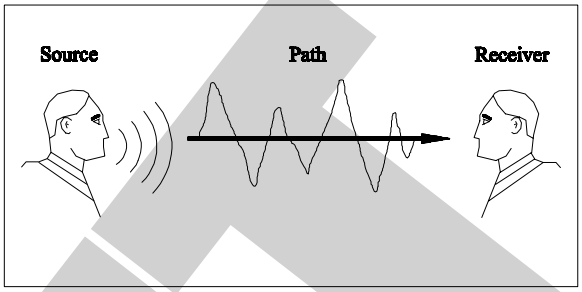


**FIGURE 10–1 SOURCE, PATH, AND RECEIVER**

## 10.2 BUILDING NOISE

Most building noise control situations involve three elements: the sound source, path, and receiver. The sound source is any device, mechanism, machine, or activity that generates sound. The path as shown in Figure 10-1 is the route the sound travels to the receiver. The receiver is a person or persons who hear the sound.

The sound source can be either desirable or undesirable. Desirable sound sources include speech, music, or any similar activities that are specifically designed to occur in an area. Undesirable sound sources generate sound that interferes with these activities. They include outdoor transportation activities, indoor and outdoor mechanical equipment, and indoor and outdoor people-related activities. Outdoor sound sources often generate sound that passes through the exterior shell of a building to indoor spaces.

Receivers are usually people who respond positively, passively, or negatively to sound to which they are exposed. Criteria that can be used as design guides must be specified to achieve acceptable sound levels in rooms. Acceptable sound criteria in buildings depends on whether the goal is to provide spaces where speech communication is easy and listening conditions are good, or to provide for reasonable acoustical comfort in work areas, homes, or vehicles.

The paths between the sound source and listener often are the only element that can be altered to improve speech or listening conditions if the sound source is desirable, or to prevent unwanted sound from reaching a listener if the sound source is undesirable.

Consider the mechanical equipment room and adjacent office shown in Figure 10-2. There are two sound sources in the mechanical equipment room: a fan and

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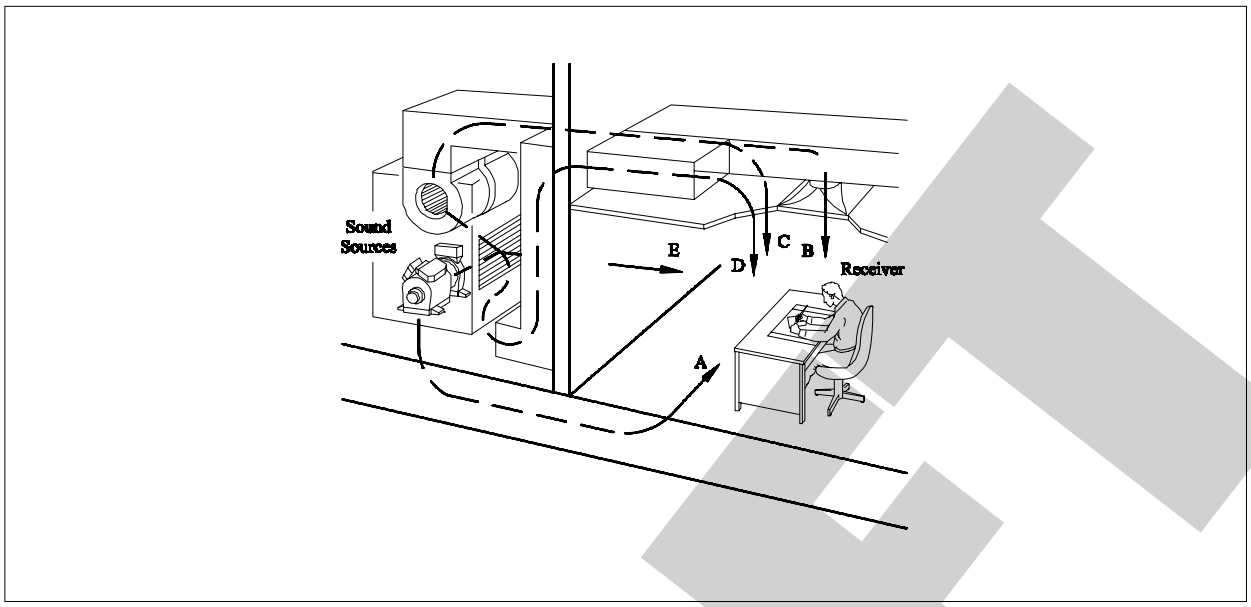


**FIGURE 10–2 MECHANICAL EQUIPMENT ROOM ADJACENT TO OFFICE AREA**

an air-conditioning compressor. There are several paths through which sound generated by these sources can travel from the mechanical equipment room to the receiver in the office area.

Path A is a structure-borne path through which vibration energy from the compressor is transmitted to the office area where it can be perceived as vibration in the floor or heard as airborne sound. Paths B through E are airborne paths. Path B is the path where sound from the fan is transmitted down the supply duct through the supply air diffuser to the office. Sound from the supply duct is transmitted through the duct wall to the office area through Path C.

Sound from both the fan and compressor is transmitted through the return air duct to the office area through path D. Sound from both the fan and compressor is transmitted through the mechanical equipment room wall to the office area through path E. All of the identified paths through which sound can travel must be properly designed and acoustically treated to achieve acceptable sound levels in the office.

The solution to most noise and vibration control problems involves examining the noise source, the sound transmission path, and receiver. These three elements often interact, and do not necessarily act independently. With respect to noise and vibration control in buildings, the receiver is always an individual or individuals. Since, it is not possible to do anything to modify the receiver, noise control efforts should be directed at modifying or changing the sound source. The design of mechanical and electrical systems in buildings is fixed, and redesign or modification of them is typically not an option. The only option normally available is to change, modify, or interrupt the sound transmission path.

## 10.3 HVAC NOISE

Background sound levels associated with HVAC and other mechanical systems in a building should be low enough so they blend into the overall background sound and do not interfere with the requirements of the areas being served. The sound should be unobtrusive, and have no audible tonal characteristics such as hum, whistle, whine, or rumble. There should be no detectable variations in the levels of different frequency components of the sound with time, which are associated with beats or other system-induced aerodynamic instabilities.

The sound and vibration sources associated with HVAC and other mechanical systems, and the many possible transmission paths between these sources and occupied areas within a building, must be examined when there are sound and vibration problems associated with these systems. Figure 10-2 indicates there are often many possible sound and vibration transmission paths between a sound or vibration source and occupied areas. For most HVAC and other mechanical systems, it generally is not possible for system designers to modify or change the source characteristics of HVAC and other mechanical systems, or the acoustical characteristics of occupied areas within a building.

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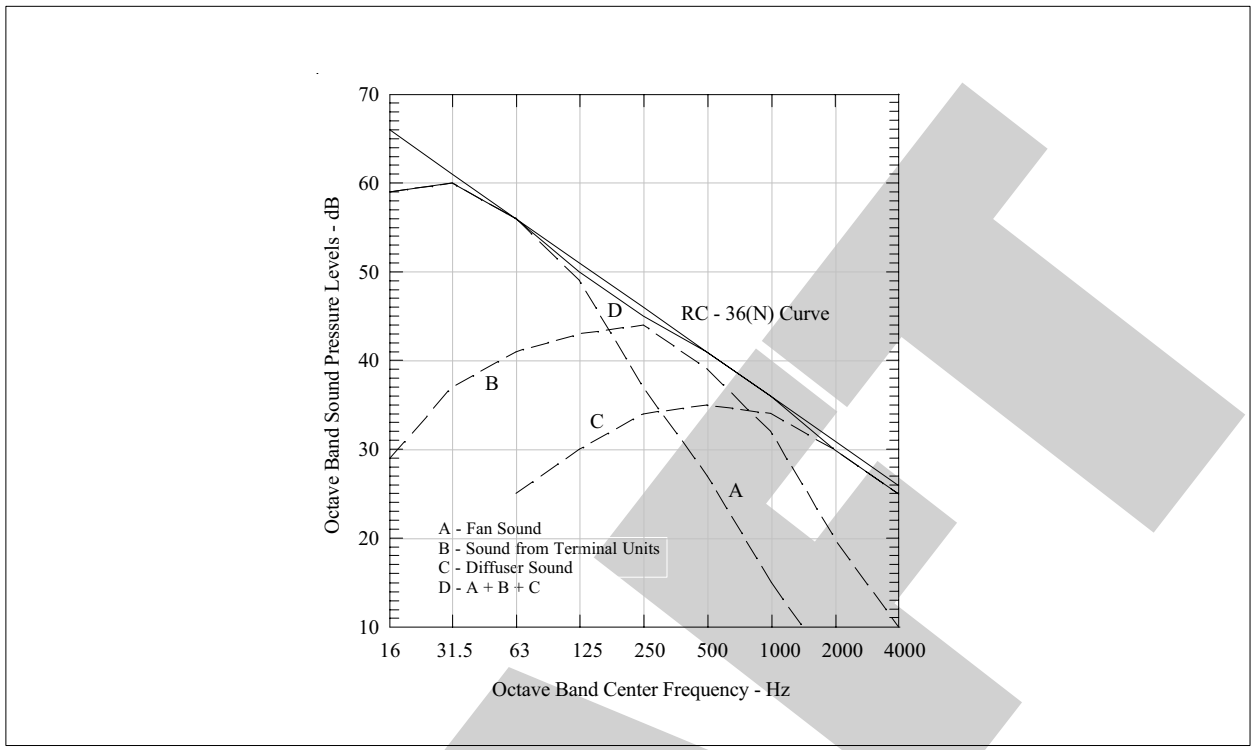


**FIGURE 10–3 ILLUSTRATION OF WELL– BALANCED HVAC SOUND SPECTRUM FOR OCCUPIED SPACES**

System designers are often constrained to modifying the sound and vibration transmission paths to achieve desired background sound levels associated with HVAC and other mechanical systems.

Different elements of HVAC systems contribute to the sound quality in a room. Fan noise normally contributes to the background sound levels in the 31.5 to 250 Hz octave frequency bands. This is shown in Figure 10-3 as curve A. Variable air volume units usually contribute to the background sound levels in the 125 to 2000 Hz octave frequency bands. This is shown as curve B in Figure 10-3. Diffuser noise usually contributes to the overall sound levels in the 250 to 8000 Hz octave frequency bands. This is shown as curve C in Figure 10-3. The overall sound pressure levels associated with all of these sound sources is shown as curve D. The Room Criterion (RC) level of the overall sound is RC 36(N). The RC 36(N) curve is superimposed over curve D. Curve D represents acceptable and desirable octave band sound pressure levels in many occupied spaces.

When a neutral RC level is not achieved in an occupied area, occupant complaints may occur. Figure 10-4 shows the frequency ranges of the most common sources of complaints associated with HVAC sound, and Figure 10-5 shows the frequencies at which different types of mechanical equipment generally control the background HVAC-related sound spectra in a room.

### 10.3.1 Design Methods to Reduce Sound Transmission

The following design methods should be considered when fans and other related mechanical equipment are selected, and when air distribution systems are designed to minimize the sound transmitted from different components of the system to the building areas they serve:

a. Design the air distribution system to minimize airflow resistance and turbulence. High airflow resistance increases the required total fan pressure, which results in higher fan noise. Turbulence increases the low frequency flow noise generated by duct fittings and dampers in the air distribution system.

b. Select a fan that will operate as near as possible to its rated peak efficiency when handling the required of airflow volume and total static pressure. Also, select a fan that generates the lowest possible sound but still meets the required design conditions for which it is selected. Oversized or undersized fans that do not operate at or near their rated peak effi-

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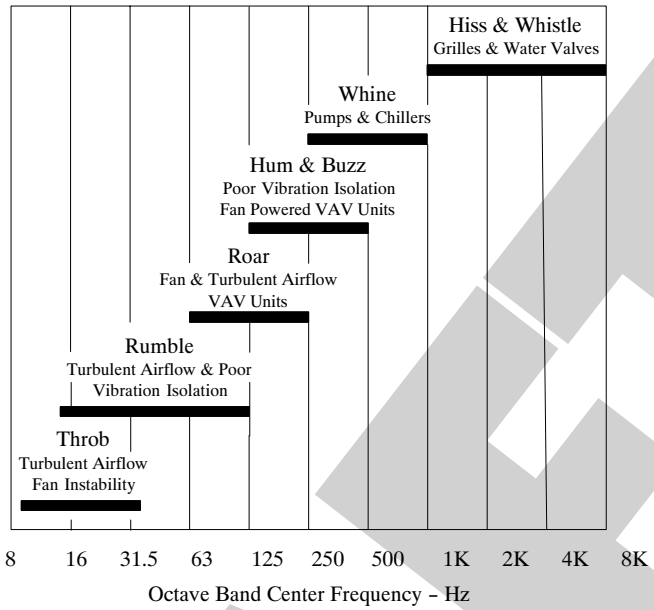


**FIGURE 10–4 FREQUENCY RANGES OF THE MOST LIKELY SOURCES OF ACOUSTICAL COMPLAINTS**

ciencies result in substantially higher fan sound levels.

c. Design the duct connection paths at the fan inlet and outlet for uniform and straight airflow. Failure to do this often results in severe air turbulence at the fan inlet and outlet and in airflow separation at the fan blades. Both can significantly increase fan sound levels.

d. Select duct silencers that do not significantly increase the required fan total static pressure. Improperly selected duct silencers can significantly increase the fan total static pressure. Silencer airflow regenerated sound can be minimized by selecting silencers with static pressure losses of 0.5 in. wg (124 Pa) or less.

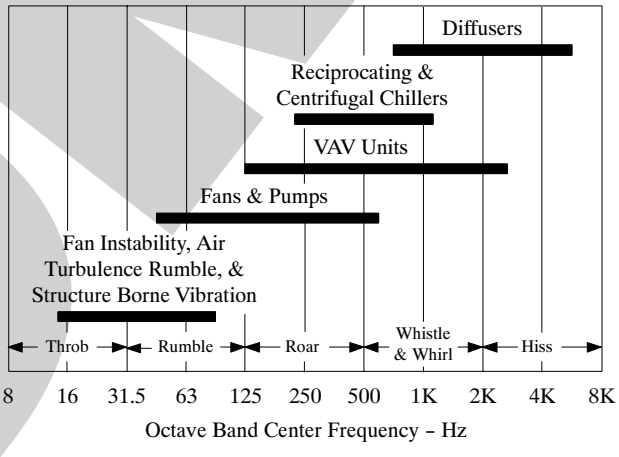


**FIGURE 10–5 FREQUENCY AT WHICH DIFFERENT TYPES OF MECHANICAL EQUIPMENT GENERALLY CONTROL SOUND SPECTRA**

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e. Do not place fan-powered mixing boxes over or near sound-sensitive areas.

f. To minimize airflow-generated sound at elbows or duct branch take-offs, locate elbows and duct branch take-offs at least four to five duct diameters from each other. For high-velocity systems, it may be necessary to increase this distance to up to 10 duct diameters in critical sound areas. Use "egg crate" grids in the necks of short-length takeoffs, which lead directly to grilles, registers, and diffusers. Do not use volume extractors, which protrude into the duct airflow.

| Sound Source | Path Nos. |
|---|---|
| Circulating fans; grilles; registers; diffusers; unitary equipment in room | 1 |
| Induction coil and fan-powered variable air-volume mixing units | 1, 2 |
| Unitary equipment located outside of room served, remotely located air-handling equipment, such as, fans, blowers, dampers, duct fittings and air washers | 2, 3 |
| Compressors, pumps, and other reciprocating and rotating equipment (excluding air-handling equipment) | 4, 5, 6 |
| Cooling towers; air-cooled condensers | 4, 5, 6, 7 |
| Exhaust fans; window air conditioners | 7, 8 |
| Sound transmission between rooms | 9, 10 |

| | Transmission Paths | Recommended Noise Reduction Methods |
|---|---|---|
| 1 | Direct sound radiated from sound source to ear | Direct sound can be controlled only by selecting quiet |
| | Reflected sound from floors, walls, and ceilings | Reflected sound is controlled by adding sound absorption to the room and to equipment location. |
| 2 | Air- and structure-borne sound radiated from casings and through walls of ducts and plenums is transmitted through walls and ceiling into room | Design duct and fittings for low turbulence; locate high velocity ducts in non-critical areas; isolate ducts and sound plenums from structure with neoprene or spring hangers. |
| 3 | Airborne sound radiated through supply and return air ducts to diffusers in room and then to listener by Path 1 | Select fans for minimum sound power; use ducts lined with sound-absorbing material; use duct silencers or sound plenums in supply and return air ducts |
| 4 | Noise is transmitted through equipment room walls and floors to adjacent rooms | Locate equipment rooms away from critical areas; use masonry blocks or concrete for equipment room walls and floor. |
| 5 | Building structure transmits vibration to adjacent walls and ceilings, from which it radiates as sound into a room by Path 1 | Mount all machines on properly designed vibration isolators; masonry blocks or concrete for equipment room walls and floor. |
| 6 | Vibration transmission along pipes and duct walls | Isolate pipe and ducts from structure with neoprene or spring hangers; install flexible connectors between pipes, ducts, and vibrating machines. |
| 7 | Noise radiated to outside enters room windows | Locate equipment away from critical areas; use barriers and covers to interrupt noise paths; select quiet equipment. |
| 8 | Inside noise follows Path 1 | Select quiet equipment. |
| 9 | Noise transmitted to an air diffuser in a room into duct and out through an air diffuser in another | Design and install duct attenuation to match transmission loss of wall between rooms. |
| 10 | Sound transmission through, over, and around | Extend partition to ceiling slab and tightly seal all around; seal all pipe, conduit, duct, and other partition penetrations. |

**Table 10–1 Sound Sources, Transmission Paths, and Recommended Noise Reduction Methods**

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g. It may be desirable to expand the duct cross-section area near critical sound areas to keep the airflow velocity in the duct as low as possible (1500 fpm or less). However, do not exceed an included expansion angle of greater than 15 degrees. Expansion angles greater than 15 degrees may result in airflow separation, producing duct rumble. Expanding the duct cross-sectional area will reduce the potential for noise associated with turbulent airflow in these areas.

h. Use turning vanes in large 90 degree rectangular elbows and 45 degree tapers in the upstream sides of the beginning of branch take-offs. This provides a smoother transition in which the air can change flow direction, reducing air turbulence.

i. Place grilles, diffusers, and registers into occupied spaces as far as possible from elbows and branch take-offs.

j. Do not use volume dampers near grills, diffusers, and registers in acoustically-critical situations.

k. Use vibration isolation for all reciprocating and rotating equipment when the equipment is located on upper floors or is roof-mounted. It often is necessary to vibration isolate mechanical equipment that is located in the basement of a building. Vibration isolate water pipes supported from the ceiling slab of a basement when tenant space is located directly above.

l. Use flexible connectors between rotating or reciprocating equipment and pipes and ducts that are connected to the equipment.

m. Use spring or neoprene hangers to vibration isolate ducts and pipes within a minimum of the first 50 ft (15.2 m) of the vibration-isolated equipment.

n. Use flexible conduit between rigid electrical conduit and reciprocating and rotating equipment.

o. Use barriers with respect to outdoor equipment when sound associated with the equipment will disturb adjacent properties if barriers are not used. Barriers typically produce no more than 15 dB of sound attenuation.

## 10.4 COMMON SOUND SOURCES

Table 10-1 lists several common sound sources associated with mechanical equipment. Anticipated sound transmission paths and recommended sound-reduction methods are also listed in the table. Airborne or structure-borne sound can follow any or all of the transmission paths associated with a specified sound source.

To effectively address each of the sound sources and related sound transmission paths associated with mechanical systems, the following design procedures are suggested:

a. Determine the desired design goal for HVAC system sound for each critical area according to its use and construction. Specify the desirable RC criterion.

b. Select quiet equipment, relative to equipment that radiates sound directly into a room.

c. Complete an initial design and layout of the HVAC system, using acoustical treatment where it appears appropriate, if central or roof-mounted mechanical equipment is used.

d. Add the sound attenuations and sound power levels associated with the central fan(s), fan-powered mixing units, if used, and duct elements between the central fan(s) and the room of interest to determine the corresponding sound pressure levels in the room. Be sure to investigate the supply and return air paths. Investigate and control possible duct sound breakout when fans are adjacent to the room of interest or roof-mounted equipment is above the room of interest.

e. Determine the sound pressure levels in the room of interest that are associated with sound transmitted through the mechanical equipment room wall, if the mechanical equipment room is adjacent to the room of interest.

f. Combine, on an energy basis, the sound pressure levels in the room of interest that are associated with all of the sound transmission paths between the mechanical equipment room or roof-mounted unit and the room of interest.

g. Determine the corresponding RC level associated with the calculated total sound pressure levels in the room of interest.

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h. Determine the octave frequency bands in which the corresponding sound pressure levels are exceeded and the sound paths that are associated with these octave frequency bands, if the RC level exceeds the design goal.

i. Add additional sound attenuation to the paths which contribute to the excessive sound pressure levels in the room of interest.

j. Repeat Steps (d.) through (i.) until the desired sound design goal is achieved.

k. Steps (c.) through (j.) must be repeated for every room that is to be analyzed.

l. Make sure that sound radiated by outdoor equipment will not disturb adjacent properties.

## 10.5 SOUND DATA STANDARDS

Accurate equipment sound data is essential when conducting acoustic design analyses of mechanical systems. This data is often available from equipment manufacturers in the form of equipment sound power levels or sound pressure levels at a specified distance from the equipment. To ensure the validity of equipment sound data, equipment manufacturers must use the most current versions of appropriate industry tests standards to obtain their data. With respect to sound measurements, these standards include:

a. ANSI S12.51/ISO 3741, American National Standard Acoustics Determination of Sound Power Levels of Noise Sources Using Sound Pressure - Precision Method for Reverberation Rooms.

b. ANSI S12.54/ISO 3744, American National Standard Acoustics - Determination of Sound Power Levels of Noise Sources Using Sound Pressure - Engineering Method in an Essentially Free Field over a Reflecting Plane.

c. AMCA 300, Reverberant Room Method for Sound Testing of Fans.

d. ASHRAE Standard 68R/AMCA 330, In-Duct Sound Power Measurement Procedure for Fans.

e. ASHRAE Standard 70, Method of Testing for Rating the Performance of Air Outlets and Inlets.

f. ASHRAE Standard 130, Methods of Testing for Rating Ducted Air Terminal Units.

g. ARI Standard 260, Standard for the Sound Rating of Ducted Air Moving and Conditioning Equipment.

h. ARI Standard 270, Standard for the Sound Rating of Outdoor Unitary Equipment

i. ARI Standard 275, Standard for Application of Sound Rated Outdoor Unitary Equipment.

j. ARI Standard 300, Standard for Rating of Sound Levels and Transmission Loss of Packaged Terminal Equipment.

k. ARI Standard 350, Standard for Sound Rating Non- Ducted Indoor Air-Conditioning Equipment.

l. ARI Standard 370, Standard for Sound Rating of Large Outdoor Refrigerating and Air-Conditioning Equipment.

m. ARI Standard 530, Standard for Method of Measuring Sound and Vibration of Refrigerant Compressors.

n. ARI Standard 575, Standard for Method of Measuring Machinery Sound within an Equipment Space.

o. ARI Standard 880, Standard for Air Terminals.

p. ARI Standard 885, Standard for the Procedure for Estimating Occupied Space Levels in the Application of Air Terminals and Air Outlets.

q. ARI Standard 890, Standard for Rating of Air Diffusers and Air Diffuser Assemblies.

r. ASTM E477, Test Method for Measuring Acoustical and Airflow Performance of Duct Liner Materials and Prefabricated Silencers.

When reviewing manufacturers' submittal sound data, require certification that the data has been obtain according to one or more of the relevant industry standards. Equipment not certified as meeting these standards should be rejected in favor of equipment whose data has been obtained according to relevant industry standards.

## 10.6 DUCT NOISE

Straight, unlined rectangular sheet metal ducts provide a small amount of sound attenuation. The attenuation

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is significant at low frequencies and tends to decrease with increasing frequency.

Unlined circular sheet metal ducts provide some sound attenuation that should be taken into account when designing a duct system. In contrast with rectangular sheet metal ducts, circular sheet metal ducts are much more rigid and do not resonate or absorb as much sound energy. Circular sheet metal ducts will only provide about one-tenth the sound attenuation at low frequencies as the sound attenuation associated with rectangular ducts.

Duct liner can be effective at reducing airborne sound, and provides some low frequency (rumble) attenuation if liner thicknesses are properly correlated to duct size.

Higher frequency sound attenuation, such as speaker tube effect, is two to four times more effective per unit length than when liner is applied to the outside throats of elbows, versus what can be achieved by just lining straight duct.

Low frequency sound attenuation requires thicker liner. The greater the need to dampen low frequency sound, the thicker the liner must be. The sound attenuation of 4 in. liner is approximately double that of 1 in. at 250 Hz, for example.

## 10.7 DUCT SILENCERS

Duct silencers are often used as a means to attenuate unwanted sound in ducts. How the air approaches a silencer, how the silencer is oriented to the fan, and where it is located in relation to the equipment room wall being penetrated is critical. Again, transitions must be designed so that airflow through silencers complies with the manufacturers design intent. The best location for a duct silencer is centered on the equipment wall. If a fire or smoke damper is required in the wall, then the silencer should be as close to the wall as possible but still within the equipment room. Otherwise, some sound may radiate from the equipment room directly into the ducts.

### 10.7.1 Duct Silencer Types

There are three types of duct silencers: dissipative, reactive (no sound absorbing material in the silencer cavities), and active silencers. When selecting silencers, the following parameters should be considered:

Insertion Loss: The difference between two sound pressure levels when measured at the same point before and after the silencer is installed.

Static Pressure Drop: The pressure drop across the silencer.

Regenerated Sound: Sound generated by the airflow through the silencer.

System effects can have a significant impact on the performance of dissipative and reactive silencers. These silencers should be placed at least three duct diameters from a fan, elbow branch take-off, or other duct element. Locating a silencer closer than three duct diameters can result in a significant increase in the static pressure drop across the silencer. This will usually cause an increase in both the fan and the silencer airflow-generated noise.

Airflow-generated noise occurs as air flows into, through, and out of a silencer. When the static pressure drop across a silencer is less than 0.35 in. wg (87.2 Pa), airflow generated noise is usually not a problem. However, when the static pressure loss across a silencer is greater than 0.35 in. wg (87.2 Pa), silencer airflow generated noise should be evaluated.

Dissipative and reactive silencers can have either a rectangular or circular cross as shown in Figure 10-6 (a) and (b). Rectangular and circular straight silencers are available in varying shapes. Straight silencers may have either side or center pods or both. Elbow silencers as shown in Figure 10-6 (c) are used when there is insufficient space for straight silencers. Special fan inlet and discharge silencers, including cone silencers, are designed to minimize system effects and to attenuate fan sound at its source.

The side and center pods of dissipative and reactive silencers are constructed of perforated sheet metal. The cavities of the pods for dissipative silencers are filled with either fiberglass or mineral wool, which gives dissipative silencers good broad-band sound attenuation characteristics. The cavities of the pods of reactive silencers are tuned chambers that are void of any fill material. Because of the tuning associated with reactive silencers, broad-band sound attenuation is often more difficult to achieve.

Dissipative and reactive silencers come in several different pressure drop configurations. The insertion loss, airflow-generated noise, and pressure drop of dissipative and reactive duct silencers are functions of silencer design and the location of the silencer in the duct system. These data are experimentally measured

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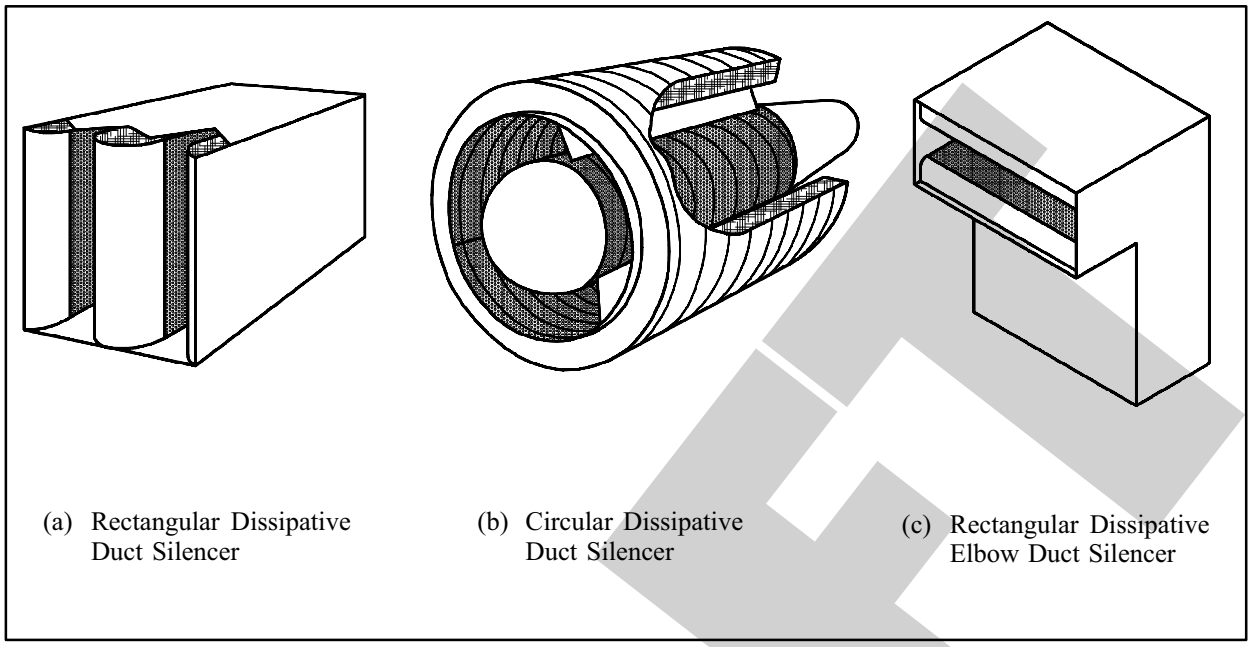

(a) Rectangular Dissipative Duct Silencer

(b) Circular Dissipative Duct Silencer

(c) Rectangular Dissipative Elbow Duct Silencer

**FIGURE 10–6 DISSIPATIVE PASSIVE DUCT SILENCERS**

and are presented as part of manufacturers' data associated with their product lines. The data should be obtained according to the procedures outlined in ASTM Standard E477, Standard Method of Testing Duct Liner Materials and Prefabricated Silencers for Acoustical and Airflow Performance.

### 10.7.2 Active Duct Silencers

Active duct sound attenuators reduce low-frequency duct noise by producing inverse sound waves that cancel the unwanted sound. They are effective in attenuating low frequency, pure-tone, and broad-band sound in a duct. Active duct silencers consist of a microprocessor, two microphones placed a specified distance apart in a duct, and a speaker that is placed between the microphones and mounted external to the duct but radiates sound into the duct, as shown in Figure 10-7. The microphone closest to a sound source that generates unwanted low-frequency sound senses the sound. The microphone signal is processed by the microprocessor which generates a signal that is 180 degrees out-of-phase with the unwanted sound which is transmitted to the speaker. The sound from the speaker destructively interferes with the unwanted sound, which causes the attenuation. The second microphone downstream of the speaker senses the attenuated sound and sends a corresponding feedback signal to the microprocessor, so the speaker signal can be adjusted. Active duct silencer systems have no com-

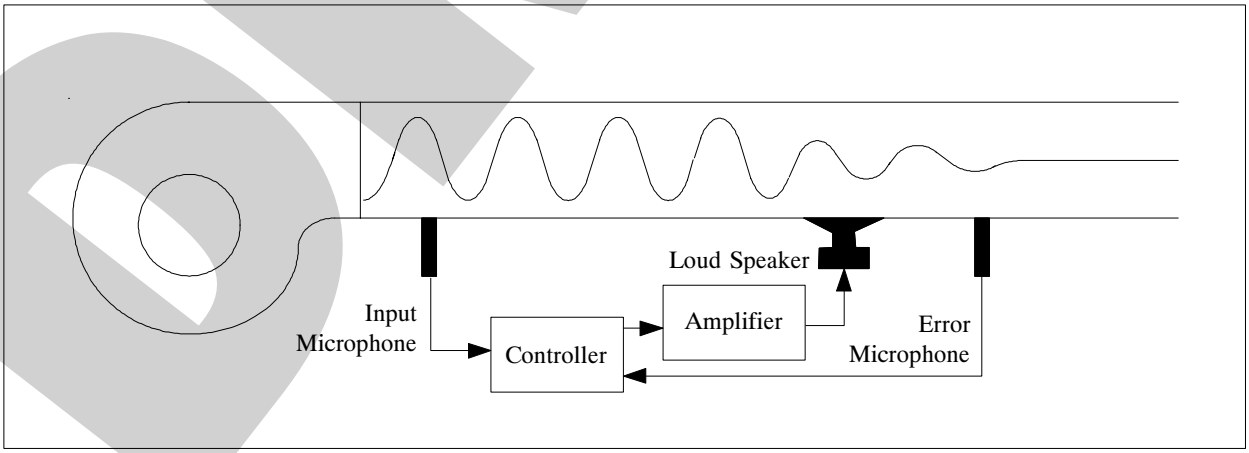


**FIGURE 10–7 ACTIVE DUCT SILENCER**

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ponents that are located within the duct. Thus, they can be used to attenuate unwanted sound without introducing a pressure loss or regenerated sound into a duct. Discrete frequency tones that typically exist between 40 and 400 Hz can be attenuated by as much as 35 dB. Broadband sound in this frequency range can be attenuated by 10 to 20 dB.

### 10.7.3 Active Attenuation Limitations

Active duct sound attenuators are limited by the presence of airflow turbulence and cross modes in the region of the microphones. The microphones detect flow noise associated with turbulence as "pseudo-noise" which inhibits the controller's ability to analyze the sound being attenuated. Thus, active systems should not be used in regions where the airflow velocity is greater than 1500 fpm or where there are duct elements or transitions that can generate significant turbulence.

Active duct sound attenuators have single microphones at the two microphone locations. Thus, they are most effective at attenuating plane waves where the sound pressure is constant across the duct cross-section. At higher frequencies where duct cross modes exist, the sound pressure across the duct cross-section is not constant and the corresponding effectiveness of active silencer in attenuating sound at these frequencies is significantly reduced.

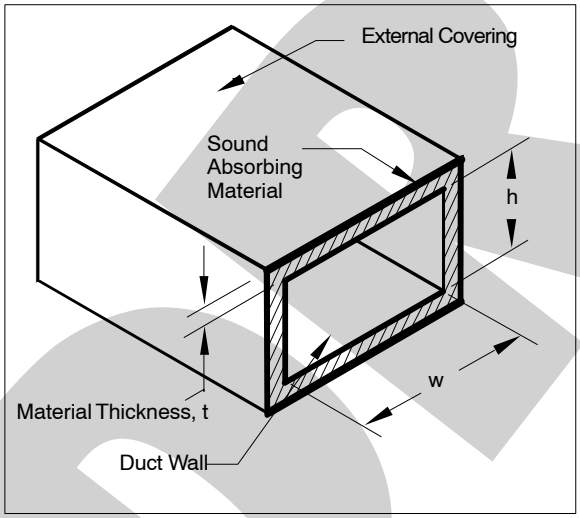


**FIGURE 10–8 EXTERNAL DUCT LAGGING ON RECTANGULAR DUCTS**

## 10.8 ACOUSTIC LAGGING

External acoustic lagging is often applied to rectangular duct work to reduce the transmission of sound energy from within the duct to surrounding areas. The lagging usually consists of a layer of soft, flexible, porous material, such as fiberglass, covered with an outer impervious layer as shown in Figure 10-8. A relatively rigid material, such as sheet metal or gypsum board, or a limp material such as sheet lead or loaded vinyl, can be used for the outer covering.

With respect to the insertion loss of externally-lagged rectangular ducts, different techniques must be used for rigid and limp outer coverings. When rigid materials are used for the outer covering, a pronounced resonance effect between the duct walls and the outer covering usually occurs. With limp materials, the variation in the separation between the duct and its outer covering dampens the resonance so that it no longer occurs.

## 10.9 DUCT SOUND BREAKOUT

Noise that is generated within a duct and then transmitted through the duct wall into the surrounding area is called duct sound breakout. This phenomenon is sometimes referred to as low-frequency duct rumble. There are two possible sources for duct sound breakout. One is associated with sound that is generated within the duct, usually by a fan or air turbulence around duct fittings associated with high-speed airflow in the duct. This sound, designated $W_i$ in Figure 10-9 is transmitted down the duct and then through the duct walls into surrounding spaces. The transmitted sound is designated $W_r$ in Figure 10-9 (a).

The second source is associated with turbulent airflow that aerodynamically excites the duct walls, causing them to vibrate. This vibration generates low-frequency duct rumble, which is then radiated as sound into the surrounding spaces. In many situations, particularly near the fan discharge, the duct sound breakout may be associated with both of these sources.

Sound that is transmitted into a duct from the surrounding area, and then transmitted within the duct is called duct sound breakout, as shown in Figure 10-9 (b). $W_i$ in the figure refers to sound in the area surrounding a duct that is incident on the duct walls, $W_t$ refers to the sound that is transmitted within the duct.

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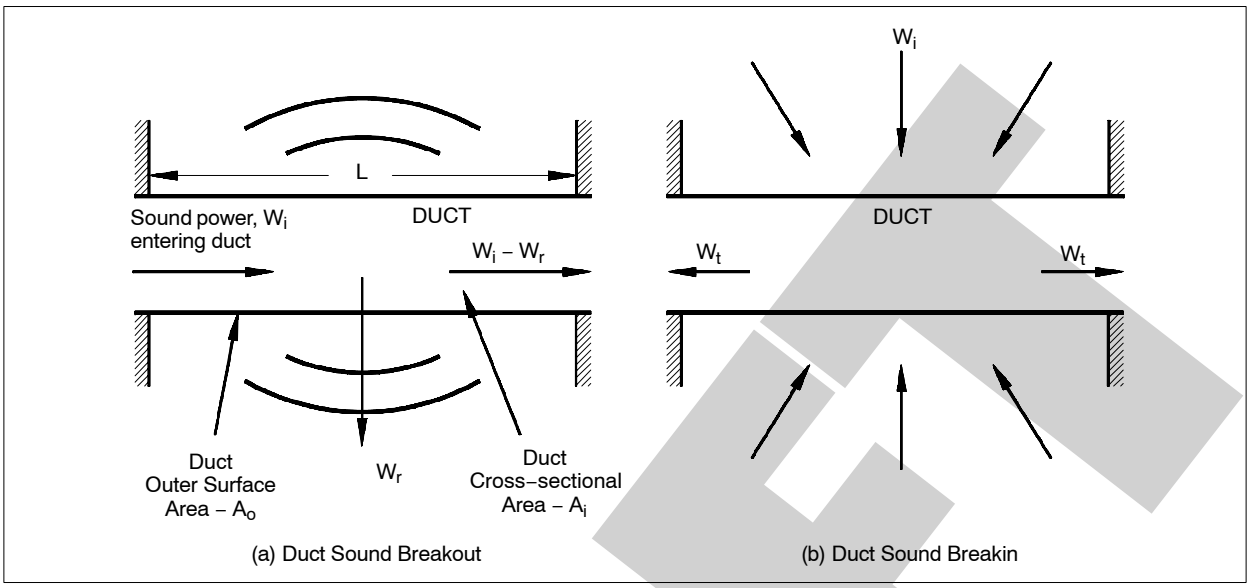


**FIGURE 10–9 BREAKOUT AND BREAK–IN OF SOUND IN DUCTS**

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CHAPTER 11

# DUCT SYSTEM CONSTRUCTION

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## 11.1 SCOPE

A variety of materials have been used in the construction of ducts. Selection of the materials used throughout the duct system, should follow the same careful consideration as the other HVAC system components. The materials used in duct systems can substantially affect the overall performance of the systems. The listed "advantages" should be evaluated with the "limiting characteristics," prior to the material selection.

Materials used for ducts include: galvanized steel, black carbon steel, aluminum, stainless steel, copper, fiberglass reinforced plastic (FRP), polyvinyl chloride (PVC), polyvinyl steel (PVS), concrete, fibrous glass (duct board), and gypsum board. Technical information will be given for each of the above materials but duct sizing and duct construction specifications will generally be stated in this manual in terms of use of galvanized steel as the material from which ducts are made. Figure A-2 and Table A-1 gives correction factors used to adjust duct friction losses for materials other than galvanized steel (multiply duct friction loss by the appropriate factor in the Table). Friction loss can be one of the most important factors to consider when selecting duct material, especially if low operating cost is a principal design objective.

Consideration must also be given to selection of duct construction components, other than those used for the duct walls. Items such as flexible ducts, duct liner, pressure-sensitive tapes, sealants, adhesives, reinforcements, and hangers, are described in individual SMACNA manuals, as well as many other publications.

Special material selection and construction may be necessary when designing systems serving nuclear projects, earthquake-prone areas, and projects with other unusual requirements.

## 11.2 DUCT SYSTEM SPECIFICATION CHECK LIST

In addition to SMACNA duct construction standards, specifications and detail drawings should include the following duct system requirements:

a. Local code requirements.

b. Duct system static pressure classifications, standard flag designation.

c. Duct material selection.

d. Allowable duct leakage, specify sealing system classification.

e. Insulation requirement, external and liner.

f. Sound control devices and methods.

g. Outlet and inlet performance.

h. Filters.

i. Dampers—fire, smoke, and volume control—and their location, size, and type.

j. Duct mounted apparatus.

k. Duct mounted equipment.

l. Special duct suspension requirements—such as seismic bracing, *see* SMACNA *Seismic Restraint Manual*.

A complete duct design specification needs to include all of the above specification requirements in sufficient detail to provide performance standards, materials, and design methods for all ducts and duct components required for the total HVAC system.

## 11.3 DUCT CONSTRUCTION MATERIALS

### 11.3.1 Galvanized Steel

#### 11.3.1.1 Applications

Widely used as a duct material for most air handling systems; not recommended for corrosive product handling or temperatures above 400 degree F (200 degree C).

#### 11.3.1.2 Advantages

High strength, rigidity, durability, rust resistance, availability, non-porous, smooth surface, workability, and weldability.

#### 11.3.1.3 Limiting Characteristics

Weldability, paintable, weight, corrosion resistance.

#### 11.3.1.4 Remarks

Galvanized steel sheet is customarily available in: commercial quality, lock-forming quality, drawing quality, special killed and physical (structural) quality. The most common material used for ductwork is lock-forming quality.

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Galvanized steel sheet is produced to various zinc-coating designations to give the service life required. Galvanizing may be accomplished by the electrolytic or hot-dipping process. Some types of galvanized coatings are: regular spangle, minimized spangle, iron-zinc alloy, and differential. Regular spangle is the most common type. Except for differential-coated sheet, the coating is always expressed as the total coating thickness of *both* surfaces. Galvanized sheets, with the surface treated for painting by phosphatizing, are commonly used.

SMACNA's *Duct Construction Standards Metal and Flexible* states: "Unless otherwise specified, steel sheet and strip used for duct and connectors shall be G-60 coated galvanized steel of lock-forming grade conforming to ASTM A653 and A924 standards. Minimum yield strength for steel sheet and reinforcements is 30,000 psi (207 kPa)."

Although SMACNA generally recommends G60 for duct construction G90 may be considered when the duct is exposed to industrial pollutants, marine atmospheres, or continuous contact with moisture.

### 11.3.2 Carbon Steel (Black Iron)

#### 11.3.2.1 Applications

Breechings, flues, stacks, hoods, other high temperature duct systems, kitchen exhaust systems, ducts requiring paint or special coating.

#### 11.3.2.2 Advantages

High strength, rigidity, durability, availability, paintable, weldability, and smooth surface non-porous.

#### 11.3.2.3 Limiting Characteristics

Corrosion resistance, weight.

#### 11.3.2.4 Remarks

Carbon steel is the designation for steel, when no minimum content is specified or required for aluminum, chromium, cobalt, columbium, molybdenum, nickel, titanium, tungsten, vanadium, zirconium or any element added to obtain a desired alloying effect.

Hot-rolled sheet is manufactured by hot-rolling slabs in a continuous mill to the required thickness. Cold-rolled sheet is manufactured from hot-rolled, descaled coils, by cold reducing to the desired thickness, generally followed by annealing to recrystalize the grain structure.

There are many different categories of black steel, with hot-rolled carbon being generally softer, less precisely rolled, less expensive, and, therefore, the most desirable for normal duct applications.

### 11.3.3 Aluminum

#### 11.3.3.1 Applications

Duct systems for moisture-laden air, louvers, special exhaust systems, ornamental duct systems. Sometimes aluminum is substituted for galvanized steel in HVAC duct systems.

#### 11.3.3.2 Advantages

Weight, resistance to moisture corrosion (salt free), non-magnetic, availability.

#### 11.3.3.3 Limiting Characteristics

Low strength, material cost, weldability, thermal expansion.

#### 11.3.3.4 Remarks

Various alloys are available in sheet form with the 3000 and 5000 temper series being the most commonly specified for duct systems. A "utility grade" sheet is normally available and widely used for HVAC system ductwork. Sheets can also be obtained with embossed or anodized surfaces.

### 11.3.4 Stainless Steel

#### 11.3.4.1 Applications

Duct systems for kitchen exhaust, moisture-laden air, fume exhaust.

#### 11.3.4.2 Advantages

High resistance to corrosion from moisture and most chemicals, ability to take a high polish.

#### 11.3.4.3 Limiting Characteristics

Labor and material cost, workability, availability.

#### 11.3.4.4 Remarks

Available in many different alloy combinations, type 304 and 316 are most commonly used. Stainless is usu-

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ally available in the following finishes as listed by ASTM:

Milled Rolled No. 1: Produced on hand sheet mills by hot rolling to specified thickness followed by annealing and de-scaling. Generally used in industrial applications, such as for heat and corrosion resistance, where smoothness of finish is not of particular importance.

Mill Rolled No. 2D: Produced on either hand sheet mills or continuous mills by cold rolling to the specified thickness, annealing and de-scaling. The dull, non-reflective finish may result from the de-scaling or pickling operation or may be developed by a final, light cold-rolled pass on dull rolls. The dull finish is favorable for retention of lubricants on the surface in deep drawing operations. This finish is generally used in forming deep-drawn articles that may be polished after fabrication.

Mill Rolled No. 2B: Commonly produced the same as 2D, except that the annealed and de-scaled sheet receives a final, light, cold-rolled pass on polished rolls. This is a general-purpose, cold-rolled finish. It is commonly used for all but exceptionally difficult deep drawing applications. This bright, moderately reflective finish is more readily polished than No. 1 or No 2D finish.

Bright Annealed Finish: A bright, cold-rolled, highly reflective finish retained by final annealing in a controlled atmosphere furnace. The purpose of the atmosphere is to prevent scaling or oxidation during annealing. The atmosphere is usually comprised of either dry hydrogen, or a mixture of dry hydrogen and dry nitrogen (sometimes known as dissociated ammonia).

Mill Polished No. 3: For use as a finish-polished surface, or as a semi-finished polished surface when it is required to receive subsequent finishing operations following fabrication. Where sheet or articles made from it will not be subjected to additional finishing or polishing operations, No. 4 finish is recommended.

Mill Polished No. 4: A bright appearance with a visible grain, but difficult to match. Widely used for restaurant equipment, kitchen equipment, store fronts, dairy equipment, etc. Following initial grinding with coarser abrasives, sheets are generally finished last with abrasives approximately 120 to 150 grit.

Mill Polished No. 6: Has a lower reflectivity than No. 4 finish. It is produced by Tampico brushing No. 4 finish sheets in a medium of abrasive and oil. It is used for architectural applications and ornamentation where high luster is undesirable. It is also used effectively to contrast with brighter finishes.

Mill Polished No. 7: Has a high degree of reflectivity. It is produced by buffing a finely ground surface, but the grit lines are not removed. It is chiefly used for architectural or ornamental purposes.

Mill Polished No. 8: The most reflective finish that is commonly produced. It is obtained by polishing with successively finer abrasives and buffing extensively with very fine buffing rouges. The surface is essentially free of grid lines from preliminary grinding operations. This finish is most widely used for press plates, as well as for small mirrors and reflectors.

### 11.3.5 Copper

#### 11.3.5.1 Applications

Duct systems for exposure to outside elements and moisture-laden air, certain chemical exhaust, ornamental ductwork.

#### 11.3.5.2 Advantages

Accepts solder readily, durable, resists corrosion, nonmagnetic.

#### 11.3.5.3 Limiting Characteristics

Cost, electrolysis, thermal expansion, stains.

#### 11.3.5.4 Remarks

Commonly used for ornamental systems and hoods. The various brown to green color shades (patina) formed by oxidation and exposure to moisture is considered a desirable characteristic.

### 11.3.6 Fiberglass Reinforced Plastic (FRP)

#### 11.3.6.1 Applications

Chemical exhaust, scrubbers, underground duct systems.

#### 11.3.6.2 Advantages

Resistance to corrosion, ease of modification.

#### 11.3.6.3 Limiting Characteristics

Cost, weight, range of chemical and physical properties, brittleness, fabrication (necessity of molds and expertise in mixing basic materials), code acceptance.

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| | | Typical Mechanical Properties | | | | | | Physical Properties | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Material | Basic Alloy Type | Yield Strength 0.2% Offset (PSI) | Tensile Strength (PSI) | Tensile Shear Strength (PSI) | Elongation (% in 2") | Hardness Rockwell | Modulus of Elasticity in Tension (PSI) | Density (lb/cu in.) | Thermal Expansion Coefficient 32° to 212° F (in/in.)° F) | Thermal Conductivity (Btu/sq ft/ft/HR/°F@68°F) | Approximate Melting Range (°F) |
| Stainless Steel | 304 | 30,000 | 75,000 | 60,000 | 40 | 92B | 29,000,000 | .288 | .0000092 | 9.2 | 2550-2650 |
| Copper | 110 | 28,000 | 36,000 | 25,000 | 30 | 60F | 17,000,000 | .322 | .0000094 | 226 | 1950-1980 |
| Aluminium | 3003 | 21,000 | 22,000 | 14,000 | 8 | 15T-60 | 10,000,000 | .099 | .0000129 | 127 | 1190-1210 |
| Galvanized Steel | | 35,000 | 50,000 | 37,000 | 33 | 55B | 29,000,000 | .284 | .0000065 | 35 | 2750-2775 |

**Table 11–1 Sheet Metal Properties**

### 11.3.7 Polyvinyl Chloride (PVC)

#### 11.3.7.1 Applications

Exhaust systems for chemical fumes and hospitals, underground duct systems.

#### 11.3.7.2 Advantages

Resistance to corrosion, weight, weldability, ease of modification.

#### 11.3.7.3 Limiting Characteristics

Cost, fabrication, code acceptance, thermal shock, weight.

### 11.3.8 Polyvinyl Steel (PVS)

#### 11.3.8.1 Applications

Underground duct systems, moisture-laden air, and corrosive air systems.

#### 11.3.8.2 Advantages

Resistance to corrosion, weight, workability fabrication, rigidity.

#### 11.3.8.3 Limiting Characteristics

Susceptible to coating damage, temperature limitations (250ºF or 120ºC Maximum), weldability, code acceptance.

#### 11.3.8.4 Remarks

Polyvinyl steel is a polyvinyl-chloride plastic coating heat fused to galvanized steel. Two-mil and four-mil coating thickness usually are standard, with steel gages (US standard) available from 26 ga through, and including 14 ga. This product is most popular in spiral formed pipe and is available in flat sheets and coil stock of lock-forming quality.

### 11.3.9 Concrete

#### 11.3.9.1 Applications

Underground ducts, air shafts.

#### 11.3.9.2 Advantages

Compressive strength, corrosion resistance.

#### 11.3.9.3 Limiting Characteristics

Cost, weight, rough surface (high friction) porous, fabrication (requires forming processes).

### 11.3.10 Rigid Fibrous Glass

#### 11.3.10.1 Applications

Interior HVAC low pressure duct systems; 2 in. wg maximum.

#### 11.3.10.2 Advantages

Weight, thermal insulation and vapor barrier, acoustical qualities, ease of modification, inexpensive tooling for fabrication.

#### 11.3.10.3 Limiting Characteristics

Cost, susceptible to damage, system pressure, code acceptance.

#### 11.3.10.4 Remarks

Joints must be properly stapled and taped, *see* SMACNA *Fibrous Glass Duct Construction Standards*.

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### 11.3.11 Gypsum Board

#### 11.3.11.1 Applications

Ceiling plenums, corridor ducts, return airshafts.

#### 11.3.11.2 Advantages

Cost, availability.

#### 11.3.11.3 Limiting Characteristics

Weight, code acceptance, leakage, deterioration when damp—not to be used in supply plenums as per code restrictions.

Not only can gypsum board become wet after installation from roof or piping leaks, it is also exposed to wetting or high moisture during on site storage and during and after installation—especially in buildings that are not yet totally enclosed with the roof and all windows in place.

Once gypsum board becomes wet, it is highly susceptible to mold growth that can seriously affect indoor air quality.

#### 11.3.11.4 Remarks

In cases where gypsum board is used as a return plenum—as a duct—it must be sealed in an equivalent manner as a duct. When used as an air duct concrete board should be considered to avoid the potential for mold growth on the organic papers used on most gypsum board. A possible, but less certain alternative, is water-resistance gypsum board.

## 11.4 ASTM STANDARDS

The American Society for Testing and Materials (ASTM) is a scientific and technical organization formed for "the development of standards on characteristics and performance of materials, products, systems and services and the promotion of related knowledge." It is the world's largest source of voluntary consensus standards. Numbered standards listed in this section refer to specifications that are defined in one of the 70 volumes of ASTM. The last two digits of the number designate the year of revision, such as the "05" in A653-05. These "yearly" designations have been dropped from this manual, but the complete list can be found on the ASTM web site, http://www.astm.org.

The following is a partial list of ASTM specifications for the construction of ductwork and is current at the time of publication of this manual.

| ASTM No. | Standard Title |
|---|---|
| A167 | Stainless and Heat-Resisting Chromium-Nickel Steel Plate, Sheet, and Strip |
| A176 | Stainless and Heat-Resisting Chromium Steel Plate, Sheet, and Strip |
| A263 | Stainless Chromium Steel-Clad Plate |
| A264 | Stainless Chromium-Nickel Steel-Clad Plate |
| A308 | Steel Sheet, Terne (Lead-Tin Alloy) Coated by the Hot-Dip Process |
| A424 | Steel, Sheet, for Porcelain Enameling |
| A463 | Steel Sheet, Aluminum-Coated, by the Hot-Dip Process |
| A480 | Flat-Rolled Stainless and Heat-Resisting Steel Plate, Sheet, and Strip |
| A505 | Steel, Sheet and Strip, Alloy, Hot-Rolled and Cold-Rolled |
| A506 | Alloy and Structural Alloy Steel, Sheet and Strip, Hot-Rolled and Cold-Rolled |
| A568 | Steel, Sheet, Carbon, and High-Strength, Low-Alloy, Hot-Rolled and Cold-Rolled |
| A599 | Tin Mill Products, Electrolytic Tin-Coated, Cold-Rolled Sheet |
| A606 | Steel, Sheet and Strip, High-Strength, Low-Alloy, Hot-Rolled and Cold-Rolled, with Improved Atmospheric Corrosion Resistance |
| A653 | Steel Sheet, Zinc-Coated (Galvanized) or Zinc-Iron Alloy-Coated (Galvannealed) by the Hot-Dip Process |
| A666 | Standard Specification for Annealed or Cold-Worked Austenitic Stainless Steel Sheet, Strip, Plate, and Flat Bar |
| A879 | Steel Sheet, Zinc Coated by the Electrolytic Process for Applications Re- |

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| | |
|---|---|
| | quiring Designation of the Coating Mass on Each Surface |
| A1008 | Steel, Sheet, Cold-Rolled, Carbon, Structural, High-Strength Low-Alloy, High-Strength Low-Alloy with Improved Formability, Solution Hardened, and Bake Hardenable |
| A1011 | Steel, Sheet and Strip, Hot-Rolled, Carbon, Structural, High-Strength Low-Alloy and High-Strength Low-Alloy with Improved Formability |
| B36 | Brass Plate, Sheet, Strip, And Rolled Bar |
| B101 | Lead-Coated Copper Sheet and Strip for Building Construction |
| B152 | Copper Sheet, Strip, Plate, and Rolled Bar |
| B209 | Aluminum and Aluminum-Alloy Sheet and Plate |
| B370 | Copper Sheet and Strip for Building Construction |
| C14 | Concrete Sewer, Storm Drain, and Culvert Pipe |
| C94 | Standard Specification for Ready-Mixed Concrete |
| C700 | Vitrified Clay Pipe, Extra Strength, Standard Strength, and Perforated |
| D2241 | Poly(Vinyl Chloride) (PVC) Pressure-Rated Pipe (SDR Series) |

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CHAPTER 12

# SPECIAL DUCT SYSTEMS

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## 12.1 SCOPE

The designer sometimes encounters ductwork requiring special attention. It is of utmost importance that the duct designer be aware of the different installation and code requirements for the specific geographical area of the installation.

Before starting the design of any special systems the designer must be thoroughly familiar with locally accepted practices and special requirements of applicable governing authorities.

This chapter provides a general description of those special duct systems frequently encountered in commercial HVAC work.

## 12.2 KITCHEN AND MOISTURE – LADEN SYSTEMS

### 12.2.1 Dishwasher Exhaust and Moisture–Laden Ducts

Exhausting moist air should be accomplished through ducts fabricated from non-corrosive materials. These ducts should be sloped toward the source of moisture or provided with proper drains. All seams and joints must be sealed watertight and oriented to not collect internal condensation. The temperature of the vapor may be excessively high, which may require the use of duct insulation or other treatment. Penetrations of the duct should be avoided.

### 12.2.2 Range and Grease Hood Exhaust Ducts

Vapor from cooking equipment must be handled exclusively through ducts designed specifically for that purpose. Care must be taken to make sure these ducts will be able to contain fire and smoke. Materials used must be heavier than standard duct and continuously welded to provide a liquid-tight system. Cleanouts should be provided at each change-of-direction in the duct. The system should be constructed so grease cannot be trapped. The duct should be sloped toward the hood or grease reservoir.

Ducts within the building should lead directly to the exterior. Where ducts pass through combustible walls and partitions, adequate clearance or protection must be provided. In the event of a fire, temperatures in excess of 2000 degrees F (1100 degrees C) may be experienced. Fire extinguishing systems may be required by local codes. Long, straight runs of duct should have provision for expansion.

Local codes governing range and grease-hood duct systems vary widely; therefore, it is imperative that the designer be familiar with these codes and construction requirements of NFPA, which is the governing code body for these duct systems.

### 12.2.3 Cage Wash Exhaust Systems

Ducts that must convey very high moisture laden exhaust air from cage cleaning and wash-down rooms.

## 12.3 SYSTEMS HANDLING SPECIAL GASES

### 12.3.1 Corrosive Vapors and Noxious Gases

Ducts that convey corrosive vapors or noxious gases, should be fabricated from materials impervious to all gases that may be handled, and must be sealed airtight. They must terminate outside the building, maintaining clearances from walls, roof, adjacent buildings, traffic areas, or equipment. The discharge airflow should not contaminate outside air intakes and other building openings under any condition.

### 12.3.2 Flammable Vapors

Ducts conveying flammable vapors must be sealed airtight and terminate outside of the building, maintaining adequate clearances from building construction and other objects. Nonflammable materials must be used for the ducts and duct supports.

## 12.4 INDUSTRIAL DUCT

Industrial process or material handling systems are covered in other SMACNA publications and will not be addressed in detail at this time. However, the following definitions and classifications for industrial duct are provided for reference.

### 12.4.1 Definition of Industrial Duct

Industrial duct is a broad classification of duct work used for many diverse applications. Air pollution control systems, industrial exhaust systems, general plant process ventilation, and systems outside the pressure range of commercial system standards are classified as industrial systems.

Independent of the operating temperature and pressure, industrial duct usually conveys air or gases contaminated with particulate or corrosive aerosols at velocities in excess of 2000 fpm (10.2 m/sec). OSHA defines the exhaust ventilation system as a sys-

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tem for removing contaminated air from a space. The general list of industrial contaminants includes dusts, fumes, mists, vapors, and gases.

Some industrial applications may include ducts operating at pressures within the range typical of systems covered by SMACNA's *HVAC Duct Construction Standards, Metal and Flexible*. While there may exist an overlap in operating pressures and temperatures, closer inspection reveals marked differences between industrial and HVAC applications in performance, maintenance, and structural requirements.

Industrial duct is found in industries such as: automotive, foundry, mining, petrochemical, pulp and paper, semiconductors, and other manufacturing. Industrial ducts are also used in pneumatic conveying systems. These systems often require special structural considerations due to the possibility of erosion, corrosion, and particulate accumulation loading. They may also include special provisions for maintenance, cleaning, fire protection, and drainage.

Industrial duct used in food processing and in systems handling materials such as rubber and paper requires smooth internal surfaces to prevent accumulation or contamination; this requirement often dictates the joining methods, which may therefore be limited to welding.

There are currently five classes of industrial duct. The designer should carefully analyze his application and specify the duct classification that most closely describes the intended use, to ensure proper construction. Round duct is preferred in industrial ventilation, air pollution control, and dust collection systems. Round duct provides for efficient air flow and has superior structural characteristics, resulting in system construction of lighter gage material and fewer, lighter reinforcing members.

Some of the more widely used sources are:

- *Industrial Ventilation, A Manual of Recommended Practice,* 25th Edition. American Conference of Governmental Industrial Hygienists, Inc., Cincinnati, OH. 2004.
- ANSI/NFPA 91, *Standard for Exhaust Systems for Air Conveying of Vapors, Gases, Mists, and Noncombustible Particulate Solids*. National Fire Protection Association, Quincy, MA. 1999.
- ANSI/AIHA Standard Z9.2, *Fundamentals Governing the Design and Operation of Local Exhaust Ventilation Systems*. American National Standards Institute, New York, New York. ibid. 2001.
- ANSI/CEMA Standard 550–2003, *Classifications and Definitions of Bulk Materials*, Conveyor Equipment Manufacturers Association, Naples, Florida. 2003.

### 12.4.2 Duct System Classification

There are five classifications of industrial duct systems:

**Class 1:** Non-abrasive, non-corrosive applications, including contaminated duct sections of make-up air and general ventilation systems, and gaseous emission control systems.

*NOTE: At the option of the specifier, Class 1 industrial duct may be used in lieu of HVAC duct for make-up air, general ventilation and distribution systems in industrial plants and production areas where the higher strength provisions of industrial duct may more adequately meet the physical demands of exposure to heavy traffic and frequent system modifications resulting from changes in production processes and plant layout.*

**Class 2:** Applications with light to moderately abrasive particulate in light concentration: buffing and polishing, woodworking, grain dust, etc.

**Class 3:** Applications with highly abrasive particulate in low concentrations or moderately abrasive particulate in higher concentrations; such as abrasive cleaning operations, dryers and kilns, boiler breaching, sand handling, etc.

**Class 4:** Applications with highly abrasive particulate in high concentrations; such as: high concentrations of particulate from any of the examples listed under Class 3 (used in heavy industrial plants, such as: steel mills, foundries, mining, and smelting).

**Class 5:** Applications with corrosives, such as acid vapors.

The first four classifications were originally adopted by the American Conference of Governmental Industrial Hygienists and published in the *Industrial Ventilation Manual of Recommended Practice*, which has received wide acceptance throughout the industrial ventilation industry. Class 5 appeared in the Uniform Mechanical Code in 1991.

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APPENDIX A

# DUCT DESIGN TABLES AND CHARTS

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This Appendix of the *HVAC Systems Duct Design* manual combines basic duct system design tables and and charts into an easy-to-use, single source text. Prior to the first edition of this manual, duct system designers had to search for fitting loss-coefficient tables and duct pressure-loss charts through many fluid flow handbooks, manufacturers' literature, and other sources.

Some of the tables come from unconfirmed sources. A worldwide literature search on fitting loss coefficients was conducted by I.E. Idel' chik in Russia and the resultant compilation was published in 1960 and updated in 1975.

Although some of these loss coefficients conflict with existing tables used in segments of the industry, data is reasonable to use for the design of HVAC duct systems. There is also a duct fitting database developed by ASHRAE.

Numbers in parantheses, when found at the end of a table or figure title, indicate the number of the reference source in the front of the manual. Where no reference number is indicated, the data was developed or obtained from SMACNA research.

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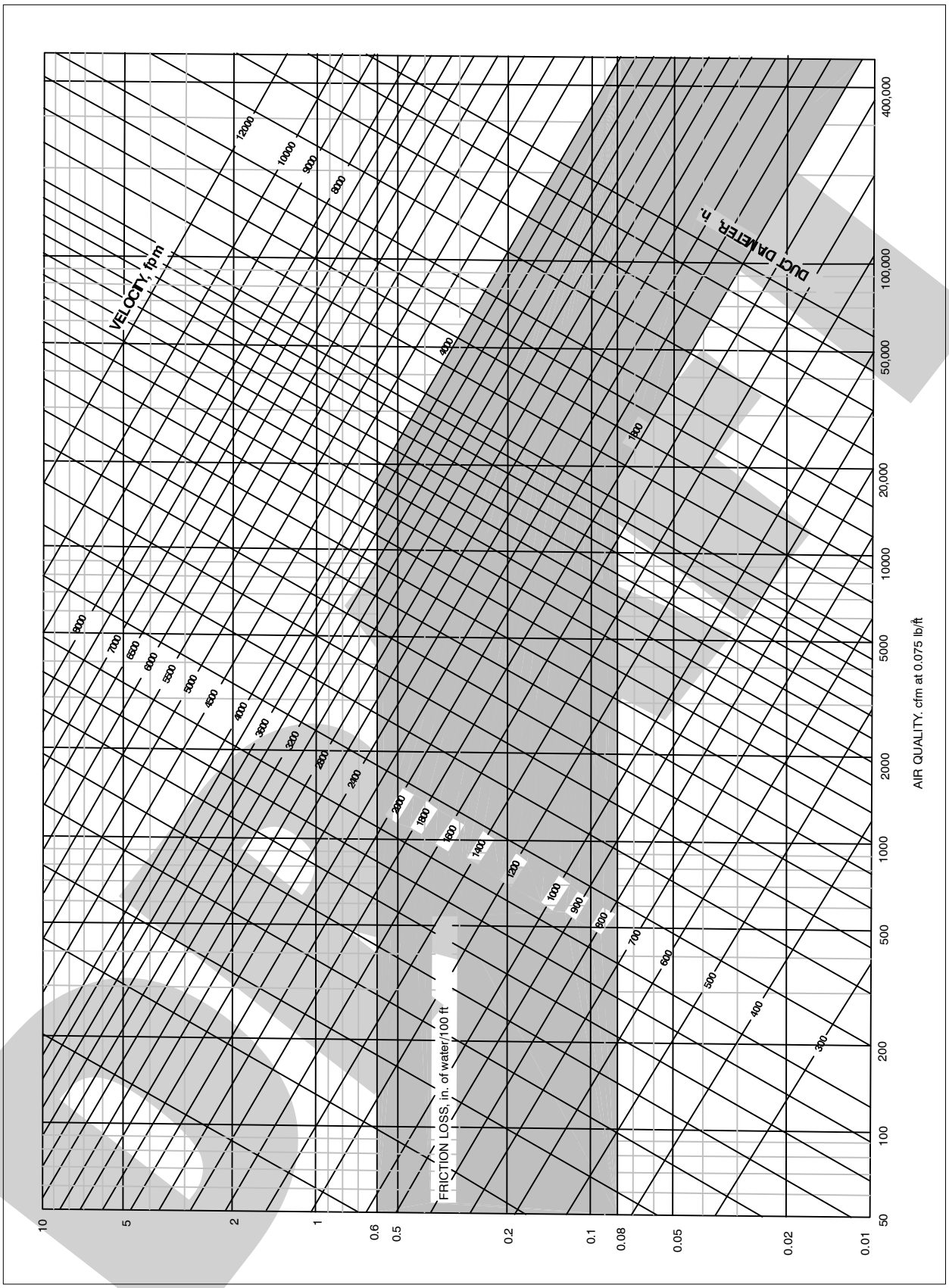


**FIGURE A–1 DUCT FRICTION LOSS CHART**

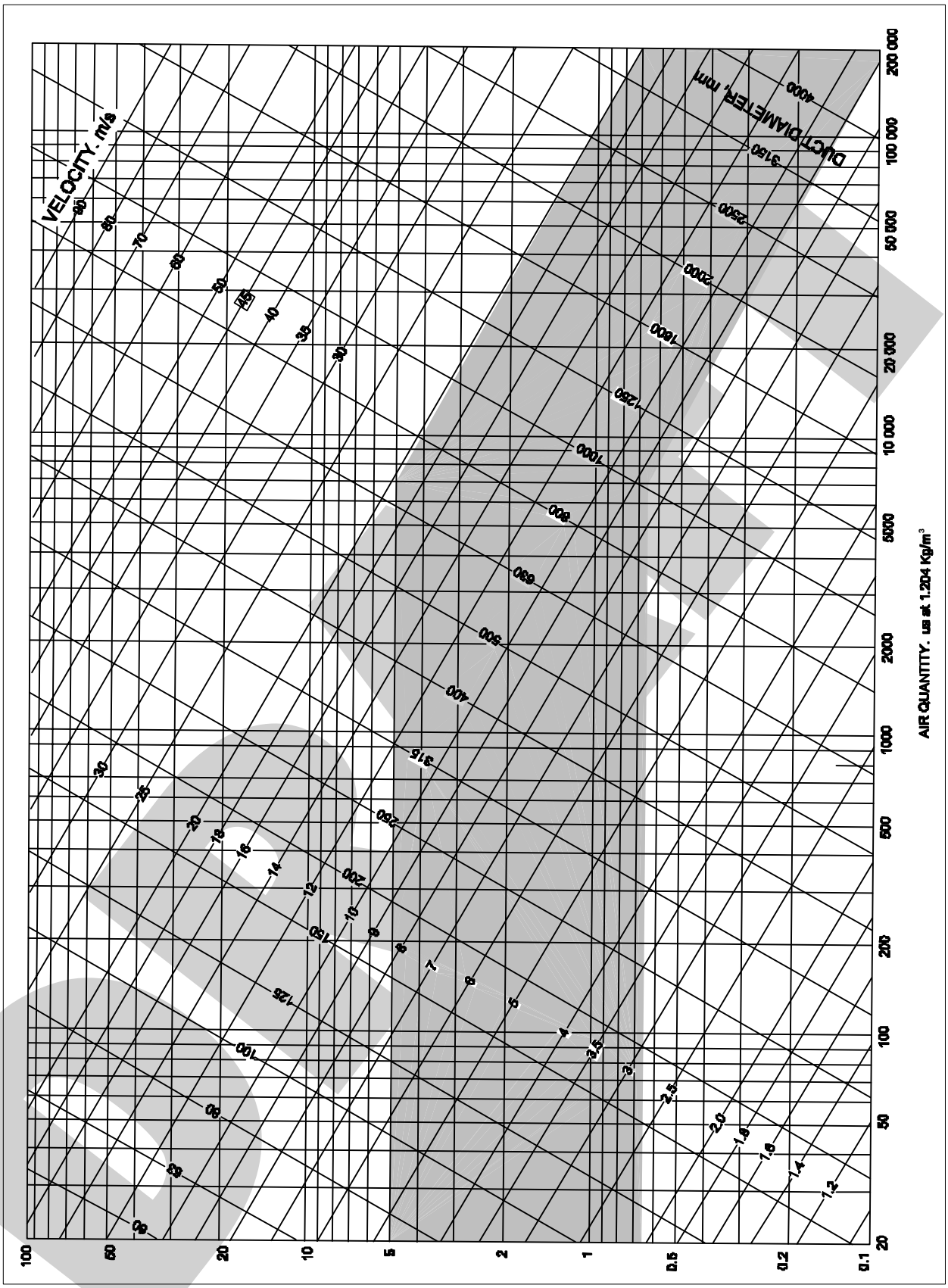


**FIGURE A–1M DUCT FRICTION LOSS CHART**

| Duct Material | Roughness Category | Absolute Roughness $\varepsilon_1$ | |
|---|---|---|---|
| | | ft | mm |
| Uncoated carbon steel, clean (Moody 1944) (0.00015 ft) (0.05 mm)<br>PVC plastic pipe (Swim 1982) (0.0003 to 0.00015 ft) (0.01 to 0.05 mm)<br>Aluminum (Hutchinson 1953) (0.00015 to 0.0002 ft) (0.04 to 0.06 mm) | Smooth | 0.0001 | 0.03 |
| Galvanized steel, longitudinal seams, 4 ft (1200 mm) joints (Griggs 1987) (0.00016 to 0.00032 ft) (0.05 to 0.1 mm) | Medium Smooth | 0.0003 | 0.09 |
| Galvanized steel, spiral seam with 1, 2, and 3ribs, 12 ft (3600 mm) joints (Jones 1979, Griggs 1987) (0.00018 to 0.00038 ft) (0.05 to 0.12 mm) | (New Duct Friction Loss Chart) | | |
| Hot-dipped galvanized steel, longitudinal seams, 2.5ft (760 mm) joints (Wright 1945) (0.0005 ft) (0.15 mm) | Old Average | 0.0005 | 0.15 |
| Fibrous glass duct, rigid<br>Fibrous glass duct liner, air side with facing material (Swim 1978) (0.005 ft) (1.5 mm) | Medium Rough | 0.003 | 0.9 |
| Fibrous glass duct liner, air side spray coated (Swim 1978) (0.015 ft) (4.5 mm)<br>Flexible duct, metallic, (0.004 to 0.007 ft (1.2 to 2.1 mm) when fully extended)<br>Flexible duct, all types of fabric and wire (0.0035 to 0.015 ft (1.0 to 4.6 mm) when fully extended)<br>Concrete (Moody 1944) (0.001 to 0.01 ft) (0.3 to 3.0 mm) | Rough | 0.01 | 3.0 |

**Table A–1 Duct Material Roughness Factors**

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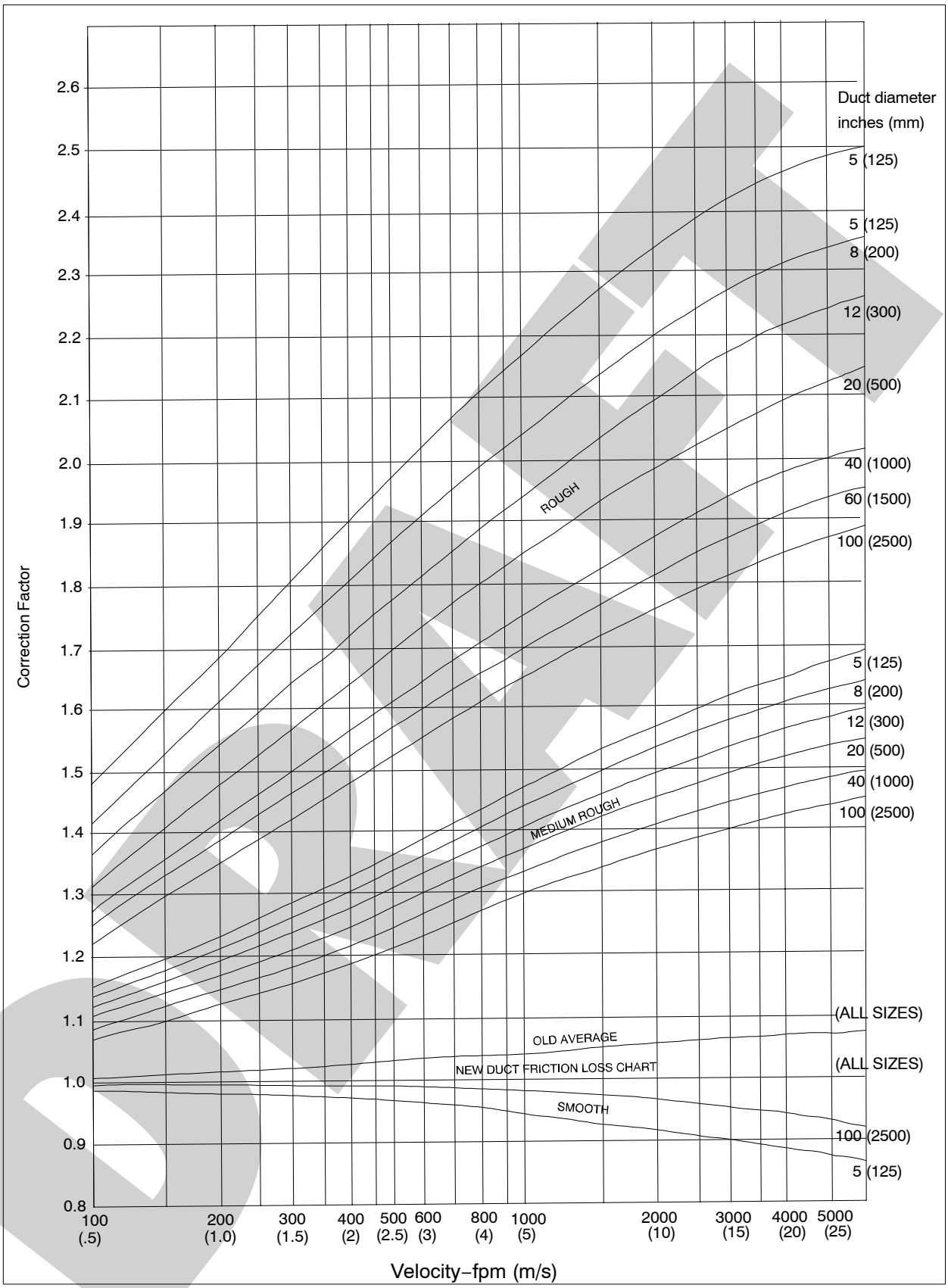


**FIGURE A–2 DUCT FRICTION LOSS CORRECTION FACTORS**

| Side Rectangular Duct | 4.0 | 4.5 | 5.0 | 5.5 | 6.0 | 6.5 | 7.0 | 7.5 | 8.0 | 9.0 | 10.0 | 11.0 | 12.0 | 13.0 | 14.0 | 15.0 | 16.0 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 3.0 | 3.8 | 4.0 | 4.2 | 4.4 | 4.6 | 4.7 | 4.9 | 5.1 | 5.2 | 5.5 | 5.7 | 6.0 | 6.2 | 6.4 | 6.6 | 6.8 | 7.0 |
| 3.5 | 4.1 | 4.3 | 4.6 | 4.8 | 5.0 | 5.2 | 5.3 | 5.5 | 5.7 | 6.0 | 6.3 | 6.5 | 6.8 | 7.0 | 7.2 | 7.5 | 7.7 |
| 4.0 | 4.4 | 4.6 | 4.9 | 5.1 | 5.3 | 5.5 | 5.7 | 5.9 | 6.1 | 6.4 | 6.7 | 7.0 | 7.3 | 7.6 | 7.8 | 8.1 | 8.3 |
| 4.5 | 4.6 | 4.9 | 5.2 | 5.4 | 5.7 | 5.9 | 6.1 | 6.3 | 6.5 | 6.9 | 7.2 | 7.5 | 7.8 | 8.1 | 8.4 | 8.6 | 8.8 |
| 5.0 | 4.9 | 5.2 | 5.5 | 5.7 | 6.0 | 6.2 | 6.4 | 6.7 | 6.9 | 7.3 | 7.6 | 8.0 | 8.3 | 8.6 | 8.9 | 9.1 | 9.4 |
| 5.5 | 5.1 | 5.4 | 5.7 | 6.0 | 6.3 | 6.5 | 6.8 | 7.0 | 7.2 | 7.6 | 8.0 | 8.4 | 8.7 | 9.0 | 9.3 | 9.6 | 9.9 |

| Side Rectangular Duct | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 22 | 24 | 26 | 28 | 30 | Side Rectangular Duct |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 6.6 | | | | | | | | | | | | | | | | | | | | 6 |
| 7 | 7.1 | 7.7 | | | | | | | | | | | | | | | | | | | 7 |
| 8 | 7.6 | 8.2 | 8.7 | | | | | | | | | | | | | | | | | | 8 |
| 9 | 8.0 | 8.7 | 9.3 | 9.8 | | | | | | | | | | | | | | | | | 9 |
| 10 | 8.4 | 9.1 | 9.8 | 10.4 | 10.9 | | | | | | | | | | | | | | | | 10 |
| 11 | 8.8 | 9.5 | 10.2 | 10.9 | 11.5 | 12.0 | | | | | | | | | | | | | | | 11 |
| 12 | 9.1 | 9.9 | 10.7 | 11.3 | 12.0 | 12.6 | 13.1 | | | | | | | | | | | | | | 12 |
| 13 | 9.5 | 10.3 | 11.1 | 11.8 | 12.4 | 13.1 | 13.7 | 14.2 | | | | | | | | | | | | | 13 |
| 14 | 9.8 | 10.7 | 11.5 | 12.2 | 12.9 | 13.5 | 14.2 | 14.7 | 15.3 | | | | | | | | | | | | 14 |
| 15 | 10.1 | 11.0 | 11.8 | 12.6 | 13.3 | 14.0 | 14.6 | 15.3 | 15.8 | 16.4 | | | | | | | | | | | 15 |
| 16 | 10.4 | 11.3 | 12.2 | 13.0 | 13.7 | 14.4 | 15.1 | 15.7 | 16.4 | 16.9 | 17.5 | | | | | | | | | | 16 |
| 17 | 10.7 | 11.6 | 12.5 | 13.4 | 14.1 | 14.9 | 15.6 | 16.2 | 16.8 | 17.4 | 18.0 | 18.6 | | | | | | | | | 17 |
| 18 | 11.0 | 11.9 | 12.9 | 13.7 | 14.5 | 15.3 | 16.0 | 16.7 | 17.3 | 17.9 | 18.5 | 19.1 | 19.7 | | | | | | | | 18 |
| 19 | 11.2 | 12.2 | 13.2 | 14.1 | 14.9 | 15.7 | 16.4 | 17.1 | 17.8 | 18.4 | 19.0 | 19.6 | 20.2 | 20.8 | | | | | | | 19 |
| 20 | 11.5 | 12.5 | 13.5 | 14.4 | 15.2 | 16.0 | 16.8 | 17.5 | 18.2 | 18.9 | 19.5 | 20.1 | 20.7 | 21.3 | 21.9 | | | | | | 20 |
| 22 | 12.0 | 13.0 | 14.1 | 15.0 | 15.9 | 16.8 | 17.6 | 18.3 | 19.1 | 19.8 | 20.4 | 21.1 | 21.7 | 22.3 | 22.9 | 24.0 | | | | | 22 |
| 24 | 12.4 | 13.5 | 14.6 | 15.6 | 16.5 | 17.4 | 18.3 | 19.1 | 19.9 | 20.6 | 21.3 | 22.0 | 22.7 | 23.3 | 23.9 | 25.1 | 26.2 | | | | 24 |
| 26 | 12.8 | 14.0 | 15.1 | 16.2 | 17.1 | 18.1 | 19.0 | 19.8 | 20.6 | 214 | 22.1 | 22.9 | 23.5 | 24.2 | 24.9 | 26.1 | 27.3 | 28.4 | | | 26 |
| 28 | 13.2 | 14.5 | 15.6 | 16.7 | 17.7 | 18.7 | 19.6 | 20.5 | 21.3 | 22.1 | 22.9 | 23.7 | 24.4 | 25.1 | 25.8 | 27.1 | 28.3 | 29.5 | 30.6 | | 28 |
| 30 | 13.6 | 14.9 | 16.1 | 17.2 | 18.3 | 19.3 | 20.2 | 21.1 | 22.0 | 22.9 | 23.7 | 24.4 | 25.2 | 25.9 | 26.6 | 28.0 | 29.3 | 30.5 | 31.7 | 32.8 | 30 |
| 32 | 14.0 | 15.3 | 16.5 | 17.7 | 18.8 | 19.8 | 20.8 | 21.8 | 22.7 | 23.5 | 24.4 | 25.2 | 26.0 | 26.7 | 27.5 | 28.9 | 30.2 | 31.5 | 32.7 | 33.9 | 32 |
| 34 | 14.4 | 15.7 | 17.0 | 18.2 | 19.3 | 20.4 | 21.4 | 22.4 | 23.3 | 24.2 | 25.1 | 25.9 | 26.7 | 27.5 | 28.3 | 29.7 | 31.0 | 32.4 | 33.7 | 34.9 | 34 |
| 36 | 14.7 | 16.1 | 17.4 | 18.6 | 19.8 | 20.9 | 21.9 | 22.9 | 23.9 | 24.8 | 25.7 | 26.6 | 27.4 | 28.2 | 29.0 | 30.5 | 32.0 | 33.3 | 34.6 | 35.9 | 36 |
| 38 | 15.0 | 16.5 | 17.8 | 19.0 | 20.2 | 21.4 | 22.4 | 23.5 | 24.5 | 25.4 | 26.4 | 27.2 | 28.1 | 28.9 | 29.8 | 31.3 | 32.8 | 34.2 | 35.6 | 36.8 | 38 |
| 40 | 15.3 | 16.8 | 18.2 | 19.5 | 20.7 | 21.8 | 22.9 | 24.0 | 25.0 | 26.0 | 27.0 | 27.9 | 28.8 | 29.6 | 30.5 | 32.1 | 33.6 | 35.1 | 36.4 | 37.8 | 40 |
| 42 | 15.6 | 17.1 | 18.5 | 19.9 | 21.1 | 22.3 | 23.4 | 24.5 | 25.6 | 26.6 | 27.6 | 28.5 | 29.4 | 30.3 | 31.2 | 32.8 | 34.4 | 35.9 | 37.3 | 38.7 | 42 |
| 44 | 15.9 | 17.5 | 18.9 | 20.3 | 21.5 | 22.7 | 23.9 | 25.0 | 26.1 | 27.1 | 28.1 | 29.1 | 30.0 | 30.9 | 31.8 | 33.5 | 35.1 | 36.7 | 38.1 | 39.5 | 44 |
| 46 | 16.2 | 17.8 | 19.3 | 20.6 | 21.9 | 23.2 | 24.4 | 25.5 | 26.6 | 27.7 | 28.7 | 29.7 | 30.6 | 31.6 | 32.5 | 34.2 | 35.9 | 37.4 | 38.9 | 40.4 | 46 |
| 48 | 16.5 | 18.1 | 19.6 | 21.0 | 22.3 | 23.6 | 24.8 | 26.0 | 27.1 | 28.2 | 29.2 | 30.2 | 31.2 | 32.2 | 33.1 | 34.9 | 36.6 | 38.2 | 39.7 | 41.2 | 48 |
| 50 | 16.8 | 18.4 | 19.9 | 21.4 | 22.7 | 24.0 | 25.2 | 26.4 | 27.6 | 28.7 | 29.8 | 30.8 | 31.8 | 32.8 | 33.7 | 35.5 | 37.2 | 38.9 | 40.5 | 42.0 | 50 |
| 52 | 17.1 | 18.7 | 20.2 | 21.7 | 23.1 | 24.4 | 25.7 | 26.9 | 28.0 | 29.2 | 30.3 | 31.3 | 32.3 | 33.3 | 34.3 | 36.2 | 37.9 | 39.6 | 41.2 | 42.8 | 52 |
| 54 | 17.3 | 19.0 | 20.6 | 22.0 | 23.5 | 24.8 | 26.1 | 27.3 | 28.5 | 29.7 | 30.8 | 31.8 | 32.9 | 33.9 | 34.9 | 36.8 | 38.6 | 40.3 | 41.9 | 43.5 | 54 |
| 56 | 17.6 | 19.3 | 20.9 | 22.4 | 23.8 | 25.2 | 26.5 | 27.7 | 28.9 | 30.1 | 31.2 | 32.3 | 33.4 | 34.4 | 35.4 | 37.4 | 39.2 | 41.0 | 42.7 | 44.3 | 56 |
| 58 | 17.8 | 19.5 | 21.2 | 22.7 | 24.2 | 25.5 | 26.9 | 28.2 | 29.4 | 30.6 | 31.7 | 32.8 | 33.9 | 35.0 | 36.0 | 38.0 | 39.8 | 41.6 | 43.3 | 45.0 | 58 |
| 60 | 18.1 | 19.8 | 21.5 | 23.0 | 24.5 | 25.9 | 27.3 | 28.6 | 29.8 | 31.0 | 32.2 | 33.3 | 34.4 | 35.5 | 36.5 | 38.5 | 40.4 | 42.3 | 44.0 | 45.7 | 60 |
| 62 | | 20.1 | 21.7 | 23.3 | 24.8 | 26.3 | 27.6 | 28.9 | 30.2 | 31.5 | 32.6 | 33.8 | 34.9 | 36.0 | 37.1 | 39.1 | 41.0 | 42.9 | 44.7 | 46.4 | 62 |
| 64 | | 20.3 | 22.0 | 23.6 | 25.1 | 26.6 | 28.0 | 29.3 | 30.6 | 31.9 | 33.1 | 34.3 | 35.4 | 36.5 | 37.6 | 39.6 | 41.6 | 43.5 | 45.3 | 47.1 | 64 |
| 66 | | 20.6 | 22.3 | 23.9 | 25.5 | 26.9 | 28.4 | 29.7 | 31.0 | 32.3 | 33.5 | 34.7 | 35.9 | 37.0 | 38.1 | 40.2 | 42.2 | 44.1 | 46.0 | 47.7 | 66 |
| 68 | | 20.8 | 22.6 | 24.2 | 25.8 | 27.3 | 28.7 | 30.1 | 31.4 | 32.7 | 33.9 | 35.2 | 36.3 | 37.5 | 38.6 | 40.7 | 42.8 | 44.7 | 46.6 | 48.4 | 68 |
| 70 | | 21.1 | 22.6 | 24.5 | 26.1 | 27.6 | 29.1 | 30.4 | 31.8 | 33.1 | 34.4 | 35.6 | 36.8 | 37.9 | 39.1 | 41.2 | 43.3 | 45.3 | 47.2 | 49.0 | 70 |
| 72 | | | 23.1 | 24.8 | 26.4 | 27.9 | 29.4 | 30.8 | 32.2 | 33.5 | 34.8 | 36.0 | 37.2 | 38.4 | 39.5 | 41.7 | 43.8 | 45.8 | 47.8 | 49.6 | 72 |
| 74 | | | 23.3 | 25.1 | 26.7 | 28.2 | 29.7 | 31.2 | 32.5 | 33.9 | 35.2 | 36.4 | 37.7 | 38.8 | 40.0 | 42.2 | 44.4 | 46.4 | 48.4 | 50.3 | 74 |
| 76 | | | 23.6 | 25.3 | 27.0 | 28.5 | 30.0 | 31.5 | 32.9 | 34.3 | 35.6 | 36.8 | 38.1 | 39.3 | 40.5 | 42.7 | 44.9 | 47.0 | 48.9 | 50.9 | 76 |
| 78 | | | 23.8 | 25.6 | 27.3 | 28.8 | 30.4 | 31.8 | 33.3 | 34.6 | 36.0 | 37.2 | 38.5 | 39.7 | 40.9 | 43.2 | 45.4 | 47.5 | 49.5 | 51.4 | 78 |
| 80 | | | 24.1 | 25.8 | 27.5 | 29.1 | 30.7 | 32.2 | 33.6 | 35.0 | 36.3 | 37.6 | 38.9 | 40.2 | 41.4 | 43.7 | 45.9 | 48.0 | 50.1 | 52.0 | 80 |
| 82 | | | | 26.1 | 27.8 | 29.4 | 31.0 | 32.5 | 34.0 | 35.4 | 36.7 | 38.0 | 39.3 | 40.6 | 41.8 | 44.1 | 46.4 | 48.5 | 50.6 | 52.6 | 82 |
| 84 | | | | 26.4 | 28.1 | 29.7 | 31.3 | 32.8 | 34.3 | 35.7 | 37.1 | 38.4 | 39.7 | 41.0 | 42.2 | 44.6 | 46.9 | 49.0 | 51.1 | 53.2 | 84 |
| 86 | | | | 26.6 | 28.3 | 30.0 | 31.6 | 33.1 | 34.6 | 36.1 | 37.4 | 38.8 | 40.1 | 41.4 | 42.6 | 45.0 | 47.3 | 49.6 | 51.7 | 53.7 | 86 |
| 88 | | | | 26.9 | 28.6 | 30.3 | 31.9 | 33.4 | 34.9 | 36.4 | 37.8 | 39.2 | 40.5 | 41.8 | 43.1 | 45.5 | 47.8 | 50.0 | 52.2 | 54.3 | 88 |
| 90 | | | | 27.1 | 28.9 | 30.6 | 32.2 | 33.8 | 35.3 | 36.7 | 38.2 | 39.5 | 40.9 | 42.2 | 43.5 | 45.9 | 48.3 | 50.5 | 52.7 | 54.8 | 90 |
| 92 | | | | | 29.1 | 30.8 | 32.5 | 34.1 | 35.6 | 37.1 | 38.5 | 39.9 | 41.3 | 42.6 | 43.9 | 46.4 | 48.7 | 51.0 | 53.2 | 55.3 | 92 |
| 96 | | | | | 29.6 | 31.4 | 33.0 | 34.7 | 36.2 | 37.7 | 39.2 | 40.6 | 42.0 | 43.3 | 44.7 | 47.2 | 49.6 | 52.0 | 54.2 | 56.4 | 96 |

## Table A–2 Circulation Equivalents of Rectangular Ducts for Equal Friction and Capacity Dimensions (I–P)

DRAFT

| Side Retan-gular Duct | 32 | 34 | 36 | 38 | 40 | 42 | 44 | 46 | 48 | 50 | 52 | 56 | 60 | 64 | 68 | 72 | 76 | 80 | 84 | 88 | Side-Retan-gular-Duct |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 32 | 35.0 | | | | | | | | | | | | | | | | | | | | 32 |
| 34 | 36.1 | 37.2 | | | | | | | | | | | | | | | | | | | 34 |
| 36 | 37.1 | 38.2 | 39.4 | | | | | | | | | | | | | | | | | | 36 |
| 38 | 38.1 | 39.3 | 40.4 | 41.5 | | | | | | | | | | | | | | | | | 38 |
| 40 | 39.0 | 40.3 | 41.5 | 42.6 | 43.7 | | | | | | | | | | | | | | | | 40 |
| 42 | 40.0 | 41.3 | 42.5 | 43.7 | 44.8 | 45.9 | | | | | | | | | | | | | | | 42 |
| 44 | 40.9 | 42.2 | 43.5 | 44.7 | 45.8 | 47.0 | 48.1 | | | | | | | | | | | | | | 44 |
| 46 | 41.8 | 43.1 | 44.4 | 45.7 | 46.9 | 48.0 | 49.2 | 50.3 | | | | | | | | | | | | | 46 |
| 48 | 42.6 | 44.0 | 45.3 | 46.6 | 47.9 | 49.1 | 50.2 | 51.4 | 52.5 | | | | | | | | | | | | 48 |
| 50 | 43.6 | 44.9 | 46.2 | 47.5 | 48.8 | 50.0 | 51.2 | 52.4 | 53.6 | 54.7 | | | | | | | | | | | 50 |
| 52 | 44.3 | 45.7 | 47.1 | 48.4 | 49.7 | 51.0 | 52.2 | 53.4 | 54.6 | 55.7 | 56.8 | | | | | | | | | | 52 |
| 54 | 45.1 | 46.5 | 48.0 | 49.3 | 50.7 | 52.0 | 53.2 | 54.4 | 55.6 | 56.8 | 57.9 | | | | | | | | | | 54 |
| 56 | 45.8 | 47.3 | 48.8 | 50.2 | 51.6 | 52.9 | 54.2 | 55.4 | 56.6 | 57.8 | 59.0 | 61.2 | | | | | | | | | 56 |
| 58 | 46.6 | 48.1 | 49.6 | 51.0 | 52.4 | 53.8 | 55.1 | 56.4 | 57.6 | 58.8 | 60.0 | 62.3 | | | | | | | | | 58 |
| 60 | 47.3 | 48.9 | 50.4 | 51.9 | 53.3 | 54.7 | 60.0 | 57.3 | 58.6 | 59.8 | 61.0 | 63.4 | 65.6 | | | | | | | | 60 |
| 62 | 48.0 | 49.6 | 51.2 | 52.7 | 54.1 | 55.5 | 56.9 | 58.2 | 59.5 | 60.8 | 62.0 | 64.4 | 66.7 | | | | | | | | 62 |
| 64 | 48.7 | 50.4 | 51.9 | 53.5 | 54.9 | 56.4 | 57.8 | 59.1 | 60.4 | 61.7 | 63.0 | 65.4 | 67.7 | 70.0 | | | | | | | 64 |
| 66 | 49.4 | 51.1 | 52.7 | 54.2 | 55.7 | 57.2 | 58.6 | 60.0 | 61.3 | 62.6 | 63.9 | 66.4 | 68.8 | 71.0 | | | | | | | 66 |
| 68 | 50.1 | 51.8 | 53.4 | 55.0 | 56.5 | 58.0 | 59.4 | 60.8 | 62.2 | 63.6 | 64.9 | 67.4 | 69.8 | 72.1 | 74.3 | | | | | | 68 |
| 70 | 50.8 | 52.5 | 54.1 | 55.7 | 57.3 | 58.8 | 60.3 | 61.7 | 63.1 | 64.4 | 65.8 | 68.3 | 70.8 | 73.2 | 75.4 | | | | | | 70 |
| 72 | 51.4 | 53.2 | 54.8 | 56.5 | 58.0 | 59.6 | 61.1 | 62.5 | 63.9 | 65.3 | 66.7 | 69.3 | 71.8 | 74.2 | 76.5 | 78.7 | | | | | 72 |
| 74 | 52.1 | 53.8 | 55.5 | 57.2 | 58.8 | 60.3 | 61.9 | 63.3 | 64.8 | 66.2 | 67.5 | 70.2 | 72.7 | 75.2 | 77.5 | 79.8 | | | | | 74 |
| 76 | 52.7 | 54.5 | 56.2 | 57.9 | 59.5 | 61.1 | 62.6 | 64.1 | 65.6 | 67.0 | 68.4 | 71.1 | 73.7 | 76.2 | 78.6 | 80.9 | 83.1 | | | | 76 |
| 78 | 53.3 | 55.1 | 56.9 | 58.6 | 60.2 | 61.8 | 63.4 | 64.9 | 66.4 | 67.9 | 69.3 | 72.0 | 74.6 | 77.1 | 79.6 | 81.9 | 84.2 | | | | 78 |
| 80 | 539 | 55.8 | 57.5 | 59.3 | 60.9 | 62.6 | 64.1 | 65.7 | 67.2 | 68.7 | 70.1 | 72.9 | 75.4 | 78.1 | 80.6 | 82.9 | 85.2 | 87.5 | | | 80 |
| 82 | 54.5 | 56.4 | 58.2 | 59.9 | 61.6 | 63.3 | 64.9 | 66.5 | 68.0 | 69.5 | 70.9 | 73.7 | 76.4 | 79.0 | 81.5 | 84.0 | 86.3 | 88.5 | | | 82 |
| 84 | 55.1 | 57.0 | 58.8 | 60.6 | 62.3 | 64.0 | 65.6 | 67.2 | 68.7 | 70.3 | 71.7 | 74.6 | 77.3 | 80.0 | 82.5 | 85.0 | 87.3 | 89.6 | 91.8 | | 84 |
| 86 | 55.7 | 57.6 | 59.4 | 61.2 | 63.0 | 64.7 | 66.3 | 67.9 | 69.5 | 71.0 | 72.5 | 75.4 | 78.2 | 80.9 | 83.5 | 85.9 | 88.3 | 90.7 | 92.9 | | 86 |
| 88 | 56.3 | 58.2 | 60.1 | 61.9 | 63.6 | 65.4 | 67.0 | 68.7 | 70.2 | 71.8 | 73.3 | 76.3 | 79.1 | 81.8 | 84.4 | 86.9 | 89.3 | 91.7 | 94.0 | 96.2 | 88 |
| 90 | 56.8 | 58.8 | 60.7 | 62.5 | 64.3 | 66.0 | 67.7 | 69.4 | 71.0 | 72.6 | 74.1 | 77.1 | 79.9 | 82.7 | 85.3 | 87.9 | 90.3 | 92.7 | 95.0 | 97.3 | 90 |
| 92 | 57.4 | 59.3 | 61.3 | 63.1 | 64.9 | 66.7 | 68.4 | 70.1 | 71.7 | 73.3 | 74.9 | 77.9 | 80.8 | 83.5 | 86.2 | 88.8 | 91.3 | 93.7 | 96.1 | 98.4 | 92 |
| 94 | 57.9 | 59.9 | 61.9 | 63.7 | 65.6 | 67.3 | 69.1 | 70.8 | 72.4 | 74.0 | 75.6 | 78.7 | 81.6 | 84.4 | 87.1 | 89.7 | 92.3 | 94.7 | 97.1 | 99.4 | 94 |
| 96 | 58A | 60.5 | 62.4 | 64.3 | 66.2 | 68.0 | 69.7 | 71.5 | 73.1 | 74.8 | 76.3 | 79.4 | 82.4 | 85.3 | 88.0 | 90.7 | 93.2 | 95.7 | 981 | 100.5 | 96 |

## Table A–2 Circulation Equivalents of Rectangular Ducts for Equal Friction and Capacity Dimensions (I–P) (continued)

Equation for Circular Equivalent of a Rectangular Duct:

$$D_e = 1.30\left[\frac{(ab)^{0.625}}{(a+b)^{0.250}}\right]$$

where

$a$ = length of one side of rectangular duct, inches.
$b$ = length of adjacent side of rectangular duct, inches.
$D_e$ = circular equivalent of rectangular duct for equal friction and capacity, inches.

DRAFT

| Side Rectangular-Duct | 100 | 125 | 150 | 175 | 200 | 225 | 250 | 275 | 300 | 350 | 400 | 450 | 500 | 550 | 600 | 650 | 700 | 750 | 800 | 900 | Side Rectangular Duct |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 100 | 109 | | | | | | | | | | | | | | | | | | | | 100 |
| 125 | 122 | 137 | | | | | | | | | | | | | | | | | | | 125 |
| 150 | 133 | 150 | 164 | | | | | | | | | | | | | | | | | | 150 |
| 175 | 143 | 161 | 177 | 191 | | | | | | | | | | | | | | | | | 175 |
| 200 | 152 | 172 | 189 | 204 | 219 | | | | | | | | | | | | | | | | 200 |
| 225 | 161 | 181 | 200 | 216 | 232 | 246 | | | | | | | | | | | | | | | 225 |
| 250 | 169 | 190 | 210 | 228 | 244 | 259 | 273 | | | | | | | | | | | | | | 250 |
| 275 | 176 | 199 | 220 | 238 | 256 | 272 | 287 | 301 | | | | | | | | | | | | | 275 |
| 300 | 183 | 207 | 229 | 248 | 266 | 283 | 299 | 314 | 328 | | | | | | | | | | | | 300 |
| 350 | 195 | 222 | 245 | 267 | 286 | 305 | 322 | 339 | 354 | 383 | | | | | | | | | | | 350 |
| 400 | 207 | 235 | 260 | 283 | 305 | 325 | 343 | 361 | 378 | 409 | 437 | | | | | | | | | | 400 |
| 450 | 217 | 247 | 274 | 299 | 321 | 343 | 363 | 382 | 400 | 433 | 464 | 492 | | | | | | | | | 450 |
| 500 | 227 | 258 | 287 | 313 | 337 | 360 | 381 | 401 | 420 | 455 | 488 | 518 | 547 | | | | | | | | 500 |
| 550 | 236 | 269 | 299 | 326 | 352 | 375 | 398 | 419 | 439 | 477 | 511 | 543 | 573 | 601 | | | | | | | 550 |
| 600 | 245 | 279 | 310 | 339 | 365 | 390 | 414 | 436 | 457 | 496 | 533 | 567 | 598 | 628 | 656 | | | | | | 600 |
| 650 | 253 | 289 | 321 | 351 | 378 | 404 | 429 | 452 | 474 | 515 | 553 | 589 | 622 | 653 | 683 | 711 | | | | | 650 |
| 700 | 261 | 298 | 331 | 362 | 391 | 418 | 443 | 467 | 490 | 533 | 573 | 610 | 644 | 677 | 708 | 737 | 765 | | | | 700 |
| 750 | 268 | 306 | 341 | 373 | 402 | 430 | 457 | 482 | 506 | 550 | 592 | 630 | 666 | 700 | 732 | 763 | 792 | 820 | | | 750 |
| 800 | 275 | 314 | 350 | 383 | 414 | 442 | 470 | 496 | 520 | 567 | 609 | 649 | 687 | 722 | 755 | 787 | 818 | 847 | 875 | | 800 |
| 900 | 289 | 330 | 367 | 402 | 435 | 465 | 494 | 522 | 548 | 597 | 643 | 686 | 726 | 763 | 799 | 833 | 866 | 897 | 927 | 984 | 900 |
| 1000 | 301 | 344 | 384 | 420 | 454 | 486 | 517 | 546 | 574 | 626 | 674 | 719 | 762 | 802 | 840 | 876 | 911 | 941 | 976 | 1037 | 1000 |
| 1100 | 313 | 358 | 399 | 437 | 473 | 506 | 538 | 569 | 598 | 652 | 703 | 751 | 795 | 838 | 878 | 916 | 953 | 988 | 1022 | 1086 | 1100 |
| 1200 | 324 | 370 | 413 | 453 | 490 | 525 | 558 | 590 | 620 | 677 | 731 | 780 | 827 | 872 | 914 | 954 | 993 | 1030 | 1066 | 1133 | 1200 |
| 1300 | 334 | 382 | 426 | 468 | 506 | 543 | 577 | 610 | 642 | 701 | 757 | 808 | 857 | 904 | 948 | 990 | 1031 | 1069 | 1107 | 1177 | 1300 |
| 1400 | 344 | 394 | 439 | 482 | 522 | 559 | 595 | 629 | 662 | 724 | 781 | 835 | 886 | 934 | 980 | 1024 | 1066 | 1077 | 1146 | 1220 | 1400 |
| 1500 | 353 | 404 | 452 | 495 | 536 | 575 | 612 | 648 | 681 | 745 | 805 | 860 | 913 | 963 | 1011 | 1057 | 1100 | 1143 | 1183 | 1260 | 1500 |
| 1600 | 362 | 415 | 463 | 508 | 551 | 591 | 629 | 665 | 700 | 766 | 827 | 885 | 939 | 991 | 1041 | 1088 | 1133 | 1177 | 1219 | 1298 | 1600 |
| 1700 | 371 | 425 | 475 | 521 | 564 | 605 | 644 | 682 | 718 | 785 | 849 | 908 | 964 | 1018 | 1069 | 1118 | 1164 | 1209 | 1253 | 1335 | 1700 |
| 1800 | 379 | 434 | 485 | 533 | 577 | 619 | 660 | 698 | 735 | 804 | 869 | 930 | 988 | 1043 | 1096 | 1146 | 1195 | 1241 | 1286 | 1371 | 1800 |
| 1900 | 387 | 444 | 496 | 544 | 590 | 633 | 674 | 713 | 751 | 823 | 889 | 952 | 1012 | 1068 | 1122 | 1174 | 1224 | 1271 | 1318 | 1405 | 1900 |
| 2000 | 395 | 453 | 506 | 555 | 602 | 646 | 688 | 728 | 767 | 840 | 908 | 973 | 1034 | 1092 | 1147 | 1200 | 1252 | 1301 | 1348 | 1438 | 2000 |
| 2100 | 402 | 461 | 516 | 566 | 614 | 659 | 702 | 743 | 782 | 857 | 927 | 993 | 1055 | 1115 | 1172 | 1226 | 1279 | 1329 | 1378 | 1470 | 2100 |
| 2200 | 410 | 470 | 525 | 577 | 625 | 671 | 715 | 757 | 797 | 874 | 945 | 1013 | 1076 | 1137 | 1195 | 1251 | 1305 | 1356 | 1406 | 1501 | 2200 |
| 2300 | 417 | 478 | 534 | 587 | 636 | 683 | 728 | 771 | 812 | 890 | 963 | 1031 | 1097 | 1159 | 1218 | 1275 | 1330 | 1383 | 1434 | 1532 | 2300 |
| 2400 | 424 | 486 | 543 | 597 | 647 | 695 | 740 | 784 | 826 | 905 | 980 | 1050 | 1116 | 1180 | 1241 | 1299 | 1355 | 1409 | 1461 | 1561 | 2400 |
| 2500 | 430 | 494 | 552 | 606 | 658 | 706 | 753 | 797 | 840 | 920 | 996 | 1068 | 1136 | 1200 | 1262 | 1322 | 1379 | 1434 | 1488 | 1589 | 2500 |
| 2600 | 437 | 501 | 560 | 616 | 668 | 717 | 764 | 810 | 853 | 935 | 1012 | 1085 | 1154 | 1220 | 1283 | 1344 | 1402 | 1459 | 1513 | 1617 | 2600 |
| 2700 | 443 | 509 | 569 | 625 | 678 | 728 | 776 | 822 | 866 | 950 | 1028 | 1102 | 1173 | 1240 | 1304 | 1366 | 1425 | 1483 | 1538 | 1644 | 2700 |
| 2800 | 450 | 516 | 577 | 634 | 688 | 738 | 787 | 834 | 879 | 964 | 1043 | 1119 | 1190 | 1259 | 1324 | 1387 | 1447 | 1506 | 1562 | 1670 | 2800 |
| 2900 | 456 | 523 | 585 | 643 | 697 | 749 | 798 | 845 | 891 | 977 | 1058 | 1135 | 1208 | 1277 | 1344 | 1408 | 1469 | 1529 | 1586 | 1696 | 2900 |
| Side Rectangular Duct | 100 | 125 | 150 | 175 | 200 | 225 | 250 | 275 | 300 | 350 | 400 | 450 | 500 | 550 | 600 | 650 | 700 | 750 | 800 | 900 | Side Rectangular Duct |

## Table A–2M Circular Equivalents of Rectangular Ducts for Equal Friction and Capacity Dimensions (SI)

DRAFT

| Side Rectangular Duct | 1000 | 1100 | 1200 | 1300 | 1400 | 1500 | 1600 | 1700 | 1800 | 1900 | 2000 | 2100 | 2200 | 2300 | 2400 | 2500 | 2600 | 2700 | 2800 | 2900 | Side Rectangular Duct |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1000 | 1093 | | | | | | | | | | | | | | | | | | | | 1000 |
| 1100 | 1143 | 1202 | | | | | | | | | | | | | | | | | | | 1100 |
| 1200 | 1196 | 1256 | 1312 | | | | | | | | | | | | | | | | | | 1200 |
| 1300 | 1244 | 1306 | 1365 | 1421 | | | | | | | | | | | | | | | | | 1300 |
| 1400 | 1289 | 1354 | 1416 | 1530 | | | | | | | | | | | | | | | | | 1400 |
| 1500 | 1332 | 1400 | 1464 | 1526 | 1584 | 1640 | | | | | | | | | | | | | | | 1500 |
| 1600 | 1373 | 1444 | 1511 | 1574 | 1635 | 1693 | 1749 | | | | | | | | | | | | | | 1600 |
| 1700 | 1413 | 1486 | 1555 | 1621 | 1684 | 1745 | 1803 | 1858 | | | | | | | | | | | | | 1700 |
| 1800 | 1451 | 1527 | 1598 | 1667 | 1732 | 1794 | 1854 | 1912 | 1968 | | | | | | | | | | | | 1800 |
| 1900 | 1488 | 1566 | 1640 | 1710 | 1778 | 1842 | 1904 | 1964 | 2021 | 2077 | | | | | | | | | | | 1900 |
| 2000 | 1523 | 1604 | 1680 | 1753 | 1822 | 1889 | 1952 | 2014 | 2073 | 2131 | 2186 | | | | | | | | | | 2000 |
| 2100 | 1558 | 1640 | 1719 | 1793 | 1865 | 1933 | 1999 | 2063 | 2124 | 2183 | 2240 | 2296 | | | | | | | | | 2100 |
| 2200 | 1591 | 1676 | 1756 | 1833 | 1906 | 1977 | 2044 | 2110 | 2173 | 2233 | 2292 | 2350 | 2405 | | | | | | | | 2200 |
| 2300 | 1623 | 1710 | 1793 | 1871 | 1947 | 2019 | 2088 | 2155 | 2220 | 2283 | 2343 | 2402 | 2459 | 2514 | | | | | | | 2300 |
| 2400 | 1655 | 1744 | 1828 | 1909 | 1986 | 2060 | 2131 | 2200 | 2266 | 2330 | 2393 | 2453 | 2511 | 2568 | 2624 | | | | | | 2400 |
| 2500 | 1685 | 1776 | 1862 | 1945 | 2024 | 2100 | 2173 | 2243 | 2311 | 2377 | 2441 | 2502 | 2562 | 2621 | 2678 | 2733 | | | | | 2500 |
| 2600 | 1715 | 1808 | 1896 | 1980 | 2061 | 2139 | 2213 | 2285 | 2355 | 2422 | 2487 | 2551 | 2612 | 2672 | 2730 | 2787 | 2842 | | | | 2600 |
| 2700 | 1744 | 1839 | 1929 | 2015 | 2097 | 2177 | 2253 | 2327 | 2398 | 2466 | 2533 | 2598 | 2661 | 2722 | 2782 | 2840 | 2896 | 2952 | | | 2700 |
| 2800 | 1772 | 1869 | 1961 | 2048 | 2133 | 2214 | 2292 | 2367 | 2439 | 2510 | 2578 | 2644 | 2708 | 2771 | 2832 | 2891 | 2949 | 3006 | 3061 | | 2800 |
| 2900 | 1800 | 1898 | 1992 | 2081 | 2167 | 2250 | 2329 | 2406 | 2480 | 2552 | 2621 | 2689 | 2755 | 2819 | 2881 | 2941 | 3001 | 3058 | 3115 | 3170 | 2900 |
| Side Rectangular Duct | 1000 | 1100 | 1200 | 1300 | 1400 | 1500 | 1600 | 1700 | 1800 | 1900 | 2000 | 2100 | 2200 | 2300 | 2400 | 2500 | 2600 | 2700 | 2800 | 2900 | Side Rectangular Duct |

## Table A–2M Circular Equivalents of Rectangular Ducts for Equal Friction and Capacity Dimensions (SI) (continued)

Equation for Circular Equivalent of a Rectangular Duct:

$$D_e = 1.30\left[\frac{(ab)^{0.625}}{(a+b)^{0.250}}\right]$$

where

$a$ = length of one side of rectangular duct, mm.
$b$ = length of adjacent side of rectangular duct, mm.
$D_e$ = circular equivalent of rectangular duct for equal friction and capacity, mm.

DRAFT

| | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 14 | 16 | 18 | 20 | 22 | 24 | 26 | 28 | 30 | 32 | 34 | 36 | 38 | 40 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 7 | | 5.7 | | | | | | | | | | | | | | | | | | | | | | |
| 8 | 5.1 | | 6.6 | 6.9 | | | | | | | | | | | | | | | | | | | | |
| 9 | 5.6 | 6.2 | | 7.7 | | | | | | | | | | | | | | | | | | | | |
| 10 | | 6.7 | 7.3 | | 8.7 | 9.0 | | | | | | | | | | | | | | | | | | |
| 11 | 6.0 | | 7.9 | 8.4 | | 9.8 | | | | | | | | | | | | | | | | | | |
| 12 | 6.4 | 7.2 | | 8.9 | 9.4 | | 10.8 | 11.0 | | | | | | | | | | | | | | | | |
| 13 | | 7.6 | 8.4 | | 10.1 | 10.6 | | 11.9 | | | | | | | | | | | | | | | | |
| 14 | 6.7 | | 8.8 | 9.6 | | 11.2 | 11.5 | | | | | | | | | | | | | | | | | |
| 15 | 7.0 | 8.0 | | 10.1 | 10.7 | | | | | 13.8 | | | | | | | | | | | | | | |
| 16 | | | 9.3 | | | | | 13.4 | 13.6 | | | | | | | | | | | | | | | |
| 17 | 7.3 | 8.4 | | | | | 12.9 | | | | | | | | | | | | | | | | | |
| 18 | | | | | 11.7 | 12.4 | | | | 15.3 | | | | | | | | | | | | | | |
| 19 | | | 10.0 | 11.0 | | | | | 15.0 | | | | | | | | | | | | | | | |
| 20 | | | | | | | 14.0 | 14.1 | | | 17.5 | | | | | | | | | | | | | |
| 21 | | | | | 12.6 | 13.5 | | | | 16.7 | | | 19.9 | | | | | | | | | | | |
| 22 | | | | 11.8 | | | | | 16.3 | | | 19.5 | | | | | | | | | | | | |
| 23 | | | | | | | 15.1 | 15.7 | | | 18.9 | | | | | | | | | | | | | |
| 24 | | | | | | 14.4 | | | | | | | 21.6 | | | | | | | | | | | |
| 25 | | | | 12.5 | | | | | | 18.0 | | 20.9 | | | 23.9 | | | | | | | | | |
| 26 | | | | | | | | 16.7 | | | | | | 23.6 | | | | | | | | | | |
| 27 | | | | | | 15.2 | | | | | 20.2 | | 23.1 | | | 25.9 | | | | | | | | |
| 28 | | | | 13.2 | | | | | | | | | | | 25.6 | | | | | | | | | |
| 29 | | | | | | | | 17.7 | | | | 22.3 | | 25.2 | | | 27.9 | | | | | | | |
| 30 | | | | | | 15.9 | | | | | 21.3 | | | | | 28.1 | | | | | | | | |
| 31 | | | | 13.8 | | | | | | 20.1 | | | 24.5 | | 27.2 | | | 29 9 | | | | | | |
| 32 | | | | | | | | 18.5 | | | | 23.5 | | | | | 29.7 | | | | | | | |
| 33 | | | | | | 18.6 | | | | | 22.4 | | | 26.6 | | 29.3 | | | 32.0 | | | | | |
| 34 | | | | 14.3 | | | | | | 20.9 | | | 25.7 | | | | | 31.7 | | | | | | |
| 35 | | | | | | | | 19.3 | | | | 24.7 | | | 28.7 | | 31.3 | | | 34.0 | | | | |
| 36 | | | | | | 17.3 | | | | | 23.4 | | | 27.9 | | | | | 33.7 | | | | | |
| 37 | | | | 14.9 | | | | | | 21.9 | | | 27.0 | | | 30.8 | | 33.4 | | | 36.0 | | | |
| 38 | | | | | | | | 20.1 | | | | 25.7 | | | 30.0 | | | | | 35.8 | | | | |
| 39 | | | | | | 17.9 | | | | | 24.4 | | | 29.2 | | | 32.8 | | 35.4 | | | 38.0 | | |
| 40 | | | | | | | | | | 22.7 | | | 28.1 | | | 32.2 | | | | | 37.8 | | | |
| 41 | | | | 15.4 | | | | 20.8 | | | | 26.8 | | | 31.3 | | | 34.9 | | 37.4 | | | 40.0 | |
| 42 | | | | | | | | | | | | | | | | | | | | | | | | |
| 43 | | | | | | 18.6 | | | | 23.5 | | | 29.1 | | | 33.5 | | | 37.0 | | 39.5 | | | 42.0 |
| 44 | | | | 15.9 | | | | | | | | 27.7 | | | 32.5 | | | 36.4 | | | | | 41.8 | |
| 45 | | | | | | | | 21.5 | | | 26.1 | | | 31.4 | | | 35.6 | | | 39.0 | | 41.5 | | |
| 46 | | | | | | 19.1 | | | | | | | 30.2 | | | 34.7 | | | 38.5 | | | | | 43.8 |
| 47 | | | | | | | | | | 24.3 | | 28.6 | | | 33.7 | | | 37.8 | | | 41.1 | | 43.5 | |
| 48 | | | | | | | | 22.1 | | | | | | 32.5 | | | 36.9 | | | 40.5 | | | | |
| 49 | | | | | | 19.6 | | | | | 26.9 | | 31.1 | | | 35.9 | | | 39.9 | | | 43.1 | | 45.6 |
| 50 | | | | | | | | | | 25.0 | | | | | 34.8 | | | 39.1 | | | 42.6 | | | |
| 51 | | | | | | | | 22.8 | | | | 29.4 | | 33.4 | | | 38.1 | | | 42.0 | | | 45.2 | |
| 52 | | | | | | 20.2 | | | | | 27.7 | | | | | 37.0 | | | 41.2 | | | 44.7 | | |
| 53 | | | | | | | | | | 25.7 | | | 32.0 | | 35.8 | | | 40.3 | | | 44.1 | | | 47.2 |
| 54 | | | | | | | | 23 3 | | | | 30.2 | | | | | 39.3 | | | 43.3 | | | 46.7 | |
| 55 | | | | | | | | | | | 28.4 | | | 34.4 | | 38.1 | | | 42.5 | | | 46.2 | | |
| 56 | | | | | | | | | | 26.3 | | | 32.9 | | | | | 41.5 | | | 45.5 | | | 48.8 |
| 57 | | | | | | | | 23.8 | | | | 31.0 | | | 36.7 | | 40.4 | | | 44.6 | | | 48.2 | |
| 58 | | | | | | | | | | | 29.1 | | | 35.3 | | | | | 43.7 | | | 47.6 | | |
| 59 | | | | | | | | | | 26.9 | | | 33.7 | | | 39.2 | | 42.6 | 44.5 | | 46.8 | | | 50.4 |
| 60 | | | | | | | | 24.4 | | | 29.6 | 31.8 | | 36.1 | 37.8 | | 41.3 | | | 45.9 | | 48.5 | 49.7 | |
| 62 | | | | | | | | | | 27.5 | | | 34.5 | | | 40.1 | | 43.5 | | | 48.1 | | | 51.8 |
| 64 | | | | | | | | 25.1 | | | 30.5 | 32.7 | | 37.1 | 38.9 | | 42.5 | | 46.0 | | | 50.2 | | |
| 66 | | | | | | | | 25.4 | | 28.3 | | 33.3 | 35.4 | | 39.6 | | | 44.8 | | 48.0 | 49.7 | | 52.4 | |
| 68 | | | | | | | | | | 28.6 | 31.2 | | | | | | | | 47.1 | | | 51.9 | | 54.5 |
| 70 | | | | | | | | 26.0 | | | 31.5 | 34.1 | 36.2 | | 40.6 | | 44.4 | 46.0 | | 49.4 | 50.9 | | 54.1 | |
| 72 | | | | | | | | 26.3 | | 29.2 | | | | | | | | | 48.6 | | 51.6 | 53.1 | | 56.2 |

## Table A–3 Spiral Flat–Oval Duct (Nominal Sizes)

**(Diameter of the round duct which will have the capacity and friction equivalent to the actual duct size)**

| | 75 | 100 | 125 | 150 | 175 | 200 | 225 | 250 | 275 | 300 | 350 | 400 | 450 | 500 | 550 | 600 | 650 | 700 | 750 | 800 | 850 | 900 | 950 | 1000 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 175 | | 145 | | | | | | | | | | | | | | | | | | | | | | |
| 200 | 130 | | 168 | 175 | | | | | | | | | | | | | | | | | | | | |
| 225 | 142 | 157 | | 196 | | | | | | | | | | | | | | | | | | | | |
| 250 | | 170 | 185 | | 221 | 229 | | | | | | | | | | | | | | | | | | |
| 275 | 152 | | 201 | 213 | | 249 | | | | | | | | | | | | | | | | | | |
| 300 | 163 | 183 | | 226 | 239 | | 274 | 279 | | | | | | | | | | | | | | | | |
| 325 | | 193 | 213 | | 257 | 269 | | 302 | | | | | | | | | | | | | | | | |
| 350 | 170 | | 224 | 244 | | 284 | 292 | | | | | | | | | | | | | | | | | |
| 375 | 178 | 203 | | 257 | 272 | | | | | 351 | | | | | | | | | | | | | | |
| 400 | | | 236 | | | | | 340 | 345 | | | | | | | | | | | | | | | |
| 425 | 185 | 213 | | | | | 328 | | | | | | | | | | | | | | | | | |
| 450 | | | | | 297 | 315 | | | | 389 | | | | | | | | | | | | | | |
| 475 | | | 254 | 279 | | | | | 381 | | | | | | | | | | | | | | | |
| 500 | | | | | | | 356 | 373 | | | 445 | | | | | | | | | | | | | |
| 525 | | | | | 320 | 342 | | | | 424 | | | 505 | | | | | | | | | | | |
| 550 | | | | 300 | | | | | 414 | | | 495 | | | | | | | | | | | | |
| 575 | | | | | | | 384 | 399 | | | 480 | | | | | | | | | | | | | |
| 600 | | | | | | 366 | | | | | | | 549 | | | | | | | | | | | |
| 625 | | | | 318 | | | | | | 457 | | 531 | | | 607 | | | | | | | | | |
| 650 | | | | | | | | 424 | | | | | | 599 | | | | | | | | | | |
| 675 | | | | | | 386 | | | | | 513 | | 587 | | | 658 | | | | | | | | |
| 700 | | | | 335 | | | | | | | | | | | 650 | | | | | | | | | |
| 725 | | | | | | | | 450 | | 485 | | 566 | | 640 | | | 709 | | | | | | | |
| 750 | | | | | | 404 | | | | | 541 | | | | | 714 | | | | | | | | |
| 775 | | | | 351 | | | | | | 511 | | | 622 | | 691 | | | 759 | | | | | | |
| 800 | | | | | | | | 470 | | | | 597 | | | | | 754 | | | | | | | |
| 825 | | | | | | 422 | | | | | 569 | | | 676 | | 744 | | | 813 | | | | | |
| 850 | | | | 363 | | | | | | 531 | | | 653 | | | | | 805 | | | | | | |
| 875 | | | | | | | | 490 | | | | 627 | | | 729 | | 795 | | | 864 | | | | |
| 900 | | | | | | 439 | | | | | 594 | | | 709 | | | | | 856 | | | | | |
| 925 | | | | 378 | | | | | | 556 | | | 686 | | | 782 | | 848 | | | 914 | | | |
| 950 | | | | | | | | 511 | | | | 653 | | | 762 | | | | | 909 | | | | |
| 975 | | | | | | 455 | | | | | 620 | | | 742 | | | 833 | | 899 | | | 965 | | |
| 1000 | | | | | | | | | | 577 | | | 714 | | | 818 | | | | | 960 | | | |
| 1025 | | | | 391 | | | | 528 | | | | 681 | | | 795 | | | 886 | | 950 | | | 1016 | |
| 1050 | | | | | | | | | | | 643 | | | 770 | | | 871 | | | | | 1011 | | |
| 1075 | | | | | | 472 | | | | 597 | | | 739 | | | 851 | | | 939 | | 1004 | | | 1067 |
| 1100 | | | | 404 | | | | | | | | 504 | | | 826 | | | 925 | | | | | 1062 | |
| 1125 | | | | | | | | 546 | | | 663 | | | 798 | | | 904 | | | 991 | | 1054 | | |
| 1150 | | | | | | 485 | | | | | | | 767 | | | 881 | | | 978 | | | | | 1113 |
| 1175 | | | | | | | | | | 617 | | 726 | | | 856 | | | 960 | | | 1044 | | 1105 | |
| 1200 | | | | | | | | 561 | | | | | | 826 | | | 937 | | | 1029 | | | | |
| 1225 | | | | | | 498 | | | | | 683 | | 790 | | | 912 | | | 1013 | | | 1095 | | 1158 |
| 1250 | | | | | | | | | | 635 | | | | | 884 | | | 993 | | | 1082 | | | |
| 1275 | | | | | | | | 579 | | | | 747 | | 848 | | | 968 | | | 1067 | | | 1148 | |
| 1300 | | | | | | 513 | | | | | 704 | | | | | 940 | | | 1046 | | | 1135 | | |
| 1325 | | | | | | | | | | 653 | | | 813 | | 909 | | | 1023 | | | 1120 | | | 1199 |
| 1350 | | | | | | | | 592 | | | | 767 | | | | | 998 | | | 1100 | | | 1186 | |
| 1375 | | | | | | | | | | | 721 | | | 874 | | 968 | | | 1080 | | | 1174 | | |
| 1400 | | | | | | | | | | 668 | | | 836 | | | | | 1054 | | | 1156 | | | 1240 |
| 1425 | | | | | | | | 605 | | | | 787 | | | 932 | | 1026 | | | 1133 | | | 1224 | |
| 1450 | | | | | | | | | | | 739 | | | 897 | | | | | 1110 | | | 1209 | | |
| 1475 | | | | | | | | | | 683 | | | 856 | | | 996 | | 1082 | 1130 | | 1189 | | | 1280 |
| 1500 | | | | | | | | 620 | | | 752 | 808 | | 917 | 960 | | 1049 | | | 1166 | | 1232 | 1262 | |
| 1550 | | | | | | | | | | 699 | | | 876 | | | 1018 | | 1105 | | | 1222 | | | 1316 |
| 1600 | | | | | | | | 638 | | | 775 | 830 | | 942 | 988 | | 1079 | | 1168 | | | 1275 | | |
| 1700 | | | | | | | | | | 726 | 792 | | | 965 | | 1067 | | | 1196 | | | 1318 | | 1384 |
| 1750 | | | | | | | | 660 | | | 800 | 867 | 919 | | 1031 | | 1128 | 1168 | | 1255 | 1293 | | 1374 | |
| 1800 | | | | | | | | 668 | | 742 | | | | | | | | | 1234 | | 1311 | 1379 | | 1427 |

## Table A–3M Spiral Flat–Oval Duct (Nominal Sizes)

**(Diameter of the round duct which will have the capacity and friction equivalent to the actual duct size)**

| Velocity fpm | Velocity Pressure in. wg | Velocity fpm | Velocity Pressure in. wg | Velocity fpm | Velocity Pressure in. wg | Velocity fpm | Velocity Pressure in. wg | Velocity fpm | Velocity Pressure in. wg |
|---|---|---|---|---|---|---|---|---|---|
| 300 | 0.01 | 2050 | 0.26 | 3800 | 0.90 | 5550 | 1.92 | 7300 | 3.32 |
| 350 | 0.01 | 2100 | 0.27 | 3850 | 0.92 | 5600 | 1.95 | 7350 | 3.37 |
| 400 | 0.01 | 2150 | 0.29 | 3900 | 0.95 | 5650 | 1.99 | 7400 | 3.41 |
| 450 | 0.01 | 2200 | 0.30 | 3950 | 0.97 | 5700 | 2.02 | 7450 | 3.46 |
| 500 | 0.02 | 2250 | 0.32 | 4000 | 1.00 | 5750 | 2.06 | 7500 | 3.51 |
| 550 | 0.02 | 2300 | 0.33 | 4050 | 1.02 | 5800 | 2.10 | 7550 | 3.55 |
| 600 | 0.02 | 2350 | 0.34 | 4100 | 1.05 | 5850 | 2.13 | 7600 | 3.60 |
| 650 | 0.03 | 2400 | 0.36 | 4150 | 1.07 | 5900 | 2.17 | 7650 | 3.65 |
| 700 | 0.03 | 2450 | 0.37 | 4200 | 1.10 | 5950 | 2.21 | 7700 | 3.70 |
| 750 | 0.04 | 2500 | 0.39 | 4250 | 1.13 | 6000 | 2.24 | 7750 | 3.74 |
| 800 | 0.04 | 2550 | 0.41 | 4300 | 1.15 | 6050 | 2.28 | 7800 | 3.79 |
| 850 | 0.05 | 2600 | 0.42 | 4350 | 1.18 | 6100 | 2.32 | 7850 | 3.84 |
| 900 | 0.05 | 2650 | 0.44 | 4400 | 1.21 | 6150 | 2.36 | 7900 | 3.89 |
| 950 | 0.06 | 2700 | 0.45 | 4450 | 1.23 | 6200 | 2.40 | 7950 | 3.94 |
| 1000 | 0.06 | 2750 | 0.47 | 4500 | 1.26 | 6250 | 2.43 | 8000 | 3.99 |
| 1050 | 0.07 | 2800 | 0.49 | 4550 | 1.29 | 6300 | 2.47 | 8050 | 4.04 |
| 1100 | 0.08 | 2850 | 0.51 | 4600 | 1.32 | 6350 | 2.51 | 8100 | 4.09 |
| 1150 | 0.08 | 2900 | 0.52 | 4650 | 1.35 | 6400 | 2.55 | 8150 | 4.14 |
| 1200 | 0.09 | 2950 | 0.54 | 4700 | 1.38 | 6450 | 2.59 | 8200 | 4.19 |
| 1250 | 0.10 | 3000 | 0.56 | 4750 | 1.41 | 6500 | 2.63 | 8250 | 4.24 |
| 1300 | 0.11 | 3050 | 0.58 | 4800 | 1.44 | 6550 | 2.67 | 8300 | 4.29 |
| 1350 | 0.11 | 3100 | 0.60 | 4850 | 1.47 | 6600 | 2.71 | 8350 | 4.35 |
| 1400 | 0.12 | 3150 | 0.62 | 4900 | 1.50 | 6650 | 2.76 | 8400 | 4.40 |
| 1450 | 0.13 | 3200 | 0.64 | 4950 | 1.53 | 6700 | 2.80 | 8450 | 4.45 |
| 1500 | 0.14 | 3250 | 0.66 | 5000 | 1.56 | 6750 | 2.84 | 8500 | 4.50 |
| 1550 | 0.15 | 3300 | 0.68 | 5050 | 1.59 | 6800 | 2.88 | 8550 | 4.56 |
| 1600 | 0.16 | 3350 | 0.70 | 5100 | 1.62 | 6850 | 2.92 | 8600 | 4.61 |
| 1650 | 0.17 | 3400 | 0.72 | 5150 | 1.65 | 6900 | 2.97 | 8650 | 4.66 |
| 1700 | 0.18 | 3450 | 0.74 | 5200 | 1.69 | 6950 | 3.01 | 8700 | 4.72 |
| 1750 | 0.19 | 3500 | 0.76 | 5250 | 1.72 | 7000 | 3.05 | 8750 | 4.77 |
| 1800 | 0.20 | 3550 | 0.79 | 5300 | 1.75 | 7050 | 3.10 | 8800 | 4.83 |
| 1850 | 0.21 | 3600 | 0.81 | 5350 | 1.78 | 7100 | 3.14 | 8850 | 4.88 |
| 1900 | 0.22 | 3650 | 0.83 | 5400 | 1.82 | 7150 | 3.19 | 8900 | 4.94 |
| 1950 | 0.24 | 3700 | 0.85 | 5450 | 1.85 | 7200 | 3.23 | 8950 | 4.99 |
| 2000 | 0.25 | 3750 | 0.88 | 5500 | 1.89 | 7250 | 3.28 | 9000 | 5.05 |

**Table A–4 Velocities/Velocity Pressures**

$$Velocity = 4005\sqrt{V_p} \text{ (or) } V_p = \left(\frac{Velocity}{4005}\right)^2$$

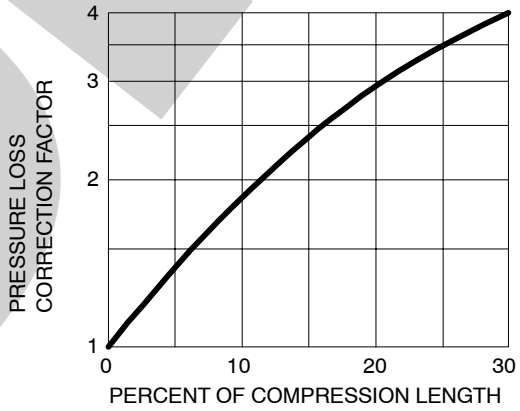


**FIGURE A–3 CORRECTION FACTOR FOR UNEXTENDED FLEXIBLE DUCT**

DRAFT

| Velocity (m/s) | Velocity Pressure (Pa) | Velocity (m/s) | Velocity Pressure (Pa) | Velocity (m/s) | Velocity Pressure (Pa) | Velocity (m/s) | Velocity Pressure (Pa) | Velocity (m/s) | Velocity Pressure (Pa) |
|---|---|---|---|---|---|---|---|---|---|
| 1.0 | 0.6 | 10.0 | 60 | 19.0 | 217 | 28.0 | 472 | 37.0 | 824 |
| 1.2 | 0.9 | 10.2 | 63 | 19.2 | 222 | 28.2 | 479 | 37.2 | 833 |
| 1.4 | 1.2 | 10.4 | 65 | 19.4 | 227 | 28.4 | 486 | 37.4 | 842 |
| 1.6 | 1.5 | 10.6 | 68 | 19.6 | 231 | 28.6 | 493 | 37.6 | 851 |
| 1.8 | 2.0 | 10.8 | 70 | 19.8 | 236 | 28.8 | 499 | 37.8 | 860 |
| 2.0 | 2.4 | 11.0 | 73 | 20.0 | 241 | 29.0 | 506 | 38.0 | 870 |
| 2.2 | 2.9 | 11.2 | 76 | 20.2 | 246 | 29.2 | 513” | 38.2 | 879 |
| 2.4 | 3.5 | 11.4 | 78 | 20.4 | 251 | 29.4 | 521 | 38.4 | 888 |
| 2.6 | 4.1 | 11.6 | 81 | 20.6 | 256 | 29.6 | 528 | 38.6 | 897 |
| 2.8 | 4.7 | 11.8 | 84 | 20.8 | 261 | 29.8 | 535 | 38.8 | 907 |
| 3.0 | 5.4 | 12.0 | 87 | 21.0 | 266 | 30.0 | 542 | 39.0 | 916 |
| 3.2 | 6.2 | 12.2 | 90 | 21.2 | 271 | 30.2 | 549 | 39.2 | 925 |
| 3.4 | 7.0 | 12.4 | 93 | 21.4 | 276 | 30.4 | 557 | 39.4 | 935 |
| 3.6 | 7.8 | 12.6 | 96 | 21.6 | 281 | 30.6 | 564 | 39.6 | 944 |
| 3.8 | 8.7 | 12.8 | 99 | 21.8 | 286 | 30.8 | 571 | 39.8 | 954 |
| 4.0 | 9.6 | 13.0 | 102 | 22.0 | 291 | 31.0 | 579 | 40.0 | 963 |
| 4.2 | 10.6 | 13.2 | 105 | 22.2 | 297 | 31.2 | 586 | 40.2 | 973 |
| 4.4 | 11.7 | 13.4 | 108 | 22.4 | 302 | 31.4 | 594 | 40.4 | 983 |
| 4.6 | 12.7 | 13.6 | 111 | 22.6 | 308 | 31.6 | 601 | 40.6 | 993 |
| 4.8 | 13.9 | 13.8 | 115 | 22.8 | 313 | 31.8 | 609 | 40.8 | 1002 |
| 5.0 | 15.1 | 14.0 | 118 | 23.0 | 319 | 32.0 | 617 | 41.0 | 1012 |
| 5.2 | 16.3 | 14.2 | 121 | 23.2 | 324 | 32.2 | 624 | 41.2 | 1022 |
| 5.4 | 17.6 | 14.4 | 125 | 23.4 | 330 | 32.4 | 632 | 41.4 | 1032 |
| 5.6 | 18.9 | 14.6 | 128 | 23.6 | 335 | 32.6 | 640 | 41.6 | 1042 |
| 5.8 | 20.3 | 14.8 | 132 | 23.8 | 341 | 32.8 | 648 | 41.8 | 1052 |
| 6.0 | 21.7 | 15.0 | 135 | 24.0 | 347 | 33.0 | 656 | 42.0 | 1062 |
| 6.2 | 23.1 | 15.2 | 139 | 24.2 | 353 | 33.2 | 664 | 42.2 | 1072 |
| 6.4 | 24.7 | 15.4 | 143 | 24.4 | 359 | 33.4 | 672 | 42.4 | 1083 |
| 6.6 | 26.2 | 15.6 | 147 | 24.6 | 364 | 33.6 | 680 | 42.6 | 1093 |
| 6.8 | 27.8 | 15.8 | 150 | 24.8 | 370 | 33.8 | 688 | 42.8 | 1103 |
| 7.0 | 29.5 | 16.0 | 154 | 25.0 | 376 | 34.0 | 696 | 43.0 | 1113 |
| 7.2 | 31.2 | 16.2 | 158 | 25.2 | 382 | 34.2 | 704 | 43.2 | 1124 |
| 7.4 | 33.0 | 16.4 | 162 | 25.4 | 389 | 34.4 | 713 | 43.4 | 1134 |
| 7.6 | 34.8 | 16.6 | 166 | 25.6 | 395 | 34.6 | 721 | 43.6 | 1145 |
| 7.8 | 36.6 | 16.8 | 170 | 25.8 | 401 | 34.8 | 729 | 43.8 | 1155 |
| 8.0 | 38.5 | 17.0 | 174 | 26.0 | 407 | 35.0 | 738 | 44.0 | 1166 |
| 8.2 | 40.5 | 17.2 | 178 | 26.2 | 413 | 35.2 | 746 | 44.2 | 1176 |
| 8.4 | 42.5 | 17.4 | 182 | 26.4 | 420 | 35.4 | 755 | 44.4 | 1187 |
| 8.6 | 44.5 | 17.6 | 187 | 26.6 | 426 | 35.6 | 763 | 44.6 | 1198 |
| 8.8 | 46.6 | 17.8 | 191 | 26.8 | 433 | 35.8 | 772 | 44.8 | 1209 |
| 9.0 | 48.8 | 18.0 | 195 | 27.0 | 439 | 36.0 | 780 | 45.0 | 1219 |
| 9.2 | 51.0 | 18.2 | 199 | 27.2 | 446 | 36.2 | 789 | 45.2 | 1230 |
| 9.4 | 53.2 | 18.4 | 204 | 27.4 | 452 | 36.4 | 798 | 45.4 | 1241 |
| 9.6 | 55.5. | 18.6 | 208 | 27.6 | 459 | 36.6 | 807 | 45.6 | 1252 |
| 9.8 | 57.8 | 18.8 | 213 | 27.8 | 465 | 36.8 | 815 | 45.8 | 1263 |

**Table A–4M Velocities/Velocity Pressures**

| Degrees | Radians | Degrees | Radians |
|---|---|---|---|
| 10° | 0.175 | 70° | 1.22 |
| 20° | 0.349 | 80° | 1.40 |
| 30° | 0.524 | 90° | 1.57 (π/2) |
| 40° | 0.698 | 135° | 2.36 |
| 50° | 0.873 | 180° | 3.14 (π) |
| 60° | 1.05 | 360° | 6.28 (2π) |

**Table A–5 Angular Conversion**

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| $V_s/V_c$ | C | |
|---|---|---|
| | 75% Regain | 90% Regain |
| 0.95 | 0.03 | 0.01 |
| .91 | .04 | .02 |
| .87 | .06 | .02 |
| .83 | .08 | .03 |
| .80 | .09 | .04 |
| 0.77 | 0.10 | 0.04 |
| .74 | .11 | .04 |
| .71 | .12 | .05 |
| .69 | .13 | .05 |
| .67 | .14 | .06 |
| 0.65 | 0.15 | 0.06 |
| .63 | .15 | .06 |
| .61 | .16 | .06 |
| .59 | .16 | .07 |
| .57 | .17 | .07 |
| 0.56 | 0.18 | 0.07 |
| .54 | .18 | .07 |
| .53 | .18 | .07 |
| .51 | .18 | .07 |
| .50 | .19 | .08 |

$V_s$ = Downstream Velocity; $V_c$ = Upstream Velocity

**Table A–6 Loss Coefficients for Straight–Through Flow**

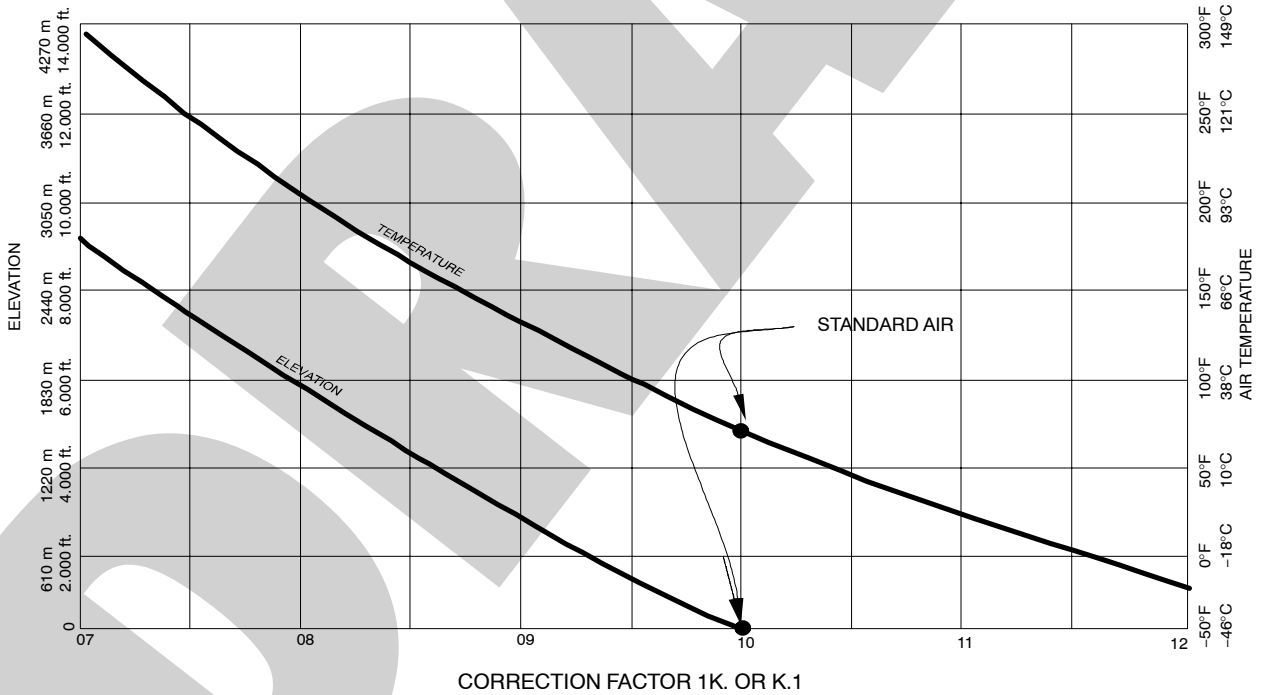


**FIGURE A–4 AIR DENSITY FRICTION CHART CORRECTION FACTORS**

*NOTE: When an air distribution system is designed to operate above 2000 feet (610 m) altitude, below 32 ° F (0 °C), or above 1200 ° F (490 ° C) temperature, the duct friction loss obtained must be corrected for the air density. The actual air flow (cfm or L/s) is used to find the duct friction loss which is multiplied by the correction factor or factors from the above chart to obtain the actual friction loss.*

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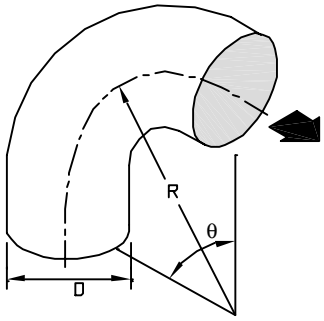
# FITTING LOSS COEFFICIENT TABLES

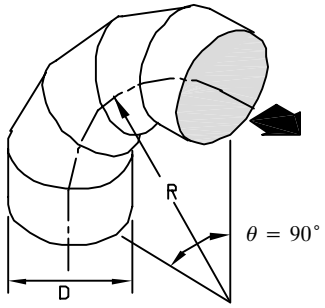
## A. ELBOW, SMOOTH RADIUS (DIE STAMPED), ROUND




| Coefficients for 90° Elbows (*See* Note 1) | | | | | | |
|---|---|---|---|---|---|---|
| R/D | 0.5 | 0.75 | 1.0 | 1.5 | 2.0 | 2.5 |
| C | 0.71 | 0.33 | 0.22 | 0.15 | 0.13 | 0.12 |

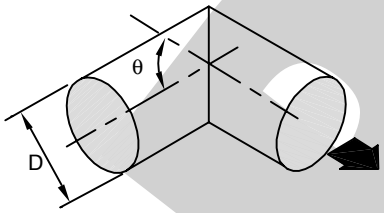
| Note 1: For angles other than 90° multiply by the following factors: | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $\theta$ | 0° | 20° | 30° | 45° | 60° | 75° | 90° | 110° | 130° | 150° | 180° |
| K | 0 | 0.31 | 0.45 | 0.60 | 0.78 | 0.90 | 1.00 | 1.13 | 1.20 | 1.28 | 1.40 |

## B. ELBOW, ROUND, 3 TO 5 PIECE — 90°




| No. of Pieces | Coefficient C | | | | |
|---|---|---|---|---|---|
| | R/D | | | | |
| | 0.5 | 0.75 | 1.0 | 1.5 | 2.0 |
| 5 | — | 0.46 | 0.33 | 0.24 | 0.19 |
| 4 | — | 0.50 | 0.37 | 0.27 | 0.24 |
| 3 | 0.98 | 0.54 | 0.42 | 0.34 | 0.33 |

## C. ELBOW, ROUND, MITERED




| Coefficient C (*See* Note 2) | | | | | | |
|---|---|---|---|---|---|---|
| $\theta$ | 20° | 30° | 45° | 60° | 75° | 90° |
| C | 0.08 | 0.16 | 0.34 | 0.55 | 0.81 | 1.2 |

| $R_e 10^{-4}$ | 1 | 2 | 3 | 4 | 6 | 8 | 10 | ≥14 |
|---|---|---|---|---|---|---|---|---|
| $K_{Re}$ | 1.40 | 1.26 | 1.19 | 1.14 | 1.09 | 1.06 | 1.04 | 1.0 |

**U.S. Units** **Metric Units**

For Standard Air:

$R_e = 8.56\ DV$ $R_e = 66.4\ DV$

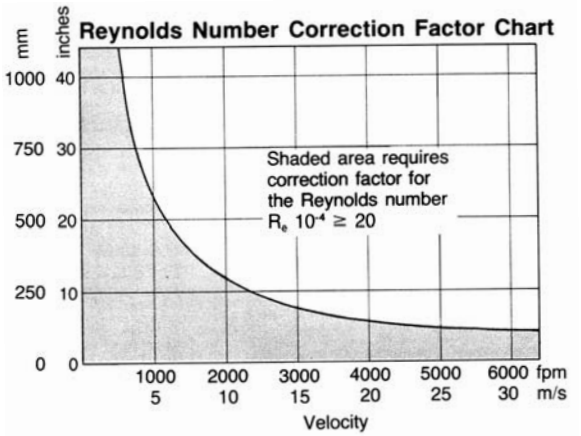
where:

$D$ = duct diameter, inches (mm)

$V$ = duct velocity, fpm (m/s)

For Rectangular Ducts:

$$D = \frac{2HW}{H + W}$$




*Note 1:* Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

*Note 2:* Correction factor for Reynolds number - $K_{Re}$

## Table A–7 Loss Coefficients, Elbows

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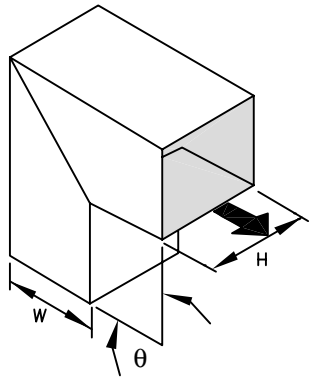
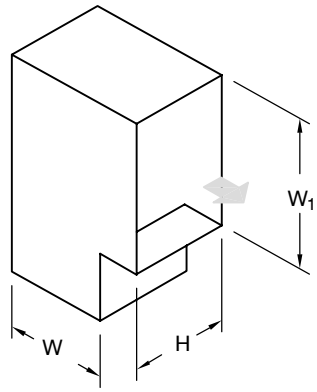
### D. ELBOW, RECTANGULAR, MITERED



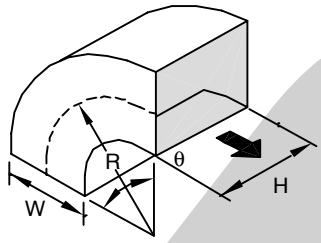

| θ | Coeffiecient C | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | H/W | | | | | | | | |
| | 0.25 | 0.5 | 0.75 | 1.0 | 1.5 | 2.0 | 4.0 | 6.0 | 8.0 |
| 20° | 0.08 | 0.08 | 0.08 | 0.07 | 0.07 | 0.07 | 0.06 | 0.05 | 0.05 |
| 30° | 0.18 | 0.17 | 0.17 | 0.16 | 0.15 | 0.15 | 0.13 | 0.12 | 0.11 |
| 45° | 0.38 | 0.37 | 0.36 | 0.34 | 0.33 | 0.31 | 0.27 | 0.25 | 0.24 |
| 60° | 0.60 | 0.59 | 0.57 | 0.55 | 0.52 | 0.49 | 0.43 | 0.39 | 0.38 |
| 75° | 0.89 | 0.87 | 0.84 | 0.81 | 0.77 | 0.73 | 0.63 | 0.58 | 0.57 |
| 90° | 1.3 | 1.3 | 1.2 | 1.2 | 1.1 | 1.1 | 0.92 | 0.85 | 0.83 |

### E. ELBOW, RECTANGULAR, MITERED WITH CONVERGING OR DIVERGIN FLOW




| H/W | Coeffiecient C | | | | | |
|---|---|---|---|---|---|---|
| | $W_1/W$ | | | | | |
| | 0.6 | 0.8 | 1.2 | 1.4 | 1.6 | 2.0 |
| 0.25 | 1.8 | 1.4 | 1.1 | 1.1 | 1.1 | 1.1 |
| 1.0 | 1.7 | 1.1 | 1.0 | 0.95 | 0.90 | 0.84 |
| 4.0 | 1.5 | 1.0 | 0.81 | 0.76 | 0.72 | 0.66 |
| ∞ | 1.5 | 1.0 | 0.69 | 0.63 | 0.60 | 0.55 |

### F. ELBOW, RECTANGULAR, SMOOTH RADIUS WITHOUT VANES




| R/W | Coeffiecients for 90° elbows (*See* Note 3) Coefficient C | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | H/W | | | | | | | | | | |
| | 0.25 | 0.5 | 0.75 | 1.0 | 1.5 | 2.0 | 3.0 | 4.0 | 5.0 | 6.0 | 8.0 |
| 0.5 | 1.5 | 1.4 | 1.3 | 1.2 | 1.1 | 1.0 | 1.0 | 1.1 | 1.1 | 1.2 | 1.2 |
| 0.75 | 0.57 | 0.52 | 0.48 | 0.44 | 0.40 | 0.39 | 0.39 | 0.40 | 0.42 | 0.43 | 0.44 |
| 1.0 | 0.27 | 0.25 | 0.23 | 0.21 | 0.19 | 0.18 | 0.18 | 0.19 | 0.20 | 0.27 | 0.21 |
| 1.5 | 0.22 | 0.20 | 0.19 | 0.17 | 0.15 | 0.14 | 0.14 | 0.15 | 0.16 | 0.17 | 0.17 |
| 2.0 | 0.20 | 0.18 | 0.16 | 0.15 | 0.14 | 0.13 | 0.13 | 0.14 | 0.14 | 0.15 | 0.15 |

**U.S. Units** **Metric Units**

For Standard Air:

$R_e = 8.56\ DV$ $R_e = 66.4\ DV$

where:

$D$ = duct diameter, inches (mm)

$V$ = duct velocity, fpm (m/s)

For Rectangular Ducts:

$$D = \frac{2HW}{H + W}$$

| R/W | $R_e\ 10^{-4}$ | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 6 | 8 | 10 | 14 | ≥20 |
| 0.5 | 1.40 | 1.26 | 1.19 | 1.14 | 1.09 | 1.06 | 1.04 | 1.0 | 1.0 |
| ≥0.75 | 2.0 | 1.77 | 1.64 | 1.56 | 1.46 | 1.38 | 1.30 | 1.15 | 1.0 |

*Note 1:* Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

*Note 3:* Correction factor for Reynolds number - $K_{Re}$

## Table A–7 Loss Coefficients, Elbows (continued)

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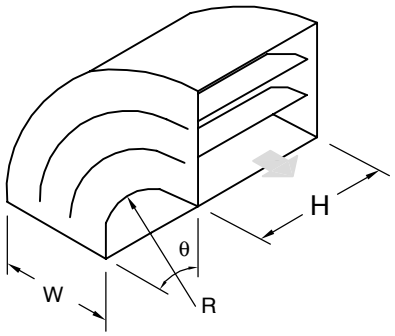
## G. ELBOW, RECTANGULAR, SMOOTH RADIUS WITH SPLITTER VANES




*NOTES:*

a. See Page x to calculate splitter vane spacing

b. CR = Curve Radio

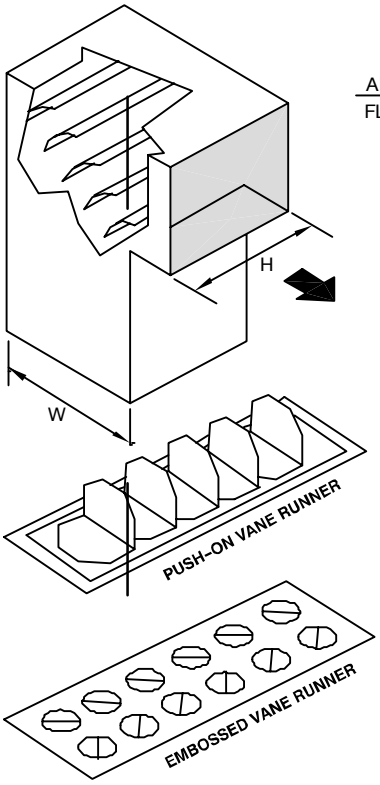
c. Use correction factors in Note 1 on Page x for elbows other than 90°

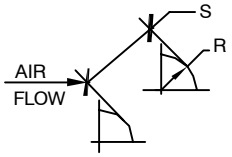
| | R/W | CR | Coefficient C H/W | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0.25 | 0.5 | 1.0 | 1.5 | 2.0 | 3.0 | 4.0 | 5.0 | 6.0 | 7.0 | 8.0 |
| 1 splitter vane | 0.05 | 0.218 | 0.52 | 0.40 | 0.43 | 0.49 | 0.55 | 0.66 | 0.75 | 0.84 | 0.93 | 1.0 | 1.1 |
| | 0.10 | 0.302 | 0.36 | 0.27 | 0.25 | 0.28 | 0.30 | 0.35 | 0.39 | 0.42 | 0.46 | 0.49 | 0.52 |
| | 0.15 | 0.361 | 0.28 | 0.21 | 0.18 | 0.19 | 0.20 | 0.22 | 0.25 | 0.26 | 0.28 | 0.30 | 0.32 |
| | 0.20 | 0.408 | 0.22 | 0.16 | 0.14 | 0.14 | 0.15 | 0.16 | 0.17 | 0.18 | 0.19 | 0.20 | 0.21 |
| | 0.25 | 0447 | 0.18 | 0.13 | 0.11 | 0.11 | 0.11 | 0.12 | 0.13 | 0.14 | 0.14 | 0.15 | 0.15 |
| | 0.30 | 0.480 | 0.15 | 0.11 | 0.09 | 0.09 | 0.09 | 0.09 | 0.10 | 0.10 | 0.11 | 0.11 | 0.12 |
| | 0.35 | 0.509 | 0.13 | 0.09 | 0.08 | 0.07 | 0.07 | 0.08 | 0.08 | 0.08 | 0.08 | 0.09 | 0.09 |
| | 0.40 | 0.535 | 0.11 | 0.08 | 0.07 | 0.06 | 0.06 | 0.06 | 0.06 | 0.07 | 0.07 | 0.07 | 0.07 |
| | 0.45 | 0.557 | 0.10 | 0.07 | 0.06 | 0.05 | 0.05 | 0.05 | 0.05 | 0.05 | 0.06 | 0.06 | 0.06 |
| | 0.50 | 0.577 | 0.09 | 0.06 | 0.05 | 0.05 | 0.04 | 0.04 | 0.04 | 0.05 | 0.05 | 0.05 | 0.05 |
| 2 splitter vanes | 0.05 | 0.362 | 0.26 | 0.20 | 0.22 | 0.25 | 0.28 | 0.33 | 0.37 | 0.41 | 0.45 | 0.48 | 0.51 |
| | 0.10 | 0.450 | 0.17 | 0.13 | 0.11 | 0.12 | 0.13 | 0.15 | 0.16 | 0.17 | 0.19 | 0.20 | 0.21 |
| | 0.15 | 0.507 | 0.12 | 0.09 | 0.08 | 0.08 | 0.08 | 0.09 | 0.10 | 0.10 | 0.11 | 0.11 | 0.11 |
| | 0.20 | 0.550 | 0.09 | 0.07 | 0.06 | 0.05 | 0.06 | 0.06 | 0.06 | 0.06 | 0.07 | 0.07 | 0.07 |
| | 0.25 | 0.585 | 0.08 | 0.05 | 0.04 | 0.04 | 0.04 | 0.04 | 0.05 | 0.05 | 0.05 | 0.05 | 0.05 |
| | 0.30 | 0.613 | 0.06 | 0.04 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.04 | 0.04 | 0.04 |
| | 0.35 | 0.638 | 0.05 | 0.04 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 |
| | 0.40 | 0.659 | 0.05 | 0.03 | 0.03 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 |
| | 0.45 | 0.677 | 0.04 | 0.92 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 |
| | 0.50 | 0.693 | 0.03 | 0.02 | 0.02 | 0.02 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| 3 splitter vanes | 0.05 | 0.467 | 0.11 | 0.10 | 0.12 | 0.13 | 0.14 | 0.16 | 0.18 | 0.19 | 0.21 | 0.22 | 0.23 |
| | 0.10 | 0.549 | 0.07 | 0.05 | 0.06 | 0.06 | 0.06 | 0.07 | 0.07 | 0.08 | 0.08 | 0.08 | 0.09 |
| | 0.15 | 0.601 | 0.05 | 0.04 | 0.04 | 0.04 | 0.04 | 0.04 | 0.04 | 0.04 | 0.04 | 0.05 | 0.05 |
| | 0.20 | 0.639 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 | 0.03 |
| | 0.25 | 0.669 | 0.03 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 | 0.02 |
| | 0.30 | 0.693 | 0.03 | 0.02 | 0.02 | 0.02 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| | 0.35 | 0.714 | 0.02 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| | 0.40 | 0.731 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| | 0.45 | 0.746 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| | 0.50 | 0.760 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

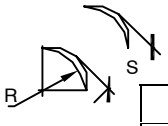
**Table A–7 Loss Coefficients, Elbows (continued)**

## H. ELBOW, RECTANGULAR, MITERED WITH TURNING VANES



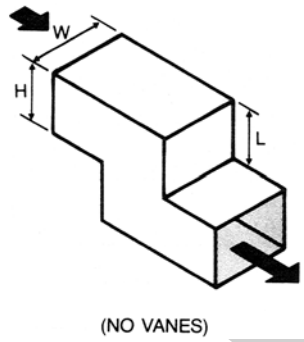




| Loss Coefficients (C) for Single Thickness Vanes | | | | | |
|---|---|---|---|---|---|
| Dimensions, Inches (mm) | | Velocity, fpm (m/s) | | | |
| R | S | 1000 (5) | 1500 (7.5) | 2000 (10) | 2500 (12.5) |
| 2.0 (50) | 1.5 (38) | 0.24 | 0.23 | 0.22 | 0.20 |
| 4.5 (114) | 3.25 (83) | 0.26 | 0.24 | 0.23 | 0.22 |




| Loss Coefficients (C) for Single Thickness Vanes | | | | | |
|---|---|---|---|---|---|
| Dimensions, Inches (mm) | | Velocity, fpm (m/s) | | | |
| R | S | 1000 (5) | 1500 (7.5) | 2000 (10) | 2500 (12.5) |
| 2.0 (50) | 1.5 (38) | 0.43 | 0.42 | 0.41 | 0.40 |
| 2.0 (50) | 2.25(56) | 0.53 | 0.52 | 0.50 | 0.49 |
| 4.5 (114) | 3.25 (83) | 0.27 | 0.25 | 0.24 | 0.23 |

## I. ELBOW, RECTANGULAR, MITERED WITH TURNING VANES




| Coefficients for W/H = 1.0 (*See* Notes 4 and 5) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| L/H | 0 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 0 | 0.62 | 0.90 | 1.6 | 2.6 | 3.6 | 4.0 | 4.2 | 4.2 | 4.2 |
| L/H | 2.4 | 2.8 | 3.2 | 4.0 | 5.0 | 6.0 | 7.0 | 9.0 | 10.0 | 8 |
| C | 3.7 | 3.3 | 3.2 | 3.1 | 2.9 | 2.8 | 2.7 | 2.6 | 2.5 | 2.3 |

*Note 4:* For W/H values other than 1.0 apply the following factor:

| W/H | 0.25 | 0.50 | 0.75 | 1.0 | 1.5 | 2.0 | 3.0 | 4.0 | 6.0 | 8.0 |
|---|---|---|---|---|---|---|---|---|---|---|
| K | 1.10 | 1.07 | 1.04 | 1.0 | 0.95 | 0.90 | 0.83 | 0.78 | 0.72 | 0.70 |

*Note 5:* Correction factor for Reynolds number — $K_{Re}$

| $R_e\ 10^{-4}$ | 1 | 2 | 3 | 4 | 6 | 8 | 10 | ≥ 14 |
|---|---|---|---|---|---|---|---|---|
| $K_{Re}$ | 1.40 | 1.40 | 1.19 | 1.14 | 1.09 | 1.06 | 1.04 | 1.0 |

**U.S. Units** **Metric Units**

For Standard Air:

$R_e = 8.56\ DV$ $R_e = 66.4\ DV$

where:

$D$ = duct diameter, inches (mm)

$V$ = duct velocity, fpm (m/s)

For Rectangular Ducts:

$$D = \frac{2HW}{H + W}$$

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

**Table A–7 Loss Coefficients, Elbows (continued)**

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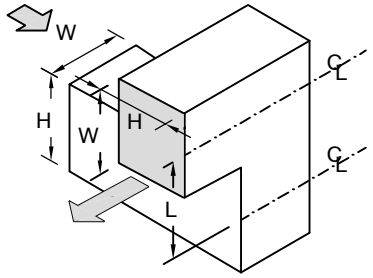
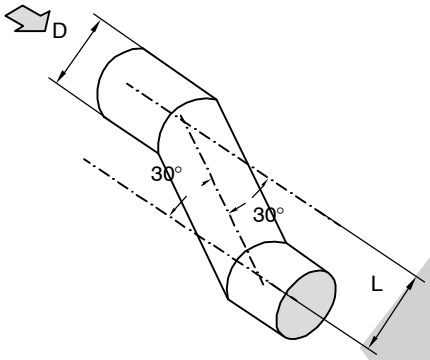
### J. ELBOWS, 90°, RECTANGULAR IN DIFFERENT PLANES



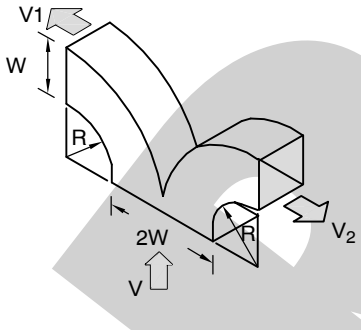

| Coefficients for H/W = 1.0: (*See* Notes 4 & 5) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| L/W | 0 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.2 | 2.4 | 2.9 | 3.3 | 3.4 | 3.4 | 3.4 | 3.3 | 3.2 | 3.1 |
| L/W | 2.4 | 2.8 | 3.2 | 4.0 | 5.0 | 6.0 | 7.0 | 9.0 | 10.0 | 8 |
| C | 3.2 | 3.2 | 3.2 | 3.0 | 2.9 | 2.8 | 2.7 | 2.5 | 2.4 | 2.3 |

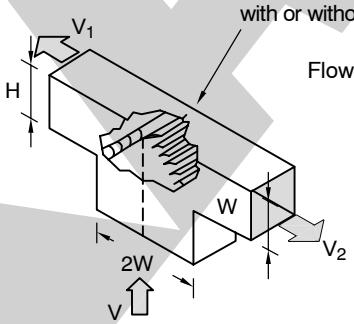
### K. ELBOWS, 30°, ROUND, OFFSET




| Coefficients C (*See* Note 5) | | | | | | | |
|---|---|---|---|---|---|---|---|
| L/D | 0 | 0.5 | 1.0 | 1.5 | 2.0 | 2.5 | 3.0 |
| C | 0 | 0.15 | 0.15 | 0.16 | 0.16 | 0.16 | 0.16 |

### L. ELBOWS, 90°, RECTANGULAR WYE OR TEE SHAPE

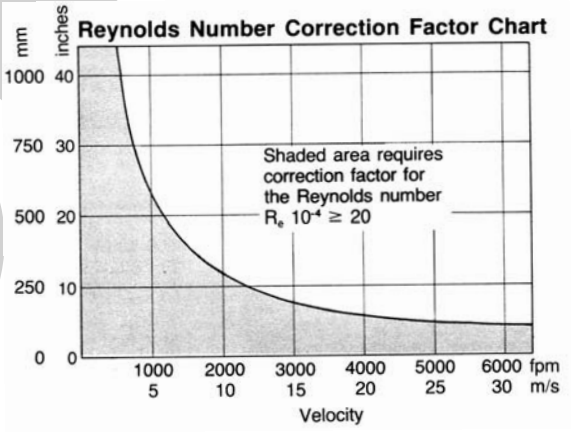



with or without divider sheet

Flow can be diverging or converging

Use value of C for a similar elbow, provided V = V1 = V2 (Velocities)




Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–7 Loss Coefficients, Elbows (continued)

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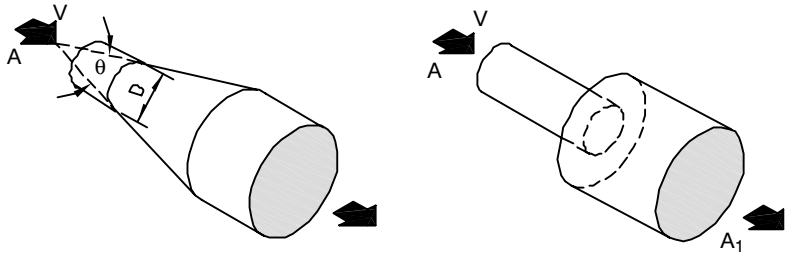
### A. TRANSITION, ROUND, CONICAL




U.S. Units: $R_e = 8.56\ DV$

Metric Units: $R_e = 66.4\ DV$

where:

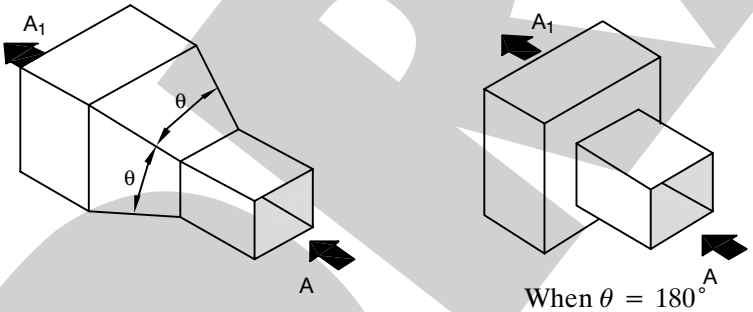
D = Upstream Diameter (inches)

V = Upstream Velocity (fpm)

When $\theta = 180^\circ$

| Coefficient C (*See* Note 6) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $R_e$ | $A_1/A$ | $\theta$ | | | | | | | |
| | | 16° | 20° | 30° | 45° | 60° | 90° | 120° | 180° |
| $0.5 \times 10^5$ | 2 | 0.14 | 0.19 | 0.32 | 0.33 | 0.33 | 0.32 | 0.31 | 0.30 |
| | 4 | 0.23 | 0.30 | 0.46 | 0.61 | 0.68 | 0.64 | 0.63 | 0.62 |
| | 6 | 0.27 | 0.33 | 0.48 | 0.66 | 0.77 | 0.74 | 0.73 | 0.72 |
| | 10 | 0.29 | 0.38 | 0.59 | 0.76 | 0.80 | 0.83 | 0.84 | 0.83 |
| | ≥16 | 0.31 | 0.38 | 0.60 | 0.84 | 0.88 | 0.88 | 0.88 | 0.88 |
| $2 \times 10^5$ | 2 | 0.07 | 0.12 | 0.23 | 0.28 | 0.27 | 0.27 | 0.27 | 0.26 |
| | 4 | 0.15 | 0.18 | 0.36 | 0.55 | 0.59 | 0.59 | 0.58 | 0.57 |
| | 6 | 0.19 | 0.28 | 0.44 | 0.90 | 0.70 | 0.71 | 0.71 | 0.69 |
| | 10 | 0.20 | 0.24 | 0.43 | 0.76 | 0.80 | 0.81 | 0.81 | 0.81 |
| | ≥16 | 0.21 | 0.28 | 0.52 | 0.76 | 0.87 | 0.87 | 0.87 | 087 |
| $\geq 6 \times 10^5$ | 2 | 0.05 | 0.07 | 0.12 | 0.27 | 0.27 | 0.27 | 0.27 | 0.27 |
| | 4 | 0.17 | 0.24 | 0.38 | 0.51 | 0.56 | 0.58 | 0.58 | 0.57 |
| | 6 | 0.16 | 0.29 | 0.46 | 0.60 | 0.69 | 0.71 | 0.70 | 0.70 |
| | 10 | 0.21 | 0.33 | 0.52 | 0.60 | 0.76 | 0.83 | 0.84 | 0.83 |
| | ≥16 | 0.21 | 0.34 | 0.56 | 0.72 | 0.79 | 0.85 | 0.87 | 0.89 |

### B. TRANSITION, RECTANGULAR, PYRAMIDAL




When $\theta = 180^\circ$

| Coefficient C (*See* Note 6) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| $A_1/A$ | $\theta$ | | | | | | | |
| | 16° | 20° | 30° | 45° | 60° | 90° | 120° | 180° |
| 2 | 0.18 | 0.22 | 0.25 | 0.29 | 0.31 | 0.32 | 0.33 | 0.30 |
| 4 | 0.36 | 0.43 | 0.50 | 0.56 | 0.61 | 0.63 | 0.63 | 0.63 |
| 6 | 0.42 | 0.47 | 0.58 | 0.68 | 0.72 | 0.76 | 0.76 | 0.75 |
| ≥10 | 0.42 | 0.49 | 0.59 | 0.70 | 0.80 | 0.87 | 0.85 | 0.86 |

*Note 6:* A = Area ( Entering airstream), $A_1$ = Area (Leaving airstream)

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–8 Loss Coefficients, Transitions (Diverging Flow)

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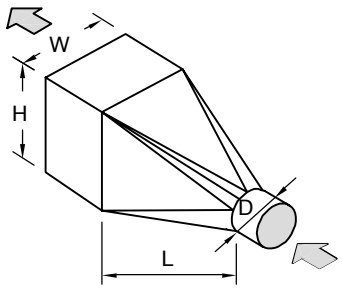
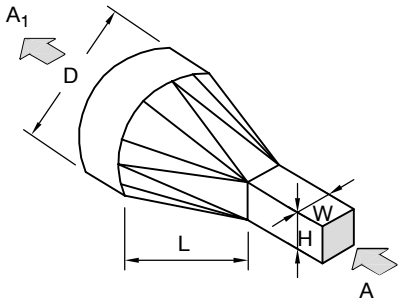
### C. TRANSITION, ROUND TO RECTANGULAR




$$\tan\left(\frac{\theta}{2}\right) = \frac{1.13\sqrt{HW} - D}{2L}$$

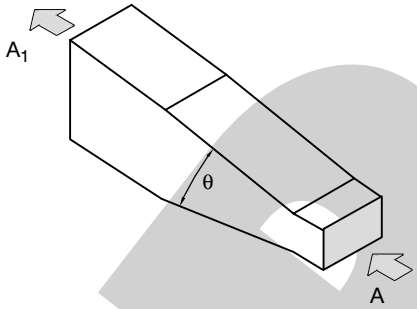
### D. TRANSITION, RECTANGULAR TO ROUND




$$\tan\left(\frac{\theta}{2}\right) = \frac{D - 1.13\sqrt{HW}}{2L}$$

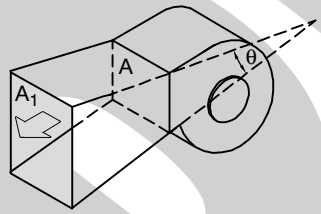
For Figures C nd D, compute θ using the equations and find the coefficient C from Table A-8B, Transition, Rectangular, Pyramidal.

### E. TRANSITION, RECTANGULAR SIDES STRAIGHT




| $A_1/A$ | Coefficient C (*See* Note 6) | | | | | | |
|---|---|---|---|---|---|---|---|
| | θ | | | | | | |
| | 14° | 20° | 30° | 45° | 60° | 90° | 180° |
| 2 | 0.09 | 0.12 | 0.20 | 0.34 | 0.37 | 0.38 | 0.35 |
| 4 | 0.16 | 0.25 | 0.42 | 0.60 | 0.68 | 0.70 | 0.66 |
| 6 | 0.19 | 0.30 | 0.48 | 0.65 | 0.76 | 0.83 | 0.80 |

### F. TRANSITION, SYMMETRIC AT FAN WITH DUCT SIDES STRAIGHT




| Coefficient C (*See* Note 6) | | | | | | |
|---|---|---|---|---|---|---|
| $A_1/A$ | 1.5 | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 |
| θ | | | | | | |
| 10° | 0.05 | 0.07 | 0.09 | 0.10 | 0.11 | 0.11 |
| 15° | 0.06 | 0.09 | 0.11 | 0.13 | 0.13 | 0.14 |
| 20° | 0.07 | 0.10 | 0.13 | 0.15 | 0.16 | 0.16 |
| 25° | 0.08 | 0.13 | 0.16 | 0.19 | 0.21 | 0.23 |
| 30° | 0.16 | 0.24 | 0.29 | 0.32 | 0.34 | 0.35 |
| 35° | 0.24 | 0.34 | 0.39 | 0.44 | 0.48 | 0.50 |

*Note 6:* A = Area ( Entering airstream), $A_1$ = Area (Leaving airstream)
Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–8 Loss Coefficients, Transitions (Diverging Flow) (continued)

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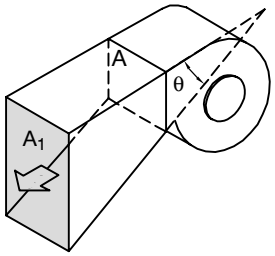
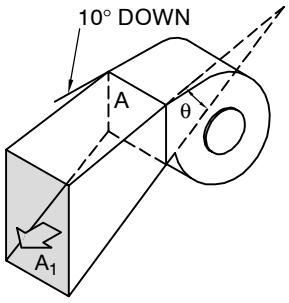
### G. TRANSITION, ASYMMETRIC AT FAN WITH DUCT SIDES STRAIGHT, TOP LEVEL



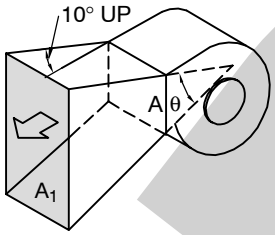

| | Coefficient C (*See* Note 6) | | | | | |
|---|---|---|---|---|---|---|
| $A_1/A$ | 1.5 | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 |
| θ | | | | | | |
| 10° | 0.08 | 0.09 | 0.10 | 0.10 | 0.11 | 0.11 |
| 15° | 0.10 | 0.11 | 0.12 | 0.13 | 0.14 | 0.15 |
| 20° | 0.12 | 0.14 | 0.15 | 0.16 | 0.17 | 0.18 |
| 25° | 0.15 | 0.18 | 0.21 | 0.23 | 0.25 | 0.26 |
| 30° | 0.18 | 0.25 | 0.30 | 0.33 | 0.35 | 0.35 |
| 35° | 0.21 | 0.31 | 0.38 | 0.41 | 0.43 | 0.44 |

### H. TRANSITION, ASYMMETRIC AT FAN WITH DUCT SIDES STRAIGHT, TOP 10° DOWN



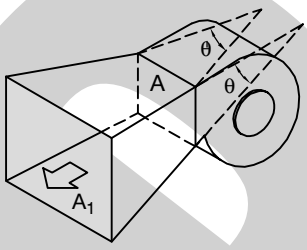

| | Coefficient C (*See* Note 6) | | | | | |
|---|---|---|---|---|---|---|
| $A_1/A$ | 1.5 | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 |
| θ | | | | | | |
| 10° | 0.11 | 0.13 | 0.14 | 0.14 | 0.14 | 0.14 |
| 15° | 0.13 | 0.15 | 0.16 | 0.17 | 0.18 | 0.18 |
| 20° | 0.19 | 0.22 | 0.24 | 0.26 | 0.28 | 0.30 |
| 25° | 0.29 | 0.32 | 0.35 | 0.37 | 0.39 | 0.40 |
| 30° | 0.36 | 0.42 | 0.46 | 0.49 | 0.51 | 0.51 |
| 35° | 0.44 | 0.54 | 0.61 | 0.64 | 0.66 | 0.66 |

### I. TRANSITION, ASYMMETRIC AT FAN WITH DUCT SIDES STRAIGHT, TOP 10° UP




| | Coefficient C (*See* Note 6) | | | | | |
|---|---|---|---|---|---|---|
| $A_1/A$ | 1.5 | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 |
| θ | | | | | | |
| 10° | 0.05 | 0.08 | 0.11 | 0.13 | 0.13 | 0.14 |
| 15° | 0.06 | 0.10 | 0.12 | 0.14 | 0.15 | 0.15 |
| 20° | 0.07 | 0.11 | 0.14 | 0.15 | 0.16 | 0.16 |
| 25° | 0.09 | 0.14 | 0.18 | 0.20 | 0.21 | 0.22 |
| 30° | 0.13 | 0.18 | 0.23 | 0.26 | 0.28 | 0.29 |
| 35° | 0.15 | 0.23 | 0.28 | 0.33 | 0.35 | 0.36 |

### J. TRANSITION, PYRAMIDAL AT FAN WITH DUCT




| | Coefficient C (*See* Note 6) | | | | | |
|---|---|---|---|---|---|---|
| $A_1/A$ | 1.5 | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 |
| θ | | | | | | |
| 10° | 0.10 | 0.18 | 0.21 | 0.23 | 0.24 | 0.25 |
| 15° | 0.23 | 0.33 | 0.38 | 0.40 | 0.42 | 0.44 |
| 20° | 0.31 | 0.43 | 0.48 | 0.53 | 0.56 | 0.58 |
| 25° | 0.36 | 0.49 | 0.55 | 0.58 | 0.62 | 0.64 |
| 30° | 0.42 | 0.53 | 0.59 | 0.64 | 0.67 | 0.69 |

*Note 6:* A = Area ( Entering airstream), $A_1$ = Area (Leaving airstream)

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

**Table A–8 Loss Coefficients, Transitions (Diverging Flow) (continued)**

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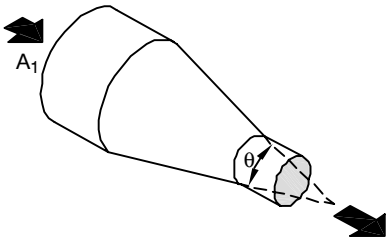
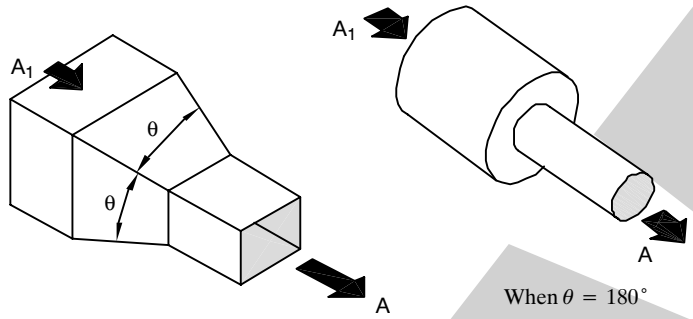
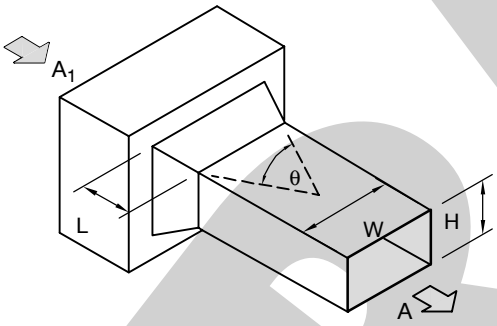
## A. CONTRACTION, ROUND AND RECTANGULAR, GRADUAL TO ABRUPT

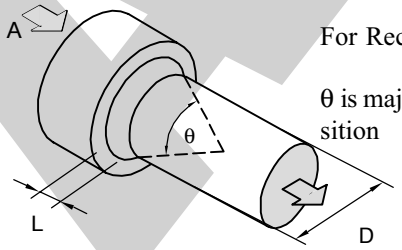



| Coefficient C (*See* Note 3) | | | | | | | |
|---|---|---|---|---|---|---|---|
| $A_1/A$ | $\theta$ | | | | | | |
| | 10° | 15°–40° | 50°–60° | 90° | 120° | 150° | 180° |
| 2 | 0.05 | 0.05 | 0.06 | 0.12 | 0.18 | 0.24 | 0.26 |
| 4 | 0.05 | 0.04 | 0.07 | 0.17 | 0.27 | 0.35 | 0.41 |
| 6 | 0.05 | 0.04 | 0.07 | 0.18 | 0.28 | 0.36 | 0.42 |
| 10 | 0.05 | 0.05 | 0.08 | 0.19 | 0.29 | 0.37 | 0.43 |




## B. CONTRACTION, CONICAL, ROUND AND RECTANGULAR




Coefficient C = K $C_o$

For Rectangular: $D = \frac{2HW}{H + W}$

θ is major angle for rectangular transition

| (*See* Note 7) | | | | | | | |
|---|---|---|---|---|---|---|---|
| A/A1 | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 0.9 | 1.0 |
| K | 1.0 | 0.85 | 0.68 | 0.50 | 0.30 | 0.18 | 0 |

| $C_o$ | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| L/D | θ | | | | | | | | |
| | 0° | 10° | 20° | 30° | 40° | 60° | 100° | 140° | 180° |
| 0.025 | 0.50 | 0.47 | 0.45 | 0.43 | 0.41 | 0.40 | 0.42 | 0.45 | 0.50 |
| 0.05 | 0.50 | 0.45 | 0.41 | 0.36 | 0.33 | 0.30 | 0.35 | 0.42 | 0.50 |
| 0.075 | 0.50 | 0.42 | 0.35 | 0.30 | 0.26 | 0.23 | 0.30 | 0.40 | 0.50 |
| 0.10 | 0.50 | 0.39 | 0.32 | 0.25 | 0.22 | 0.18 | 0.27 | 0.38 | 0.50 |
| 0.15 | 0.50 | 0.37 | 0.27 | 0.20 | 0.16 | 0.15 | 0.25 | 0.37 | 0.50 |
| 0.60 | 0.50 | .0.27 | 0.18 | 0.13 | 0.11 | 0.12 | 0.23 | 0.36 | 0.50 |

*Note 7:* $A_1$ = Area ( Entering airstream), A= Area (Leaving airstream)

Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–9 Loss Coefficients, Transitions (Converging Flow)

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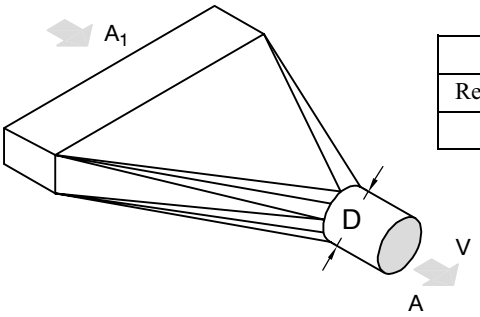
### C. CONTRACTION, RECTANGULAR SLOT TO ROUND




| Coefficient C (*See* Note 7) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Re $10^{-4}$ | 1 | 2 | 4 | 6 | 8 | 10 | 20 | ≥40 |
| C | 0.27 | 0.25 | 0.20 | 0.17 | 0.14 | 0.11 | 0.04 | 0 |

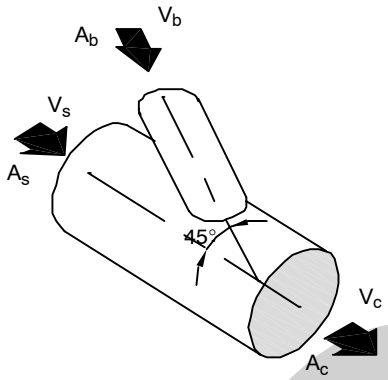
$A < A_1$

| U.S. Units | Metric Units |
|---|---|
| Re = 8.56 DV | 66.4 DV |

*Note 7:* $A_1$ = Area ( Entering airstream), A= Area (Leaving airstream)

## Table A–9 Loss Coefficients, Transitions (Converging Flow) (continued)

### A. CONVERGING WYE, ROUND




| Branch, Coefficient C (*See* Note 8) | | | | | | | |
|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | $A_b/A_c$ | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.6 | 0.8 | 1.0 |
| 0.4 | -0.56 | -0.44 | -0.35 | -0.28 | -0.15 | -0.04 | 0.05 |
| 0.5 | -0.48 | -0.37 | -0.28 | -0.21 | -0.09 | 0.02 | 0.11 |
| 0.6 | -0.38 | -0.27 | -0.19 | -0.12 | 0 | 0.10 | 0.18 |
| 0.7 | -0.26 | -0.16 | -0.08 | -0.01 | 0.10 | 0.20 | 0.28 |
| 0.8 | -0.21 | -0.02 | 0.05 | 0.12 | 0.23 | 0.32 | 0.40 |
| 0.9 | 0.04 | 0.13 | 0.21 | 0.27 | 0.37 | 0.46 | 053 |
| 1.0 | 0.22 | 0.31 | 0.38 | 0.44 | 0.53 | 0.62 | 0.69 |
| 1.5 | 1.4 | 1.5 | 1.5 | 1.6 | 1.7 | 1.7 | 1.8 |
| 2.0 | 3.1 | 3.2 | 3.2 | 3.2 | 3.3 | 3.3 | 3.3 |
| 2.5 | 5.3 | 5.3 | 5.3 | 5.4 | 5.4 | 5.4 | 5.4 |
| 3.0 | 8.0 | 8.0 | 8.0 | 8.0 | 8.0 | 8.0 | 8.0 |

| Main Coefficient C | | | | | | | |
|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | $A_b/A_c$ | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.6 | 0.8 | 1.0 |
| 0.1 | -8.6 | -4.1 | -2.5 | -1.7 | -0.97 | -0.58 | -0.34 |
| 0.2 | -6.7 | -3.1 | -1.9 | -1.3 | -0.67 | -0.36 | -0.18 |
| 0.3 | -5.0 | -2.2 | -1.3 | -0.88 | -0.42 | -0.19 | -0.05 |
| 0.4 | -3.5 | -1.5 | -0.88 | -0.55 | -0.21 | -0.05 | 0.05 |
| 0.5 | -2.3 | -0.95 | -0.51 | -0.28 | -0.06 | 0.06 | 0.13 |
| 0.6 | -1.3 | -0.50 | -0.22 | -0.09 | 0.05 | 0.12 | 0.17 |
| 0.7 | -0.63 | -0.18 | -0.03 | 0.04 | 0.12 | 0.16 | 0.18 |
| 0.8 | -0.18 | 0.01 | 0.07 | 0.10 | 0.13 | 0.15 | 0.17 |
| 0.9 | 0.03 | 0.07 | 0.08 | 0.09 | 0.10 | 0.11 | 0.13 |
| 1.0 | -0.01 | 0 | 0 | 0.10 | 0.02 | 0.04 | 0.05 |

*Note 8*: A = Area (sq. in.). Q = Airflow (cfm). V = Velocity (fpm)

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–10 Loss Coefficients, Converging Junctions (Tees, Wyes)

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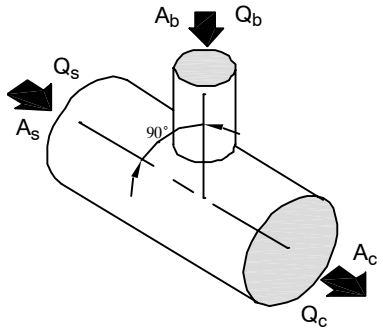
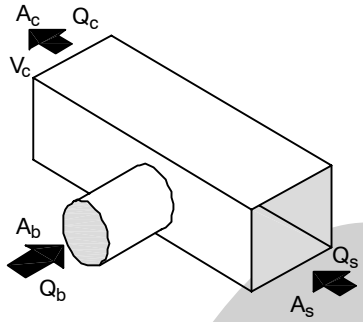
### B. CONVERGING TEE, 90°, ROUND




| Branch, Coefficient C (*See* Note 8) | | | | | | | |
|---|---|---|---|---|---|---|---|
| $O_b/Q_c$ | $A_b/A_c$ | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.6 | 0.8 | 1.0 |
| 0.1 | 0.40 | -0.37 | -0.51 | -0.46 | -0.50 | -0.51 | -0.52 |
| 0.2 | 3.8 | 0.72 | 0.17 | -0.02 | -0.14 | -0.18 | -0.24 |
| 0.3 | 9.2 | 2.3 | 1.0 | 0.44 | 0.21 | 0.11 | -0.08 |
| 0.4 | 16 | 4.3 | 2.1 | 0.94 | 0.54 | 0.40 | 0.32 |
| 0.5 | 26 | 6.8 | 3.2 | 1.1 | 0.66 | 0.49 | 0.42 |
| 0.6 | 37 | 9.7 | 4.7 | 1.6 | 0.92 | 0.69 | 0.57 |
| 0.7 | 43 | 13 | 6.3 | 2.1 | 1.2 | 0.88 | 0.72 |
| 0.8 | 65 | 17 | 7.9 | 2.7 | 1.5 | 1.1 | 0.86 |
| 0.9 | 82 | 21 | 9.7 | 3.4 | 1.8 | 1.2 | 0.99 |
| 1.0 | 101 | 26 | 12 | 4.0 | 2.1 | 1.4 | 1.1 |

| Main, Coefficient C (*See* Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $O_b/Q_c$ | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 |
| C | 0.16 | 0.27 | 0.38 | 0.46 | 0.53 | 0.57 | 0.59 | 0.60 | 0.59 | 0.55 |

### C. CONVERGING TEE, ROUND BRANCH TO RECTANGULAR MAIN

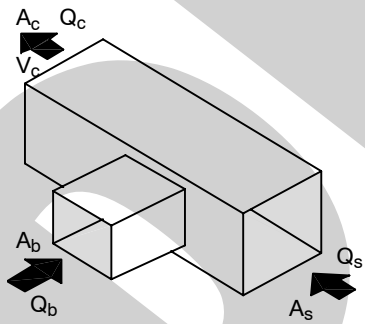



For Main Loss Coefficient (C) *see* A-10B.

| When: | $A_b/A_s$ | $A_s/A_c$ | $A_b/A_c$ |
|---|---|---|---|
| | 0.5 | 1.0 | 0.5 |

| Branch, Coefficient C (See Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $V_c$ | $Q_b/Q_c$ | | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 |
| < 1200 fpm | -0.63 | -0.55 | -0.13 | 0.23 | 0.78 | 1.30 | 1.93 | 3.10 | 4.88 | 5.60 |
| > 1200 fpm | -0.49 | -0.21 | -0.23 | 0.60 | 1.27 | 2.06 | 2.75 | 3.70 | 4.93 | 5.95 |

### D. CONVERGING TEE, RECTANGULAR MAIN AND BRANCH




| When: | $A_b/A_s$ | $A_s/A_c$ | $A_s/A_c$ |
|---|---|---|---|
| | 0.5 | 1.0 | 0.5 |

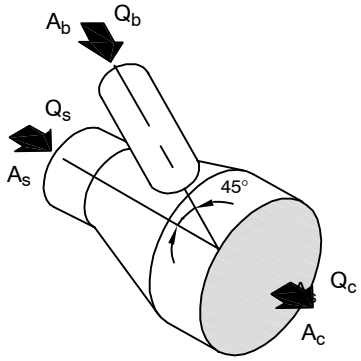
| Branch, Coefficient C (See Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $V_c$ | $Q_b/Q_c$ | | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 |
| < 1200 fpm | -0.75 | -0.53 | -0.03 | 0.33 | 1.03 | 1.10 | 2.15 | 2.93 | 4.18 | 4.78 |
| > 1200 fpm | -0.69 | -0.21 | -0.23 | 0.67 | 1.17 | 1.66 | 2.67 | 3.36 | 3.93 | 5.13 |

For Main Loss Coefficient (C) *see* A-10B.

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

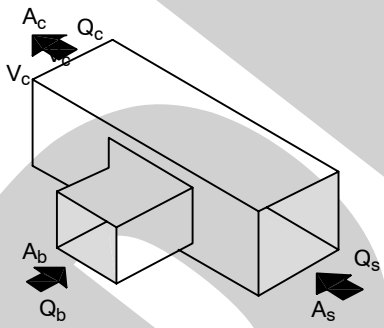
## Table A–10 Loss Coefficients, Converging Junctions (Tees, Wyes) (continued)

## E. CONVERGING WYE, CONICAL ROUND




| Main, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $A_s/A_c$ | $A_b/A_c$ | $Q_b/Qs$ | | | | | | | | | |
| | | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| 0.3 | 0.2 | -2.4 | -0.01 | 2.0 | 3.8 | 5.3 | 6.6 | 7.8 | 8.9 | 9.8 | 11 |
| | 0.3 | -2.8 | -1.2 | 0.12 | 1.1 | 1.9 | 2.6 | 3.2 | 3.7 | 4.2 | 4.6 |
| 0.4 | 0.2 | -1.2 | 0.93 | 2.8 | 4.5 | 5.9 | 7.2 | 8.4 | 9.5 | 10 | 11 |
| | 0.3 | -1.6 | -0.27 | 0.81 | 1.7 | 2.4 | 3.0 | 3.6 | 4.1 | 4.5 | 4.9 |
| | 0.4 | -1.8 | -0.72 | 0.07 | 0.66 | 1.1 | 1.5 | 1.8 | 2.1 | 2.3 | 2.5 |
| 0.5 | 0.2 | -.046 | 1.5 | 3.3 | 4.9 | 6.4 | 7.7 | 8.8 | 9.9 | 11 | 12 |
| | 0.3 | -.094 | 0.25 | 1.2 | 2.0 | 2.7 | 3.3 | 3.8 | 4.2 | 4.7 | 5.0 |
| | 0.4 | -1.1 | -0.24 | 0.42 | 0.92 | 1.3 | 1.6 | 1.9 | 2.1 | 2.3 | 2.5 |
| | 0.5 | -1.2 | -0.38 | 0.18 | 0.58 | 0.88 | 1.1 | 1.3 | 1.5 | 1.6 | 1.7 |
| 0.6 | 0.2 | -0.55 | 1.3 | 3.1 | 4.4 | 6.1 | 7.4 | 6.6 | 9.6 | 11 | 12 |
| | 0.3 | -1.1 | 0 | 0.88 | 1.6 | 2.3 | 2.8 | 3.3 | 3.7 | 4.1 | 4.5 |
| | 0.4 | -1.2 | -0.48 | 0.10 | 0.54 | 0.89 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| | 0.5 | -1.3 | -0.62 | -0.14 | 0.21 | 0.47 | 0.68 | 0.85 | 0.99 | 1.1 | 1.2 |
| | 0.6 | -1.3 | -0.69 | -0.26 | 0.04 | 0.26 | 0.42 | 0.57 | 0.66 | 0.75 | 0.82 |
| 0.8 | 0.2 | 0.06 | 1.8 | 3.5 | 5.1 | 6.5 | 7.8 | 8.9 | 10 | 11 | 12 |
| | 0.3 | -0.52 | 0.35 | 1.1 | 1.7 | 2.3 | 2.8 | 3.2 | 3.6 | 3.9 | 4.2 |
| | 0.4 | -0.67 | -0.05 | 0.43 | 0.80 | 1.1 | 1.4 | 1.6 | 1.8 | 1.9 | 2.1 |
| | 0.5 | -0.73 | -0.19 | 0.18 | 0.46 | 0.68 | 0.85 | 0.99 | 1.1 | 1.2 | 1.3 |
| | 0.6 | -0.75 | -0.27 | 0.05 | 0.28 | 0.45 | 0.58 | 0.68 | 0.76 | 0.83 | 0.88 |
| | 0.7 | -0.77 | -0.31 | -0.02 | 0.18 | 0.32 | 0.43 | 0.50 | 0.56 | 0.61 | 0.65 |
| | 0.8 | -0.78 | -0.34 | -0.07 | 0.12 | 0.24 | 0.33 | 0.39 | 0.44 | 0.47 | 0.50 |
| 1.0 | 0.2 | — | 2.1 | 3.7 | 5.2 | 6.6 | 7.8 | 9.0 | 11 | 11 | 12 |
| | 0.3 | — | 0.54 | 1.2 | 1.8 | 2.3 | 2.7 | 3.1 | 3.7 | 3.7 | 4.0 |
| | 0.4 | — | 0.21 | 0.62 | 0.96 | 1.2 | 1.5 | 1.7 | 2.0 | 2.0 | 2.1 |
| | 0.5 | — | 0.05 | 0.37 | 0.60 | 0.79 | 0.93 | 1.1 | 1.2 | 1.2 | 1.3 |
| | 0.6 | — | -0.02 | 0.23 | 0.42 | 0.55 | 0.66 | 0.73 | 0.80 | 0.85 | 0.89 |
| | 0.8 | — | -0.10 | 0.11 | 0.24 | 0.33 | 0.39 | 0.43 | 0.46 | 0.47 | 0.48 |
| | 1.0 | — | -0.14 | 0.05 | 0.16 | 0.23 | 0.27 | 0.29 | 0.30 | 0.30 | 0.29 |

## F. CONVERGING TEE, 45° ENTRY BRANCH TO RECTANGULAR MAIN




| When: | $A_b/A_s$ | $A_s/A_c$ | $A_s/A_s$ |
|---|---|---|---|
| | 0.5 | 1.0 | 0.5 |

| Branch, Coefficient C (*See* Note 4) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $V_c$ | $Q_b/Q_c$ | | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 |
| < 1200 fpm | -0.83 | -0.68 | -0.30 | 0.28 | 0.55 | 1.03 | 1.50 | 1.93 | 2.50 | 3.03 |
| > 1200 fpm | -0.72 | -0.52 | -0.23 | 0.34 | 0.76 | 1.14 | 1.83 | 2.01 | 2.90 | 3.63 |

For Main Loss Coefficient (C) see A-10B.

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

**Table A–10 Loss Coefficients, Converging Junctions (Tees, Wyes) (continued)**

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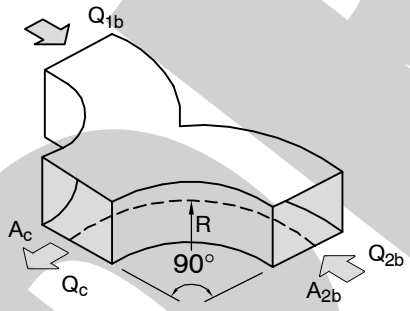
| Main, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $A_s/A_c$ | $A_b/A_c$ | $Q_b/Q_s$ | | | | | | | | | |
| | | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| 0.3 | 0.2 | 5.3 | -.01 | 2.0 | 1.1 | 0.34 | -.20 | -.61 | -.93 | -1.2 | -1.4 |
| | 0.3 | 5.4 | 3.7 | 2.5 | 1.6 | 1.0 | 0.53 | 0.16 | -.14 | -.38 | -.58 |
| 0.4 | 0.2 | 1.9 | 1.1 | 0.46 | -.07 | -.49 | -.83 | -1.1 | -1.3 | -1.5 | -1.7 |
| | 0.3 | 2.0 | 1.4 | 0.81 | 0.42 | 0.08 | -.20 | -.43 | -.62 | -.78 | -.92 |
| | 0.4 | 2.0 | 1.5 | 1.0 | 0.68 | 0.39 | 0.16 | -.04 | -.21 | -.35 | -.47 |
| 0.5 | 0.2 | 0.77 | 0.34 | -.09 | -.48 | -.81 | -1.1 | 1.3 | -1.5 | -1.7 | -1.8 |
| | 0.3 | 0.85 | 0.56 | 0.25 | -.03 | -.27 | -.48 | -.67 | -.82 | -.96 | -1.1 |
| | 0.4 | 0.88 | 0.66 | 0.43 | 0.21 | 0.02 | -.15 | -.30 | -.42 | -.54 | -.64 |
| | 0.5 | 0.91 | 0.73 | 0.54 | 0.36 | 0.21 | 0.06 | -06 | -.17 | -.26 | -.35 |
| 0.6 | 0.2 | 0.30 | 0 | -.34 | -.67 | -.96 | -1.2 | -1.4 | -1.6 | -1.8 | -1.9 |
| | 0.3 | 0.37 | 0.21 | -.02 | -.24 | -.44 | -.63 | -.79 | -.93 | -1.1 | -1.2 |
| | 0.4 | 0.40 | 0.31 | 0.16 | -.01 | -.16 | -.30 | -.43 | -.54 | -.64 | -.73 |
| | 0.5 | 0.43 | 0.37 | 0.26 | 0.14 | 0.02 | -.09 | -.20 | -.29 | -.37 | -.45 |
| 0.8 | 0.6 | 0.44 | 0.41 | 0.33 | 0.24 | 0.14 | 0.05 | -.03 | -.11 | -.18 | -.25 |
| | 0.2 | -.06 | -.27 | -.57 | -.86 | -1.1 | -1.4 | -1.6 | -1.7 | -1.9 | -2.0 |
| | 0.3 | 0 | -.08 | -.25 | -.43 | -.62 | -.78 | -.93 | -1.1 | -1.2 | -1.3 |
| | 0.4 | 0.04 | 0.02 | -.08 | -.21 | -.34 | -.46 | -.57 | -.67 | -.77 | -.85 |
| | 0.5 | 0.06 | 0.08 | 0.02 | -.06 | -.16 | -.25 | -.34 | -.42 | -.50 | -.57 |
| | 0.6 | 0.07 | 0.12 | 0.09 | 0.03 | -.04 | -.11 | -.18 | -.25 | -.31 | -.37 |
| | 0.7 | 0.08 | 0.15 | 0.14 | 0.10 | 0.05 | -.01 | -.07 | -.12 | -.17 | -.22 |
| | 0.8 | 0.09 | 0.17 | 0.18 | 0.16 | 0.11 | 0.07 | 0.02 | -.02 | -.07 | -.11 |
| 1.0 | 0.2 | — | -.39 | -.67 | -.96 | -1.2 | -1.5 | -1.6 | -1.8 | -2.0 | -2.1 |
| | 0.3 | — | -.19 | -.35 | -.54 | -.71 | -.87 | -1.0 | -1.2 | -1.3 | -1.4 |
| | 0.4 | — | -.10 | -.19 | -.31 | -.43 | -.55 | -.66 | -.77 | -.86 | -.94 |
| | 0.5 | — | -.04 | -.09 | -.17 | -.26 | -.35 | -.44 | -.52 | -.59 | -.66 |
| | 0.6 | — | 0 | -.02 | -.07 | -.14 | -.21 | -.28 | -.34 | -.40 | -.46 |
| | 0.8 | — | 0.06 | 0.07 | 0.05 | 0.02 | -.03 | -.07 | -.12 | -.16 | -.20 |
| | 1.0 | — | 0.09 | 0.13 | 0.13 | 0.11 | 0.08 | 0.06 | 0.03 | -.01 | -.03 |

### G. SYMMETRICAL WYE, DOVETAIL, RECTANGULAR




| Coefficient C (*See* Note 8) | | |
|---|---|---|
| $A_{1b}/A_c$ or $A_{2b}/A_c$ | 0.50 | 1.0 |
| C | 0.23 | 0.07 |

$$\frac{R}{W} = 1.5 \qquad \frac{Q_{1b}}{W_c} = \frac{Q_{2b}}{W_c} = 0.5$$

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–10 Loss Coefficients, Converging Junctions (Tees, Wyes) (continued)

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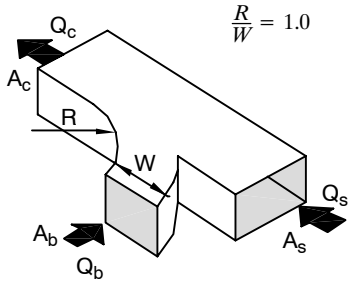
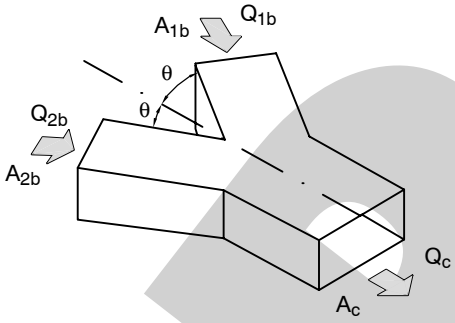
### H. CONVERGING WYE, RECTANGULAR

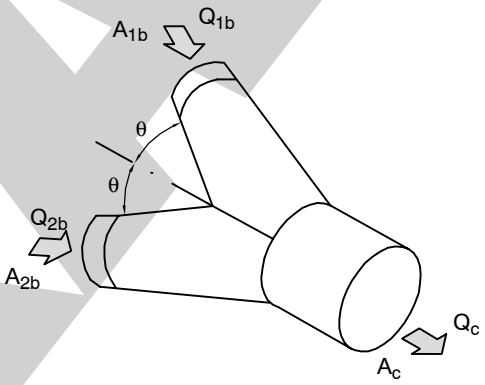



| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $A_b/A_s$ | $A_b/A_c$ | $Q_b/Q_c$ | | | | | | | | |
| | | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.25 | 0.25 | -0.50 | 0 | 0.50 | 1.2 | 2.2 | 3.7 | 5.8 | 8.4 | 11 |
| 0.33 | 0.25 | -1.2 | -0.40 | 0.40 | 1.6 | 3.0 | 4.8 | 6.8 | 8.9 | 11 |
| 0.5 | 0.5 | -0.50 | -0.20 | 0 | 0.25 | 0.45 | 0.70 | 1.0 | 1.5 | 2.0 |
| 0.67 | 0.5 | -1.0 | -0.60 | -0.20 | 0.10 | 0.30 | 0.60 | 1.0 | 1.5 | 2.0 |
| 1.0 | 0.5 | -2.2 | -1.5 | -0.95 | -0.50 | 0 | 0.40 | 0.80 | 1.3 | 1.9 |
| 1.0 | 1.0 | -0.60 | -0.30 | -0.10 | -0.04 | 0.13 | 0.21 | 0.29 | 0.36 | 0.42 |
| 1.33 | 1.0 | -1.2 | -0.80 | -0.40 | -0.20 | 0 | 0.16 | 0.24 | 0.32 | 0.38 |
| 2.0 | 1.0 | -2.1 | -1.4 | -0.90 | -0.5 | -.20 | 0 | 0.20 | 0.25 | 0.30 |

| Main, Coefficient C (*See* Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $A_S/A_C$ | $Ab/A_C$ | $Q_b/Q_c$ | | | | | | | | |
| | | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.75 | 025 | 30 | 30 | 0.20 | -0.1 | -0.45 | -0.92 | -1.5 | -.20 | -.26 |
| 1.05 | 0.5 | 0.17 | 0.16 | 0.10 | 0 | -0.08 | -0.18 | -.027 | -.37 | -.46 |
| 0.75 | 0.5 | 0.27 | 0.35 | 0.32 | 0.25 | 0.12 | -0.03 | -0.23 | -.42 | -.58 |
| 0.5 | 0.5 | 1.2 | 1.1 | 0.90 | 0.65 | 0.35 | 0 | -0.40 | -.80 | -1.3 |
| 1.0 | 1.0 | 0.18 | 0.24 | 0.27 | 0.26 | 0.23 | 0.18 | 0.10 | 0 | -.12 |
| 0.75 | 1.0 | 0.75 | 0.36 | 0.38 | 0.35 | 0.27 | 0.18 | 0.05 | -.08 | -.22 |
| 0.5 | 1.0 | 0.80 | 0.87 | 0.80 | 0.68 | 0.55 | 0.40 | 0.25 | 0.08 | -.10 |

### I. WYE, RECTANGULAR AND ROUND




| θ | Coefficient C (*See* Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | $Q_{1b}/Q_c$ or $Q_{2b}/Q_c$ | | | | | | | | | | |
| | 0 | 10 | 0.20 | 0.30 | 0.40 | 0.50 | 0.60 | 0.70 | 0.80 | 0.90 | 1.0 |
| 15° | -2.6 | -19 | -13 | -.77 | -.30 | 0.10 | 041 | 0.67 | 0.85 | 097 | 1.0 |
| 30° | -2.1 | -15 | -10 | -.53 | -.1 | 0.28 | 069 | 0.91 | 1.1 | 1.4 | 1.6 |
| 45° | -1.3 | -.93 | -.55 | -.16 | 0.20 | 0.56 | 0.92 | 1.26 | 1.6 | 2.0 | 2.3 |

When:
$A_{1b} = A_{2b}$
$A_c = A_{1b} + A_{2b}$
*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–10 Loss Coefficients, Converging Junctions (Tees, Wyes) (continued)

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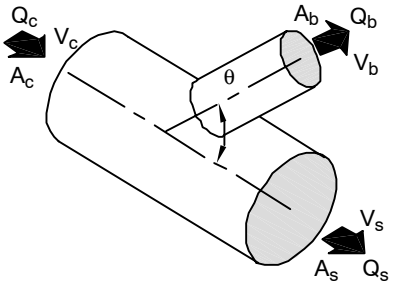
## A. TEE OR WYE, 30° DEGREES TO 90°, ROUND




| Main, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_S/V_c$ | 0 | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.8 | 1.0 |
| C | 0.35 | 0.28 | 0.22 | 0.17 | 0.13 | 0.09 | 0.06 | 0.02 | 0 |

Wye $\theta = 30°$

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $A_b/A_c$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.8 | 0.95 | 0.92 | 0.92 | 0.93 | 0.94 | 0.95 | 1.1 | 1.2 | 1.4 |
| 0.7 | 0.95 | 0.94 | 0.95 | 0.98 | 1.0 | 1.1 | 1.2 | 1.4 | 1.6 |
| 0.6 | 0.96 | 0.97 | 1.0 | 1.1 | 1.1 | 1.2 | 1.4 | 1.7 | 2.0 |
| 0.5 | 0.97 | 1.0 | 1.1 | 1.2 | 1.4 | 1.5 | 1.8 | 2.1 | 2.5 |
| 0.4 | 0.99 | 1.1 | 1.3 | 1.5 | 1.7 | 2.0 | 2.4 | — | — |
| 0.3 | 1.1 | 1.4 | 1.8 | 2.3 | — | — | — | — | — |
| 0.2 | 1.3 | 1.9 | 2.9 | — | — | — | — | — | — |
| 0.1 | 2.1 | — | — | — | — | — | — | — | — |

Wye $\theta = 45°$

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $A_b/A_c$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.8 | 0.75 | 0.55 | 0.40 | 0.28 | 0.21 | 0.16 | 0.15 | 0.16 | 0.19 |
| 0.7 | 0.72 | 0.51 | 0.36 | 0.25 | 0.18 | 0.15 | 0.16 | 0.20 | 0.26 |
| 0.6 | 0.69 | 0.46 | 0.31 | 0.21 | 0.17 | 0.16 | 0.20 | 0.28 | 0.39 |
| 0.5 | 0.65 | 0.41 | 0.26 | 0.19 | 0.18 | 0.22 | 0.32 | 0.47 | 0.67 |
| 0.4 | 0.59 | 0.33 | 0.21 | 0.20 | 0.27 | 0.40 | 0.62 | 0.92 | 1.3 |
| 0.3 | 0.55 | 0.28 | 0.24 | 0.38 | 0.76 | 1.3 | 2.0 | — | — |
| 0.2 | 0.40 | 0.26 | 0.58 | 1.3 | 2.5 | — | — | — | — |
| 0.1 | 0.28 | 1.5 | — | — | — | — | — | — | — |

Wye $\theta = 60°$

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $A_b/Ac$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.8 | 0.83 | 0.71 | 0.62 | 0.56 | 0.52 | 0.50 | 0.53 | 0.60 | 0.68 |
| 0.7 | 0.82 | 0.69 | 0.61 | 0.56 | 0.54 | 0.54 | 0.60 | 0.70 | 0.82 |
| 0.6 | 0.81 | 0.68 | 0.60 | 0.58 | 0.58 | 0.61 | 0.72 | 0.87 | 1.1 |
| 0.5 | 0.79 | 0.66 | 0.61 | 0.62 | 0.68 | 0.76 | 0.94 | 1.2 | 1.5 |
| 0.4 | 0.76 | 0.65 | 0.65 | 0.74 | 0.89 | 1.1 | 1.4 | 1.8 | 2.3 |
| 0.3 | 0.80 | 0.75 | 0.89 | 1.2 | 1.8 | 2.6 | 3.5 | — | — |
| 0.2 | 0.77 | 0.96 | 1.6 | 2.5 | — | — | — | — | — |
| 0.1 | 1.0 | 2.9 | — | — | — | — | — | — | — |

Wye $\theta = 90°$

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $A_b/Ac$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.8 | 0.78 | 0.62 | 0.49 | 0.40 | 0.34 | 0.31 | 0.32 | 0.35 | 0.40 |
| 0.7 | 0.77 | 0.59 | 0.47 | 0.38 | 0.34 | 0.32 | 0.35 | 0.41 | 0.50 |
| 0.6 | 0.74 | 0.56 | 0.44 | 0.37 | 0.35 | 0.36 | 0.43 | 0.54 | 0.68 |
| 0.5 | 0.71 | 0.52 | 0.41 | 0.38 | 0.40 | 0.45 | 0.59 | 0.78 | 1.0 |
| 0.4 | 0.66 | 0.47 | 0.40 | 0.43 | 0.54 | 0.69 | 0.95 | 1.3 | 1.7 |
| 0.3 | 0.66 | 0.48 | 0.52 | 0.73 | 1.2 | 1.8 | 2.7 | — | — |
| 0.2 | 0.56 | 0.56 | 1.0 | 1.8 | — | — | — | — | — |
| 0.1 | 0.60 | 2.1 | — | — | — | — | — | — | — |

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–11 Loss Coefficients, Diverging Junctions (Tees, Wyes)

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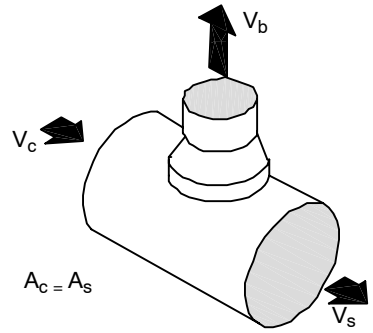
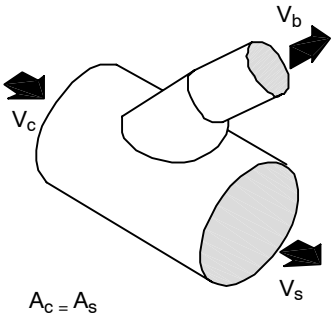
### B. 90° CONICAL TEE, ROUND



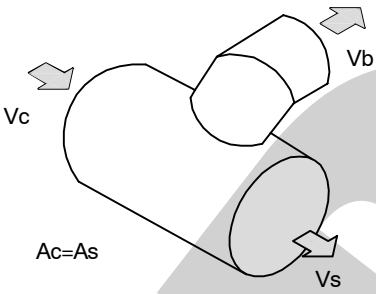

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 0.85 | 0.74 | 0.62 | 0.52 | 0.42 | 0.36 | 0.32 | 0.32 | 0.37 | 0.52 |

### C. 45° CONICAL WYE, ROUND



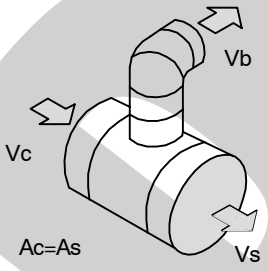

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 0.84 | 0.61 | 0.41 | 0.27 | 0.17 | 0.12 | 0.12 | 0.14 | 0.18 | 0.27 |

### D. 90° TEE, ROUND, ROLLED 45° WITH 45° ELBOW, BRANCH 90° TO MAIN




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 1.32 | 1.51 | 1.60 | 1.65 | 1.74 | 1.87 | 2.0 | 2.2 | 2.5 | 2.7 |

### E. 90° TEE, ROUND, WITH 90° ELBOW, BRANCH 90° TO MAIN




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 1.03 | 1.08 | 1.18 | 1.33 | 1.56 | 1.86 | 2.2 | 2.6 | 3.0 | 3.4 |

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–11 Loss Coefficients, Diverging Junctions (Tees, Wyes) (Continued)

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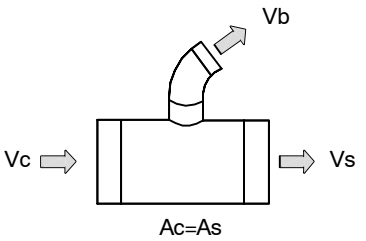
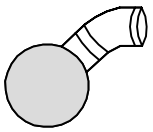
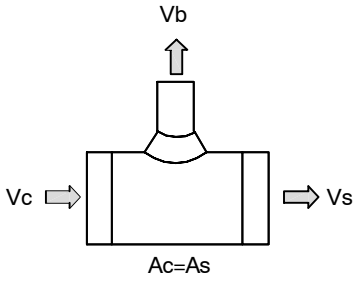
### F. 90° TEE, ROUND, ROLLED 45° WITH 60° ELBOW, BRANCH 45° TO MAIN

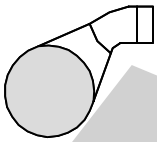


| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 1.06 | 1.15 | 1.29 | 1.45 | 1.65 | 1.89 | 2.2 | 2.5 | 2.9 | 3.3 |

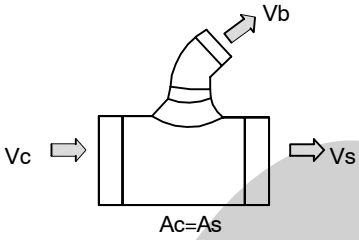
### G. 90° CONICAL TEE, ROUND, ROLLED 45° WITH 45° ELBOW, BRANCH 90° TO MAIN

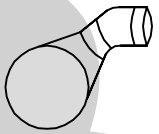


| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 0.94 | 0.88 | 0.84 | 0.80 | 0.82 | 0.84 | 0.87 | 0.90 | 0.95 | 1.02 |

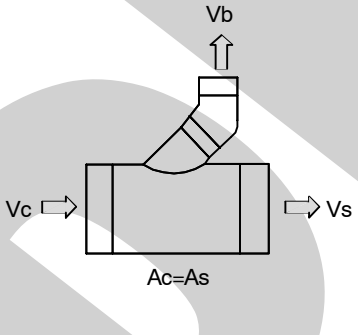
### H. 90° CONICAL TEE, ROUND, ROLLED 45° WITH 60° ELBOW, BRANCH 45° TO MAIN

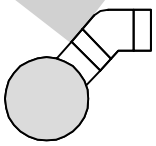


| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.8 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 0.95 | 0.90 | 0.86 | 0.81 | 0.79 | 0.79 | 0.81 | 0.86 | 0.96 | 1.10 |

### I. 45° WYE, ROUND, ROLLED 45° WITH 60° ELBOW, BRANCH 90° TO MAIN



| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 0.88 | 0.77 | 0.68 | 0.65 | 0.69 | 0.73 | 0.88 | 1.14 | 1.54 | 2.2 |

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–11 Loss Coefficients, Diverging Junctions (Tees, Wyes) (Continued)

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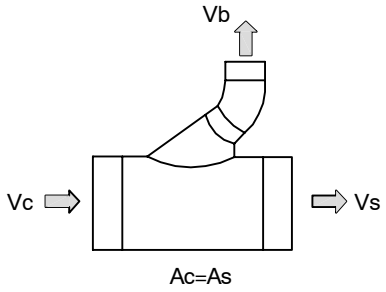
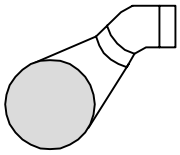
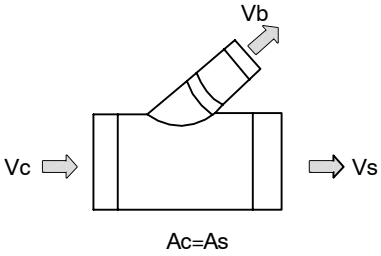
### J. 45° CONICAL WYE ROUND, ROLLED 45° WITH 60° ELBOW, BRANCH 90° TO MAIN



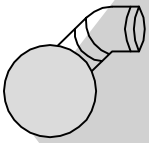

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 0.82 | 0.63 | 0.52 | 0.45 | 0.42 | 0.41 | 0.40 | 0.41 | 0.45 | 0.56 |



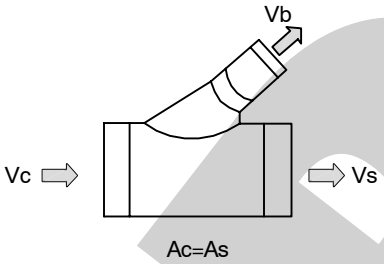
### K. 45° WYE, ROUND, ROLLED 45° WITH 30° ELBOW, BRANCH 45° TO MAIN



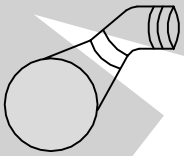

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 0.84 | 0.72 | 0.62 | 0.54 | 0.50 | 0.56 | 0.71 | 0.92 | 1.22 | 1.66 |



### L. 45° CONICAL WYE, ROUND, ROLLED 45° WITH 30° ELBOW, BRANCH 45° TO MAIN




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0 | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 1.0 | 0.93 | 0.71 | 0.55 | 0.44 | 0.42 | 0.42 | 0.44 | 0.47 | 0.54 | 0.62 |



*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–11 Loss Coefficients, Diverging Junctions (Tees, Wyes) (Continued)

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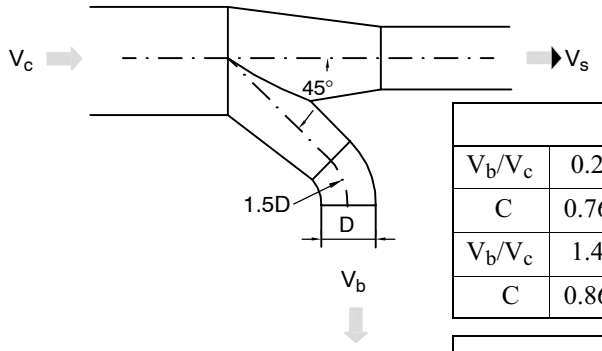
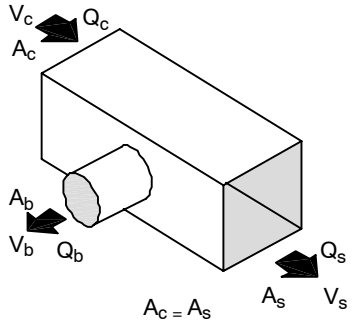
### M. 45° CONICAL MAIN AND BRANCH WITH 45° ELBOW, BRANCH 90° TO MAIN




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0.2 | 0.4 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 | 1.1 | 1.2 |
| C | 0.76 | 0.60 | 0.52 | 0.50 | 0.51 | 0.52 | 0.56 | 2.8 | 3.0 |
| $V_b/V_c$ | 1.4 | 1.6 | 1.8 | 2.0 | 2.2 | 2.4 | 2.6 | 2.8 | 3.0 |
| C | 0.86 | 1.1 | 1.4 | 1.8 | 2.2 | 2.6 | 3.1 | 3.7 | 4.2 |

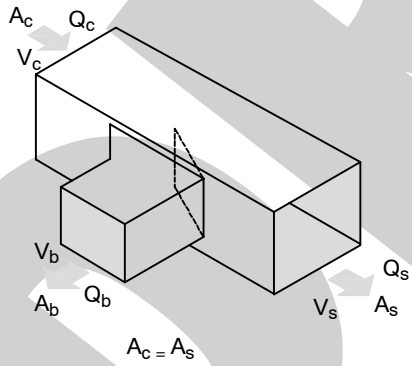
| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| C | 0.14 | 0.06 | 0.05 | 0.09 | 0.18 | 0.30 | 0.46 | 0.64 | 0.84 | 1.0 |

### N. TEE, 45° RECTANGULAR MAIN AND BRANCH




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.2 | 0.91 | | | | | | | | |
| 0.4 | 0.81 | 0.79 | | | | | | | |
| 0.6 | 0.77 | 0.72 | 0.70 | | | | | | |
| 0.8 | 0.78 | 0.73 | 0.69 | 0.66 | | | | | |
| 1.0 | 0.78 | 0.98 | 0.85 | 0.79 | 0.74 | | | | |
| 1.2 | 0.90 | 1.11 | 1.16 | 1.23 | 1.03 | 0.86 | | | |
| 1.4 | 1.19 | 1.22 | 1.26 | 1.29 | 1.54 | 1.25 | 0.92 | | |
| 1.6 | 1.35 | 1.42 | 1.55 | 1.59 | 1.63 | 1.50 | 1.31 | 1.09 | |
| 1.8 | 1.44 | 1.50 | 1.75 | 1.74 | 1.72 | 2.24 | 1.63 | 1.40 | 1.17 |

### O. TEE, 45° ENTRY, RECTANGULAR MAIN AND BRANCH WITH DAMPER




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.2 | 0.61 | | | | | | | | |
| 0.4 | 0.46 | 0.61 | | | | | | | |
| 0.6 | 0.43 | 0.50 | 0.54 | | | | | | |
| 0.8 | 0.39 | 0.43 | 0.62 | 0.53 | | | | | |
| 1.0 | 0.34 | 0.57 | 0.77 | 0.73 | 0.68 | | | | |
| 1.2 | 0.37 | 0.64 | 0.85 | 0.98 | 1.07 | 0.83 | | | |
| 1.4 | 0.57 | 0.71 | 1.04 | 1.16 | 1.54 | 1.36 | 1.18 | | |
| 1.6 | 0.89 | 1.08 | 1.28 | 1.30 | 1.69 | 2.09 | 1.81 | 1.47 | |
| 1.8 | 1.33 | 1.34 | 2.04 | 1.78 | 1.90 | 2.40 | 2.77 | 2.23 | 1.92 |

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–11 Loss Coefficients, Diverging Junctions (Tees, Wyes) (Continued)

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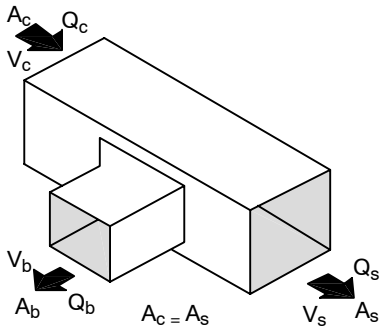
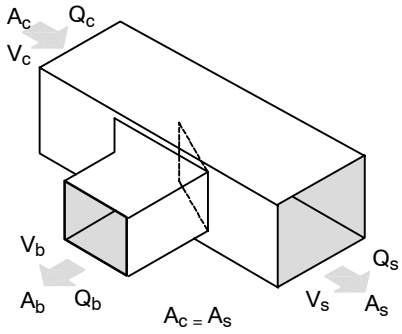
### P. TEE RECTANGULAR MAIN AND BRANCH



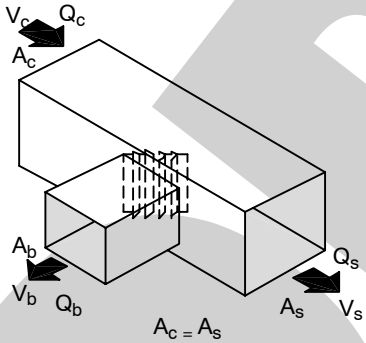

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.2 | 1.03 | | | | | | | | |
| 0.4 | 1.04 | 1.01 | | | | | | | |
| 0.6 | 1.11 | 1.03 | 1.05 | | | | | | |
| 0.8 | 1.16 | 1.21 | 1.17 | 1.12 | | | | | |
| 1.0 | 1.38 | 1.40 | 1.30 | 1.36 | 1.27 | | | | |
| 1.2 | 1.52 | 1.61 | 1.68 | 191 | 1.47 | 1.66 | | | |
| 1.4 | 1.79 | 2.01 | 1.90 | 2.31 | 2.28 | 2.20 | 1.95 | | |
| 1.6 | 2.07 | 2.28 | 2.13 | 2.71 | 2.99 | 2.81 | 2.09 | 2.20 | |
| 1.8 | 2.32 | 2.54 | 2.64 | 3.09 | 3.72 | 3.48 | 2.21 | 2.29 | 2.57 |

### Q. TEE RECTANGULAR MAIN AND BRANCH WITH DAMPER




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.2 | 0.58 | | | | | | | | |
| 0.4 | 0.67 | 0.64 | | | | | . | | |
| 0.6 | 0.78 | 0.76 | 0.75 | | | | | | |
| 0.8 | 0.88 | 0.98 | 0.81 | 1.01 | | | | | |
| 1.0 | 1.12 | 1.05 | 1.08 | 1.18 | 1.29 | | | | |
| 1.2 | 1.49 | 1.48 | 1.40 | 1.51 | 1.70 | 1.91 | | | |
| 1.4 | 2.10 | 2.21 | 2.25 | 2.29 | 2.32 | 2.48 | 2.53 | | |
| 1.6 | 2.72 | 3.30 | 2.84 | 3.09 | 3.30 | 3.19 | 3.29 | 3.16 | |
| 1.8 | 3.42 | 4.58 | 3.65 | 3.92 | 4.20 | 4.15 | 4.14 | 4.10 | 4.05 |

### R. TEE RECTANGULAR MAIN AND BRANCH WITH EXTRACTOR




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.2 | 0.60 | | | | | | | | |
| 0.4 | 0.62 | 0.69 | | | | | | | |
| 0.6 | 0.74 | 0.80 | 0.82 | | | | | | |
| 0.8 | 0.99 | 1.10 | 0.95 | 0.90 | | | | | |
| 1.0 | 1.48 | 1.12 | 1.41 | 1.24 | 1.21 | | | | |
| 1.2 | 1.91 | 1.33 | 1.43 | 1.52 | 1.55 | 1.64 | | | |
| 1.4 | 2.47 | 1.67 | 1.70 | 2.04 | 1.86 | 1.98 | 2.47 | | |
| 1.6 | 3.17 | 2.40 | 2.33 | 2.53 | 2.31 | 2.51 | 3.13 | 3.25 | |
| 1.8 | 3.85 | 3.37 | 2.89 | 3.23 | 3.09 | 3.03 | 3.30 | 3.74 | 4.11 |

| Main, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0.2 | 0.4 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 |
| C | 0.03 | 0.04 | 0.07 | 0.12 | 0.13 | 0.14 | 0.27 | 0.30 | 0.25 |

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–11 Loss Coefficients, Diverging Junctions (Tees, Wyes) (Continued)

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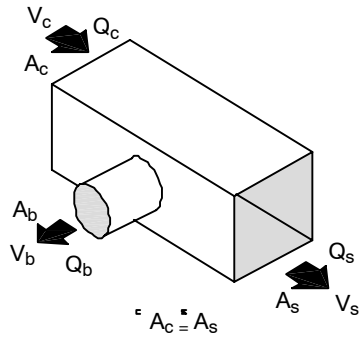
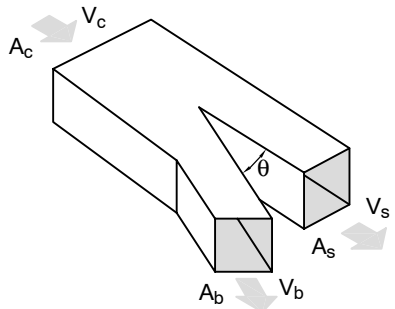
## S. TEE RECTANGULAR MAIN TO ROUND BRANCH




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $V_b/V_c$ | $Q_b/Q_c$ | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.2 | 1.00 | | | | | | | | |
| 0.4 | 1.01 | 1.07 | | | | | | | |
| 0.6 | 1.14 | 1.10 | 1.08 | | | | | | |
| 0.8 | 1.18 | 1.31 | 1.12 | 1.13 | | | | | |
| 1.0 | 1.30 | 1.38 | 1.20 | 1.23 | 1.26 | | | | |
| 1.2 | 1.46 | 1.58 | 1.45 | 1.31 | 1.39 | 1.48 | | | |
| 1.4 | 1.70 | 1.82 | 1.65 | 1.51 | 1.56 | 1.64 | 1.71 | | |
| 1.6 | 1.93 | 2.06 | 2.00 | 1.85 | 1.70 | 1.76 | 1.80 | 1.88 | |
| 1.8 | 2.06 | 2.17 | 2.20 | 2.13 | 2.06 | 1.98 | 1.99 | 2.00 | 2.07 |

## T. WYE RECTANGULAR




$\theta$ = 15° to 90°
$A_c = A_s + A_b$

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $\theta$ | $V_b/V_c$ | | | | | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.8 | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| 15° | 0.81 | 0.65 | 0.51 | 0.38 | 0.28 | 0.20 | 0.11 | 0.06 | 0.14 | 0.30 | 0.51 | 0.76 | 1.0 |
| 30° | 0.84 | 0.69 | 0.56 | 0.44 | 0.34 | 0.26 | 0.19 | 0.15 | 0.15 | 0.30 | 0.51 | 0.76 | 1.0 |
| 45° | 0.87 | 0.74 | 0.63 | 0.54 | 0.45 | 0.38 | 0.29 | 0.24 | 0.23 | 0.30 | 0.51 | 0.76 | 1.0 |
| 60° | 0.90 | 0.82 | 0.79 | 0.66 | 0.59 | 0.53 | 0.43 | 0.36 | 0.33 | 0.39 | 0.51 | 0.76 | 1.0 |
| 90° | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |

| Branch, Coefficient C (*See* Note 8) | | | | | | |
|---|---|---|---|---|---|---|
| $\theta$ | 15°-60° | 90° | | | | |
| $V_s/V_c$ | $A_s/A_c$ | | | | | |
| | 0-1.0 | 0-0.4 | 0.5 | 0.6 | 0.7 | ≥0.8 |
| 0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |
| 0.1 | 0.81 | 0.81 | 0.81 | 0.81 | 0.81 | 0.81 |
| 0.2 | 0.64 | 0.64 | 0.64 | 0.64 | 0.64 | 0.64 |
| 0.3 | 0.50 | 0.50 | 0.52 | 0.52 | 0.50 | 0.50 |
| 0.4 | 0.36 | 0.36 | 0.40 | 0.38 | 0.37 | 0.36 |
| 0.5 | 0.25 | 0.25 | 0.30 | 0.28 | 0.27 | 0.25 |
| 0.6 | 0.16 | 0.16 | 0.23 | 0.20 | 0.18 | 0.16 |
| 0.8 | 0.04 | 0.04 | 0.17 | 0.10 | 0.07 | 0.04 |
| 1.0 | 0 | 0 | 0.20 | 0.10 | 0.05 | 0 |
| 1.2 | 0.07 | 0.07 | 0.36 | 0.21 | 0.14 | 0.07 |
| 1.4 | 0.39 | 0.39 | 0.79 | 0.59 | 0.39 | — |
| 1.6 | 0.90 | 0.90 | 1.4 | 1.2 | — | — |
| 1.8 | 1.8 | 1.8 | 2.4 | — | — | — |
| 2.0 | 3.2 | 3.2 | 4.0 | — | — | — |

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

**Table A–11 Loss Coefficients, Diverging Junctions (Tees, Wyes) (Continued)**

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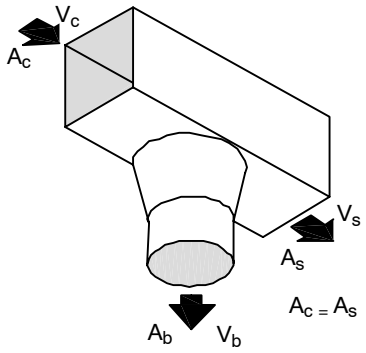
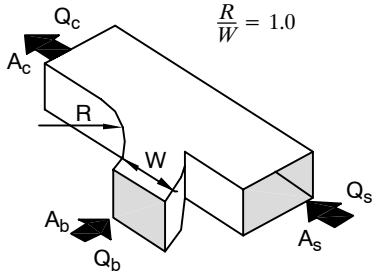
### U. TEE RECTANGULAR MAIN TO CONICAL BRANCH




| Branch, Coefficient C (*See* Note 8) | | | | | | |
|---|---|---|---|---|---|---|
| $V_b/V_c$ | 0.40 | 0.50 | 0.75 | 1.0 | 1.3 | 1.5 |
| C | 0.80 | 0.83 | 0.90 | 1.0 | 1.1 | 1.4 |

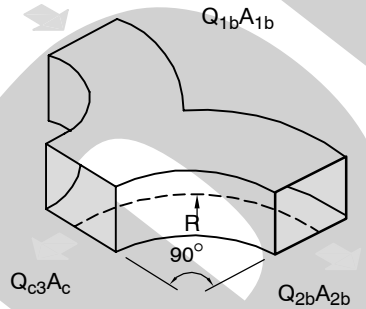
### V. WYE, RECTANGULAR




| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $A_b/A_s$ | $A_b/A_s$ | $Q_b/Q_c$ | | | | | | | | |
| | | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.25 | 0.25 | 0.55 | 0.50 | 0.60 | 0.85 | 1.2 | 1.8 | 3.1 | 4.4 | 6.0 |
| 0.33 | 0.25 | 0.35 | 0.35 | 0.50 | 0.80 | 1.3 | 2.0 | 2.8 | 3.8 | 5.0 |
| 0.5 | 0.5 | 0.62 | 0.48 | 0.40 | 0.40 | 0.48 | 0.60 | 0.78 | 1.1 | 1.5 |
| 0.67 | 0.5 | 0.52 | 0.40 | 0.32 | 0.30 | 0.34 | 0.44 | 0.62 | 0.92 | 1.4 |
| 1.0 | 0.5 | 0.44 | 0.38 | 0.38 | 0.41 | 0.52 | 0.68 | 0.92 | 1.2 | 1.6 |
| 1.0 | 1.0 | 0.67 | 0.55 | 0.46 | 0.37 | 0.32 | 0.29 | 0.29 | 0.30 | 0.37 |
| 1.33 | 1.0 | 0.70 | 0.60 | 0.51 | 0.42 | 0.34 | 0.28 | 0.26 | 0.26 | 0.29 |
| 2.0 | 1.0 | 0.60 | 0.52 | 0.43 | 0.33 | 0.24 | 0.17 | 0.15 | 0.17 | 0.21 |

| Branch, Coefficient C (*See* Note 8) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $A_b/A_s$ | $A_b/A_s$ | $Q_b/Q_c$ | | | | | | | | |
| | | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.25 | 0.25 | -.01 | -.03 | -.01 | 0.05 | 0.13 | 0.21 | 0.29 | 0.38 | 0.46 |
| 0.33 | 0.25 | 0.08 | 0 | -.02 | -.01 | 0.02 | 0.08 | 0.16 | 0.24 | 0.34 |
| 0.5 | 0.5 | -.03 | -.06 | -.05 | 0 | 0.06 | 0.12 | 0.19 | 0.27 | 0.35 |
| 0.67 | 0.5 | 0.04 | -.02 | -.04 | -.03 | -.01 | 0.04 | 0.12 | 0.23 | 0.37 |
| 1.0 | 0.5 | 0.72 | 0.48 | 0.28 | 0.13 | 0.05 | 0.04 | 0.09 | 0.18 | 0.30 |
| 1.0 | 1.0 | -.02 | -.04 | -.04 | -.01 | 0.06 | 0.13 | 0.22 | 0.30 | 0.38 |
| 1.33 | 1.0 | 0.10 | 0 | 0.01 | -.03 | -.01 | 0.03 | 0.10 | 0.20 | 0.30 |
| 2.0 | 1.0 | 0.62 | 0.38 | 0.23 | 0.13 | 0.08 | 0.05 | 0.06 | 0.10 | 0.20 |

### W. SYMMETRICAL WYE, DOVETAIL, RECTANGULAR




| Coefficient C (*See* Note 8) | | |
|---|---|---|
| $\mathbf{A_{1b}/A_c}$ or $A_{2b}/A_c$ | 0.50 | 1.0 |
| C | 0.30 | 0.25 |

When: $\frac{R}{W} = 1.5$

$$\frac{Q_{1b}}{Q_c} = \frac{Q_{2b}}{Q_c} = 0.5$$

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–11 Loss Coefficients, Diverging Junctions (Tees, Wyes) (Continued)

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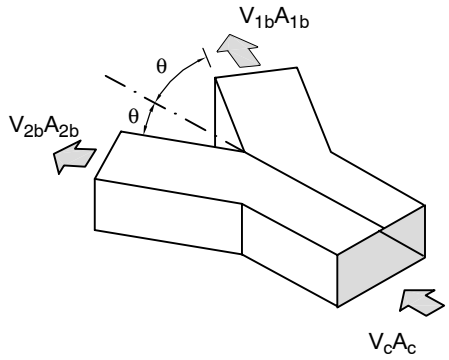
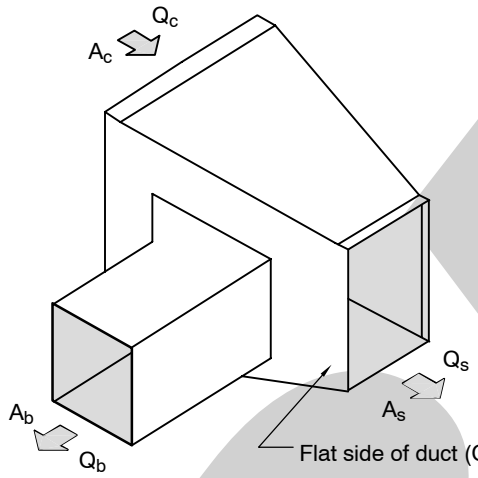
### X. WYE, RECTANGULAR AND ROUND




| θ | Coefficient C (*See* Note 8) | | | | | | |
|---|---|---|---|---|---|---|---|
| | $V_{lb}/V_c$ or $V_{2b}/V_c$ | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.8 |
| 15° | 0.81 | 0.65 | 0.51 | 0.38 | 0.28 | 0.20 | 0.11 |
| 30° | 0.84 | 0.69 | 0.56 | 0.44 | 0.34 | 0.26 | 0.19 |
| 45° | 0.87 | 0.74 | 0.63 | 0.54 | 0.45 | 0.38 | 0.29 |
| 60° | 0.90 | 0.82 | 0.79 | 0.66 | 0.59 | 0.53 | 0.43 |
| 90° | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |

| θ | $V_{lb}/V_c$ or $V_{2b}/V_c$ | | | | | |
|---|---|---|---|---|---|---|
| | 1.0 | 1.2 | 1.4 | 1.6 | 1.8 | 2.0 |
| 15° | 0.06 | 0.14 | 0.30 | 0.51 | 0.76 | 1.0 |
| 30° | 0.15 | 0.15 | 0.30 | 0.51 | 0.76 | 1.0 |
| 45° | 0.24 | 0.23 | 0.30 | 0.51 | 0.76 | 1.0 |
| 60° | 0.36 | 0.33 | 0.39 | 0.51 | 0.76 | 1.0 |
| 90° | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |

### Y. TEE, RECTANGULAR REDUCING, 45° ENTRY BRANCH




| Branch, Coefficient C | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $Q_b/Q_c$ | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 |
| C | 1.16 | 0.96 | 0.82 | 0.68 | 0.56 | 0.49 | 0.47 | 0.48 | 0.50 | 0.54 |

| Main, Coefficient C | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $Q_s/Q_c$ | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 |
| C | 0.21 | 0.20 | 0.20 | 0.20 | 0.20 | 0.20 | 0.22 | 0.25 | 0.35 | 0.53 |

When:

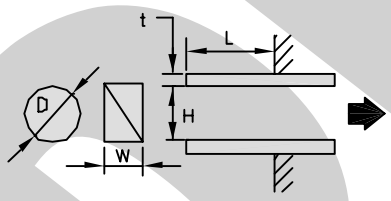
$\frac{A_b}{A_c}$ = 0.5 to 1.0

$\frac{A_s}{A_c}$ = 0.5 to 1.0

*Note 8:* A = Area ( sq. in.). Q= airflow (cfm). V= Velocity (fpm)
Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–11 Loss Coefficients, Diverging Junctions (Continued)

### A. DUCT MOUNTED IN WALL, ROUND AND RECTANGULAR



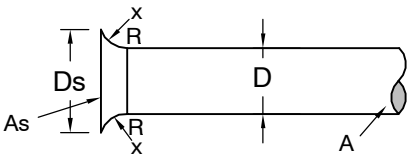

$Rectangular: D = \frac{2HW}{(H + W)}$

| t/D | Coefficient C | | | | | | |
|---|---|---|---|---|---|---|---|
| | L/D | | | | | | |
| | 0 | 0.002 | 0.01 | 0.05 | 0.2 | 0.5 | ≥1.0 |
| ≈0.05 | 0.50 | 0.57 | 0.68 | 0.80 | 0.92 | 1.0 | 1.0 |
| 0.02 | 0.50 | 0.51 | 0.52 | 0.55 | 0.66 | 0.72 | 0.72 |
| ≥0.05 | 0.50 | 0.50 | 0.50 | 0.50 | 0.50 | 0.50 | 0.50 |

With Screen or Perforated Plate:
a. Sharp Edge ($t/D_e \leq 0.05$): $C_s = 1 + C_1$
b. Thick Edge ($t/D_e > 0.05$): $C_s = C + C_1$
where:
$C_s$ is new coefficient of fitting with a screen or perforated plate at the entrance.
C is from above table
$C_1$ is from Table 14-17A (screen) or Table 14-17B (perforated plate)
Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–12 Loss Coefficients, Entries

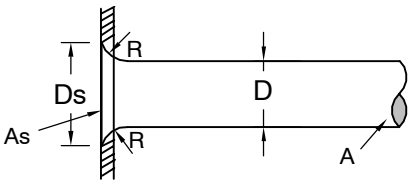
### B. SMOOTH COVERING BELLMOUTH, ROUND, WITHOUT END WALL




| Coefficient C (*See* Note 9) | | | | | | |
|---|---|---|---|---|---|---|
| R/D | 0 | 0.01 | 0.02 | 0.03 | 0.04 | 0.05 |
| C | 1.0 | 0.87 | 0.74 | 0.81 | 0.51 | 0.40 |

| R/D | 0.06 | 0.08 | 0.10 | 0.12 | 0.16 | 0.20 & greater |
|---|---|---|---|---|---|---|
| C | 0.32 | 0.20 | 0.15 | 0.10 | 0.06 | 0.03 |

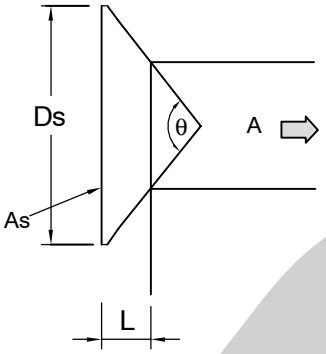
### C. SMOOTH COVERING BELLMOUTH, ROUND, WITH END WALL

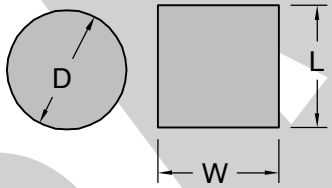



| Coefficient C (*See* Note 9) | | | | | | |
|---|---|---|---|---|---|---|
| R/D | 0 | 0.01 | 0.02 | 0.03 | 0.04 | 0.05 |
| C | 0.50 | 0.43 | 0.36 | 0.31 | 0.26 | 0.22 |

| R/D | 0.06 | 0.08 | 0.10 | 0.12 | 0.16 | 0.20 & greater |
|---|---|---|---|---|---|---|
| C | 0.20 | 0.15 | 0.12 | 0.09 | 0.06 | 0.03 |

### D. CONICAL, CONVERGING BELLMOUTH, ROUND AND RECTANGULAR WITHOUT END WALL




Rectangular $D = \dfrac{2HW}{(H + W)}$

$\theta$ is a major angle for rectangular entry

| Coefficient C (*See* Note 9) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| L/D | θ | | | | | | | | |
| | 0° | 10° | 20° | 30° | 40° | 60° | 10° | 140° | 180° |
| 0.025 | 1.0 | 0.96 | 0.93 | 0.90 | 0.86 | 0.80 | 0.69 | 0.59 | 0.50 |
| 0.05 | 1.0 | 0.93 | 0.86 | 0.80 | 0.75 | 0.67 | 0.58 | 0.53 | 0.50 |
| 0.10 | 1.0 | 0.80 | 0.67 | 0.55 | 0.48 | 0.41 | 0.41 | 0.44 | 0.50 |
| 0.25 | 1.0 | 0.68 | 0.45 | 0.30 | 0.22 | 0.17 | 0.22 | 0.34 | 0.50 |
| 0.60 | 1.0 | 0.46 | 0.27 | 0.18 | 0.14 | 0.13 | 0.21 | 0.33 | 0.50 |
| 1.0 | 1.0 | 0.32 | 0.20 | 0.14 | 0.11 | 0.10 | 0.18 | 0.30 | 0.50 |

*Note 9:* With screening in opening at $D_s$, $C_s$ = C (from table + $\dfrac{\text{C (Screen coef. Table A} - 14)}{\left(\frac{A_s}{A}\right)^2}$

where A= Area at D; $A_s$ at $D_s$

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–12 Loss Coefficients, Entries (Continued)

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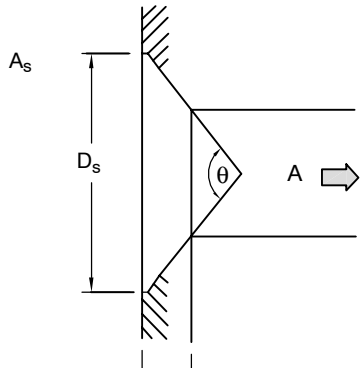
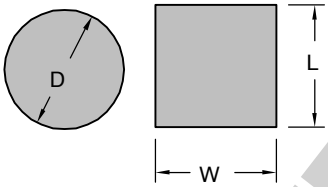
## E. CONICAL, CONVERGING BELLMOUTH, ROUND AND RECTANGULAR WITH END WALL



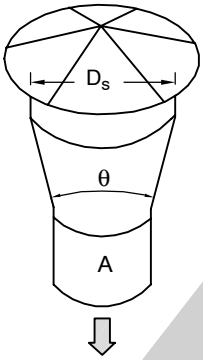




Rectangular $D = \frac{2HW}{(H + W)}$

$\theta$ is a major angle for rectangular entry

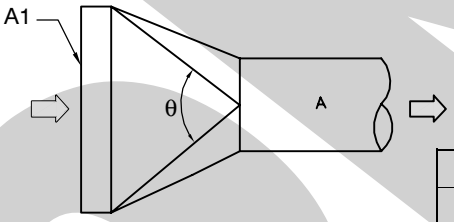
| Coefficient C (*See* Note 9) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| L/D | $\theta$ | | | | | | | | |
| | 0° | 10° | 20° | 30° | 40° | 60° | 100° | 140° | 180° |
| 0.025 | 0.50 | 0.47 | 0.45 | 0.43 | 0.41 | 0.40 | 0.42 | 0.45 | 0.50 |
| 0.05 | 0.50 | 0.45 | 0.41 | 0.36 | 0.33 | 0.30 | 0.35 | 0.42 | 0.50 |
| 0.075 | 0.50 | 0.42 | 0.35 | 0.30 | 0.26 | 0.23 | 0.30 | 0.40 | 0.50 |
| 0.10 | 0.50 | 0.39 | 0.32 | 0.25 | 0.22 | 0.18 | 0.27 | 0.38 | 0.50 |
| 0.15 | 0.50 | 0.37 | 0.27 | 0.20 | 0.16 | 0.15 | 0.25 | 0.37 | 0.50 |
| 0.60 | 0.50 | 0.27 | 0.18 | 0.13 | 0.11 | 0.12 | 0.23 | 0.36 | 0.50 |

## F. INTAKE HOOD




| Coefficient C (*See* Note 9) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| $\theta$ | L/D | | | | | | | | |
| | 0.1 | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 or greater |
| 0° | 2.6 | 1.8 | 1.5 | 1.4 | 1.3 | 1.2 | 1.2 | 1.1 | 1.1 |
| 15° | 1.3 | 0.77 | 0.60 | 0.48 | 0.41 | 0.30 | 0.29 | 0.28 | 0.25 |

## G. HOOD, TAPERED, FLNAGED OR UNFLANGED




$A_1 \geq 2A$

$\theta$ is a major angle for rectangular entry

| Coefficient C (*See* Note 9) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| $\theta$ | 0° | 20° | 40° | 60° | 80° | 100° | 120° | 140° | 160° | 180° |
| Round Hood | 1.0 | 0.11 | 0.06 | 0.09 | 0.14 | 0.18 | 0.27 | 0.32 | 0.43 | 0.50 |
| Square Hood or Rect. Hood | 1.0 | 0.19 | 0.13 | 0.16 | 0.21 | 0.27 | 0.33 | 0.43 | 0.53 | 0.62 |

*Note 9:* With screening in opening at $D_s$, $C_s$ = C (from table + $\frac{\text{C (Screen coef. Table ?)}}{\left(\frac{A_s}{A}\right)^2}$

where $A$ = Area at D; $A_s$ at $D_s$

Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

**Table A–12 Loss Coefficients, Entries (Continued)**

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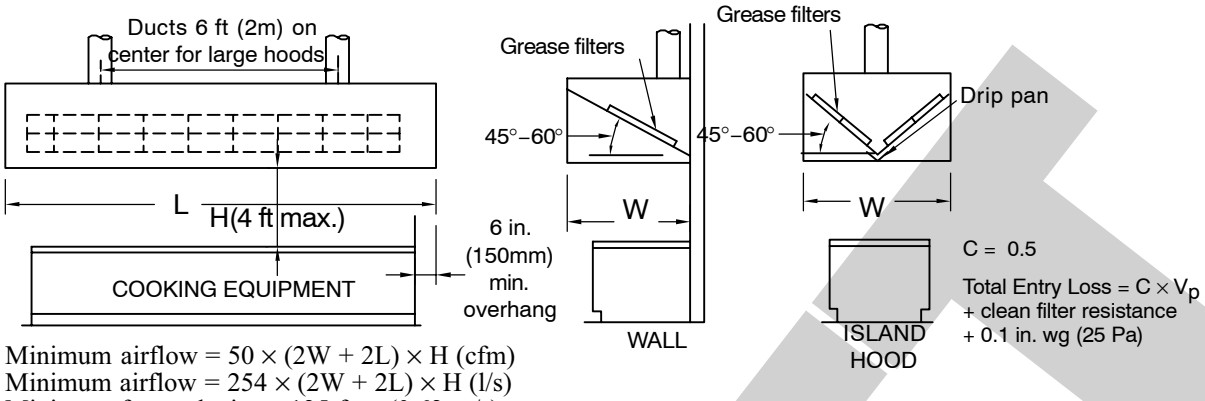
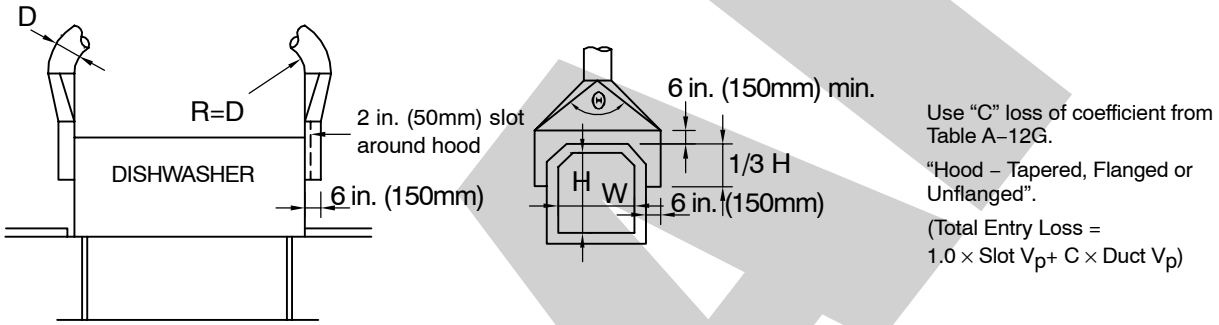
### H. HOOD, CANOPY ISLAND OR RANGE




Minimum airflow = 50 × (2W + 2L) × H (cfm)
Minimum airflow = 254 × (2W + 2L) × H (l/s)
Minimum face velocity = 125 fpm (0.62 m/s)
Minimum duct velocity = 1000 fpm (5 m/s)
(Use minimums from local codes where applicable)
W,L,H in feet (U.S.), in meters (metric)

### I. HOOD, SLOT (DISHWASHER)

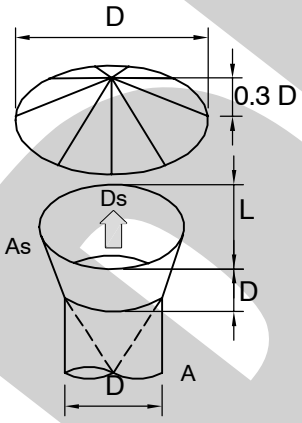



Minimum airflow = 300 × H × W (cfm) = 1524 × H × W (l/s)
Minimum duct velocity = 1000 fpm (5 m/s)
(Use minimums from local codes where applicable)

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–12 Loss Coefficients, Entries (Continued)

### A. EXHAUST HOOD




| θ | Coefficient C (*See* Note 9) | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | L/D | | | | | | | | | |
| | 0.1 | 0.2 | 0.25 | 0.3 | 0.35 | 0.4 | 0.5 | 0.6 | 0.8 | 1.0 |
| 0° | 4.0 | 2.3 | 1.9 | 1.6 | 1.4 | 1.3 | 1.2 | 1.1 | 1.0 | 1.0 |
| 15° | 2.6 | 1.2 | 1.0 | 0.80 | 0.70 | 0.65 | 0.60 | 0.60 | 0.60 | 0.60 |

*Note 9:* With screening in opening at $D_s$, $C_s$ = C (from table + $\frac{\text{C (Screen coef. Table A – 14)}}{\left(\frac{A_s}{A}\right)^2}$

where *A* = Area at D; $A_s$ at $D_s$

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–13 Loss Coefficients, Exits

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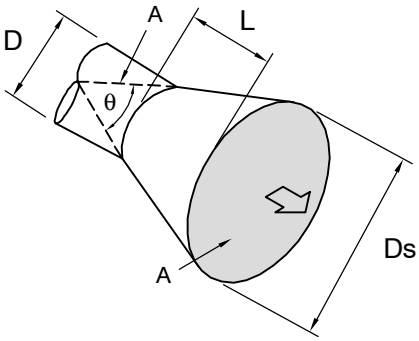
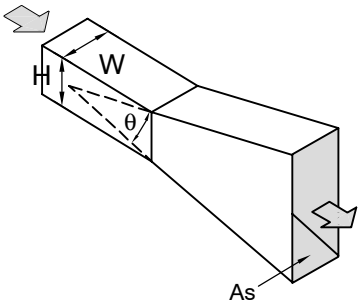
### B. EXIT, CONICAL, ROUND, WITH OR WITHOUT A WALL



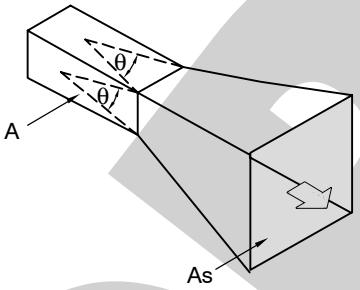

| Coefficient C (*See* Note 9) | | | | | | | |
|---|---|---|---|---|---|---|---|
| $A_s/A$ | $\theta$ | | | | | | |
| 2 | 0.33 | 0.36 | 0.44 | 0.74 | 0.97 | 0.99 | 1.0 |
| 4 | 0.24 | 0.28 | 0.36 | 0.54 | 0.94 | 1:0 | 1.0 |
| 6 | 0.22 | 0.25 | 0.32 | 0.49 | 0.94 | 0.98 | 1.0 |
| 10 | 0.19 | 0.23 | 0.30 | 0.50 | 0.94 | 0.72 | 1.0 |
| 16 | 0.17 | 0.20 | 0.27 | 0.49 | 0.94 | 1.0 | 1.0 |

### C. EXIT, PLANE DIFFUSER, RECTANGULAR WITH OR WITHOUT A WALL




| Coefficient C (*See* Note 10) | | | | | | |
|---|---|---|---|---|---|---|
| $A_s/A$ | $\theta$ | | | | | |
| | 14° | 20° | 30° | 45° | 60° | ≥ 90° |
| 2 | 0.37 | 0.38 | 0.50 | 0.75 | 0.90 | 1.1 |
| 4 | 0.25 | 0.37 | 0.57 | 0.82 | 1.0 | 1.1 |
| 6 | 0.28 | 0.47 | 0.64 | 0.87 | 1.0 | 1.1 |

### D. EXIT, PYRAMIDAL DIFFUSER, RECTANGULAR WITH OR WITHOUT A WALL




| Coefficient C (*See* Note 10) | | | | | | |
|---|---|---|---|---|---|---|
| $A_s/A$ | $\theta$ | | | | | |
| | 10° | 14° | 20° | 30° | 45° | ≥ 60° |
| 2 | 0.44 | 0.58 | 0.70 | 0.86 | 1.0 | 1.1 |
| 4 | 0.31 | 0.48 | 0.61 | 0.76 | 0.94 | 1.1 |
| 6 | 0.29 | 0.47 | 0.62 | 0.74 | 0.94 | 1.1 |
| 10 | 0.26 | 0.45 | 0.60 | 0.73 | 0.89 | 1.0 |

*Note 9:* With screening in opening at $D_s$, $C_s$ = C (from table + $\dfrac{C\ (\text{Screen coef. Table A} - 14)}{\left(\frac{A_s}{A}\right)^2}$

where $A$ = Area at D; $A_s$ at $D_s$

*Note 10:* With screening in opening at $A_s$, $C_s$ = C (from table + $\dfrac{C\ (\text{Screen coef. Table A} - 14)}{\left(\frac{A_s}{A}\right)^2}$

Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = $C \times V_p$

## Table A–13 Loss Coefficients, Exits (Continued)

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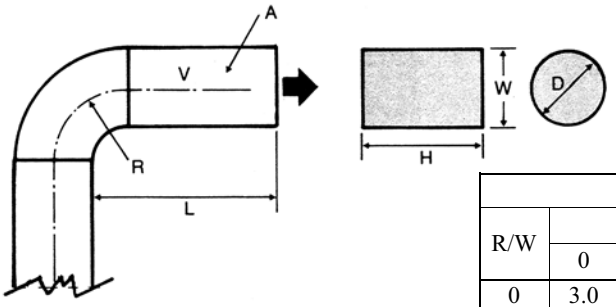
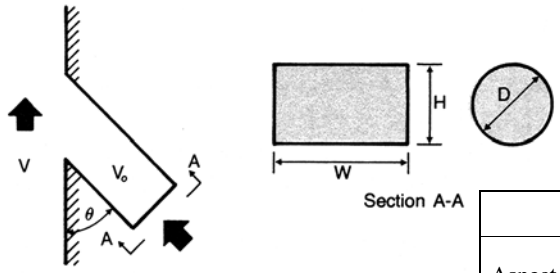
### E. EXIT, DISCHARGE TO ATMOSPHERE FROM A 90° ELBOW, ROUND AND RECTANGULAR




| Rectangular: Coefficient C (*See* Note 11) | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| R/W | L/W | | | | | | | | | |
| | 0 | 0.5 | 1.0 | 1.5 | 2.0 | 3.0 | 4.0 | 6.0 | 8.0 | 12.0 |
| 0 | 3.0 | 3.1 | 3.2 | 3.0 | 2.7 | 2.4 | 2.2 | 2.1 | 2.1 | 2.0 |
| 0.75 | 2.2 | *2.2* | 2.1 | 1.8 | 1.7 | 1.6 | 1.6 | 1.5 | 1.5 | 1.5 |
| 1.0 | 1.8 | 1.5 | 1.4 | 1.4 | 1.3 | 1.3 | 1.2 | 1.2 | 1.2 | 1.2 |
| 1.5 | 1.5 | 1.2 | 1.1 | 1.1 | 1.1 | 1.1 | 1.1 | 1.1 | 1.1 | 1.1 |
| 2.5 | 1.2 | 1.1 | 1.1 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 | 1.0 |

| Round: Coefficient C (*See* Note 10) | | |
|---|---|---|
| L/D | 0.9 | 1.3 |
| C | 1.5 | 1.4 |

### F. EXIT, DUCT FLUSH WITH WALL, FLOW ALONG WALL




| Rectangular: Coefficient C (*See* Note 11) | | | | | | |
|---|---|---|---|---|---|---|
| Aspect Ratio (H/W) | θ | $V/V_0$ | | | | |
| | | 0 | 0.5 | 1.0 | 1.5 | 2.0 |
| 0.1-0.2 | 30°-90° | 1.0 | 0.95 | 1.2 | 1.5 | 1.8 |
| | 120° | 1.0 | 0.92 | 1.1 | 1.4 | 1.9 |
| | 150° | 1.0 | 0.75 | 0.95 | 1.4 | 1.8 |
| 0.5-2.0 | 30°-45° | 1.0 | 1.0 | 1.1 | 1.3 | 1.6 |
| | 60° | 1.0 | 0.90 | 1.1 | 1.4 | 1.6 |
| | 90° | 1.0 | 0.80 | 0.95 | 1.4 | 1.7 |
| | 120° | 1.0 | 0.80 | 0.95 | 1.3 | 1.7 |
| | 150° | 1.0 | 0.82 | 0.83 | 1.0 | 1.3 |
| 5-10 | 45° | 1.0 | 0.92 | 0.93 | 1.1 | 1.3 |
| | 60° | 1.0 | 0.87 | 0.87 | 1.0 | 1.3 |
| | 90° | 1.0 | 0.82 | 0.80 | 0.97 | 1.2 |
| | 120° | 1.0 | 0.80 | 0.76 | 0.90 | 0.98 |

| Round: Coefficient C (*See* Note 10) | | | | | |
|---|---|---|---|---|---|
| θ | $V/V_0$ | | | | |
| | 0 | 0.5 | 1.0 | 1.5 | 2.0 |
| 30°-45° | 1.0 | 1.0 | 1.1 | 1.3 | 1.6 |
| 60° | 1.0 | 0.90 | 1.1 | 1.4 | 1.6 |
| 90° | 1.0 | 0.80 | 0.95 | 1.4 | 1.7 |
| 120° | 1.0 | 0.80 | 0.95 | 1.3 | 1.7 |
| 150° | 1.0 | 0.82 | 0.83 | 1.0 | 1.3 |

*Note 11:* With screening in opening at $C_s = C$ (from table + C (Screen Coef. from Table A-14)
Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss $TP = C \times V_p$

## Table A–13 Loss Coefficients, Exits (Continued)

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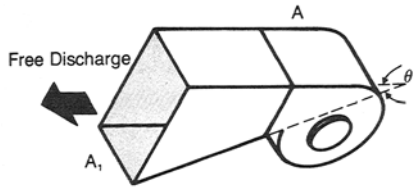
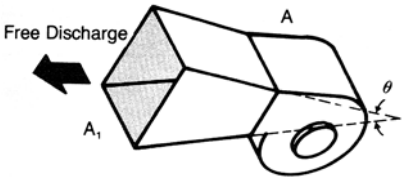
### G. PLANE ASYMMETRIC DIFFUSER AT FAN OUTLET WITHOUT DUCTWORK



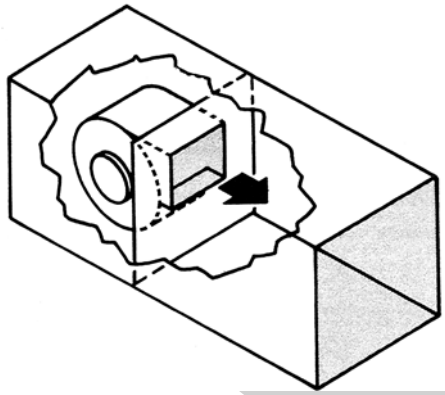

| θ | Coefficient C (*See* Note 12) | | | | | |
|---|---|---|---|---|---|---|
| | $A_1/A$ | | | | | |
| | 1.5 | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 |
| 10° | 0.51 | 0.34 | 0.25 | 0.21 | 0.18 | 0.17 |
| 15° | 0.54 | 0.36 | 0.27 | 0.24 | 0.22 | 0.20 |
| 20° | 0.55 | 0.38 | 0.31 | 0.27 | 0.25 | 0.24 |
| 25° | 0.59 | 0.43 | 0.37 | 0.35 | 0.33 | 0.33 |
| 30° | 0.63 | 0.50 | 0.46 | 0.44 | 0.43 | 0.42 |
| 35° | 0.65 | 0.56 | 0.53 | 0.52 | 0.51 | 0.50 |

### H. PYRAMIDAL DIFFUSER AT FAN OUTLET WITHOUT DUCTWORK



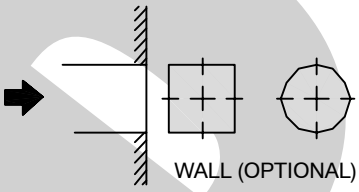

| θ | Coefficient C (*See* Note 12) | | | | | |
|---|---|---|---|---|---|---|
| | $A_1/A$ | | | | | |
| | 1.5 | 2.0 | 2.5 | 3.0 | 3.5 | 4.0 |
| 10° | 0.54 | 0.42 | 0.37 | 0.34 | 0.32 | 0.31 |
| 150 | 0.67 | 0.58 | 0.53 | 0.51 | 0.50 | 0.51 |
| 20° | 0.75 | 0.67 | 0.65 | 0.64 | 0.64 | 0.65 |
| 25° | 0.80 | 0.74 | 0.72 | 0.70 | 0.70 | 0.72 |
| 30° | 0.85 | 0.78 | 0.76 | 0.75 | 0.75 | 0.76 |

### I. FAN, FREE DISCHARGE, PLENUM



C = 1.0 (Airfoil wheels - backwardly inclined)
C = 1.5 (Backwardly inclined wheels)
C = 1.8 (Forward curved wheel)

### J. EXIT, ABRUPT, ROUND AND RECTANGULAR, WITH OR WITHOUT A WALL




C = 1.0
With Screen: Cs = 1 + (C from Table A-14)

*Note 12:* With screening in opening at $A_1$, $C_s$ = C (from table + $\dfrac{C\ (\text{Screen coef. Table A} - 14)}{\left(\frac{A_1}{A}\right)^2}$

Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–13 Loss Coefficients, Exits (Continued)



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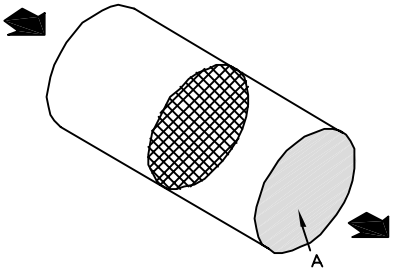
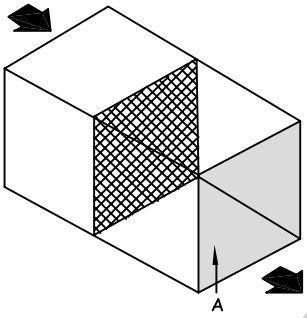
### A. SCREEN IN DUCT, ROUND AND RECTANGULAR

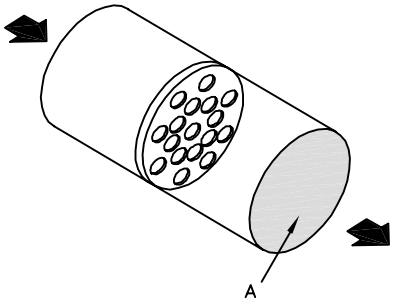


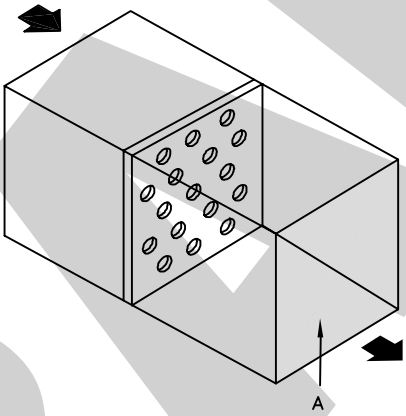




$n = A_s/A$
where:
n = Free area ratio of screen
$A_s$ = Total flow area of screen
A = Area of Duct

| Coefficient C | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| n | 0.30 | 0.40 | 0.50 | 0.55 | 0.60 | 0.65 | 0.70 | 0.75 | 0.80 | 0.90 | 1.0 |
| C | 6.2 | 3.0 | 1.7 | 1.3 | 0.97 | 0.75 | 0.58 | 0.44 | 0.32 | 0.14 | 0 |

### B. PERFORATED PLATE IN DUCT, THICK, ROUND AND RECTANGULAR




| Coefficient C | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| t/D | n | | | | | | | | |
| | 0.20 | 0.25 | 0.30 | 0.40 | 0.50 | 0.60 | 0.70 | 0.80 | 0.90 |
| 0.015 | 52 | 30 | 18 | 8.2 | 4.0 | 2.0 | 0.97 | 0.42 | 0.13 |
| 0.2 | 48 | 28 | 17 | 7.7 | 3.8 | 1.9 | 0.91 | 0.40 | 0.13 |
| 0.4 | 46 | 27 | 17 | 7.4 | 3.6 | 1.8 | 0.88 | 0.39 | 0.13 |
| 0.6 | 42 | 24 | 15 | 6.6 | 3.2 | 1.6 | 0.80 | 0.36 | 0.13 |

t/D > 0.015

$$n = \frac{A_p}{A}$$

where: t = plate thickness
d = duameter of perforated holes
n = free area ratio of plate
Ap = total flow area of perforated plate
A = area of duct

Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

## Table A–14 Loss Coefficients, Screens and Plates

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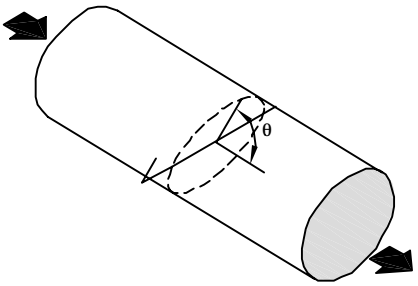
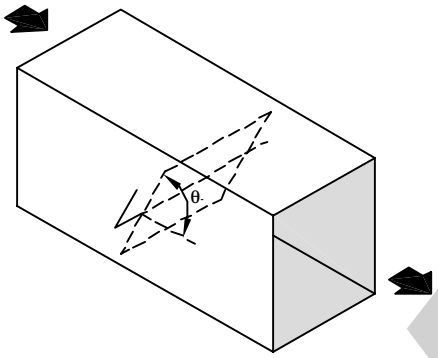
### A. DAMPER, BUTTERFLY, THIN PLATE, ROUND




| Coefficient C | | | | | | | |
|---|---|---|---|---|---|---|---|
| $\theta$ | 0° | 10° | 20° | 30° | 40° | 50° | 60° |
| C | 0.20 | 0.52 | 1.5 | 4.5 | 11 | 29 | 108 |

0° is full open

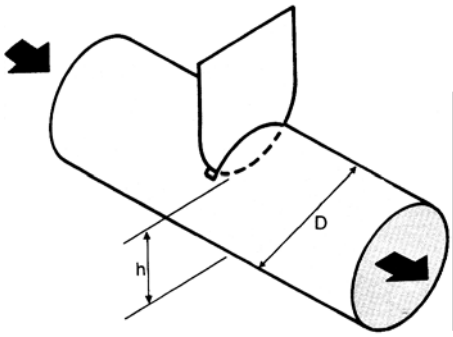
### B. DAMPER, BUTTERFLY, THIN PLATE, RECTANGULAR




| Coefficient C | | | | | | | |
|---|---|---|---|---|---|---|---|
| $\theta$ | 0° | 10° | 20° | 30° | 40° | 50° | 60° |
| C | 0.04 | 0.33 | 1.2 | 3.3 | 9.0 | 26 | 70 |

0° is full open

### C. DAMPER, GATE, ROUND




| Coefficient C | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| h/D | 0.2 | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| $A_h/A$ | 0.25 | 0.38 | 0.50 | 0.61 | 0.71 | 0.81 | 0.90 | 0.96 |
| C | 35 | 10 | 4.6 | 2.1 | 0.98 | 0.44 | 0.17 | 0.06 |

$A_h$ = Free Area

A = Area of Duct

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–15 Loss Coefficients, Obstructions (Constant Velocities)

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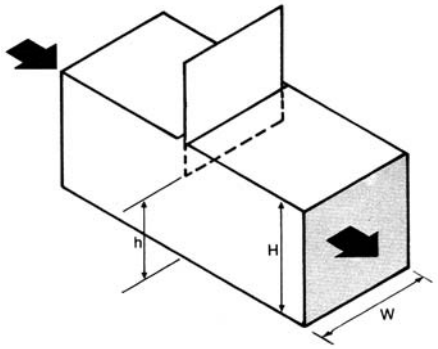
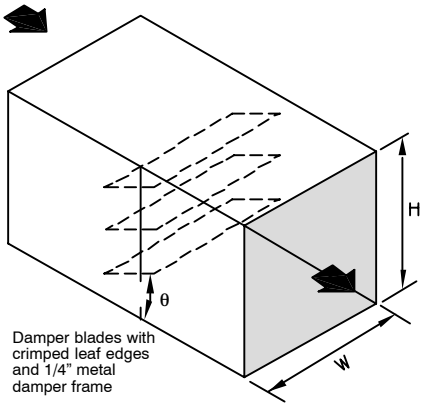
### D. DAMPER, GATE, RECTANGULAR




| Coefficient C | | | | | | | |
|---|---|---|---|---|---|---|---|
| | h/H | | | | | | |
| H/W | 0.3 | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 |
| 0.5 | 14 | 6..9 | 3.3 | 1.7 | 0.83 | 0.32 | 0.09 |
| 1.0 | 19 | 8.8 | 4.5 | 2.4 | 1.2 | 0.55 | 0.17 |
| 1.5 | 20 | 9.1 | 4.7 | 2.7 | 1.2 | 0.47 | 0.11 |
| 2.0 | 18 | 8.8 | 4.5 | 2.3 | 1.1 | 0.51 | 0.13 |

### E. DAMPER, RECTANGULAR, PARALLEL BLADES

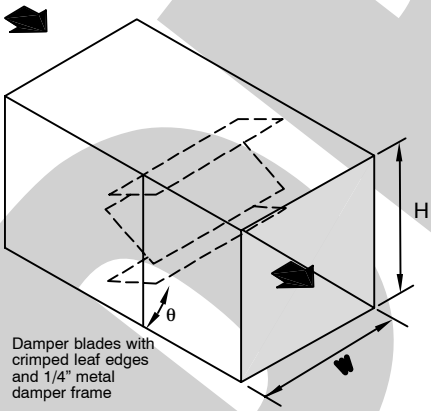



| Coefficient C | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| L/R | θ | | | | | | | | |
| | 80° | 70° | 60° | 50° | 40° | 30° | 20° | 10° | 0° Fully open |
| 0.3 | 116 | 32 | 14 | 9.0 | 5.0 | 2.3 | 1.4 | 0.79 | 0.52 |
| 0.4 | 152 | 38 | 16 | 9.0 | 6.0 | 2.4 | 1.5 | 0.85 | 0.52 |
| 0.5 | 188 | 45 | 18 | 9.0 | 6.0 | 2.4 | 1.5 | 0.92 | 0.52 |
| 0.6 | 245 | 45 | 21 | 9.0 | 5.4 | 2.4 | 1.5 | 0.92 | 0.52 |
| 0.8 | 284 | 55 | 22 | 9.0 | 5.4 | 2.5 | 1.5 | 0.92 | 0.52 |
| 1.0 | 361 | 65 | 24 | 10 | 5.4 | 2.6 | 1.6 | 1.0 | 0.52 |
| 1.5 | 576 | 102 | 28 | 10 | 5.4 | 2.7 | 1.6 | 1.0 | 0.52 |

$$\frac{L}{R} = \frac{NW}{2(H + W)}$$ where:

N is number of damper blades
W is duct dimension parallel to blade axis
L is sum of damper blade lengths
R is perimeter of duct
H is duct dimension on perpendicular to blade axis

### F. DAMPER, RECTANGULAR, OPPOSED BLADES




| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| L/R | θ | | | | | | | | |
| | 80° | 70° | 60° | 50° | 40° | 30° | 20° | 10° | 0° Fully open |
| 0.3 | 807 | 284 | 73 | 21 | 9.0 | 4.1 | 2.1 | 0.85 | 0.52 |
| 0.4 | 915 | 332 | 100 | 28 | 11 | 5.0 | 2.2 | 0.92 | 0.52 |
| 0.5 | 1045 | 377 | 122 | 33 | 13 | 5.4 | 2.3 | 1.0 | 0.52 |
| 0.6 | 1121 | 411 | 148 | 38 | 14 | 6.0 | 2.3 | 1.0 | 0.52 |
| 0.8 | 1299 | 495 | 188 | 54 | 18 | 6.6 | 2.4 | 1.1 | 0.52 |
| 1.0 | 1521 | 547 | 245 | 65 | 21 | 7.3 | 2.7 | 1.2 | 0.52 |
| 1.5 | 1654 | 677 | 361 | 107 | 28 | 9.0 | 3.2 | 1.4 | 0.52 |

$$\frac{L}{R} = \frac{NW}{2(H + W)}$$ where:

N is number of damper blades
W is duct dimension parallel to blade axis
L is sum of damper blade lengths
R is perimeter of duct
H is duct dimension on perpendicular to blade axis

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–15 Loss Coefficients, Obstructions (Continued)

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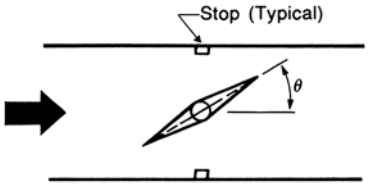
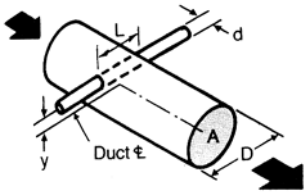
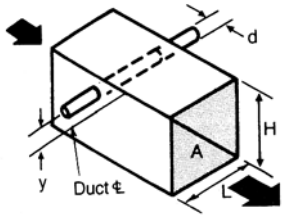
## D. DAMPER, RECTANGULAR, OPPOSED BLADES




| Coefficient C | | | | | | | |
|---|---|---|---|---|---|---|---|
| è | 0° | 10° | 20° | 30° | 40° | 50° | 60° |
| C | 0.50 | 0.65 | 1.6 | 4.0 | 9.4 | 24 | 67 |

## D. OBSTRUCTION, SMOOTH CYLINDER IN ROUND AND RECTANGULAR DUCTS




where:

$S_m$ = dL

$R_e$ = 8.56 (DV (Std. Air - US)

$R_e$ = 66.4 DV (Std. Air - Metric)

$S_m/A$ < 0.3

C = KC0

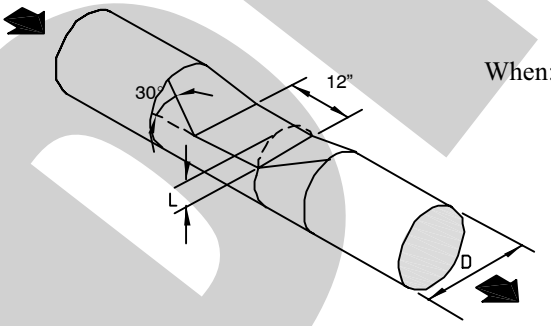
A = Area of duct

| Coefficient C | | | | |
|---|---|---|---|---|
| Re | $S_m/A$ | | | |
| | .05 | .10 | .15 | .20 |
| 0.1 | 3.9 | 8.4 | 14 | 19 |
| 0.5 | 1.5 | 3.2 | 5.2 | 7.1 |
| 1 | 0.66 | 1.4 | 2.3 | 3.2 |
| 5 | 0.30 | 0.64 | 1.1 | 1.4 |
| 10 | 0.17 | 0.38 | 0.62 | 0.84 |
| 50 | 0.11 | 0.24 | 0.38 | 0.52 |
| 100 | 0.10 | 0.21 | 0.35 | 0.47 |
| $(0.5\text{-}200) \times 10^3$ | 0.07 | 0.15 | 0.24 | 0.33 |
| $3 \times 10^5$ | 0.07 | 0.16 | 0.26 | 0.35 |
| $4 \times 10^5$ | 0.05 | 0.11 | 0.19 | 0.25 |
| $5 \times 10^5$ | 0.04 | 0.09 | 0.14 | 0.19 |
| $(6\text{-}10) \times 10^5$ | 0.02 | 0.05 | 0.07 | 0.10 |

For obstruction offset from the centerline use the following factors:

| y/D or y/H | 0 | 0.05 | 0.10 | 0.15 | 0.20 | 0.25 | 0.30 | 0.35 | 0.40 |
|---|---|---|---|---|---|---|---|---|---|
| K | 1.0 | 0.97 | 0.93 | 0.89 | 0.84 | 0.79 | 0.74 | 0.67 | 0.58 |

## I. ROUND DUCT, DEPRESSED TO AVOID AN OBSTRUCTION




When: L/D = 0.33

C = 0.24

Use the velocity pressure ($V_p$) of the downstream section. Fitting loss TP = C × $V_p$

**Table A–15 Loss Coefficients, Obstructions (Continued)**

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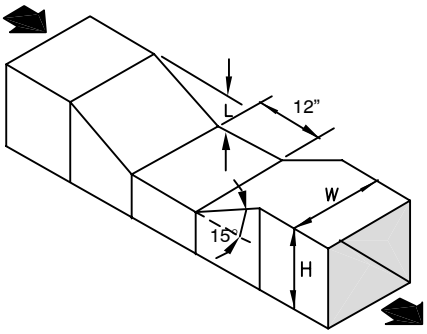
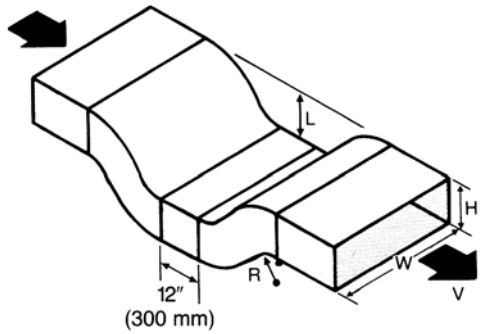
### J. RECTANGULAR DUCT, DEPRESSED TO AVOID AN OBSTRUCTION




| Coefficient C | | | | |
|---|---|---|---|---|
| W/H | L/H | | | |
| | 0.125 | 0.15 | 0.25 | 0.30 |
| 1.0 | 0.26 | 0.30 | 0.33 | 0.35 |
| 0.4 | 0.10 | 0.14 | 0.22 | 0.30 |

### K. RECTANGULAR DUCT WITH 4–45° SMOOTH RADIUS ELLS TO AVOID AN OBSTRUCTION




| Coefficient C | | | | | |
|---|---|---|---|---|---|
| Velocity fpm (m/s) | 800 (4) | 1200 (6) | 1600 (8) | 2000 (10) | 2400 (12) |
| C | 0.18 | 0.22 | 0.24 | 0.25 | 0.26 |

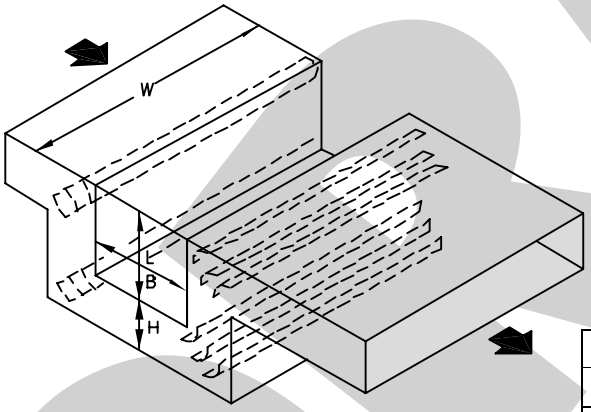
Where:

W/H = 4

R/H = 1

L/H = 1.5

### L. RECTANGULAR DUCT WITH 4–90° MITERED ELLS TO AVOID OBSTRUCTION




| Coefficient C | | | | |
|---|---|---|---|---|
| L/H Ratio | 0.5 | 1.0 | 1.5 | 2 |
| Single Blade Turning Vanes | — | 0.86 | 0.83 | 0.77 |
| Double Blade Turning Vanes | — | 1.85 | 2.84 | 2.91 |
| "S" type Splitter Vanes | 0.61 | 0.65 | — | — |
| No Vanes - Up to 1200 fpm | 0.88 | 5.26 | 6.92 | 7.56 |
| No Vanes - Over 1200 fpm | 1.26 | 6.22 | 8.82 | 9.24 |

Where:

W/H = 1.0 to 3.0

B = 12" to 12"

Use the velocity pressure ($V_{p)}$ of the downstream section. Fitting loss TP = C × $V_p$

## Table A–15 Loss Coefficients, Obstructions (Continued)

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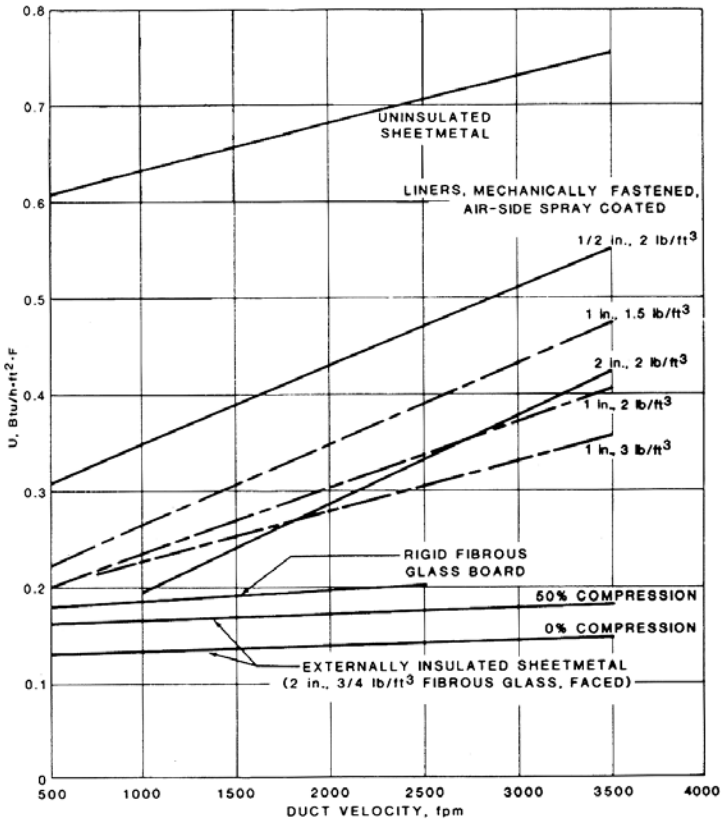
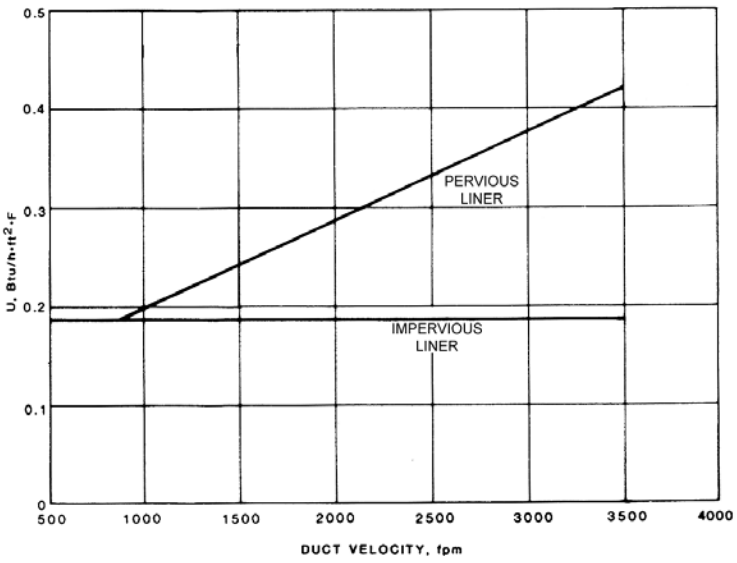
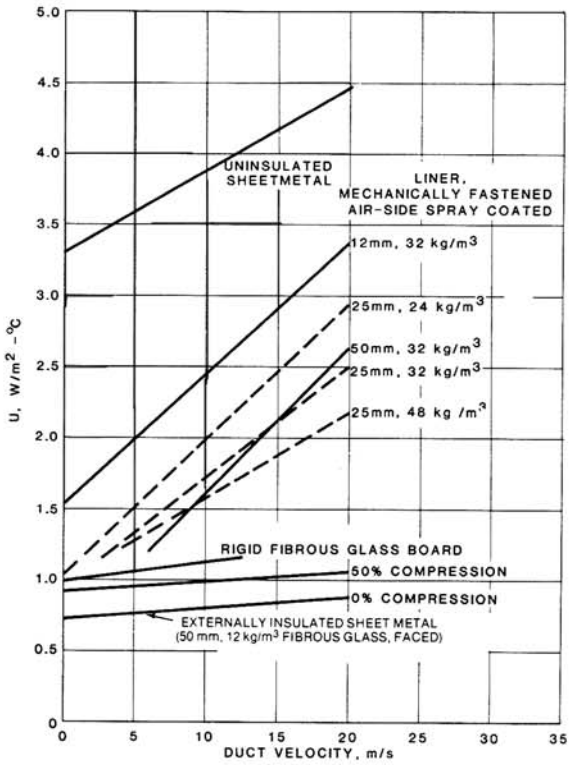



**a. Rigid Ducts**

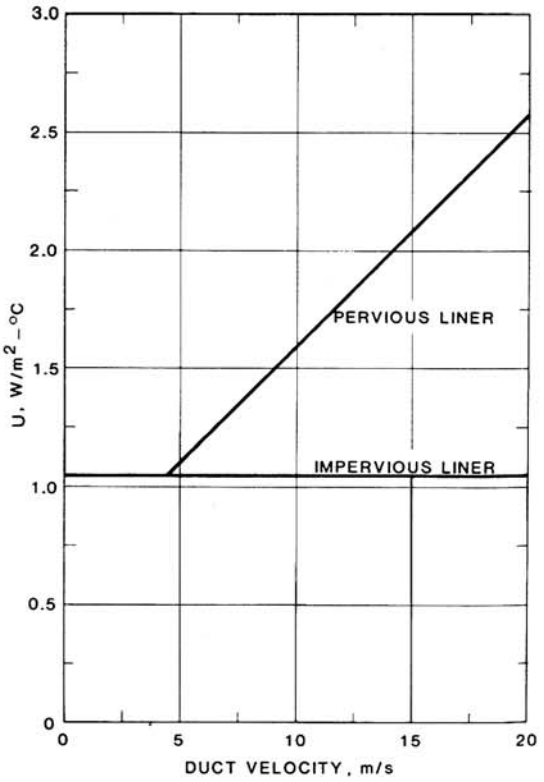



**b. Flexible Ducts**

**FIGURE A–5 DUCT HEAT TRANSFER COEFFICIENTS**




**a. Rigid Ducts**




**b. Flexible Ducts**

**FIGURE A–5M DUCT HEAT TRANSFER COEFFICIENTS**

# HVAC EQUATIONS (I–P)

## Air Equations (I–P)

a. $V = 1096 \frac{\sqrt{V_p}}{d}$

or for standard air ($d$ = 0.075 lb/cu ft):

$V = 4005 \sqrt{V_p}$

to solve for $d$:

$d = 1.325 \frac{P_b}{T}$

$V$ = Velocity (fpm)
$V_p$ = Velocity Pressure (in. wg)
$d$ = Density (lb/cu ft)
$P$ = Absolute Static Pressure (in. Hg) (Barometric pressure + static pressure)
$T$ = Absolute Temperature (460° + °F)

b. $Q\ (sens.) = 60 \times C_p \times d \times cfm \times \Delta t$

or for standard air ($Cp$ = 0.24 Btu/lb × °F):

$Q\ (sens.) = 1.08 \times cfm \times \Delta t$

$Q$ = Heat Flow (Btu/hr)
$Cp$ = Specific Heat (Btu/lb × °F)
$d$ = Density (lb/cu ft)
$\Delta t$ = Temperature Difference (°F)

c. $Q\ (lat.) = 4750 \times cfm \times \Delta W$ (lb.)

$Q\ (lat.) = 0.67 \times cfm \times \Delta W$ (gr.)

$\Delta W$ = Humidity Ratio (lb or gr $H_2O$/lb dry air)

d. $Q\ (total) = 4.5 \times cfm \times \Delta h$

$\Delta h$ = Enthalpy Difference (Btu/lb dry air)

e. $Q = A \times U \times \Delta t$

A = Area of Surface (sq ft)
U = Heat Transfer Coefficient (Btu/sq ft × hr × °F)

f. $R = \frac{1}{U}$

R = Sum of Thermal Resistance (sq ft × hr × °F / Btu)

g. $P_1 \frac{V_1}{T_1} = P_2 \frac{V_2}{T_2} = RM$

P = Absolute Pressure (lb/sq ft)
V = Total Volume (cu ft)
R = Gas Constant
T = Absolute Temperature (460° + °F = °R)
M = Mass (lb)

h. $TP = V_p + SP$

TP = Total Pressure (in. wg)
$V_p$ = Velocity Pressure (in. wg)
SP = Static Pressure (in. wg)

i. $V_p = \left(\frac{V}{4005}\right)^2$

j. $V = V_m \left[\frac{d\ (other\ than\ standard)}{0.075\ (d = std.air)}\right]$

V = Velocity (fpm)
$V_m$ = Measured Velocity (fpm)
d = Density (lb/cu ft)

k. $cfm = A \times V$

A = Area of Duct cross section (sq ft)

l. $TP = C \times V_p$

C = Duct Fitting Loss Coefficient

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### Fan Equations (I–P)

a. $\frac{cfm_2}{cfm_1} = \frac{rpm_2}{rpm_1}$

b. $\frac{P_2}{P_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$

c. $\frac{bhp_2}{bhp_1} = = \left(\frac{rpm_2}{rpm_1}\right)^3$

d. $\frac{d_2}{d_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$

e. $\frac{rpm\ fan}{rpm\ motor} = \frac{Pitch\ diam.\ motor\ pulley}{Pitch\ diam.\ fan\ pulley}$

$cfm$ = Cubic feet per minute
$rpm$ = Revolutions per minute
$P$ = Static or Total Pressure (in. wg)
$bhp$ = Brake horsepower
$d$ = Density (lb/cu ft)

### Pump Equations (I–P)

a. $\frac{gpm_2}{gpm_1} = \frac{rpm_2}{rpm_1}$

b. $\frac{gpm_2}{gpm_1} = \frac{D_2}{D_1}$

c. $\frac{H_2}{H_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$

d. $\frac{H_2}{h_1} = \left(\frac{D_2}{D_1}\right)^2$

e. $\frac{bhp_2}{bhp_1} = \left(\frac{rpm_2}{rpm_1}\right)^3$

f. $\frac{bhp_2}{bhp_1} = \left(\frac{D_2}{D_1}\right)^3$

$rpm$ = Revolutions per minute
$D$ = Impeller diameter
$H$ = Head (ft wg)
$bhp$ = Brake horsepower

### Hydronic Equivalents (I–P)

a. One gallon water = 8.33 pounds

b. Specific heat ($C_p$) water = 1.00 Btu/lb · °F (@68°F)

c. Specific heat ($C_p$) water vapor = 0.45 Btu/lb · °F (@68°F)

d. One ft of water = 0.433 psi

e. One psi = 2.3 ft wg = 2.04 in. Hg

f. One cu ft of water = 62.4 lb = 7.49 gal.

g. One in. of mercury (Hg) = 13.6 in. wg = 1.13 ft. wg

h. Atmospheric Pressure = 29.92 in. Hg = 14.696 psi

i. One psi = 2.31 ft. wg = 2.04 in. hg

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### Hydronic Equations (I–P)

a. $Q = 500 \times gpm \times \Delta t$

$Q$ = Heat Flow (Btu/hr)
$gpm$ = Gallons per minute
$\Delta t$ = Temperature Difference (°F)

b. $\frac{\Delta P_2}{\Delta P_1} = \left(\frac{gpm_2}{gpm_1}\right)^2$

c. $\Delta P = \left(\frac{gpm}{C_v}\right)^2$

$\Delta P$ = Pressure difference (°F)
$C_v$ = Valve constant (dimensionless)

d. $whp = \frac{gpm \times H \times Sp.Gr.}{3960}$

e. $bhp = \frac{gpm \times H \times Sp.Gr.}{3960 \times E_p} = \frac{whp}{E_p}$

$whp$ = Water horsepower
$gpm$ = Gallons per minute
$bhp$ = Brake horsepower
$H$ = Head (ft wg)
$Sp.Gr.$ = Specific gravity (use 1.0 for water)
$E_p$ = Efficiency of pump

f. $E_p = \frac{whp \times 100}{bhp} (in\ percent)$

g. $NPSHA = P_a \pm P_s + \frac{V_2}{2g} - P_{vp}$

$NPSHA$ = Net positive suction head available
$P_a$ = Atm. pressure (use 34 ft wg)
$P_s$ = Pressure at pump centerline (ft wg)
$g$ = Gravity acceleration (32.2 ft/sec$^2$)

h. $h = f \times \frac{L}{D} \times \frac{V_2}{2g}$

$h$ = Head Loss (ft)
$f$ = Friction factor (dimensionless)
$L$ = Length of pipe (ft)
$D$ = Internal diameter (ft)
$V$ = Velocity (ft/sec)

### Electric Equations (I–P)

a. $Bhp = \frac{I \times E \times P.F. \times Eff.}{746}$ (Single Phase)

b. $Bhp = \frac{I \times E \times P.F. \times Eff. \times 1.73}{746}$ (Three Phase)

c. $E = IR$

d. $P = EI$

e. $\frac{F.L.Amps \times Voltage\ *}{Actual\ Voltage} = Actual\ F.L.Amps$

*NOTE:* ***** *refers to Nameplate ratings.*

| Water Temperature °F | 60°F | 150°F | 200°F | 250°F | 300°F | 340°F |
|---|---|---|---|---|---|---|
| Ft. head differential per inch Hg. differential | 1.046 | 1.07 | 1.09 | 1.11 | 1.15 | 1.165 |

**Table A–16 Converting Pressure In Inches of Mercury to Feet of Water at Various Water Temperatures**

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| Altitude (ft) | Sea Level | 1000 | 2000 | 3000 | 4000 | 5000 | 6000 | 7000 | 8000 | 9000 | 10,000 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Barometer (in. Hg) | 29.92 | 28.86 | 27.82 | 26.82 | 25.84 | 24.90 | 23.98 | 23.09 | 22.22 | 21.39 | 20.58 |
| (in. wg) | 407.5 | 392.8 | 378.6 | 365.0 | 351.7 | 338.9 | 326.4 | 314.3 | 302.1 | 291.1 | 280.1 |
| Air Temp. °F | | | | | | | | | | | |
| -40° | 1.26 | 1.22 | 1.17 | 1.13 | 1.09 | 1.05 | 1.01 | 097 | 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 | 0.58 | 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 |
| Standard Air Density, Sea Level, 70°F = 0.0 75 lb/cu ft at 29. 92 in. Hg | | | | | | | | | | | |

**Table A–17 Air Density Correction Factors**

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| Altitude (m) | Sea Level | 250 | 500 | 750 | 1000 | 1250 | 1500 | 1750 | 2000 | 2500 | 3000 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Barometer (kPa) | 101.3 | 98.3 | 96.3 | 93.2 | 90.2 | 88.2 | 85.1 | 83.1 | 80.0 | 76.0 | 71.9 |
| Air Temp °C | | | | | | | | | | | |
| 0° | 1.08 | 1.05 | 1.02 | 0.99 | 0.96 | 0.93 | 0.91 | 0.88 | 0.86 | 0.81 | 0.76 |
| 20° | 1.00 | 0.97 | 0.95 | 0.92 | 0.89 | 0.87 | 0.84 | 0.82 | 0.79 | 0.75 | 0.71 |
| 50° | 0.91 | 0.89 | 0.86 | 0.84 | 0.81 | 0.79 | 0.77 | 0.75 | 0.72 | 0.68 | 0.64 |
| 75° | 0.85 | 0.82 | 0.80 | 0.78 | 0.75 | 0.73 | 0.71 | 0.69 | 0.67 | 0.63 | 0.60 |
| 100° | 0.79 | 0.77 | 0.75 | 0.72 | 0.70 | 0.68 | 0.66 | 0.65 | 0.63 | 0.59 | 0.56 |
| 125° | 0.74 | 0.72 | 0.70 | 0.68 | 0.66 | 0.64 | 0.62 | 0.60 | 0.59 | 0.55 | 0.52 |
| 150° | 0.70 | 0.68 | 0.66 | 0.64 | 0.62 | 0.60 | 0.59 | 0.57 | 0.55 | 0.52 | 0.49 |
| 175° | 0.66 | 0.64 | 0.62 | 0.62 | 0.59 | 0.57 | 0.55 | 0.54 | 0.52 | 0.44 | 0.46 |
| 200° | 0.62 | 0.61 | 0.59 | 0.57 | 0.56 | 0.54 | 0.52 | 0.51 | 0.49 | 0.47 | 0.44 |
| 225° | 0.59 | 0.58 | 0.56 | 0.54 | 0.53 | 0.51 | 0.50 | 0.48 | 0.47 | 0.44 | 0.42 |
| 250° | 0.56 | 0.55 | 0.53 | 0.52 | 0.50 | 0.49 | 0.47 | 0.46 | 0.45 | 0.42 | 0.40 |
| 275° | 0.54 | 0.52 | 0.51 | 0.49 | 0.48 | 0.47 | 0.45 | 0.44 | 0.43 | 0.40 | 0.38 |
| 300° | 0.51 | 0.50 | 0.49 | 0.47 | 0.46 | 0.45 | 0.43 | 0.42 | 0.41 | 0.38 | 0.36 |
| 325° | 0.49 | 0.48 | 0.47 | 0.45 | 0.44 | 0.43 | 0.41 | 0.40 | 0.39 | 0.37 | 0.35 |
| 350° | 0.47 | 0.46 | 0.45 | 0.43 | 0.42 | 0.41 | 0.40 | 0.39 | 0.38 | 0.35 | 0.33 |
| 375° | 0.46 | 0.44 | 0.43 | 0.42 | 0.41 | 0.39 | 0.38 | 0.37 | 0.36 | 0.34 | 0.32 |
| 400° | 0.44 | 0.43 | 0.41 | 0.40 | 0.39 | 0.38 | 0.37 | 0.36 | 0.35 | 0.33 | 0.31 |
| 425° | 0.42 | 0.41 | 0.40 | 0.39 | 0.38 | 0.37 | 0.35 | 0.34 | 0.33 | 0.32 | 0.30 |
| 450° | 0.41 | 0.40 | 0.38 | 0.37 | 0.36 | 0.35 | 0.34 | 0.33 | 0.32 | 0.31 | 0.29 |
| 475° | 0.39 | 0.38 | 0.37 | 0.36 | 0.35 | 0.34 | 0.33 | 0.32 | 0.31 | 0.29 | 0.28 |
| 500° | 0.38 | 0.37 | 0.36 | 0.35 | 0.34 | 0.33 | 0.32 | 0.31 | 0.30 | 0.28 | 0.27 |
| 525° | 0.37 | 0.36 | 0.35 | 0.34 | 0.33 | 0.32 | 0.31 | 0.30 | 0.29 | 0.27 | 0.26 |

**Table A–17M Air Density Correction Factors**

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# HVAC EQUATIONS (SI)

## Air Equations (SI)

a. $V = 1.414\sqrt{\frac{V_p}{d}}$

or for standard air (d = 1.204 kg/m³):

$V = \sqrt{1.66V_p}$

To solve for *d*:

$d = 3.48\frac{P_b}{T}$

$V$ = Velocity (m/s)
$V_p$ = Velocity Pressure (pascals or Pa)
$d$ = Density (kg/m³)
$P_b$ = Absolute Static Pressure (kPa) (Barometric pressure + static pressure)
$T$ = Absolute Temp. (273° + °C = °K)

b. $Q = C_p \times d \times L/s \times \Delta t$

or for standard air ($C_P$ = 1.005 kJ/kg · °C)

$Q(sens.) = 1.23 \times L/s \times \Delta t$ (in watts)

$Q(sens.) = 1.23 \times m^3/s \times \Delta t$ (in kilowatts)

$Q$ = Heat Flow (watts or kilowatts)
$C_p$ = Specific Heat (kJ/kg · °C)
$d$ = Density (kg/m³)
$\Delta t$ = Temperature Difference (°C)
$m^3/s$ = Airflow (cubic meters per second)
$\Delta W$ = Humidity Ratio ($gH_2O$/kg dry air)

c. $Q(lat.) = 3.0 \times L/s \times \Delta W$

d. $Q(total\ heat) = 1.20 \times L/s \times \Delta h$

Δh = Enthalpy Diff. (kJ/kg dry air)

e. $Q = A \times U \times \Delta t$

$A$ = Area of Surface (m²)
$U$ = Heat Transfer Coefficient (W/m² · °C)

f. $R = \frac{1}{U}$

$R$ = Sum of Thermal Resistances (m² · °C/W)

g. $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = RM$

$P$ = Absolute Pressure (kPa)
$V$ = Total Volume (m³)
$T$ = Absolute Temp. (273° + °C = °K)
$R$ = Gas Constant
$M$ = Mass (kg)

h. $TP = V_P + SP$

$TP$ = Total Pressure (Pa)
$V_p$ = Velocity Pressure (Pa)
$SP$ = Static Pressure (Pa)

i. $V_p = \frac{d}{2} \times V^2 = 0.602\,V^2$

$d$ = Density (kg/m³)
$V$ = Velocity (m/s)
$V_m$ = Measured Velocity (m/s)

j. $V = V_m\left[\frac{d\ (other\ than\ standard)}{1.204(d\ =\ std.\ air)}\right]$

k. $L/s = 1000 \times A \times V$

l. $TP = C \times V_P$

A = Area of duct cross section (m2)
C = Duct Fitting Loss Coefficient

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## Fan Equations (SI)

a. $\frac{L/s_2}{L/s_1} = \frac{m^3/s_2}{m^3/s_1} = \frac{rad/s_2}{rad/s_1} = \frac{rev/s_2}{rev/s_1}$

b. $\frac{P_2}{P_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^2$

c. $\frac{kW_2}{kW_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^3$

d. $\frac{d_2}{d_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^2$

e. $\frac{rad/s\ (fan)}{rad/s\ (motor)} = \frac{\text{Pitch diam. motor pulley}}{\text{Pitch. diam. fan pulley}}$

$L/s$ = Liters per second
$m^3/s$ = Cubic meters per second
$rad/s$ = Radians per second
$P$ = Static or Total Pressure (Pa)
$kW$ = Kilowatts
$d$ = Density (kg/m³)

## Pump Equations (SI)

a. $\frac{L/s_2}{L/s_1} = \frac{m^3/s_2}{m^3/s_1} = \frac{rad/s_2}{rad/s_1} = \frac{rev/s_2}{rev/s_1}$

b. $\frac{m^3/s_2}{m^3/s_1} = \frac{d_2}{d_1}$

c. $\frac{H_2}{H_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^2$

d. $\frac{H_2}{H_1} = \left(\frac{D_2}{D_1}\right)^2$

e. $\frac{BP_2}{BP_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^3$

$L/s$ = Liters per second
$m^3/s$ = Cubic meters per second
$rad/s$ = Radians per second
$D$ = Impeller diameter
$H$ = Head (kPa)
$BP.$ = Brake horsepower

## Hydronic Equations (SI)

a. $Q = 4190 \times m^3/s \times \Delta t$

$Q$ = Heat flow (kilowatts)
$\Delta t$ = Temperature difference (°C)
$m^3/s$ = Cubic meters per second (used for large volumes)

b. $\frac{\Delta P_2}{\Delta P_1} = \left(\frac{m^3/s_2}{m^3/s_1}\right)^2 = \left(\frac{L/s_2}{L/s_1}\right)^2$

$L/s$ = Liters per second

c. $\Delta P = \left(\frac{m^3/s}{C_v}\right)^2 = \left(\frac{L/s}{C_v}\right)^2$

$\Delta P$ = Pressure diff. (Pa or kPa)
$C_v$ = Valve constant (dimensionless)

d. $WP\ (\text{kW}) = 9.81 \times m^3/s \times H\ (\text{m}) \times Sp.Gr.$

or $WP(\text{W}) = \frac{L/s \times H\ (\text{Pa}) \times Sp.Gr.}{1002}$

$WP$ = Water power (kW or W)
$m^3/s$ = Cubic meters per second
$H$ = Head (Pa or m)
$L/s$ = Liters per second
$Sp.Gr.$ = Specific gravity (use 1.0 for water)

e. $BP = \frac{WP}{E_p}$

BP = Brake power (kW)
$E_p$ = Efficiency of Pump

f. $E_p = \frac{WP \times 100}{BP}$ (in percent)

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g. $NPSHA = P_a \pm P_s + \frac{V^2}{2g} - P_{vp}$

*NPSHA* = Net positive suction head available
$P_a$ = Atm. press (Pa)
(Std. Atm. press. = 101,325 Pa)
$P_s$ = Pressure at pump centerline (Pa)
$\frac{V^2}{2g}$ = Velocity head at point $P_s$ (m)
$P_{vp}$ = Absolute vapor pressure (Pa)

h. $h = f \times \frac{L}{D} \times \frac{V^2}{2g}$

$h$ = Head loss (m)
$g$ = Gravity acceleration (9.807 m/s²)
$f$ = Friction factor (dimensionless)
$L$ = Length of pipe (m)
$D$ = Internal diameter (m)
$V$ = Velocity (m/s)

a. $kW = \frac{I \times E \times P.F. \times Eff.}{1000}$ (Single Phase)

b. $kW = \frac{I \times E \times P.F. \times Eff. \times 1.73}{1000}$
(Three Phase)

c. $E = IR$

d. $P = EI$

e. $\frac{F.L.\ Amps\ * \times Voltage\ *}{Actual\ Voltage} = Actual\ F.L.Amps$

$kW$ = Kilowatts
$I$ = Amps (A)
$E$ = Volts (V)
$P.F.$ = Power factor
$R$ = ohms (Ω)
$P.$ = watts (W)

*NOTE: * Refers to Nameplate Ratings*

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## SI UNITS AND EQUIVALENTS

| UNIT | SYMBOL | QUANTITY | EQUIVALENT OR RELATIONSHIP |
|---|---|---|---|
| ampere | A | Electric current | Same as I-P |
| candela | cd | Luminous intensity | 1 cd/$m^2$ = 0.292 ft. lamberts |
| Celsius | °C | Temperature | °F = 1.8 °C + 32° |
| coulomb | C | Electric charge | Same as I-P |
| farad | F | Electric capacitance | Same as I-P |
| henry | H | Electric inductance | Same as I-P |
| hertz | Hz | Frequency | Same as cycles per second |
| joule | J | Energy, work, heat | 1 J = 0.7376 ft-lb<br>= 0.000948 Btu |
| kelvin | K | Thermodynamic temperature | °K = °C + 273.15°<br>= $\frac{°F + 459.67}{1.8}$ |
| kilogram | kg | Mass | 1 kg = 2.2046 lb |
| liter | L | Liquid volume | 1 L = 1.056 qt = 0.264 gal |
| lumens | lm | Luminous flux | 1 lm/$m^2$ = 0.0929 ft candles |
| lux | lx | Illuminance | 1 lx = 0.0929 ft candles |
| meter | m | Length | 1 m = 3.281 ft |
| mole | mol | Amount of substance | — |
| newton | N | Force | 1 N = kg • $m/s^2$ = 0.2248 lb (force) |
| ohm | Ω | Electrical resistance | Same as I-P |
| pascal | Pa | Pressure, stress | 1 Pa = $N/m^2$ = 0.000145 psi<br>= 0.004022 in.wg |
| radian | rad | Plane angle | 1 rad = 57.29° |
| second | s | Time | Same as I-P |
| siemens | S | Electric conductance | — |
| steradian | sr | Solid angle | — |
| volt | v | Electric potential | Same as I-P |
| watt | w | Power, heat flow | 1 W = J/s = 3.4122 Btu/hr<br>1 W = 0.000284 tons of refrigeration |

**Table A–18 SI Units (Basic and Derived)**

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| QUANTITY | SYMBOL | UNIT | I-P RELATIONSHIP |
|---|---|---|---|
| acceleration | $m/s^2$ | meters per second squared | 1 $m/s^2$ = 3.281 $ft/sec^2$ |
| angular velocity | rad/s | radians per second | 1 rad/sec = 9.549 rpm = 0.159 rps |
| area | $m^2$ | square meter | 1 $m^2$ = 10.76 $ft^2$ |
| atmospheric pressure | — | 101.325 kPa | 29.92 in Hg = 14.696 psi |
| density | $kg/m^3$ | kilograms per cubic meter | 1 $kg/m^3$ = 0.0624 $lb/ft^3$ |
| density, air | — | 1.2 $kg/m^3$ | 0.075 $lb/ft^3$ |
| density, water | — | 1000 $kg/m^3$ | 62.4 $lb/ft^3$ |
| duct friction loss | Pa/m | pascals per meter | 1 Pa/m = 0.1224 in.wg/100 ft |
| enthalpy | kJ/kg | kilojoule per kilogram | 1 kJ/kg = 0.4299 Btu/lb dry air |
| gravity | | 9.8067 $m/s^2$ | 32.2 $ft/sec^2$ |
| heat flow | W | watt | 1 W = 3.412 Btu/hr |
| length (normal) | m | meter | 1 m = 3.281 ft = 39.37 in. |
| linear velocity | m/s | meters per second | 1 m/s = 196.9 fpm |
| mass flow rate | kg/s | kilograms per second | 1 kg/s = 7936.6 lb/hr |
| moment of inertia | $kg{\cdot}m^2$ | kilograms × square meter | 1 $kg{\cdot}m^2$ = 23.73 lb $ft^2$ |
| power | W | watt | 1 W = 0.00134 hp |
| pressure | kPa<br>Pa | kilopascal (1000 pascals)<br>pascal | 1 kPa = 0.296 in Hg = 0.145 psi<br>1 Pa = 0.004015 in.wg |
| specific heat—air ($C_p$) | — | 1000 J/kg °C | 1000 J/kg °C = 1 kJ/kg °C<br>= 0.2388 Btu/lb °F |
| specific heat—air ($C_v$) | — | 717 J/kg °C | 0.17 Btu/lb °F |
| specific heat—water | — | 4190 J/kg °C | 1.0 Btu/lb °F |
| specific volume | $m^3/kg$ | cubic meters per kilogram | 1 $m^3/kg$ = 16.019 $ft^3/lb$ |
| thermal conductivity | W $mm/m^2{\cdot}°C$ | watt millimeter per square meter °C | 1 W $mm/m^2 \cdot °C$<br>= 0.0069 Btuh $in/ft^2$ °F |
| volume flow rate | $m^3/s$<br>L/s<br>—<br>— | cubic meters per second<br>liters per second<br>1 $m^3/s$ = 1000 L/s<br>1 mL = liters/1000 | 1 $m^3/s$ = 2118.88 cfm (air)<br>1 L/s = 2.12 cfm (air)<br>1 $m^3/s$ = 15,850 gpm (water)<br>1 mL/s = 1.05 gph (water) |

**Table A–19 SI Equivalents**

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| Type of Area | NC-RC Level | Approx. dBA |
|---|---|---|
| RESIDENCES | | |
| Private home (rural and suburban) | 20-30 | 25-35 |
| Private home (urban) | 25-35 | 30-40 |
| Apartment house | 30-40 | 35-45 |
| HOTELS | | |
| Individual rooms | 30-40 | 35-45 |
| Ballroom, Banquet Am | 30-40 | 35-45 |
| Halls, corridors, lobbies | 35-45 | 40-50 |
| Garages | 40-50 | 45-55 |
| Kitchens, laundries | 40-50 | 45-55 |
| HOSPITALS & CLINICS | | |
| Private rooms | 25-35 | 30-40 |
| Operating rooms | 30-40 | 35-45 |
| Wards, corridors | 30-40 | 35-45 |
| Laboratories | 30-40 | 35-45 |
| Lobbies, waiting rms | 35-45 | 40-50 |
| Washrooms, toilets | 40-50 | 45-55 |
| OFFICES | | |
| Board rooms | 20-30 | 25-35 |
| Conference rooms | 25-35 | 30-40 |
| Executive offices | 30-40 | 35-45 |
| General offices | 30-45 | 35-50 |
| Reception rooms | 30-45 | 35-50 |
| General open offices | 35-45 | 40-50 |
| Drafting rooms | 35-45 | 40-50 |
| Halls & corridors | 40-60 | 45-65 |
| Tabulation and computation areas | 40-50 | 45-55 |
| AUDITORIUMS AND MUSIC HALLS | | |
| Concert, opera halls | 15-25 | 20-30 |
| Sound record studios | 15-25 | 20-30 |
| Legitimate theaters | 25-30 | 30-40 |
| Multi-purpose halls | 25-30 | 30-30 |
| Movie theaters | 30-35 | 35-40 |
| TV audience studios | 30-35 | 35-40 |
| Amphitheaters | 30-35 | 35-40 |
| Lecture halls | 30-35 | 35-40 |
| Planetariums | 30-35 | 35-40 |
| Lobbies | 35-45 | 40-50 |
| CHURCHES AND SCHOOLS | | |
| Sanctuaries | 20-30 | 25-35 |
| Libraries | 30-40 | 35-45 |
| Schools & Classrooms | 30-40 | 35-45 |
| Laboratories | 35-45 | 40-50 |
| Recreation halls | 35-50 | 40-55 |
| Corridors & halls | 35-50 | 40-55 |
| PUBLIC LIBRARIES | | |
| Libraries, museums | 30-40 | 35-45 |
| Court rooms | 30-40 | 35-45 |
| Post offices, lobbies | 35-45 | 40-50 |
| Gen. banking areas | 35-45 | 40-50 |
| Washrooms, toilets | 40-50 | 45-55 |
| RESTAURANTS, LOUNGES, CAFETERIAS | | |
| Restaurants | 35-45 | 40-50 |
| Cocktail lounges | 35-40 | 40-45 |
| Night clubs | 35-45 | 40-50 |
| Cafeterias | 40-50 | 45-55 |
| RETAIL STORES | | |
| Clothing stores | 35-45 | 40-50 |
| Department stores (upper floors) | 35-40 | 40-45 |
| Department stores (main floors) | 35-45 | 40-50 |
| Small retail stores | 40-50 | 45-55 |
| Supermarkets | 40-50 | 45-55 |
| INDOOR SPORTS ACTIVITIES | | |
| Coliseums | 30-40 | 35-45 |
| Bowling alleys | 35-45 | 40-50 |
| Gymnasiums | 35-46 | 40-50 |
| Swimming pools | 40-55 | 45-60 |
| TRANSPORTATION (RAIL, BUS, PLANES) | | |
| Ticket sales offices | 30-40 | 35-45 |
| Lounges, waiting rms | 35-50 | 40-55 |

**Table A–20 Recommended NC–RC Levels For Different Indoor Activity Areas**

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| NC-AC Level | Environment | Telephone Use | Listening Conditions |
|---|---|---|---|
| <20 | Silent | Excellent | Critical |
| 20-30 | Quiet | Excellent | Excellent |
| 30-40 | Moderately Noisy | Good | Good to Satisfactory |
| 40-50 | Noisy | Satisfactory | Satisfactory to Slightly Difficult |
| 50-60 | Very Noisy | Difficult | Difficult |

**Table A–21 Listening Conditions And Telephone Use As A Function Of NC–RC Levels**

| Sound Source | Path Nos. |
|---|---|
| Circulating fans; grills; diffusers; registers; unitary equipment in room | 1 |
| Induction coil and fan-powered mixing units | 1,2 |
| Unitary equipment located outside of room served; remotely located air handling equipment, such as, fans and blowers, dampers, duct fittings, and air washers | 2,3 |
| Compressors and pumps | 4,5,6 |
| Cooling towers; air cooled condensers | 4,5,6,7 |
| Exhaust fans; window air conditioners | 7,8 |
| Sound transmission between rooms | 9,10 |

| Transmission Paths | | Recommended Noise Reduction Methods |
|---|---|---|
| No. | Description | |
| 1 | Direct sound radiated from sound source to ear<br>Reflected sound from walls, ceiling, and walls | Direct sound can be controlled only by selecting quiet equipment<br>Reflected sound is controlled by adding sound absorption to room and to location of equipment |
| 2 | Air and structure borne sound radiated from casings and through walls of ducts and plenums is transmitted through walls and ceiling into room | Design ducts and fittings for low turbulence; locate high velocity ducts in noncritical areas; isolate ducts and sound plenums from structure with neoprene or spring hangers |
| 3 | Airborne sound radiated through supply and return air ducts to diffusers in room and then to listener by Path 1 | Select fans for minimum sound power; use ducts lined with sound absorbing material; use duct silencers or sound plenums in supply and return air ducts |
| 4 | Noise is transmitted through equipment room walls and floors to adjacent rooms | Locate equipment rooms away from critical areas; use masonry blocks or concrete for equipment room walls and floor |
| 5 | Building structure transmits vibration to adjacent walls and ceilings from which it is radiated as noise into room by Path 1 | Mount all machines on properly designed vibration isolators; design mechanical equipment room for dynamic loads; balance rotating and reciprocating equipment |
| 6 | Vibration transmission along pipe and duct walls | Isolate pipe and ducts from structure with neoprene or spring hangers; install flexible connectors between pipes, ducts, and vibrating machines |
| 7 | Noise radiated to outside enters room windows | Locate equipment away from critical areas; use barriers and covers to interrupt noise paths; select quiet equipment |
| 8 | Inside noise follows Path 1 | Select quiet equipment |
| 9 | Noise transmitted to diffuser in a room into ducts and out through an air diffuser in another room | Design and install duct attenuation to match transmission loss of wall between rooms |
| 10 | Sound transmission through, over, and around room partitions | Extend partition to ceiling slab and tightly seal all around; seal all pipe, conduit, and duct penetrations |

**Table A–22 Sound Sources, Transmission Paths, And Recommended Noise Reduction Methods**

DRAFT

| Fan Type | | 1/1 Octave Band Center Frequency - Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| Centrifugal | | | | | | | | | |
| Airfoil, Backward Curved, Backward Inclined | Wheel Diameter (in.) | | | | | | | | |
| | >36 in. | 40 | 40 | 39 | 34 | 30 | 23 | 19 | 17 |
| | <36 in. | 45 | 45 | 43 | 39 | 34 | 28 | 24 | 19 |
| Forward Curved | All | 53 | 53 | 43 | 36 | 36 | 31 | 26 | 21 |
| Radial | Total Press (in. wg) | | | | | | | | |
| Material | 4-10 | 56 | 47 | | 39 | 37 | 32 | | |
| Medium Pressure | 6-15 | | | | 42 | 38 | 33 | | |
| High Pressure | 15-60 | 61 | 58 | 53 | 48 | 46 | 44 | 41 | 38 |
| Vaneaxial | Hub Ratio | | | | | | | | |
| | 0.3-0.4 | 49 | 43 | 43 | 48 | 47 | 45 | 38 | 34 |
| | 0.4-0.6 | 49 | 43 | 46 | 43 | 41 | 36 | 30 | 28 |
| | 0.6-0.8 | 53 | 52 | 51 | 51 | 49 | 47 | 43 | 40 |
| Tubeaxial | Wheel Diameter (in.) | | | | | | | | |
| | >40 | 51 | 46 | 47 | 49 | 47 | 46 | 39 | 37 |
| | >40 | 48 | 47 | 49 | 53 | 52 | 51 | 43 | 40 |
| Propeller | | | | | | | | | |
| General ventilation and Cooling towers | All | 48 | 51 | 58 | 56 | 55 | 52 | 46 | 42 |

**Table A–23 Specific Sound Power Levels, $K_W$, For Fan Total Sound Power**

| Fan Type | 1/1 Octave Band in which BFI occurs | BFI dB |
|---|---|---|
| Centrifugal | | |
| Airfoil, backward curved, backward inclined | 250 Hz | 3 |
| Forward curved | 500 Hz | 2 |
| Radial blade, pressure blower | 125 Hz | 8 |
| Vaneaxlal | 125 Hz | 6 |
| Tube axial | 63 Hz | 7 |
| Propeller | | |
| Cooling Tower | 3 Hz | 5 |

**Table A–24 Blade Frequency Increments (BFI)**

| Static Efficiency % of Peak | Correction Factor dB |
|---|---|
| 90 to 100 | 0 |
| 85 to 89 | 3 |
| 75 to 85 | 6 |
| 65 o 74 | 9 |
| 55 to 64 | 12 |
| 50 to 54 | 15 |
| Below 50 | 16 |

**Table A–25 Correction Factor, C, For Off–Peak Operation**

DRAFT

| Duct Size (in. × in.) | Gage | 1/1 Octave Band Center Frequency—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 12 × 12 | 24 | 21 | 24 | 27 | 30 | 33 | 36 | 41 | 45 |
| 12 × 24 | 24 | 19 | 22 | 25 | 28 | 31 | 35 | 41 | 45 |
| 12 × 48 | 22 | 19 | 22 | 25 | 28 | 31 | 37 | 43 | 45 |
| 24 × 24 | 22 | 20 | 23 | 26 | 29 | 32 | 37 | 43 | 45 |
| 24 × 48 | 20 | 20 | 23 | 26 | 29 | 31 | 39 | 45 | 45 |
| 48 × 48 | 18 | 21 | 24 | 27 | 30 | 35 | 41 | 45 | 45 |
| 48 × 96 | 18 | 19 | 22 | 25 | 29 | 35 | 41 | 45 | 45 |

**Table A–26 $TL_{out}$ vs. Frequency For Various Rectangular Ducts**

Data are for duct lengths of 20 feet, but values may be used for the cross section shown regardless of length.

| Duct Size (in. × in.) | Gage | 1/1 Octave Band Center Frequency—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 12 × 12 | 24 | 16 | 16 | 16 | 25 | 30 | 33 | 38 | 42 |
| 12 × 24 | 24 | 15 | 15 | 17 | 25 | 28 | 32 | 38 | 42 |
| 12 × 48 | 22 | 14 | 14 | 22 | 25 | 28 | 34 | 40 | 42 |
| 24 × 24 | 22 | 13 | 13 | 21 | 26 | 29 | 34 | 40 | 42 |
| 24 × 48 | 20 | 12 | 15. | 23 | 26 | 28 | 36 | 42 | 42 |
| 48 × 48 | 18 | 10 | 19 | 24 | 27 | 32 | 38 | 42 | 42 |
| 48 × 96 | 18 | 11 | 19 | 22 | 26 | 32 | 38 | 42 | 42 |

**Table A–27 $TL_{in}$ vs. Frequency For Various Rectangular Ducts**

Data are for duct lengths of 20 feet, but values may be used for the cross section shown regardless of length.

| Duct Size (in.) | Length (ft.) | Gage | 1/1 Octave Band Center Frequency—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| Long Seam Ducts | | | | | | | | | | |
| 8 | 15 | 26 | 45 | 53 | 55 | 52 | 44 | 35 | 34 | 26 |
| 14 | 15 | 24 | 50 | 60 | 54 | 36 | 34 | 31 | 25 | 38 |
| 22 | 15 | 22 | 47 | 53 | 37 | 33 | 33 | 27 | 25 | 43 |
| 32 | 15 | 22 | 51 | 46 | 26 | 26 | 24 | 22 | 38 | 43 |
| Spiral Wound Ducts | | | | | | | | | | |
| 8 | 10 | 26 | 48 | 64 | 75 | 72 | 56 | 56 | 46 | 29 |
| 14 | 10 | 26 | 43 | 53 | 55 | 33 | 34 | 35 | 25 | 40 |
| 26 | 10 | 24 | 45 | 50 | 26 | 26 | 25 | 22 | 36 | 43 |
| 26 | 10 | 16 | 48 | 53 | 36 | 32 | 32 | 28 | 41 | 36 |
| 32 | 10 | 22 | 43 | 42 | 28 | 25 | 26 | 24 | 40 | 35 |

**Table A–28 Experimentally Measured $TL_{out}$ vs. Frequency For Round Ducts**

Data are for duct lengths of 20 feet, but values may be used for the cross section shown regardless of length.

DRAFT

| Duct Size (in.) | Length (ft.) | Gage | 1/1 Octave Band Center Frequency—Hz | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 |
| Long Seam Ducts | | | | | | | | | |
| 8 | 15 | 26 | 50 | 50 | 50 | 44 | 42 | 40 | 38 |
| 14 | 15 | 24 | 50 | 50 | 48 | 37 | 35 | 33 | 31 |
| 22 | 15 | 22 | 50 | 50 | 38 | 32 | 30 | 28 | 26 |
| 32 | 15 | 22 | 50 | 44 | 29 | 26 | 24, | 22 | 37 |
| Spiral Wound Duct | | | | | | | | | |
| 8 | 10 | 26 | 50 | 50 | 50 | 46 | 42 | 40 | 38 |
| 14 | 10 | 26 | 50 | 50 | 48 | 35 | 33 | 31 | 29 |
| 26 | 10 | 24 | 45 | 45 | 35 | 27 | 25 | 23 | 38 |
| 26 | 10 | 16 | 50 | 50 | 42 | 34 | 32 | 30 | 45 |
| 32 | 10 | 22 | 50 | 47 | 32 | 26 | 24 | 22 | 37 |

**Table A–29 Calculated $TL_{out}$ vs. Frequency For Round Ducts**

| Duct Size (in.) | Length (ft.) | Gage | 1/1 Octave Band Center Frequency—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| Long Seam Ducts | | | | | | | | | | |
| 8 | 15 | 26 | 17 | 31 | 39 | 42 | 41 | 32 | 31 | 23 |
| 14 | 15 | 24 | 27 | 43 | 43 | 31 | 31 | 28 | 22 | 35 |
| 22 | 15 | 22 | 28 | 40 | 30 | 30 | 30 | 24 | 22 | 40 |
| 32 | 15 | 22. | 35 | 36 | 23 | 23 | 21 | 19 | 35 | 40 |
| Spiral Wound Ducts | | | | | | | | | | |
| 8 | 10 | 26 | 20 | 42 | 59 | 62 | 53 | 43 | 26 | 26 |
| 14 | 10 | 26 | 20 | 36 | 44 | 28 | 31 | 32 | 22 | 37 |
| 26 | 10 | 24 | 27 | 38 | 20 | 23 | 22 | 19 | 33 | 40 |
| 26 | 10 | l6 | 30 | 41 | 30 | 29 | 29 | 25 | 38 | 33 |
| 32 | 10 | 22 | 27 | 32 | 25 | 22 | 23 | 21 | 37 | 42 |

**Table A–30 Experimentally Determined $TL_{in}$ vs. Frequency For Round Ducts**

| Duct Size (in.) | Length (ft.) | Gage | 1/1 Octave Band Center Frequency—Hz | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 |
| Long Seam Ducts | | | | | | | | | |
| 8 | 15 | 26 | 17 | 23 | 29 | 34 | 39 | 37 | 35 |
| 14 | 15 | 24 | 22 | 28 | 34 | 32 | 32 | 30 | 28 |
| 22 | 15 | 22 | 26 | 32 | 31 | 29 | 27 | 25 | 23 |
| 32 | 15 | 22 | 29 | 34 | 26 | 23 | 21 | 19 | 34 |
| Spiral Wound Ducts | | | | | | | | | |
| 8 | 10 | 26 | 17 | 23 | 29 | 35 | 39 | 37 | 35 |
| 14 | 10 | 26 | 27 | 38 | 37 | 30 | 30 | 28 | 26 |
| 26 | 10 | 24 | 27 | 33 | 29 | 24 | 22 | 20 | 35 |
| 26 | 10 | l6g | 27 | 33 | 36 | 31 | 29 | 27 | 42 |
| 32 | 10 | 22 | 29 | 35 | 29 | 23 | 21 | 19 | 34 |

**Table A–31 Calculated $TL_{in}$ vs. Frequency For Round Ducts**

DRAFT

| Duct Size (in. × in.) | Gage | 1/1 Octave Band Center Frequency—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 12 x 6 | 24 | 31 | 34 | 37 | 40 | 43 | — | — | |
| 24 x 6 | 24 | 24 | 27 | 30 | 33 | 36 | — | — | — |
| 24 x 12 | 24 | 28 | 31 | 34 | 37 | — | — | — | — |
| 48 x 12 | 22 | 23 | 26 | 29 | 32 | — | — | — | — |
| 48 x 24 | 22 | 27 | 30 | 33 | — | — | — | — | — |
| 96 x 24 | 2O | 22 | 25 | 28 | — | — | — | — | — |
| 96 x 48 | 18 | 28 | 31 | — | — | — | — | — | — |

**Table A–32 $TL_{out}$ vs. Frequency For Various Flat–Oval Ducts**

| Duct Size (in. × in.) | Gage | 1/1 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 | 2O | 11 | 22 | 25 | — | — | — | — | — |
| 96 x 48 | 18 | 19 | 28 | — | — | — | — | — | — |

**Table A–33 $TL_{in}$ vs. Frequency For Various Flat–Oval Ducts**

| | 1/1 Octave Band Center Frequency—Hz | | | | | | |
|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 |
| Non-Sound Absorbing Materials | | | | | | | |
| Concrete | 0.01 | 0.01 | 0.01 | 0.02 | 0.02 | 0.02 | 0.03 |
| Bare Sheet Metal | 0.04 | 0.04 | 0.04 | 0.05 | 0.05 | 0.05 | 0.07 |
| Sound Absorbing Materials | | | | | | | |
| 1 in. 3.0 lb/ft$^3$ Fiberglass Insulation Board | 0.02 | 0.03 | 0.22 | 0.69 | 0.91 | 0.96 | 0.99 |
| 2 in. 3.0 lb/ft$^3$ Fiberglass Insulation Board | 0.18 | 0.22 | 0.82 | 1.00 | 1.00 | 1.00 | 1.00 |
| 3 in. 3.0 lb/ft$^3$ Fiberglass Insulation Board | 0.48 | 0.53 | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 |
| 4 in. 3.0 lb/ft$^3$ Fiberglass Insulation Board | 0.76 | 0.84 | 1.00 | 1.00 | 1.00 | 1.00 | 0.97 |

**Table A–34 Absorption Coefficients For Selected Plenum Materials**

| Duct Size in. × in. | P/A 1/ft | Attenuation - dB/ft 1/1 Octave Band Center Freq.—Hz | | | |
|---|---|---|---|---|---|
| | | 63 | 125 | 250 | Above 250 |
| 6 × 6 | 8.0 | 0.30 | 0.20 | 0.10 | 0.10 |
| 12 × 12 | 4.0 | 0.35 | 0.20 | 0.10 | 0.06 |
| 12 × 24 | 3.0 | 0.40 | 0.20 | 0.10 | 0.05 |
| 24 × 24 | 2.0 | 0.25 | 0.20 | 0.10 | 0.03 |
| 48 × 48 | 1.0 | 0.15 | 0.10 | 0.07 | 0.02 |
| 72 × 72 | 0.7 | 0.10 | 0.10 | 0.05 | 0.02 |

**Table A–35 Sound Attenuation In Unlined Rectangular Sheet Metal Ducts**

If duct is externally lined, multiply results associated with 63 Hz, 125 Hz and 250 Hz by 2.

DRAFT

| Dimensions in. × in. | Insertion Loss - dB/ft 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 4 × 4 | 2.00 | 2.06 | 2.18 | 3.66 | 10.10 | 10.36 | 4.80 | 2.87 |
| 6 | 1.49 | 1.63 | 1.89 | 3.33 | 8.90 | 8.95 | 4.41 | 2.74 |
| 8 | 1.28 | 1.43 | 1.75 | 3.16 | 8.27 | 8.22 | 4.21 | 2.67 |
| 10 | 1.16 | 1.32 | 1.66 | 3.05 | 7.80 | 7.78 | 4.07 | 2.63 |
| 6 × 6 | 1.08 | 1.24 | 1.60 | 2.98 | 7.62 | 7.48 | 3.98 | 2.60 |
| 10 | 0.82 | 0.96 | 1.35 | 2.66 | 6.52 | 6.26 | 3.59 | 2.46 |
| 12 | 0.77 | 0.90 | 1.29 | 2.57 | 6.23 | 5.94 | 3.49 | 2.42 |
| 18 | 0.69 | 0.79 | 1.18 | 2.42 | 5.74 | 5.41 | 3.31 | 2.36 |
| 8 × 8 | 0.77 | 0.90 | 1.29 | 2.57 | 6.23 | 5.94 | 3.49 | 2.42 |
| 12 | 0.65 | 0.74 | 1.12 | 2.34 | 5.49 | 5.13 | 3.21 | 2.32 |
| 18 | 0.60 | 0.67 | .1.04 | 2.22 | 5.10 | 4.72 | 3.06 | 2.26 |
| 24 | 0.56 | 0.60 | 0.96 | 2.09 | 4.70 | 4.29 | 2.90 | 2.20 |
| 10 × 10 | 0.63 | 0.71 | 1.09 | 2.29 | 5.34 | 4.97 | 3.15 | 2.30 |
| 16 | 0.55 | 0.59 | 0.94 | 2.06 | 4.62 | 4.21 | 2.86 | 2.19 |
| 20 | 0.53 | 0.55 | 0.89 | 1.98 | 4.37 | 3.94 | 2.76 | 2.15 |
| 30 | 0.51 | 0.51 | 0.82 | 1.87 | 4.02 | 3.59 | 2.62 | 2.09 |
| 12 × 12 | 0.56 | 0.60 | 0.96 | 2.09 | 4.70 | 4.29 | 2.90 | 2.20 |
| 18 | 0.51 | 0.52 | 0.85 | 1.90 | 4.14 | 3.71 | 2.67 | 2.11 |
| 24 | 0.50 | 0.48 | 0.79 | 1.81 | 3.85 | 3.41 | 2.54 | 2.06 |
| 36 | 0.40 | 0.43 | 0.74 | 1.70 | 3.54 | 3.10 | 2.41 | 2.00 |
| 15 × 15 | 0.51 | 0.51 | 0.82 | 1.87 | 4.02 | 3.59 | 2.62 | 2.09 |
| 22 | 0.40 | 0.44 | 0.75 | 1.71 | 3.57 | 3.12 | 2.42 | 2.01 |
| 30 | 0.36 | 0.39 | 0.68 | 1.61 | 3.29 | 2.85 | 2.29 | 1.96 |
| 45 | 0.32 | 0.34 | 0.62 | 1.52 | 3.03 | 2.59 | 2.17 | 1.90 |
| 18 × 18 | 0.40 | 0.43 | 0.74 | 1.70 | 3.54 | 3.10 | 2.41 | 2.00 |
| 28 | 0.33 | 0.35 | 0.64 | 1.54 | 3.09 | 2.65 | 2.20 | 1.91 |
| 36 | 0.30 | 0.32 | 0.59 | 1.47 | 2.90 | 2.46 | 2.11 | 1.87 |
| 54 | 0.27 | 0.28 | 0.54 | 1.38 | 2.67 | 2.24 | 2.00 | 1.82 |
| 24 × 24 | 0.30 | 0.32 | 0.59 | 1.47 | 2.90 | 2.46 | 2.11 | 1.87 |
| 36 | 0.25 | 0.26 | 0.51 | 1.34 | 2.55 | 2.13 | 1.94 | 1.80 |
| 48 | 0.23 | 0.24 | 0.47 | 1.27 | 2.37 | 1.95 | 1.85 | 1.76 |
| 72 | 0.21 | 0.21 | 0.43 | 1.20 | 2.19 | 1.78 | 1.75 | 1.71 |
| 30 × 30 | 0.24 | 0.25 | 0.49 | 1.31 | 2.48 | 2.06 | 1.91 | 1.78 |
| 45 | 0.21 | 0.21 | 0.43 | 1.20 | 2.19 | 1.78 | 1.75 | 1.71 |
| 60 | 0.19 | 0.19 | 0.39 | 1.13 | 2.03 | 1.63 | 1.67 | 1.67 |
| 36 × 36 | 0.21 | 0.21 | 0.43 | 1.20 | 2.19 | 1.78 | 1.75 | 1.71 |
| 54 | 0.18 | 0.17 | 0.37 | 1.09 | 1.93 | 1.54 | 1.61 | 1.64 |
| 72 | 0.16 | 0.16 | 0.34 | 1.03 | 1.79 | 1.41 | 1.54 | 1.60 |
| 42 × 42 | 0.18 | 0.18 | 0.38 | 1.11 | 1.96 | 1.57 | 1.63 | 1.65 |
| 64 | 0.16 | 0.15 | 0.32 | 1.01 | 1.72 | 1.35 | 1.50 | 1.58 |
| 84 | 0.14 | 0.14 | 0.30 | 0.96 | 1.61 | 1.25 | 1.43 | 1.55 |
| 48 × 48 | 0.16 | 0.16 | 0.34 | 1.03 | 1.79 | 1.41 | 1.54 | 1.60 |
| 72 | 0.14 | 0.13 | 0.29 | 0.94 | 1.58 | 1.22 | 1.42 | 1.54 |

**Table A–36 Insertion Loss For Rectangular Ducts With 1 in. Of Fiberglass Lining**

DRAFT

| Dimensions in. × in. | Insertion Loss - dB/ft 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 4 × 4 | 3.54 | 3.81 | 4.52 | 7.61 | 10.10 | 10.36 | 4.80 | 2.87 |
| 6 | 2.57 | 2.99 | 3.91 | 6.94 | 8.90 | 8.95 | 4.41 | 2.74 |
| 8 | 2.15 | 2.60 | 3.59 | 6.58 | 8.27 | 8.22 | 4.21 | 2.67 |
| 10 | 1.92 | 2.38 | 3.40 | 6.36 | 7.88 | 7.78 | 4.07 | 2.63 |
| 6 × 6 | 1.78 | 2.23 | 3.27 | 6.20 | 7.62 | 7.48 | 3.98 | 2.60 |
| 10 | 1.27 | 1.69 | 2.74 | 5.54 | 6.52 | 6.26 | 3.59 | 2.46 |
| 12 | 1.16 | 1.56 | 2.61 | 5.55 | 5.36 | 5.94 | 3.49 | 2.42 |
| 18 | 1.00 | 1.35 | 2.38 | 5.05 | 5.74 | 5.41 | 3.31 | 2.36 |
| 8 × 8 | 1.16 | 1.56 | 2.61 | 5.36 | 6.23 | 5.94 | 3.49 | 2.42 |
| 12 | 0.93 | 1.25 | 2.26 | 4.89 | 5.49 | 5.13 | 3.21 | 2.32 |
| 18 | 0.83 | 1.11 | 2.08 | 4.64 | 5.10 | 4.72 | 3.06 | 2.26 |
| 24 | 0.74 | 0.98 | 1.90 | 4.37 | 4.70 | 4.29 | 2.90 | 2.20 |
| 10 × 10 | 0.88 | 1.20 | 2.19 | 4.79 | 5.34 | 4.97 | 3.15 | 2.30 |
| 16 | 0.72 | 0.95 | 1.87 | 4.32 | 4.62 | 4.21 | 2.86 | 2.19 |
| 20 | 0.68 | 0.87 | 1.76 | 4.15 | 4.37 | 3.94 | 2.76 | 2.15 |
| 30 | 0.62 | 0.78 | 1.61 | 3.91 | 4.02 | 3.59 | 2.62 | 2.09 |
| 12 × 12 | 0.74 | 0.98 | 1.90 | 4.37 | 4.70 | 4.29 | 2.90 | 2.20 |
| 18 | 0.64 | 0.81 | 1.66 | 3.99 | 4.14 | 3.71 | 2.67 | 2.11 |
| 24 | 0.60 | 0.73 | 1.53 | 3.78 | 3.85 | 3.41 | 2.54 | 2.06 |
| 36 | 0.48 | 0.64 | 1.42 | 3.56 | 3.54 | 3.10 | 2.41 | 2.00 |
| 15 × 15 | 0.62 | 0.78 | 1.61 | 3.91 | 4.02 | 3.59 | 2.62 | 2.09 |
| 22 | 0.48 | 0.65 | 1.43 | 3.58 | 3.57 | 3.12 | 2.42 | 2.01 |
| 30 | 0.42 | 0.57 | 1.30 | 3.38 | 3.29 | 2.85 | 2.29 | 2.96 |
| 45 | 0.37 | 0.49 | 1.18 | 3.19 | 3.03 | 2.59 | 2.17 | 1.90 |
| 18 × 18 | 0.48 | 0.64 | 1.42 | 3.56 | 3.54 | 3.10 | 2.41 | 2.00 |
| 28 | 0.38 | 0.51 | 1.21 | 3.23 | 3.09. | 2.65 | 2.20 | 1.91 |
| 36 | 0.34 | 0.46 | 1.12 | 3.08 | 2.90 | 2.46 | 2.11 | 1.87 |
| 54 | 0.30 | 0.40 | 1.02 | 2.91 | 2.67 | 2.24 | 2.00 | 1.82 |
| 24 × 24 | 0.34 | 0.46 | 1.12 | 3.08 | 2.90 | 2.46 | 2.11 | 1.87 |
| 36 | 0.28 | 0.37 | 0.97 | 2.81 | 2.55 | 2.13 | 1.94 | 1.80 |
| 48 | 0.26 | 0.33 | 0.89 | 2.67 | 2.37 | 1.95 | 1.85 | 1.76 |
| 72 | 0.23 | 0.29 | 0.81 | 2.51 | 2.19 | 1.78 | 1.75 | 1.71 |
| 30 × 30 | 0.27 | 0.35 | 0.94 | 2.76 | 2.48 | 2.06 | 1.91 | 1.78 |
| 45 | 0.23 | 0.29 | 0.81 | 2.51 | 2.19 | 1.78 | 1.75 | 1.71 |
| 60 | 0.21 | 0.26 | 0.74 | 2.38 | 2.03 | 1.63 | 1.67 | 1.67 |
| 36 × 36 | 0.23 | 0.29 | 0.81 | 2.51 | 2.19 | 1.78 | 1.75 | 1.71 |
| 54 | 0.19 | 0.24 | 0.70 | 2.29 | 1.93 | 1.54 | 1.61 | 1.64 |
| 72 | 0.17 | 0.21 | 0.64 | 2.18 | 1.79 | 1.41 | 1.54 | 1.60 |
| 42 x 42 | 0.20 | 0.24 | 0.71 | 2.33 | 1.96 | 1.57 | 1.63 | 1.65 |
| 64 | 0.17 | 0.20 | 0.61 | 2.12 | 1.72 | 1.35 | 1.50 | 1.58 |
| 84 | 0.15 | 0.17 | 0.54 | 1.97 | 1.56 | 1.21 | 1.41 | 1.53 |
| 48 × 48 | 0.17 | 0.21 | 0.64 | 2.18 | 1.79 | 1.41 | 1.54 | 1.60 |
| 72 | 0.15 | 0.17 | 0.55 | 1.99 | 1.58 | 1.22 | 1.42 | 1.54 |

**Table A–37 Insertion Loss For Rectangular Ducts With 2 in. Of Fiberglass Lining**

DRAFT

| Dimensions in. | Attenuation - dB/ft 1/1 Octave Band Center Freq.—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 < 0 ≤ 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 A–38 Sound Attenuation In Straight Round Ducts**

| Diameter in. | Insertion Loss - dB/ft 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 6 | 0.38 | 0.59 | 0.93 | 1.53 | 2.17 | 2.31 | 2.04 | 1.26 |
| 8 | 0.32 | 0.54 | 0.89 | 1.50 | 2.19 | 2.17 | 1.83 | 1.18 |
| 10 | 0.27 | 0.50 | 0.85 | 1.48 | 2.20 | 2.04 | 1.64 | 1.12 |
| 12 | 0.23 | 0.46 | 0.81 | 1.45 | 2.18 | 1.91 | 1.48 | 1.05 |
| 14 | 0.19 | 0.42 | 0.77 | 1.43 | 2.14 | 1.79 | 1.34 | 1.00 |
| 16 | 0.16 | 0.38 | 0.73 | 1.40 | 2.08 | 1.67 | 1.21 | 0.95 |
| 18 | 0.13 | 0.35 | 0.69 | 1.37 | 2.01 | 1.56 | 1.10 | 0.90 |
| 20 | 0.1.1 | 0.31 | 0.65 | 1.34 | 1.92 | 1.45 | 1.00 | 0.87 |
| 22 | 0.08 | 0.28 | 0.61 | 1.31 | 1.82 | 1.34 | 0.92 | 0.83 |
| 24 | 0.07 | 0.25 | 0.57 | 1.28 | 1.71 | 1.24 | 0.85 | 0.80 |
| 26 | 0.05 | 0.22 | 0.53 | 1.24 | 1.59 | 1.14 | 0.79 | 0.77 |
| 28 | 0.03 | 0.19 | 0.49 | 1.20 | 1.46 | 1.04 | 0.74 | 0.74 |
| 30 | 0.02 | 0.16 | 0.45 | 1.16 | 1.33 | 0.95 | 0.69 | 0.71 |
| 32 | 0.01 | 0.14 | 0.42 | 1.12 | 1.20 | 0.87 | 0.66 | 0.69 |
| 34 | 0 | 0.11 | 0.38 | 1.07 | 1.07 | 0.79 | 0.63 | 0.66 |
| 36 | 0 | 0.08 | 0.35 | 1.02 | 0.93 | 0.71 | 0.60 | 0.64 |
| 38 | 0 | 0.06 | 0.31 | 0.96 | 0.80 | 0.64 | 0.58 | 0.61 |
| 40 | 0 | 0.03 | 0.28 | 0.91 | 0.68 | 0.57 | 0.55 | 0.58 |
| 42 | 0 | 0.01 | 0.25 | 0.84 | 0.56 | 0.50 | 0.53 | 0.55 |
| 44 | 0 | 0 | 0.23 | 0.78 | 0.45. | 0.44 | 0.51 | 0.52 |
| 46 | 0 | 0 | 0.20 | 0.71 | 0.35 | 0.39 | 0.48 | 0.48 |
| 48 | 0 | 0 | 0.18 | 0.63 | 0.26 | 0.34 | 0.45 | 0.44 |
| 50 | 0 | 0 | 0.15 | 0.55 | 0.19 | 0.29 | 0.41 | 0.40 |
| 52 | 0 | 0 | 0.14 | 0.46 | 0.13 | 0.25 | 0.37 | 0.34 |
| 54 | 0 | 0 | 0.12 | 0.37 | 0.09 | 0.22 | 0.31 | 0.29 |
| 56 | 0 | 0 | 0.10 | 0.28 | 0.08 | 0.18 | 0.25 | 0.22 |
| 58 | 0 | 0 | 0.09 | 0.17 | 0.08 | 0.16 | 0.18 | 0.15 |
| 60 | 0 | 0 | 0.08 | 0.06 | 0.10 | 0.14 | 0.09 | 0.07 |

**Table A–39 Insertion Loss For Acoustically Lined Round Ducts – 1 in. Lining**

DRAFT

| Diameter in. | Insertion Loss - dB/ft 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 6 | 0.56 | 0.80 | 1.37 | 2.25 | 2.17 | 2.31 | 2.04 | 1.26 |
| 8 | 0.51 | 0.75 | 1.33 | 2.23 | 2.19 | 2.17 | 1.83 | 1.18 |
| 10 | 0.46 | 0.71 | 1.29 | 2.20 | 2.20 | 2.04 | 1.64 | 1.12 |
| 12 | 0.42 | 0.67 | 1.25 | 2.18 | 2.18 | 1.91 | 1.48 | 1.05 |
| 14 | 0.38 | 0.63 | 1.21 | 2.15 | 2.14 | 1.79 | 1.34 | 1.00 |
| 16 | 0.35 | 0.59 | 1.17 | 2.12 | 2.08 | 1.67 | 1.21 | 0.95 |
| 18 | 0.32 | 0.56 | 1.13 | 2.10 | 2.01 | 1.56 | 1.10 | 0.90 |
| 20 | 0.29 | 0.52 | 1.09 | 2.07 | 1.92 | 1.45 | 1.00 | 0.87 |
| 22 | 0.27 | 0.49 | 1.05 | 2.03 | 1.82 | 1.34 | 0.92 | 0.83 |
| 24 | 0.25 | 0.46 | 1.01 | 2.00 | 1.71 | 1.24 | 0.85 | 0.80 |
| 26 | 0.24 | 0.43 | 0.97 | 1.96 | 1.59 | 1.14 | 0.79 | 0.77 |
| 28 | 0.22 | 0.40 | 0.93 | 1.93 | 1.46 | 1.04 | 0.74 | 0.74 |
| 30 | 0.21 | 0.37 | 0.90 | 1.88 | 1.33 | 0.95 | 0.69 | 0.71 |
| 32 | 0.20 | 0.34 | 0.86 | 1.84 | 1.20 | 0.87 | 0.66 | 0.69 |
| 34 | 0.19 | 0.32 | 0.82 | 1.79 | 1.07 | 0.79 | 0.63 | 0.66 |
| 36 | 0.18 | 0.29 | 0.79 | 1.74 | 0.93 | 0.71 | 0.60 | 0.64 |
| 38 | 0.17 | 0.27 | 0.76 | 1.69 | 0.80 | 0.64 | 0.58 | 0.61 |
| 40 | 0.16 | 0.24 | 0.73 | 1.63 | 0.68 | 0.57 | 0.55 | 0.58 |
| 42 | 0.15 | 0.22 | 0.70 | 1.57 | 0.56 | 0.50 | 0.53 | 0.55 |
| 44 | 0.13 | 0.20 | 0.67 | 1.50 | 0.45 | 0.44 | 0.51 | 0.52 |
| 46 | 0.12 | 0.17 | 0.64 | 1.43 | 0.35 | 0.39 | 0.48 | 0.48 |
| 48 | 0.11 | 0.15 | 0.62 | 1.36 | 0.26 | 0.34 | 0.45 | 0.44 |
| 50 | 0.09 | 0.12 | 0.60 | 1.28 | 0.19 | 0.29 | 0.41 | 0.40 |
| 52 | 0.07 | 0.10 | 0.58 | 1.19 | 0.13 | 0.25 | 0.37 | 0.34 |
| 54 | 0.05 | 0.08 | 0.56 | 1.10 | 0.09 | 0.22 | 0.31 | 0.29 |
| 56 | 0.02 | 0.05 | 0.55 | 1.00 | 0.08 | 0.18 | 0.25 | 0.22 |
| 58 | 0 | 0.03 | 0.53 | 0.90 | 0.08 | 0.16 | 0.18 | 0.15 |
| 60 | 0 | 0 | 0.53 | 0.79 | 0.10 | 0.14 | 0.09 | 0.07 |

**Table A–40 Insertion Loss For Acoustically Lined Round Ducts—2 in. Lining**

DRAFT

| Diameter in. | Insertion Loss - dB/ft 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| 6 | 0.64 | 1.00 | 1.58 | 2.25 | 2.17 | 2.31 | 2.04 | 1.26 |
| 8 | 0.59 | 0.95 | 1.54 | 2.21 | 2.19 | 2.17 | 1.83 | 1.18 |
| 10 | 0.54 | 0.91 | 1.50 | 2.18 | 2.20 | 2.04 | 1.64 | 1.12 |
| 12 | 0.50 | 0.87 | 1.46 | 2.16 | 2.18 | 1.91 | 1.48 | 1.05 |
| 14 | 0.46 | 0.83 | 1.42 | 2.13 | 2.14 | 1.79 | 1.34 | 1.00 |
| 16 | 0.43 | 0.79 | 1.38 | 2.10 | 2.08 | 1.67 | 1.21 | 0.95 |
| 18 | 0.40 | 0.75 | 1.34 | 2.07 | 2.01 | 1.56 | 1.10 | 0.90 |
| 20 | 0.38 | 0.72 | 1.30 | 2.04 | 1.92 | 1.45 | 1.00 | 0.87 |
| 22 | 0.35 | 0.69 | 1.26 | 2.01 | 1.82 | 1.34 | 0.92 | 0.83 |
| 24 | 0.33 | 0.66 | 1.22 | 1.98 | 1.71 | 1.24 | 0.85 | 0.80 |
| 26 | 0.32 | 0.63 | 1.18 | 1.94 | 1.59 | 1.14 | 0.79 | 0.77 |
| 28 | 0.30 | 0.60 | 1.14 | 1.90 | 1.46 | 1.04 | 0.74 | 0.74 |
| 30 | 0.29 | 0.57 | 1.11 | 1.86 | 1.33 | 0.95 | 0.69 | 0.71 |
| 32 | 0.28 | 0.54 | 1.07 | 1.82 | 1.20 | 0.87 | 0.66 | 0.69 |
| 34 | 0.27 | 0.52 | 1.04 | 1.77 | 1.07 | 0.79 | 0.63 | 0.66 |
| 36 | 0.26 | 0.49 | 1.00 | 1.72 | 0.93 | 0.71 | 0.60 | 0.64 |
| 38 | 0.25 | 0.47 | 0.97 | 1.67 | 0.80 | 0.64 | 0.58 | 0.61 |
| 40 | 0.24 | 0.44 | 0.94 | 1.61 | 0.68 | 0.57 | 0.55 | 0.58 |
| 42 | 0.23 | 0.42 | 0.91 | 1.55 | 0.56 | 0.50 | 0.53 | 0.55 |
| 44 | 0.22 | 0.39 | 0.88 | 1.48 | 0.45 | 0.44 | 0.51 | 0.52 |
| 46 | 0.20 | 0.37 | 0.85 | 1.41 | 0.35 | 0.39 | 0.48 | 0.48 |
| 48 | 0.19 | 0.35 | 0.83 | 1.33 | 0.26 | 0.34 | 0.45 | 0.44 |
| 50 | 0.17 | 0.32 | 0.81 | 1.25 | 0.19 | 0.29 | 0.41 | 0.40 |
| 52 | 0.15 | 0.30 | 0.79 | 1.17 | 0.13 | 0.25 | 0.37 | 0.34 |
| 54 | 0.13 | 0.27 | 0.77 | 1.07 | 0.09 | 0.22 | 0.31 | 0.29 |
| 56 | 0.10 | 0.25 | 0.76 | 0.98 | 0.08 | 0.18 | 0.25 | 0.22 |
| 58 | 0.07 | 0.22 | 0.75 | 0.87 | 0.08 | 0.16 | 0.18 | 0.15 |
| 60 | 0.04 | 0.20 | 0.74 | 0.76 | 0.10 | 0.14 | 0.09 | 0.07 |

**Table A–41 Insertion Loss For Acoustically Lined Round Ducts—3 in. Lining**

| Fw | 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 |

**Table A–42 Insertion Loss Of Unlined And Lined Square Elbows Without Turning Vanes**

$f_w$ = f x w where f is the VI octave center frequency (kHz) and w is the width of the elbow (inches).

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| Fw | Insertion Ioss—dB |
|---|---|
| fw < 1.9 | 0 |
| 1.9 < fw < 3.8 | 1 |
| 3.8 < fw < 7.5 | 2 |
| fw > 7.5 | 3 |

## Table A–43 Insertion Loss Of Round Elbows

$f_w$ = f x w where f is the VI octave center frequency (kHz) and w is the width of the elbow (inches).

| Fw | Insertion Ioss—dB | |
|---|---|---|
| | unlined elbows | line 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 |

## Table A–44 Insertion Loss Of Unlined And Lined Square Elbows With Turning Vanes

$f_w$ = f × w where f is the VI octave center frequency (kHz) and w is the width of the elbow (inches).

| Face Velocity fpm | 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| | Insertion Loss—dB | | | | | | | |
| + 1000 | 5 | 14 | 31 | 45 | 51 | 53 | 51 | 33 |
| + 2000 | 4 | 12 | 26 | 43 | 47 | 48 | 47 | 30 |
| - 1000 | 7 | 19 | 36 | 44 | 48 | 50 | 48 | 29 |
| - 2000 | 4 | 19 | 34 | 42 | 46 | 48 | 46 | 28 |
| | Regenerated Sound Power Levels—dB | | | | | | | |
| + 1000 | 58 | 52 | 42 | 36 | 37 | 35 | 30 | 30 |
| + 1500 | 64 | 57 | 58 | 49 | 45 | 49 | 48 | 46 |
| + 2000 | 72 | 65 | 64 | 63 | 55 | 56 | 57 | 56 |
| - 1000 | 55 | 52 | 54 | 55 | 55 | 64 | 64 | 55 |
| - 1500 | 61 | 58 | 59 | 61 | 61 | 66 | 75 | 66 |
| - 2000 | 69 | 66 | 65 | 75 | 71 | 73 | 79 | 76 |

## Table A–45 7 ft, Rectangular, Standard Pressure Drop Duct Silencers

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| Face Velocity fpm | 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| | Insertion Loss—dB | | | | | | | |
| +1000 | 2 | 8 | 18 | 33 | 41 | 47 | 24 | 15 |
| +2000 | 1 | 8 | 17 | 32 | 39 | 44 | 24 | 16 |
| +2500 | 1 | 7 | 17 | 30 | 37 | 42 | 24 | 15 |
| -1000 | 1 | 11 | 21 | 35 | 41 | 45 | 22 | 12 |
| -2000 | 1 | 11 | 21 | 36 | 40 | 43 | 21 | 11 |
| -3000 | 1 | 9 | 21 | 34 | 38 | 40 | 20 | 10 |
| | Regenerated Sound Power Levels—dB | | | | | | | |
| +1000 | 58 | 51 | 40 | 34 | 35 | 28 | 27 | 19 |
| +2000 | 67 | 61 | 58 | 53 | 51 | 54 | 52 | 45 |
| +2500 | 74 | 68 | 65 | 62 | 56 | 59 | 60 | 55 |
| -1000 | 58 | 49 | 46 | 44 | 49 | 45 | 34 | 25 |
| -2000 | 70 | 61 | 59 | 56 | 57 | 62 | 58 | 50 |
| -2500 | 74 | 67 | 64 | 62 | 61 | 65 | 65 | 57 |

**Table A–46 7 ft, Rectangular, Low Pressure Drop Duct Silencers**

| Face Velocity fpm | 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| | Insertion Loss—dB | | | | | | | |
| +1000 | 4 | 7 | 21 | 32 | 38 | 38 | 26 | 20 |
| +2000 | 4 | 7 | 20 | 32 | 38 | 38 | 27 | 21 |
| +3000 | 4 | 7 | 19 | 31 | 38 | 38 | 27 | 21 |
| -1000 | 5 | 9 | 23 | 33 | 39 | 37 | 25 | 19 |
| -2000 | 5 | 9 | 23 | 34 | 37 | 36 | 24 | 16 |
| -3000 | 6 | 10 | 24 | 34 | 37 | 36 | 24 | 16 |
| | Regenerated Sound Power Levels—dB | | | | | | | |
| +1000 | 62 | 43 | 38 | 39 | 36 | 25 | 22 | 28 |
| +2000 | 62 | 58 | 53 | 54 | 53 | 51 | 45 | 35 |
| +3000 | 72 | 66 | 62 | 64 | 63 | 64 | 61 | 54 |
| -1000 | 56 | 43 | 41 | 38 | 37 | 31 | 23 | 28 |
| -2000 | 66 | 56 | 54 | 54 | 57 | 54 | 49 | 41 |
| -3000 | 76 | 64 | 63 | 64 | 67 | 67 | 65 | 60 |

**Table A–47 Round, High Pressure Drop Duct Silencers**

DRAFT

| Face Velocity fpm | 1/1 Octave Band Center Freq.—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| | Insertion Loss—dB | | | | | | | |
| +1000 | 4 | 6 | 13 | 26 | 32 | 24 | 16 | 14 |
| +2000 | 4 | 5 | 13 | 25 | 32 | 24 | 16 | 13 |
| +3000 | 4 | 6 | 13 | 23 | 31 | 24 | 16 | 13 |
| -1000 | 5 | 7 | 16 | 28 | 35 | 25 | 24 | 30 |
| -2000 | 4 | 7 | 16 | 28 | 35 | 25 | 17 | 15 |
| -3000 | 6 | 7 | 16 | 29 | 35 | 25 | 16 | 15 |
| | Regenerated Sound Power Levels—dB | | | | | | | |
| +1000 | 60 | 44 | 39 | 34 | 29 | 25 | 24 | 30 |
| +2000 | 62 | 58 | 48 | 47 | 49 | 45 | 38 | 31 |
| +3000 | 69 | 63 | 55 | 55 | 57 | 58 | 54 | 47 |
| -1000 | 58 | 46 | 43 | 38 | 33 | 30 | 24 | 30 |
| -2000 | 65 | 52 | 50 | 49 | 48 | 44 | 36 | 33 |
| -3000 | 72 | 57 | 57 | 57 | 56 | 57 | 52 | 49 |

**Table A–48 Round, Low Pressure Drop Duct Silencers**

| Rectangular | C1 | C2 | C3 | C4 |
|---|---|---|---|---|
| High Pressure Drop | 0.6464 | 0.3971 | $2.637 \times 10\text{-}7$ | 2.012 |
| Low Pressure Drop | 0.6015 | 0.4627 | $9.802 \times 10\text{-}8$ | 2.011 |
| Circular | | | | |
| High Pressure Drop | $5.108 \times 10\text{-}^{3}$ | 1.999 | $4.007 \times 10^{-8}$ | 2.002 |
| Low Pressure Drop | $5.097 \times 10^{-3}$ | 2.000 | $1.104 \times 10^{-8}$ | 2.022 |

**Table A–49 Coefficients For Determining Static Pressure Drop Across Duct Silencers**

| Distance between silencer and closest edge of system component (fan, elbow, branch TO, etc. | C(up) Upstream from entering edge of silencer | C(down) Downstream from leaving edge of silencer |
|---|---|---|
| $D_{eq} \times 3$ or greater | 1.0 | 1.0 |
| $D_{eq} \times 2$ | 1.4 | 1.4 |
| $D_{eq} \times 1$ | 1.9 | 1.9 |
| $D_{eq} \times 0.5$ | 3.0 | 3.0 |
| Directly Connected | 4.0 | Not recommended |

**Table A–50 Coefficient For System Component Effect On Duct Silencers**

DRAFT

| Thickness (in.) | Surface Weight (lb/ft$^2$) | Transmission Loss—dB 1/1 Octave Band Center Freq.—Hz | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 |
| Gypsum Board | | | | | | | | |
| ⅜ | 1.6 | 6 | 11 | 17 | 22 | 28 | 32 | 24 |
| ½ | 2.1 | 9 | 14 | 20 | 24 | 30 | 31 | 27 |
| ⅝ | 2.7 | 10 | 15 | 22 | 26 | 31 | 28 | 30 |
| 1 | 4.6 | 13 | 18 | 26 | 30 | 30 | 29 | 37 |

| Dimensions (ft × ft × in.) | Surface Weight (lb/ft$^2$) | Transmission Loss—dB 1/1 Octave Band Center Freq.—Hz | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 |
| Acoustical Ceiling Tile - Exposed T-Bar Grid Suspended Lay-in-Ceilings | | | | | | | | |
| 2 × 4 × ⅝ | 0.6 – 0.7 | 4 | 9 | 9 | 14 | 19 | 24 | 26 |
| 2 × 4 × 1-½ | 0.9 | 4 | 9 | 10 | 11 | 15 | 20 | 25 |
| 2 × 4 × ⅝ | 0.95 – 1.1 | 5 | 11 | 13 | 15 | 21 | 24 | 28 |
| 2 × 2 × ⅝ | 1.2 – 1.3 | 5 | 10 | 11 | 15 | 19 | 22 | 24 |
| Acoustical Ceiling Tile - Concealed Spline Suspended Ceiling | | | | | | | | |
| 1 × 1 × ⅝ | 1.2 | 6 | 14 | 14 | 18 | 22 | 27 | 30 |

**Table A–51 Transmission Loss Values For Ceiling Materials**

| Ceiling Configuration | τ |
|---|---|
| Gypboard: Gypboard: No Ceiling Diffusers or Penetrations | 0.0001 |
| Gypboard: Few Ceiling Diffusers and Penetrations Well Sealed | 0.001 |
| Lay-in-Suspended Tile: No Integrated Lighting of Diffuser System | 0.001 |
| Lay-in-Suspended Tile: Integrated Lighting and Diffuser System | 0.03 |

**Table A–52 Correction Coefficient "τ" For Different Types Of Ceilings**

| Room Absorption Coefficient 1/1 Octave Band Center Frequency—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|
| Type of Room | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 |
| Dead<br>Acoustical ceiling, plush carpet, soft furnishings, people | 0.26 | 0.30 | 0.35 | 0.4 | 0.43 | 0.46 | 0.52 |
| Medium Dead<br>Acoustical ceiling, commercial carpet, people | 0.24 | 0.22 | 0.18 | 0.25 | 0.30 | 0.36 | 0.42 |
| Average<br>Acoustical ceiling or commercial carpet, people | 0.25 | 0.23 | 0.17 | 0.20 | 0.24 | 0.29 | 0.34 |
| Medium Live<br>Some acoustical material, people | 0.25 | 0.23 | 0.15 | 0.15 | 0.17 | 0.20 | 0.23 |
| Live<br>People only | 0.26 | 0.24 | 0.12 | 0.10 | 0.09 | 0.11 | 0.13 |

**Table A–53 Average Sound Absorption Coefficients, α, For Typical Receiving Rooms**

DRAFT

| Air Absorption Coefficient 1/1 Octave Band Center Frequency—Hz | | | | | | | |
|---|---|---|---|---|---|---|---|
| | 63 | 125 | 250 | 500 | 1000 | 2000 | 4000 |
| M (1/ft) | 0 | 0 | 0 | 0 | 0 | 0.0009 | 0.0029 |

**Table A–54 Air Absorption Coefficients**

| N | 10 Log N (dB) | 20 Log N (dB) | N | 10 Log N (dB) | 20 Log N (dB) |
|---|---|---|---|---|---|
| .1 | -10 | -20 | 100 | 20 | 40 |
| .12 | -9 | -18 | 125 | 21 | 42 |
| .16 | 8 | -16 | 160 | 22 | 44 |
| .2 | -7 | -14 | 200 | 23 | 46 |
| .25 | -6 | -12 | 250 | 24 | 48 |
| .32 | -5 | -10 | 320 | 25 | 50 |
| .4 | -4 | -8 | 400 | 26 | 52 |
| .5 | -3 | -6 | 500 | 27 | 54 |
| .63 | -2 | -4 | 630 | 28 | 56 |
| .8 | -1 | -2 | 800 | 29 | 58 |
| 1.0 | 0 | 0 | 1,000 | 30 | 60 |
| 1.25 | 1 | 2 | 1,250 | 31 | 62 |
| 1.6 | 2 | 4 | 1.600 | 32 | 64 |
| 2.0 | 3 | 6 | 2.000 | 33 | 66 |
| 2.5 | 4 | 8 | 2.500 | 34 | 68 |
| 3.2 | 5 | 10 | 3,200 | 35 | 70 |
| 4.0 | 6 | 2 | 4.000 | 36 | 72 |
| 5.0 | 7 | 14 | 5.000 | 37 | 74 |
| 6.3 | 8 | 16 | 6.300 | 38 | 76 |
| 8 | 9 | 18 | 8.000 | 39 | 78 |
| 10 | 10 | 20 | 10.000 | 40 | 80 |
| 12 | 11 | 22 | 12.500 | 41 | 82 |
| 16 | 12 | 24 | 16.000 | 42 | 84 |
| 20 | 13 | 26 | 20.000 | 43 | 86 |
| 25 | 14 | 28 | 25.000 | 44 | 88 |
| 32 | 15 | 30 | 32.000 | 45 | 90 |
| 40 | 16 | 32 | 40.000 | 46 | 92 |
| 50 | 17 | 34 | 50,000 | 47 | 94 |
| 63 | 18 | 36 | 63.000 | 48 | 96 |
| 80 | 19 | 38 | 80.000 | 49 | 98 |
| 100 | 20 | 40 | 100,000 | 50 | 100 |

**Table A–55 Decibel Equivalents Of Numbers (N)**

DRAFT

| No. | Logarithm | No | Logarithm | No | Logarithm | No | Logarithm | No | Logarithm |
|---|---|---|---|---|---|---|---|---|---|
| .01 | -2.00000 | .50 | -1.69897 | 1 | 0.00000 | 50 | 1.69897 | 100 | 2.00000 |
| .02 | -2.30103 | .51 | -1.70757 | 2 | 0.30103 | 51 | 1.70757 | 125 | 2.09691 |
| .03 | -2.47712 | .52 | -1.71600 | 3 | 0.47712 | 52 | 1.71600 | 150 | 2.17609 |
| .04 | -2.60206 | .53 | -1.72428 | 4 | 0.60206 | 53 | 1.72428 | 175 | 2.24304 |
| .05 | -2.69897 | .54 | -1.73239 | 5 | 0.69897 | 54 | 1.73239 | 200 | 2.30103 |
| .06 | -2.77815 | .55 | -1.74036 | 6 | 0.77815 | 55 | 1.74036 | 225 | 2.35218 |
| .07 | -2.84510 | .56 | -1.74819 | 7 | 0.84510 | 56 | 1.74819 | 250 | 2.39794 |
| .08 | -2.90309 | .57 | -1.75587 | 8 | 0.90309 | 57 | 1.75587 | 275 | 2.43933 |
| .09 | -2.95424 | .58 | -1.76343 | 9 | 0.95424 | 58 | 1.76343 | 300 | 2.47712 |
| .10 | -1.00000 | .59 | -1.77085 | 10 | 1.00000 | 59 | 1.77085 | 325 | 2.51188 |
| .11 | -1.04139 | .60 | -1.77815 | 11 | 1.04139 | 60 | 1.77815 | 350 | 2.54407 |
| .12 | -1.07918 | .61 | -1.78533 | 12 | 1.07918 | 61 | 1.78533 | 375 | 2.57403 |
| .13 | -1.11394 | .62 | -1.79239 | 13 | 1.11394 | 62 | 1.79239 | 400 | 2.60206 |
| .14 | -1.14613 | .63 | -1.79934 | 14 | 1.14613 | 63 | 1.79934 | 425 | 2.62839 |
| .15 | -1.17609 | .64 | -1.80618 | 15 | 1.17609 | 64 | 1.80618 | 450 | 2.65321 |
| .16 | -1.20412 | .65 | -1.81291 | 16 | 1.20412 | 65 | 1.81291 | 475 | 2.67669 |
| .17 | -1.23045 | .66 | -1.81954 | 17 | 1.23045 | 66 | 1.81954 | 500 | 2.69897 |
| .18 | -1.25527 | .67 | -1.82607 | 18 | 1.25527 | 67 | 1.82607 | 525 | 2.72016 |
| .19 | -1.27875 | .68 | -1.83251 | 19 | 1.27875 | 68 | 1.83251 | 550 | 2.74036 |
| .20 | -1.30103 | .69 | -1.83885 | 20 | 1.30103 | 69 | 1.83885 | 575 | 2.75967 |
| .21 | -1.32222 | .70 | -1.84510 | 21 | 1.32222 | 70 | 1.84510 | 600 | 2.77815 |
| .22 | -1.34242 | .71 | -1.85126 | 22 | 1.34242 | 71 | 1.85126 | 625 | 2.79588 |
| .23 | -1.36173 | .72 | -1.85733 | 23 | 1.36173 | 72 | 1.85733 | 650 | 2.81291 |
| .24 | -1.38021 | .73 | -1.86332 | 24 | 1.38021 | 73 | 1.86332 | 675 | 2.82930 |
| .25 | -1.39794 | .74 | -1.86923 | 25 | 1.39794 | 74 | 186923 | 700 | 2.84510 |
| .26 | -1.41497 | .75 | -1.87506 | 26 | 1.41497 | 75 | 1.87506 | 725 | 2.86034 |
| .27 | -1.43136 | .76 | -1.88081 | 27 | 1.43136 | 76 | 1.88081 | 750 | 2.87506 |
| .28 | -1.44716 | .77 | -1.88649 | 28 | 1.44716 | 77 | 1.88649 | 800 | 2.90309 |
| .29 | -1.46240 | .78 | -1.89209 | 29 | 1.46240 | 78 | 1.89209 | 850 | 2.92942 |
| .30 | -1.47712 | .79 | -1.89763 | 30 | 1.47712 | 79 | 1.89763 | 900 | 2.95424 |
| .31 | -1.49136 | 80 | -1.90309 | 31 | 1.49136 | 80 | 1.90309 | 950 | 2.97772 |
| .32 | -1.50515 | .81 | -1.90849 | 32 | 1.50515 | 81 | 1.90849 | 1000 | 3.00000 |
| .33 | -1.51851 | .82 | -1.91381 | 33 | 1.51851 | 82 | 1.91381 | 2000 | 3.30103 |
| .34 | -1.53148 | .83 | -1.91908 | 34 | 1.53148 | 83 | 1.91908 | 3000 | 3.47712 |
| .35 | -1.54407 | .84 | -1.92428 | 35 | 1.54407 | 84 | 1.92428 | 4000 | 3.60206 |
| .36 | -1.55630 | .85 | -1.92942 | 36 | 1.55630 | 85 | 1.92942 | 5000 | 3.69897 |
| .37 | -1.56820 | .86 | -1.93450 | 37 | 1.56820 | 86 | 1.93450 | 6000 | 3.77815 |
| .38 | -1.57978 | .87 | -1.93952 | 38 | 1.57978 | 87 | 1.93952 | 7000 | 3.84510 |
| .39 | -1.59106 | .88 | -1.94448 | 39 | 1.59106 | 88 | 1.94448 | 8000 | 3.90309 |
| .40 | -1.60206 | .89 | -1.94939 | 40 | 1.60206 | 89 | 1.94939 | 9000 | 3.95424 |
| .41 | -1.61278 | .90 | -1.95424 | 41 | 1.61278 | 90 | 1.95424 | 10,000 | 4.00000 |
| .42 | -1.62325 | .91 | -1.94904 | 42 | 1.62325 | 91 | 1.95904 | 20,000 | 4.30103 |
| .43 | -1.63347 | .92 | -1.96379 | 43 | 1.63347 | 92 | 1.96379 | 30,000 | 4.47712 |
| .44 | -1.64345 | .93 | -1.96848 | 44 | 1.64345 | 93 | 1.96848 | 40,000 | 4.60206 |
| .45 | -1.65321 | .94 | -1.97313 | 45 | 1.65321 | 94 | 1.97313 | 50,000 | 4.69897 |
| .46 | -1.66276 | .95 | -1.97772 | 46 | 1.66276 | 95 | 1.97772 | 60,000 | 4.77815 |
| .47 | -1.67210 | .96 | -1.98227 | 47 | 1.67210 | 96 | 1.98227 | 70,000 | 4.84510 |
| .48 | -1.68124 | .97 | -1.98677 | 48 | 1.68124 | 97 | 1.98677 | 80,000 | 4.90309 |
| .49 | -1.69020 | .98 | -1.99123 | 49 | 1.69020 | 98 | 1.99123 | 90,000 | 4.95424 |
| | | .99 | -1.99564 | | | 99 | 1.99564 | 100,000 | 5.00000 |

**Table A–56 Five Place Logarithms**

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**SHEET METAL AND AIR CONDITIONING CONTRACTORS'**
**NATIONAL ASSOCIATION, INC.**

# Technical Paper
# on
# Duct Leakage

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**Rev. 12/92**

## Foreword

This technical paper was prepared in response to increasing concerns over the inappropriate and impractical reference to specifications for low levels of duct leakage. Such specifications are apparently a result of misunderstanding or misinformation about realistically attainable and cost effective levels of duct leakage.

In addition to the article, a sample specification is included after the technical article.

For further information on HVAC duct leakage and testing, please refer to SMACNA's *HVAC Air Duct Leakage Test Manual*, 1985.


## Acknowledgement

SMACNA wishes to acknowledge and thank the author of this technical paper, Robert S. Deeds of METCO Spiral Pipe and Fittings of Salt Lake City, UT. Mr. Deeds has been an active member and chairman on the Duct Construction Standard Committee and served as chairman of the Technical Resources Committee and Seismic Restraint Task Force. He was also involved in the development of the Duct Leakage Test Manual.


Mr. Deeds, a registered professional engineer, is a graduate of the University of Utah with a Masters Degree in mechanical engineering. He has lectured on duct construction standards, duct leakage, and seismic restraint at the national and local levels. Mr. Deeds has also made presentations to joint SMACNA/ASHRAE educational seminars on technical subjects at the local level.

## Introduction

In 1985, SMACNA published the *HVAC Air Duct Leakage Test Manual* (referred to as the Test Manual) in conjunction with the *HVAC Duct Construction Standards—Metal and Flexible* manual. A key to the data presented in the Test Manual is the concept of defining duct leakage as a function of the pressure in the duct and the surface area of the duct. This allowed the introduction of a Leakage Class that could be used to calculate and predict far more accurate leakage rates for systems of various sizes and configurations. Analysis of the data available at that time indicated that this approach was far superior to the more common approach of specifying leakage as a percentage of fan or system air flow rate.

## Leakage Class Approach

The major advantage of using the Leakage Class approach is that it takes into account the physical size of the system in determining leakage. Looking only at a percentage of system air flow rates can be misleading and result in inaccurate leakage predictions and expectations. It can also lead to impractical specifications that add excessive expense to a project and that, in some cases, are simply unattainable.

The Leakage Class can be defined using the equation:

$$C_L = F / (P)^N \qquad \text{Equation 1}$$

where:

F = Leakage Factor
$C_L$ = Constant
P = Static pressure in the duct
N = Exponent relating to turbulence

Analysis of data compiled by SMACNA, ASHRAE, and TIMA resulted in the following conclusions:

a) N = 0.65 is a reliable average
b) Leakage factor F is best related to cubic feet per minute (cfm) per 100 square feet of duct surface
c) Leakage is not duct velocity dependent
d) Leakage in the same type of construction is not significantly different in positive and negative pressure modes

Using a static pressure (P) of 1 inch water gage (w.g.) and substituting into Equation 1:

$$C_L = F / (1)^{.65}$$

and:

$C_L$ = F at 1 in. w.g. pressure

Mathematically the constant $C_L$ is equal to the Leakage Factor when the pressure (P) in the system is equal to 1 in. w.g. This constant can thus be defined as the Leakage Class.

Leakage Class = Amount of Leakage (cfm/100 sq. ft.) when the pressure (P) = 1 in. w.g.

The significance of this definition is understood when used in conjunction with Equation 1. By rearranging Equation 1 and solving for the Leakage Factor (F) we find the following:

$$F = C_L \times (P)^{.65} \qquad \text{Equation 2}$$

where:

F = Leakage Factor (cfm/100 sq. ft. of duct surface area)
$C_L$ = Leakage Class
P = Static pressure in the duct (in. w.g.)

It is thus possible to predict the Leakage Factor (F) at any duct static pressure. The square footage of duct surface area can be easily calculated and is typically available as an output from the design or estimating process. The total leakage in cfm is then calculated by multiplying the calculated Leakage Factor times the duct surface area. This can be calculated for the entire system or independent sections.

## Duct Sealing

The importance of the Leakage Class can be appreciated further by reviewing the sealing provisions contained in the SMACNA *HVAC Duct Construction Standards —Metal and Flexible,* 1985 edition. These provisions are reproduced in Table 1 for reference.

**Table 1**

**DUCT SEALING REQUIREMENTS**

| Seal Class | Sealing Required | Static Pressure Construction Class |
|---|---|---|
| A | All transverse joints, longitudinal seams and duct wall penetrations | 4" w.g. and up |
| B | All transverse joints and longitudinal seams | 3" w.g. |
| C | Transverse joints | 2" w.g. |

In addition to the above any variable air volume system duct of 1" and 1/2" w.g. construction class that is upstream of the VAV boxes shall also meet Seal Class C.

An analysis of the available data allows for the estimation of a Leakage Class corresponding to a given Seal Class for both round and rectangular ductwork. Table 2 has been extracted from the Test Manual and predictions are based on test research averages using SMACNA's standards for construction and skilled, trained workers.

**Table 2**

| APPLICABLE LEAKAGE CLASSES | | | |
|---|---|---|---|
| DUCT CLASS | 1/2", 1", 2" W.G. | 3" W.G. | 4", 6", 10" W.G. |
| SEAL CLASS | C | B | A |
| SEALING APPLICABLE | TRANSVERSE JOINTS ONLY | TRANSVERSE JOINTS AND SEAMS | JOINTS, SEAMS AND ALL WALL PENETRATIONS |
| LEAKAGE CLASS | | | |
| RECTANGULAR METAL | 24 | 12 | 6 |
| ROUND METAL | 12 | 6 | 3 |

Referring to Table 2, ductwork that is sealed according to Seal Class C has its transverse joints sealed only. Rectangular ductwork that is sealed according to Seal Class C can be expected to exhibit leakage characteristics associated with Leakage Class 24. Round ductwork that is sealed according to Seal Class C can be expected to exhibit leakage characteristics associated with Leakage Class 12. Ductwork that is sealed according to Seal Class B has its transverse joints and seams sealed. Rectangular ductwork that is sealed according to Seal Class B can be expected to exhibit leakage characteristics associated with Leakage Class 12. Round ductwork that is sealed according to Seal Class B can be expected to exhibit leakage characteristics associated with Leakage Class 6. Ductwork that is sealed according to Seal Class A has all joints, seams and wall penetrations sealed. Rectangular ductwork that is sealed according to Seal Class A can be expected to exhibit leakage characteristics associated with Leakage Class 6. Round ductwork that is sealed according to Seal Class A can be expected to exhibit leakage characteristics associated with Leakage Class 3.

However, an assumption that imposing Seal Class A on 1 in. w.g. duct class, for example, will automatically result in the leakage rate/class associated with Seal Class A in Table 2 is not valid because the joints and seams allowed and used at the higher pressure level are not customary for the lower pressure classes. Restricting joint selection and specifying Seal Class A or B on 1 in. w.g. and 0.5 in. w.g. construction class or raising the construction pressure class beyond the structurally needed level are not as cost effective as allowing

freedom of choice in construction detail with appropriate control of leakage. Unsealed ductwork can be expected to exhibit average leakage characteristics associated with Leakage Class 48 for rectangular ductwork and Leakage Class 24 for round ductwork. It thus becomes possible to predict Leakage Factors based on duct construction and this provides a basis for evaluating duct conforming to SMACNA's duct construction standards.

## Duct Leakage and Pressure

A detailed graph of leakage classifications 3, 6, 12, 24, and 48 is printed in the Test Manual. A simplified copy of this graph is shown in Figure 1. Leakage factors and classes from Figure 1 are listed in Table 3 for convenient use. Class 48 represents average unsealed ductwork.

**Figure 1**

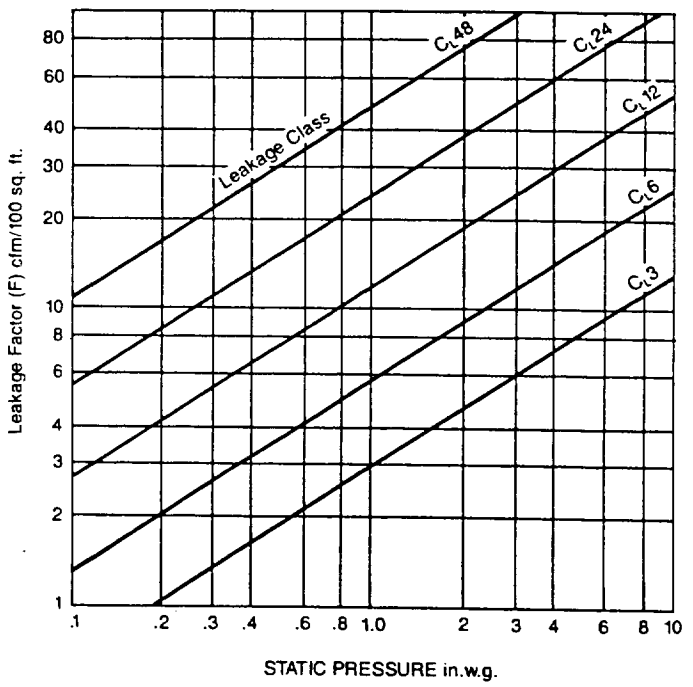

Table 3

## LEAKAGE FACTOR (F) IN CFM/100 S.F. DUCT

| PRESSURE W.G. $P^{.65}$ | PRESSURE W.G. P" | LEAKAGE CLASS ($C_L$) CLASS 3 | CLASS 6 | CLASS 12 | CLASS 24 | UNSEALED CLASS 48 |
|---|---|---|---|---|---|---|
| .143 | .05 | .4 | .9 | 1.7 | 3.4 | 6.7 |
| .224 | .10 | .7 | 1.3 | 2.7 | 5.4 | 10.7 |
| .351 | .20 | 1.1 | 2.1 | 4.2 | 8.4 | 16.8 |
| .457 | .30 | 1.4 | 2.7 | 5.5 | 11.0 | 21.9 |
| .551 | .40 | 1.7 | 3.3 | 6.6 | 13.2 | 26.4 |
| .637 | .50 | 1.9 | 3.8 | 7.6 | 15.3 | 30.6 |
| .717 | .60 | 2.2 | 4.3 | 8.6 | 17.2 | 34.4 |
| .793 | .70 | 2.4 | 4.8 | 9.5 | 19.0 | 38.1 |
| .865 | .80 | 2.6 | 5.2 | 10.4 | 20.8 | 41.5 |
| .934 | .90 | 2.8 | 5.6 | 11.2 | 22.4 | 44.8 |
| 1 | 1 | 3 | 6 | 12 | 24 | 48 |
| 1.30 | 1.5 | 3.9 | 7.8 | 15.6 | 31.2 | 62.4 |
| 1.57 | 2.0 | 4.7 | 9.4 | 18.8 | 37.7 | 75.4 |
| 1.81 | 2.5 | 5.4 | 10.9 | 21.7 | 43.4 | 86.8 |
| 2.04 | 3.0 | 6.1 | 12.2 | 24.5 | 49.0 | 98.0 |
| 2.26 | 3.5 | 6.7 | 13.6 | 27.1 | 54.2 | 108.5 |
| 2.46 | 4.0 | 7.4 | 14.8 | 29.5 | 59.0 | 118.1 |
| 2.66 | 4.5 | 8.0 | 16.0 | | | |
| 2.85 | 5.0 | 8.6 | 17.1 | | | |
| 3.03 | 5.5 | 9.1 | 18.2 | | | |
| 3.20 | 6.0 | 9.6 | 19.2 | | | |
| 3.54 | 7.0 | 10.6 | 21.2 | | | |
| 3.86 | 8.0 | 11.6 | 23.2 | | | |
| 4.17 | 9.0 | 12.5 | 25.0 | | | |
| 4.47 | 10.0 | 13.4 | 26.8 | | | |
| 4.75 | 11.0 | 14.3 | 28.5 | | | |
| 5.03 | 12.0 | 15.1 | 30.2 | | | |

$$C_L = \frac{F}{P^{.65}}$$

When P = 1

$$C_L = F$$

$$F = C_L(P)^{.65}$$

Figure 1 (or Table 3) is useful in determining the Leakage Factor when the Leakage Class and the pressure are known. As an example, rectangular ductwork that is sealed according to Seal Class B can be expected to exhibit leakage characteristics associated with Leakage Class 12 (see Table 2). At a duct static pressure of 1 in. w.g., the Leakage Factor can be read from Figure 1 as 12 cfm per 100 sq. ft. This is consistent with the definition of the Leakage Class above. If that same ductwork (Leakage Class 12) sees a pressure of 3 in. w.g., the Leakage Factor becomes 24.5 cfm per 100 sq. ft. The total leakage in cfm is then calculated by multiplying the Leakage Factor times the duct surface area.

Leakage as a percent of flow can be related to Leakage Class and Pressure Class. Table 4 (as extracted from the Test Manual) depicts this relationship and is useful in understanding system performance.

## Table 4

### LEAKAGE AS % OF FLOW IN SYSTEM

| LEAKAGE CLASS | FAN CFM PRORATED* PER S.F. | STATIC PRESSURE (IN. W.G.) 1/2 | 1 | 2 | 3 | 4 | 6 |
|---|---|---|---|---|---|---|---|
| 48 | 2 | 15 | 24 | 38 | | | |
| | 2-½ | 12 | 19 | 30 | | | |
| | 3 | 10 | 16 | 25 | | | |
| | 4 | 7.7 | 12 | 19 | | | |
| | 5 | 6.1 | 9.6 | 15 | | | |
| 24 | 2 | 7.7 | 12 | 19 | | | |
| | 2-½ | 6.1 | 9.6 | 15 | | | |
| | 3 | 5.1 | 8.0 | 13 | | | |
| | 4 | 3.8 | 6.0 | 9.4 | | | |
| | 5 | 3.1 | 4.8 | 7.5 | | | |
| 12 | 2 | 3.8 | 6 | 9.4 | 12 | | |
| | 2-½ | 3.1 | 4.8 | 7.5 | 9.8 | | |
| | 3 | 2.6 | 4.0 | 6.3 | 8.2 | | |
| | 4 | 1.9 | 3.0 | 4.7 | 6.1 | | |
| | 5 | 1.5 | 2.4 | 3.8 | 4.9 | | |
| 6 | 2 | 1.9 | 3 | 4.7 | 6.1 | 7.4 | 9.6 |
| | 2-½ | 1.5 | 2.4 | 3.8 | 4.9 | 5.9 | 7.7 |
| | 3 | 1.3 | 2.0 | 3.1 | 4.1 | 4.9 | 6.4 |
| | 4 | 1.0 | 1.5 | 2.4 | 3.1 | 3.7 | 4.8 |
| | 5 | .8 | 1.2 | 1.9 | 2.4 | 3.0 | 3.8 |
| 3 | 2 | 1.0 | 1.5 | 2.4 | 3.1 | 3.7 | 4.8 |
| | 2-½ | .8 | 1.2 | 1.9 | 2.4 | 3.0 | 3.8 |
| | 3 | .6 | 1.0 | 1.6 | 2.0 | 2.5 | 3.2 |
| | 4 | .5 | .8 | 1.3 | 1.6 | 2.0 | 2.6 |
| | 5 | .4 | .6 | .9 | 1.2 | 1.5 | 1.9 |

*TYPICALLY $\frac{\text{FAN CFM}}{\text{DUCT SURFACE AREA}}$ WILL BE 2 to 5 CFM/SQUARE FOOT.

% OF FLOW = LEAKAGE FACTOR (IN CFM/100 AT THE PRESSURE)

DIVIDED BY $\frac{\text{FAN CFM}}{\text{S.F. SURFACE}} = \frac{\text{CFM}_A}{100\ \text{S.F.}} \times \frac{\text{S.F.}}{\text{CFM}_F}$

For the average job, the ratio of fan CFM to duct surface area is in the range of 2 to 5 CFM per square foot of duct surface area. This value is indicated in column 2 of Table 4. Systems with relatively small amounts of ductwork and large air flow requirements are nearer to 5 CFM per square foot of duct surface area. Systems with relatively large amounts of ductwork and small air flow requirements are nearer to 2 CFM per square foot of duct surface area. As an example of reading Table 4, consider an average size rectangular duct system that has 3 cfm per square foot, is designed for 2 in. w.g. static pressure, and is sealed according to Seal Class C. Table 1 indicates that the Leakage Class for this system is Class 24. Referring to Table 4, the leakage for this system will be 13 percent of the system air flow.

Appraisal of actual leakage by Table 4 can best be done by averaging upstream and downstream operating pressures since pressure declines in the direction of flow. For example, 2 in. w.g. static pressure at fan discharge will reduce to about 0.1 in. w.g. at a diffuser. The rate for 1 in. w.g. in Table 4 would be a better estimate than that for 2 in. w.g. in this case. Furthermore, it is not appropriate to evaluate leakage as percent of flow in separate sections of the same duct system. Allocations must be made from an aggregate allowable for the total system. Further review of Table 4 shows that 1 percent leakage is only attainable in relatively small systems when all joints, seams and wall penetrations are carefully sealed and the static pressure is less than or equal to 2 in. w.g.

## Conclusion

The assumption that 1 percent leakage is easily attainable at higher pressures and under less stringent sealing requirements is misleading to both designers and contractors. Care must be taken in specifying and interpreting allowable leakage requirements to avoid unnecessary expense and confusion.

In addition, there is no need to conduct leakage tests at pressures above duct construction class. Twenty-five years ago, "high pressure" leakage tests were done at 125% of rated pressure with 10% of the cubic volume of the test specimen as a maximum allowable rate. With today's knowledge leakage assessment can be done on a more accurate and economical basis. However, strict reliance on Equation 2 could lead to an assumption that leakage tests could always be done at 1 in. w.g. with the derivative conclusion that leakage at any other pressure level would precisely fit the curve for Equation 2. This would not account for the exponent and constant being averages. The practical approach to testing is to use a pressure equal to the average operating pressure or, conservatively, the Duct Construction Pressure Class level.

SMACNA's *HVAC Air Duct Leakage Test Manual* contains additional information referring to duct leakage and testing. Use of the information and guidance in the Test Manual should facilitate design, improve system performance, and reduce the difficulty of testing and balancing newly installed systems. Use of the Leakage Class can also help to alleviate the confusion and misunderstandings associated with leakage requirements.

# APPENDIX D

## SAMPLE PROJECT SPECIFICATION

NOTICE TO DESIGNERS:

WHEN TESTS ARE DEEMED NECESSARY, A TEST OF A REPRESENTATIVE SAMPLE OF THE DUCT IS RECOMMENDED. IF SAMPLE IS DEFECTIVE, THE CONTRACTOR SHOULD REPAIR OR MODIFY THE CONSTRUCTION. IF RESULTS OF SAMPLE TEST ARE GOOD, CONTRACTOR CAN BE PERMITTED TO PROCEED WITHOUT FURTHER TESTING. VISUAL INSPECTION AND EXAMINATION OF OPERATING CONDITIONS SHOULD SUFFICE TO JUSTIFY FAITH IN METHODS USED.

1.1 Contractor shall, at the beginning of the work construct, erect and leak test a representative sample of the duct construction to be used at the ____ pressure class. The sample specimen shall include at least five transverse joints, typical seams, an access door and at least two typical branch connections plus an elbow.

1.2 The leakage amount shall not exceed the allotted amount for the pressure class or the allotted amount for that portion of the system, whichever is applicable.

| DUCT CONSTRUCTION CLASS | LEAKAGE CLASS |
|---|---|
| 10″ w.g. | 3 |
| 6″ w.g. | 6 |
| 4″ w.g. | 6 |
| 3″ w.g. | 12 |

NOTE: See Section 4 of the SMACNA leakage test manual for normal classification.

1.3 Leakage test procedures shall follow the outlines and classifications in the SMACNA HVAC Duct Leakage Test manual.

1.4 If specimen fails to meet allotted leakage level, the contractor shall modify to bring it into compliance and shall retest it until acceptable leakage is demonstrated.

1.5 Tests and necessary repair shall be completed prior to concealment of ducts.

# PROJECT COORDINATION:

## *Everyone Can Win When It's Done Right*

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**January 1998**

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# 1. Introduction

Construction projects have become larger, more complex, and more challenging over the past several years.

- Projects include more building systems than they used to. We've come a long way from steam radiators and hanging incandescent light bulbs.
- Building systems are increasingly more sophisticated. Internal communications used to mean installing telephone lines. Now owners want exotic cables and fiber optics – and their requirements grow more complex every month.
- Projects are often complex facilities that accommodate multiple uses, each with its own sophisticated requirements.
- Budget pressures mean that project owners question everything.
- The pressures place space at a premium and mean that more systems and equipment must be squeezed into less room.
- Telescoping design and construction schedules make it hard to do calm, thorough, deliberate project planning.

The project team now includes many technically sophisticated trades. Their work must share the same limited space without interference. In many cases, one trades' systems must interconnect with those of other trades.

In a typical situation, a number of systems will share a plenum or mechanical room:

- sheet metal
- mechanical
- electrical
- plumbing
- fire sprinklers
- telecommunications
- data communications
- and more.

The systems will be provided and installed by several subcontractors.

This makes coordinating the efforts of the trades on the project team a critical issue. Final designs must recognize everyone's space requirements. Work planning must establish the sequence in which the trades will perform and install their work. Plans should establish a precedence of trades to be followed in case of conflicts.

Technology has provided improved tools to help the project team with the coordination process. Project design documents once consisted of hand-drafted drawings. The first two-dimensional CAD systems automated this process and made it easier to manipulate overlays and share information. Three-dimensional CAD systems made it possible to go beyond plan views and see designs in three dimensions. Current three-dimensional systems let the users:

- develop models of the design that include three dimensional components from libraries
- link components into functional systems and evaluate their performance
- evaluate multiple systems in a tight space and check for interferences.

These sophisticated tools represent a great opportunity, yet there are still problems with project coordination.

- Shortened production schedules and reduced consultant fees mean that original design drawings are: often schematic in nature, filled with errors, not well coordinated among systems.
- The ultimate responsibility for project coordination is seldom clearly defined in the original construction contract or among the subs.
- General contractors are often unwilling to exercise responsibility for coordination, reluctant to assign the responsibility to anyone else, and unready to pay anyone else to perform coordination tasks.
- Some smaller subs do not have automated tools and cannot participate in a technically sophisticated coordination process.
- Some larger subs find themselves taking on coordination responsibilities by default, largely in order to protect their own interests on the construction site.

Members of the project team are somehow getting their work done, but many feel they are having to do extra work for which they are not being fairly compensated.

Beyond that, they are concerned that the lack of formal agreements on coordination responsibility might unfairly expose them to liability if something goes wrong.

## 2. The Big Problem

Most commercial construction contracts include little, if any, language about project coordination:

- They do not establish the coordination process.
- They do not set requirements.
- They do not assign responsibility (or extra compensation where necessary).

The provisions that should appear in a contract are discussed in more detail below.

Where should these provisions appear?

Coordination provisions may be included in the prime contract between the project owner and the general contractor. This inclusion is rare, though, unless the owner has some very specific requirements or unless the project team includes a construction manager or other party to oversee contracts with multiple prime contractors.

Most standard construction contracts assign the coordination responsibility to the general contractor in a non-specific way and leave it at that. They seldom include requirements about how coordination is to be performed.

The general contractor must make project coordination plans, procedures, and requirements clear to all subcontractors. Subs must know what they are expected to do and how they are expected to perform their work.

For this reason, the contracts between the general contractor and the subcontractors must include specific coordination provisions. Whatever the

contract between the general and the owner may include, the contracts between the general and his subs are the documents that establish requirements for the subs.

There are a number of ways to conduct project coordination – two of them will be discussed below. Some ways work better than others. Some apply best in specific situations.

The most important thing, though, is not which approach to coordination is followed in a specific project. More important is that:

- the methods, requirements, and responsibilities for project coordination must be established before contracts are signed
- this information must be clearly communicated to all parties who are bidding on the work.

The most predictable, successful team efforts occur when all the team members are in agreement and share a common understanding of what they will do and how they will do it – and when they reach this understanding before the work begins.

## 3. Two Coordination Methods

Two distinct approaches to the coordination process exist and can be potentially successful. This document will refer to these approaches as:

- Single Coordinator
- Coordination Team

### 3.1. Single Coordinator

Under this approach, one member of the project team is assigned overall responsibility for coordination. This party's responsibilities include:

- receiving design documents from all the parties working in a shared area (such as the ceiling plenum or a mechanical room)
- preparing a coordination model/ document that locates everyone's work in the shared space and assures that there are no conflicts
- preparing a work plan that includes a schedule and sequence for the trades' work in the shared space
- overseeing the work as it is performed, verifying that all the parties work to the coordination plan and that the work stays coordinated
- mediating disputes among the coordinating parties
- coordinating changes that occur during construction and verifying that changes work for all parties

The Single Coordinator method requires significant planning and coordination effort before actual construction begins.

It requires the participation of all parties who have work in the shared space, even though one party has overall responsibility.

The general contractor may be the Single Coordinator.

A typical project construction contract assigns overall project responsibility, including the responsibility for coordination, to the general contractor.

At his option, the general can do the work himself or he can sub-contract it. However, the general will often instruct

the trades working in shared space to coordinate among themselves.

More frequently, the Single Coordinator is one of the subs who are working in the shared space.

The Single Coordinator must be a strong and capable team leader providing leadership during the planning process followed by a strong presence on the job site.

In today's environment, the Single Coordinator must also have sophisticated design tools to receive design information from all the trades and enter it into a 3D model for coordination and interference checking.

Larger sheet metal contractors assume the role of the Single Coordinator. They have the necessary management and leadership skills. They also have the necessary high-tech tools. In addition, they feel that assuming the coordination responsibility themselves protects them from having another trade get into the shared space early and making it hard for them to do their work.

Much of the sheet metal contractor's work must be custom fabricated. Once it is made, none of it can be effectively used anywhere beside the project for which it was fabricated.

The fact that some of this work is custom-made means that before he can install anything on the job site the sheet metal contractor must design his layout, prepare shop drawings, submit them for approval, and then fabricate the duct-work from the approved drawings. This process takes a great deal of time.

The other contractors' work includes installing standard items and fixtures, which may be complex in their own right, but are linked together with commodity materials like plumbing pipe or conduit.

The other contractors' design submittals usually include catalog cuts of the fixtures. The other contractors can prepare these submittals quickly and the turnaround is fast. These contractors can buy most of their fixtures from local suppliers without much delay.

These contractors can be ready to install their work much faster than can the sheet metal contractor.

The sheet metal contractor's concern is that he will go through the design, submittal, fabrication process and ship his work to the project site only to find that other contractors working in the same space have already installed their work — and installed it in a way that means his work will not fit.

The sheet metal contractor wants to be responsible for coordinating all the trades' work because it gives him overall control of where systems go in the plenum and the sequence in which they are installed. This control assures him that everything will fit and that it will be installed in the correct sequence. Most importantly, it assures him that his own work, which takes longer to fabricate and is harder to modify on-site than anyone else's, will fit. Such coordination is advantageous to the entire construction team, especially the owner.

Much as some may want this responsibility, there are often problems when it is assigned to them:

- Assignment of coordination responsibility is often made after agreements are signed. The Single Coordinator may not be compensated

for performing the additional work. Other trades may not go along with the assignment of coordination responsibility and they may not cooperate with the process.

- Some parties, particularly smaller trades who do not have automated design tools, may not participate in the coordination process at all – figuring that they will just do their work on their own and work out problems on the job site.

The Single Coordinator method does have strong points:

- There is a single point of contact and responsibility for coordination issues. Everyone knows it and everyone knows where to take questions, complaints, problems, and suggestions.
- The coordination process has a leader.
- One party has ongoing coordination responsibility on the job site – even after the coordination documents are prepared.
- Coordination work that is done by a single party in a continuous effort will probably be more consistent.

**Recommendations**

- Assign coordination responsibility before proposals are solicited. This way, all proposals will include the same tasks. Proposals will include the cost of performing the work.
- Specifically assign coordination responsibility – include this assignment in the contract between the general contractor and the Single Coordinator and in contracts between the general and all subs.
- Require that all parties working in the shared space participate in the coordination process.
- Specify the sequence of work in the original contracts

### 3.2. Coordination Team

Under this approach, coordination responsibility is shared among all the parties working in the shared space.

In a typical Coordination Team situation:

- There is a set of CAD files for the design drawings.
- There is an assigned order in which the coordination work is to take place.
- The design drawings go to the first trade in the sequence, who enters his own work into the drawing files and then passes a copy of his work on to the next party in the sequence.
- The next party adds his own work and then passes the growing design files to the next party.
- If one party discovers a conflict or problem with work done by a previous party, he contacts that party and they reach an accommodation.
- When the last party has entered his work, everyone can assume that all parties are in agreement and that each party can perform or install its own work as indicated on the coordination model.

This approach is popular among most medium-sized contractors. They liked this approach because:

- They do not have to assume the responsibility of performing overall coordination work. Instead, they can focus their efforts on doing their own work.
- The coordination work is shared among all the trades and no single trade is required to do an unfair portion of the work.
- Each trade is ultimately responsible for coordinating its own work.
- If one trade discovers coordination problems with work of the other trades, they must work out accommodations with the parties involved (without taking up the time of uninvolved parties).

The Coordination Team approach works best when:

- The precedence and priority order of the trades is clear before the work begins – either through a general understanding of a "customary" precedence or through specific contract provisions.
- The trades understand the process and commit to making it work.
- Each trade has the tools required to participate in the process – CAD system, 3-D modeling software, etc.
- The trades know and are used to working with one another. This approach requires mutual trust and a lot of cooperation.
- One party is responsible for overseeing the whole coordination process:
    - Making certain all parties participate fully
    - Assuring that the work flows through the sequence of trades in a timely way
    - Mediating disputes when a third party is necessary.

**Recommendations**

- When using the Coordination Team method, establish it in the proposal documents and spell out just how it will work. Announce this to all the subs preparing bids. If all the subs prepare their bids from the same assumptions, the bidding will be fairer and the bids will be easier to compare.
- Assign a coordination overseer and establish his specific responsibilities. Make the assignment early in the proposal phase.
- Require all parties doing work in the shared space to participate in the coordination process.
- Establish a sequence for the trades to follow in performing their coordination work. Establish a schedule for the coordination work.

## 4. Summary

Both the Single Coordinator and the Coordination Team approaches to project coordination have proved to be successful in the industry. [i]Still, both approaches can be implemented more successfully, as is described in the recommendations above.

Other coordination methods may prove successful for specific participants in particular situations.

The key recommendations of the report are these:

- Establish detailed requirements, procedures, and responsibility for project coordination in the project contracts
- Compensate the parties for their coordination efforts.

When these recommendations are implemented:

- important decisions are made at the beginning of the project.
- all the trades know how the project work will be coordinated and performed before they make their proposals.
- trades will know their coordination responsibilities and will include appropriate fee in their proposals (knowing that competitors will base their own proposals on the same assumptions).
- since proposals will be based on the same assumptions, they can be realistically compared with each other.
- project team members know how the project will be organized and can be ready to start their work as soon as Notice to Proceed is given.
- delays and false starts are minimized.

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[1]Preparations for writing this document included conducting interviews with several contractors and discussing their experiences with the coordination process. The contracts for the projects discussed were either competitively bid or negotiated with a "traditional" project construction team consisting of contractors and subcontractors. No design/build projects were discussed.