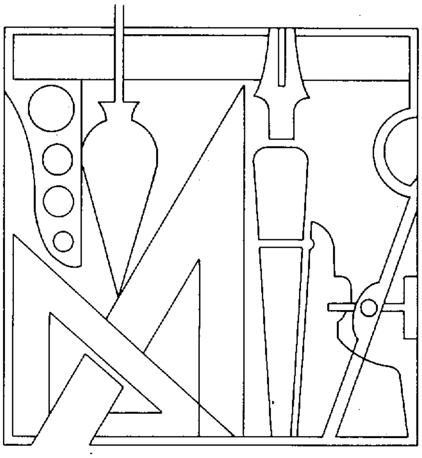
HVAC SYSTEMS DUCT DESIGN

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

HVAC SYSTEMS DUCT DESIGN



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U.S. & Metric Units

SMACNA

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Contractors' National Association, Inc.
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HVAC SYSTEMS—DUCT DESIGN

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FOREWORD

The Sheet Metal and Air Conditioning Contractors' National Association, Inc. (SMACNA), in keeping with its policy of disseminating information and providing standards of design and construction, offers this comprehensive and fundamental "HVAC Systems-Duct Design" manual as part of the continuing effort to upgrade the heating, ventilating and air conditioning (HVAC) industry. This manual presents the basic methods and procedures needed to design HVAC air distribution systems. It does not deal with the determination of air conditioning loads and room air quantities.

This manual is part one of a three set "HVAC Systems" Library. The second is the SMACNA "HVAC Systems—Applications" manual which 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 state-of-the-art publication on air and hydronic system testing and balancing.

The HVAC duct system designer is faced with many considerations once load calculations are completed and the type of distribution system to be used has been determined. This manual provides not only the basic engineering guides for the sizing of HVAC ductwork systems, but guides in the areas of:

a. Materials
b. Methods of Construction
c. Economics of Duct Systems
d. Duct System Layout
e. Calculation of System 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 sizes between the spaces allocated and the duct system pressure losses (which directly affect the fan power and thus the operating costs). Materials, equipment, and construction methods must be chosen with respect to system first costs and life cycle costing. This manual has been structured to offer options in design, materials and construction methods, so as to allow the designer to cope with and solve increasingly complex design problems using either U.S. units or metric units.

The SMACNA "HVAC Systems—Duct Design" manual was written to be totally compatible with chapter 32 of the ASHRAE 1989 "Fundamentals Handbook", although some new fitting loss coefficients found in this SMACNA manual may be from more recent research projects. The basic fluid flow equations (Bernoulli, Darcy, Colebrook, Altshul, etc.) are not included, but may be found in the ASHRAE Handbook. Practical applications of these equations are available through use of included tables and charts. Some of the text in this manual has been taken with permission from various ASHRAE publications. Some was used as published, some edited, some revised, and some expanded with the addition of newer data.

Although most HVAC systems are constructed to pressure classifications between minus 3 in. w.g. to 10 in. w.g., (-750 Pa to 2500 Pa), the design methods, tables, charts, and equations may be used to design other types of duct systems operating at much higher pressures and temperatures. Air density correction factors for both higher altitudes and temperatures are included.

SMACNA recognizes that in the future, this manual must be expanded and updated. As need arises, manuals on related subjects may be developed. A continuing effort will be made to provide the industry with a compilation of the latest construction methods and engineering data from recognized sources, and from SMACNA research, supplemented by the services of local SMACNA Chapters and SMACNA Contractors.

W. David Bevirt, P.E.
Director of Technical Research

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11 NOISE CONTROL

REFERENCES

Data from some publications from the following organizations have been used in developing this manual and may be used by the reader to further expand the methods or procedures found herein. Numbers in parentheses at the end of figures or table titles refer to the numbers preceding the reference.

A. Associations and Corporations

1. Air Movement and Control Association, Inc. (AMCA) — Fan Application Manuals, Standards
2. American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE) — Handbooks, Standards
3. American Society for Testing and Materials (ASTM) — Annual Book of ASTM Standards
4. Carrier Corporation — System Design Manuals, Publications
5. National Environmental Balancing Bureau (NEBB) — Manuals, Standards, Study Courses
6. National Fire Protection Association (NFPA) — Standards
7. Sheet Metal and Air Conditioning Contractors' National Association (SMACNA) — Manuals, Standards
8. Trane Co. — Publications
9. United Sheet Metal, United McGill Corporation — Publications

B. Publications

10. "Fan Engineering" — Buffalo Forge Company
11. "Procedural Standards for Measuring Sound and Vibration" — National Environmental Balancing Bureau (NEBB)
12. "Sound and Vibration in Environmental Systems" — National Environmental Balancing Bureau (NEBB)
13. "Study Course for Measuring Sound and Vibration" — National Environmental Balancing Bureau (NEBB)
14. "Handbook of Noise Control" edited by Cyril M. Harris. McGraw-Hill Book Company
15. "Handbook of Hydraulic Resistance" by I.E. Idelchik. Hemisphere Publishing Corp.

CHAPTER 1
INTRODUCTION

A PURPOSE

The purpose of the heating, ventilating and air conditioning (HVAC) duct system is to provide building occupants with:

1. thermal comfort,
2. humidity control,
3. ventilation,
4. air filtration.

However, a poorly designed or constructed HVAC duct system may result in systems that are costly to operate, that cause discomfort, that are noisy, and that permit contamination to occur to the conditioned spaces.

This manual, when used with other SMACNA publications, will provide the necessary information and data to properly design and install HVAC systems. They economically will provide clean, conditioned air unobtrusively to building occupants.

B GENERAL REQUIREMENTS

The HVAC duct system is a structural assembly whose primary function is to convey air between specific points. In fulfilling this function, the duct assembly must perform satisfactorily with certain fundamental performance characteristics. Elements of the assembly include an envelope of sheet metal (or other materials), reinforcements, seams, and joints; and theoretical and/or practical performance limits must be established for:

1. dimensional stability—deformation and deflection.
2. containment of the air being conveyed.
3. vibration.
4. noise generation, transmission and/or attenuation.
5. exposure to damage, weather, temperature extremes, flexure cycles, chemical corrosion, or other in-service conditions.
6. support.
7. emergency conditions such as fire and seismic occurrence.
8. heat gain or loss to the airstream.
9. adherence to duct walls of dirt or contaminants.

In establishing limitations for these factors, due consideration must be given to effects of the pressure differential across the duct wall, airflow friction losses, dynamic losses, air velocities, leakage, as well as the inherent strength characteristics of the duct components. Design and construction criteria, which will permit an economical attainment of the predicted and desired performance, must be determined.

C HVAC SYSTEMS LIBRARY

In addition to this "HVAC Systems—Duct Design" manual, there are many other SMACNA publications that directly or indirectly relate to the design and installation of HVAC systems. A listing with a brief description follows. They may be ordered from SMACNA using the order form found in the back of this manual.

1. HVAC Air Duct Leakage Test Manual

A companion to HVAC Duct Construction Standards, this new manual contains duct construction leakage classifications, expected leakage rates for sealed and unsealed ductwork, duct leakage test procedures, recommendations on use of leakage testing, types of test apparatus and test setup and sample leakage analysis. 1st Edition—1985.

2. HVAC Duct Construction Standards—Metal and Flexible

Primarily for commercial and institutional projects, but usable for residential and certain industrial work, this set of construction standards is a collection of material from earlier editions of SMACNA's low pressure, high pressure, flexible duct and duct liner standards.

It comprehensively prescribes construction detail alternatives for uncoated steel, galvanized steel, aluminum and stainless steel ductwork consisting of straight sections, transitions, elbows and united and divided flow fittings plus accessory items such as access doors, volume dampers, belt guards, hangers, casing, louvers and vibration isolation. For -3″ to + 10″ w.g. pressures (−750 to 2500 Pascals). 1st Edition—1985.

3. HVAC Systems—Applications

This manual, new to the "HVAC Systems Library" contains information and data needed by the designer and installer of more specialized HVAC systems used in commercial and institutional buildings. 1st Edition—1986.

4. HVAC Systems—Testing, Adjusting and Balancing

This manual is a "state-of-the-art" publication on air and hydronic balancing and adjusting. A contractor using the methods and principles described can properly supervise the balancing of any system. 1st Edition—1983.

5. Indoor Air Quality Manual

A "state-of-the-art" manual that identifies indoor air quality (IAQ) problems as they currently are defined. Also contains: The methods and procedures used to solve IAQ problems. The equipment and instrumentation necessary. The changes that must be made to the building and its HVAC systems. 1st Edition-1988.

6. Installation Standards for Residential Heating and Air Conditioning Systems

For residential and light commercial installations. This publication incorporates complete and comprehensive installation standards for conventional heating and cooling systems as well as solar assisted space conditioning and domestic water heating systems. 6th Edition—1988.

7. Energy Conservation Guidelines

Guidelines to familiarize the HVAC Contractor with the potential energy savings that can be made in new and existing buildings. Energy conservation information combined with good industry practice that an owner or systems designer should consider prior to selecting building equipment and systems. 1st Edition—1984.

8. Energy Recovery Equipment and Systems Air-to-Air

This comprehensive manual is an "A to Z State-of-the-Art" publication which has been developed by leading experts in the energy recovery industry so that anyone with a technical background can obtain a complete understanding of energy recovery equipment and systems. 1st Edition—1978.

9. Fibrous Glass Duct Construction Standards

Pressure Sensitive Tape Standards, performance of the fibrous glass board, fabrication of the fibrous glass board, fabrication of duct and fittings, closures of seams and joints, reinforcements with tee bars, channels, and tie-rods, and hangers and supports are covered in detail. 6th Edition.—1990.

10. Fire, Smoke and Radiation Damper Guide for HVAC Systems

An application and installation study guide for architects, engineers, code officials, manufacturers and contractors. Covers fire dampers, combination fire and smoke dampers, heat stops, fire doors, framing of structural openings, contract plan marking, installation instructions, and special applications. 3rd Edition—1986.

D CODES AND ORDINANCES

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, property protection, environmental concerns, and energy conservation regulations. Figure 1-1 illustrates relationships between laws, ordinances, codes, and standards that can affect the

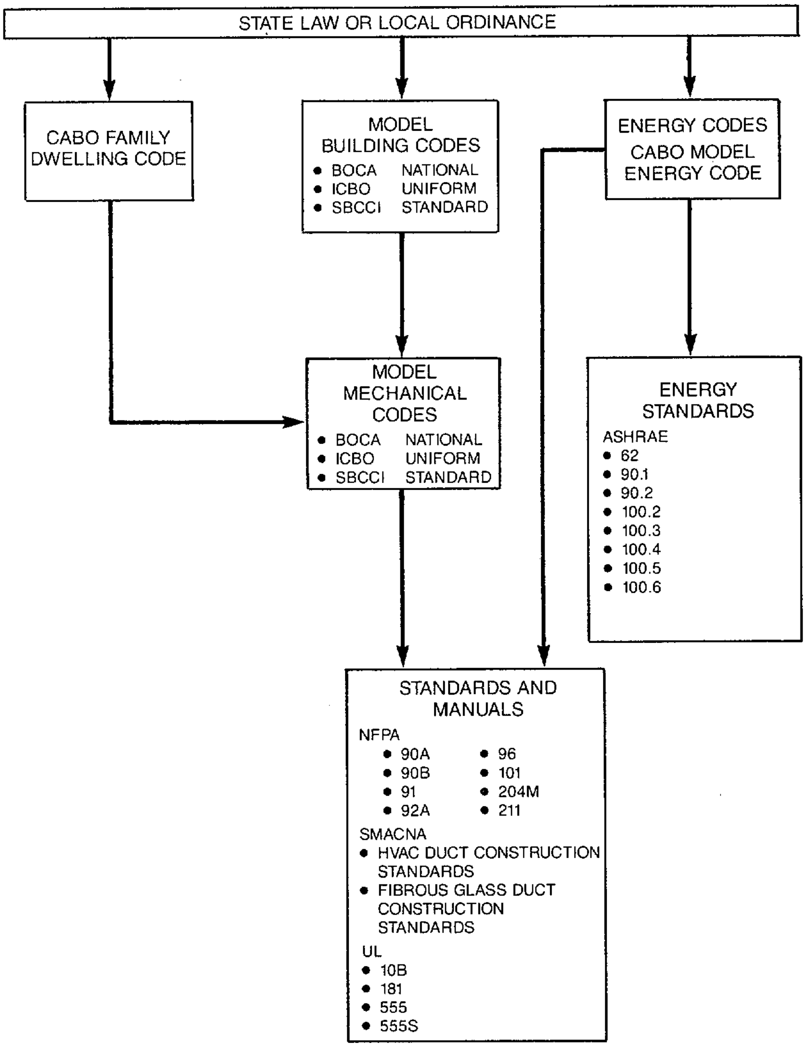


Figure 1-1 U.S.A. BUILDING CODES AND ORDINANCES

design and construction of HVAC duct systems; however, Figure 1-1 may not list 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.

Model code changes require long cycles for approval by the consensus process. Since the development of safety codes, energy codes and 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 compliance obligations affect their designs. If a provision is in conflict with the design intent, the designer should resolve the issue with local building officials. New or different construction methods can be accommodated by the provisions for equivalency that are incorporated into codes. Staff engineers from the model code agencies are available to assist in the resolution of conflicts, ambiguities, and equivalencies.

2. Fire and Smoke Codes

Fire and smoke control is covered in Chapter 47 of the 1991 ASHRAE "HVAC Applications" handbook. The designer should consider flame spread, smoke development, and toxic gas production from duct smoke development, and toxic gas production from duct and duct insulation materials. Code documents for ducts in certain locations within buildings rely on a criterion of "limited combustible material" (see Chapter 15—"Glossary") that is independent of 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 new materials; little research is reported on ratings of duct materials that have aged or of systems that are poorly maintained for cleanliness. Fibrous and other porous materials exposed to airflow in ducts may accumulate more dirt than nonporous materials.

National, state and local codes usually require fire and/or smoke dampers or radiation dampers wherever ducts penetrate fire-rated walls, floors, ceiling, partitions or smoke barriers. Any required fire, radiation or smoke dampers must be identified on the plans by the duct designer, and their location clearly shown. Before specifying dampers for installation in any vertical shafts or in any smoke evacuation systems, consult with local authorities having jurisdiction. Also review NFPA 92A "Recommended Practice for Smoke Control Systems".

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

1. The BOCA Basic Mechanical Code of Building Officials and Code Administrators International, Inc. Homewood, Illinois.
2. The Uniform Mechanical Code of International Conference of Building Officials (ICBO), Whittier, California.
3. The Standards Mechanical Code of Southern Building Code Congress International, Birmingham, Alabama.
4. The National Building Code of American Insurance Association, New York, Chicago and San Francisco.
5. National Fire Protection Association (NFPA), Quincy, Massachusetts.
6. National Building Code (by the National Research Council of Canada), Ottawa, Ontario, Canada.
7. Building Code of Australia, Australian Uniform Building Regulations Council, Federal Department of Industry, Technology and Commerce, Canberra, ACT., Australia.

Note: Federal state, and local codes or ordinances may modify or supercede the above listed codes.

CHAPTER 2

ECONOMICS OF DUCT SYSTEMS

A INTRODUCTION

All too often first cost has preoccupied the minds of both the building owner and the HVAC System designer, causing them to neglect giving proper consideration to system life and operating cost. A building that is inexpensive to build may contain systems that are expensive to operate and maintain.

With normal inflation building construction costs continue to escalate. The cost of money and energy continue to increase dramatically, but not always in the same proportion. These factors require a more rational and factual approach to the real costs of a system, by analyzing both owning and operating costs over a fixed time period (life cycle costs).

Chapter 49—"Owning and Operating Costs" of the 1987 ASHRAE "Systems and Applications Handbook" has a complete and detailed analysis of this subject. The basic elements are described as follows:

1. Annual Owning Costs

a) Initial Costs—The amortization period must be determined in which the initial costs are to be recovered and converted by use of a capital recovery factor (CRF) into an equivalent annual cost (see Table 2-1).

b) Interest

c) Taxes
   1. Property or real estate taxes.
   2. Building management personal property taxes.
   3. Other building taxes.

d) Insurance

2. Annual Operating Costs

a) Annual Energy Costs
   1. Energy and fuel costs.
   2. Water charges.
   3. Sewer charges.
   4. Chemicals for water treatment.

b) Annual Maintenance Costs
   1. Maintenance contracts.
   2. General housekeeping costs.
   3. Labor and material for replacing worn parts and filters.
   4. Costs of refrigerant, oil and grease.
   5. Cleaning & painting.
   6. Periodic testing and rebalancing.
   7. Waste disposal.

c) Operators—The annual wages of building engineers and/or operators should not be included as part of maintenance, but entered as a separate cost item.

Table 2-1 COST OF OWNING AND OPERATING A TYPICAL COMMERCIAL BUILDING

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

B INITIAL SYSTEM COSTS

The first financial impact of the HVAC duct system is the initial cost of the system. A careful evaluation of all cost variables entering into the duct system should be made if maximum economy is to be achieved. The designer has a great influence on these costs when specifying the duct system material, system operating pressures, duct sizes and complexity, fan horsepower, sound attenuation and determining the space requirements for both ductwork and apparatus.

Chapters 7 and 8 describe duct sizing methods in detail, and in Chapter 12, duct construction materials, are discussed. Other items, which are important in controlling first costs, are given later in this chapter. The amortization period or useful life for HVAC duct systems is normally considered to be the same as the life of the building, thus minimizing the annual effect of first cost of duct systems in comparison with other elements of an HVAC system which have a shorter useful life.

In Table 2-2, data is given for capital recovery factors based on years of useful life and the rate of return or interest rate. The purpose of this table is to give a factor which, when multiplied by the initial cost of a system or component thereof, will result in an equivalent uniform annual owning cost for the period of years chosen.

Example 2-1

Find the uniform annual owning cost if a $10,000 expenditure is amortized over 30 years at 12 percent.

Solution

The capital recovery factor (CRF) from Table 2-2 for 30 years at 12 percent is 0.12414. The uniform annual owning cost = 0.12414 x $10,000 = $1241.40.

Section XIV—"Energy Recovery System Investment Analysis" of the SMACNA "Energy Recovery Equipment and Systems" manual contains 19 pages of HVAC systems investment analysis text, equations, examples and financial tables.

C OPERATION COSTS

Since one normally considers that a duct system does not require any allowance for annual maintenance expense, except for equipment which may be a part of it, attention should be directed to energy costs which are created by the duct system. The important determining factor for fan size and power, other than air quantity, is system total pressure. In other sections of this manual, data will be given which will allow for the calculation of the system total pressure.

Since fans normally operate continuously when the building is occupied, the energy demand of various air distribution systems is one of the major contributors to the total building HVAC system annual energy costs. Fan energy cost can be minimized by reducing duct velocities and static pressure losses; however, this has a direct bearing on the system first cost and could influence building cost. Extra space might be required by the resultant enlarged ductwork throughout the building and larger HVAC equipment rooms also might be required. It is extremely important for the designer to adequately investigate and calculate the impact of operating costs versus system first cost.

Table 2-2 CAPITAL RECOVERY FACTORS (CRF)

	Rate of Return or Interest Rate, percent						
Years	6	8	10	12	15	20	25
2	0.54544	0.56077	0.57619	0.59170	0.61512	0.65455	0.69444
4	0.28859	0.30192	0.31547	0.32923	0.35027	0.38629	0.42344
6	0.20336	0.21632	0.22961	0.24323	0.26424	0.30071	0.33882
8	0.16104	0.17401	0.18744	0.20130	0.22285	0.26061	0.30040
10	0.13587	0.14903	0.16275	0.17698	0.19925	0.23852	0.28007
12	0.11928	0.13270	0.14676	0.16144	0.18448	0.22526	0.26845
14	0.10758	0.12130	0.13575	0.15087	0.17469	0.21689	0.26150
16	0.09895	0.11298	0.12782	0.14339	0.16795	0.21144	0.25724
18	0.09236	0.10670	0.12193	0.13794	0.16319	0.20781	0.25459
20	0.08718	0.10185	0.11746	0.13388	0.15976	0.20536	0.25292
25	0.07823	0.09368	0.11017	0.12750	0.15470	0.20212	0.25095
30	0.07265	0.08883	0.10608	0.12414	0.15230	0.20085	0.25031
35	0.06897	0.08580	0.10369	0.12232	0.15113	0.20034	0.25010
40	0.06646	0.08386	0.10226	0.12130	0.15056	0.20014	0.25006

For example, computations have confirmed that a continuously operating HVAC system costs 3 cents per cfm (6 cents per l/s) per 0.25 in w.g. (62 Pa) static pressure annually, based on 9 cents per kW/Hr cost of electrical energy. Therefore a 0.25 in. w.g. (62 Pa) increase in static pressure for a 100,000 cfm (50,000 l/s) system would add $3000 to the cost of the HVAC operation for one year. An increase in the design HVAC system operating static pressure also may add to the first costs of the system, by increasing the duct system pressure classification.

Table 2-3 INITIAL SYSTEM COSTS

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 (See Chapter 10 of this volume.)
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 auxiliaries
4. **Heat Distribution Equipment**
    a. Pumps, reducing valves, piping, piping insulation, etc.
    b. Terminal units or devices
5. **Cooling Distribution Equipment**
    a. Pumps, piping, piping insulation, condensate drains, etc.
    b. Terminal units, mixing boxes, diffusers, grilles, etc.
6. **Air Treatment and Distribution Equipment**
    a. Air heaters, humidifiers, dehumidifiers, filters, etc.
    b. Fans, ducts, duct insulation, dampers, etc.
    c. 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

D CONTROLLING COSTS

Some time proven industry practices which have generally proved to lower first costs are:

1. Use the minimum number of fittings possible. Fittings may be expensive and the dynamic pressure loss of fittings is far greater than straight duct sections of equal centerline length; i.e. one 24″ x 24″ (600 mm x 600 mm) R/W ratio = 1.0 radius elbow has a pressure loss equivalent to 29 feet (8.8 m) of straight duct.
2. Consider the use of semi-extended plenums (see Chapters 7 and 8).
3. Seal ductwork to minimize air leakage. This could even reduce equipment and ductwork sizes.
4. Consider using round duct where space and initial cost allows, as round ductwork has the lowest possible duct friction loss for a given perimeter.
5. When using rectangular ductwork, maintain the aspect ratio as close to 1 to 1 as possible to minimize duct friction loss and initial cost.

Table 2-4 ASPECT RATIO EXAMPLE (Same Airflows and Friction Loss Rates)

Duct Dimensions		Duct Area			Metal Thickness			Duct Weight*	
Inches	Millimetres	Square Inches	Square Metres	Aspect Ratio	Gauge	Inches	Millimetres	Pounds per Foot	Kilograms per Metre
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 × 300	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

*Duct Weight Based on 2 in.w.g. (500 Pa) Pressure Classification, 4 foot (1.22 m) Reinforcement Spacing. (Weight of Reinforcement and Hanger Materials **Not** Included.)

E DUCT ASPECT RATIOS

It is very important to emphasize the impact that increased aspect ratios of rectangular ducts have on both initial costs and operational costs. 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 rate. It is obvious from making a comparison of the weight of the higher aspect ratio ducts per foot (metre), that the cost of labor and material will be greater.

However, the cost of different types of duct work (and the use of taps versus divided flow fittings) can materially affect installation costs as shown by the average costs of different duct system segments 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 these tables and charts, as duct construction materials and methods, system operating pressures, duct system location, etc. may vary the cost relationships considerably!**

F PRESSURE CLASSIFICATION AND LEAKAGE

Repeatedly throughout this publication and other SMACNA publications, attention is drawn to the fact that the HVAC system designer should indicate the operating pressures of the various sections of the duct system on the plans. This is done in an effort to insure that each system segment will have the struc-

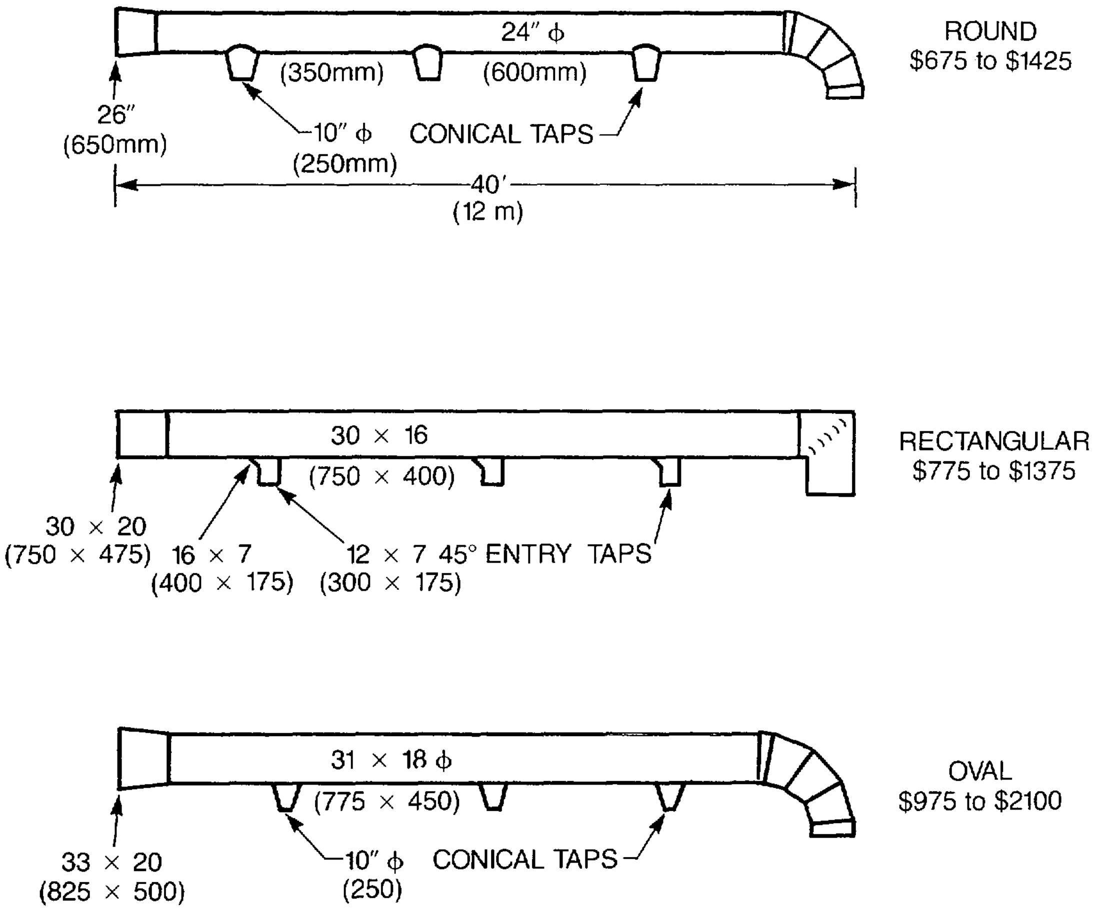


Figure 2-1 RELATIVE COSTS OF DUCT SEGMENTS INSTALLED
(Average costs of from several market areas to be used for comparison only)

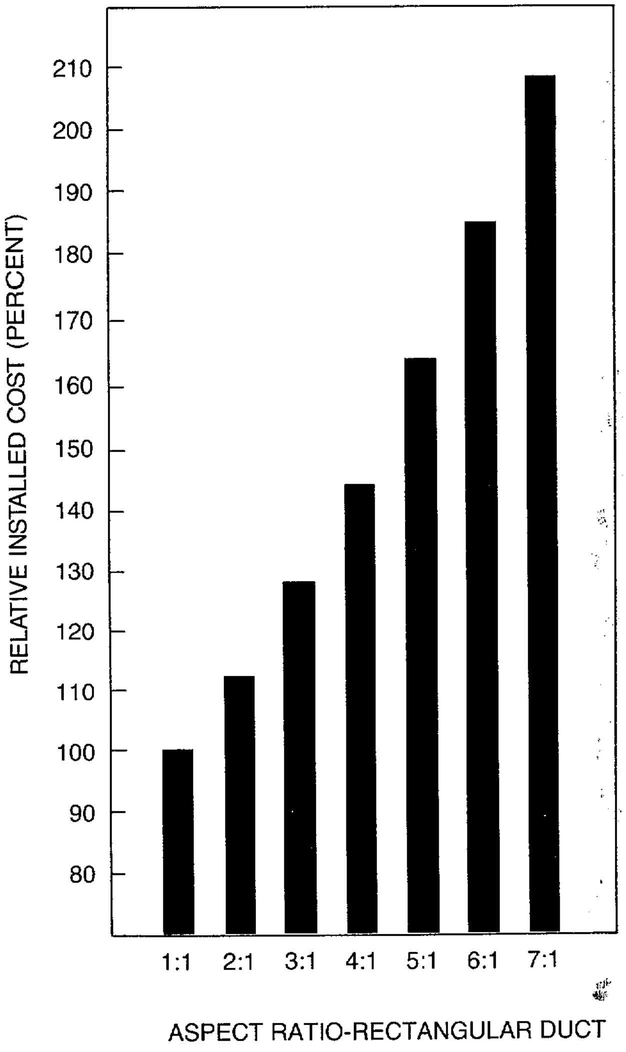


Figure 2-2 RELATIVE INSTALLED COST VS ASPECT RATIO

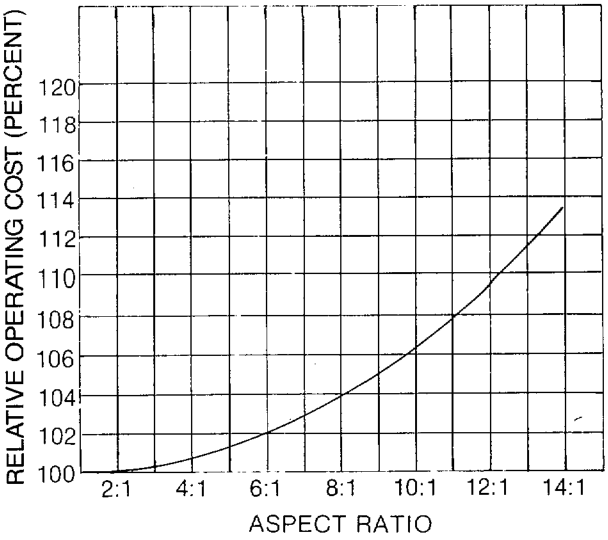


Figure 2-3 RELATIVE OPERATING COST VS ASPECT RATIO (based on equal duct area)

tural strength to meet the pressure classifications in SMACNA standards, but will keep initial duct system construction costs as low as possible. Each advancement to the next duct pressure class increases duct system construction costs.

Since the installed cost per system varies greatly, depending on local labor rates, cost of materials, area practice, shop and field equipment, and other variables, it is virtually impossible to present definite cost data. Therefore, a system of relative cost has been developed. Considering the lowest pressure classification, 0. to 0.5 in w.g. (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 classes.

Table 2-5 RELATIVE DUCT SYSTEM COSTS (Fabrication and Installation of Same Size Duct)

Duct Pressure Class		
In. w.g.	Pa	Cost Ratio
0- 0.5	0- 125	1.00
0.5- 1.0	125- 250	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

The comparison in Table 2-5 is made on the basis of galvanized sheet metal ductwork, and all ductwork being sealed in accordance with the minimum classifications as listed in the SMACNA "HVAC Duct Construction Standards—Metal and Flexible", First Edition 1985.

The amount of duct air leakage now may be determined in advance by the HVAC system designer, so that the estimated amount of leakage can be added to the system airflow total when selecting the system supply air fan. The amount of duct air leakage, in terms of cfm per 100 square feet (l/s per square metre), is based on the amount of ductwork in each "seal class". Additional information may be found in Chapter 5 of the SMACNA "HVAC Air Duct Leakage Test Manual", First Edition—1985, and in Chapter 32 of the 1989 ASHRAE "Fundamentals Handbook" It is important to note that a one percent (1%) air leakage rate for large HVAC duct systems is almost impossible to attain, and that large unsealed duct systems may develop leakage well above 30 percent of the total system airflow . The cost of sealing ductwork may add approximately 5 to 10 percent to the HVAC duct system fabrication and installation costs, but these costs may vary considerably, depending on job conditions and contractor plant facilities.

G COST OF FITTINGS

Chapter 14—"Duct Design Tables and charts contains fitting loss coefficients from which the HVAC system designer may select the one best suited for the situation. However, the fitting that gives the lowest, i.e. efficient dynamic loss, may also be the most expensive to make. A higher aspect ratio rectangular duct fitting might cost very little more to make than a square fitting, and much less to make than some round fittings. Variables apply here, probably more than in all previous discussions.

Without trying to develop a complete estimating procedure, using a 5 foot (1.5m) section of ductwork as a base, the relative cost of a simple full radius elbow of constant cross-sectional area is approximately from 4 to 8 times that of the straight section of ductwork. The relative cost of a vaned, square-throated elbow of constant size might even be greater.

The HVAC system designer should bear in mind that much of the ductwork fabricated today is done from automated equipment, whereby fabrication labor is reduced to a minimum by the purchase of an expensive piece of capital equipment. However, many fittings are still handmade, which results in very high labor to material costs.

Table 2-6 ESTIMATED EQUIPMENT SERVICE LIFE (2)

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	Evaporative 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^b	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 conditioners		Axial	20	Condensate	15
Single-zone	15	Propeller	15	Reciprocating engines	20
Multizone	15	Ventilating roof-mounted	20	Steam turbines	30
Boilers, hot water (steam)		Coils		Electric motors	18
Steel water-tube	24 (30)	DX, water, or steam	20	Motor starters	17
Steel fire-tube	25 (25)	Electric	15	Electric transformers	30
Cast iron	35 (30)	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		

CHAPTER 3

ROOM AIR DISTRIBUTION

A COMFORT CONDITIONS

An understanding of the principles of room air distribution helps in the selection, design, control and operation of HVAC air duct systems. The real evaluation of air distribution in a space, however, requires an affirmative answer to the question: "Are the occupants comfortable?" The object of good air distribution in HVAC systems 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 feet (2m) above floor level. To obtain comfort conditions within this zone, standard limits have been established as acceptable *effective draft temperature.* This term includes air temperature, air motion, relative humidity, and their physiological effects on the human body. Any variation from accepted standards of one of these elements causes 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 effects.

Although the percentage of room occupants who object to certain conditions may change over the years, Figures 3-1 and 3-2 provide insight into possible objectives of room air distribution. The data 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 inches (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 3-1 and 3-2 show that even higher velocities may be acceptable to some occupants. ASHRAE Standard 55-1981R 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 3-1 shows that up to 20 percent of occupants will not accept an ankle-to-sitting-level gradient of about 4°F (2°C). Poorly designed or operated systems in a heating mode can create this condition, which emphasizes the importance of proper selection and operation of perimeter systems.

To define the difference (θ) 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 (U.S. Units):

θ = effective draft temperature, °F
t_x = local airstream dry-bulb temperature, °F
t_c = average room dry-bulb temperature, °F
V_x = local airstream velocity, fpm
$a = 0.07$
$b = 30$

where (Metric Units):

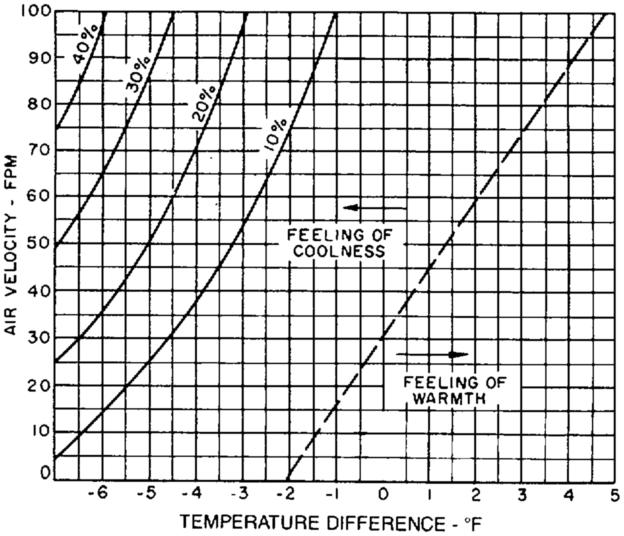
θ = effective draft temperature, °C
t_x = local airstream dry-bulb temperature, °C
t_c = average room dry-bulb temperature, °C
V_x = local airstream velocity, m/s
$a = 8$
$b = 0.15$

Equation 3-1 accounts for the feeling of "coolness" produced by air motion and is used to establish the neutral line in Figures 3-1 and 3-2. 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) and both of them 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.

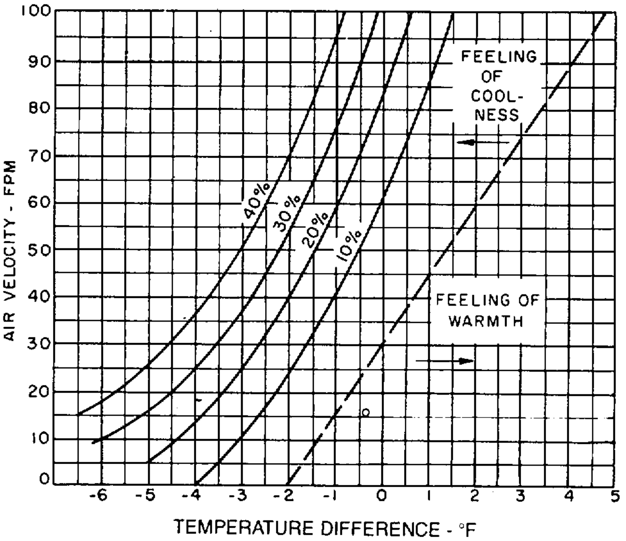
B AIR DIFFUSION PERFORMANCE INDEX (ADPI)

1. Comfort Criteria

A high percentage of people are comfortable in sedentary (office) occupations where the *effective draft temperature* (θ), as defined in Equation 3-1, is between -3°F and + 2°F (-1.7°C and + 1.1°C) and the

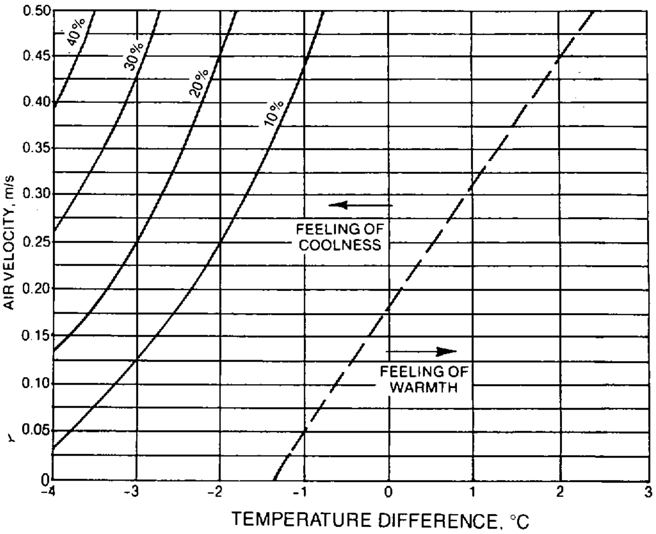


(A) ANKLE REGION

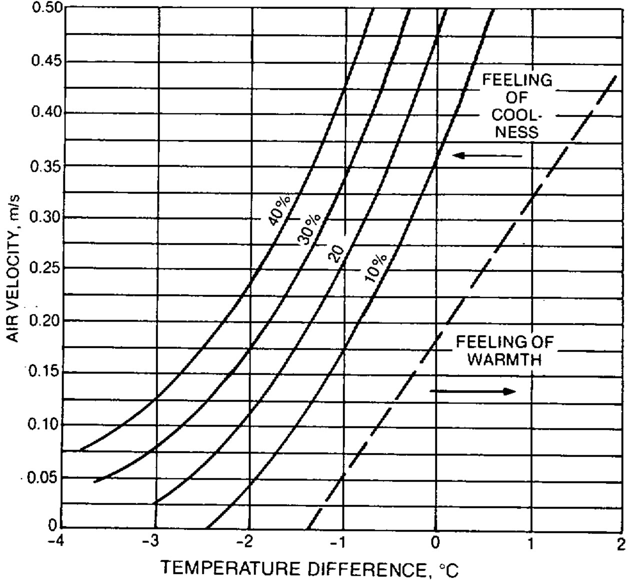


(B) NECK REGION

Figure 3-1 PERCENTAGE OF OCCUPANTS OBJECTING TO DRAFTS IN AIR-CONDITIONED ROOMS (U.S. UNITS) (2)



(A) ANKLE REGION



(B) NECK REGION

Figure 3-2 PERCENTAGE OF OCCUPANTS OBJECTING TO DRAFTS IN AIR-CONDITIONED ROOMS (METRIC UNITS) (2)

air velocity is less than 70 fpm (0.35m/s). If many measurements of air velocity and air temperature were made throughout the occupied zone of an office, the 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 is 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.

ADPI is a measure of cooling mode conditions. Heating conditions can be evaluated using ASHRAE Standard 55-1981R guidelines or the ISO Standard 7730-83, "Comfort Equations."

The following cooling zone design criteria for the various air diffusion devices maximize the ADPI and comfort. These criteria also account for airflow rate, outlet size, manufacturer's design qualities, and dimensions of the room for which the system is designed.

2. Definitions

A. 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. For all devices, the terminal velocity, V_t, was selected as 50 fpm (0.25 m/s) except in the case of ceiling slot diffusers, where the terminal velocity was selected as 100 fpm (0.5 m/s). Data for the throw of a jet from various outlets are generally given by each manufacturer for isothermal jet conditions and without boundary walls interfering with the jet. Throw data certified under Air Diffusion Council (ADC) Equipment Test Code 1062GRD-84 must be taken under isothermal conditions. Throw data not certified by ADC may be isothermal or not, as the manufacturer chooses. ASHRAE Standard 70-72R also includes specifications for reporting throw data.

B. 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.

C. 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. Table 3-1 summarizes definitions of characteristic length for various devices.

D. MIDPLANE

The *midplane* between outlets also can be considered the module line when outlets serve equal modules throughout a space, and characteristic length consideration can then be based on module dimensions.

3. Load Considerations

These recommendations cover cooling loads of up to 80 Btu/h·ft² (250 W/m²) of floor surface. The loading is distributed uniformly over the floor up to about 7 Btu/h·ft² (22 W/m²), lighting contributes about 10 Btu/h·ft² (31 W/m²) and the remainder is supplied by a concentrated load against one wall that simulated a business machine or a large sunloaded 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.

Table 3-1 CHARACTERISTIC ROOM LENGTH FOR DIFFUSERS

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

4. 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 (1) high sidewall grille, (2) sill grille, (3) two and four-slot ceiling diffusers, (4) conetype circular ceiling diffusers, (5) light troffer diffusers, and (6) square-faced perforated and louvered ceiling diffusers. Table 3-2 summarizes the results of the recommendations on values of T_v/L by giving the value of T_v/L where 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.

5. Outlet Type Selection

No criteria have been established for choosing among the six types of outlets to obtain an optimum ADPI. All outlets tested, when used according to these recommendations, can have ADPI values that are satisfactory [greater than 90 percent for loads less than 40 Btu/h·ft² (126 W/m²)].

Table 3-2 AIR DIFFUSION PERFORMANCE INDEX (ADPI) SELECTION GUIDE (2)

Terminal Device	Room Load Btu/h·ft²	Room Load W/m²	$T_{0.25}/L$ for Max. ADPI	Maximum ADPI	For ADPI Greater Than	Range of $T_{0.25}/L$
High Sidewall Grilles	80	250	1.8	68	—	—
	60	190	1.8	72	70	1.5-2.2
	40	125	1.6	78	70	1.2-2.3
	20	65	1.5	85	80	1.0-1.9
Circular Ceiling Diffusers	80	250	0.8	76	70	0.7-1.3
	60	190	0.8	83	80	0.7-1.2
	40	125	0.8	88	80	0.5-1.5
	20	65	0.8	93	90	0.7-1.3
Sill Grille Straight blades	80	250	1.7	61	60	1.5-1.7
	60	190	1.7	72	70	1.4-1.7
	40	125	1.3	86	80	1.2-1.8
	20	65	0.9	95	90	0.8-1.3
Sill Grille Spread blades	80	250	0.7	94	90	0.8-1.5
	60	190	0.7	94	80	0.6-1.7
	40	125	0.7	94	—	—
	20	65	0.7	94	—	—
Ceiling Slot Diffusers (for T_{100}/L)	80	250	0.3*	85	80	0.3-0.7
	60	190	0.3*	88	80	0.3-0.8
	40	125	0.3*	91	80	0.3-1.1
	20	65	0.3*	92	80	0.3-1.5
Light Troffer Diffusers	60	190	2.5	86	80	<3.8
	40	125	1.0	92	90	<3.0
	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

* Given for $T_{0.5}/L$

6. 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 3-1).

d) Select the recommended T_v/L ratio from Table 3-2.

e) Calculate the throw distance (T_v) by multiplying the recommended T_v/L ratio from Table 3-2 by the room length (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 (U.S. Units)

Specifications:

Room Size: 20 ft by 12 ft with 9 ft. ceiling
Type device: High sidewall grille, located at the center of 12 ft endwall, 9 in. from ceiling.
Loading: Uniform, 10 Btu/h·.ft² or 2400 Btu/h
Air Volume: 1 cfm/ft² or 240 cfm for the one outlet

Data Required:
- Characteristic length: (L) = 20 ft (length of room: Table 3-1
- Recommended Tv/L = 1.5 (Table 3-2)
- Throw to 50 fpm = T_{50} = 1.5 x 20 = 30 ft

Solution

Refer to the manufacturer's catalog for a size that gives this isothermal throw to 50 fpm. Manufacturer recommends the following sizes, when blades are straight, discharging 240 cfm: 16 in. by 4 in., 12 in. by 5 in. or 10 in. by 6 in.

Example 3-1 (Metric Units)

Specifications:
- Room size: 6000 by 4000 mm with 2500 mm high ceiling
- Type Device: High sidewall grille, located at the center of 4000 mm endwall, 230 mm from ceiling
- Loading: Uniform, 30 W/m² or 720 W
- Air Volume: 0.5 l/s per m² or 120 l/s per outlet

Data Required:
- Characteristic length L = 6000 mm (length of room: Table 3-1).
- Recommended T_v/L = 1.5 (Table 3-3)
- Throw to 0.25 m/s = $T_{0.25}$ = 1.5 x 6 = 9m

Solution

Refer to the manufactuer's catalog for a size that gives this isothermal throw to 0.25 m/s. Manufacturer recommends the following sizes, when blades are straight, discharging 120 l/s: 400 mm by 100 mm, 300 mm by 125 mm or 250 mm by 125 mm.

C AIR DISTRIBUTION FUNDAMENTALS

1. Air Diffusion

Conditioned air normally is 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, therefore, calls for entrainment of room air by the primary airstream outside the zone of occupancy to reduce air motion 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 also will 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.

2. Surface (Coanda) Effect

Drawings A and B of Figure 3-3 illustrate the *Coanda effect* phenomenon. Since turbulent jet airflow from a

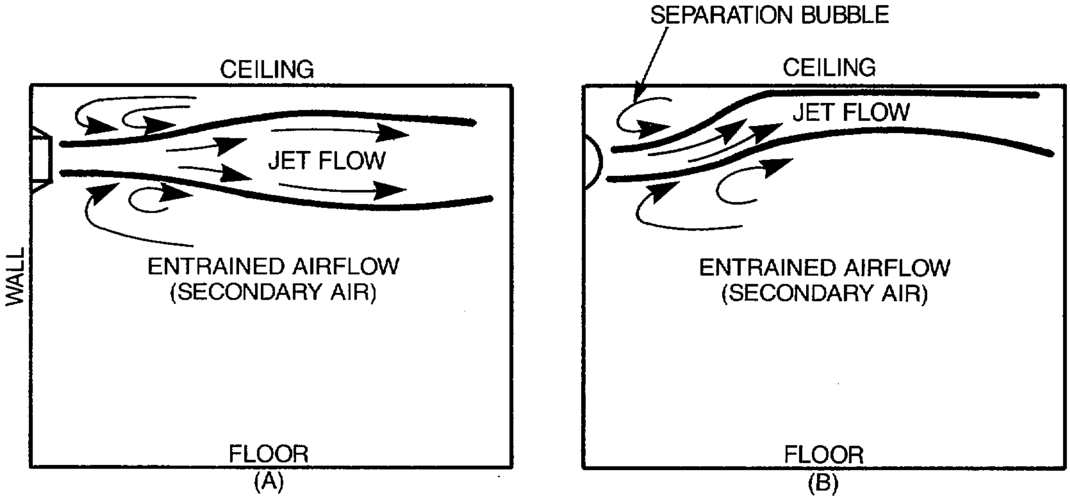


Figure 3-3 SURFACE (COANDA) EFFECT

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 ceilings, the surface interrupts the flow path on that side as shown in Figure 3-3 (B). The result is that no more secondary air is flowing on that side to replace the air being entrained with the jet airflow. This causes a lowering of the pressure on that side of the outlet device, creating a low-pressure bubble that causes the 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 horizontally projected cool 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 33 percent. Airflow per unit area for these types of outlets can, therefore, be increased. 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 airconditioning systems must be restricted to a range of 15°F to 20°F (8°C to 11°C). Airstreams from slot diffusers and grilles show a marked tendency to drop because of the lack of surface effect.

3. Smudging

Smudging may 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 (lobbies, stores, etc.) 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.

4. Sound Level

The sound level of an outlet is a function of the discharge velocity and the transmission of systemic 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.

The cause of higher frequency sounds can be pinpointed as outlet or systemic sounds by removing the outlet during operation. A reduction in sound level indicated that the outlet is causing noise. If the sound level remains essentially unchanged, the system is at fault. Chapter 42 "Sound and Vibration Control" in the 1991 ASHRAE "HVAC Applications" handbook has more information on design criteria, acoustic treatment, and selection procedures.

5. Effect of Blades

Blades 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 straight position) has a spread of 14° to 24°, 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 at each end of 45°) has a spread of about 60°, and 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 uniformily 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.

In addition to vertical blades that normally spread the air horizontally, horizontal blades may spread the air vertically. However, spreading the air vertically risks hitting beams or other obstructions or blowing primary air at excessive velocities into the occupied zone. On the other hand, 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 or arched air path. The air path should be arched sufficiently to miss the beams and prevent the primarily or induced airstream from striking furniture and obstacles and producing objectionable drafts.

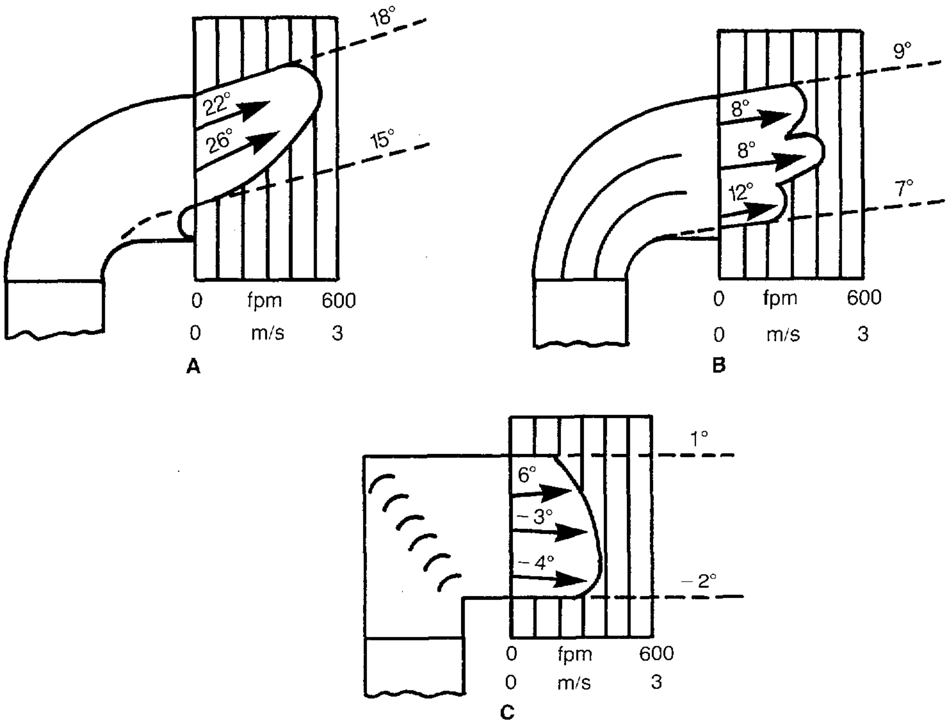
6. Duct Approaches to Outlets

The manner in which the airstream is introduced into the outlet is important. To obtain 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 air flow 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 receive the air perpendicularly and at 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, and 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. In all these cases, special devices for directing and equalizing the airflow are necessary for proper direction and diffusion of the air.

A. STACK HEADS

Tests conducted with the 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 types). Cushion chambers at the top of the stack heads are not beneficial. Figure 3-4 shows



Stack, 14 · 6 in. (350 × 150 mm); Outlets, 14 · 9 in. (350 × 225 mm); Stack Velocity, 500 fpm (2.5 m/s)

A. Rounded Throat and Rounded Back.
B. Rounded Throat and Back and 2 Splitters.
C. Square Throat and Back and Turning Vanes.

Figure 3-4 OUTLET VELOCITY AND AIR DIRECTION DIAGRAMS FOR STACK HEADS WITH EXPANDING OUTLETS

the direction of flow, diffusion, and velocity [measured 12 inches (300 mm) from opening] of the air for various stack heads tested, expanding from a 14 in. by 6 in. (350 mm x 150 mm) stack to a 14 in. by 9 in. (350 mm x 225 mm) opening, without 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 nonexpanding elbows of similar shapes, for velocities from 200 to 400 fpm. (1 to 2 m/s).

In tests conducted with 3 in. by 10 in. (75 mm x 250 mm), 4 in by 9 in., and 6 in. by 6 in. (150 mm x 150 mm), side outlets in a 6 in. by 20 in. 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, therefore, was not desirable.

B. BRANCH TAKEOFFS

SMACNA duct fitting research at the ETL Laboratories and the SMACNA "bubble" airflow research video have shown, both from duct traverse pressure readings and from visual observation of airflow with 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 found 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 swirling turbulence. See the discussion in Chapter 5, Section E "Dynamic Losses".

D 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, therefore, determine both the type and location of the supply outlets.

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, rather than dissipated into the conditioned space. Where lighting loads are heavy [5 W/ft^2 (54 W/m^2)] and ceilings relatively high [above 15 ft (4.6m)], 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 (See Figures 3-5 to 3-9).

1. Group A Outlets.

Outlets mounted in or near the ceiling with horizontal air discharge should not be used with temperature differentials exceeding 25°F (14°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° deflection has a longer throw and a great drop than a ceiling diffuser with a single 360° 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, which increases the throw and decreases the drop. To prevent smudging, the total air should should be directed away from the ceiling, but this rarely is practical, except for very high ceilings. For optimum air

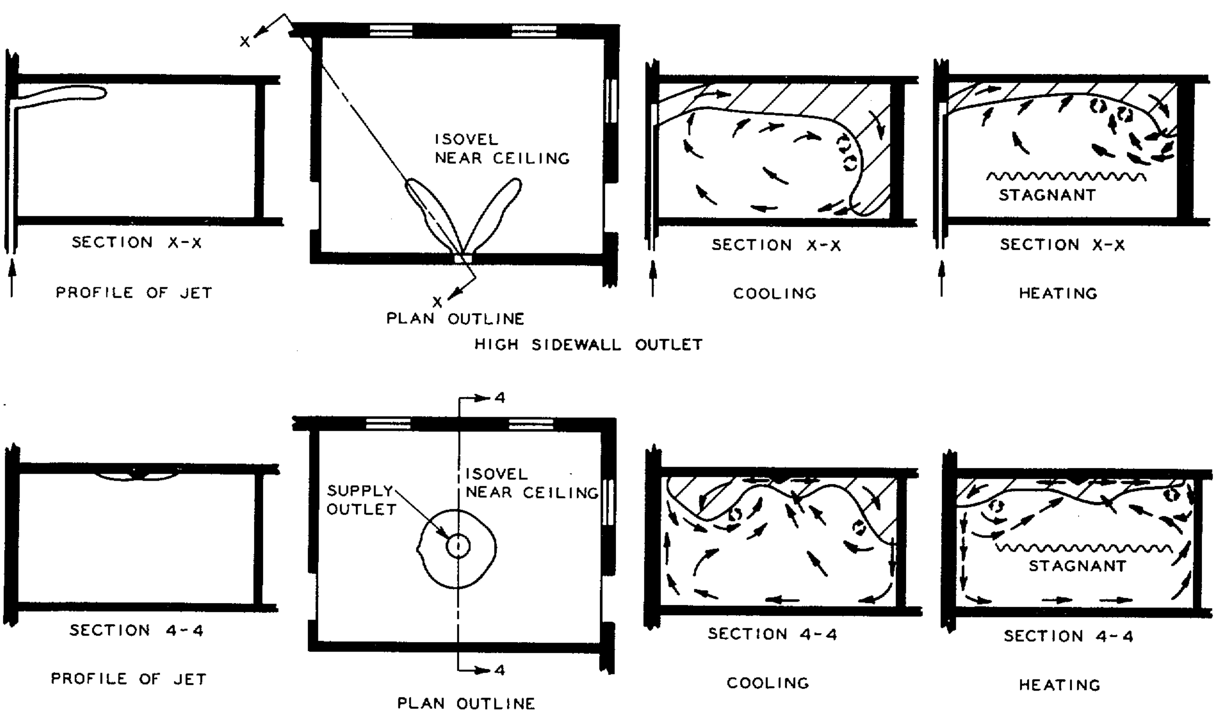


Figure 3-5 AIR MOTION CHARACTERISTICS OF GROUP A OUTLETS (2)

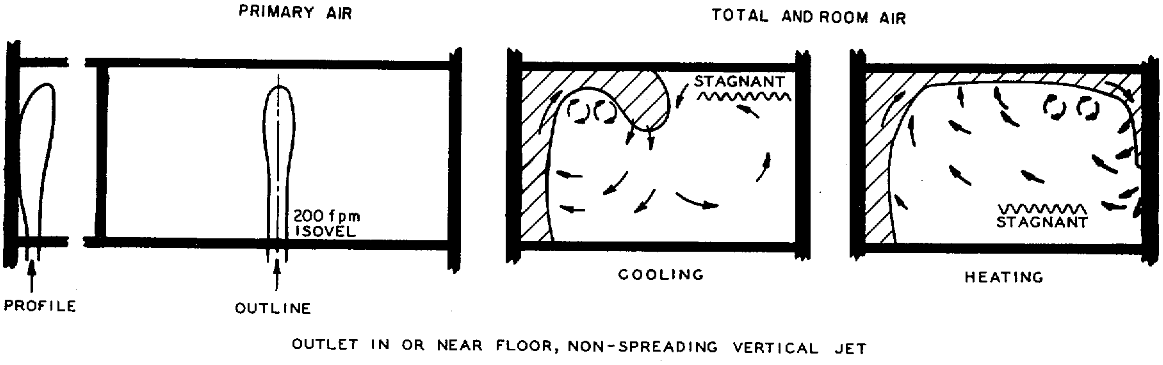


FIGURE 3-6 AIR MOTION CHARACTERISTICS OF GROUP B OUTLETS (2)

diffusion in areas without high ceilings, total air should scrub the ceiling surface.

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

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 6 foot (2m) level.

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. Ade-

Table 3-3 GUIDE FOR SELECTION OF SUPPLY OUTLETS

Type of Outlet	Floor Space CFM/per Sq. Foot	Floor Space l/s per Sq. Metre	Approximate Maximum Air Changes/Hour For 10 Ft. (3 m) Ceiling Height
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

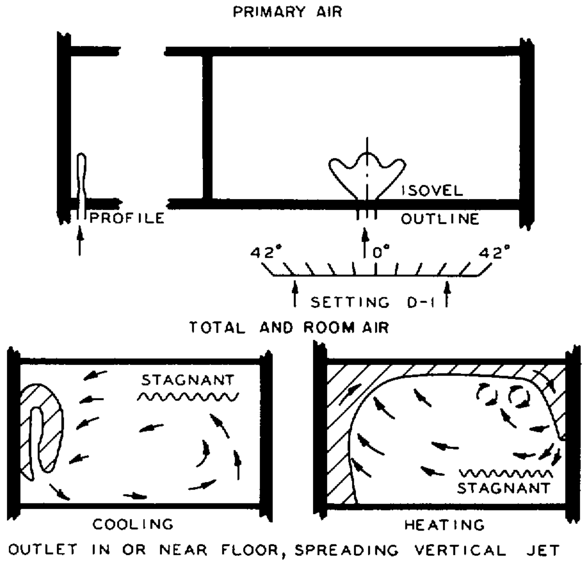


Figure 3-7 AIR MOTION CHARACTERISTICS OF GROUP C OUTLETS (2)

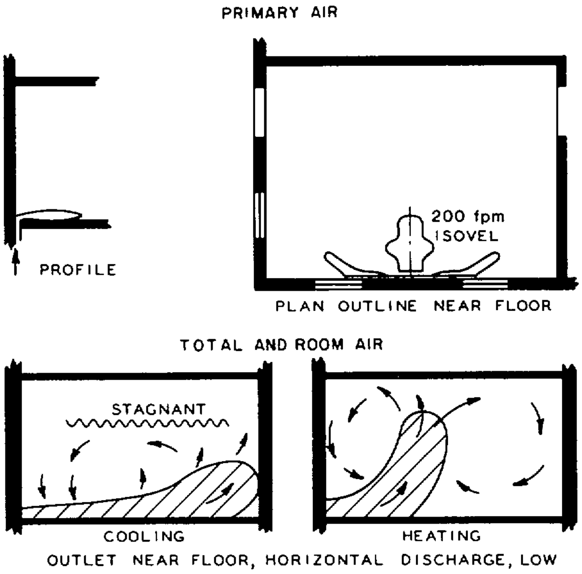


Figure 3-8 AIR MOTION CHARACTERISTICS OF GROUP D OUTLETS (2)

quate induction by a ceiling-mounted diffuser prevents shortcircuiting of unmixed supply air between supply outlet and ceiling-mounted returns.

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 inches (150 mm) and an angle of 15° is 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.

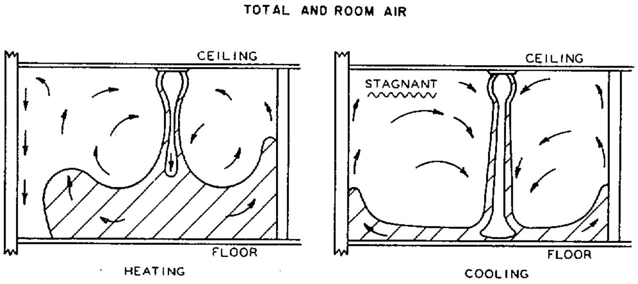


Figure 3-9 AIR MOTION CHARACTERISTICS OF GROUP E OUTLETS (2)

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°F (8°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.

The 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.

4. Group D Outlets

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

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.—seldom for both.

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

6. Ventilating Ceilings

ASHRAE Investigations indicate that air temperatures and velocities throughout a room cooled by a ventilating ceiling are a linear function of room load (heat load per unit area), and are not affected significantly by variations in ceiling type, total air temperature differential, or air volume flow rate. Higher room loading produces wider room air temperature variations and higher velocities, which decreases performance.

These studies also found no appreciable difference in the performance of air diffusing ceilings and circular ceiling diffusers for lower room loads [20 Btu/h·ft^2 (65 W/m^2)] For higher room loads [80 Btu·ft^2, (250 W/m^2)] an air-diffusing ceiling system has only slightly larger vertical temperature variations and slightly lower room air velocities than a ceiling diffuser system.

When the ventilating ceiling is used at exterior exposures, the additional load at the perimeter must be considered. During heating operation, the designer must provide for the cold wall effect, as with any ceiling supply diffusion system. Cold air in plenums also may cause condensation to form on the exterior facade of the plenum. The sound generated by the air supply device must also be considered in total system analysis to ensure that room sound levels do not exceed the design criteria. Check local codes for maximum plenum sizes, fire dampers, and other restrictions to the use of ceiling plenums.

E OUTLET CRITERIA

1. General

Outlets with higher induction rates move (throw) 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 velocities.

Outlets with high induction characteristics can also be used advantageously in air-conditioning systems with low supply-air temperatures and consequent 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°F to 35°F (17°C to 19°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°F (14°C). Grilles may generally be used in well-designed systems with cooling temperature differentials up to 20°F (11°C).

2. Selection Procedures

The following procedure is generally used in selecting outlet locations and types:

a) Determine the amount of air to be supplied to each room. (Refer to Chapters 25 and 26 in the 1989 ASHRAE "FUNDAMENTALS" Handbook to determine air quantities for heating and cooling.

b) Select the type and quantity of outlets for each room, considering such factors as air quantity required, distance available for throw or radius of diffusion, structural characteristics, and architectural concepts. Table 3-3 is based on experience and typical ratings of various outlets. It may be used as a guide to the outlets applicable for use with various room air loadings. Special conditions, such as ceiling heights greater than the normal 8 to 12 feet (2.4 to 3.6 m) and exposed duct mounting, as well as product modifications and unusual conditions of room occupancy, can modify this table. Manufacturers' rating data should be consulted for final determination of the suitability of the outlets used.

c) Locate outlets in the room to distribute the air as uniformly as possible. Outlets may be sized and located to distribute air in proportion to the heat gain or loss in various portions of the room.

d) Select proper outlet size from manufacturers' ratings according to air quantities, discharge velocities, distribution patterns, and sound levels. Note manufacturers' recommendations with regard to use. In an open space configuration, the interaction of airstreams from multiple diffuser sources may alter single diffuser throw data or single diffuser air temperature air velocity data, and it may not be sufficient to predict particular levels of air motion in a space. Also, obstructions to the primary air distribution pattern require special consideration.

3. Grille and Register Applications

Properly selected grilles operate satisfactorily from high side and perimeter locations in the sill, curb, 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.

A. 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.

B. 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 inches (200 mm) from the perimeter surface, it is usually desirable to deflect the airstream toward the perimeter wall. This can be done with adjustable or fixed deflecting blade grilles.

C. CEILING INSTALLATIONS

Ceiling installations generally are limited to grilles having curved blades, which, because of their design, provide a horizontal pattern. Curved blade grilles may also be used satisfactorily in high side wall or perimeter installations.

4. Slot Diffuser Applications

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

A. HIGH SIDE WALL INSTALLATION

The perpendicular-flow slot diffuser is best suited to high side wall installations and perimeter installations in sills, curbs, and floors. The air discharged from a perpendicular slot diffuser will not drop if the diffuser is located within 6 to 12 inches (150 to 300 mm) from the ceiling and is long enough to establish surface effect. Under these conditions, air travels along the ceiling to the end of the throw. If the slot diffuser is mounted 1 to 2 feet (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 feet (600 mm) below the ceiling, premature drop of cold air into the occupied zone will probably result.

B. 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 transiently 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.

C. SILL INSTALLATION

The perpendicular-flow slot diffuser is well suited to sill installation, but it may also be installed in the curb and floor. When the diffuser is located within 8 inches (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 inches (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 of jet, as recommended by the manufacturer and also as shown by 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.

D. AIR-LIGHT FIXTURES

Slot diffusers, having a single-slot discharge and nominal 2, 3 and 4 feet (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 one another's equipment.

5. Ceiling Diffuser Applications

A. CEILING INSTALLATIONS

Ceiling diffusers should be mounted in the center of the space that 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.

B. SIDE WALL INSTALLATIONS

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

Table 3-4 SUPPLY AIR OUTLET TYPES

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 High sidewall installation Perimeter installation
Perpendicular-flow slot diffuser	Generally 25 to 1 dimensional aspect ratio with maximum height of 3 inches (75 mm)	High sidewall installation Perimeter installation in sills, curbs and floors
Parallel-flow slot diffuser	Generally 25 to 1 dimensional aspect ratio with maximum height of 3 inches (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— 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

Perforated face diffuser	Perforated face plate with or without deflection device to obtain a horizontal discharge pattern	Install in center of area served or control discharge pattern when installed off center of area served
Variable area diffuser	Parallel or concentric passages or perforated face with means to vary discharge area	Use with variable volume system to minimize variation to throw with variable supply air volume
Air distributing ceilings	Ceiling system provided with round holes or slots	Use with ceiling supply plenum—particularly suited to large zones of uniform room temperature
Linear grille	Linear slot width ½ to 1 inch (12 to 25 mm), continuous length with adjustable airflow blades	Ceiling and perimeter with air deflection adjustable from 1-way horizontal to vertical to 2-way horizontal
Egg crate grille	Fixed square grid	Ceiling or sidewall (no pattern adjustment)
High capacity double deflection blade grille	One set vertical and one set horizontal adjustable blades. Blades are deep & wide spaced	High sidewall installation where high capacity and low discharge velocity are required
Drum louvers and adjustable high capacity diffusers	Adjustable direction core	High sidewall or ceiling installation, where directional and/or long throw required provides spot heating or spot cooling to areas of high load requirements

C. EXPOSED DUCT INSTALLATION

Some ceiling diffuser types, particularly stepped-down units, perform satisfactorily on exposed ducts. Consult manufacturers' catalogs for specific types.

D. 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 toward the downblow pattern position, the surface effect maintains the horizontal discharge pattern until the discharge airstream is effectively deflected at the diffuser face, resulting in a vertical pattern. However, when adjustable pattern 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.

6. Air-Distributing Ceilings

The air-distributing ceiling uses the confined space above the ceiling as a supply plenum that receives air from stub ducts. The plenum should be designed to achieve uniform plenum pressure, resulting in uniform delivery of air to the conditioned space below.

Air is delivered through round holes or slots in the ceiling material or suspension system. These holes and slots vary in shape and size among manufacturers. Various manufacturers have developed a number of products based on the principle of a supply plenum, with sizes ranging mainly from 1 by 1 foot (300 by 300 mm) tile for a concealed grid to 2 by 4 feet (600 by 1200 mm) lay-in panels for an exposed grid. Sometimes, the slots are equipped with adjustable dampers to facilitate changing the open area after installation.

The upper limit of plenum pressure must be that recommended by the ceiling manufacturer. It generally ranges from 0.10 to 0.15 in. w.g. (25 to 35 Pa), dictated by resistance to sag, to a lower pressure limit of about 0.01 in. w.g. (2.5 Pa), where uniformity of plenum pressure becomes more important. The range of air rates extends from about 15 cfm (7.0 l/s) down to about 1 cfm per square foot (5 l/s per m^2 of floor area. High flow rates are recommended only for low-temperature differentials.

Active portions of the air-distributing ceilings should be located with respect to room load distribution, with higher airflow rates at the exterior exposures. This

Table 3-5 SUPPLY AIR OUTLET PERFORMANCE

Group—Type	Mounting	Discharge Direction	Characteristics	
			Cooling	Heating
A High Sidewall Grilles Sidewall Diffusers Ceiling Diffusers Slot Diffusers (Parallel Flow)	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
Variable Area Grille Variable Area Diffuser	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
B Floor Grilles Baseboard Units Fixed Bar Grilles Linear Grilles	Floor, Low Sidewall, Sill	Vertical Non Spreading Air Jet	Small amount of stagnant air generally above occupied zone	Smaller amount of stagnant air than Group A outlets
C Floor Grilles Adjustable Bar Grilles Linear Diffusers	Floor, Low Sidewall, Sill	Vertical Spreading Air Jet	Larger amount of stagnant air than Group B outlets	Smaller amount of stagnant air than group B outlets—particularly suited to heating applications
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 Linear Grilles Grilles Slot Diffusers (Vertical Flow) Sidewall 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

method of air distribution is particularly suited to large zones of uniform room temperature. Where different room temperatures are desired, a separate ceiling plenum is required for each zone. Construction of the ceiling plenum requires care with regard to air tightness, obstructions causing unequal plenum pressure and temperature, heat storage effect of the structure, and the influence of a roof or the areas surrounding the plenum.

7. 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 percent 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.

F INLET CRITERIA

1. General

Return air or exhaust 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, therefore, 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 branch ducts are necessary to balance the airflow in the return air duct system.

2. Types of Inlets

A. ADJUSTABLE BLADE GRILLES

The same grilles used for air supply are used to match the deflection setting of the blades with that of the supply outlets.

B. FIXED BLADE GRILLES

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 sightproof; 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 rest rooms and utility areas.

F. DIFFUSERS

Ceiling and Slot Diffusers may also be used as return and exhaust inlets.

3. Selection Procedures

Select return 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 does not significantly affect air motion. The location of return and exhaust inlets will not compensate for ineffective supply air distribution.

The selection of return and exhaust inlets depends on (a) velocity in the occupied zone near the inlet, (b) permissible pressure drop through the inlet, and (c) noise.

A. VELOCITY

Control of the room air motion to maintain comfort conditions depends on proper supply outlet selection. The effect of air flow through return inlets on 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. Drafty conditions rarely occur near return inlets. Table 3-6 shows recommended return air inlet face velocities.

Table 3-6 RECOMMENDED RETURN AIR INLET FACE VELOCITIES

Inlet Location	Velocity Over Gross Inlet Area Feet per Minute	Metres 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

B. 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.

C. NOISE

The problem of return air inlet noise is the same as that for supply outlets. In computing resultant room noise levels from 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 is the frequent installation of the latter at ear level. When they are so located, the return inlet velocity should not exceed 75 percent maximum permissible outlet velocity.

4. 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.

A. HVAC SYSTEM LOADS

An HVAC system operating in the cooling mode performs best when generated heat is removed at its source rather than distributed throughout the conditioned space. Heat from solar and miscellaneous loads such as 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 that heat from being distributed into the conditioned space. Combination return air/lighting fixtures, besides increasing the HVAC system efficiency, improve light output and extend the lamp life. The manufacturers of fixtures, ceiling grids, and grilles give performance information (airflow rate, pressure drop, and heat removal rate) of their product.

B. EXHAUST OUTLETS

Exhaust outlets located in walls and doors, depending on their 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. The pressure drop through door returns should not be excessive, or the air diffusion to the room may be seriously unbalanced by opening or closing the doors. Outward leakage through doors or windows cannot be counted on for dependable results.

C. SPECIAL SITUATIONS

The designer should consider special situation requirements in locating return and exhaust inlets in bars, kitchens, lavatories, dining rooms, club rooms, conference rooms, etc. These normally should be located near or at the ceiling level to collect the warm air "build-up", odors, smoke and fumes.

G SUMMARY

1. General

Approximate values of pressure drop requirements for various types of air outlet and inlet devices may be found in Chapter 9. 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:

1. Pressure loss through outlet
2. Throw
3. Spread
4. Drop
5. 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 dampers to control the excessive air distribution may create unacceptable noise levels. Additional data can be found on this subject in Chapters 7, 8, and 9.

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 fixtures to avoid deflection of the airstream. Outlets should be located to neutralize undesirable convection currents set up by concentrated loads (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.

2. Supply Outlets

To summarize the procedure for supply outlet location and selection:

a) Determine room supply air quanitity 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

Table 3-7 RETURN & EXHAUST AIR INLET TYPES

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 pattern to match supply outlets	Return, exhaust and transfer grilles
V-blade grille	Sight proof	Particularly 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 3-8 ACCESSORY DEVICES

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
Diffuser 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.

*Do not use as a duct system balancing damper

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 (see Chapter 11).

3. Accessories

Accessory Devices should be chosen to obtain the desired design performance of air outlet and inlet devices (see Table 3-8).

4. Return & Exhaust Inlets

To summarize the procedure for inlet location and selection:

a) Determine room return and an exhaust air quantity from design load calculations.
b) Select type and quantitiy of inlets for each room evaluating:
    1) Inlet airflow
    2) Inlet velocity
    3) Architectural requirements
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 11).

CHAPTER 4

GENERAL APPROACH TO DUCT DESIGN

A DUCT SYSTEM SELECTION

HVAC system duct design follows after the room loads and air quantities have been determined. Design procedure details will be examined more minutely in succeeding 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 for a particular type. In any event, the specific type of system will affect the type of air handling apparatus that is selected.

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:

1. Air cleanliness
2. Odor
3. Space temperature
4. Means of temperature control
5. Air motion and distribution
6. Mean radiant temperature
7. Quality of ventilation
8. Humidity control
9. Noise level

Air systems may be separated into two main categories—single duct systems and dual duct systems. Single duct systems are those in which the main heating and cooling sources are 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 (multizone unit). A dual duct system contains the main heating and cooling sources in a parallel airflow, using a separate cold and warm air duct distribution system which 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 to the design of variable volume duct systems, dual-duct systems, etc. These fundamental design methods may be used when designing special duct systems, which are found in the SMACNA "HVAC systems—Applications" manual.

B AIR DISTRIBUTION

First, locate the supply air outlets, and then select the size and type 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 assured by proper consideration of the basic factors in the selection of the outlet terminal devices.

The outlet terminal devices should provide the proper air velocities within the room occupied zone (floor to six feet (2m) above the floor) and the proper temperature equalization. Entrainment of the room air by the primary (or supply) airstream at the outlet terminal to attain the required temperature equalization and to counteract the effects of natural room air convection is very important.

It is recommended that air distribution terminal devices be selected from industry standard types or configurations so that they can be obtained from many sources. Most terminal device manufacturers' catalogs furnish data on airflow throw, drop, air pattern, terminal velocities, and on acoustics, ceiling heights, etc.

Supply outlets on the same branch should be chosen with approximately the same pressure loss [no more than 0.05 in. w.g.(12.5 Pa) variation] through 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 application engineering data. However, some of the basic procedures used in the selection of air distribution equipment are:

1. Consider the ambient conditions that could affect comfort.
2. Decide on the location of air supply outlets, such as in the floor, sill, sidewall, exposed duct or ceiling, taking into account the type system serving them. Locate return and exhaust air devices.
3. Consider the special requirements affecting outlets when used with systems such as a variable air volume (VAV) system.
4. Select straightening vanes and dampers to be used with outlet devices to provide uniform face velocity and minor balancing.
5. Refer to manufacturer's data regarding throw, spread, drop, noise level, etc.

C ZONING

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

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 with two exposed walls.

Similarly, the interior zones may also be broken down into control zones to satisfy individual room requirements or variations created by internal loads, such as lights, people or equipment.

D PRELIMINARY LAYOUT

The next step is to draw preliminary schematic diagram for the ductwork which will convey the design air quantity to the selected zones and outlets by the most efficient and economical path. It is suggested that this layout be made on a reproducible tracing of the architectural floor plans. By doing this, the designer, will have a better feel for the final relationship of air terminals, branch ducts, main ducts, risers and apparatus. This procedure will help the designer coordinate the ductwork with the structural limitations of the building and other building systems and services.

On this preliminary layout, the designer should indicate the design airflows throughout the system. If a constant volume system is chosen, it will be the arithmetic sum of the CFM (l/s) of 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 to allow a summarization 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 air and exhaust air systems. This is not only to size the ductwork properly, but it also permits the designer to evaluate the effect of the total HVAC system design, balancing the proper proportions of supply air return air, exhaust air and outside makeup air.

E DUCT SIZING

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 Chapters 7 and 8. 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 efforts are completed, 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.

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. At this point, the designer must consider the additional space required beyond the bare sheet metal sizes for reinforcing and circumferential joints. In addition, consideration must be given to external insulation or duct liner which may be required, clearance for piping, conduit, light fixtures, etc., where applicable, and clearance for the removal of ceiling tiles.

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

F DESIGN METHODS

There is no design method that will automatically provide the most economical duct system for all conditions. Duct systems have been designed using one or more of the following methods or their variations (some of which are obsolete):

- Equal friction.
- Static regain.
- Extended plenum or semi-extended plenum (modification of equal friction or other design methods.)
- T-Method.
- Velocity reduction.
- Total pressure.
- Constant velocity.
- 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 the 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 Chapters 7 and 8 (Duct Sizing Procedures), along with the "semiextended 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. Traditionally used duct design methods follow.

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 air, return air and exhaust air duct systems. It normally is not used for higher pressure systems. With supply air duct systems, this design method "automatically" reduces air velocities in the direction of the airflow, thus reducing the possibility of generating noise (against the air flow in return or exhaust duct systems).

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

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 air 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 requirements 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 which offsets the friction loss 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 that 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 the oversized ducts that can occur at the ends of long branches, especially if one 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 coefticients for duct fittings found in Chapter 14 of this manual or in Chapter 32 of the ASHRAE—1989 Fundamentals Handbook, 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.**

3. Extended Plenums

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

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

A disadvantage is that 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 Chapters 7 and 8.

4. T-Method

The *T-Method* of duct sizing is a recently developed duct design optimization procedure that includes system initial costs and operating costs, energy costs, hours of operation, annual escalation, interest rates, etc. Manual procedures and equations may be found in Chapter 32 of the ASHRAE 1989 Fundamentals Handbook, but the method is best used with the proper computer software.

5. Seldom Used Methods

A. VELOCITY REDUCTION

An experienced designer who can use sound judgement in selecting arbitrary velocities is qualified to design a relatively simple duct system using the velocity reduction method. All others should not attempt to use this method except for estimating purposes unless the system has only 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 resultant pressure loss differences in the various sections of the duct system are not taken into account and balancing is attempted mainly by the use of good dampers at strategic locations.

B. 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.

C. CONSTANT VELOCITY

With adequate experience, many designers are able to select an optimum velocity that is used through out 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.

6. 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.

G 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, such as the ASHRAE Handbook of Fundamentals. In this section, some of the practical considerations in duct design which affect duct heat gain or loss are noted.

Consider first a conditioned air supply system with the air handling apparatus 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 running conditioned air 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 cost and operating cost. However, the designer must also realize if the velocities are reduced too much as a result of this, duct wall heat transfer increases, indicating that additional duct insulation might be required.

The use of additional insulation on duct work 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 the duct heat gain or loss.

H SOUND AND VIBRATION

With the design for the duct system 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 11 contains data and methods needed to make this determination. Some duct resizing may be required at this stage to incorporate the necessary acoustical treatment into the duct system design.

I PRESSURE CLASSIFICATION

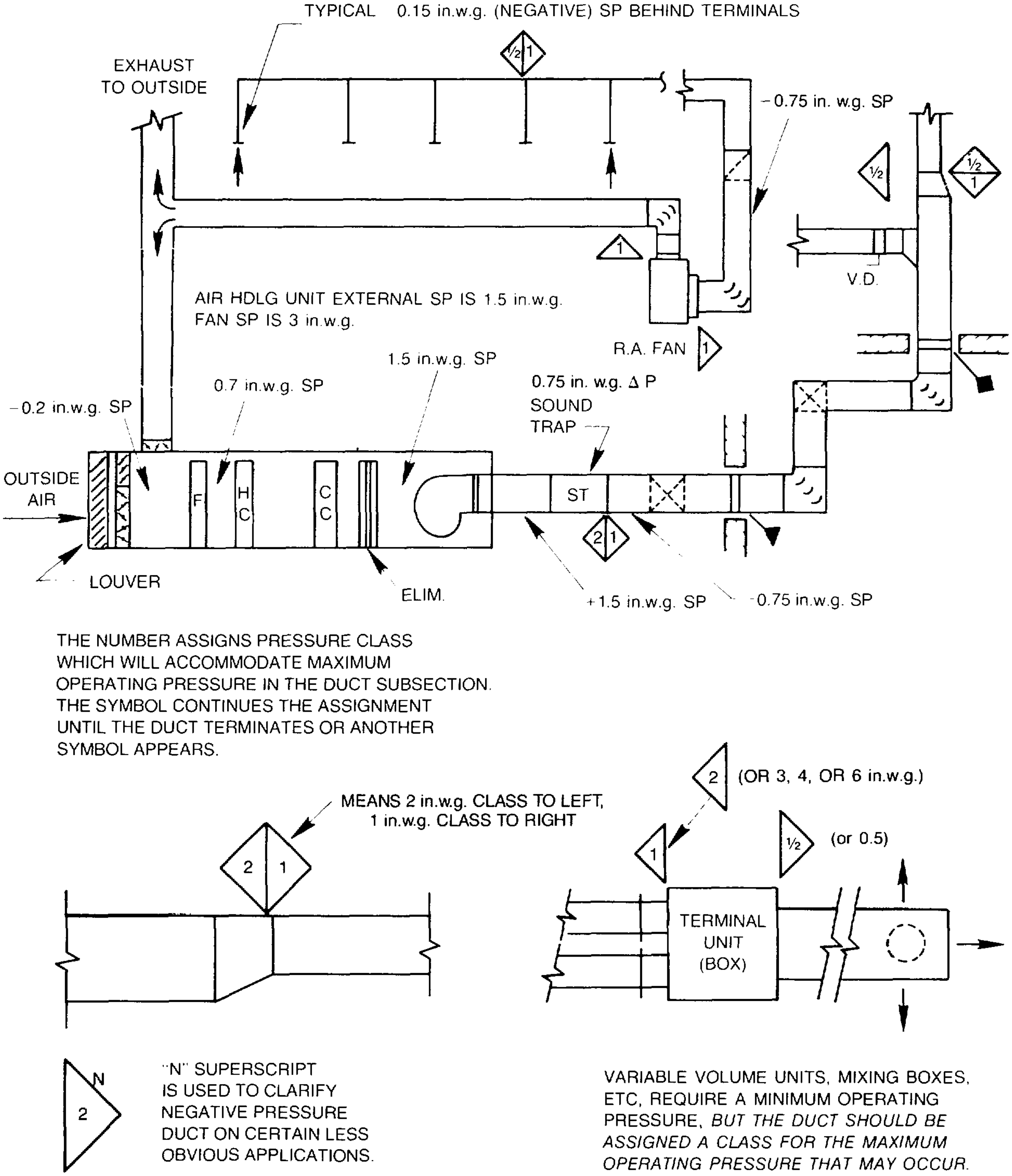
It is beneficial to all concerned to have the designer indicate all ductwork static pressure classification changes on the drawings. For clear interpretation of the requirements for ductwork and economical attainment of performance objectives, it is essential that the contract plans depict the portion of each duct system to be constructed for a particular static pressure classification (see Table 4-1). These static pressure rating changes are shown 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 and 8.

Special consideration must be given to the pressure classes of ductwork used for some 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 under minimum airflow conditions. Under these conditions, the maximum duct construction classification should remain the same

Table 4-1 HVAC DUCT PRESSURE-VELOCITY CLASSIFICATION

Static Pressure Class		Operating Pressure		Type of Pressure	Seal Class	Maximum Velocity	
U.S. Units in. w.g.	Metric Pa	U.S. Units in. w.g.	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

SAMPLE SITUATION: WITH A TERMINAL REQUIRING 0.15 in.w.g. STATIC PRESSURE, A BRANCH DAMPER REQUIRING 0.15 in.w.g. SP, DUCT DESIGNED FOR 0.1 in.w.g. (SP) LOSS per 100 ft. AND FITTING LOSSES EQUAL TO THE STRAIGHT DUCT LOSS, THE CIRCUIT CAN BE 100 LINEAL FEET LONG BEFORE 0.5 in.w.g. LOSS IS EXCEEDED.



AIR HDLG UNIT EXTERNAL SP IS 1.5 in.w.g. FAN SP IS 3 in.w.g.

THE NUMBER ASSIGNS PRESSURE CLASS WHICH WILL ACCOMMODATE MAXIMUM OPERATING PRESSURE IN THE DUCT SUBSECTION. THE SYMBOL CONTINUES THE ASSIGNMENT UNTIL THE DUCT TERMINATES OR ANOTHER SYMBOL APPEARS.

"N" SUPERSCRIPT IS USED TO CLARIFY NEGATIVE PRESSURE DUCT ON CERTAIN LESS OBVIOUS APPLICATIONS.

VARIABLE VOLUME UNITS, MIXING BOXES, ETC, REQUIRE A MINIMUM OPERATING PRESSURE, *BUT THE DUCT SHOULD BE ASSIGNED A CLASS FOR THE MAXIMUM OPERATING PRESSURE THAT MAY OCCUR.*

Figure 4-1 DUCT PRESSURE CLASS DESIGNATION (U.S. UNITS)

throughout the supply duct system upstream of the VAV boxes.

Special consideration also must be given to emergency mode operations such as when smoke control systems go into operation or fire dampers close against full system airflow. Select duct pressure classifications that will handle the sudden pressure changes without damage to the duct distribution system.

J DUCT LEAKAGE

Until recent joint research projects between ASHRAE and SMACNA on duct leakage, HVAC system designers arbitrarily established percentage leakage rates for duct systems, that were impossible to attain by the installing contractor. The anticipated amount of duct system leakage may now be calculated once the duct pressure and seal classifications are determined. These leakage rates, in terms of cfm/100 sq. ft. (l/s per m^2) of duct surface may also be expressed in 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 which varies with the average pressure of the system, must be added to the total airflow capacity of the HVAC system fans(s).

K FAN SIZING

With the various elements of the HVAC duct system selected, the duct system laid out, and the sizing finalized, the designer now must calculate the total pressure of the systems which the fan(s) must overcome. In Chapters 7 and 8, there is a detailed description of 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 Chapter 9, enable the designer to sum up the total pressure requirements for the fan(s). Estimated system air leakage must be added to the system airflow at this time.

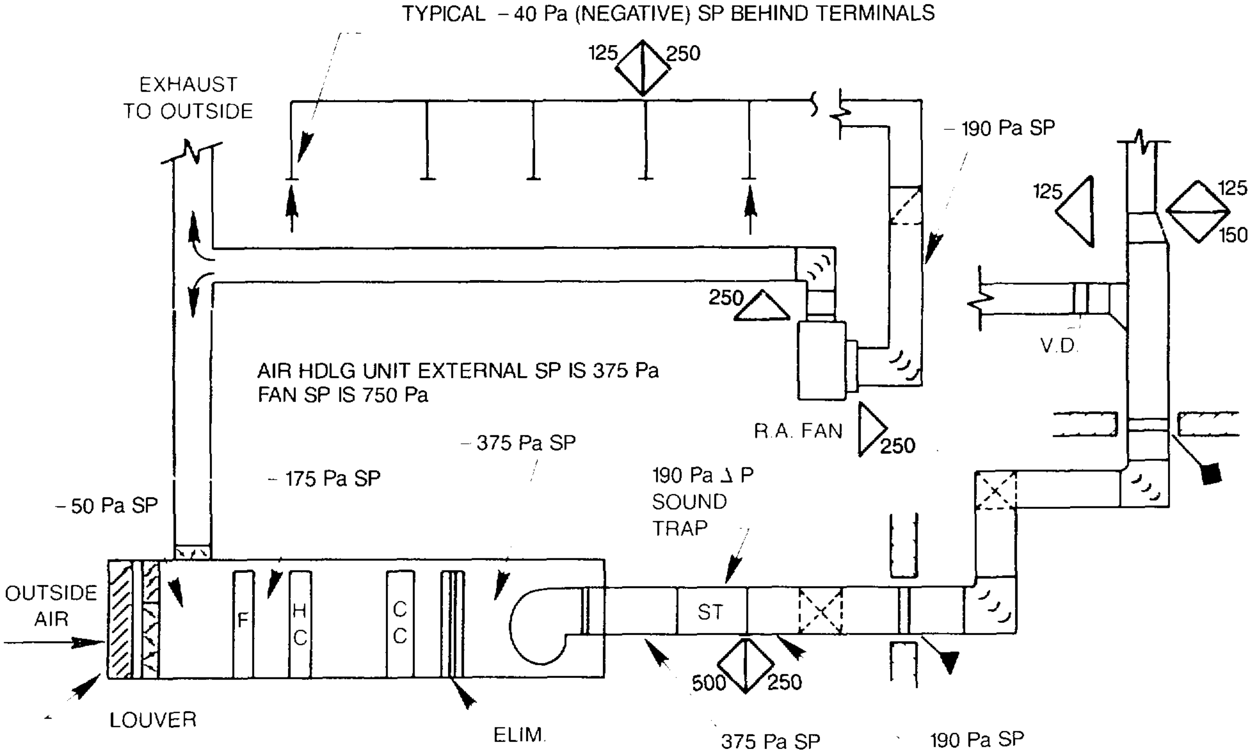
L TESTING, ADJUSTING & BALANCING (TAB)

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 that the TAB personnel can perform their function properly to determine the total system airflow with a reasonable degree of accuracy. This same thought also applies to critical branch ducts of the supply air system. This subject will be discussed in more detail in Chapter 10; however, it is important that the designer indicate all necessary balancing dampers and devices on the drawings.

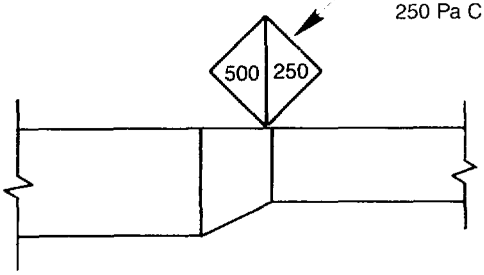
M FINAL DESIGN DOCUMENTS

Assuming that all of the steps mentioned earlier in this chapter have been followed, the final plans can be drawn and the specifications completed. Using the pre-liminary design (usually a single line drawing) as a guide, a double line duct system is shown on the final mechanical drawings, employing the symbols commonly used for ventilation and air conditioning (see 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, etc., so that the installed system will function within the design parameters and meet applicable code requirements.

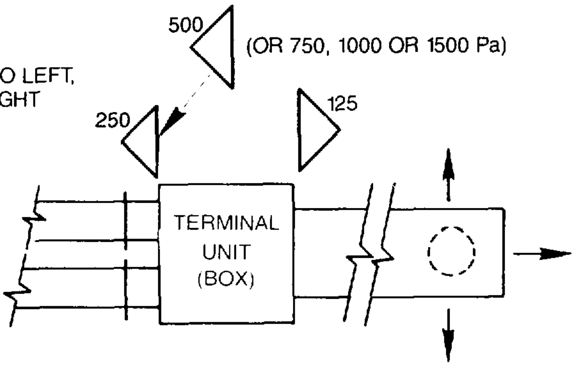
SAMPLE SITUATION: WITH A TERMINAL REQUIRING 40Pa STATIC PRESSURE. A BRANCH DAMPER REQUIRING 40 Pa SP, DUCT DESIGNED FOR 0.8Pa/m SP LOSS AND FITTING LOSSES EQUAL TO THE STRAIGHT DUCT LOSS. THE CIRCUIT CAN BE 30 METRES LONG BEFORE 125 Pa LOSS IS EXCEEDED.

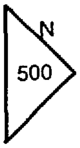


THE NUMBER ASSIGNS PRESSURE CLASS WHICH WILL ACCOMMODATE MAXIMUM OPERATING PRESSURE IN THE DUCT SUBSECTION. THE SYMBOL CONTINUES THE ASSIGNMENT UNTIL THE DUCT TERMINATES OR ANOTHER SYMBOL APPEARS.



MEANS 500 Pa CLASS TO LEFT, 250 Pa CLASS TO RIGHT

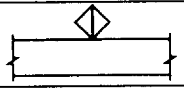
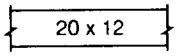
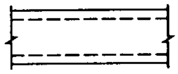
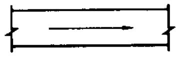
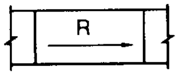
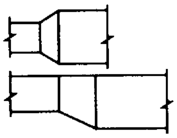
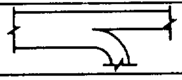
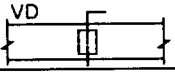
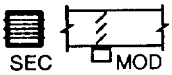
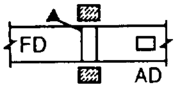
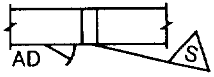
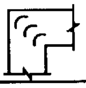
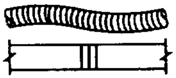
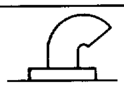




"N" SUPERSCRIPT IS USED TO CLARIFY NEGATIVE PRESSURE DUCT ON CERTAIN LESS OBVIOUS APPLICATIONS.

VARIABLE VOLUME UNITS, MIXING BOXES, ETC. REQUIRE A MINIMUM OPERATING PRESSURE, *BUT THE DUCT SHOULD BE ASSIGNED A CLASS FOR THE MAXIMUM OPERATING PRESSURE THAT MAY OCCUR.*

Figure 4-2 DUCT PRESSURE CLASS DESIGNATION (METRIC UNITS)

SYMBOL MEANING	SYMBOL
POINT OF CHANGE IN DUCT CONSTRUCTION (BY STATIC PRESSURE CLASS)	
DUCT (1ST FIGURE, SIDE SHOWN 2ND FIGURE, SIDE NOT SHOWN)	
ACOUSTICAL LINING DUCT DIMENSIONS FOR NET FREE AREA	
DIRECTION OF FLOW	
DUCT SECTION (SUPPLY)	
DUCT SECTION (EXHAUST OR RETURN)	
INCLINED RISE (R) OR DROP (D) ARROW IN DIRECTION OF AIR FLOW	
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)	
WYE JUNCTION	
VOLUME DAMPER MANUAL OPERATION	
AUTOMATIC DAMPERS MOTOR OPERATED	
ACCESS DOOR (AD) ACCESS PANEL (AP)	
FIRE DAMPER: SHOW —◀ VERTICAL POS. SHOW —◆ HORIZ. POS.	
SMOKE DAMPER △S	
FIRE & SMOKE DAMPER - ▲ SMOKE DAMPER - △S RADIATION DAMPER - △R	
TURNING VANES	
FLEXIBLE DUCT FLEXIBLE CONNECTION	
GOOSENECK HOOD (COWL)	
BACK DRAFT DAMPER	BDD 

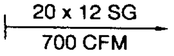
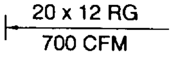
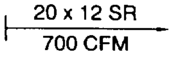
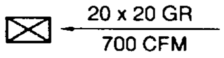
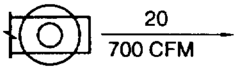
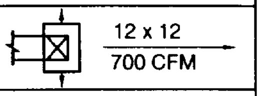
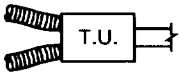
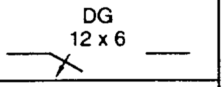
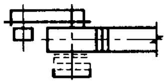
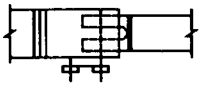
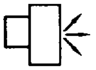
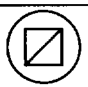
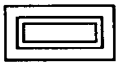
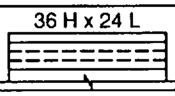
SYMBOL MEANING	SYMBOL
SUPPLY GRILLE (SG)	
RETURN (RG) OR EXHAUST (EG) GRILLE (NOTE AT FLR OR CLG)	
SUPPLY REGISTER (SR) (A GRILLE + INTEGRAL VOL. CONTROL)	
EXHAUST OR RETURN AIR INLET CEILING (INDICATE TYPE)	
SUPPLY OUTLET. CEILING, SQUARE (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION	
SUPPLY OUTLET. CEILING,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

Figure 4-3 SYMBOLS FOR VENTILATION & AIR CONDITIONING (U.S. and/or Metric Units)

CHAPTER 5

DUCT DESIGN FUNDAMENTALS

A 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, turning vanes, etc. 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 Section A 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").

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 9.

2. System Curves

At a fixed volume air flow rate through a given air distribution system, a corresponding pressure loss or resistance to this flow will exist. If the flow rate is changed, the resulting pressure loss or resistance to flow also will change. The relationship governing this change is given by the following system equation:

Equation 5-1

$$\frac{\text{Pressure}_2}{\text{Pressure}_1} = \left(\frac{\text{airflow rate}_2}{\text{airflow rate}_1}\right)^2$$

Where:

Pressure = in. w.g. (Pa)
Airflow rate = cfm (l/s)

Typical plots of the resistance to flow versus the airflow rate establish the *system curves* for three different and arbitrary fixed systems, (A, B and C), illustrated in Figure 5-1. For a fixed system, an increase or decrease in the system airflow rate volume will increase or decrease the system resistance along the given system curve only.

Refer to System Curve A on Figure 5-1. Assume a system design point at 100 percent volume and 100 percent resistance. If the airflow rate volume is increased to 120 percent of design volume, the system resistance will increase to 144 percent of the design resistance in accordance with the system equation. A further increase in volume results in a corresponding increase in system pressure. A decrease in volume flow to 50 percent of design airflow 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 HVAC systems.

3. System Curve/Fan Curve Interaction

If the system curve, composed of the resistance to flow of the system and the appropriate "System Effect Factors," (discussed later in this section) has been accurately determined, then it is assumed that the fan selected will develop the necessary pressure to meet the system requirements at the designed airflow (cfm or l/s).

The point of intersection of the system curve and the fan performance curve determines the actual airflow volume. If the system resistance has been accurately determined and the fan properly selected, their performance curves will intersect at the design airflow. (See Figure 5-2). The normalized System Curve A from Figure 5-1 has been plotted with a normalized fan performance curve. The 100 percent design airflow volume of the system curve was arbitrarily selected to intersect at 60 percent of the free delivery airflow volume of the fan.

The airflow rate volume through the system in a given installation may vary from changes in the system re-

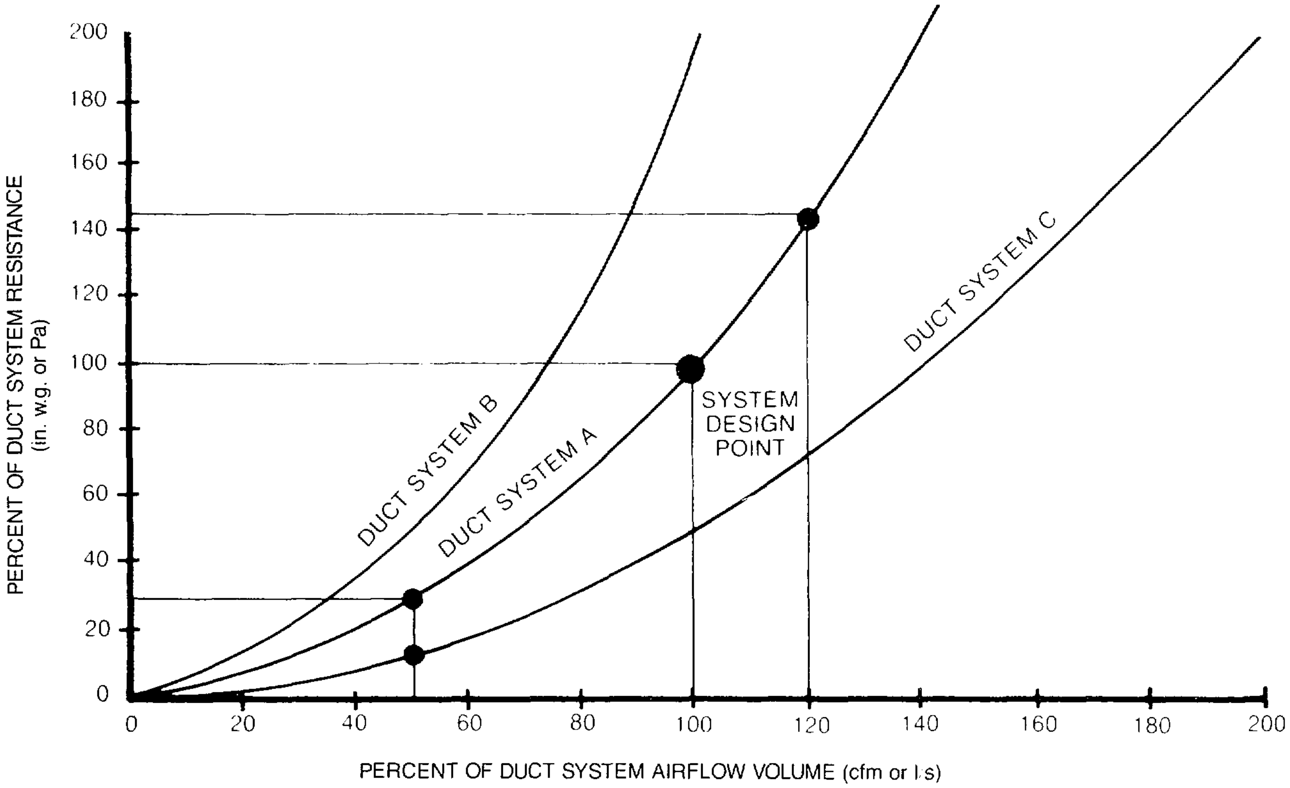


Figure 5-1 NORMALIZED DUCT SYSTEM CURVES (1)

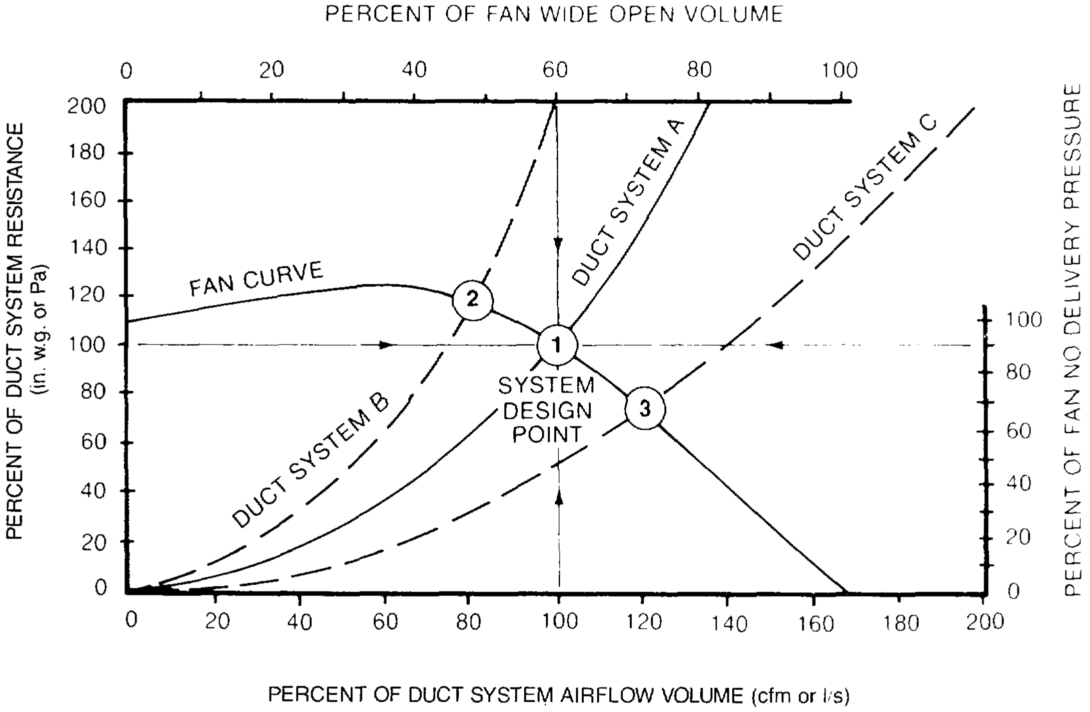


Figure 5-2 INTERACTION OF SYSTEM CURVES AND FAN CURVE (1)

sistance, usually from fan dampers, duct dampers, mixing boxes, terminal units, etc. Referring to Figure 5-2, the airflow volume rate may vary from 100 percent design airflow (Point 1, Curve A), to approximately 80 percent of the design airflow by increasing the resistance to flow, thus changing the system curve characteristic to Curve B. This results in fan operation at Point 2 (the intersection of the fan curve and the new System Curve B). Similarly, the airflow rate can be increased to approximately 120 percent of the design airflow volume by decreasing the resistance to flow, 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).

To review; when system losses have been estimated accurately, when the duct systems have been fabricated and installed exactly as shown on the drawings with specified components, then the design airflow volume can be expected as illustrated in Figure 5-3 at Point 1.

However, when the duct systems have not been estimated accurately or installed as shown, a higher pressure loss causes the fan to operate at Point 2 of Figure 5-3, and a lower system pressure loss at Point 3. Again note that the interaction of the installed duct system curve and the fan curve from actual operating conditions determine the duct system airflow volume rate.

4. Fan Speed Change Effects

A change in fan speed will alter the airflow volume rate through a given system as shown by Equation 5-2:

Equation 5-2

$$\frac{\text{Airflow rate}_2}{\text{Airflow rate}_1} = \frac{\text{Fan Speed}_2}{\text{Fan Speed}_1}$$

Where:

Airflow rate = cfm (l/s)
Fan Speed = rpm (rad/s)

Figure 5-4 illustrates the increase in system airflow when the fan speed is increased 10 percent. *Any change in fan speed creates a new fan curve.* The system operating point then moves along the system curve from Point 1 to Point 2. **The 10 percent increase in airflow extracts a severe fan power penalty. According to the fan laws, the fan power output must then increase 33 percent.**

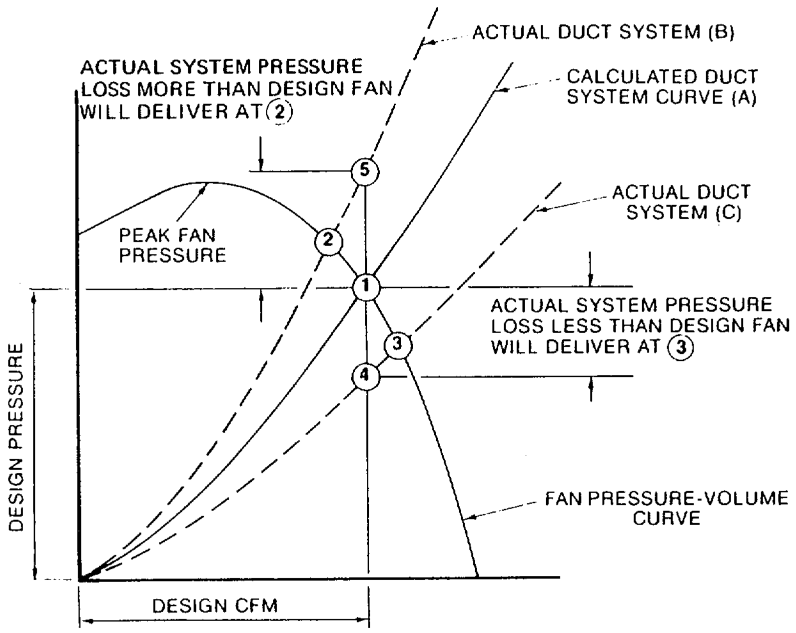


Figure 5-3 DUCT SYSTEM NOT AT DESIGN POINT (1)

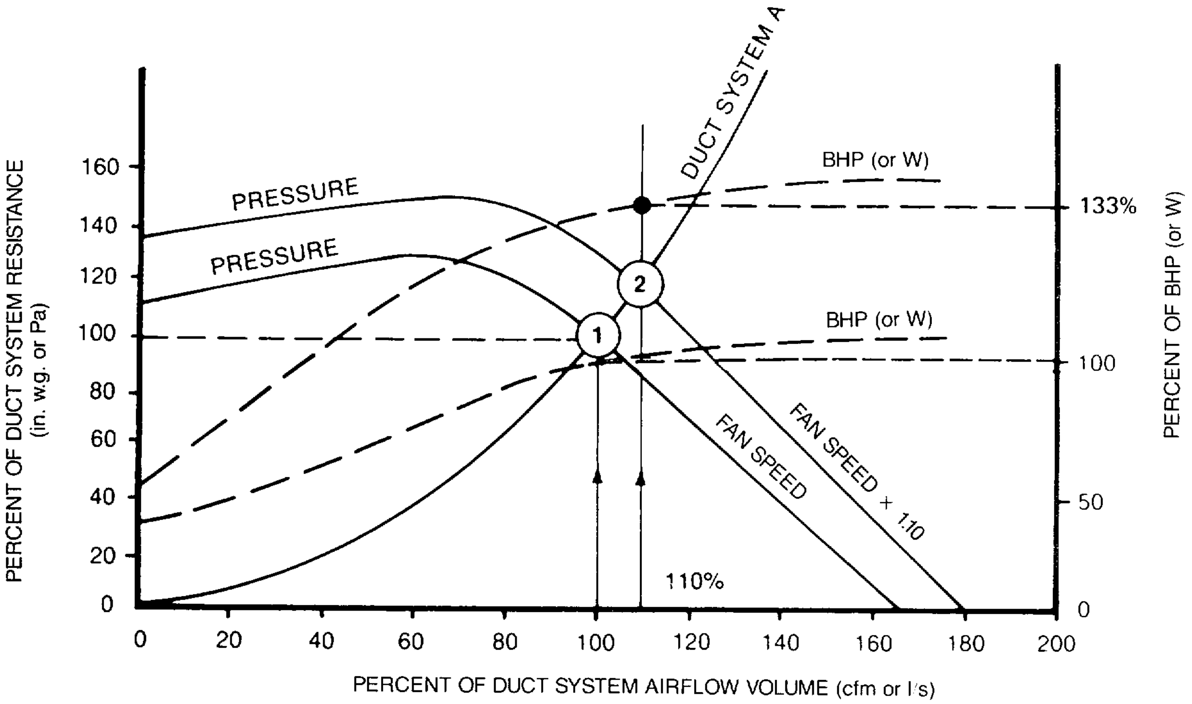


Figure 5-4 EFFECT OF 10 PERCENT INCREASE IN FAN SPEED (1)

Equation 5-3

$$\frac{\text{Fan Power}_2}{\text{Fan Power}_1} = \left(\frac{\text{Fan Speed}_2}{\text{Fan Speed}_1}\right)^3$$

Where:

Fan Power = HP (kW or W)
Fan Speed = rpm (rad/s)

Example 5-1 (U.S.)

A 10 HP fan runs at 500 rpm.

Calculate the HP at 550 rpm.

Solution

Using Equation 5-3:

$$\text{Fan Power}_2 = 10\left(\frac{550}{500}\right)^3 = 13.31 \text{ HP}$$

Example 5-1 (Metric)

A 7.5 kW fan runs at 50 rad/s.

Calculate the fan power at 55 rad/s.

Solution

Using Equation 5-3:

$$\text{Fan Power}_2 = 7.5\left(\frac{55}{50}\right)^3 = 9.98 \text{ kW}$$

Frequently, the extra horsepower (Watts) is not available from the existing fan motor, and the motor power wiring is too small to add a larger motor.

This fact is often startling to the system designer who finds the system short of air. Only 10 percent more air is needed, but the selected motor horsepower is not capable of a 33 percent increase in load. The increased power requirements are the result of increased work done. The greater volume flow rate of air moved by the fan against the resulting higher system resistance to the flow, causes increased work to be done. In the same system, the fan power increases as the cube of the speed ratio, and fan efficiency remains the same at all points on the same system curve. (See HVAC Fan Equations in Chapter 14.)

Increasing the fan speed also may create problems for the fan by putting it and possibly the ductwork into a higher pressure classification. Be sure to review the fan rating table for pressure class limits or contact the fan manufacturer to determine if the fan speed may be increased safely.

5. Air Density Effects

The resistance of a duct system is dependent on the density of the air (or gas) flowing through the system. Air at standard conditions has a density of 0.075 lb/cu.ft. (1.204 kg/m^3). Figure 5-5 illustrates the effect on the fan performance of a density variation from this standard value.

The fan pressure and horsepower vary directly as the ratio of the gas density at the fan inlet to standard density. This density ratio must always be considered when selecting fans from manufacturers' catalogs or curves (fan airflow volume is constant).

Equation 5-4

$$\text{Density Ratio} = \frac{d_2}{d_1} = \frac{TP_2}{TP_1} = \frac{\text{Fan Power}_2}{\text{Fan Power}_1}$$

Where:

d = Density—lb/cu.ft. (kg/m^3)
TP = Total pressure—in. w.g. (Pa)
Fan Power = bhp (kW)

6. "Safety Factor" Cautions

System designers sometimes add "Safety Factors" to their estimate of the system resistance to compensate for unknown field conditions. These "Safety Fac-

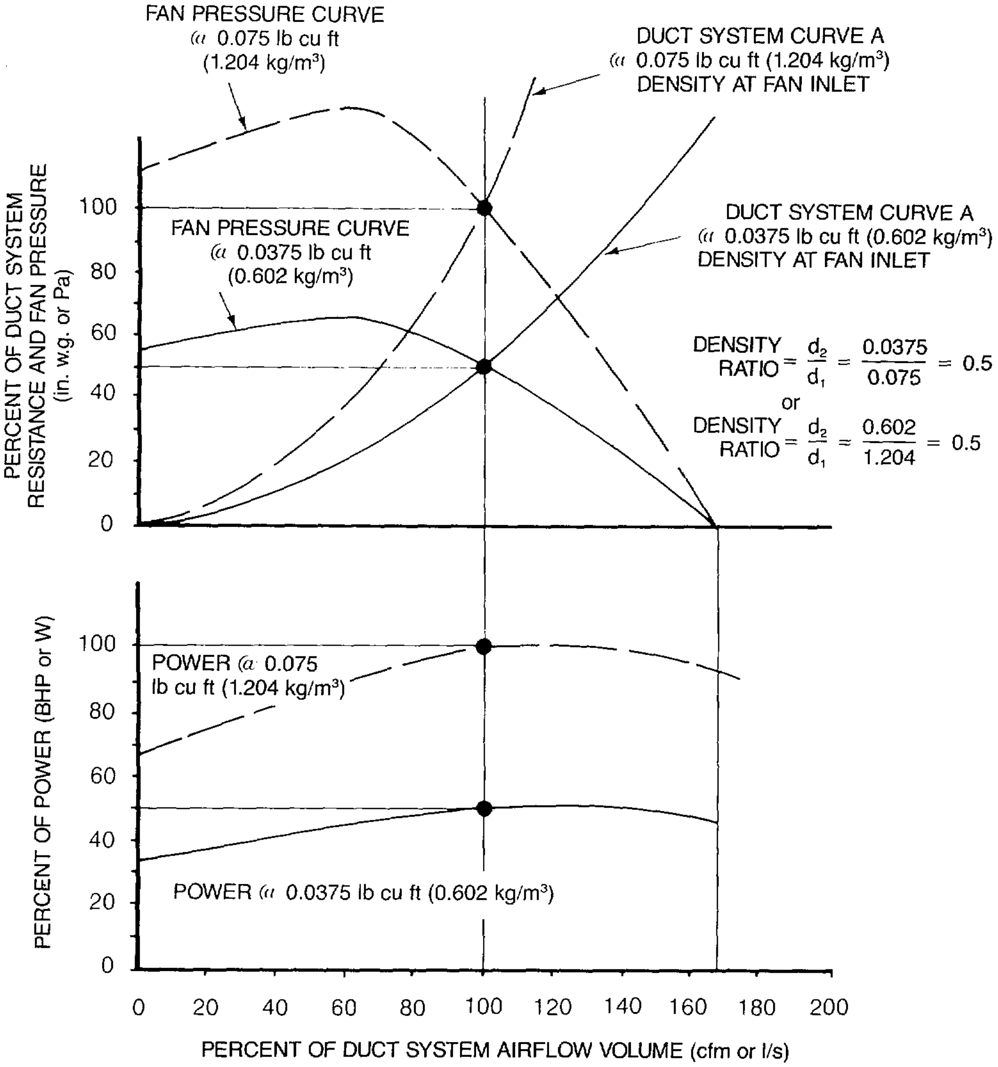


Figure 5-5 DENSITY EFFECT (1)

tors" may compensate for resistance losses that were overlooked and the actual system will deliver design flow (Point 1, Figure 5-3). Occasionally, however, the estimated system resistance, including the "Safety Factors," is 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 flow rate is less than design (Point 4).

This result may not necessarily be an advantage because the fan will usually be operating at a less efficient point on the performance curve and may require more horsepower than at design flow. Under these conditions, it may be necessary to reduce the fan speed or to adjust a damper to increase the actual system resistance (Curve C) to the original design characteristic (Curve A).

B OTHER FACTORS AFFECTING DUCT SYSTEM PRESSURES

1. System Effect

A "derating" of the HVAC system fan, called "System Effect" must be taken into account by the system designer 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. It will be necessary, therefore, to apply judgement based on actual experience in applying the System Effect Factors.

Figure 5-6 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 5-6, 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—"Fan—Duct Connection Pressure Losses" contains the necessary data, charts and tables needed to determine the System Effect Factors required by duct connections to HVAC system fans. The System Effect Factor is given in inches of water gauge (Pascals) and may be added to the total system pressure losses as shown on Figure 5-6. However, System Effect can *not* 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 should derate the HVAC system supply fan by *deducting* the System Effect Factor from the fan rated capacity (in. w.g. or Pa).

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, but for centrifugal fans, it may be necessary to calculate the inlet velocity. The necessary inlet dimensions usually are included in the fan catalog.

2. Wind Effect

With few exceptions, building intakes and exhausts cannot be located or oriented for a prevailing wind to assure HVAC system operation. Wind can assist or hinder supply air and exhaust air fans depending on their position on the building, but even in locations with a predominant wind direction, the ventilating system must perform adequately for all other directions.

Airflow through a wall opening results from positive or negative external and internal pressures. Such differential pressures may exceed 0.5 in. w.g. (125 Pa)

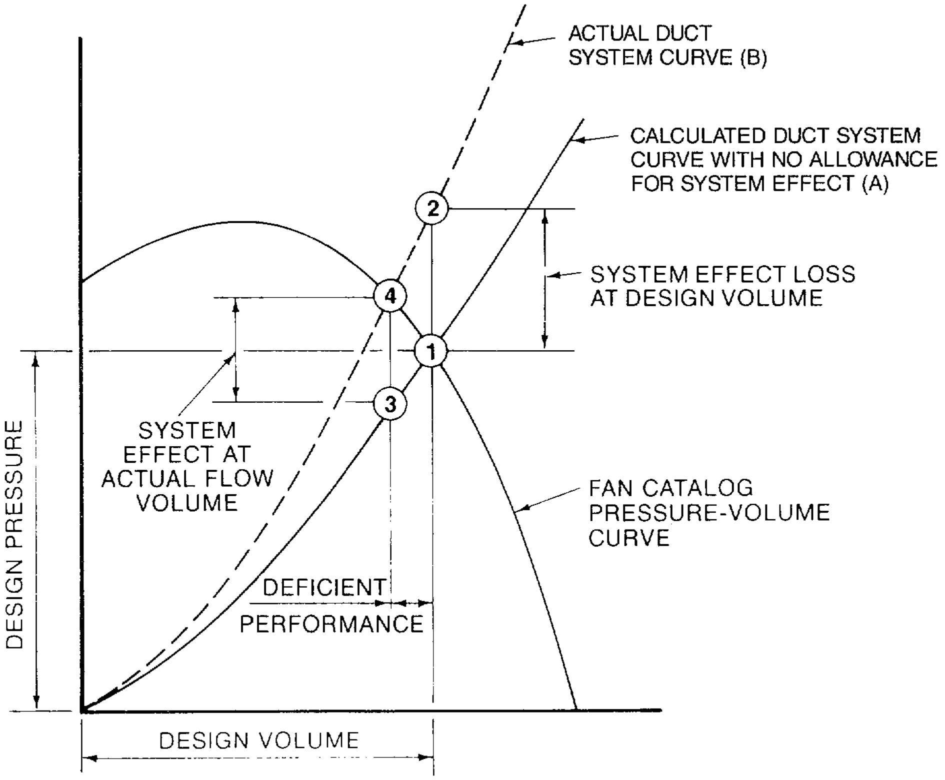


Figure 5-6 CHANGES FROM "SYSTEM EFFECT" (1)

during high winds. Supply and exhaust systems, and openings, dampers, louvers, doors, and windows make the building flow conditions too complex for most 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 w.g. (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 and other sides, Clarke (1967) when measuring HVAC Systems operating at 1 to 1.5 in. [w.g. (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. Such changes in flow rate can cause noise at the supply outlets and drafts in the space served.

For mechanical systems, the wind can be thought of as producing a pressure in series with a system fan, either assisting or opposing it (Houlihan 1965).

Where system stability is essential, the supply air and exhaust air systems must be designed for higher [pressures about 3 to 4 in. w.g. (750 to 1000 Pa)] to minimize unacceptable variations in flow rate. 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 pressure adequate to minimize wind effects.

b) Controls to regulate flow rate or pressure or both.

c) Separate supply and exhaust systems to serve each building area requiring control or balance.

d) Doors (possibly self-closing) 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.

3. 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 normal stack effect, is caused by the air in the building being warmer and less dense than the outside air. "Normal stack effect" is greater when outside temperatures are low and when buildings are taller. However, "normal 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 5-5

$$\Delta p = k_s \left(\frac{1}{T_o} - \frac{1}{T_i}\right) h \text{ (or) } \Delta p\ h = k_s \left(\frac{1}{T_o} - \frac{1}{T_i}\right)$$

Where:

Δp = pressure difference, in w.g. (Pa)
T_o = absolute temperature of outside air, °R (K)
T_i = 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°F (−18°C) and an inside temperature of 70°F (21°C), the maximum pressure difference due to stack effect would be 0.22 in. w.g. (55 Pa). This means that at the top of the building, a shaft would have a pressure of 0.22 in. w.g. (55 Pa) greater than the outside pressure. At the bottom of the shaft, the shaft would have a pressure of 0.22 in. w.g. (55 Pa) less than the outside pressure. Figure 5-7 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 pressures 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 both normal and reverse stack effect is illustrated in Figure 5-8. In this case, the pressure difference expressed in Equation 5-5 refers to the pressure difference between the shaft and the outside of the building.

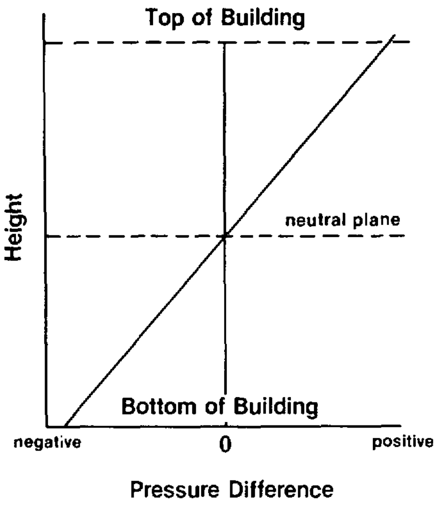
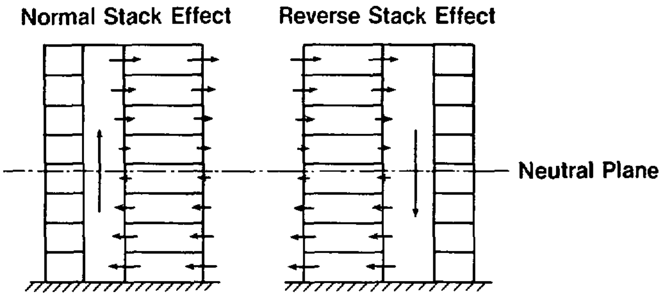


Figure 5-7 PRESSURE DIFFERENCE DUE TO NORMAL STACK EFFECT (2)

Figure 5-9 can be used to determine the pressure difference due to stack effect. For normal stack effect, Δp/h is positive and the pressure difference is positive, above the neutral plane and negative below it. For reverse stack effect, Δp/h is negative and the pressure difference is negative above the neutral plane and positive below it.

In unusally 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 was the cold column of air, and other shafts within the building were the warm columns of air.



Note: Arrows Indicate Direction of Air Movement

Figure 5-8 AIR MOVEMENT DUE TO NORMAL AND REVERSE STACK EFFECT (2)

If the 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. McGuire and Tamura (1975) provide methods for calculating the location of the neutral plane for some vented conditions.

C SYSTEM PRESSURE CHANGES

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 datum line.

At any cross-section, the total pressure (TP) is the sum of the static pressure (SP) and the velocity pressure (V_p).

Equation 5-6

$$TP = SP + V_p$$

where:

TP = Total Pressure—in. w.g. (Pa)
SP = Static Pressure—in. w.g. (Pa)
V_p = Velocity Pressure—in. w.g. (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/cu. ft. (1.2041 kg/m^3), and corrections are necessary for significant differences in density due to altitude 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.*

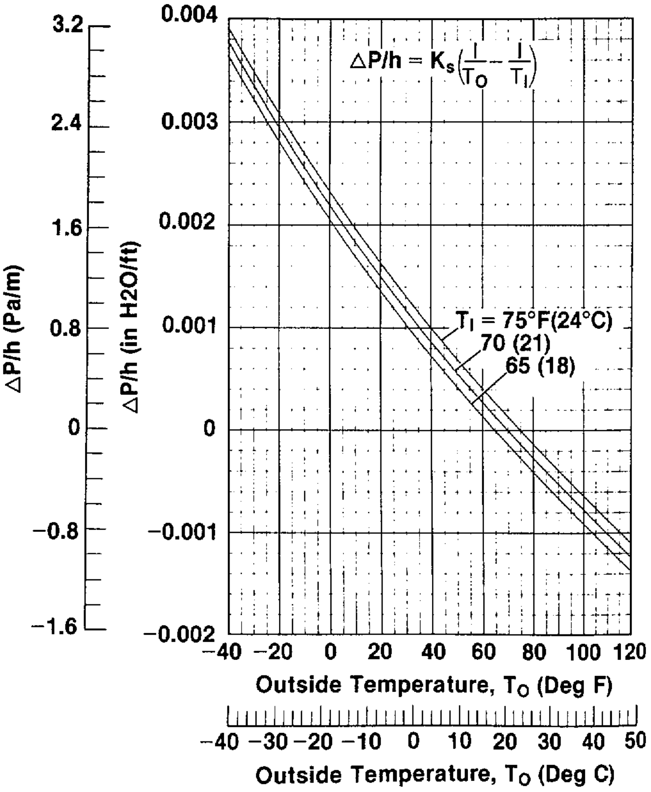


Figure 5-9 PRESSURE DIFFERENCE DUE TO STACK EFFECT (2)

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).

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, etc.* 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.

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. 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, as may be seen 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 a 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.

6. Entrance Fittings

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 referenced 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, the static pressure is negative with respect to the atmospheric pressure. *However, velocity pressure is always a positive value.*

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, but *total pressure* determines the actual amount 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).

8. Fan Pressures

The total resistance to airflow is noted by ΔTP_{sys} in Figure 5-10. Since the prime mover is a vane-axial fan, the inlet and outlet velocity pressures are equivalent; i.e. $\Delta TP_{sys} = \Delta SP_{sys}$. 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 $\Delta TP_{sys} \neq \Delta SP_{sys}$. If one needs to know the static pressure requirements of a centrifugal fan, and the total pressure requirements are known, the following relationship may be used:

Equation 5-7

$$\text{Fan SP} = TP_d - TP_s - V_{pd}$$
$$(\text{or as SP} = TP - V_p)$$
$$\text{Fan SP} = SP_d - TP_s$$

where the subscripts "d" and "s" refer to the discharge and suction sections, respectively, of the fan. 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.

9. Return Air System Pressures

There are many persons in the HVAC industry (and elsewhere) that believe that return air in a duct system is "sucked back" by the fan; therefore the ductwork and fittings do not need the use of good design practices (i.e. no turning vanes for mitred elbows, the lack of smooth air flow into the fan inlet, the use of "panned" joists in residential systems, etc.). How *wrong* they are!

A diagram is shown in Figure 5-11 of a simple return air system. Converting to absolute pressures, an atmospheric pressure of 14.7 psi or 407 in. w.g. (101,325 pascals) 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. w.g.—100,575 Pascals) at the system fan inlet. The total pressure drop of 3 in.w.g. (750 Pa) could be reduced substantially if the 90° mitered elbows had turning vanes and the fan inlet connection was better designed. In reality, a return air or exhaust air duct behaves exactly

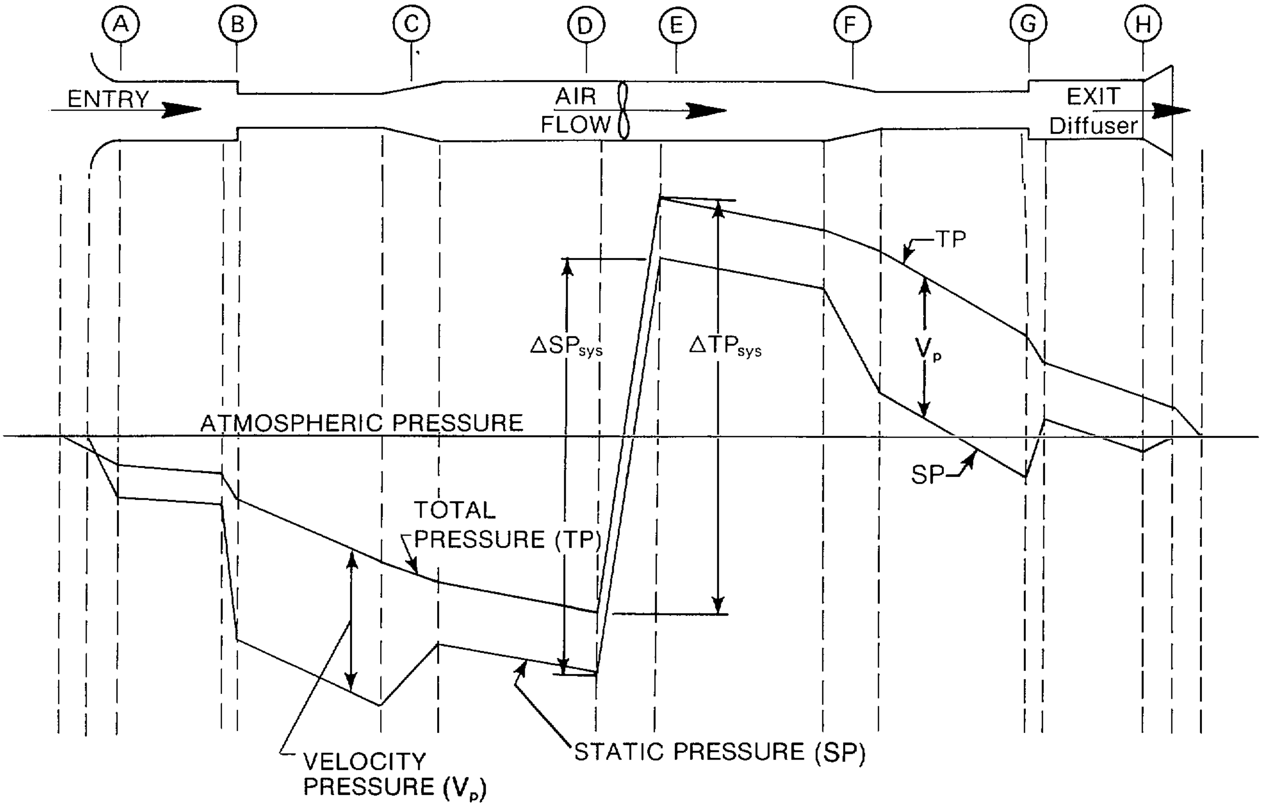


Figure 5-10 PRESSURE CHANGES DURING FLOW IN DUCTS

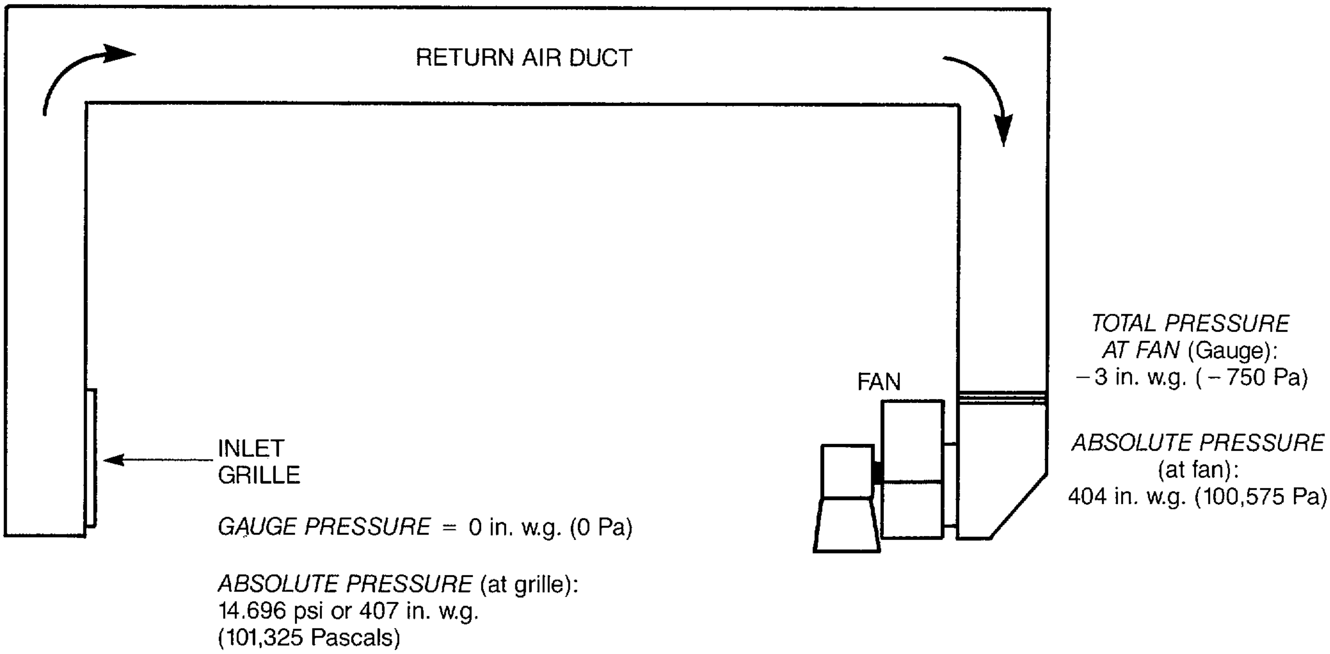


Figure 5-11 RETURN AIR DUCT EXAMPLE

as a supply air duct with atmospheric pressure *pushing* the air to the lower pressure area created by the fan suction.

D STRAIGHT DUCT LOSSES

1. Duct Friction 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 14-1 and 14-2) in Chapter 14. They are based on standard air with a density of 0.075 lb/cu. ft. (1.204 kg/m^3) flowing through average clean round galvanized metal ducts with beaded slip couplings on 48 inch (1220 mm) centers, equivalent to an absolute roughness of 0.0003 feet (0.09 mm). The previous duct friction loss charts were based on 30 inch (760 mm) joints and an absolute roughness of 0.0005 (0.15 mm), and most computer software programs and duct calculators still contain these older values. The SMACNA Duct Design Calculators (both U.S. and Metric) contain the newer data.

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°F to 140°F (10°C to 60°C) and up to 2000 feet (600m) altitude. Figure 14-5 and Tables 14-26 and 14-32 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/or altitude. The actual air volume (cfm or l/s) is used to find the duct friction loss using Figures 14-1 and 14-2. This loss is multiplied by the correction factor(s) to obtain the adjusted duct friction loss.

2. Circular Equivalents

HVAC duct systems usually are sized first 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 14-2 and 14-3 in Chapter 14 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 in. x 26 in. (2030 mm x 660 mm) duct is required, it will be twice that of a 40 in. x 13 in. (1015 mm x 330 mm) that has a circular equivalent of 24 inches (610 mm) diameter or 2 x 24 = 48 inches (1220 mm) diameter.

Rectangular ducts should not be sized directly from actual duct cross-sectional areas. Instead, Tables 14-2 and 14-3 must be used, or the resulting rectangular duct sizes will be smaller creating greater duct velocities for a given airflow.

E DYNAMIC LOSSES

Wherever turbulent flow is present, brought about by sudden changes in the direction or magnitude of the velocity of the air flowing, 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 excess of 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 crosssectional area* of the duct.

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 Chapter 14. *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 14-6 and 14-7, which show 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 5-8

$$TP = C \times V_p$$

Where:

TP = Total Pressure loss (in w.g. or Pa)
C = Fitting Loss coefficient
V_p = Velocity Pressure (in. w.g. or Pa)

The velocity pressure (Vp) used for rectangular duct fittings must be obtained from the velocity (V) obtained by using the following equation:

Equation 5-9 (U.S.)

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

Where:

V = Velocity (fpm)
Q = Airflow (cfm)
A = Cross-sectional Area (sq. ft.)

Equation 5-9 (Metric)

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

Where:

V = Velocity (m/s)
Q = Airflow (m^3/s), 1000 l/s = 1 m^3/s
A = Cross-sectional Area (m^2)

(or)

$$V = \frac{1000\,Q}{A}$$

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_o" for orifice area, etc.

Velocity pressure (V_p) may be calculated from Equation 5-10 or obtained from Tables 14-6 and 14-7 in Chapter 14.

Equation 5-10 (U.S.)

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

Equation 5-10 (Metric)

$$V_p = 0.602\,V^2$$

Where:

V_p = Velocity Pressure (in. w.g. or Pa)
V = Velocity (fpm or m/s)

Example 5-2 (U.S.)

An elbow in a 24 in. x 20 in. duct conveying 7000 cfm has a loss coefficient (C) of 0.40. Find the elbow pressure loss.

Solution

Using Equation 5-9:

$$V = \frac{Q}{A} = \frac{7000}{24 \times 20/144} = 2100 \text{ fpm}$$

Using Equation 5-10:

$$V_p = \left(\frac{2100}{4005}\right)^2 = 0.275 \text{ in. w.g.}$$

Using Equation 5-8:

$TP = C \times V_p = 0.40 \times 0.275$
$TP = 0.11$ in. w.g.
(Elbow pressure loss)

Example 5-2 (Metric)

An elbow in a 600 mm × 500 mm duct conveying 3500 l/s has a loss coefficient (C) of 0.40. Find the elbow pressure loss.

Solution

Using Equation 5-9:

(3500 l/s = 3.5 m^3/s), (600 mm = 0.6 m), (500 mm = 0.5 m)

$$V = \frac{Q}{A} = \frac{3.5}{0.6 \times 0.5} = 11.67 \text{ m/s}$$

(or)

$$V = \frac{1000Q}{A} = \frac{1000 \times 3500}{600 \times 500}$$

$V = 11.67$ m/s

Using Equation 5-10:

$V_p = 0.602 \times (11.67)^2 = 81.99$ Pa (Use 82)

Using Equation 5-8:

$TP = C \times V_p = 0.40 \times 82$
$TP = 32.8$ Pa

2. Pressure Losses in Elbows

Dynamic-loss coefficients for elbows (see Table 14-10) are nearly independent of the air velocity and are affected by the roughness of the duct walls only in the case of the bends. In tables used in other texts, the dynamic losses often are grouped with the friction losses to facilitate design calculations by determining bend losses in terms of additional equivalent lengths of straight duct or inches of water. However, the elbow loss coefficients in Table 14-10 are used with the duct velocity pressure to calculate the "total pressure" loss of each fitting. The additional duct friction loss (if any) of 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 use of the sum of the losses for the individual elbows. Actually, 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 14-10.

Loss coefficients for some elbows with angle bends other than 90° may be computed from the table in Note 1 on page 14-19. Loss coefficients for elbows discharging air directly into a large space are higher than those given for elbows within duct systems (see Table 14-16 figure E).

A. 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 14-10, figure F). However, most installations do not have ample 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 5-3 (U.S.)

A 48 in. (H) x 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.

Solution:

Using Figure 5-12 and Table 14-10, figure G:

a) $\frac{R}{W} = \frac{6}{24} = 0.25$;

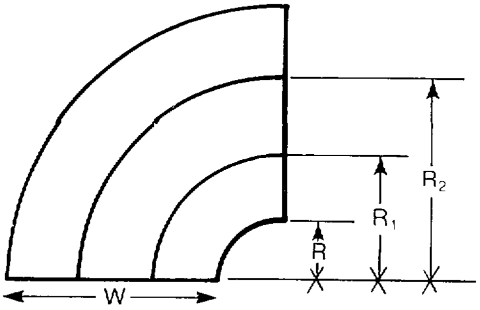
From Table 14-10. Figure G for two splitter vanes, CR = 0.585

$$R_1 = \frac{R}{CR} = \frac{6}{0.585} = 10.26 \text{ inches}$$

$$R_2 = \frac{R}{CR^2} = \frac{6}{(0.585)^2} = 17.53 \text{ inches}$$

b) $\frac{H}{W} = \frac{48}{24} = 2.0$

c) From the fitting loss coefficient table for two splitter vanes (opposite R/W = 0.25), C = 0.04



Elbow with two splitter vanes
(Section View)

1. Select the number of splitter vanes to be used (1, 2 or 3).
2. Referring to Table 14-10, figure G (Page 14.21), calculate the R/W Ratio and select the *Curve Ratio* (CR) from the proper table.
3. Calculate Splitter Vane Spacing (for the number of vanes required):

   a) $R_1 = \frac{R}{CR}$

   b) $R_2 = \frac{R}{CR^2}$

   c) $R_3 = \frac{R}{CR^3}$
4. The proper fitting loss coefficient (C) can be selected from Table 14-10, figure G after determining the aspect ratio (H/W).

Figure 5-12 TO CALCULATE SPLITTER VANE SPACING FOR A SMOOTH RADIUS RECTANGULAR ELBOW

Example 5-3 (Metric)

A 1200 mm (H) x 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.

Solution

Using Figure 5-12 and Table 14-10, Figure G:

a) $\frac{R}{W} = \frac{150}{600} = 0.25$

From Table 14-10, Figure G for two splitter vanes, CR = 0.585

$$R_1 = \frac{R}{CR} = \left(\frac{150}{0.585}\right) = 256.4 \text{ mm}$$

$$R_2 = \frac{R}{CR^2} = \frac{150}{(0.585)^2} = 438.3 \text{ mm}$$

b) $\frac{H}{W} = \frac{1200}{600} = 2.0$

c) From the fitting loss coefficient table for two splitter vanes) (opposite R/W = 0.25), C = 0.04

B. TURNING VANES

1. Single vs 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 ASHRAE Fundamentals Handbook (American Society of Heating, Refrigeration, Air Conditioning Engineers) since 1977. SMACNA research on duct fitting turning vanes still indicates that using double thickness turning vanes instead of single thickness vanes, increases the pressure loss of elbows (see new data in Chapter 14, Table 14-10H).

Single thickness vanes have a maximum length of 36 in. (914 mm) as outlined on page 2-5 of the 1985 Edition of the SMACNA "HVAC Duct Construction Standards." Turning vanes over 36 inches (914 mm) are used in a double thickness configuration to keep their curved shape with the higher airstream velocities found in some HVAC system ductwork and to prevent vibration or fluttering. They are not more aerodynamic than single-blade vanes as originally thought, as the loss coefficients in Table 14-10H indicate.

Of course, 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). But, multiple, single thickness turning vane sections with vanes 36 inches (914 mm) long or less can be installed in large elbows instead of using double thickness vanes.

2. Trailing Edges

Trailing edges shown on single thickness vanes, design numbers 1 and 3 in Figure 3-8 of ASHRAE 1989 Fundamentals Handbook Chapter 32 also have become an industry problem. SMACNA research has shown that unless these turning vanes are made and installed perfectly, trailing edged vanes, when made with average workmanship, actually have a *higher loss* than vanes without them. And when the vanes are accidentally installed with the airflow reversed, much higher losses develop.

Because of this research, the SMACNA Duct Design Committee has recommended that turning vanes with trailing edges be eliminated from fitting loss coefficient tables and duct construction manuals when manuals are revised. They have been eliminated from Table 14-10H in this manual.

3. Vanes Missing

For many years contractors, often with the system designer's approval, have eliminated every other turning vane from the vane runners installed in rectangular mitred duct elbows. Some contractors even believed that they would lower the pressure loss of the elbow by doing this. But they were wrong! This practice more than doubles elbow pressure losses, and definitely is not recommended.

Figure 5-13 is a chart developed from SMACNA-sponsored research performed by ETL Laboratories in Cortland New York. ETL tested single thickness turning vanes with a radius of 4½ in. (114 mm). The distance between vanes was varied from 3 in. to 6½ in. (75 mm to 165 mm) in increments of ¼ in. (6mm) using embossed rail runners. Airflow velocities varied from 1,000 to 2,500 fpm (5 to 12.5 m/s) in the 24-in. × 24-in. (600 mm × 600 mm) test elbow. The loss coefficient of 0.18 for the standard spacing of 3¼ in. (82 mm) may be compared with the loss coefficient of 0.46 at a 6½ in. (165 mm) spacing (every other vane missing). *The pressure loss of the elbow with missing turning vanes was over 2½ times the pressure loss of a properly fabricated elbow containing all of the vanes.*

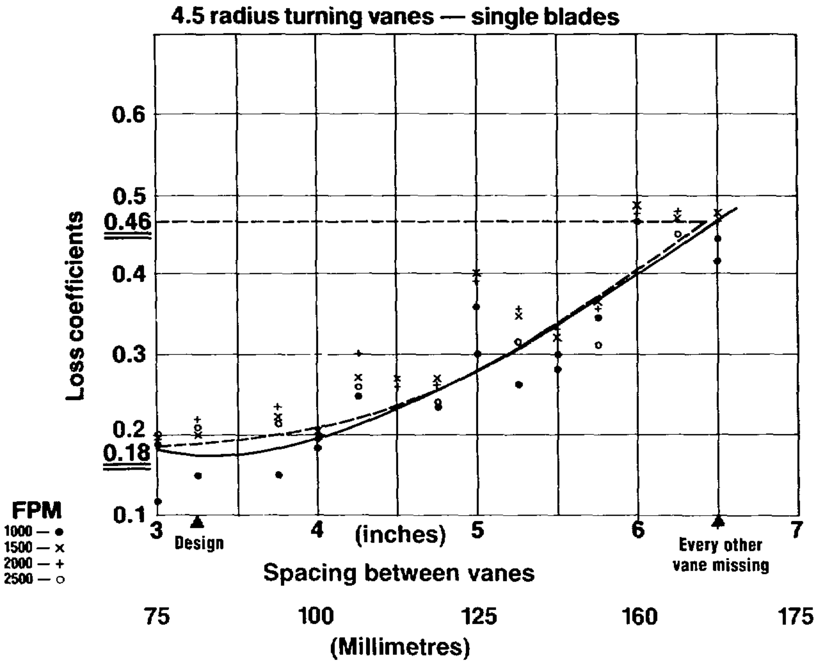


Figure 5-13 TURNING VANES RESEARCH

Example 5-4 (U.S.)

In a 2-in. w.g. pressure HVAC duct system that has six 90° elbows, an airflow velocity of 2,200 fpm, the velocity pressure (V_p) for 2,200 fpm is 0.30-in. w.g. Calculate the pressure loss of the 6 elbows, a) using 4½ in. turning vanes, single thickness, with all vanes present (Table 14-10, Figure H), b) with every other vane missing (see Figure 5-13), and c) with 2 inch double thickness turning vanes on 2.25 inch centers (Table 14-10, Figure H).

Solution

a) *Single, Standard Spacing*
The loss coefficient for a 90° elbow with 4½ in. single thickness vanes is 0.23. Using Equation 5-6:
$TP = C \times V_p = 0.23 \times 0.30$
$TP = 0.069$ in. w.g.
Loss for 6 Elbows = 0.414 in. w.g.

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. w.g.
Loss for 6 Elbows = 0.828 in. w.g.

c) *Double, Standard Spacing*
The loss coefficient for the 2 in. double thickness vane is 0.50 (2000 fpm).
$TP = C \times V_p = 0.50 \text{ x } 0.30$
$TP = 0.15$ in. w.g.
Loss for 6 Elbows = 0.90 in. w.g.

Example 5-4 (Metric)

In a 500 Pascal pressure HVAC duct system that has six 90° 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 14-10, 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. (Table 14-10, Figure H, No. 3).

Solution

a) *Single, Standard Spacing*
The loss coefficient for a 90° elbow with 114 mm single thickness vanes is 0.23. Using Equation 5-6:
$P = C \times V_p = 0.23 \times 71.6$
$TP = 16.47$ Pa
Loss for 6 elbows = 98.82 Pa

b) *Single, Every Other Vane Missing*
From Figure 5-13, C = 0.46
TP = C × V_p = 0.46 × 71.6
TP = 32.94 Pa
Loss for 6 elbows = 197.6 Pa

c) *Double, Standard Spacing*
The loss coefficient for the 50 mm double thickness vane is 0.50 (10 m/s).
TP = C × V_p = 0.50 × 71.6
TP = 35.8 Pa
Loss for 6 elbows = 214.8 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 w.g.(500 Pa) system static pressure to work with. The a) single thickness vane elbows used 0.414 in. w.g. (98.82 Pa) or 20.7 percent of the available pressure. The b) elbows, with half of the turning vanes missing, consumed 0.828 in. w.g. (197.6 Pa) or 41.4 percent of the system pressure. The c) double thickness vane elbows used 0.90 in. w.g. (214.8 Pa) or 45.0 percent.

Another turning vane problem occurs when a rectangular duct mitred elbow changes size from inlet to outlet. Until research data is available, 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 so that they are tangent to the airflow. However, the critical and rather common problem is that turning vanes are put into the vane rails as they are for a normal 90° elbow, as shown in Figure 5-14. Vanes that are not tangent to the airflow direction can cause a high pressure loss. This "non-tangent to the airflow problem" also happens in normal 90° 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.

3. Pressure Losses in Divided-Flow Fittings

A. 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.

It has been found by tests that the regain coefficient across a takeoff can be as high as 0.90 for well designed and 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 following 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.

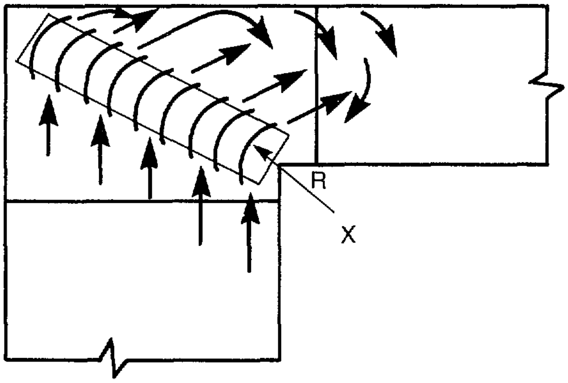


Figure 5-14 TURBULENCE CAUSED BY IMPROPER MOUNTING AND USE OF TURNING VANES

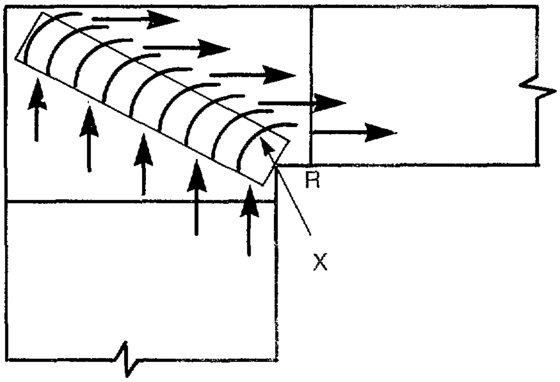


Figure 5-15 PROPER INSTALLATION OF TURNING VANES
(Vanes do not have "trailing edges," but have been moved in the vane runner to remain tangent to the airstream.)

B. 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 Tables 14-13 and 14-14 of Chapter 14. 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 in the 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 downsteam (converging) velocity using the nomenclature (V_b/V_c) shown in the figures in Tables 14-13 and 14-14. 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, etc.).

For example, data from the SMACNA Duct Fitting Research Program shows that an inexpensive 45° entry branch from a rectangular main (Table 14-14, figure N) is a far more efficient fitting to use than a rectangular branch with an expensive extractor (Table 14-14, figure S). 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 14-14, figure T) 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 14-14, figure W) 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 tape entitled "Duct Research Destroys Design Myths" was produced, which demonstrates that turbulence is directly related to fitting loss coefficients. Helium filled soap bubbles in the airstream of a lighted duct with one side of clear plastic dramatically shows the efficiency of the 45° entry fitting over the other types of branch duct tap fittings (see Figures 5-16 to 5-21).

4. Losses Due to Area Changes

Area changes in ducts, which are generally unavoidable, are frequently necessitated by the building construction or changes in the volume of air carried. Experimental investigations of pressure changes and of pressure losses at changes of the area in duct cross sections indicate 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 14-11 and 14-12 of Chapter 14.

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. Other Loss Coefficients

Loss coefficients for most commonly used entries, discharges, screens and plates, dampers and obstructions are found in Tables 14-15 to 14-18. 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 profiles across filters, coils, etc., when irregular velocities are present due to approach angle or mixing conditions, and in front of fan dis-

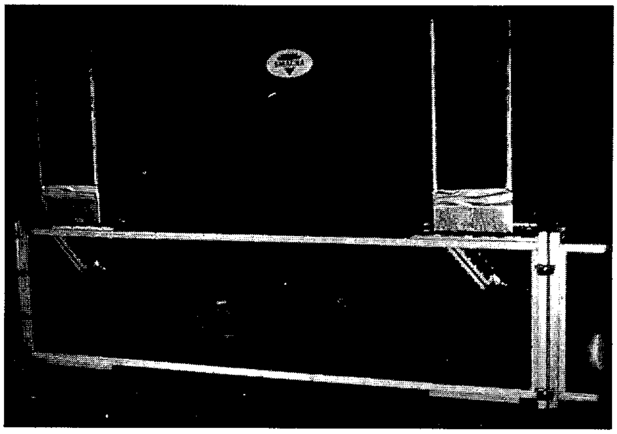


Figure 5-16 TEST DUCT WITH CLEAR PLASTIC SIDE AND EXTRACTORS

Figure 5-17 TEST DUCT WITH SMOKE (No airflow indicated)

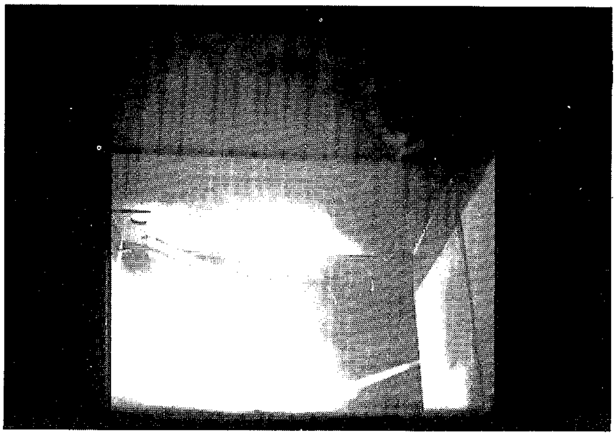


Figure 5-18 TURBULENT FLOW FROM EXTRACTOR (Looking from Branch Duct into Main Duct)

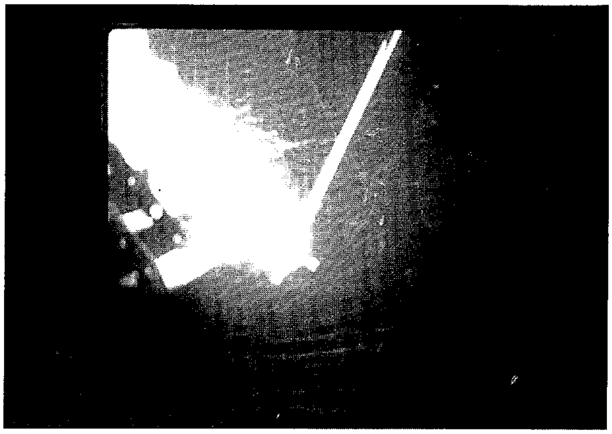


Figure 5-19 TURBULENT FLOW FROM EXTRACTOR (Looking at Main Duct)

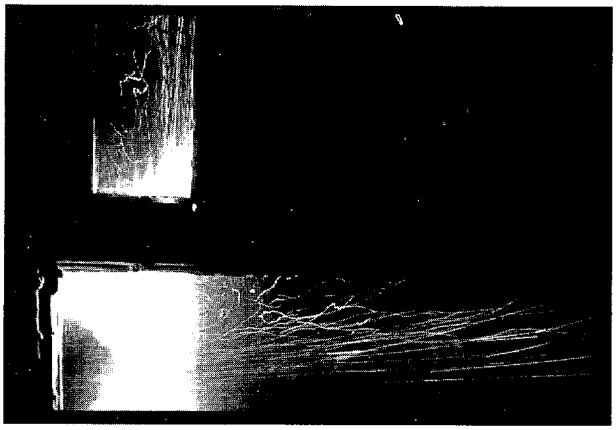


Figure 5-20 TURBULENT FLOW FROM SPLITTER DAMPER

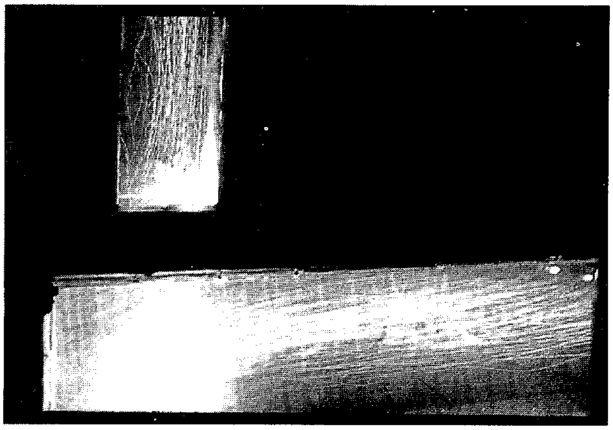


Figure 5-21 SMOOTH FLOW WITH 45° ENTRY TAP FITTING

charge 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 14-18 are based on a constant velocity. Use of these coefficients will be found in the duct design examples in Chapters 7 and 8. 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 5-22).

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 14-16, 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 5-5 using Table 14-16 G indicates why marginally sized exhaust systems have suffered through the years.

Example 5-5 (U.S.)

A small vent set has an outlet velocity of 1790 fpm at 0.25 in. w.g. static pressure. Calculate the capacity loss of the "free discharge" roof mounted fan. (θ = 30°, A_1/A = 1.5).

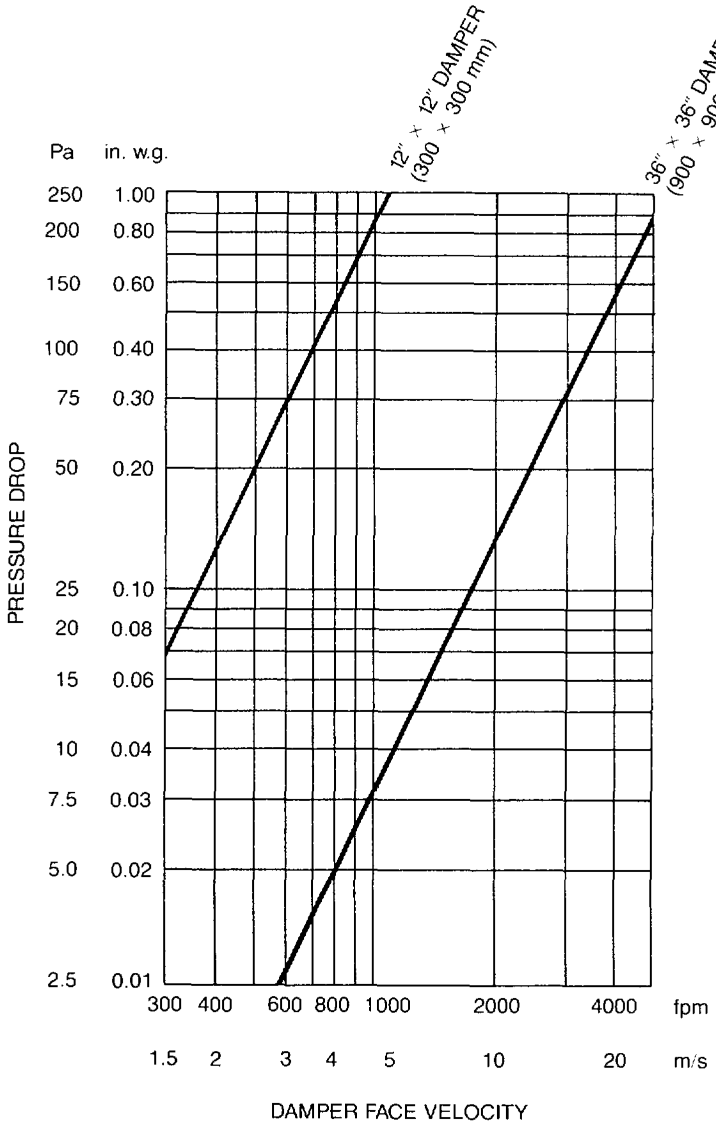


Figure 5-22 AMCA DAMPER TESTS (1)

Solution

Find V_p for 1790 fpm (using Equation 5-10):

$$V_p = \left(\frac{1790}{4005}\right)^2 = 0.20 \text{ in. w.g.}$$

From Table 14-16 G, C = 0.63. Using Equation 5-8:

$TP = C \times V_p = 0.63 \times 0.20$

$TP = 0.126$ in. w.g.

The "free discharge" consumes 50 percent of the rated 0.25 in. w.g. fan capacity.

Example 5-5 (Metric)

A small vent set has an outlet velocity of 9 m/s at 60 pascals static pressure. Calculate the capacity loss of the "free discharge" roof mounted fan.

($\theta = 0.5$ rad, $A_1/A = 1.5$)

Solution

Find V_p for 9 m/s (using Equation 5-10):

$$V_p = \left(\frac{9}{1.30}\right)^2 = 47.9 \text{ Pa}$$

From Table 14-16 G, C= 0.63

Using Equation 5-8:

$TP = C \times V_p = 0.63 \times 47.9$

$TP = 30.2$ Pa

The "free discharge" consumes about 50 percent of the rated 60 Pa fan capacity.

6. 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. Table 14-18, figures I to L are the result of this work. An offshoot of this project was the discovery of the need for new duct friction loss charts (now found in Figures 14-1 and 14-2).

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 w.g. 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. w.g. (15 Pa) loss.

Rectangular—$C \times V_p = 0.35 \times 0.25\ (62) = 0.09$ in w.g. (22 Pa) loss.

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 14-18 figure L can be used. This configuration was tested over an extensive time period with every conceivable variation of dimensions, aspect ratios, beam heights and widths, etc. plus the turning vane variations. Unfortunately, some of these fittings have been installed without turning vanes (usually because some sheet metal contractors have found that they do not get paid for furnishing expensive fittings which were not shown on the project mechanical drawings.)

Nevertheless, this type of fitting installed *without* turning vanes totally can destroy the airflow in a duct system as is shown (and compared with the same fitting with turning vanes) in the following example:

Example 5-6 (U.S.)

An average low pressure duct system might be designed to develop a velocity of 2000 fpm at 2.5 in. w.g. total pressure in the main supply duct leaving the fan. What would be the pressure loss of the fitting found in Table 14-18, Figure L if the beam/duct height ratio (L/H) was 2 (with and without single thickness turning vanes)?

Solution:

From Table 14-6, V_p = 0.25 for 2000 fpm.

From Table 14-18, 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. w.g.

Without Turning Vanes:

Fitting loss $= C \times V_p = 9.24 \times 0.25$

$= 2.31$ in. w.g.

One can see that the 0.19 in w.g. pressure loss of the fitting with turning vanes is but 8 percent of the initial 2.5 in w.g. in the duct system. *The 2.31 in. w.g. pressure loss of the fitting without turning vanes theoretically destroys the system airflow by wiping out 92% of the 2.5 in. w.g. total system pressure!* Actually, the operating point of the system/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 5-6 (Metric)

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 14-18, Figure L if the beam/duct height ratio (L/H) was 2 (with and without single thickness turning vanes)?

Solution:

From Table 14-7, V_p = 62 Pa for 10 m/s.
From Table 14-18, Figure L,
C = 0.77 for single thickness turning vanes
C = 9.24 without turning vanes

With Turning Vanes:
Fitting loss = C × V_p = 0.77 × 62 = 47.7Pa

Without Turning Vanes:
Fitting loss = C × V_p = 9.24 × 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/fan curve interchange moves up and to the left on the fan curve, substantially reducing the system airflow (see Figure 5-23).

F 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. Each has a relatively short shelf life and no documented research has identified the in-service aging characteristics of sealant applications. Many sealants contain volatile solvents that evaporate and introduce shrinkage and curing factors. Surface cleanliness and sealant application in relation to air pressure direction are other variables. With the exception of pressure-sensitive adhesive tapes, no standard tests exist to evaluate performance and 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:

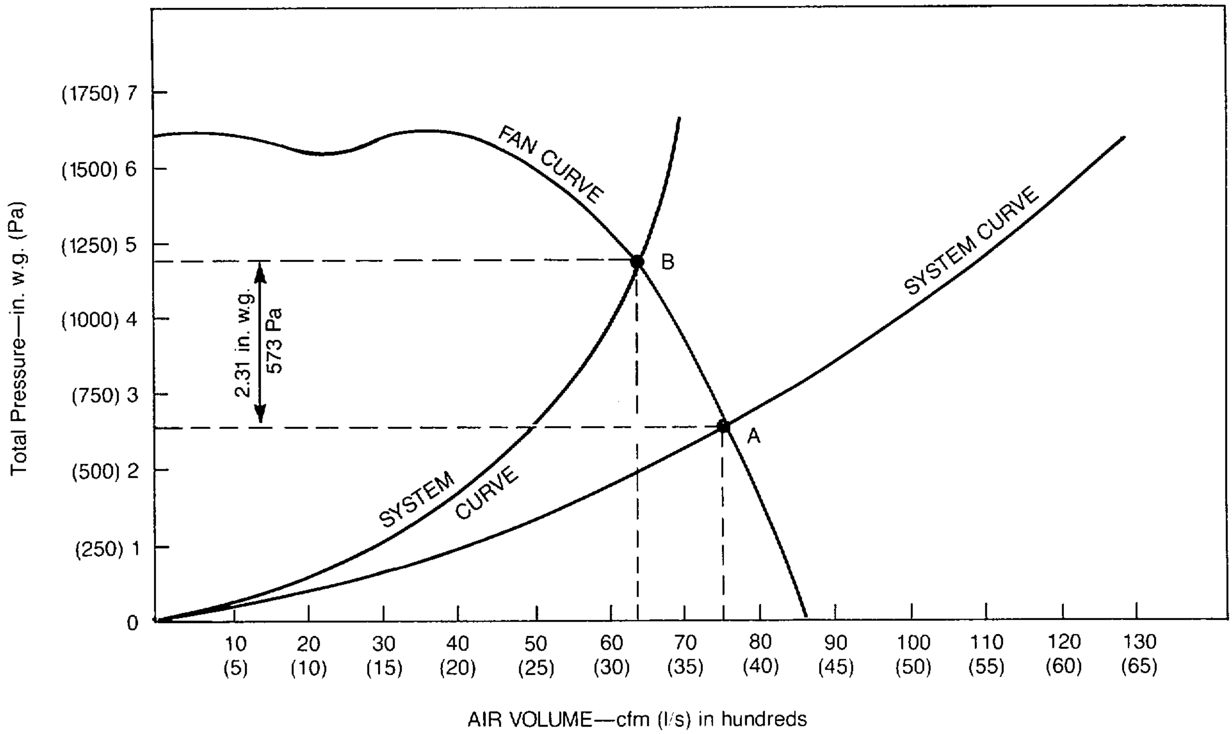


Figure 5-23 EXAMPLE 5-5 FAN/SYSTEM CURVE

Table 5-1 Unsealed Longitudinal Seam Leakage for Metal Ducts

Type of Duct/Seam	Leakage,[a]			
	cfm/ft (seam length)		l/s per metre (seam length)	
	Range	Average	Range	Average
Rectangular				
Pittsburgh Lock	0.01 to 0.56	0.16	0.015 to 0.87	0.25
Button Punch Snap Lock	0.01 to 0.16	0.08	0.015 to 0.25	0.10
Round				
Snap Lock	0.04 to 0.14	0.11	0.06 to 0.22	0.17
Grooved	0.11 to 0.18	0.12	0.17 to 0.28	0.19

[a] Leakage rate is at 1 in. w.g. (250 Pa) static pressure

Equation 5-11

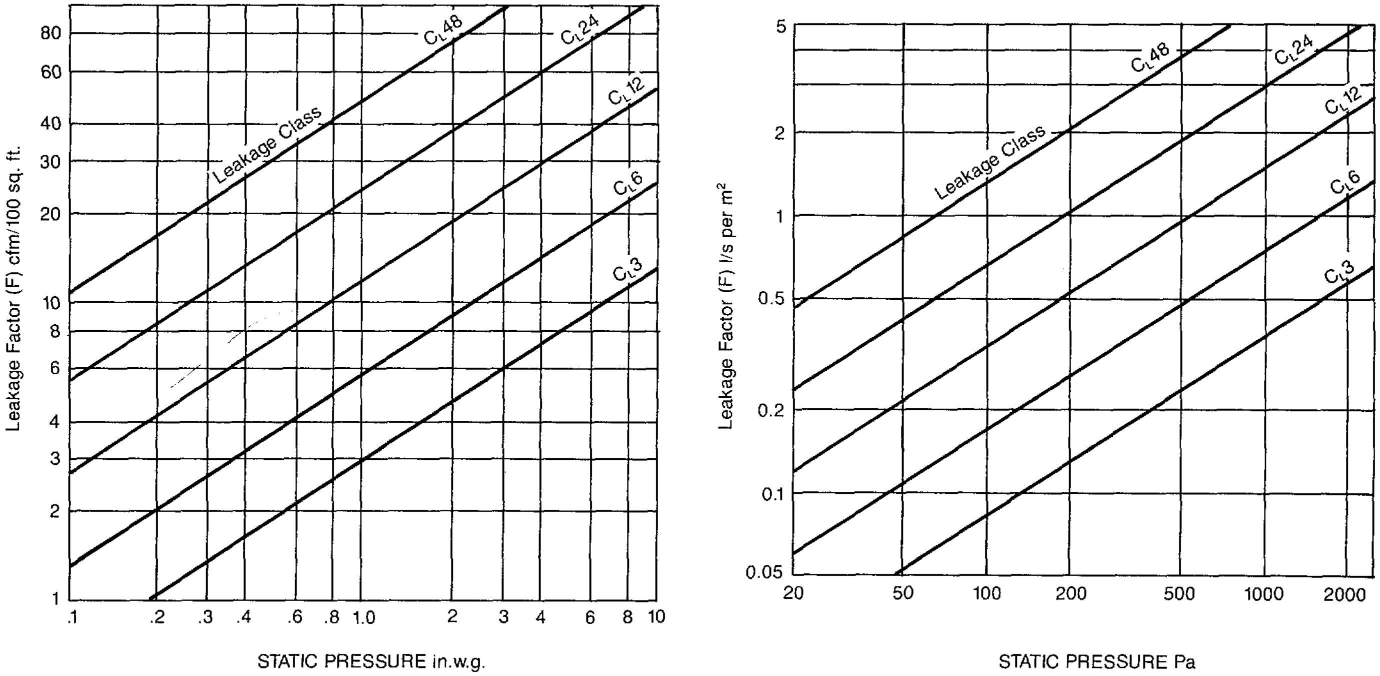
$$F = C_L P^N$$

where:

F = Leak rate per unit of duct 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 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.

Analysis of the SMACNA/ASHRAE/TIMA data resulted in the categorization of duct systems into a *leakage class (C_L)* based on Equation 5-12, where the exponent *N* is assumed to be 0.65. A selected series of leakage classes based on Equation 5-12 is shown in Figure 5-24.



A (U.S. Units)

B (Metric Units)

Figure 5-24 DUCT LEAKAGE CLASSIFICATIONS

Table 5-2 APPLICABLE LEAKAGE CLASSES[a]

DUCT CLASS (SEE TABLE 4-1)	1/2, 1, 2 in.w.g. (125, 250, 500 Pa)		3 in.w.g. (750 Pa)	4, 6, 10 in.w.g. (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 (C_L) cfm/100 sq. ft (l/s per m^2) at 1 in.w.g. (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

[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.

Equation 5-12 (U.S.)

$$C_L = F/P^{0.65}$$

Equation 5-12 (Metric)

$$C_L = 720\ F/P^{0.65}$$

where:

C_L = Leakage class at 1 in.w.g. (250 Pa) static pressure—cfm/100 sq. ft. (l/s per m^2)

F = Leakage rate—cfm/100 sq. ft. (l/s per m^2) duct surface

P = Static Pressure—in.w.g. (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 cost and life cycle cost of a duct system so the owner may benefit. In performing an analysis, the designer should independently account for air leakage in casings and frames of equipment in the duct system. Casings or volume-controlling air terminal units may leak 2 to 5% 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 and by good workmanship.

Where zero leakage is required, designers should understand that contractors may have difficulty meeting their requirements. Zero leakage is not a practical objective except in critical situations such as nuclear safety-related applications. One (1) percent leakage also is difficult or impossible to attain in larger systems. The shaded area in Table 5-3 predicts that one (1) percent leakage in duct systems is only attainable up to 2 in.w.g. (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 (U.S.)

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.

Solution

a) The average pressure from A to F is 2.5 in.w.g. [(3 + 2)/2]. From Table 5-2, the leakage class for a 3 in.w.g. duct class round metal duct is "6". Using Figure 5-24(A), the leakage factor would be 10.6 cfm/100 sq. ft. The 34 inch diameter duct from A to F has 800 square feet of duct surface.

Leakage = 10.6/100 × 800 = 85 cfm

b) The average pressure from F to J and P is 1.5 in.w.g. [(2 + 1)/2]. From Table 5-2, the leakage class for a 2 in.w.g. duct class round metal duct is "12". Using Figure 5-24(A), the leakage factor would be 15.5 cfm/100 sq. ft. The total calculated duct surface is 900 square feet.

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.

Table 5-3 LEAKAGE AS A PERCENTAGE OF SYSTEM AIRFLOW

LEAKAGE CLASS	System Airflow		STATIC PRESSURE in.w.g. (Pa)					
	cfm/ft^2	l/s per m^2	1/2 (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

c) The average pressure from J to M is 1 in.w.g. because the VAV boxes require 1 in.w.g. inlet pressure. From Table 5-2, the leakage class will remain at "12", as it is based on 1 in.w.g. The calculated duct surface for all four (4) branches (M, S, W, and X) is 1320 square feet.

Leakage = 12/100 × 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 × 100/20,000 cfm
= 2.62%

Example 5-7 (Metric)

Using a typical duct system shown in Figure 8-2 of Chapter 8, find the total leakage of the supply ductwork in both l/s and percentage of airflow.

Solution

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 5-24(B), the leakage factor would be 0.6 l/s per square metre. The 900 mm diameter duct from A to F has 90 square metres of surface.
Leakage = 0.6 × 90 = 54 l/s

b) The average pressure from F to I and O 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 5-24(B), the leakage factor would be 0.75 l/s per square metre. The total calculated duct surface (including all four branches) would be 146 square metres.
Leakage = 0.75 × 146 = 110 l/s

c) The average pressure from I 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 (4) branches (M, S, W and X) is 208 square metres. Using Figure 5-24(B), the leakage factor is 0.65 l/s per square metre.
Leakage = 0.65 × 208 = 135 l/s

d) The total duct system leakage (not counting the flexible connections) is:

a) AF = 54 l/s
b) F/IO = 110 l/s
c) IM/etc. = 135 l/s
Total = 299 l/s duct leakage

Percent leakage = 299 × 100/10,000 l/s
= 2.99%

Although the duct systems in Figures 7-2 and 8-2 are similar, the metric unit dimensions are *not* conversions from the U.S. unit dimensions, so the percentage leakages from the two examples cannot be compared.

If a VAV system was built to a duct class of 3 in.w.g. (750 Pa) throughout and 1 in.w.g. (250 Pa) was required at the VAV boxes, the average pressure would be 2 in.w.g. (500 Pa). From Table 5-2 and Figure 5-24 the following is obtained:

Round metal duct,
C_L = 6 & F = 9 cfm/100 ft^2 (0.5 l/s per m^2)

Rectangular metal duct,
C_L = 12 & F = 18 cfm/100 ft^2 (0.95 l/s per m^2)

To obtain a one (1) percent leakage rate using a 10,000 cfm (5000 l/s) fan, the system would be limited in size to the following:

a) *U.S. Units*
1% of 10,000 cfm = 100 cfm
Round duct = 100 cfm/9 cfm/100 sq. ft.
= 1111 square feet (maximum)

Rectangular duct = 100 cfm/18 cfm/100 sq. ft.
= 556 square feet (maximum)

b) *Metric Units*
1% of 5000 l/s = 50 l/s
Round duct = 50 l/s/0.5 l/s per m^2
= 100 square metres (maximum)

Rectangular duct = 50 l/s/0.95 l/s per m^2
= 53 square metres (maximum)

It becomes obvious that to obtain a one (1) 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 ductwork must be welded or soldered, with the resultant extreme increase in costs of fabrication and erection.

G DUCT HEAT GAIN/LOSS

ANSI/ASHRAE/IES *Standard* 90A (1980) requires thermal insulation of all duct systems and their components (*i.e.*, ducts, plenums, and enclosures) installed in or on buildings. Adequate thermal insulation is determined by:

Equation 5-13 (U.S.)

$$R = \Delta t/15$$

Equation 5-13 (Metric)

$$R = \Delta t/47.3$$

where:

R = thermal resistance excluding film resistances, ft^2·°F·h/Btu (m^2·°C/W)
Δt = design temperature differential between duct air and duct surface, °F (°C)

Duct insulation is not required in any of the following cases:

1. Where supply or return air ducts are installed in basements, cellars, or unventilated crawl spaces with insulated walls in one- and two-family dwellings.
2. When the heat gain or loss of the ducts, without insulation, will not increase the energy requirements of the building.
3. Exhaust air ducts.

Since Standard 90A 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 5-14 to 5-16.

Equation 5-14 (U.S.)

$$Q_l = \frac{UPL}{12}\left[\left(\frac{t_e + t_l}{2}\right) - t_a\right]$$

Equation 5-14 (Metric)

$$Q_l = \frac{UPL}{1000}\left[\left(\frac{t_e + t_l}{2}\right) - t_a\right]$$

Equation 5-15 (U.S. & Metric)

$$t_e = \frac{t_l\,(y + 1) - 2t_a}{(y - 1)}$$

Equation 5-16 (U.S. & Metric)

$$t_l = \frac{t_e\,(y - 1) + 2t_a}{(y + 1)}$$

where:

y = $2.4AV\rho/UPL$ for rectangular ducts ($2.01\ AV\rho/UPL$)
y = $0.6DV\rho/UL$ for round ducts ($0.5DV\rho/UL$)
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_l = 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)
ρ = density, lb$_m$/ft^3 (kg/m^3)
t_e = temperature of air entering duct, °F (°C)
t_l = temperature of air leaving duct, °F (°C)
t_a = temperature of air surrounding duct, °F (°C)

Use Figure 14-6 (14-7) in Chapter 14 to determine the "U-values" for insulated and uninsulated ducts. For a 2 inch (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 14-6(a) [14-7(a)]. Pervious flexible duct liners also influence heat transfer significantly as shown in Figure 14-6(b) [14-7(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 5-8 (U.S.)

A 65 foot length of 24 inch by 36 inch uninsulated sheet metal duct, freely suspended, conveys heated air through a space maintained above freezing at 40°F. Based on heat loss calculations for the heated zone, 17,200 cfm of standard air at a supply air temperature of 122°F is required. The duct is connected directly to the heated zone. Determine the air temperature entering the duct and the duct heat loss.

Solution

a) Calculate the duct velocity using Equation 5-9:

$$V = \frac{Q}{A} = \frac{17{,}200}{24 \times 36/144} = 2900 \text{ fpm}$$

Select $U = 0.73$ Btuh/ft^2·°F (from Figure 14-6).

Calculate $P = 2(24 \text{ in.} + 36 \text{ in.}) = 120 \text{ in.}$

$$y = 2.4A\ V\rho/UPL$$

$$y = \frac{2.4 \times 24'' \times 26'' \times 2900 \times 0.075 \text{ lb/ft}^3}{0.73 \times 120'' \times 65'}$$

$$y = 79.2$$

b) Calculate the entering air temperature using Equation 5-15:

$$t_e = \frac{122°F\ (79.2\ + 1)\ -\ (2\ \times\ 40°F)}{(79.2\ -\ 1)}$$

$$t_e = 124.1°F$$

c) Calculate the duct heat loss using Equation 5-14:

$$Q_l = \frac{0.73 \times 120'' \times 65'}{12} \times \left[\left(\frac{124.1°F + 122°F}{2}\right) - 40°F\right]$$

$$Q_l = 39{,}200 \text{ Btuh}$$

Example 5-8 (Metric)

A 20 metre length of 600 mm by 900 mm uninsulated sheet metal duct, freely suspended, conveys heated air through a space maintained above freezing at 5°C. Based on heat loss calculations for the heated zone, 8100 l/s of standard air at a supply air temperature of 50°C is required. The duct is connected directly to the heated zone. Determine the air temperature entering the duct and the duct heat loss.

Solution

a) Calculate the duct velocity using Equation 5-9:

$$V = \frac{Q}{A} = \frac{1000 \times 8100}{600 \times 900} = 15 \text{ m/s}$$

Select U = 4.16 W/m²·°C (from Figure 14-7).

Calculate P = 2 (600 + 900) = 3000 mm

$$y = 2.01\ AV\rho/UPL$$

$$y = \frac{2.01 \times 600 \times 900 \times 15 \times 1.204 \text{ kg/m}^3}{4.16 \times 3000 \times 20 \text{ m}}$$

$$y = 78.5$$

b) Calculate the entering air temperature using Equation 5-15:

$$t_e = \frac{50°C\ (78.5\ +\ 1)\ -\ (2\ \times\ 5°C)}{78.5\ -\ 1}$$

$$t_e = 51.2°C$$

c) Calculate the duct heat loss using Equation 5-14:

$$Q_l = \frac{4.16 \times 3000 \times 20 \text{ m}}{1000} \times \left[\left(\frac{51.2°C + 50°C}{2}\right) - 5°C\right]$$

$$Q_l = 11.4 \text{ kW}$$

Example 5-9 (U.S)

Same as Example 5-8, except the duct is insulated externally with 2 in. thick fibrous glass with a density of 0.75 lb/ft³. The insulation is wrapped with 0% compression.

Solution

All values, except U and P, remain the same as Example 5-8. From Figure 14-6(a), U = 0.15 btu/h·ft²·°F at 2900 fpm. P = 136 in. Therefore:

$$y = 441$$

$$t_e = 122.4°F$$

$$Q_l = 9083 \text{ Btuh}$$

Insulating this duct reduces heat loss to 20 percent of the uninsulated duct.

Example 5-9 (Metric)

Same as Example 5-8, except the duct is insulated externally with 50 mm thick fibrous glass with a density of 12 kg/m³. The insulation is wrapped with 0% compression.

Solution

All values, except U and P, remain the same as Example 5-8. From Figure 14-7(a), U = 0.83 W/(m²·°C) at 15 m/s. P = 3400 mm. Therefore:

$$y = 394$$

$$t_e = 50.2°C$$

$$Q_l = 2300 \text{ W } (2.3 \text{ kW})$$

Insulating this duct reduces heat loss to 20 percent of the uninsulated duct.

H SMACNA DUCT RESEARCH

For over 10 years, the Research Department of SMACNA has worked independently with universities and testing laboratories, and jointly with ASHRAE in various duct system research projects. At SMACNA and/or ASHRAE chapter meetings, HVAC system designers and contractors were asked to submit ideas for test projects based on their perceived need from experience or problems found in their area or region. Many of these research projects are in various stages of completion, with the results in some cases, still undetermined.

Although the SMACNA Duct Design Committee has incorporated new fitting loss coefficient data for turning vanes into Table 14-10H in Chapter 14 after many years of testing, the balance of the fitting loss coefficient data in this section did not have a sufficient range of testing to be *totally* reliable under all conditions. However, the data is accurate within the research test parameters listed.

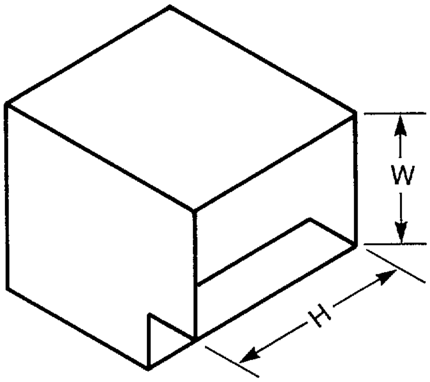
1. Other Elbow Configurations

In addition to the various test projects on turning vane elbows discussed earlier in Section E, other types of mitered rectangular duct elbows without turning vanes were tested. Loss coefficients were obtained for three sizes of 90° mitered elbows—12 × 12 inch (300 × 300 mm), 22 × 8 inch (550 × 200 mm), and 8 × 22 inch (200 × 550 mm). These are compared with existing data from Table 14-10D in Chapter 14 in Figure 5-25. 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.

Figure 5-26 shows three unusual configuration 90° 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° rectangular elbow (Table 14-10F from Chapter 14) with R/W = 0.75 and 1.0 is compared with that of Elbows A, B, and C in Figure 5-26. Note that when the throat of the 90° mitered elbow (Figure 5-25) is changed from 90° to 45° (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).

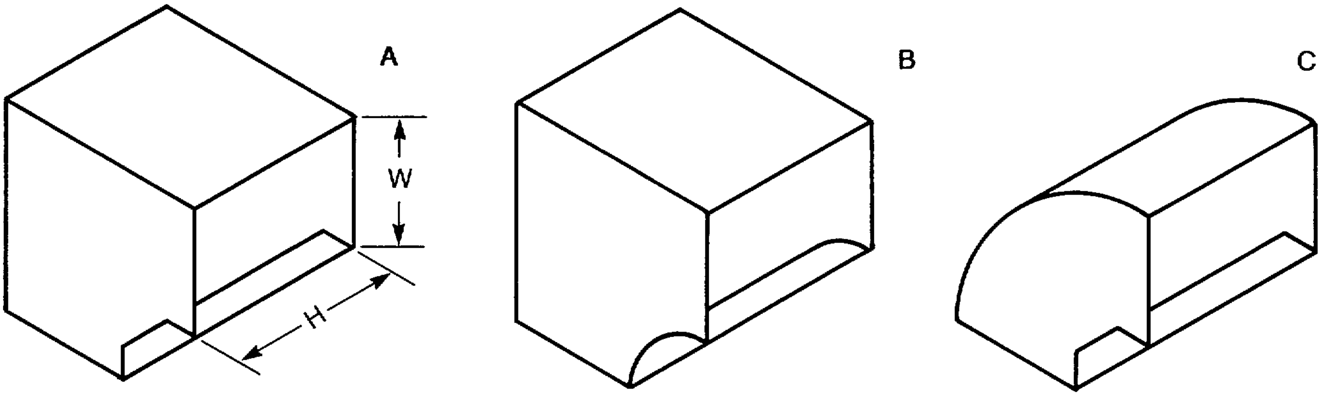
2. Taps at End of Ducts

Many new duct systems are installed without supply outlets in place until the tenant space is leased and



	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 5-25 RECTANGULAR ELBOW WITH 90° THROAT, 90° HEEL



45° Throat, 90° Heel

Radius Throat 90° Heel

45° Throat Radius Heel

Elbow	Fitting Loss Coefficient (C)		
	H/W		
	0.36	1.00	2.75
A B C	0.41 0.36 0.34	0.60 0.58 0.33	0.53 0.45 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)

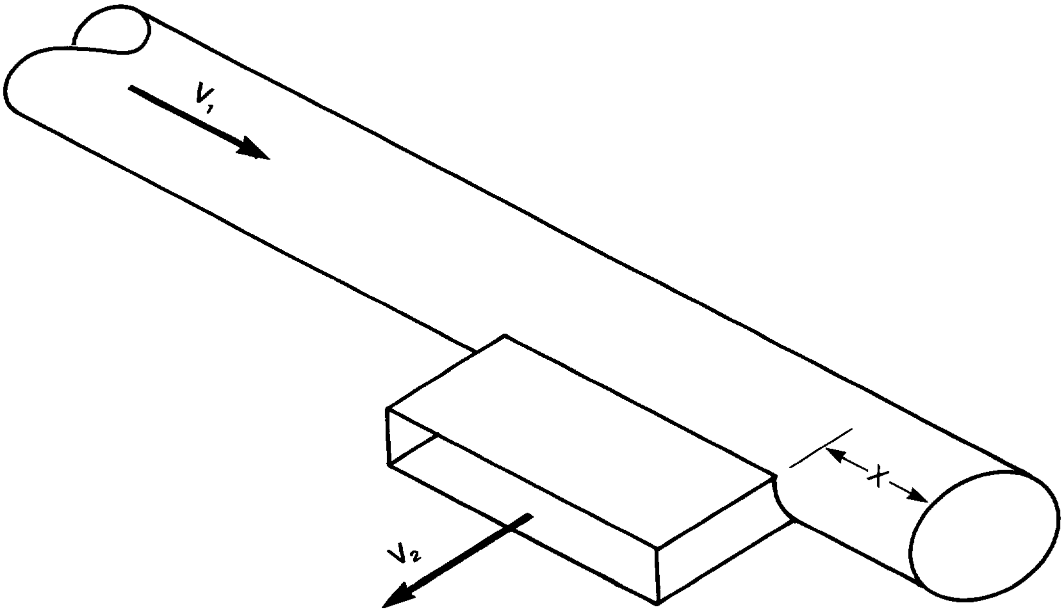
Figure 5-26 DIFFERENT CONFIGURATION ELBOW RESEARCH

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 common question, tests were made of 10 and 12 inch (250 and 300 mm) diameter ducts with 14 × 6 and 22 × 8 inch (350 × 150 and 550 × 200 mm) taps located 1, 6, and 12 inches (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 5-27. 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, compare the values with Tables 14-10D and 14-14Q in Chapter 14.

3. Future Test Results

SMACNA unilaterally has additional research projects underway along with joint research projects with ASHRAE. Between editions of this manual, Technical Bulletins will be issued to allow SMACNA Contractors access to the latest in HVAC system design information resulting from these projects.



Rectangular tap near end of round duct

	Fitting Loss Coefficient (C)		
V_2/V_1	X = 1 in. (25mm)	X = 6 in. (150mm)	X = 12 in. (300mm)
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 5-27 END TAP RESEARCH

CHAPTER 6

FAN-DUCT CONNECTION PRESSURE LOSSES

Most of the text material and accompanying tables and figures in this section were developed by the Air Moving and Conditioning Association, 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 (on the abcissa), it is possible to read across from any curve (to the ordinate) to find the "System Effect Factor" for a particular configuration. System Effect Curve "letter designations" (such as R, S, T, etc.) 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 inches of water gauge (in. w.g.) or Pascals (Pa) and it must be added to the total system pressure losses or subtracted from the fan performance pressure rating.

The velocity rate used in entering the chart will be either the inlet or the outlet velocity of the fan, dependent on whether the configuration in question is related to the fan inlet or the 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). The necessary 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.

A FAN OUTLET CONDITIONS

1. 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 with only 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° for converging elements nor greater than 7° for diverging elements.

Figure 6-2 shows the changes 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 6,000 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.

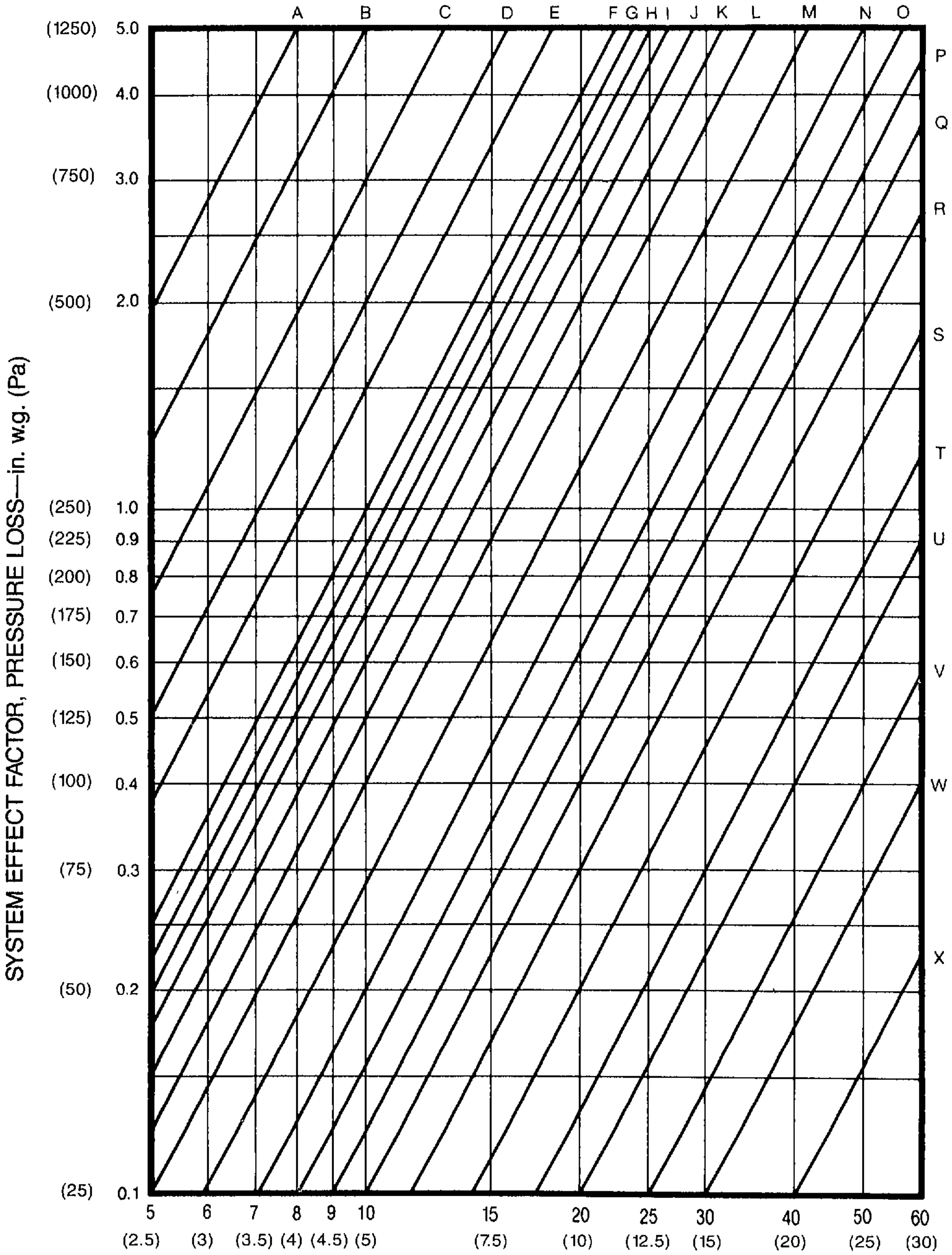
To determine the applicable System Effect Factor, calculate the average velocity in the outlet duct and enter the System Effect Curves (Figure 6-1) at this velocity. Select the appropriate System Effect Curve from Table 6-1. The ratio of blast area to outlet area is not usually included in fan catalog data and it will be necessary to obtain this from the fan manufacturer.

NOTE: 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.

2. Outlet Diffusers or Evasés

The process which takes 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.

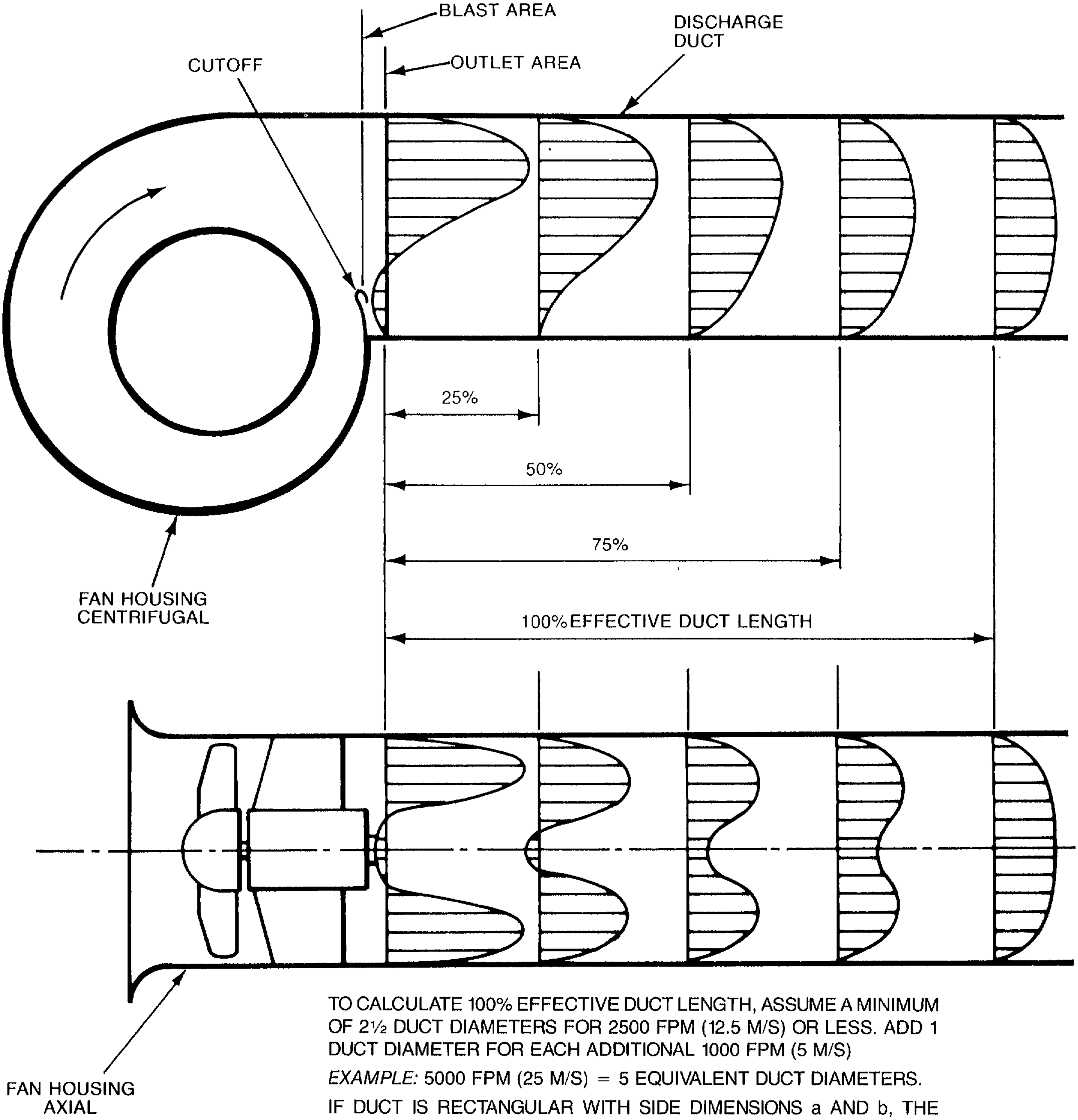
In many systems, it may be feasible to use an outlet duct which is considerably larger than the fan outlet. In these cases, the static pressure available to over-



AIR VELOCITY—FPM IN HUNDREDS (m/s)

Air Density = 0.075 lb per cu ft (1.204 kg/m^3)

Figure 6-1 SYSTEM EFFECT CURVES (1)



TO CALCULATE 100% EFFECTIVE DUCT LENGTH, ASSUME A MINIMUM OF 2½ 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° for converging and 7° 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 (1)

Table 6-1 SYSTEM EFFECT CURVES FOR OUTLET DUCTS (1)

	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	—	—	—	—	—

come system resistance can be increased by converting some of the fan outlet velocity pressure to static pressure.

To achieve this conversion efficiently, it is necessary to use a connection piece between the fan outlet and the duct which allows the airstream to expand gradually. This is called a diffuser or evasé.

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.

3. 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 velocity profile at the outlet of a fan is not uniform and an elbow located at or near the fan outlet will, therefore, develop a pressure loss greater than its "table" value.

The amount of this 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 be to further distort the outlet velocity profile of the fan. This will increase the losses and may result in such uneven flow in the duct that branch takeoffs near the elbow will not deliver their designated airflow.

Wherever 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, then 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 which 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.

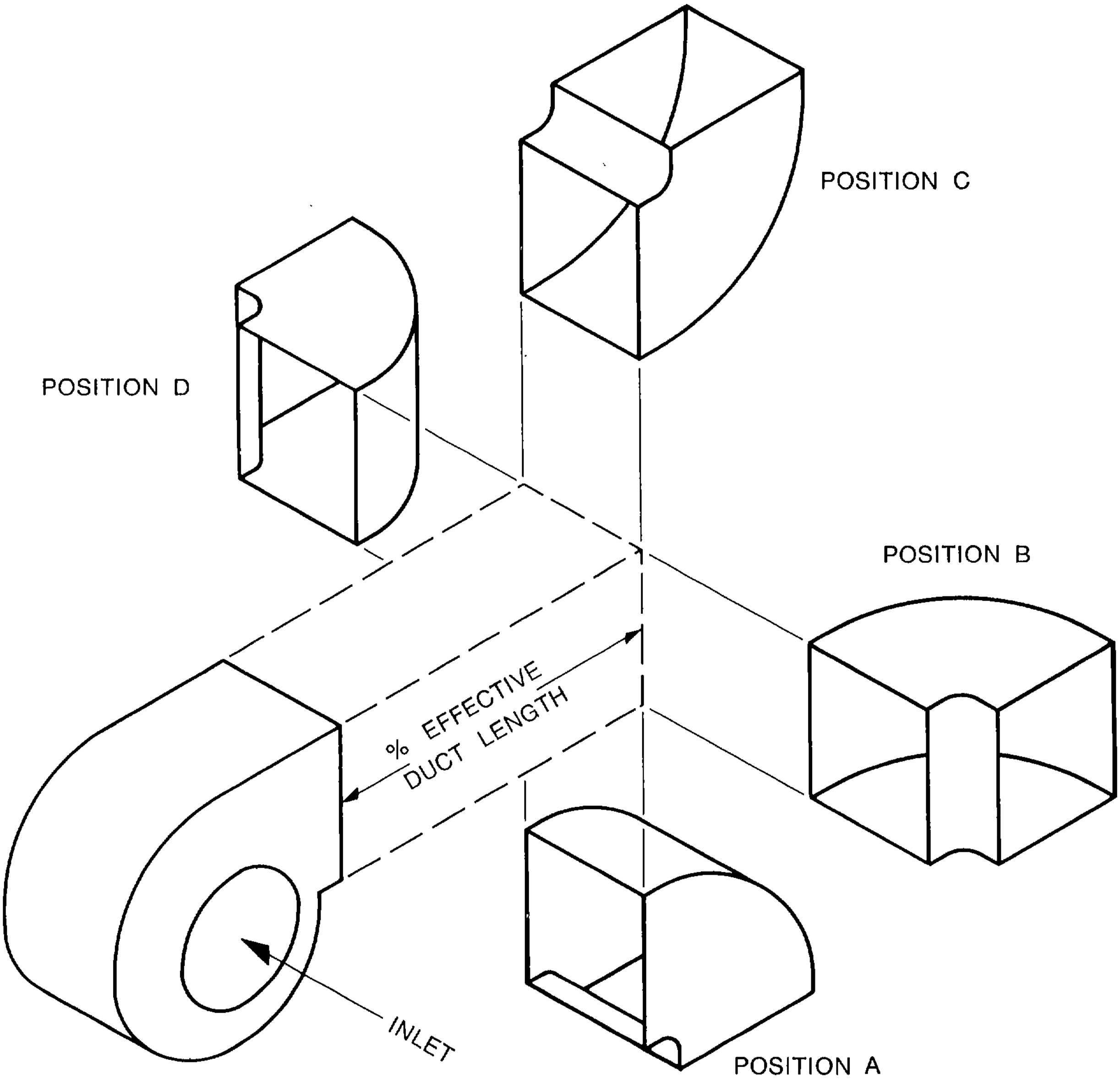
4. Turning Vanes

Turning vanes will usually reduce the pressure loss through an elbow, but where a non-uniform approach velocity profile exists, such as at 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.

5. Fan Volume Control Dampers

Dampers can be furnished as accessory equipment by the fan manufacturer; 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" blades or "parallel" blades. When partially closed, the parallel bladed damper diverts the airstream to the side of the duct. This results in



SWSI Centrifugal Fan Shown

Note: Fan Inlet and Elbow Positions Must be Oriented as shown for Proper Application of System Effect Factors (Table 6-2)

Figure 6-3 OUTLET DUCT ELBOWS (1)

a non-uniform velocity profile beyond the damper, and flow to branch ducts close to the downstream side 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 to free space, a parallel blade damper may be satisfactory.

For a centrifugal fan, best air performance usually will be achieved by installing the damper with its

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	

SYSTEM EFFECT FACTOR CURVES FOR SWSI FANS

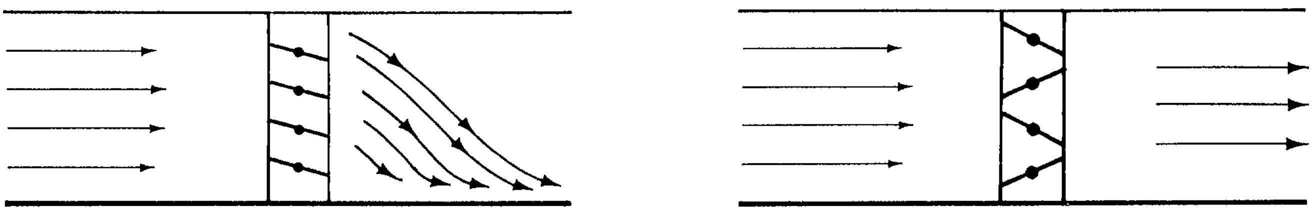
FOR DWDI FANS DETERMINE SYSTEM EFFECT FACTOR CURVE USING THE ABOVE TABULATION FOR SWSI FANS. NEXT DETERMINE SYSTEM EFFECT FACTOR (ΔP) BY USING FIGURE 6-1 THEN APPLY APPROPRIATE MULTIPLIER FROM TABULATION BELOW:

MULTIPLIERS FROM DWDI FANS

ELBOW POSITION B = Δ × 1.25
ELBOW POSITION D = Δ P × 0.85
ELBOW POSITIONS A AND C = ΔP × 1.00

REFER TO FIGURE 6-3 FOR ELBOW POSITION DESIGNATION CURVES

Table 6-2 **SYSTEM EFFECT FACTOR CURVES FOR OUTLET ELBOWS** (1)



PARALLEL BLADED DAMPER ILLUSTRATING DIVERTED FLOW

OPPOSED BLADED DAMPER ILLUSTRATING NON-DIVERTING FLOW

Figure 6-4 PARALLEL VS. OPPOSED DAMPERS (1)

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 which should be applied to the damper manufacturer's cataloged pressure loss when the damper is installed at the outlet of a centrifugal fan.

6. Duct Branches

Standard procedures for the design of duct systems are all based on the assumption of uniform flow profiles in the system (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. Wherever possible, a length of straight duct should be installed between the fan outlet and any split or branch takeoff.

B 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 useable performance of a fan. Cabinet clearance effect or plenum effect is considered a component part of the entire system and the pressure losses through the cabinet or plenum must be considered as a System Effect when determining system characteristics.

1. Inlet Ducts

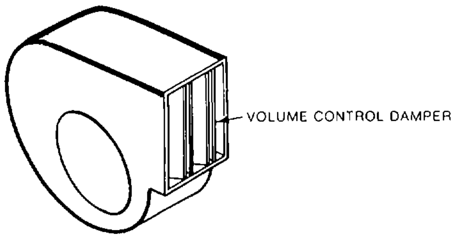
Some fans intended primarily for use as "exhausters" may be tested with an inlet duct in place or with a special bell-mouth inlet to stimulate the effect of a duct. Figure 6-8 illustrates the variations in inlet flow which will occur. A ducted inlet condition is shown as (a), the unducted condition as (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 which 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 and even a simple flat flange on the end of a duct will reduce the loss to about one-half of the loss through an unflanged entry.

AMCA Standard 210 limits an inlet duct to a cross-sectional area not greater than 112½ percent nor less than 92½ percent of the fan inlet area. The included angle of transition elements is limited to 15° converging and 7° diverging.

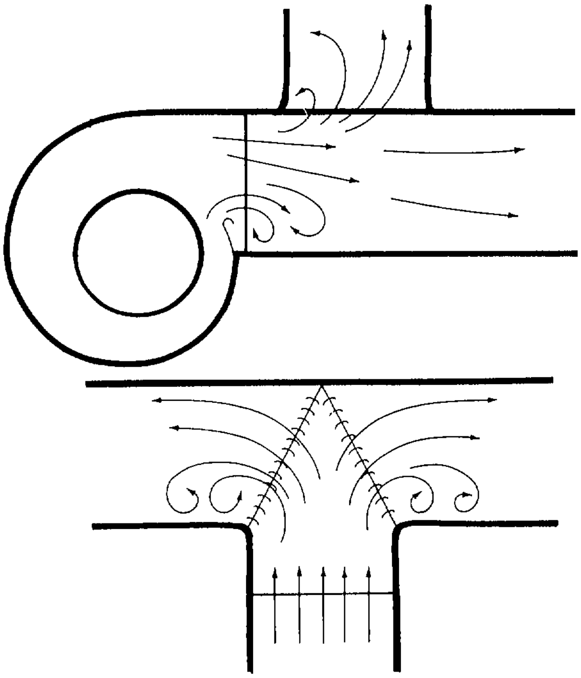
2. Inlet Elbows

Non-uniform flow into the inlet is the most common cause of deficient fan performance. An elbow or a 90° duct turn located at the fan inlet will not allow the air



BLAST AREA / OUTLET AREA	ΔSP MULTIPLIER
0.4	7.5
0.5	4.8
0.6	3.3
0.7	2.4
0.8	1.9
0 9	1.5
1.0	1.2

Figure 6-5 PRESSURE LOSS MULTIPLIERS FOR VOLUME CONTROL DAMPERS (1)



AVOID LOCATION OF SPLIT OR DUCT BRANCH CLOSE TO FAN DISCHARGE. PROVIDE A STRAIGHT SECTION OF DUCT TO ALLOW FOR AIR DIFFUSION.

(See Figure 6-2 for corrective calculations)

Figure 6-6 BRANCHES LOCATED TOO CLOSE TO FAN (1)

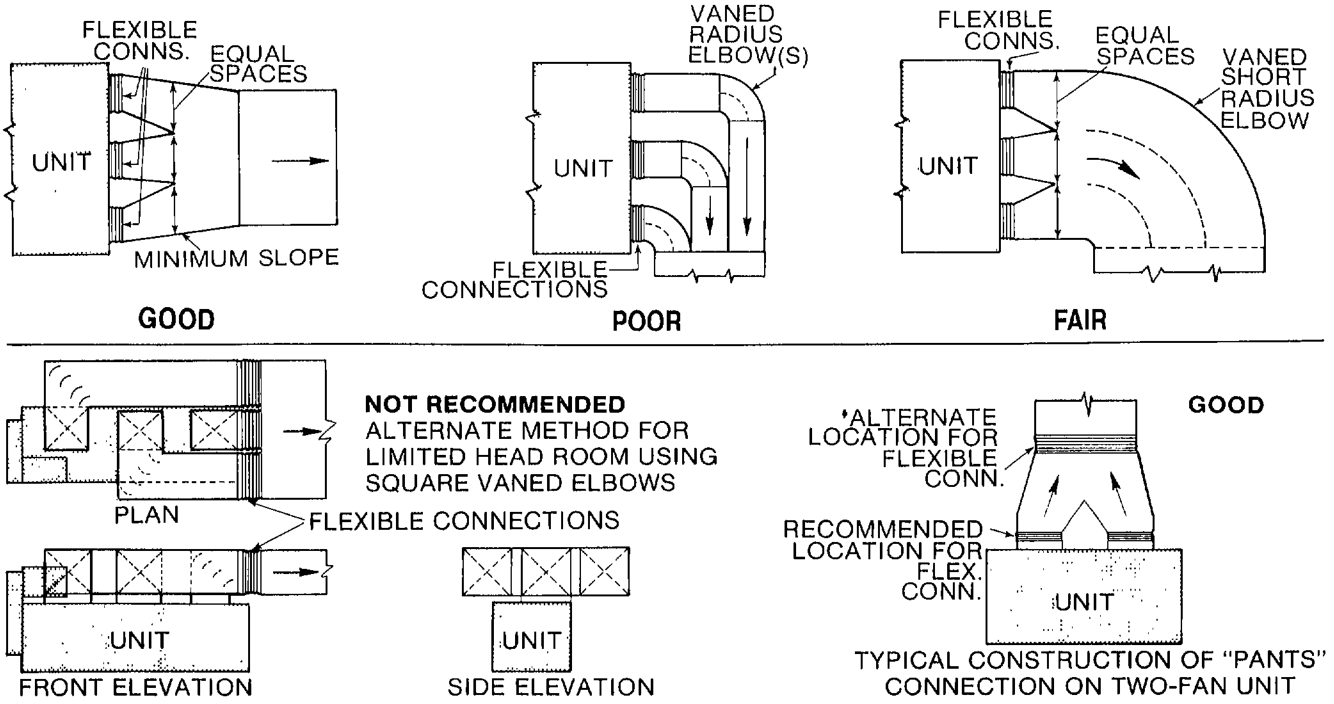
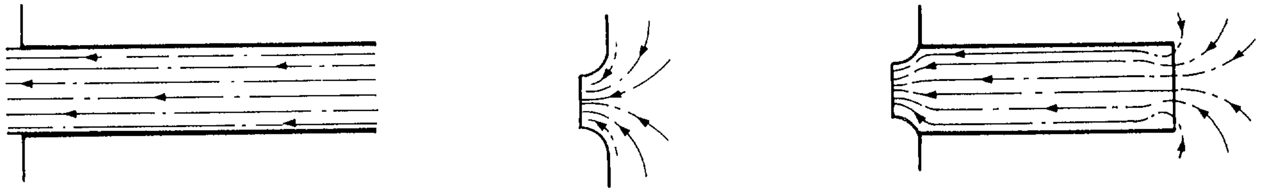


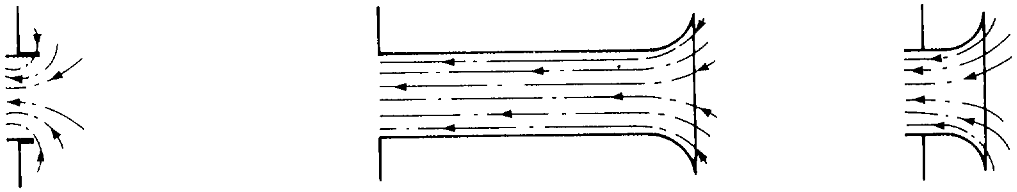
Figure 6-7 TYPICAL HVAC UNIT CONNECTIONS



a UNIFORM FLOW INTO FAN ON A DUCT SYSTEM

b UNIFORM FLOW INTO FAN WITH SMOOTH CONTOURED INLET

c VENA CONTRACTA AT DUCT INLET REDUCES PERFORMANCE



d VENA CONTRACTA AT INLET REDUCES EFFECTIVE FAN INLET AREA

e IDEAL SMOOTH ENTRY TO DUCT

f BELL MOUTH INLET PRODUCES FULL FLOW INTO FAN

Figure 6-8 TYPICAL INLET CONNECTIONS FOR CENTRIFUGAL AND AXIAL FANS (1)

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 and, therefore, the airstream resists a change in direction within an elbow as illustrated in Figures 6-9 & 6-10.

The System Effect Curves for round section elbows of 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.

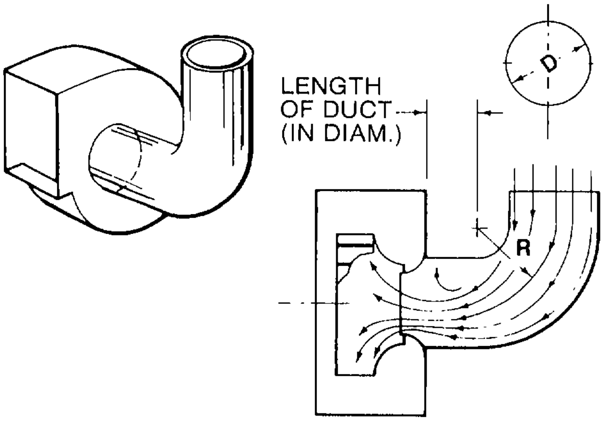
Refer to Figures 6-11 and 6-12 for the System Effect Curves for other inlet elbows and 90° duct turns which produce non-uniform inlet flow. Note that when duct turning vanes and/or a suitable length of duct is used (three to eight diameters long, depending on velocities) between the fan inlet and the 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, thereby, 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).

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 is one which 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 so 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.

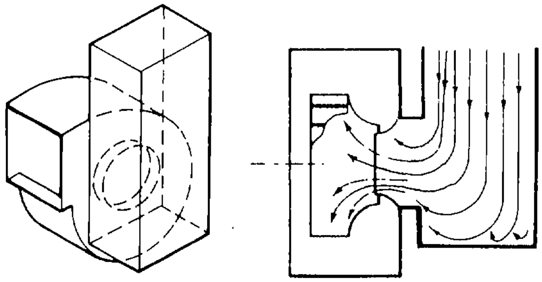
Inlet spin may arise from a great variety of approach conditions and sometimes the cause is not obvious. Some common duct connections which cause inlet spin are illustrated in Figure 6-14, but since the variations are many, no System Effect Factors are tabulated. It is recommended that these types of duct



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 6-9 NON-UNIFORM FLOW INTO A FAN INLET INDUCED BY A 90° ROUND SECTION ELBOW—NO TURNING VANES (1)



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 DEGREES AND THEREFORE THIS INLET SHOULD BE AVOIDED. CAPACITY LOSSES AS HIGH AS 45 PERCENT HAVE BEEN OBSERVED. EXISTING INSTALLATIONS CAN BE IMPROVED WITH VANES OR THE CONVERSION TO SQUARE OR MITERED ELBOWS WITH VANES.

Figure 6-10 NON-UNIFORM FLOW INDUCED INTO FAN INLET BY A RECTANGULAR INLET DUCT (1)

connections be avoided, but if this is not possible, inlet conditions can usually be improved by the use of vanes to break the spinning vortex (Figure 6-15).

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, from a single curved sheet metal van to multi-bladed "airfoil" vanes.

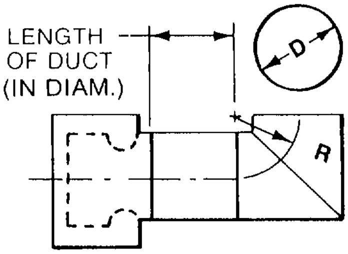
The pressure drop through elbows with these devices are part of the system pressure losses. The cataloged pressure loss of proprietary vanes will be based upon uniform airflow at the entry to the elbow. If the airflow approaching the elbow is significantly non-uniform because of the disturbance further upstream in the system, the pressure 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.

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 6-16, is reproduced from AMCA Standard 210.

6. Enclosures

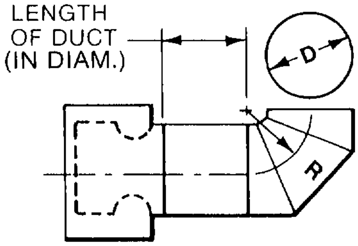
Fans within plenums and cabinets or next to walls should be located so that air may flow unobstructed into the inlets. Fan performance is reduced if the space between the fan inlet and the enclosure is too restrictive. It is common practice to allow at least one-half impeller diameter between an enclosure 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



SYSTEM EFFECT CURVES

R/D	NO DUCT	2D DUCT	5D DUCT
—	N	P	R-S

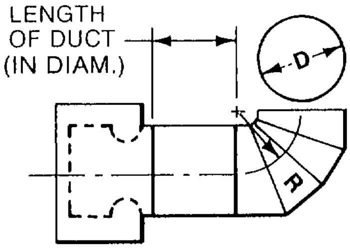
a TWO PIECE MITERED 90 ROUND SECTION ELBOW – NOT VANED



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

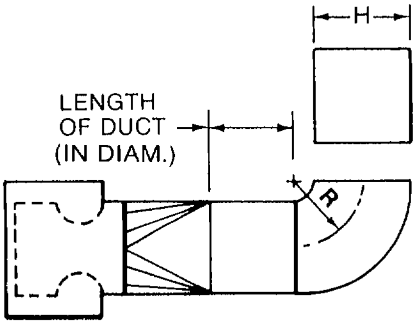
Figure 6-11 SYSTEM EFFECTS FOR VARIOUS MITERED ELBOWS WITHOUT VANES (1)

performance is to be expected. Figure 6-17 illustrates fans located 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/or inlet spin. Figure 6-18 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 6-19.

7. Obstructed Inlets

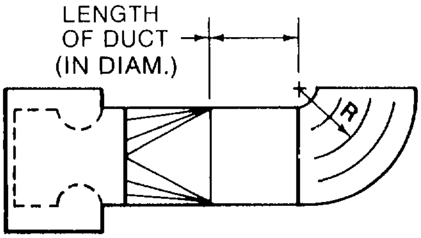
A reduction in fan performance can be expected when an obstruction to airflow is located in the plane of the fan inlet. Structural members, columns, butterfly valves, blast gates and pipes are examples of more common inlet obstructions.



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°, AND FOR A DIVERGING ELEMENT 7½°. $D = \frac{2H}{\sqrt{\pi}}$

Figure 6-12 SYSTEM EFFECTS FOR SQUARE DUCT ELBOWS (1)

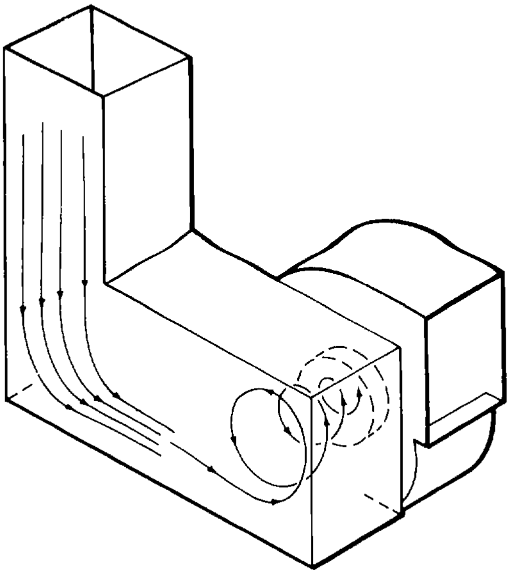
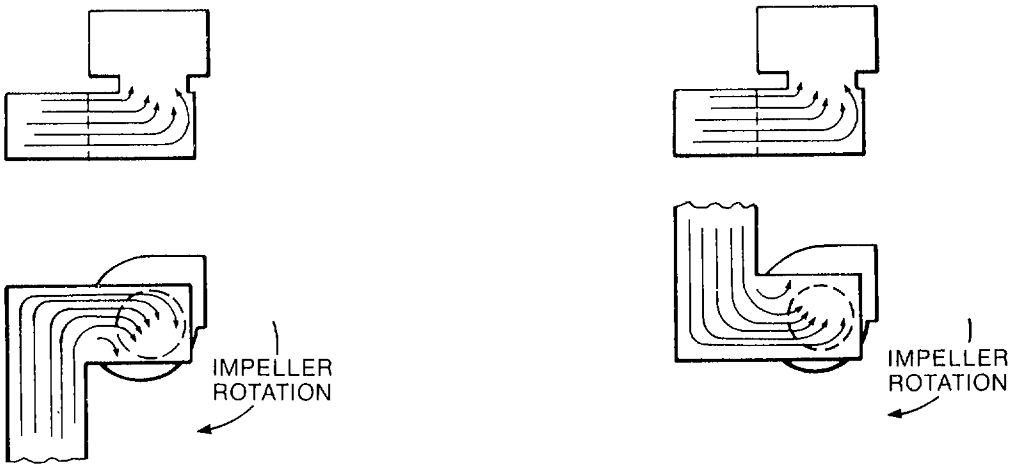


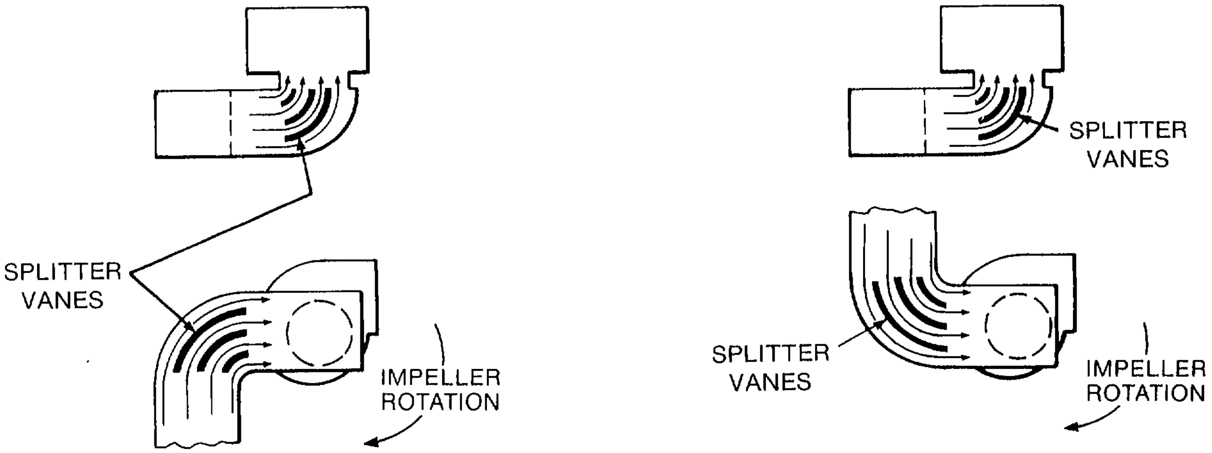
Figure 6-13 EXAMPLE OF A FORCED INLET VORTEX (SPIN-SWIRL) (1)



ROTATING SWIRL

COUNTER-ROTATING SWIRL

Figure 6-14 INLET DUCT CONNECTIONS CAUSING INLET SPIN (1)



CORRECTED
ROTATING SWIRL

CORRECTED COUNTER-
ROTATING SWIRL

Figure 6-15 CORRECTIONS FOR INLET SPIN (1)

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 conveniently in terms of the unobstructed percentage of the inlet area. Because of the shape of inlet cones of many fans, it is sometimes difficult to establish the area of the fan inlet. Figures 6-21 and 6-22 illustrate the convention adopted for this purpose. Where an inlet collar is provided (Figure 6-21) the inlet area is calculated from 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 (Figure 6-22).

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

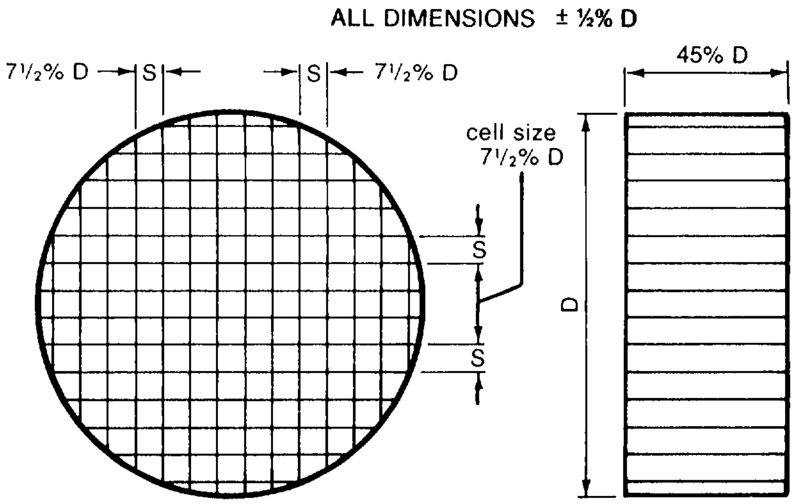
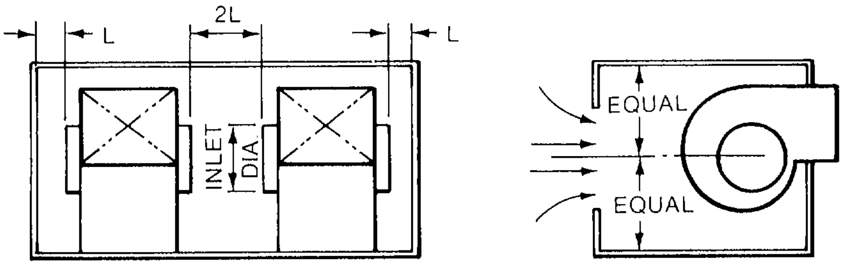


Figure 6-16 AMCA STANDARD 210 FLOW STRAIGHTENER (1)



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 6-17 SYSTEM EFFECT CURVES FOR FANS LOCATED IN PLENUMS AND CABINET ENCLOSURES AND FOR VARIOUS WALL TO INLET DIMENSIONS (1)

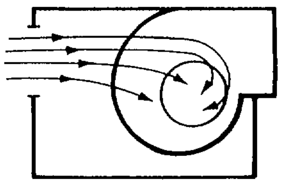


Figure 6-18 ENCLOSURE INLET NOT SYMMETRICAL WITH FAN INLET, PREROTATIONAL VORTEX INDUCED (1)

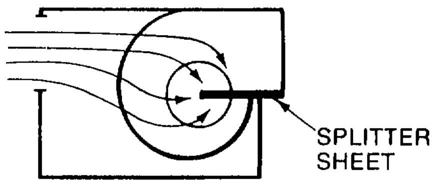


Figure 6-19 FLOW CONDITION OF FIGURE 6-18 IMPROVED WITH A SPLITTER SHEET (1)

Curve chart and the System Effect Factor determined from the curve listed for that unobstructed percentage of the inlet area.

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 6-20 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° connection to the fan with almost no horizontal duct.

The inlet box should be made of a metal gauge 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 connection to the inlet of the box. This requires the box to be adequately supported by the fan base and the 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 used 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.

C 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.

If possible, the necessary information should be obtained directly from the fan manufacturer. The data presented in this section are offered only as a guide in the absence of specific data from the fan manufacturer.

1. Bearing Supports

Some fans require that the fan shaft be supported by a bearing and bearing support in the fan inlet or just adjacent to it.

These components may have an effect on the airflow to the fan inlet and, consequently, on the fan performance, depending on the size of the bearings and supports in relation to the fan inlet opening. The location of the bearing and support, that is, whether it is located in the actual inlet sleeve or "stepped out" from the inlet, will also have an effect.

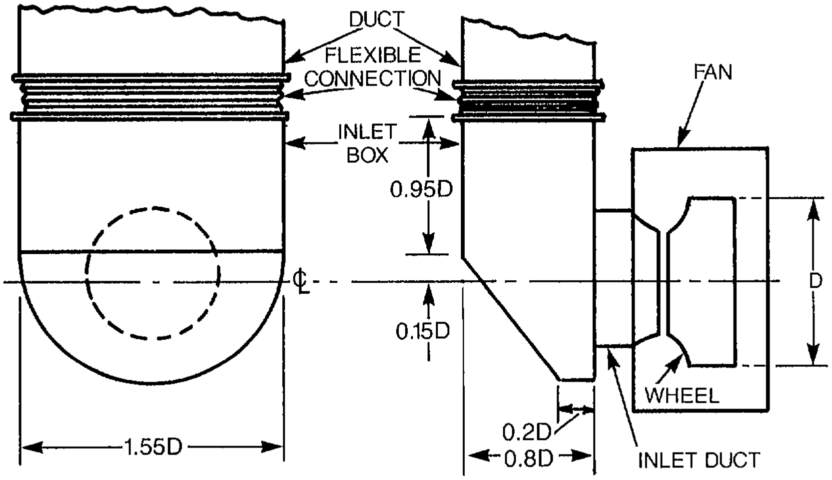


Figure 6-20 CENTRIFUGAL FAN INLET BOX

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, if possible by use of the fan manufacturer's allowance for bearings in the fan inlet.

If no better data is available, an approximation may be made as described under "Obstructed Inlets" in subsection B of this section.

2. Drive Guards

Most fans may 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 "stepped out" bearing pedestals.

In any case, depending on the location of the guard and on the inlet velocity, the fan performance may be significantly affected by this obstruction.

It is desirable that a drive guard located in this position 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 6-21, 6-22, and Table 6-3.

Where possible, open construction on guards is recommended to allow free air passage to the inlet.

Guards and sheaves should be designed to obstruct as little of the inlet as possible and in no case should the obstruction be more than 1/3 of the inlet area.

3. Belt Tube in Axial Fans

With a belt-driven axial flow fan, it is usually necessary that the fan motor be mounted outside the fan housing.

To protect the belts from the airstream and also to prevent any leakage from the fan housing, manufacturers, in many cases, provide a belt tube.

Most manufacturers include the effects of this belt tube in their rating tables; however, in cases where this is not reflected, the appropriate System Effect Curves obtained from Table 6-3 may be used.

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 that, 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.

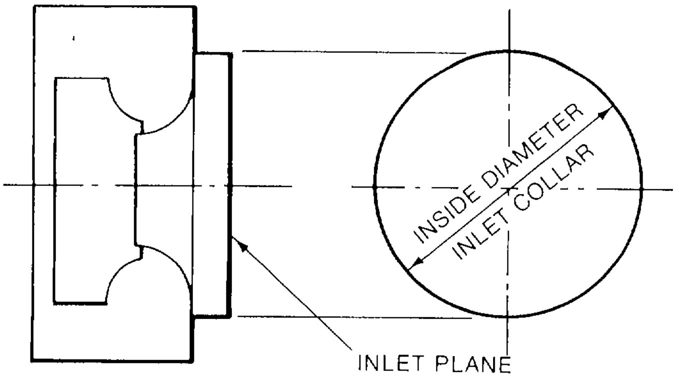


Figure 6-21 FREE INLET AREA PLANE—FAN WITH INLET COLLAR (1)

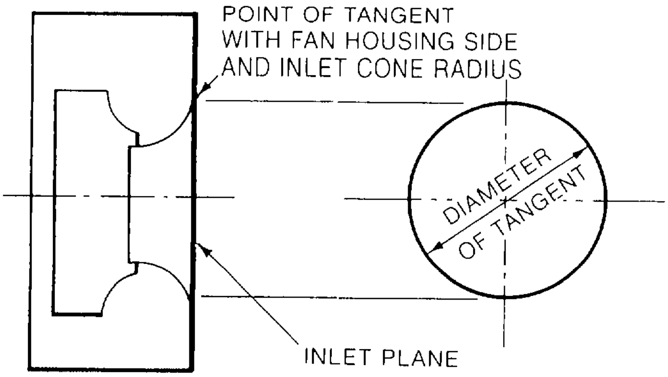


Figure 6-22 FREE INLET AREA PLANE—FAN WITHOUT INLET COLLAR (1)

Table 6-3 SYSTEM EFFECT CURVES FOR INLET OBSTRUCTIONS (1)

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

The opposed blade type is used to control airflow volume by changing the system by 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" or "T" from Figure 6-1 should be applied in making the fan selection.

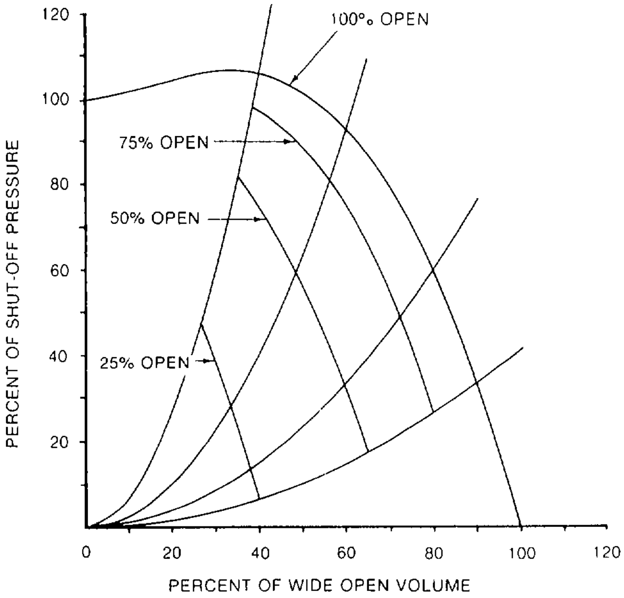


Figure 6-23 TYPICAL NORMALIZED INLET VALVE CONTROL PRESSURE—VOLUME CURVE (1)

5. Inlet Vane Control

To maintain fan efficiency at reduced flow conditions, airflow quantity is often controlled by variable vanes mounted in the fan inlet (see Figure 6-24).

These are arranged to generate a forced inlet vortex which rotates in the same direction as the fan impeller.

Inlet vanes may be of two different basic types:

1. Integral (built-in)
2. 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, the System Effect Curves of Table 6-4 should be applied in making the fan selection using Figure 6-23.

D 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 Factors can be noted and included in the system total pressure loss/fan capacity calculations.

Using a fan similar to that in the duct system example in Figures 7-2 or 8-2 of Chapters 7 or 8, 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), Sys-

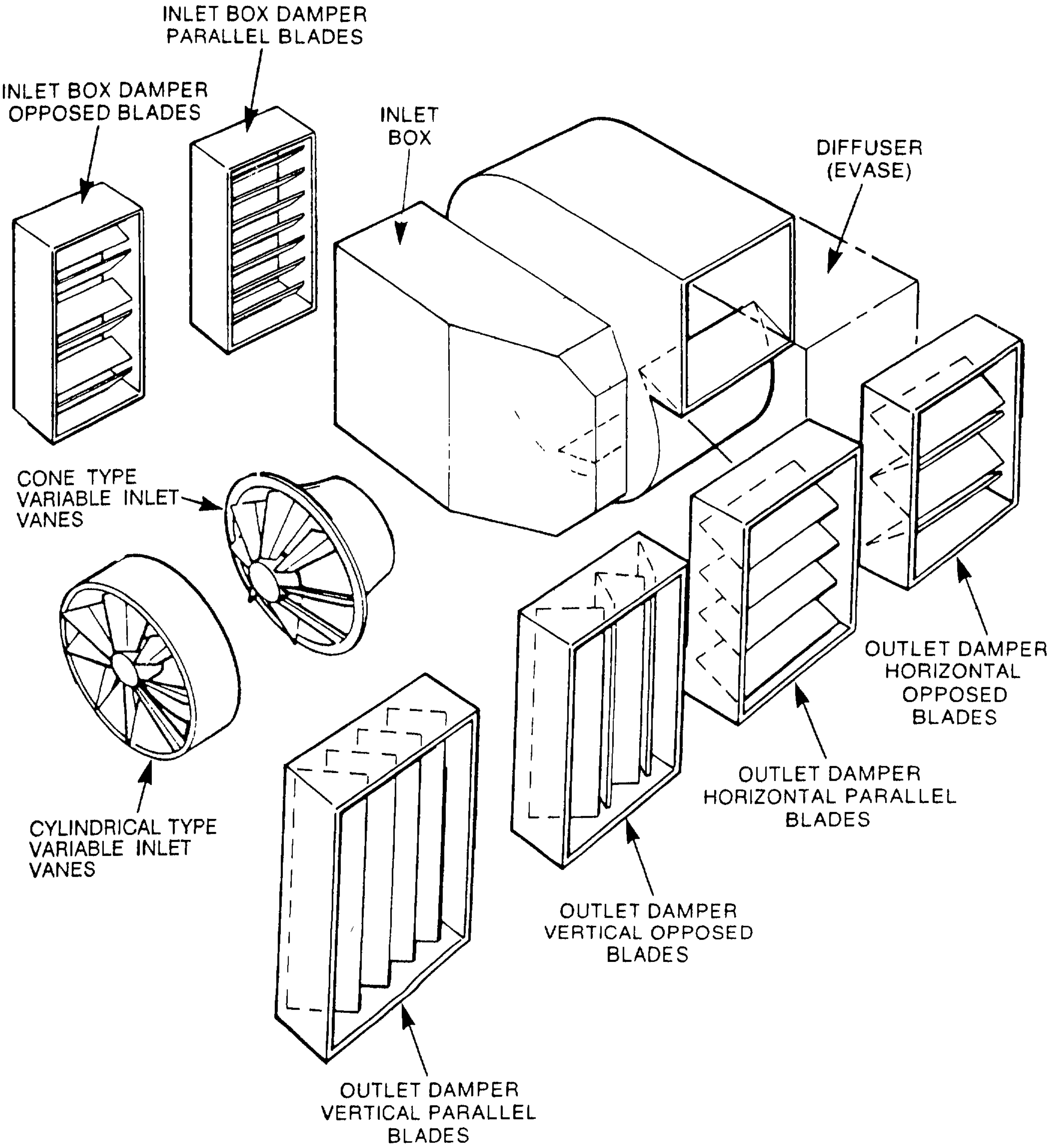


Figure 6-24 COMMON TERMINOLOGY FOR CENTRIFUGAL FAN APPURTENANCES (1)

Table 6-4 (1)

Vane Type	System Effect Curve
1. Integral (built in)	"Q" or "R"
2. Cylindrical (add on)	"S"

tem 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 inch (1220 mm) SWSI fan has an inlet and outlet area of 13.1 square feet (1.22 m^2) each. At 20,000 cfm (10,000 l/s) and 2.4 in. w.g. (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. w.g. (57 Pa) for the inlet side of the fan. This becomes part of the static pressure *derating* of the fan.

The fan discharge size for *this* example is 43 inches (1092 mm) wide by 44 inches (1118 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 inches (760 mm) from the fan discharge, which would result in an approximately "25% 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. w.g. (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). Then the velocity in the duct has no relationship with the fan discharge velocity unless it falls within the parameters discussed earlier in "Outlet Ducts" of Subsection A.

It is important to note again 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.

CHAPTER 7

DUCT SIZING PROCEDURES (U.S. UNITS)

A DESIGN FUNDAMENTALS

For duct sizing procedures using S.I. units or the metric system, see chapter 8.

1. The total pressure (TP) at any location within a system is the sum of the static pressure (SP)and the velocity pressure (V_p).
2. Total pressure *always* decreases algebraically in the direction of airflow (negative values of return air or exhaust systems increase in the direction of airflow, and positive values of supply air systems decrease in the direction of airflow). See *Figure 5-10* and the text on page 5.11.
3. The losses in total pressure between the fan and the end of *each* branch of a system are the same.
4. Static pressure and velocity pressure are mutually convertible and either can increase or decrease in the direction of flow.

B DESIGN OBJECTIVES

1. 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.
2. Consider energy conservation in the fan selection, duct configuration, duct wall heat gain or loss, etc.
3. Special consideration should be given to the need for sound attenuation and breakout noise.
4. Testing, adjusting and balancing equipment and dampers should be shown on the drawings.
5. Locations of all life safety devices such as fire dampers, smoke dampers, etc. should be shown on the drawings.
6. The designer should consider the pressure losses that occur from tie rods and other duct obstructions.
7. If the ductwork is well designed and constructed, at least 75 to 90 percent of the original velocity pressure can be regained.
8. Round ducts generally are preferred for higher pressure systems.
9. Branch takeoffs and fittings with low loss coefficients should be used. Both 90° and 45° duct takeoffs can be used. However, the use of conical tees or angular takeoffs can reduce pressure losses.
10. Use of the SMACNA Duct Design Calculators would aid the duct design process, especially when making changes in the field.

C DUCT SYSTEM SIZING PROCEDURES

1. Introduction

The "equal friction" method of duct sizing probably has been the most universally used means of sizing low pressure supply air, return air and exhaust air duct systems and it is being adapted by many for use in medium pressure systems. It normally has not been used for sizing high pressure systems. This design method "automatically" reduces air velocities in the direction of the airflow, so that 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 readily may be checked at any point. There is then the opportunity to reduce velocity created noise by increasing duct size or adding sound attenuation materials (such as duct lining).

The major disadvantages of the equal friction method are: (1) there is no natural provision for equalizing pressure drops in the branches (except in the few cases of a symmetrical layout); and (2) there is no provision for providing the same static pressure behind each supply or return terminal device. Consequently, balancing can be difficult, even with a considerable amount of dampering in short duct runs. However, the equal friction method can be modified by designing portions of the longest run with different friction rates from those used for the shorter runs (or branches from the long run).

Static regain (or loss) due to velocity changes, has been added to the equal friction design procedure by using fitting pressure losses calculated with new loss coefficient tables in Chapter 14. Otherwise, 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 the 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 Chapter 14.**

2. Modified Equal Friction Design Procedures

"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 equivalent feet of duct is selected before the ductwork is laid out, and that this loss per 100 feet is 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 air duct leaving the fan, the usual procedure is to select an initial velocity from the chart in Figure 14-1. This velocity could be selected above the shaded section of Figure 14-1 if higher sound levels and energy conservation are not limiting factors. The chart in Figure 14-1 is used to determine the friction loss by using the design air quantity (cfm) and the selected velocity (fpm). A friction loss value commonly used for lower pressure duct sizing is 0.1 in. of water (in.w.g.) per 100 equivalent feet of ductwork, although 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 14-1.

The friction losses of each duct section should be corrected for other materials and construction methods by use of Table 14-1 and Figure 14-3. The correction factor from Figure 14-3 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 are then used to select the equivalent rectangular duct sizes from Table 14-2, unless round ductwork is to be used.

The flow rate (cfm) in the second section of the main supply duct, after the first branch takeoff, is the original cfm supplied by the fan reduced by the amount of cfm into the first branch. Using Figure 14-1, the new flow rate value (using the recommended friction rate of 0.1 in. w.g. per 100 ft.) will determine the duct velocity and diameter for that section. The equivalent rectangular size of that duct section again is obtained from Table 14-2 (if needed). All subsequent sections of the main supply duct and all branch ducts can be sized from Figure 14-1 using the same friction loss rate and the same procedures.

The total pressure drop measured at each terminal device or 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. w.g. If the pressure difference between the terminals exceeds that amount, dampering would be required that could create objectionable air noise levels.

The modified equal friction method is used for sizing duct systems that are not symmetrical or that 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 (as nearly as possible) 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 a good design practice to include balancing dampers in HVAC duct systems to balance the airflow to each branch.

3. Fitting Pressure Loss Tables

Tables 14-10 to 14-18 contain 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 picked up 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

$$TP = C \times V_p$$

Where:

TP = Total Pressure (in. w.g.)
C = Dimensionless Loss Coefficient
V_p = Velocity Pressure (in. w.g.)

By using the duct fitting loss coefficients in Chapter 14 which include static pressure regain or loss, accurate duct system fitting pressure losses are obtained. When combined with the static pressure friction losses of the straight duct sections 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 which has a velocity of 2550 fpm. Using Table 14-6, the velocity pressure (V_p) is found to be 0.41 in. w.g. The total pressure (TP) loss of each fitting is determined as follows:

Example A:

36″ (H) × 12″ (W), 90° Radius Elbow (R/W = 1.5), no vanes. From Table 14-10, Figure F, the loss coefficient of 0.14 is obtained using H/W = 3.0.

The loss coefficient should not be used without checking to see if a correction is required for the Reynolds number (Note 3):

$$D = \frac{2\,HW}{H + W} = \frac{2 \times 36'' \times 12''}{36'' + 12''} = 18''$$

$$R_e = 8.56 \times DV$$

$$R_e = 8.56 \times 18 \times 2550 = 392{,}904$$

$$R_e 10^{-4} = \frac{392{,}904}{10^4} = 39.29$$

The correction factor of 1.0 is found where R/W > 0.75 and $R_e\ 10^{-4} > 20$; so the loss coefficient remains at 0.14. Then:

$$TP = C \times V_p = 0.14 \times 0.41 = 0.057 \text{ in. w.g.}$$

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 14-19 had been checked, as the plotted point is outside the shaded area requiring correction (using the duct diameter and velocity to plot the point).

If the elbow was 45° instead of 90°, another correction factor of 0.60 (See the reference to Note 1 on page 14.19) would be used: 0.60 × 0.057 = 0.034 in. w.g.

Example B:

45° Round Wye, 20″ diameter main duct, (2500 fpm); 10″ diameter branch duct, branch velocity of 1550 fpm. Determine the fitting pressure losses. (Figure A of Table 14-14).

$$A_b = \pi r^2 = \pi 5^2 = \pi 25$$

$$A_c = \pi r^2 = \pi 10^2 = \pi 100$$

$$A_b/A_c = 25/100 = 0.25$$

From Figure 14-1:

For 10″ diameter, 1550 fpm; $Q_b = 850$ cfm

For 20″ diameter, 2500 fpm; $Q_c = 5500$ cfm

$$Q_b/Q_c = 850/5500 = 0.155$$

Interpolating in the table between $A_b/A_c = 0.2$ and 0.3; and $Q_b/Q_c = 0.1$ and 0.2; 0.56 is selected as the branch fitting loss coefficient. The branch pressure loss is calculated.

Obtain V_p of 0.39 for 2500 fpm from Table 14-6.

$$TP = C \times V_p = 0.56 \times 0.39 = 0.218 \text{ in. w.g.}$$

The main pressure loss is calculated by first establishing V_s:

$$Q_s = Q_c - Q_b = 5500 - 850 = 4650 \text{ cfm}$$

Using Figure 14-1, 20″ diameter:

$$V_s = 2120 \text{ fpm}$$

$$V_s/V_c = 2120/2500 = 0.85$$

From the Table 14-14, Figure A, $C = 0.02$

$$TP = C \times V_p = 0.02 \times 0.39 = 0.008 \text{ in. w.g.}$$

Example C:

36″ × 12″ rectangular to 20″ diameter round transition where $\Theta = 30°$ (Table 14-12, Figure A), $V_p = 0.4$.

$$A_1 = 36 \times 12 = 432 \text{ sq. in.}$$

$$A = \pi r^2 = \pi 10^2 = 314 \text{ sq. in.}$$

$$A_1/A = 1.38 \text{ (use 2)}$$

0.05 is selected as the loss coefficient.

$$TP = C \times V_p = 0.05 \times 0.4 = 0.02 \text{ in. w.g.}$$

Fortunately, there usually are not too many "complicated" fittings in most duct systems, but when there are, the systems usually are part of a large complex.

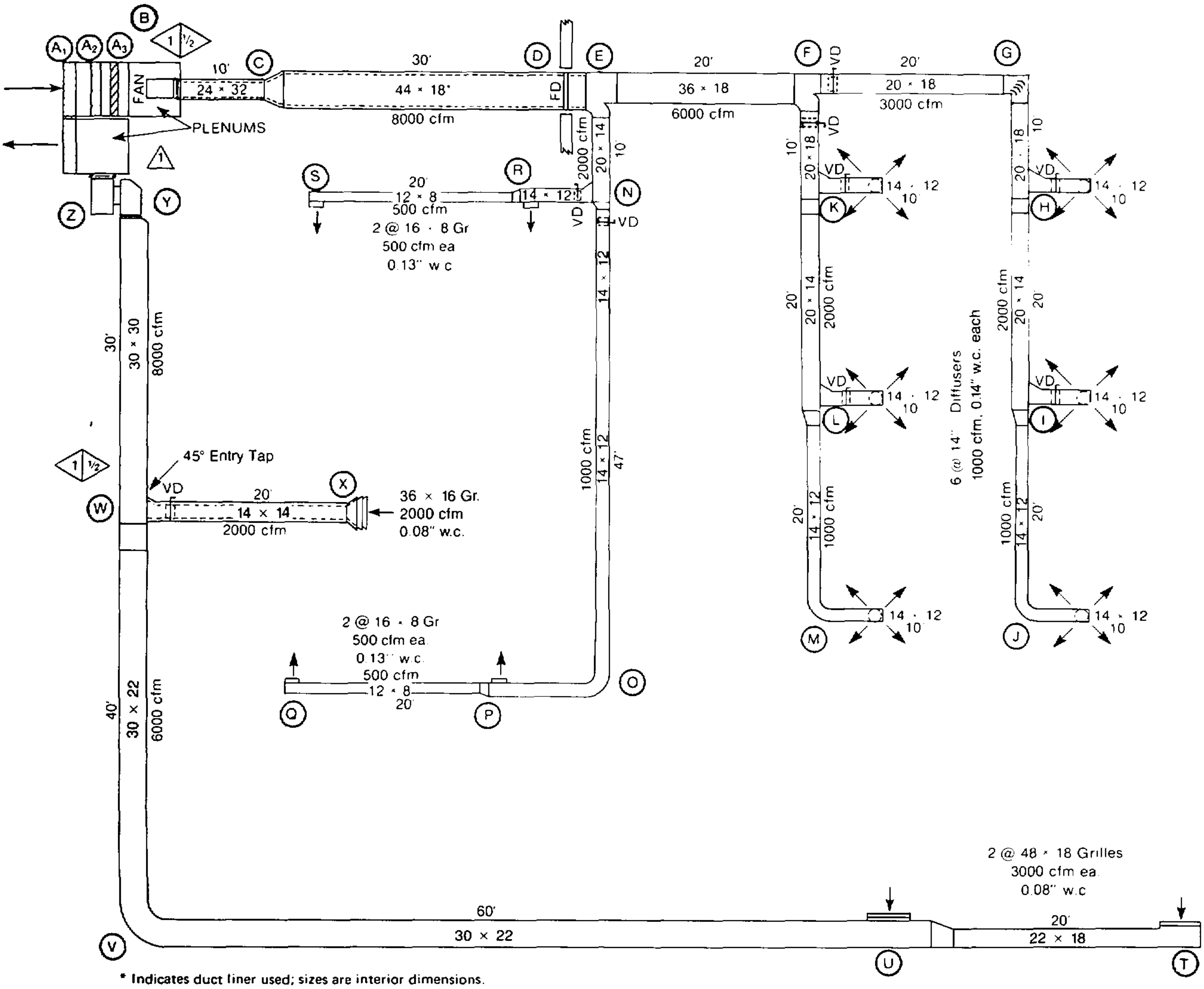
A computer programmed for the above calculations can facilitate the duct system design procedure.

D SUPPLY AIR DUCT SYSTEM-SIZING EXAMPLE NO. 1

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 at the end of this 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 8000 cfm each. The grilles and diffusers have been tentatively sized to provide the required flow, throw, noise level, etc., 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.

1. Supply Fan Plenum

From manufacturer's data sheets or from the Figures or Tables in Chapter 9, 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 feet of duct discharging directly from fan "B" (duct is fan outlet size), no "System Effect Factor" (see Chapter 6) needs to be added for either side of the fan. As the plenum



* Indicates duct liner used; sizes are interior dimensions.

Figure 7-1 DUCT SYSTEMS FOR DUCT SIZING EXAMPLES NO 1 AND 2.

Table 7-1 DUCT SIZING, SUPPLY AIR SYSTEM—
EXAMPLE NO. 1

DUCT SIZING WORK SHEET
(U.S. Units)

DATE ______ PAGE 1 OF 2

PROJECT SAMPLE BUILDING LOCATION FIRST FLOOR SYSTEM SUPPLY AIR

	A DUCT RUN	B SEC-TION	C ITEM		D FLOW CFM	E FRICTION PER 100'	F VELOCITY FPM	G Vp	H LOSS COEFF.	I EQUIV. DIAM.	J RECTANGULAR SIZE	K CORR. FACT.	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1	PLENUM B	A₁	–	H.R. DEVICE	8000	–	500	–	–	–	–	–	0.30		1.382	1.382
2	"	A₂	–	FILTERS	8000	–	400	–	–	–	–	–	0.15			(A,J)
3	"	A₃	–	COIL	8000	–	550	–	–	–	–	–	0.45	0.90		
4	RUN BJ	BC	10'	DUCT	8000	0.095	1600	–	–	30.2	*24x32	1.40	0.013	0.013	0.492	
5	"	CE	30'	DUCT	8000	0.095	1600	–	–	30.2	*44x18	1.40	0.040		0.479	
6	"	C	–	TRANS.	8000	–	1500	0.14	0.25	–	24x32/44x18	–	0.035			
7	"	D	–	F. DAMP.	8000	–	–	–	–	–	44x18	–	0.06	0.135		
8	"	EF	20'	DUCT	6000	0.095	1500	–	–	27.4	36x18	–	0.019		0.344	
9	"	E	90°	WYE	8000/6000	–	1455	0.13	-0.01	–	44x18/36x18	–	-0.001	0.018		
10	"	FH	30'	DUCT	3000	0.095	1260	–	–	20.7	20x18	–	0.029		0.326	
11	"	F	90°	WYE	6000/3000	–	1333	0.11	0.05	–	36x18/20x18	–	0.006			
12	"	F	–	VOL. DAMP	3000	–	1200	0.09	0.04	–	20x18	–	0.004			
13	"	G	90°	ELBOW	3000	–	1200	0.09	0.24	–	20x18	–	0.022	0.061		
14	"	HI	20'	DUCT	2000	0.095	1140	–	–	18.2	20x14	–	0.019		0.265	
15	"	H	–	TRANS.	3000/2000	–	1200	0.09	0.05	–	20x18/20x14	–	0.005	0.024		
16	"	IJ	30'	DUCT	1000	0.080	900	–	–	14.2	14x12	–	0.024		0.241	
17	"	I	–	TRANS	2000/1000	–	1029	0.07	0.05	–	20x14/14x12	–	0.004			
18	"	J	90°	ELBOW	1000	–	857	0.05	0.16	14.2	14x12	1.32	0.011			
19	"	J	–	VOL. DAMP	1000	–	857	0.05	0.04	–	14x12	–	0.002			
20	"	J	–	TAP FIT.	1000	–	857	0.05	1.20	–	14x12/14φ	–	0.060			
21	"	J	–	DIFF.	1000	–	–	–	–	–	14φ	–	0.140	0.241		
22																
23																
24																
25																

NOTES: *Indicates duct lining used. Sizes are interior dimensions.

static pressure (SP) loss is negligible, the losses for the inlet air portion of the fan system entered in column L are added, and the loss of 0.90 in. w.g. is entered in column M on line 3.

2. Supply Air System

a) Duct Section BC—The 24″ × 32″ fan discharge size has a circular equivalent of 30.2 inches (Table 14-2). Using the chart in Figure 14-1, a velocity of 1600 fpm and a friction loss of 0.095 in. w.g. per 100 ft. of duct is 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 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 entered in column J is marked with an asterisk and the fibrous glass liner "medium rough" correction factor of 1.40 is obtained from Table 14-1 and Figure 14-3 and entered in column K. Duct sections BC static pressure (SP) loss is computed as follows:

$$\text{SP (duct section)} = 10' \text{ (duct)} \times \frac{0.095 \text{ in. w.g.}}{100 \text{ ft.}}$$

$$\times\ 1.40 \text{ (corr. factor)} = 0.013 \text{ in. w.g.}$$

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.

b) Duct Section CE—At point C, building construction conditions require that the duct aspect ratio change, so a duct transition is needed. Using the same 0.095 in. w.g. per 100 ft. duct friction loss and 30.2 in. duct diameter for the 8000 cfm airflow, a 44″

Table 7-1(a) DUCT SIZING, SUPPLY AIR SYSTEM— EXAMPLE NO. 1 (CONT.)

DUCT SIZING WORK SHEET (U.S. Units)

DATE ______ PAGE 2 OF 2

PROJECT SAMPLE BUILDING LOCATION FIRST FLOOR SYSTEM SUPPLY AIR

	A DUCT RUN	B SECTION	C ITEM		D FLOW CFM	E FRICTION PER 100'	F VELOCITY FPM	G Vp	H LOSS COEFF	I EQUIV DIAM	J RECTANGULAR SIZE	K CORR FACT	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1	A, F	-	-	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	–	36x18/20x18	-	0.057		0.336	(A,M)
3	"	F		VOL DAMP	3000	–	1200	0.09	0.04	–	20x18	-	0.004			
4	"	FK	10'	DUCT	3000	0.095	1260	–	–	20.7	20x18	-	0.010	0.071		
5	"	KM	-	TOTAL OF LINES 14-21 (SHEET 1 OF 2)								–	–	0.265	0.265	
6																
7	A, E	-	-	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	20x14	-	0.010		0.318	(A,Q)
9	"	E	90°	WYE	8000/2000	-	1455	0.13	0.43	–	44x18/20x14	-	0.056	0.066		
10	"	NP	55'	DUCT	1000	0.080	900	–	–	14.2	14x12	-	0.044		0.252	
11	"	N	-	TRANS	2000/1000	–	1029	0.07	0.05	–	20x14/14x12	-	0.004			
12	"	N	-	VOL DAMP	1000	-	857	0.05	0.04	–	14x12	–	0.002			
13	"	O	90°	ELBOW	1000	–	857	0.05	0.12	–	14x12	–	0.006	0.056		
14	"	PQ	20'	DUCT	500	0.095	810	–	–	10.7	12x8	–	0.019		0.196	
15	"	P	–	TRANS	1000/500	-	857	0.05	0.06	–	14x12/12x8	–	0.003			
16	"	Q	90°	ELBOW	500	–	750	0.04	1.00	–	12x8	1.09	0.044			
17	"	Q	-	GRILLE	500	–	–	-	–	–	16x8	-	0.130	0.196		
18	A, N	–	-	TOTAL OF LINES 7-9 (ABOVE)								–	–	1.115	1.371	1.371
19	RUN NS	NR	8'	DUCT	1000	0.080	900	–	–	14.2	14x12	–	0.006		0.256	(A,S)
20	"	N	45°	ENT TAP	2000/1000	-	1029	0.07	0.74	–	20x14/14x12	–	0.052			
21	"	N	-	VOL DAMP	1000	-	857	0.05	0.04	–	14x12	–	0.002	0.060		
22	"	RS	20'	DUCT	500	0.095	810	–	–	10.7	12x8	-	0.019		0.196	
23	"	R	–	TRANS	1000/500	–	857	0.05	0.06	–	14x12/12x8	–	0.003			
24	"	S	90°	ELBOW	500	-	750	0.04	1.00	–	12x8	1.09	0.044			
25	"	S	–	GRILLE	500	-	–	–	–	–	16x8	-	0.130	0.196		

NOTES: *Indicates duct lining used. Sizes are interior dimensions.

× 18″ duct is selected from Table 14-2 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:

$$SP = 30' \times \frac{0.095 \text{ in. w.g.}}{100 \text{ ft.}} \times 1.40$$
$$= 0.040 \text{ in w.g.}$$

(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 × 32 = 768 and A_1 = 44 × 18 = 792, $A_1/A = \frac{792}{768} = 1.03$, so it is diverging (greater than 1.0).

The average velocity of the entering airstream (Equation 5-7) = Q/A or cfm/Area (ft.) = 8000/24 × 32/144 = 1500 fpm.

From Table 14-11, Figure B, using Θ = 30° 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 0.14 in. w.g. is obtained from Table 14-6 for 1500 fpm and entered in column G. The transition fitting pressure loss of 0.035 in. w.g. ($C \times V_p = 0.25 \times 0.14$) 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. w.g. for the fire damper at D is obtained from Chapter 9 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 totalled (0.133 in. w.g.) and entered in column M on line 7. This is the total pressure loss of the 44″ × 18″ duct section CE (inside dimensions) and its components.

c) 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 6,000 cfm (for duct section EF) and 0.095 in. w.g. per 100 ft. to the chart in Figure 14-1, a duct diameter of 27.1 in. and 1500 fpm velocity is obtained and entered on line 8. Table 14-2 is used to select a 36″ × 18″ rectangular duct size needed by keeping the duct height 18 inches (equivalent duct diam. = 27.4 in.). Normally, duct size changes are made changing only one dimension (for ease and economy of fabrication) and keeping the aspect ratio as low as possible. The use of 27.4 in. instead of 27.1 in. does not change the velocity (including velocity pressure) or duct friction losses significantly to require the use of different values. A review of the chart in Figure 14-1 will verify this, so 1500 fpm and 0.095 in. w.g. will continue to be used.

As the continuous rolled galvanized duct system is being fabricated in 4 foot sections, the degree of roughness (Table 14-1) indicates "medium smooth". No correction factor is needed, as the chart in Figure 14-1 is based on an Absolute Roughness of 0.0003 ft. as a result of recent SMACNA assisted ASHRAE research.

The static pressure loss for duct section EF is:

$$SP = \frac{20 \text{ ft.} \times 0.095}{100 \text{ ft.}} = 0.019 \text{ in. w.g.}$$

(enter on line 8 column L)

The diverging 90° wye fitting used at E can be found in Table 14-14, 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 (Prelim.) for 2,000 cfm @ 0.095 in. w.g. = 254 sq. in. (area of 18.0 in. diameter duct obtained from Figure 14-1). Then:

$A_b/A_s = (9.0)^2\pi/(13.7)^2\pi = 254/590 = 0.43,$

$A_b/A_c = 254/707 = 0.36,$ and

$Q_b/Q_c = 2000/8000 = 0.25.$

Using $A_b/A_s = 0.33$; and $A_b/A_c = 0.25$ (the closest figures), C (Main) = − 0.01 (obtained by interpolation). The V_p for 1455 fpm (8000/44 × 18/144) is 0.13 in. w.g.

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. w.g.) and entered on line 9 in column M.

d) 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. w.g. and 3000 cfm in Figure 14-1, 1260 fpm and 20.7 inches diameter are obtained from Figure 14-1; 20″ × 18″ equiv. duct size from Table 14-2:

$$\text{FH duct section loss} = \frac{30 \text{ ft.} \times 0.095}{100 \text{ ft.}}$$
$$= 0.029 \text{ in. w.g.}$$

(enter on line 10)

For the wye fitting at F, Table 14-14, figure W is again used. With the 6000 cfm airflow dividing equally into two 3000 cfm airstream ducts, $A_b = A_s$. Therefore,

$A_b/A_s = 1.0;\ A_b/A_c = (10.5)^2\ \pi/(13.7)^2\ \pi$
$= 346/590 = 0.59$

$Q_b/Q_c = 3000/6000 = 0.5$

Using $A_b/A_s = 1.0$; $A_b/A_c = 0.5$;

C (Main) = 0.05,
velocity = 1333 fpm (6000/36 × 18/144)

$V_p = 0.11$ (From Table 14-6)

Fitting loss = C × V_p = 0.05 × 0.11
= 0.006 in. w.g. (line 11)

The loss coefficient for the thin plate volume damper near F can be obtained from Table 14-18, Figure B (Set wide open, i.e. 0°). The velocity pressure (V_p) of 0.09 in. w.g. for 1200 fpm (3000/20 × 18/144) is obtained from Table 14-6.

Damper Loss = C × V_p = 0.04 × 0.09
= 0.004 in. w.g. (line 12)

Elbow G in the FH duct run is a square elbow with 4.5 inch single thickness turning varies on 3 ¼ inch centers. The loss coefficient of 0.24 is obtained from Table 14-10, Figure H for the 20" × 18" elbow and entered on line 13 along with the other data (cfm, fpm, V_P, etc.)

G fitting loss = C × V_P = 0.24 × 0.09
= 0.022 in w.g. (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. w.g. is entered on line 13 in column M.

e) 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 inch diameter (and the duct size of 20″ × 14″) are obtained (again changing only one duct dimension where possible).

Then, HI duct section loss $= \dfrac{20 \text{ ft.} \times 0.095}{100}$
$= 0.019$ in. w.g. (line 14).

The loss coefficient for transition H (converging flow) is obtained from Table 14-12, Figure A using $\Theta = 30°$ (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).

$\text{Vel} = 3000/20 \times 18/144 = 1200 \text{ fpm}; V_p = 0.09,$

$$\frac{A_1}{A} = \frac{20'' \times 18''}{20'' \times 14''} = 1.29; C = 0.05$$

H fitting loss $= C \times V_p = 0.05 \times 0.09$
$= 0.005$ in. w.g. (line 15)

The loss values in column L (0.019 and 0.005) are again totalled and entered on line 15 in column M (0.024 in. w.g.).

f) Duct Section IJ—Duct section IJ is calculated as the above duct sections and the same type of transition is used (1000 cfm, 970 fpm, 13.9 inch diam.; with a 14″ × 12″ duct size being selected at a 14.2 inch diameter Equivalent):

$$\text{IJ duct loss} = \frac{30 \text{ ft.} \times 0.095}{100 \text{ ft.}} = 0.029 \text{ in. w.g.}$$

If the 14.2 in. circular equivalent of the 14″ × 12″ duct is reploted on the chart in Figure 14-1 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:

$$\text{IJ duct loss} = \frac{30 \times 0.080}{100 \text{ ft.}} = 0.024 \text{ in. w.g.}$$

As the new value is 0.003 in. w.g. less (a somewhat significant amount), the 0.024 in w.g. is entered on line 16. However, if this were done on a computer, the larger (safer) amount would be used.

Transition at I (Table 14-12, Figure A):

$$\frac{A_1}{A} = \frac{20 \times 14}{14 \times 12} = 1.67; C = 0.05$$

$\text{Velocity} = 2000/20 \times 14/144 = 1029 \text{ fpm};$
$V_p = 0.07,$

I fitting loss $= C \times V_p = 0.05 \times 0.07$
$= 0.004$ in. w.g. (line 17).

The "J" elbow is smooth, long radius without vanes (Table 14-10, Figure F) having a R/W ratio of 2.0. As H/W = 12/14 = 0.86, the loss coefficient of 0.16 is used.

By applying values of the 14.2 inch equivalent duct diameter and the duct velocity of 900 fpm to the "Reynolds Number Correction Factor Chart" on page 14.19, it is found that a correction factor must be used. The actual average velocity is:

V = 1000/14 × 12/144 = 857 fpm

The equations under Note 3 on page 14,20 are solved to allow the correction factor to be obtained.

$$D = \frac{2\text{ HW}}{H + W} = \frac{2 \times 12 \times 14}{12 + 14} = 12.92;$$

$R_e = 8.56 \text{ DV} = 8.56 \times 12.92 \times 857$

$R_e = 94{,}780$
$R_e 10^{-4} = 9.48$

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 $= C \times V_p \times K_{Re}$
$= 0.16 \times 0.05 \times 1.32 = 0.011$ in. w.g.
(enter on line 18)

If the K_{Re} correction factor was not used, the calculated loss of 0.008 in. w.g. (0.16 × 0.05) is 0.003 in. w.g. lower than the value used. On a long, winding 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 $= C \times V_p = 0.04 \times 0.05$
$= 0.002$ in. w.g. (Line 19)

Figure T of Table 14-14 (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 in Chapter 14 would be the mitered elbows in Table 14-10, Figures C, D or E. The average loss coefficient value for a 90° turn from these figures is 1.2, which is the recommended value to use until additional research in the SMACNA program establishes duct fitting loss coefficients for these configurations.

Obviously, if there was ample room in the ceiling, the use of a vaned elbow 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:

Fitting loss = $C \times V_p = 1.2 \times 0.05 = 0.060$ in. w.g. (enter on line 20)

The diffuser pressure loss on the drawing (Figure 7-1) for the diffuser at J includes the pressure losses for the damper with the diffuser. The 0.14 in. w.g. is entered on line 21 in column L.

In Table 7-1, the pressure losses on lines 16 through 21 in column L are totalled (0.241 in. w.g.) and the value entered on line 21 in column M (in black) and on line 16 in column N (in red). 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. w.g. (line 4) for the duct run B to J (the assumed main duct run). This total is added to the 0.90 in. w.g. on line 3 of column M (Fan Plenum B) for the total pressure loss of 1.382 in w.g., the design total pressure at which supply fan B must operate for 8000 cfm. The value of 1.382 in. w.g. is entered on line 1 in columns N and O. (The numbers in column N and O are shown in red to indicate that they 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 per 100 feet remain constant, as these losses are dependent only on the friction loss rate selected. The minor modification at the last duct section was made because of the rectangular duct size that was selected.

The last section of duct (IJ), with all of its fittings and the terminal device, had over half of the pressure loss generated by the complete duct run (BJ). The primary reason for this is that all of the fittings in the main run had a static regain (included in the loss coefficients) with each lowering of the airstream velocity which reduced the actual pressure loss of each section.

g) 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 20 feet of duct from F to G (0.019 in. w.g.) and the elbow at G (0.014 in. w.g. for a total loss of 0.033 in. w.g.). 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. w.g.) 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 14-14, Figure W (used before for the wye at F), and using the same ratios as before, ($A_b/A_s = 1.0$; $A_b/A_c = 0.5$; $Q_b/Q_c = 0.5$), the *branch* loss coefficient $C = 0.52$.

F fitting loss = $C \times V_p = 0.52 \times 0.11$
= 0.057 in. w.g. (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. w.g. for fitting F is compared to the 0.033 in. w.g. computed above for duct EG and elbow G. As the difference between them of 0.024 in. w.g. is within the 0.05 in. w.g. allowable design difference, the fitting used at F was a good selection. However, the A_1M duct run will have a 0.024 in. w.g. greater pressure loss than the A_1J duct run. So the assumed "longest run" did not have the greatest pressure loss although again the difference was within 0.05 in. w.g. This also confirms the need for the use of balancing dampers in each of the 20″ × 18″ ducts at F.

The information for the "branch" volume damper at F can be copied from line 12 of Table 7-1 (as all conditions are the same) and entered on line 3 of Table 7-1(a). The calculations then are made for the 10 ft. of 20″ × 18″ duct (FK):

$$\text{FK duct loss} = \frac{10 \text{ ft.} \times 0.095}{100} = 0.010 \text{ in. w.g.}$$

(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. w.g.) 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), so lines 15 and 21 in column M of Table 7-1 are totalled (0.265 in. w.g.) and entered on line 5 in columns M and N of table 7-1(a).

Finally, the figures in column M are accumulated in column N (starting from the bottom) to obtain the new total pressure loss of 1.403 in. w.g. for the fan B duct system (line 1, column 0). *This loss only is 0.021 in. w.g. higher than the A_1J 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.*

h) 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 totalling lines 1 to 7 of Table 7-1 and entering the 1.049 in. w.g. value on line 7 in column M.

Data for duct section EN is obtained (2000 cfm, 1140 fpm, 18.2 inch diam., with 20″ × 14″ being the selected rectangular size) using the same 0.095 in. w.g. friction loss rate which has changed only once in this example to this point:

$$\text{EN duct loss} = \frac{10 \text{ ft.} \times 0.095}{100 \text{ ft.}} = 0.010 \text{ in. w.g.}$$

(enter on line 8)

The data used before for computing the "main" loss coefficient for wye E (Table 14-14, figure W) is again used to obtain the "branch" loss coefficient (see "Duct Section EF")

$A_b/A_s = 0.33$, $A_b/A_c = 0.25$, $Q_b/Q_c = 0.25$

(the preliminary calculations to branch EN are verified).

C (branch) = 0.43 (by interpolation)

$$\text{E fitting loss} = C \times V_p = 0.43 \times 0.13$$
$$= 0.056 \text{ in. w.g. (line 9)}$$

The loss values in column L (0.010 + 0.056) are totalled and entered on line 9 in column M (0.066 in. w.g.).

i) 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" rectangular size again is selected using 14.2 in. diameter, 0.08 in. w.g. per 100 ft. friction loss rate, and 900 fpm velocity.

$$\text{NP duct loss} = \frac{55 \text{ ft.} \times 0.08}{100 \text{ ft.}} = 0.044 \text{ in. w.g.}$$

(enter on line 10)

At N, a 45° entry tap is used for branch duct NS and a 30° transition is used to reduce the duct size for the run to P. From Table 14-12, Figure A:

$A_1/A = 20 \times 14/14 \times 12 = 1.67$

$C = 0.05$ for $\Theta = 30°$,

Vel. = 2000/20 × 14/144 = 1029 fpm, $V_p = 0.07$

$$\text{N fitting loss} = C \times V_p = 0.05 \times 0.07$$
$$= 0.004 \text{ in. w.g. (line 11).}$$

The volume damper at N has the same numbers as used above for the damper at J:

$$\text{Damper loss} = C \times V_p = 0.04 \times 0.05$$
$$= 0.002 \text{ in. w.g. (Line 12)}$$

At O, a smooth radius elbow with one splitter vane is selected (Table 14-10, Figure G):

R/W = 0.25, H/W = 12/14 = 0.86, C = 0.12 (by interpolation)

$$\text{O fitting loss} = C \times V_p = 0.12 \times 0.05$$
$$= 0.006 \text{ in. w.g. (line 13)}$$

The cumulative loss of 0.056 in. w.g. (0.044 + 0.004 + 0.002 + 0.006) is entered on line 13 in column M.

j) Duct Section PQ—Data for the last 20 feet of duct is obtained from Figure 14-1 and Table 14-2 (500 cfm, 810 fpm, 10.7 inch diameter, which is the equivalent of a 12″ × 8″ rectangular size):

$$\text{PQ duct loss} = \frac{20 \text{ ft.} \times 0.095}{100 \text{ ft.}} = 0.019 \text{ in. w.g.}$$

(enter on line 14)

The loss coefficient for transition P is obtained from Table 14-12, Figure A (converging flow) using Θ = 45°:

$A_1/A = 14 \times 12/12 \times 8 = 1.75$;

C = 0.06, Vel. = 857 (from the 14″ × 12″ duct)

$$\text{P fitting loss} = C \times V_p = 0.06 \times 0.05$$
$$= 0.003 \text{ in. w.g. (line 15)}$$

The fitting at Q is a mitered 90° change of-size elbow (Table 14-10, Figure E).

$H/W = 8/12 = 0.67$; $W_1/W = 16/12 = 1.33$

Velocity = 500/12 × 8/144 = 750 fpm, $V_p = 0.04$

A fitting loss coefficient of 1.0 is selected. Then referring to Note 2 on Page 14.17, plotting the data on the "Reynolds Number Correction Factor Chart" indicates that a correction factor will be required.

$$D = \frac{2 \times 8 \times 12}{8 + 12} = 9.6$$

$R_e = 8.56\,DV = 8.56 \times 9.6 \times 750 = 61{,}632$

$R_e 10\ ^{4} = 6.16$; $K_{Re} = 1.09$

$$\text{Q fitting loss} = 1.0 \times 0.04 \times 1.09$$
$$= 0.044 \text{ in. w.g.}$$

(enter on line 16)

The pressure loss of 0.13 in. w.g. on the drawing (Figure 7-1) for the 16″ × 8″ grille is entered on line 17.

The pressure losses on lines 14-17 in column L are totalled (0.196 in. w.g.) 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. w.g. which is entered on line 7 in columns N and O.

The A_1M duct run pressure loss of 1.430 in. w.g. is 0.063 in. w.g. higher than the 1.367 in w.g. pressure loss of the A_1Q duct run, giving a system that is slightly above the 0.05 in. w.g. suggested good design difference. Nevertheless, balancing dampers in the branch ducts at N should allow the TAB technician to properly balance the system.

k) Duct Section NS—The pressure losses from A_1 to N (lines 7 to 9) are totalled (1.115 in. w.g.) 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:

$$\text{NR duct loss} = \frac{8 \text{ ft.} \times 0.080}{100 \text{ ft.}} = 0.006 \text{ in. w.g.}$$

(enter on line 19)

A 45° entry rectangular tap is used for the branch duct at N. From Table 14-14, Figure N:

$V_b/V_c = 857/1029 = 0.83$ (Use 1.0)

$Q_b/Q_c = 1000/2000 = 0.5; C = 0.74$

Velocity = 1029 fpm; $V_p = 0.07$

N Fitting Loss $= C \times V_p = 0.74 \times 0.07$
$= 0.052$ in. w.g.

(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:

$$\text{RS duct loss} = \frac{20 \text{ ft.} \times 0.095}{100 \text{ ft.}} = 0.019 \text{ in. w.g.}$$

(Enter on line 22)

R Transition loss $= C \times V_p = 0.06 \times 0.05$
(from line 15)
$= 0.003$ in. w.g.

(Enter on line 23)

S Elbow loss $= C \times V_p \times K_{Re}$ (from line 16)
$= 1.0 \times 0.04 \times 1.09$
$= 0.044$ in. w.g.

(Enter on line 24)

S Grille loss (from Figure 7-1) = 0.13 in. w.g.
(Enter on line 25)

The losses for Run RS in column L are totalled and 0.196 in. w.g. 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. w.g.) 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.

l) Additional Discussion—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 $\Theta = 0°$ (wide open). The preliminary damper setting angle Θ can be calculated in this situation as follows (*assuming* a total system loss difference of 0.038 in. w.g. between points S and Q for this example):

System loss difference = 0.038 in. w.g.

N damper loss (set at 0°) = 0.002 in. w.g.

N damper loss (set at ?) = 0.040 in. w.g.
(0.038 + 0.002)

Damper loss $= C \times V_p$ or $C = \text{Damper loss}/V_p$

$C = 0.040/0.50 = 0.80$

Referring back to Table 14-18, Figure B, the loss coefficient when C = 0.80 would require a damper angle Θ of about 15° (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).

Other advantages of the above duct sizing procedures are that 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 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 costs.

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. w.g. 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, fire and smoke code requirements, etc.

Finally, the system pressure loss check list in Figure 9-1 of Chapter 9 should be used to verify that all system component pressure losses have been included in the fan total pressure requirements, and that some allowance has been added for possible changes in the field. These additional items should be shown on the duct sizing work sheets.

E RETURN AIR (EXHAUST AIR) DUCT SYSTEM-SIZING EXAMPLE NO. 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 brake horsepower requirements. This will conserve energy and, therefore, lower the daily 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 B—"Other Factors Affecting Duct System Pressures" of Chapter 5. 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:

$\text{Fan SP} = TP_d - TP_s - Vp_d$ (Equation 5-4)

$\text{Fan SP} = TP_d - (-TP_s) - Vp_d$

$\text{Fan SP} = TP_d + TP_s - Vp_d$

as $TP = SP + Vp$, 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 work sheet as positive numbers.

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 7-2 in column L (0.30 in. w.g.). 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, $V_p = 0.16$ and $C = 1.5$ from Table 14-16, Figure I:

$$\text{Z Fan pressure loss} = C \times V_p = 1.5 \times 0.16 = 0.24 \text{ in. w.g.}$$

(Enter on line 2)

The plenum loss total of 0.54 in. w.g. is entered on line 2 in Column M.

2. Exhaust Air System

a) Duct Section YW—Using 8,000 cfm, 1500 fpm is selected from the chart in Figure 14-1 which establishes the duct friction loss at 0.08 in. w.g. per 100 ft. of duct and the diameter at 32.8 inches. From Table 14-2, a 30" × 30" retangular duct can be selected for the YW duct section and the computed friction loss value entered in column L.

$$\text{YW duct loss} = \frac{30' \times 0.08}{100} = 0.024 \text{ in. w.g.}$$

(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. w.g. (entered on line 4).

The use of an inlet box (see Section B-8 of Chapter 6) would reduce the loss, but many fans are connected in this manner.

The dynamic friction loss of the elbow/transition must also be computed. Table 14-10, Figure F can be used for the elbow, and Table 14-11, Figure D for the transition.

Transition Y:

$$\tan\left(\frac{\theta}{2}\right) = \frac{D - 1.13\sqrt{HW}}{2L} = \frac{33 - 1.13\sqrt{30 \times 30}}{2 \times 2}$$

$$\tan\left(\frac{\theta}{2}\right) = \frac{33 - 33.9}{4} = 0.225$$

$$\frac{\theta}{2} = 12.68°;\ \theta = 25.36°$$

From Table 14-11, Figure B:

$A_1/A < 2$; $C = 0.24$ (by interpolation)

$\text{Velocity} = 8000/30 \times 30/144 = 1280 \text{ fpm}$

From Table 14-6 or by calculation, $V_p = 0.10$ in. w.g.

Table 7-2 DUCT SIZING, EXHAUST AIR SYSTEM— EXAMPLE NO. 2

DUCT SIZING WORK SHEET
(U.S. Units)

DATE ______ PAGE 1 OF 1

PROJECT SAMPLE BUILDING LOCATION FIRST FLOOR SYSTEM EXHAUST AIR

	A DUCT RUN	B SECTION	C ITEM		D FLOW CFM	E FRICTION PER 100'	F VELOCITY FPM	G Vp	H LOSS COEFF.	I EQUIV. DIAM.	J RECTANGULAR SIZE	K CORR. FACT.	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1	PLENUM Z	A,Z	-	H.R. DEVICE	8000	–	500	–	–	–	–	–	0.30		1.272	1.272
2	"	Z	-	FAN	8000	-	1595	0.16	-	–	-	-	0.24	0.54		(A,T)
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	-	30×30/33ϕ	-	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°	TRANS	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																

NOTES: *Indicates duct lining used. Sizes are interior dimensions.

$$\text{Y Transition loss} = C \times V_p = 0.24 \times 0.10 = 0.024 \text{ in. w.g.}$$

(Enter on line 5)

Elbow Y:

H/W = 1.0, R/W = 0.75, C = 0.44

Using the equivalent diameter, a quick check of the "Reynolds Number Correction Factor Chart" on page 14.17 indicates that no correction is needed.

$$\text{Y Elbow loss} = C \times V_p = 0.44 \times 0.10 = 0.044 \text{ in. w.g.}$$

(Enter on line 6)

Note that the combined pressure loss of 0.348 in. w.g. (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 = $C x V_p = 1.0 \times 0.10 = 0.10$ in. w.g.

The total for YW (0.372) is entered on line 6 in column M.

b) Duct Section WU—Using 6000 cfm and 0.8 in. w.g./100, 1400 fpm is established with 28.0 inch diameter. Using Table 14-2, a rectangular size of 30″ × 20″ is selected (keeping one side the same size).

$$\text{WU duct loss} = \frac{100' \times 0.08}{100} = 0.08 \text{ in. w.g.}$$

(Enter on line 7)

A converging 45° entry fitting will be used at W (see Table 14-13, Figure F).

To obtain the "main" loss coefficient, the note in Fitting 14-13F refers to Fitting 14-13B:

Using Table 14-13B (Main Coefficient):

$Q_b/Q_c = 2000/8000 = 0.25$; $C = 0.33$ (by interpolation)

W fitting loss $= C \times V_p = 0.33 \times 0.10$
$= 0.033$ in. w.g.
(Enter on line 8)

Note that 0.10 is the velocity pressure of the 30″ × 30″ downstream section (note direction of flow).

The diverging flow transition at W with an included angle of 30° uses Table 14-11, Figure E because of the change of only one duct dimension.

$A_1/A = 30 \times 30/30 \times 22 = 1.36$ (use 2);
$C = 0.20$;
upstream section velocity $= 6000/30 \times 22/144 =$ 1309 fpm.

$V_p = 0.11$ (From Table 14-6 or by calculation)

W trans. fitting loss $= C \times V_p = 0.20 \times 0.11$
$= 0.022$ in. w.g.
(Enter on line 9)

Using a radius elbow without vanes (Table 14-10, Figure F) at V, the following data is used:

$H/W = 22/30 = 0.73$, $R/W = 2.0$,
$C = 0.16$

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 $= C \times V_p = 0.16 \times 0.11$
$= 0.018$ in. w.g.
(Enter on line 10)

As before, the total section loss of 0.145 in. w.g. is entered in column M.

c) Duct Section UT—The static pressure loss (the total pressure loss is always the same as the static pressure when there is no velocity change) for the duct section UT is:

$$\text{Ut duct loss} = \frac{20' \times 0.08}{100'} = 0.016 \text{ in. w.g.}$$

(Enter on line 11)

From Figure 14-1 where a 21.7 inch diameter duct and 1180 fpm was obtained for 3000 cfm, a 22″ × 18″ rectangular duct is selected from Table 14-2. A converging 90° tee fitting (Table 14-13, Figure D) will be used at U, but again the "main" loss coefficient is obtained from Figure 14-13B.

$Q_b/Q_c = 3000/6000 = 0.5$; $C = 0.53$
U Fitting loss $= 0.53 \times 0.11$ (downstream V_p)
$= 0.058$ in. w.g.
(Enter on line 12)

The U transition loss coefficient is found in Table 14-11, Figure B, and the following data computed:

$A_1/A = 30 \times 22/22 \times 18 = 1.67$ (use 2), $\theta = 30°$, $C = 0.25$
Vel. $= 3000/22 \times 18/144 = 1091$ fpm
$V_p = 0.07$ (upstream duct)

U fitting loss $= C \times V_p = 0.25 \times 0.07$
$= 0.018$ in. w.g.
(Enter on line 13)

The pressure loss for the change of size elbow at T will again be computed using Table 14-10, Figure E (Caution should be used to determine airflow direction):

$H/W = 18/48 = 0.38$, $W_1/W = 24/48 = 0.5$,

$C = 1.75$ (by interpolation and extrapolation)
Vel. of the upstream section (grille size)
$= 3000/48 \times 18/144 = 500$ fpm, $V_p = 0.02$

T fitting loss $= C \times V_p = 1.75 \times 0.02$
$= 0.035$ in. w.g.
(Enter on line 14)

Turning vanes could be added to the change of size mitered elbow, but no loss coefficient tables are available. One could speculate that if single-blade turning vanes reduce the $C = 1.2$ of a standard 90° 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. w.g. 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. w.g. entered on line 1 in columns N and O.

d) 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. w.g. for duct run W to T (line 7, column N) is also the total pressure loss requirement for the short 20′ duct run (0.05 in. w.g. is the acceptable pressure difference between outlets or inlets on the same duct run).

In an attempt to dissipate this pressure, a velocity of 1,550 fpm and a duct friction loss rate of 0.2 in. w.g. per 100 ft. (15.3 inch diameter) is selected for the 2,000 cfm flow rate (Figure 14-1). One inch thick duct lining (correction factor = 1.93 from Figure 14-3 and Table 14-1 [Rough]) also can be added for noise control and increased friction, and a balancing damper is to be used for final adjustments. The computations using this modification of the design method are:

$$\text{WX duct loss} = \frac{20' \times 0.2}{100'} \times 1.93$$
$$= 0.077 \text{ in. w.g.}$$

(Enter on line 17)

Select the rectangular size of 14″ × 14″ from Table 14-2.

The converging 45° entry fitting used at W (Table 14-13, Figure F) is reviewed again to determine the branch loss coefficient.

$Q_b/Q_c = 0.25$, Velocity (V_c) = 1280 fpm,
$C = -0.37$,
W fitting loss = $-0.37 \times 0.10 = -0.037$ in. w.g.

As 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. A smaller grille with a higher pressure loss could be used if a greater noise level could be tolerated.

If a straight rectangular tap was used (Table 14-13, Figure D) instead of the 45° 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 14-12. With $A_1/A = \frac{36 \times 36}{14 \times 14} = 2.94$ and $\Theta = 180°$ (abrupt), $C = 0.33$ by interpolation. The downstream velocity must be used to determine the V_p used in the computations:

Velocity = 2000/14 × 14/144 = 1469 fpm;
$V_p = 0.13$;
X fitting loss = $C \times V_p = 0.33 \times 0.13$
$= 0.043$ in. w.g.

(Enter on line 19)

X grille loss (from Figure 7-1) = 0.08 in. w.g. (line 20)

Subtracting 0.163 in. w.g. (the total of lines 17 to 20) from the 0.360 in. w.g. duct run WT pressure loss shown on line 7 in column N, leaves 0.197 in. w.g. of pressure for the balancing damper to dissipate.

Damper loss coefficient $C = TP/V_p = 0.197/0.13$
$= 1.52$

From Table 14-18, Figure B, a damper set at 22° (by interpolation) has the loss coefficient of 1.53 that will balance the branch duct WX. The total of 0.360 in. w.g. (adding lines 17-21) is entered on line 21 in column M and on line 17 in column N.

A perforated plate (Table 14-17, Figure B) is a non-adjustable alternate solution. If a 1/18″ thick perforated plate was used instead of the balancing damper, the calculation procedure would be as follows (see Table 14-17, Figure B):

Assuming 5/8″ diameter holes,
t/d = 0.125/0.625 = 0.2.

With C = 1.53 (from above),
n = 0.64 (by interpolation).

$$n = \frac{A_p}{A};\ A_p = n \times A$$

A_p (flow area of perf. plate) = $0.64 \times 16 \times 13$
$= 133.12$ sq. in.

No. of holes = A_p/area of a 5/8″ hole
= 133.12/0.307
= 434 (5/8″ diameter)

F SUPPLY AIR DUCT SYSTEM SIZING EXAMPLE NO. 3

1. Introduction

Higher pressure supply air systems (over 3 in. w.g.) usually are required for the large central station HVAC supply air duct distribution systems. Because of higher fan brake horsepower requirements, ASHRAE Standard 90.1-1989 provisions will cause the designer to analyze lower pressure duct systems against the on-going (and 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. However, the choice of fitting loss coefficient tables in Chapter 14 require some designers to use a new approach when designing these systems.

2. Design Procedures

After analyzing a duct system layout, the chart in Figure 14-1 of Chapter 14 is used to select an "approximate" initial velocity and a pressure loss per 100 feet 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) of the duct sections and the selected velocity (fpm), the duct diameters and friction loss rates also may be obtained from Figure 14-1. When rectangular duct sizes are to be used, selection may be made from the chart in Table 14-2, 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.

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 14-4 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 Chapter 14 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 9. 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 (comparing loss coefficients to obtain the most efficient fitting). If the initial system airflow is over 30,000 cfm, the velocity can be held constant (with an increase in the duct friction rate) until the system airflow drops below 30,000 cfm. Then the duct friction rate generally should remain constant (equal friction).

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 totalled 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 which 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 brake horsepower and cost of operation. This is becoming more critical with rising energy rates, and a life cycle cost analysis will probably dictate that lower operating costs be considered more important than lower first costs and space saving requirements.

3. Supply Air System

Table 7-3 is the tabulation of design and computation data obtained when sizing the 20,000 cfm supply duct system shown in Figure 7-2. The 290 foot 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 vary from 20,000 cfm to 1000 cfm. Selecting an initial velocity of approximately 3200 fpm and a friction rate of 0.30 in. w.g. per 100 feet would indicate (by following the 0.30 in. w.g. line horizontally to 1000 cfm) that the duct velocities would gradually be reduced to almost 1500 fpm at an airflow of 1000 cfm.

a) 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 43″ × 32″, a discharge velocity of 2190 fpm (velocity pressure = 0.30 in. w.g.), and a blast area/outlet area ratio of 0.6.

Elbow B is sized 44″ × 32″ (so that it is similar to the outlet size) and a radius elbow (R/W = 1.5) is selected. It is located 26 inches 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:

$$\text{Equiv. Diam.} = \sqrt{\frac{4 \times 44 \times 32}{\pi}} = 42.3 \text{ in.}$$

Table 7-3 DUCT SIZING, SUPPLY AIR SYSTEM— EXAMPLE NO. 3

DUCT SIZING WORK SHEET (U.S. Units)

DATE ____ PAGE 1 OF 2

PROJECT SAMPLE BUILDING LOCATION CONF. AREA SYSTEM SUPPLY AIR

	A DUCT RUN	B SEC-TION	C ITEM	D FLOW CFM	E FRICTION PER 100'	F VELOCITY FPM	G Vp	H LOSS COEFF.	I EQUIV. DIAM.	J RECTANGULAR SIZE	K CORR. FACT.	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1	PLENUM A	A	- SYSTEM EFFECT	20,000	–	2190	–	–	40.9	44X32	RS	0.290		2.875	2.875
2	"	B	90° ELBOW	20,000	–	2045	0.26	0.15	40.9	44X32	–	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	–	44X32/34Ø	–	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° WYE	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 OF 2								–	—	0.650	0.650	
25															

NOTES: *Indicates duct lining used. Sizes are interior dimensions.

(or use 40.9 in. from Table 14-2),

% Effective duct

$$= \frac{\text{straight duct length} \times 100}{\text{Vel./1000 (2.5 min.)} \times \text{Equiv. Diam.}}$$

$$\text{\% Effective duct} = \frac{26 \times 100}{2.5 \times 42.3} = 24.6\%$$

From Table 6-2, System Effect Curve R-S for a 0.6 blast area ratio and 25% Effective Duct is used with Figure 6-1 to find the System Effect pressure loss of 0.29 in. w.g. (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. w.g. 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 14-10, figure F) with R/W = 1.5 and H/W = 44/32 = 1.38.

$$\text{Average Velocity} = 20{,}000/44 \times 32/144$$
$$= 2045 \text{ fpm}$$

The velocity pressure (V_p) of 0.26 in. w.g. is obtained from Table 14-4 for a velocity of 2045 fpm. A quick check of the "Reynolds Number Correction Factor" chart on page 14.17 shows that no correction is needed.

$$\text{B fitting loss} = C \times V_p = 0.15 \times 0.26$$
$$= 0.039 \text{ (line 2)}$$

The total pressure loss of 0.329 in. w.g. for the plenum is entered on line 2 in column M.

b) Duct Section CF—Round spiral duct with an absolute roughness of 0.0003 feet will be used in this supply duct system. For the 80 feet of duct in section

Table 7-3(a) DUCT SIZING, SUPPLY AIR SYSTEM— EXAMPLE NO. 3 (CONT.)

DUCT SIZING WORK SHEET (U.S. Units)

DATE ______ PAGE 2 OF 2

PROJECT SAMPLE BUILDING LOCATION CONF. AREA SYSTEM SUPPLY AIR

	A DUCT RUN	B SEC-TION	C ITEM		D FLOW CFM	E FRICTION PER 100'	F VELOCITY FPM	G Vp	H LOSS COEFF	I EQUIV DIAM	J RECTANGULAR SIZE	K CORR FACT	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O 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																
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NOTES: *Indicates duct lining used. Sizes are interior dimensions.

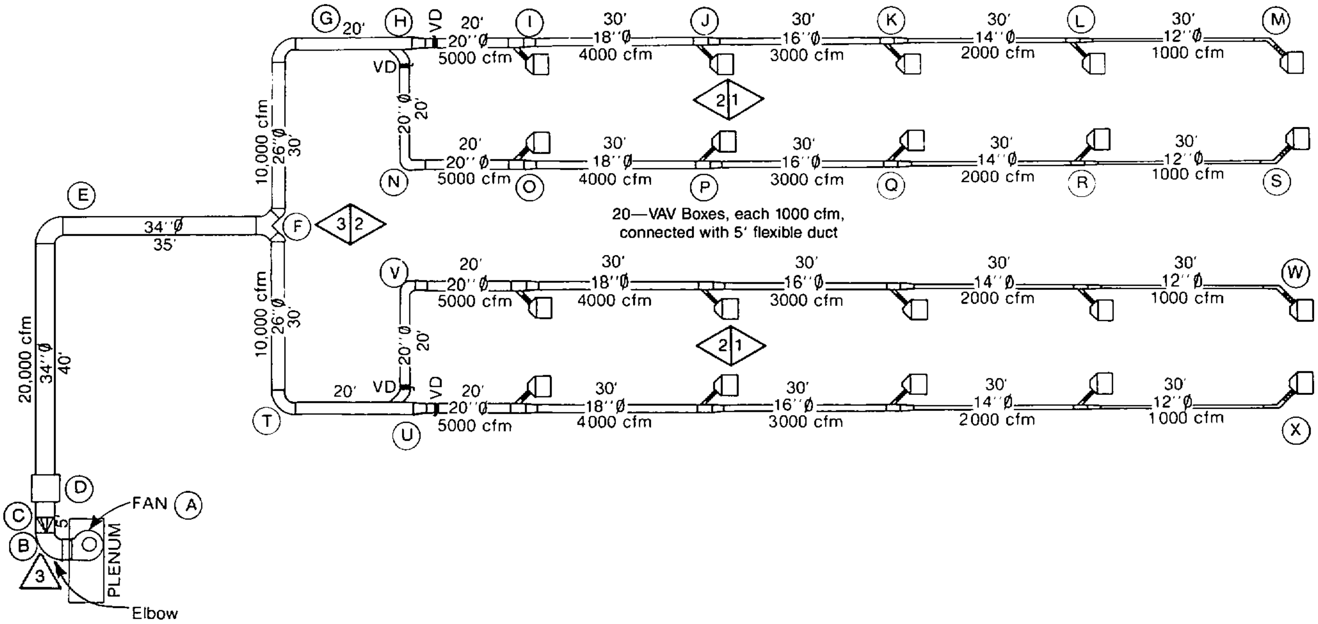

Figure 7-2 SUPPLY AIR DUCT SYSTEM FOR SIZING EXAMPLE NO. 3

CF and using an assumed velocity of 3200 fpm, it falls right on the closest standard size duct diameter of 34 inches (in the chart of Figure 14-1). The selected velocity of 3200 fpm has a friction loss rate of 0.30 in. w.g. per 100 ft. A duct friction correction factor is not required, as the chart in Figure 14-1 is based on the same absolute roughness.

$$\text{CF duct loss} = \frac{80' \times 0.30}{100'}$$
$$= 0.240 \text{ in. w.g. (line 3)}$$

The transition at C will be converging, rectangular to round (Table 14-12, figure A) with $A_1/A = 44 \times 32 / (17)^2\pi = 1.55$ and $\Theta = 20°$; C = 0.05. The velocity pressure used is that of the downstream section: 0.64 in. w.g. for 3200 fpm (Table 14-4).

$$\text{C transition loss} = C \times V_p$$
$$= 0.05 \times 0.64 \text{ (leaving } V_p\text{)}$$
$$= 0.032 \text{ in. w.g. (line 4).}$$

The pressure loss for a medium attenuation 34 inch diameter sound trap of 0.26 in. w.g. is obtained from Chapter 9. A preliminary loss also can be obtained from manufacturer's data sheets. The data is entered on line 5.

The smooth radius, 90° round elbow at E has an R/D ratio = 1.5; C = 0.15 (Table 14-10, Figure A).

$$\text{E elbow loss} = C \times V_p = 0.15 \times 0.64$$
$$= 0.096 \text{ in. w.g. (line 6).}$$

The pressure losses of the four items in duct section CF are added and the 0.628 in. w.g. total is entered in column M on line 6.

c) Duct Section FH—Using the same procedure as above, the closest standard size for 10,000 cfm at 0.30 in. w.g. friction loss per 100 feet is 26 inch diameter (Figure 14-1). A velocity of 2700 fpm and the related V_p of 0.45 is used for further calculations.

$$\text{FH duct loss} = \frac{50' \times 0.30}{100}$$
$$= 0.150 \text{ in. w.g. (line 7)}$$

Using a 45° round wye fitting (Table 14-14, Figure Y) with 45° elbows at F, $V_{1b}/V_c = 2700/3200 = 0.84$; C = 0.28 (by interpolation).

$$\text{F wye fitting loss} = C \times V_p = 0.28 \times 0.64$$
$$= 0.179 \text{ in. w.g. (line 8).}$$

The 45° round elbow (RD = 1.5) at F will use the same loss coefficient as the 90° elbow above (Table 14-10, Figure A) multiplied by the 0.6 correction factor for 45° (Note 1). V_p for 2700 fpm = 0.45 in. w.g.

$$\text{F elbow fitting loss} = 0.15 \times 0.45 \times 0.6$$
$$= 0.041 \text{ in. w.g. (line 9).}$$

The 90° round elbow at G uses the same values without the correction factor.

$$\text{G elbow fitting loss} = 0.15 \times 0.45$$
$$= 0.068 \text{ in. w.g. (line 10).}$$

The losses in column L again are totalled in column M (0.438 in. w.g.).

d) Duct Section HO—The following values are obtained using the same procedures as above (5000 cfm at 0.30 in. w.g. per 100 feet friction loss): 20 inch diameter duct size, 2300 fpm velocity. Note that the duct velocity continues to decrease as the duct volume (cfm) becomes lower using the same duct friction loss rate (0.30 in. w.g. per 100 ft.).

$$\text{HO duct loss} = \frac{40' \times 0.30}{100'}$$
$$= 0.120 \text{ in. w.g. (line 11)}$$

At point H in the duct system, the branch coefficient is obtained for the diverging 45° round wye with a conical main and branch with a 45° elbow (Table 14-14, Figure M):

$$V_b/V_c = \frac{2300}{2700} = 0.85,\ C = 0.51,$$
$$V_p = 0.45 \text{ (2700 fpm).}$$
$$\text{H wye (branch) loss} = C \times V_p = 0.51 \times 0.45$$
$$= 0.230 \text{ in. w.g. (line 12).}$$

The 90° round elbow is calculated as the above 90° ell and the loss coefficient for the balancing damper is obtained from Table 14-18, Figure A ($\Theta = 0°$); C = 0.20.

$$\text{H damper loss} = C \times V_p = 0.20 \times 0.33$$
$$= 0.066 \text{ in. w.g. (line 13)}$$

$$\text{N elbow fitting loss} = C \times V_p = 0.15 \times 0.33$$
$$= 0.050 \text{ in. w.g. (line 14)}$$

The total for the HO duct section (0.466 in. w.g.) is entered in column M.

e) Duct Section OP—For 4000 cfm at 0.30 in. w.g., the closest standard size duct is 18 inch diameter. Using the 18 inch duct, the friction rate then becomes 0.34 in. w.g. per 100 feet and the duct velocity is 2250 fpm.

$$\text{OP duct loss} = \frac{30' \times 0.34}{100'}$$
$$= 0.102 \text{ in. w.g. (line 15)}$$

V_p for 2250 fpm = 0.32 (Table 14-6)

The 45° round diverging conical wye at point 0 (Table

14-14, figure C) requires that the "main" coefficient C be obtained from Table 14-14A.

V_s/V_c = 2250/2300 = 0.98; but when there is no change in velocity, the table indicates that there is no dynamic loss, i.e. C = 0 for V_s/V_c = 1.0. Interpolating gives a questionable loss coefficient of 0.004, which multiplied by the V_p of 0.33 gives a loss of 0.001. However, a minimum loss coefficient of 0.01 is used.

0 Wye (main) loss = 0.01 × 0.33 = 0.003 in. w.g. (enter on line 16)

The 60° transition from 20 inch diameter to 18 inch diameter does have a dynamic pressure loss and the fitting loss coefficient is obtained from Table 14-12, Figure A.

$A_1/A = 10^2\pi/9^2\pi = 1.23$, C = 0.06

0 transition loss = C × V_p (downstream)
= 0.06 × 0.32
= 0.019 in. w.g. (line 17)

The section loss of 0.124 in. w.g. is entered in column M on line 17.

f) Duct Section PQ—The same calculations as used in duct section OP are repeated using 3000 cfm and a 0.30 in. w.g. per 100 feet friction loss rate to obtain the closest standard duct size of 16 inch diameter (Figure 14-1). Using the exact 16 inch duct size, the new velocity is 2150 fpm and the friction loss rate is 0.36 in. w.g. per 100 feet:

$$\text{PQ duct loss} = \frac{30' \times 0.36}{100'}$$
= 0.108 in. w.g. (line 18)

V_p for 2150 fpm = 0.29 (Table 14-6)

For the 45° round conical wye at P (Table 14-14A):

V_s/V_c = 2150/2250 = 0.96,
C = 0.01 (0.004 by interpolation),

P wye (main) loss = C × V_p = 0.01 × 0.32
= 0.003 in. w.g. (line 19).

60° transition at P: $A_1/A = 9^2\pi/8^2\pi = 1.27$,
C = 0.06

P transition loss = C × V = 0.60 × 0.29
= 0.017 in. w.g. (line 20)

This section loss of 0.128 is entered in column M.

g) Duct Section QR—The selection of a standard 14 inch diameter duct for 2000 cfm (Figure 15-1 indicates a 0.32 in. w.g. per 100 feet friction loss rate and a velocity of 1880 fpm.

$$\text{QR duct loss} = \frac{30' \times 0.32}{100'}$$
= 0.096 in. w.g. (line 21)

V_p for 1880 fpm = 0.22 (by calculation)

Again using the same type of wye at Q:

V_s/V_c = 1880/2150 = 0.87, C = 0.01 (0.013 by interpolation)

Q wye (main) loss = C × V_p = 0.01 × 0.29
= 0.003 in. w.g. (line 22)

Q Transition: $A_1/A = 8^2\pi/7^2\pi = 1.31$, C = 0.06

Q Trans. fitting loss = C × V_p = 0.06 × 0.22
= 0.013 in. w.g. (line 23)

The section loss of 0.112 is entered in column M.

h) Duct Section RS—Using an addition duct sizing form to record the data [Table 7-3(a)], the 1000 cfm duct size at 0.30 in. w.g. would be half way between the 10 inch and 12 inch standard duct sizes. The 10 inch diameter duct would be in a much higher pressure loss category, so the 12 inch duct at 0.19 in. w.g. per 100 feet and 1290 fpm velocity would be the better selection.

$$\text{RS duct loss} = \frac{30' \times 0.19}{100'}$$
= 0.057 in. w.g.
[Enter on line 1 of Table 7-3(a)]

V_p for 1290 fpm = 0.10 in. w.g. (by calculation)

R wye fitting: V_s/V_c = 1290/1880 = 0.69, C = 0.04

R wye (main) loss = C × V_p = 0.04 × 0.22
= 0.009 in. w.g. (line 2)

R transition: $A_1/A = 7^2\pi/6^2\pi = 1.36$, C = 0.06

R transition loss = C × V_p = 0.06 × 0.10
= 0.006 in. w.g. (line 3)

A 45° elbow at the end of the duct is connected to the VAV box by a 5 foot piece of 12 inch diameter flexible duct. The correction factor for the flexible duct is obtained from the chart in Figure 14-3 using Table 14-1 as a guide. Bends of 30° 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 14-4 also contains a correction factor for unextended (compressed) flexible duct.

S 45° elbow fitting loss = 0.15 × 0.10 × 0.60
= 0.009 in. w.g. (line 4)

$$\text{S flex. duct loss} = \frac{5' \times 0.19}{100'} \times 1.95 \text{ (rough)}$$
= 0.019 in. w.g. (line 5)

An estimate of 0.25 in. w.g. 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. w.g. and the total (0.55 in. w.g.) 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 of 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 totalled in column N with the *total pressure* loss for the supply duct system of 2.875 in. w.g. being entered on line 1 in column 0.

i) Recap—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 No. 1, several fittings with higher pressure losses or "high loss" VAV boxes can allow a duct run that was not the originally selected run for design computations, actually to be the duct run with the greatest pressure loss.

Assuming that the return air duct system of Example No. 3 (not shown) had an approximate total pressure loss of 2.0 in. w.g., the output of the system supply fan would need to be 20,000 cfm at 4.88 in. w.g. (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 past the wye fitting at F is in the low pressure range (under 2 in. w.g.), even though there are velocities up to 2700 fpm.

This is the reason that 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. 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.

In the first edition of this "HVAC Duct System Design" manual, this same duct system example was sized using a "constant" velocity of approximately 2800 fpm. The duct sizes ranged from 36 inch to 10 inch diameter at 3.22 in. w.g. total pressure, instead of from 34 inch to 12 inch diameter at 2.88 in. w.g. total pressure. The "modified" equal friction method of design allowed a 10 percent lower system total pressure, a smaller main duct near the fan where construction costs are proportionately the greatest, and a yearly savings of approximately $816.00 based on the example in Chapter 2 where electrical energy costs were 9 cents per kW-hour. On-going costs of operating a system are extremely important, but savings in initial system costs also can conserve energy.

G EXTENDED PLENUM DUCT SIZING

1. Introduction

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; i.e., equal friction method, etc. Though there is a lack of published data concerning extended plenum use and design, extensive field testing, both in experimental form and in many actual installations throughout the country, have proven the concept. An *extended plenum* is a trunk duct of constant size, usually at the discharge of a fan, fan-coil unit, mixing box, variable air volume (VAV) box, etc., extended as a plenum to serve multiple outlets and/or branch ducts.

A *semi-extended plenum* is a trunk design *system* utilizing the concept of extended plenum incorporating a *minimum* number of size reductions due to decreasing volume.

2. 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 length of straight duct to fitting ratio.

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.

d) 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 that low velocities, which could develop, might result in excessive heat gain or loss through the duct walls.

3. 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. w.g. through 6 in. w.g. and duct velocities up through 3000 feet per minute. Other specific design considerations include:

a) Branch takeoffs from the trunk duct should preferably be round duct connecting at a 45° angle. If rectangular branches are used, a 45° 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.

4. 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. w.g. per 100 feet. Note that reducing fittings have been used at each branch takeoff.

In Figure 7-4, 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 suzes 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 Chapter 14, 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. w.g. (1.93-1.30) lower pressure loss than the system sized by the equal friction method.

The brake horsepower 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 brake horsepower and, there-

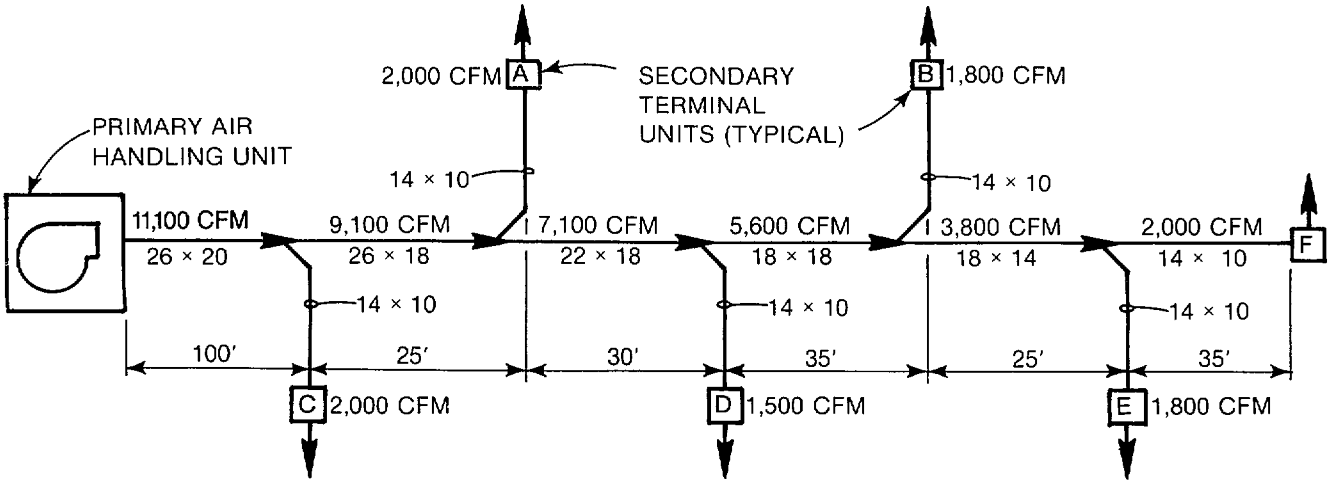

Figure 7-3 SYSTEM "A"—SIZED BY EQUAL FRICTION METHOD

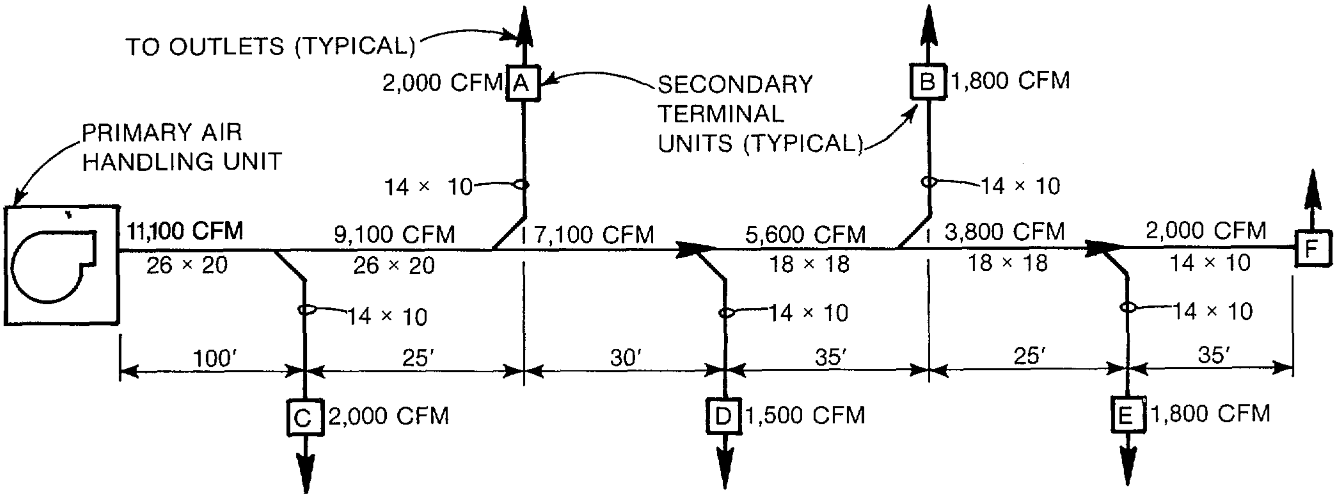

Figure 7-4 SYSTEM "B"—MODIFIED BY SEMI-EXTENDED PLENUM CONCEPT

Table 7-4 SEMI-EXTENDED PLENUM COMPARISON

System	Duct Losses (Inches w.g.)	Fitting Losses (Inches w.g.)	Total Losses (Inches w.g.)	Fan Bhp Required
"A"—equal friction method	1.27	.66	1.93	6.3
"B" semi-extended plenum method	1.08	.22	1.30	4.6

Table 7-5 SEMI-EXTENDED PLENUM INSTALLATION COST 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	

fore, 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.

5. Cost Comparison

Although energy conservation holds the "spotlight," installation costs are still of 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 can be seen that 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.

DUCT SIZING WORK SHEET
(U.S. Units)

DATE ______ PAGE ______

PROJECT ______ LOCATION ______ SYSTEM ______

	A DUCT RUN	B SEC-TION	C ITEM		D FLOW CFM	E FRICTION PER 100'	F VELOCITY FPM	G Vp	H LOSS COEFF.	I EQUIV. DIAM.	J RECTANGULAR SIZE	K CORR. FACT.	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1																
2																
3																
4																
5																
6																
7																
8																
9																
10																
11																
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NOTES: *Indicates duct lining used. Sizes are interior dimensions.

Figure 7-5 DUCT SIZING WORK SHEET (U.S.)

CHAPTER 8

DUCT SIZING PROCEDURES (METRIC UNITS)

A DESIGN FUNDAMENTALS

1. Metric Design

The easiest way that 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 U.S. and the metric 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 examples used in this chapter are in the same general range of values as the examples in U.S. Units in Chapter 7. However, they are not "soft conversions", i.e. the numbers multiplied by the conversion factors found in Chapter 14, Section F—"Metric Units and Equivalents". For example, dividing 3.5 in. w.g. by 0.004022 in. w.g., which equals 1 pascal (1 Pa), the answer would equal 870.21 Pa. A "hard conversion" would be 870 or 875 Pa, a rounded off number.

Some of the easy to remember "round number" conversions generally used to check calculations or where exact conversions are not required are:

2 cfm = 1 litre per second (1 l/s)
200 fpm = 1 metre per second (1 m/s)
1 in. w.g. = 250 pascals (250 Pa)
1 inch = 25 millimetres (25 mm)

Some duct friction loss charts being circulated in the HVAC industry are using "mm w.g./m" (millimetres water gauge per metre) instead of "Pa/m". One Pascal equals 0.1022 mm water gauge, so for practical purposes: 1 Pa = 0.1 mm w.g.—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 Chapter 14.

2. Design Criteria

For duct sizing procedures using U.S. Units, see Chapter 7.

1. The total pressure (TP) at any location within a system is the sum of the static pressure (SP) and the velocity pressure (V_p).
2. Total pressure *always* decreases algebraically in the direction of airflow (negative values of return air or exhaust systems increase in the direction of airflow, and positive values of supply air systems decrease in the direction of airflow—see *Figure 5-10*).
3. The losses in total pressure between the fan and the end of *each* branch of a system are the same.
4. Static pressure and velocity pressure are mutually convertible and either can increase or decrease in the direction of flow.

B DESIGN OBJECTIVES

1. 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.
2. Consider energy conservation in the fan selection, duct configuration, duct wall heat gain or loss, etc.
3. Special consideration should be given to the need for sound attenuation and breakout noise.
4. Testing, adjusting and balancing equipment and dampers should be shown on the drawings.
5. Locations of all life safety devices such as fire dampers, smoke dampers, etc. should be shown on the drawings.
6. The designer should consider the pressure losses that occur from tie rods and other duct obstructions.
7. If the ductwork is well designed and constructed, at least 75 to 90 percent of the original velocity pressure can be regained.
8. Round ducts generally are preferred for higher pressure systems.

9. Branch takeoffs and fittings with low loss coefficients should be used. Both 90° and 45° duct takeoffs can be used. However, the use of conical tees or angular takeoffs can reduce pressure losses.

C DUCT SYSTEM SIZING PROCEDURES

1. Introduction

The "equal friction" method of duct sizing probably has been the most universally used means of sizing low pressure supply air, return air and exhaust air duct systems and it is being adapted by many for use in medium pressure systems. It normally has not been used for sizing high pressure systems. This design method "automatically" reduces air velocities in the direction of the airflow, so that 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 readily may be checked at any point. There is then the opportunity to reduce velocity created noise by increasing duct size or adding sound attenuation materials (such as duct lining).

The major disadvantages of the equal friction method are: (1) there is no natural provision for equalizing pressure drops in the branches (except in the few cases of a symmetrical layout); and (2) there is no provision for providing the same static pressure behind each supply or return terminal device. Consequently, balancing can be difficult, even with a considerable amount of dampering in short duct runs. However, the equal friction method can be modified by designing portions of the longest run with different friction rates from those used for the shorter runs (or branches from the long run).

Static regain (or loss) due to velocity changes, has been added to the equal friction design procedure by using fitting pressure losses calculated with new loss coefficient tables in Chapter 14. Otherwise, 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 the 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 Chapter 14.**

2. Modified Equal Friction Design Procedures

"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 equivalent metre of duct is selected before the ductwork is laid out, and that this pressure loss in pascals per metre is 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 air duct leaving the fan, the usual procedure is to select an initial velocity from the chart in Figure 14-2. This velocity could be selected above the shaded section of Figure 14-2 if higher sound levels and energy conservation are not limiting factors. The chart in Figure 14-2 is used to determine the friction loss by using the design air quantity (litres per second) and the selected velocity (metres per second). A friction loss value commonly used for lower pressure duct sizing is in the range of 0.8 to 1.0 pascals per metre (Pa/m), although 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 14-2.

The friction losses of each duct section should be corrected for other materials and construction methods by use of Table 14-1 and Figure 14-3. The correction factor from Figure 14-3 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 are then used to select the equivalent rectangular duct sizes from Table 14-3, unless round ductwork is to be used.

The flow rate (l/s) in the second section of the main supply duct, after the first branch takeoff, is the original airflow (l/s) supplied by the fan reduced by the amount of airflow (l/s) into the first branch. Using Figure 14-2, the new flow rate value (using the recommended friction rate of 0.8 to 1.0 pascals per metre) will determine the duct velocity and diameter for that section. The equivalent rectangular size of that duct section again is obtained from Table 14-3 (if needed). All subsequent sections of the main supply duct and all branch ducts can be sized from Figure 14-2 using the same friction loss rate and the same procedures.

The total pressure drop measured at each terminal device or air outlet (or inlet) of a small duct system, or of branch ducts of a larger system, should not differ more than 12 pascals. If the pressure difference between the terminals exceeds that amount, dampering would be required that could create objectional air noise levels.

The modified equal friction method is used for sizing duct systems that are not symmetrical or that 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 (as nearly as possible) 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 a good design practice to include balancing dampers in HVAC duct systems to balance the airflow to each branch.

3. Fitting Pressure Loss Tables

Tables 14-10 to 14-18 contain 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 picked up 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 8-1

$$TP = C \times V_p$$

Where:

TP = Total Pressure (Pa)
C = Dimensionless Loss Coefficient
V_p = Velocity Pressure (Pa)

By using the duct fitting loss coefficients in Chapter 14 which include static pressure regain or loss, accurate duct system fitting pressure losses are obtained. When combined with the static pressure friction losses of the straight duct sections 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 which has a velocity of 13 m/s. Using Table 14-7, the velocity pressure (V_p) is found to be 100 pascals. The total pressure (TP) loss of each fitting is determined as follows:

Example A:

900mm (H) × 300mm (W), 90° Radius Elbow (R/W = 1.5), no vanes. From Table 14-10, Figure F, the loss coefficient of 0.14 is obtained using H/W = 3.0.

The loss coefficient should not be used without checking to see if a correction is required for the Reynolds number (Note 3):

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

$$R_e = 66.4 \times DV$$

$$R_e = 66.4 \times 450 \times 13 = 388{,}440$$

$$R_e 10^{-4} = \frac{388{,}440}{10^4} = 38.84$$

The correction factor of 1.0 is found where 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 14-17 had been checked, as the plotted point is outside the shaded area requiring correction (using the duct diameter and velocity to plot the point).

If the elbow was 45° instead of 90°, another correction factor of 0.60 (See the reference to Note 1 on page 14.17) would be used: 0.60 × 14 = 8.4 Pa.

Example B:

45° Round Wye, 500mm diameter main duct, (12.5 m/s); 250mm diameter branch duct, branch velocity of 7.7 m/s. Determine the fitting pressure losses. (Figure A of Table 14-14).

$$A_b = \pi r^2 = \pi(125)^2 = \pi 15{,}625$$

$$A_c = \pi r^2 = \pi(250)^2 = \pi 62{,}500$$

$$A_b/A_c = 15{,}625/62{,}500 = 0.25$$

From Figure 14-2:

For 250mm diameter, 7.7 m/s; $Q_b = 370$ l/s

For 500mm diameter, 12.5 m/s; $Q_c = 2400$ l/s

$Q_b/Q_c = 370/2400 = 0.154$

Interpolating in the table between $A_b/A_c = 0.2$ and 0.3; and $Q_b/Q_c = 0.1$ and 0.2; 0.56 is selected as the branch fitting loss coefficient. The branch pressure loss is calculated.

Obtain V_p of 92.5 for 12.5 m/s from Table 14-7 or by using Equation 5-8 (Metric).

$TP = C \times V_p = 0.56 \times 92.5 = 51.8$ Pa (Equation 5-6).

The main pressure loss is calculated by first establishing V_s:

$Q_s = Q_c - Q_b = 2400 - 370 = 2030$ l/s (downstream airflow).

From Figure 14-2, 500mm diameter and 2030 l/s:

$V_s = 10.5$ m/s.

$V_s/V_c = 10.5/12.5 = 0.84$

From the Table 14-14, Figure A, $C = 0.02$

$TP = C \times V_p = 0.02 \times 92.5 = 1.85$ Pa.

Example C:

900mm × 300mm rectangular to 500mm diameter round transition where $\Theta = 30°$ (Table 14-12, Figure A), $V_p = 100$ Pa.

$A_1 = 900 \times 300 = 270{,}000\ m^2$

$A = \pi r^2 = \pi(250)^2 = 196{,}350\ m^2$

$A_1/A = 270{,}000/196{,}350 = 1.37$ (use 2)

0.05 is selected as the loss coefficient.

$TP = C \times V_p = 0.05 \times 100 = 5$ Pa

Fortunately, there usually are not too many "complicated" fittings in most duct systems, but when there are, the systems usually are part of a large complex.

A computer programmed for the above calculations can facilitate the duct system design procedure.

D SUPPLY AIR DUCT SYSTEM-SIZING EXAMPLE NO. 1

A plan of a sample building HVAC duct system is shown in Figure 8-1 and the tabulation of the computations can be found in Table 8-1. A full size "Duct Sizing Work Sheet" may be found in Figure 8-5 at the end of this 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, noise level, etc., 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.

1. Supply Fan Plenum

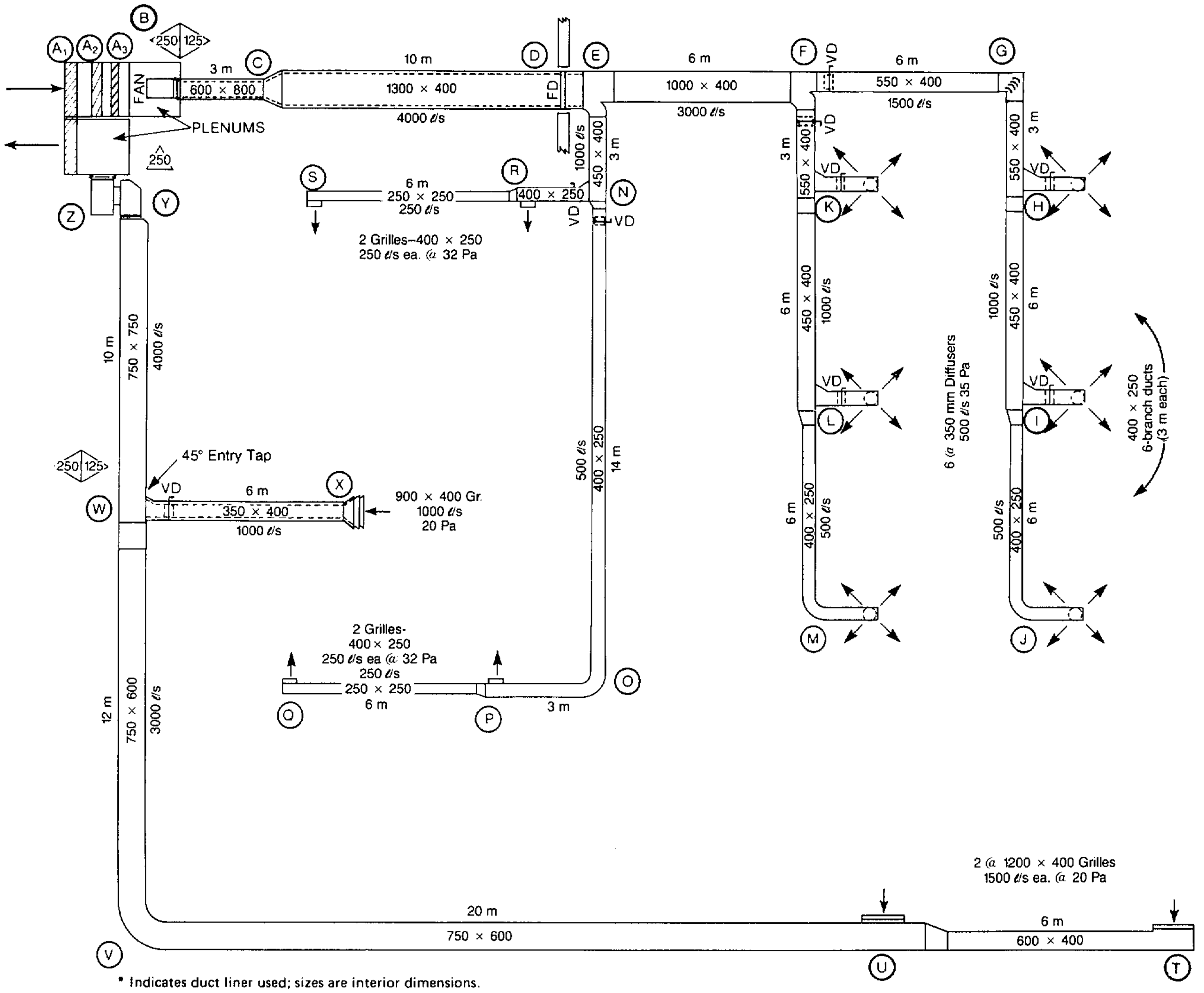
From manufacturer's data sheets or from the Figures or Tables in Chapter 9, the static pressure losses of the energy recovery device, filter bank and heating-cooling coil are entered in Table 8-1 in column L. (Velocities, if available, are entered in column F for reference information only.) With 3 metres of duct discharging directly from fan "B" (duct is fan outlet size), no "System Effect Factor" (see Chapter 6) 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, and the loss of 225 pascals (Pa) is entered in column M on line 3.

2. Supply Air System

a) Duct Section BC—The 600mm × 800mm fan discharge size has a circular equivalent of 755mm inches (Table 14-3). Using the chart in Figure 14-2, a velocity of 9.0 m/s and a friction loss of 0.95 Pa/m of duct 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 600mm × 800mm 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 14-1 and Figure 14-3 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 (corr. 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.



* Indicates duct liner used; sizes are interior dimensions.

Figure 8-1 DUCT SYSTEMS FOR DUCT SIZING EXAMPLES NO 1 AND 2 (METRIC)

b) Duct Section CE—At point C, building construction conditions require that the duct aspect ratio change, so a duct transition is needed. Using the same 0.95 Pa/m duct friction loss and 755mm duct diameter for the 4000 l/s or 4.0 m^3/s airflow, a 1300mm × 400mm duct is selected from Table 14-3 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 = 10 × 0.95 × 1.35 = 12.8 pascals

(enter 13 pascals on line 5 in column L)

The transition loss coefficient can be obtained after determining if the fitting is diverging or converging. $A = 600 \times 800 = 480{,}000$ and $A_1 = 1300 \times 400 = 520{,}000$, $A_1/A = 520{,}000/480{,}000 = 1.08$, so it is diverging (greater than 1.0).

The average velocity of the entering airstream (Equation 5-9) $V = \dfrac{Q}{A} = \dfrac{4.0m^3/s}{1.3m \times 0.4m} = 7.7$ m/s.

From Table 14-11, Figure B, using $\Theta = 30°$ 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 pascals is calculated using Equation 5-10 ($V_p = 0.602\ V^2$) or is obtained from Table 14-7 for 7.7 m/s and entered in column G. The transition fitting pressure loss of 9 Pa ($C \times V_p =$

0.25 × 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 15 pascals for the fire damper at D is obtained from Chapter 9 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 totalled (37 Pa) and entered in column M on line 7. This is the total pressure loss of the 1300mm × 400 mm duct section CE (inside dimensions) and its components.

c) 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 14-2, a duct diameter of 676mm and 8.4 m/s velocity is obtained and entered on line 8. Table 14-3 is used to select a 1000mm × 400mm rectangular duct size needed by keeping the duct height 400mm. Normally, duct size changes are made changing only one dimension (for ease and economy of fabrication) and keeping the aspect ratio as low as possible.

As the continuous rolled galvanized duct system is being fabricated in 1200mm sections, the degree of roughness (Table 14-1) indicates "medium smooth". No correction factor is needed, as the chart in Figure 14-2 is based on an Absolute Roughness of 0.09mm as a result of recent SMACNA assisted ASHRAE research.

The static pressure loss for duct section EF is:

SP = 6m × 0.95 = 5.7 Pa (Use 6)

(enter on line 8 in column L).

Table 8-1 DUCT SIZING, SUPPLY AIR SYSTEM—EXAMPLE NO. 1

DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE ______ PAGE 1 of 2

PROJECT Sample Project LOCATION First Floor SYSTEM Supply Air

	A DUCT RUN	B SECTION	C ITEM		D FLOW L/S	E FRICTION PER METRE	F VELOCITY (M/S)	G Vp (PA)	H LOSS COEFF	I EQUIV DIAM	J RECTANGULAR SIZE	K CORR FACT	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O 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 ϕ	–	18			
21	"	J	-	Diff.	500	–	–	–	–	–	350 ϕ	-	35	68		
22																
23																
24																
25																

NOTES *Indicates duct lining used. Sizes are interior dimensions

The diverging 90° wye fitting used at E can be found in Table 14-14, 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 (Prelim.) for 1000 l/s at 0.95 Pa/m = 196,350 mm^2 (area of 450mm diameter duct obtained from Figure 14-2). Then:

$A_b/A_s = (225)^2\pi/(338)^2\pi = 159{,}043/358{,}908 = 0.44$,

$A_b/A_c = (225)^2\pi/(378)^2\pi = 159{,}043/448{,}883 = 0.35$, and

$Q_b/Q_c = 1000/4000 = 0.25$.

$$\text{Velocity} = \frac{Q}{A} = \frac{4.0 \text{ m}^3/\text{s}}{1.3 \text{ m} \times 0.4 \text{ m}}$$
$$= 7.7 \text{ m/s (Equation 5-9).}$$

Using $A_b/A_s = 0.5$; and $A_b/A_c = 0.5$ (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" thus has a negative value (TP = C × V_p = −0.05 × 35.7 = −1.79) and minus 2 pascals 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.

d) 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 14-2, 7 m/s and 510mm diameter are obtained from Figure 14-2; 550mm × 400mm equiv. duct size from Table 14-3: FH duct section loss = 10 × 0.95 = 9.5 pascals; (enter 10 pascals on line 10).

Table 8-1(a) DUCT SIZING, SUPPLY AIR SYSTEM—EXAMPLE NO. 1 (CONT.)

SMACNA

DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE ____________ PAGE 2 of 2

PROJECT Sample Project LOCATION First Floor SYSTEM Supply Air

	A DUCT RUN	B SECTION	C ITEM		D FLOW (L/S)	E FRICTION PER METRE	F VELOCITY (M/S)	G Vp (PA)	H LOSS COEFF	I EQUIV DIAM	J RECTANGULAR SIZE	K CORR FACT	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1	A, F	—	-	TOTAL OF LINES 1 TO 9 IN TABLE 8-1 (PAGE 1)									—	270	367	367
2	Run FM	F	90°	Wye	3000/1500	-	7.5	33.9	0.52	-	1000 x 400/550 x 400	-	18		97	
3	"	F	-	Vol. Damp	1500	-	6.8	27.8	0.04	-	550 x 400	-	1			
4	"	FK	3m	Duct	1500	0.95	7.0	—	-	510	550 x 400	-	3	22		
5	"	KM	-	TOTAL OF LINES 14 TO 21 IN TABLE 8-1 (PAGE 1)									-	75	75	
6																
7	A, E	—	-	TOTAL OF LINES 1 TO 7 IN TABLE 8-1 (PAGE 1)									-	266	348	348
8	Run EQ	EN	3m	Duct	1000	0.95	6.4	—	-	456	450 x 400	-	3		82	
9	"	E	90°	Wye	4000/1000	—	7.7	35.7	0.44	-	1300 x 400/450 x 400	-	16	19		
10	"	NP	17m	Duct	500	0.95	5.4	—	-	340	400 x 250	-	16		63	
11	"	N	-	Trans	1000/500	-	5.6	18.9	0.05	-	450 x 400/400 x 250	-	1			
12	"	N	-	Vol Damp	500	-	5.0	15.1	0.04	—	400 x 250	-	1			
13	"	O	90°	Elbow	500	-	5.0	15.1	0.12	—	400 x 250	-	2	20		
14	"	PQ	6m	Duct	250	0.95	4.6	—	—	270	250 x 250	-	6		43	
15	"	P	—	Trans	500/250	-	5.0	15.1	0.06	—	400 x 250/250 x 250	-	1			
16	"	Q	90°	Elbow	250	-	4.0	9.6	0.90	-	250 x 250/400 x 250	1.08	4			
17	"	Q	-	Grille	250	-	—	-	-	-	400 x 250	-	32	43		
18	A, N	—	-	TOTAL OF LINES 7 TO 9 ABOVE							—	—	-	235	346	346
19	Run NS	NR	3m	Duct	500	0.95	5.4	—	-	340	400 x 250	—	3		61	
20	"	N	45°	Ent. Tap	1000/500	-	5.6	18.9	0.74	-	450 x 400/400 x 250	—	14			
21	"	N	-	Vol Damp	500	-	5.0	15.1	0.04	-	400 x 250	—	1	18		
22	"	RS	6m	Duct	250	0.95	4.6	—	—	270	250 x 250	-	6		43	
23	"	R	-	Trans	500/250	—	5.0	15.1	0.06	—	400 x 250/250 x 250	—	1			
24	"	S	90°	Elbow	250	-	4.0	9.6	0.90	-	250 x 250/400 x 250	1.08	4			
25	"	S	-	Grille	250	—	—	-	-	-	400 x 250	—	32	32		

NOTES *Indicates duct lining used. Sizes are interior dimensions.

For the wye fitting at F, Table 14-14, figure W is again used. With the 3000 l/s airflow dividing equally into two 1500 l/s airstream ducts, $A_b = A_s$. Therefore,

$A_b/A_c = (255)^2\pi/(338)^2\pi$
$= 204,282/358,908 = 0.57$

$Q_b/Q_c = 1500/3000 = 0.5$

Using $A_b/A_s = 1.0$; $A_b/A_c = 0.5$;

C (Main) = 0.05,
velocity = 3/1.0 × 0.4 = 7.5 m/s

$V_p = 33.9$ Pa (From Table 14-7)

Fitting loss $= C \times V_p = 0.05 \times 33.9$
= 1.70 pascals
(enter 2 pascals on line 11).

The loss coefficient for the thin plate volume damper near F can be obtained from Table 14-18, Figure B (Set wide open, i.e. 0°). The velocity pressure (V_p) of 27.8 Pa for 6.8 m/s (1.5/0.55 × 0.4) is obtained from Table 14-7 or calculated.

Damper Loss $= C \times V_p = 0.04 \times 27.8$
= 1.1 Pa (Use 1 pascal on line 12).

Elbow G in the FH duct run is a square elbow with 114mm single thickness turning vanes on 57mm centers. The loss coefficient of 0.15 is obtained from Table 14-10, Figure H for the 550mm × 400 mm elbow (single thickness vanes—No. 2) and entered on line 13 along with the other data (cfm, fpm, V_p, etc.)

G fitting loss $= C \times V_p = 0.15 \times 27.8$
= 4.2 Pa (Use 4 pascals 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.

e) 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 450mm × 400mm) are obtained (again changing only one duct dimension where possible).

HI duct section loss = 6m × 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 14-12, Figure A using Θ = 30°. 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 = 1.5/0.55 × 0.40 = 6.8 m/s;

Velocity pressure (V_p) = 27.8 Pa;

$\frac{A_1}{A} = \frac{500 \times 400}{450 \times 400} = 1.22$ (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 totalled and entered on line 15 in column M (7 Pa).

f) 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, 340mm diam, with a 400mm × 250mm duct size being selected):

IJ duct loss = 10m × 0.95 Pa/m = 9.5 Pa
(enter 10 pascals on line 16).

It might be obvious by now that using a duct friction loss of 0.95 to 1.0 Pa/m, the calculations are quite simple, i.e. 1 pascal pressure loss for each metre of duct!

Transition at I (Table 14-12, Figure A):

$\frac{A_1}{A} = \frac{450 \times 400}{400 \times 250} = 1.8$ (Use 2); C = 0.05;

Velocity = 1.0/0.45 × 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 smooth, long radius without vanes (Table 14-10, Figure F) having a R/W ratio of 2.0. As H/W = 250/400 = 0.63, the loss coefficient of 0.17 (by interpolation) is used.

By applying the values of the 340mm duct diameter and the duct velocity of 5.6 m/s to the "Reynolds Number Correction Factor Chart" on page 14.17, it is found that a correction factor must be used. The actual average velocity is:

V = 0.5/0.4 × 0.25 = 5.0 m/s (Equation 5-9).

The equations under Note 3 on page 14.18 are solved to allow the correction factor to be obtained.

$D = \frac{2\ HW}{H + W} = \frac{2 \times 400 \times 250}{400 + 250} = 307.7\text{mm};$
$R_e = 66.4\ DV = 66.4 \times 307.7 \times 5.0$
$R_e = 102,156$
$R_e 10^{-4} = 10.22$

From the table (Note 3) when R/W > 0.75, the correction factor of 1.29 is obtained and the V_p of 15.1 Pa for 5 m/s is used.

Fitting loss $= C \times V_p \times K_{Re}$
$= 0.17 \times 15.1 \times 1.29 = 3.31$ 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 × 15.1) is 0.74 pascals lower than the value used. On a long, winding 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:

$$\text{Damper Loss} = C \times V_p = 0.04 \times 15.1$$
$$= 0.60 \text{ Pa (enter 1 Pa on line 19).}$$

Figure T of Table 14-14 (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 system airflow is going through the tap. The closest duct configurations found in Chapter 14 would be the mitered elbows in Table 14-10, Figures C, D or E. The average loss coefficient value for a 90° turn from these figures is 1.2, which is the recommended value to use until additional research in the SMACNA program establishes duct fitting loss coefficients for these configurations (see Section H of Chapter 5).

Obviously, if there was ample room in the ceiling, the use of a vaned elbow 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:

$\text{Fitting loss} = C \times V_p = 1.2 \times 15.1 = 18.1$ Pa; (enter 18 Pa on line 20).

The pressure loss of the 350mm diameter diffuser on the drawing (Figure 8-1) at J includes the pressure losses for the damper behind the diffuser. The 35 pascal loss is entered on line 21 in column L.

In Table 8-1, the pressure losses on lines 16 through 21 in column L are totalled (68 Pa) and the value entered on line 21 in column M (in black) and on line 16 in column N (in red). 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 pascals for the duct run B to J (the assumed main duct run) being entered on line 4. This total is added to the 225 pascals on line 3 of column M (Fan Plenum B) for the total pressure loss of 362 pascals, the design total pressure at which supply fan B must supply 4000 litres per second. The value of 362 pascals is entered on line 1 in columns N and O. (The numbers in columns N and O are shown in red to indicate that they 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 per 100 feet 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 that all of the fittings in the main run had a static regain (included in the loss coefficients) with each lowering of the airstream velocity which reduced the actual pressure loss of each section.

g) 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 metres of duct from F to G (6 Pa) and the elbow at G (4 Pa), a total loss of 10 pascals. 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 (270 Pa) and the result entered on line 1 (column M) of the table in Figure 8-1(a) using a new duct sizing form.

Referring again to Table 14-14, Figure W (used before for the wye at F), and using the same ratios as before, ($A_b/A_s = 1.0$; $A_b/A_c = 0.5$; $Q_b/Q_c = 0.5$), the *branch* loss coefficient $C = 0.52$.

$$\text{F fitting loss} = C \times V_p = 0.52 \times 33.9$$
$$= 17.6 \text{ Pa (Enter 18 Pa on line 2).}$$

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 pascals for fitting F is compared to the 10 pascals computed above for duct EG and elbow G. As the difference between them of 8 pascals is within the 12 pascals allowable design difference, the fitting used at F was a good selection. However, the A_1M duct run will have a 8 pascals greater pressure loss than the A_1J duct run. So the assumed "longest run" did not have the greatest pressure loss although again the difference was within 12 pascals. This also confirms the need for the use of balancing dampers in each of the 550mm × 400mm ducts at F.

The information for the "branch" volume damper at F can be copied from line 12 of Table 8-1 (as all conditions are the same) and entered on line 3 of Table

8-1(a). The calculations then are made for the 3 metres of 550mm × 400mm duct from F to K:

FK duct loss = 3 × 0.95 = 2.9 Pa;
(enter 3 pascals 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 8-1(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 8-1 are totalled (75 Pa) and entered on line 5 in columns M and N of Table 8-1(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 pascals for the fan B duct system (line 1, column 0). *This loss only is 5 pascals higher than the A_1J duct system pressure loss (Table 8-1), but it is the total pressure loss value to be used in the selection of Fan B.*

h) Duct Section EN—Using the balance of the duct sizing form (Table 8-1(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 totalling lines 1 to 7 of Table 8-1 and entering the 266 pascals 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 450mm × 400mm being the selected rectangular size) using the same 0.95 Pa/m friction loss rate.

EN duct loss = 3 × 0.95 = 2.9 Pa;
(enter 3 pascals on line 8).

The data used before for computing the "main" loss coefficient for wye E (Table 14-14, figure W) is again used to obtain the "branch" loss coefficient (see "Duct Section EF").

$A_b/A_s = 0.5$, $A_b/A_c = 0.5$, $Q_b/Q_c = 0.25$

The preliminary calculations to branch EN are verified (see text of "Duct Section EF").

C (branch) = 0.44 (by interpolation)

E fitting loss = $C \times V_p = 0.44 \times 35.7$
$= 15.7$ pascals;
(enter 16 pascals on line 9).

The loss values in column L (3 + 16 = 19) are totalled and entered on line 9 in column M.

i) Duct Section NP—Data for the 17 metre 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, 340mm diameter and 400mm × 250mm equivalent rectangular size.

NP duct loss = 17 × 0.95 = 16.2 pascals;
(enter 16 pascals on line 10).

At N, a 45° entry tap is used for branch duct NS and a 30° transition is used to reduce the duct size for the run to P. From Table 14-12, Figure A:

$A_1/A = 450 \times 400/400 \times 250 = 1.8$ (Use 2)

$C = 0.05$ for $\Theta = 30°$,

Velocity = 1.0/0.45 × 0.4 = 5.6 m/s;

V_p (Table 14-7) = 18.9 pascals;

N fitting loss = $C \times V_p = 0.05 \times 18.9$
$= 0.95$ pascals;
(enter 1 pascal on line 11).

The volume damper at N has the same numbers as used above for the damper at J:

Damper loss = $C \times V_p = 0.04 \times 15.1$
$= 0.6$ pascals;
(enter 1 pascal on line 12).

At O, a smooth radius elbow with one splitter vane is selected (Table 14-10, Figure G):

R/W = 0.25, H/W = 250/400 = 0.63; C = 0.12 (by interpolation);

O fitting loss = $C \times V_p = 0.12 \times 15.1$
$= 1.8$ pascals;
(enter 2 pascals on ine 13).

The cumulative loss of 20 pascals (16 + 1 + 1 + 2) is entered on line 13 in column M.

j) Duct Section PQ—Data for the last 6 metres of duct is obtained from Figure 14-2 and Table 14-3: 250 l/s, 4.6 m/s, 270mm diameter, and 250mm × 250mm equivalent rectangular size.

PQ duct loss = 6 × 0.95 = 5.7 pascals;
(enter 6 pascals on line 14).

The loss coefficient for transition P is obtained from Table 14-12, Figure A (converging flow) using $\Theta = 45°$:

$A_1/A = 400 \times 250/250 \times 250 = 1.6$ (Use 2);

C = 0.06, Vel. = 5.0 m/s (from the 400 × 250 duct), V_p = 15.1 pascals;

P fitting loss = $C \times V_p = 0.06 \times 15.1$
$= 0.9$ pascals;
(enter 1 pascal on line 15).

The fitting at Q is a mitered 90° change of-size elbow (Table 14-10, Figure E):

$H/W = 250/250 = 1.0$; $W_1/W = 400/250 = 1.6$;

Velocity $= 0.25/0.25 \times 0.25 = 4.0$ m/s; $V_p = 9.6$ pascals.

A fitting loss coefficient of 0.90 is selected. Then referring to Note 2 on Page 14.17, plotting the data on the "Reynolds Number Correction Factor Chart" indicates that a correction factor will be required.

$$D = \frac{2 \times 250 \times 250}{250 + 250} = 250 \text{ mm;}$$

$R_e = 66.4\ DV = 66.4 \times 250 \times 4.0 = 66{,}400$;

$R_e 10^{-4} = 6.64$; $K_{Re} = 1.08$

Q fitting loss $= 0.90 \times 4 \times 1.08 = 3.9$ pascals;
(enter 4 pascals on line 16).

The pressure loss of 32 pascals on the drawing (Figure 8-1) for the 400 × 250 grille is entered on line 17.

The pressure losses on lines 14-17 in column L are totalled (43 pascals) 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 pascals, which is entered on line 7 in columns N and O.

The A_1M duct run pressure loss of 367 pascals is 19 pascals higher than the 348 pascals pressure loss of the A_1Q duct run, giving a system that is above the 12 pascals suggested good design difference for branch ducts. Nevertheless, balancing dampers in the branch ducts at N should allow the TAB technician to properly balance the system.

k) Duct Section NS—The pressure losses from A_1 to N (lines 7 to 9) are totalled (285 pascals) 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 $= 3 \times 0.95 = 2.9$ pascals;
(enter 3 pascals on line 19).

A 45° entry rectangular tap is used for the branch duct at N. From Table 14-14, Figure N:

$V_b = 0.5/0.4 \times 0.25 = 5.0$ m/s;

$V_c = 1.0/0.45 \times 0.4 = 5.6$ m/s;

$V_b/V_c = 5.0/5.6 = 0.89$ (Use 1.0);

$Q_b/Q_c = 500/1000 = 0.5$; $C = 0.74$;

V_p (upstream) of 5.6 m/s = 18.9 pascals;

N Fitting Loss $= C \times V_p = 0.74 \times 18.9$
$= 14$ Pascals;
(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 pascals can be entered on line 21 in column M.

Using similar data from line 14:

RS duct loss $= 6 \times 0.95 = 5.9$ pascals;
(enter 6 pascals on line 22).

Using similar data from line 15:

R Transition loss $= C \times V_p = 0.06 \times 15.1$
$= 0.91$ pascals;
(enter 1 pascal on line 23).

S Elbow loss $= C \times V_p \times K_{Re}$ (from line 16)
$= 0.90 \times 9.6 \times 1.08$
$= 3.9$ pascals;
(enter 4 pascals on line 24).

S Grille loss (from Figure 8-1) = 32 pascals;
(Enter on line 25).

The losses for Run RS in column L are totalled and the 43 pascal 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 A_1 to S of 346 pascals 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.

l) Additional Discussion—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 $\theta = 0°$ (wide open). The preliminary damper setting angle θ can be calculated in this situation as follows (*assuming* a total system loss difference of 20 pascals between points S and Q for this example):

System loss difference = 20 pascals

N damper loss (set at 0°) = 1 pascal

N damper loss (set at ?) = 21 pascals (20 + 1)

Damper loss $= C \times V_p$ or $C =$ Damper loss$/V_p$

$C = 21\ \text{Pa}/15.1 = 1.39$

Referring back to Table 14-18, Figure B, the loss coefficient of $C = 1.39$ would require a damper angle θ of about 21° (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).

Other advantages of the above duct sizing procedures are that 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 in Chapter 4). After the duct system is sized, these static pressure "flags" should be noted on the drawings as shown on Figure 8-1 to obtain the most economical duct fabrication and installation costs.

Building pressure allowance for supply air duct systems should be determined from building ventilation requirements considering normal building infiltration. Allowance in the range of 5 to 25 pascals 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, fire and smoke code requirements, etc.

Finally, the system pressure loss check list in Figure 9-1 of Chapter 9 should be used to verify that all system component pressure losses have been included in the fan total pressure requirements, and that some allowance has been added for possible changes in the field. These additional items should be shown on the duct sizing work sheets.

E RETURN AIR (EXHAUST AIR) DUCT SYSTEM-SIZING EXAMPLE NO. 2

The exhaust air duct system of fan "Y" shown in Figure 8-1 will be sized using lower main duct velocities to reduce the fan power requirements. This will conserve energy and, therefore, lower the daily 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 B—"Other Factors Affecting Duct System Pressures" of Chapter 5. 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 $= TP_d - TP_s - Vp_d$ (Equation 5-4)

Fan SP $= TP_d - (-TP_s) - Vp_d$

Fan SP $= TP_d + TP_s - Vp_d$

as $TP = SP + Vp$, then:

Equation 8-2

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 8-2, 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.

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 8-2 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 14-16, Figure I:

Z Fan pressure loss $= C \times V_p = 1.5 \times 40$

$= 60$ pascals;

(enter in column L on line 2).

The plenum loss total of 135 Pa is entered on line 2 in Column M.

2. Exhaust Air System

a) Duct Section YW—The 750mm inlet duct to the fan is connected to an inlet box (see Figure 6-20 of Chapter 6). The inlet box exhaust duct connection size is 600mm (0.8 × 750) × 1160mm (1.55 × 750). The given loss coefficient (C) is 1.0. The return air duct connection velocity is:

Velocity $= 4/0.6 \times 1.16 = 5.8$ m/s (Equation 5-9);

From Table 14-7, $V_p = 20.3$ pascals;

Inlet box loss $= C \times V_p = 1.0 \times 20.3$

$= 2.03$ pascals;

(enter 20 pascals on line 3).

Using 4000 l/s and 8 m/s from the chart in Figure 14-2, the following is established: duct friction loss of 0.7 Pa/m, 800mm duct diameter; and from Table 14-3, a 750mm × 750mm rectangular equivalent size duct.

YW duct loss $= 10\text{m} \times 0.7$ Pa/m $= 7$ pascals;

(enter on line 4).

The transition at Y (Table 14-11, Figure B) has a 30° total slope.

$A_1/A = 600 \times 1160/750 \times 750 = 1.24$ (Use 2);

Loss Coefficient (C) = 0.25 for 30°;
Velocity = 4.0/0.75 × 0.75 = 7.1 m/s;
V_p = 30.4 pascals;
Y transition loss = $C \times V_p = 0.25 \times 30.4$
= 7.6 pascals;
(enter 8 pascals on line 5).

The total for section YW (20 + 7 + 8 = 35 Pa) is entered on line 5 in column M.

b) Duct Section WU—Using 3000 l/s and 0.7 Pa/m, 7.4 m/s is established along with a duct diameter of 730mm. Using Table 14-3, a rectangular size of 750mm × 600mm is selected (keeping one side the same size).

WU duct loss = 32m × 0.7 Pa/m = 22.4 pascals; (enter 22 pascals on line 6).

A converging 45° entry tee will be used at W (see Table 14-13, Figure F) with the velocity pressure of the downstream airflow velocity (4000 m/s).

To obtain the "main" loss coefficient, the note in Fitting 14-13 F refers to Fitting 14-13B:

Using Table 14-13B (Main Coefficient):

$Q_b/Q_c = 1000/4000 = 0.25$; C = 0.33
(by interpolation);

W fitting loss = $C \times V_p = 0.33 \times 30.4$
= 10.0 pascals;
(enter on line 7).

Table 8-2 DUCT SIZING, EXHAUST AIR SYSTEM—EXAMPLE NO. 2

DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE ______ PAGE 1 of 1
PROJECT Sample Building LOCATION First Floor SYSTEM Exhaust Air

	A DUCT RUN	B SECTION	C ITEM		D FLOW (L/S)	E FRICTION PER METRE	F VELOCITY (M/S)	G Vp (PA)	H LOSS COEFF	I EQUIV DIAM	J RECTANGULAR SIZE	K CORR FACT	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1	Plenum Z	A, 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	32m	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	6m	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	–	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	180°	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																

NOTES: *Indicates duct lining used. Sizes are interior dimensions.

The diverging flow transition at W with an included angle of 30° uses Table 14-11, Figure E because of the change of only one duct dimension.

$A_1/A = 750 \times 750/750 \times 600$
$= 1.25$ (Use 2);
$C = 0.20$;
upstream section velocity $= 3.0/0.75 \times 0.6$
$= 6.7$ m/s;
$V_p = 27.0$ for 6.7 m/s
(From Table 14-7 or by calculation);
W trans. fitting loss $= C \times V_p = 0.20 \times 27.0$
$= 5.4$ pascals;
(enter 5 pascals on line 8).

Using a radius elbow without vanes (Table 14-10, Figure F) at V, the following data is used:

$H/W = 600/750 = 0.8$, $R/W = 2.0$, $C = 0.16$

Again, using the equivalent diameter of 730mm 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 $= C \times V_p = 0.16 \times 27.0$
$= 4.3$ pascals
(enter 4 pascals on line 9).

As before, the total section loss of 41 pascals is entered in column M.

c) Duct Section UT—The static pressure loss (the total pressure loss is always the same as the static pressure when there is no velocity change) for the duct section UT is:

UT duct loss $= 6 \times 0.7 = 4.2$ pascals;
(enter 4 pascals on line 10).

From Figure 14-2 where a 540mm diameter duct and 6.2 m/s was obtained for 1500 l/s, a 600mm × 400mm rectangular duct is selected from Table 14-3. A converging 90° tee fitting (Table 14-13, Figure D) will be used at U, but again the "main" loss coefficient is obtained from Figure 14-13B.

$Q_b/Q_c = 1500/3000 = 0.5$; $C = 0.53$;
U Fitting loss $= 0.53 \times 27.0$ (downstream V_p)
$= 14.3$ pascals;
(enter 14 pascals on line 11).

The U transition loss coefficient is found in Table 14-11, Figure B, and the following data computed:

$A_1/A = 750 \times 600/600 \times 400 = 1.88$ (use 2),
$\theta = 30°$, $C = 0.25$;
Vel. $= 1.5/0.6 \times 0.4 = 6.3$ m/s
$V_p = 23.9$ Pa (upstream duct).

U fitting loss $= C \times V_p = 0.25 \times 23.9$
$= 6.0$ pascals;
(enter on line 12)

The pressure loss for the change of size elbow at T will again be computed using Table 14-10, Figure E (Caution should be used to determine airflow direction):

$H/W = 400/1200 = 0.33$, $W_1/W = 600/1200 = 0.5$;

$C = 1.8$
Vel. of the upstream section (grille size)
$= 1.5/1.2 \times 0.4 = 3.1$ m/s, $V_p = 5.8$ pascals;

T fitting loss $= C \times V_p = 1.8 \times 5.8$
$= 10.4$ pascals;
(enter 10 pascals on line 13).

Turning vanes could be added to the change of size mitered elbow, but no loss coefficient tables are available. One could speculate that if single-blade turning vanes reduce the $C = 1.2$ of a standard 90° 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 pascals for the exhaust grille at T is taken from Figure 8-1 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 pascals entered on line 1 in columns N and O.

d) 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 pascals for duct run W to T (line 6, column N) is also the total pressure loss requirement for the short 6 metre duct run (12 pascals 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, and a duct friction loss rate of 1.7 Pa/m, and 400mm diameter is selected for the 1000 l/s flow rate (Figure 14-2). Duct lining of 25mm thickness (correction factor = 1.93 from Figure 14-3 and Table 14-1 [Rough]) 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 $= 6\text{m} \times 1.7 \text{ Pa/m} \times 1.93$
$= 19.7$ pascals;
(enter 20 pascals on line 17).

Select the rectangular size of 350mm × 400mm from Table 14-3.

The converging 45° entry fitting used at W (Table 14-13, Figure F) is reviewed again to determine the branch loss coefficient.

$Q_b/Q_c = 0.25$, Velocity (V_c) $= 7.1$ m/s,
$V_p = 30.4$ Pa, $C = -0.37$;

W fitting loss $= C \times V_p = -0.37 \times 30.4$
$= -11.2$ pascals.

As 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. A smaller grille with a higher pressure loss could be used if a greater noise level could be tolerated.

If a straight rectangular tap was used (Table 14-13, Figure D) instead of the 45° entry tap, the loss coefficient would then become 0.01, a more appropriate selection. This is one of the reasons why higher loss fittings remain in the tables.

An inefficient transition at grille X also will help build up the loss (note airflow direction). Figure A is a rectangular converging transition in Table 14-12. With $A_1/A = \dfrac{900 \times 400}{350 \times 400} = 2.57$ and $\Theta = 180°$ (abrupt), $C = 0.30$. The downstream velocity must be used to determine the V_p used in the computations:

Velocity (downstream) $= 1.0/0.35 \times 0.4$
$= 7.1$ m/s;
$V_p = 30.4$ pascals;
X fitting loss $= C \times V_p = 0.30 \times 30.4$
$= 9.1$ pascals
(enter 9 pascals on line 20).

X grille loss (from Figure 8-1) = 20 pascals
(enter on line 20).

Subtracting 38 pascals (the total of lines 17 to 20) from the 95 pascal duct run WT pressure loss shown on line 6 in column N, leaves 57 pascals of pressure for the balancing damper to dissipate.

Damper loss coefficient $C = TP/V_p$
$= 57\text{Pa}/30.4\text{ Pa}$
$= 1.88.$

From Table 14-18, Figure B, a damper set about 23° (by interpolation) has a loss coefficient of 1.88 that will balance the branch duct WX. The total of 95 pascals (adding lines 17-21) is shown on line 21 in column M and on line 17 in column N.

A perforated plate (Table 14-17, Figure B) is a non-adjustable alternate solution. If a 3mm thick perforated plate was used instead of the balancing damper, the calculation procedure would be as follows (see Table 14-17, Figure B):

Assuming 16mm diameter holes,
$t/d = 3/16 = 0.19$

With $C = 1.88$ (from above),
$n = 0.60$

$n = \dfrac{A_p}{A}; \; A_p = n \times A$

A_p (flow area of perf. plate) $= 0.60 \times 350 \times 400$
$= 84{,}000 \text{ mm}^2$

No. of holes $= A_p$/area of a 16mm diameter hole
No. of holes $= 84{,}000/8^2\pi = 418$

F SUPPLY AIR DUCT SYSTEM SIZING EXAMPLE NO. 3

1. Introduction

Higher pressure supply air systems (over 750 pascals) usually are required for the large central station HVAC supply air duct distribution systems. Because of higher fan power requirements, ASHRAE Standard 90.1-1989 provisions will cause the designer to analyze lower pressure duct systems against the on-going (and 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. However, the choice of fitting loss coefficient tables in Chapter 14 require some designers to use a new approach when designing these systems.

2. Design Procedures

After analyzing a duct system layout, the chart in Figure 14-2 of Chapter 14 is used to select an "approximate" initial velocity and a pressure loss (pascals per metre) 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 (litres per second) of the duct sections and the selected velocity (metres per second), the duct diameters and friction loss rates also may be obtained from Figure 14-2. When rectangular duct sizes are to be used, selection may be made from the chart in Table 14-3, 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.

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 14-7 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 Chapter 14 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 9. 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 (comparing loss coefficients to obtain the most efficient fitting). If the initial system airflow is over 15,000 l/s, the velocity can be held constant (with an increase in the duct friction rate) until the system airflow drops below 15,000 l/s. Then the duct friction rate generally should remain constant (equal friction).

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 totalled 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 which 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 rates, and a life cycle cost analysis will probably dictate that lower operating costs be considered more important than lower first costs and space saving requirements.

3. Supply Air System

Table 8-3 is the tabulation of design and computation data obtained when sizing the 10,000 l/s supply duct system shown in Figure 8-2. The 95 metre 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 l/s 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 an airflow of 500 l/s.

a) 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 1100mm × 810mm, a discharge velocity of 11 m/s (velocity pressure = 75 pascals), and a blast area/outlet area ratio of 0.6.

Elbow B is sized 1100mm × 810mm (so that it is similar to the outlet size) and a radius elbow (R/W = 1.5) is selected. It is located 660mm 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:

$$\text{Equiv. Diam.} = \sqrt{\frac{4 \times 1100 \times 810}{\pi}} = 1065\text{mm}$$

% Effective duct

$$= \frac{\text{straight duct length} \times 100}{\text{Vel./5 (2.5 min.)} \times \text{Equiv. Diam.}}$$

$$\text{\% Effective duct} = \frac{660 \times 100}{2.5 \times 1065} = 24.8\%$$

From Table 6-2, System Effect Curve R-S for a 0.6 blast area ratio and 25% Effective Duct is used with Figure 6-1 to find the System Effect pressure loss of 72 pascals (based on 11 m/s). 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 8-3. *Again it is noted that the 72 pascals 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 14-10, figure F) with R/W = 1.5 and H/W = 1100/810 = 1.36.

Average Velocity = Q/A (Equation 5-9)
= 10/1.1 × 0.81 = 11.2 m/s

The velocity pressure (V_p) of 76 pascals is obtained from Table 14-7 for a velocity of 11.2 m/s. A quick check of the "Reynolds Number Correction Factor" chart on page 14.17 shows that no correction is needed.

B fitting loss = C × V_p = 0.15 × 76
= 11.4 pascals (Use 11 Pa).

The total pressure loss of 83 pascals for the plenum is entered on line 2 in column M.

b) Duct Section CF—Round spiral duct with an absolute roughness of 0.0003 feet will be used in this supply duct system. For the 27 metres of duct in section CF and using an assumed velocity of 16.0 m/s, it falls right on the closest standard size duct diameter of 900mm (from the chart of Figure 14-2). 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 14-2 is based on the same absolute roughness.

CF duct loss = 27 × 2.4 = 64.8 pascals;
(enter 65 pascals on line 3.)

The transition at C will be converging, rectangular to round (Table 14-12, figure A) with $A_1/A = 1100 \times 810/(450)^2\pi = 1.40$ and $\Theta = 20°$; C = 0.05. The velocity

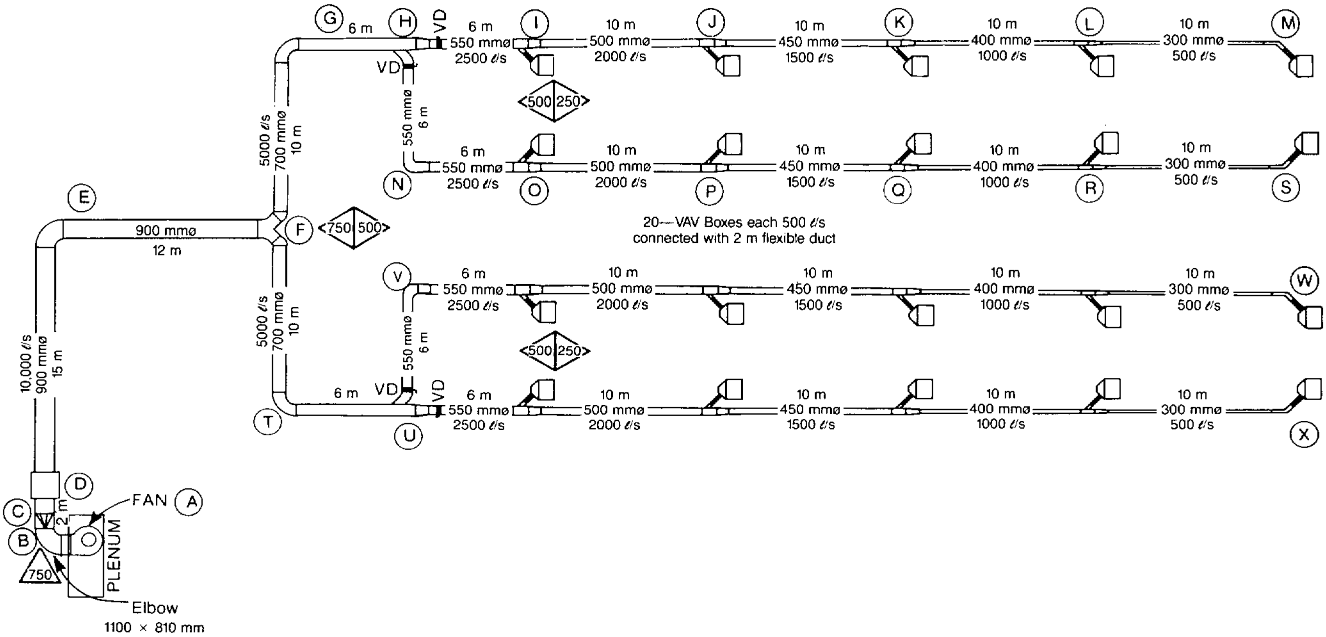


Figure 8-2 SUPPLY AIR DUCT SYSTEM FOR SIZING EXAMPLE NO. 3 (METRIC)

pressure used is that of the downstream section: 154 pascals for 16 m/s (Table 14-7).

C transition loss $= C \times V_p$
$= 0.05 \times 154$ (leaving V_p)
$= 7.7$ Pa
(enter 8 Pa on line 4).

The pressure loss for a medium attenuation 900mm diameter sound trap of 65 pascals is obtained from Chapter 9. A preliminary loss also can be obtained from manufacturer's data sheets. The data is entered on line 5.

The smooth radius, 90° round elbow at E has an R/D ratio = 1.5; C = 0.15 (Table 14-10, Figure A).

E elbow loss $= C \times V_p = 0.15 \times 154$
$= 23.1$ pascals (line 6).

The pressure losses of the four items in duct section CF are added and the 161 pascals total is entered in column M on line 6.

c) 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 680mm (use 700) diameter (Figure 14-2). A velocity of 13.0 m/s, 2.0 Pa/m and the related V_p of 102 pascals is used for further calculations based on the 700mm diameter standard size duct.

FH duct loss $= 16 \times 2.0 = 32$ Pa;
(enter 32 pascals on line 7).

Using a 45° round wye fitting (Table 14-14, Figure Y) with 45° elbows at F, $V_{lb}/V_c = 13.0/16 = 0.81$; C = 0.29 (by interpolation).

F wye fitting loss $= C \times V_p = 0.29 \times 154$
$= 44.7$ pascals;
(enter 45 pascals on line 8.)

The 45° round elbow (RD = 1.5) at F will use the same loss coefficient as the 90° elbow above (Table 14-10, Figure A) multiplied by the 0.6 correction factor for 45° (Note 1). V_p for 13.0 m/s = 102 pascals.

F elbow fitting loss $= 0.15 \times 102 \times 0.6$
$= 9.2$ Pa (line 9).

The 90° round elbow at G uses the same values without the correction factor.

G elbow fitting loss $= 0.15 \times 102$
$= 15.3$ Pa (line 10).

The losses in column L again are totalled and 101 pascals is entered in column M.

d) 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 525mm diameter duct size. Using a standard size of 550mm, velocity = 10.6 m/s; $V_p = 68$ pascals, and the friction loss rate = 1.9 m/s.

HO duct loss $= 12 \times 1.9 = 22.8$ Pa;
(enter 23 Pa on line 11).

At point H in the duct system, the branch coefficient is obtained for the diverging 45° round wye with a conical main and branch with a 45° elbow (Table 14-14, Figure M):

$$V_b/V_c = \frac{10.6}{13.0} = 0.82, \ C = 0.51;$$

$V_p = 102$ Pa (13.0 m/s).
H wye (branch) loss $= C \times V_p = 0.51 \times 102$
$= 52.0$ pascals (line 12).

The 90° round elbow is calculated as the above 90° ell and the loss coefficient for the balancing damper is obtained from Table 14-18, Figure A ($\Theta = 0°$); C = 0.20; V_p for 10.6 m/s = 68 pascals.

H damper loss $= C \times V_p = 0.20 \times 68$
$= 13.6$ Pa (line 13)

N elbow fitting loss $= C \times V_p = 0.15 \times 68$
$= 10.2$ Pa (line 14)

The total for the HO duct section (99 pascals) is entered in column M.

e) Duct Section OP—For 2000 l/s at 2.4 Pa/m, the closest standard size duct is 500mm diameter. Using the 500mm 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 14-7).

The 45° round diverging conical wye at point 0 (Table 14-14, figure C) requires that the "main" coefficient C be obtained from Table 14-14A.

$V_s/V_c = 10.2/10.6 = 0.96$; but when there is little or no change in velocity, the table indicates that there is no dynamic loss, i.e. C = 0 for $V_s/V_c = 1.0$. Interpolating gives a questionable loss coefficient of 0.004, which multiplied by the V_p of 68 pascals gives a loss of 0.3 pascals. However, a minimum loss coefficient of 0.01 is used to be on the safe side.

0 Wye (main) loss $= 0.01 \times 68 = 0.7$ pascals;
(enter 1 pascal on line 16).

The 60° transition from 550mm diameter to 500mm diameter does have a dynamic pressure loss and the fitting loss coefficient is obtained from Table 14-12, Figure A.

$A_1/A = (275)^2\pi/(250)^2\pi = 1.21$ (use 2); $C = 0.06$;

0 transition loss $= C \times V_p$ (downstream)
$= 0.06 \times 63$
$= 3.8$ pascals (line 17).

The section loss of 25 pascals is entered in column M.

f) 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 450mm diameter (Figure 14-2). Using the 450mm 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 14-7).

For the 45° round conical wye at P (Table 14-14A):

$V_s/V_c = 9.0/10.2 = 0.88$, $C = 0.01$;

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° transition at P: $A_1/A = (250)^2\pi/(225)^2\pi = 1.23$ (Table 14-12A), $C = 0.06$;

P transition loss $= C \times V = 0.60 \times 49 = 2.9$ pascals;
(enter 3 pascals on line 20).

This section loss of 23 pascals is entered in column M.

g) Duct Section QR—The selection of a standard 400mm diameter duct for 1000 l/s (Figure 15-2 indicates a 1.8 Pa/m friction loss rate and a velocity of 8.0 m/s).

QR duct loss $= 10 \times 1.8 = 18$ pascals (line 21).

V_p for 8.0 m/s = 39 Pa (Table 14-7).

Again using the same type of wye at Q:

$V_s/V_c = 8.0/9.0 = 0.89$, $C = 0.01$;

Q wye (main) loss $= C \times V_p = 0.01 \times 49$
(upstream V_p)
$= 0.5$ Pa
(enter 1 Pa on line 22).

Q Transition: $A_1/A = (225)^2\pi/(200)^2\pi$
$= 1.27$ (Table 14-12A),
$C = 0.06$;

Q Trans. fitting loss $= C \times V_p = 0.06 \times 39$
$= 2.3$ (line 23).

The section loss of 21 pascals is entered in column M.

h) Duct Section RS—Using an addition duct sizing form to record the data [Table 8-3(a)], the 500mm duct size at 2.4 Pa/m would be between the 250mm and 300mm standard duct sizes. The 250mm diameter duct would have a much higher pressure loss, so the 300mm duct at 1.7 Pa/m friction loss and 6.8 m/s velocity would be the better selection.

RS duct loss $= 10 \times 1.7 = 17$ Pa (line 1).

V_p for 6.8 m/s = 28 pascals (Table 14-7).

R wye fitting: $V_s/V_c = 6.8/8.0 = 0.85$, $C = 0.01$ (Figure 14-14A);

R wye (main) loss $= C \times V_p = 0.01 \times 39$
$= 0.04$ pascals
(Enter 1 pascal on line 2).

R transition: $A_1/A = (200)^2\pi/(150)^2\pi = 1.78$;
$C = 0.06$ (Table 14-12A);

R transition loss $= C \times V_p = 0.06 \times 28$
$= 1.7$ pascals (line 3).

A 45° elbow at the end of the duct is connected to the VAV box by a 2 metre piece of 300mm diameter flexible duct. The correction factor for the flexible duct is obtained from the chart in Figure 14-3 using Table 14-2 as a guide. Bends of 30° 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 14-4 also contains a correction factor for unextended (compressed) flexible duct.

S 45° elbow fitting loss $= 0.15 \times 28 \times 0.60$
$= 2.5$ Pa (line 4)

S flex. duct loss $= 2\text{m} \times 1.7\ \text{Pa/m} \times 1.95$
$= 6.6$ pascals (line 5).

An estimate of 65 pascals is made for the downstream side ductwork and diffuser from the VAV box. This is added to the VAV box pressure loss of 75 pascals and the total (140 pascals) is entered on line 6. The total pressure loss of 170 pascals for the RS section is entered in columns M and N and also in column N on line 24 of page 1 (Table 8-3) of the duct sizing work sheet. Working from the bottom to the top of the form, the section pressure losses are totalled in column N with the *total pressure* loss for the supply duct system of 683 pascals being entered on line 1 in columns N and O.

i) Recap—the same procedure is used to size the other segments of the supply duct system; or if the

Table 8-3 DUCT SIZING, SUPPLY AIR SYSTEM—EXAMPLE NO. 3

DUCT SIZING WORK SHEET (METRIC UNITS)

DATE ______ PAGE 1 of 2
PROJECT Sample Building LOCATION Conf. Area SYSTEM Supply Air

	A DUCT RUN	B SECTION	C ITEM		D FLOW (L/S)	E FRICTION PER METRE	F VELOCITY (M/S)	G Vp (PA)	H LOSS COEFF	I EQUIV DIAM	J RECTANGULAR SIZE	K CORR FACT	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1	Plenum A	A	–	System effect	10,000	–	11.0	–	–	1065	1100 x 810	RS	72		683	683
2	"	B	90°	Elbow	10,000	–	11.2	76	0.15	1065	1100 x 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 x 810/900 ϕ	–	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	1500/1000	–	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)							- (Page 2)	–	–		170	
25																

NOTES: *Indicates duct lining used. Sizes are interior dimensions.

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 No. 1, several fittings with higher pressure losses or "high loss" VAV boxes can allow a duct run that was not the originally selected run for design computations, actually to be the duct run with the greatest pressure loss.

Assuming that the return air duct system of Example No. 3 (not shown) had an approximate total pressure loss of 500 pascals, the output of the system supply fan would need to be 10,000 l/s at 1182 pascals (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 past the wye fitting at F is in the low pressure range (under 500 pascals), even though there are velocities up to 13 m/s.

This is the reason that it is extremely important to indicate static pressure "flags" on the drawings after the duct system is sized (as in indicated in Figure 8-2). Table 2-5 indicates the relative costs of fabrication and installation of the 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.

In the first edition of this "HVAC Duct System Design" manual, this same duct system example was sized in U.S. units using a "constant" velocity of approximately 14 m/s. The duct sizes ranged from 900mm to 250mm diameter at 800 pascals total pressure, instead of from 900mm to 300mm at 683 pascals total pressure. The "modified" equal friction method

Table 8-3(a) DUCT SIZING, SUPPLY AIR SYSTEM—EXAMPLE NO. 3 (CONT.)

DUCT SIZING WORK SHEET
(METRIC UNITS)

DATE ______ PAGE 2 of 2

PROJECT Sample Building LOCATION Conf. Area SYSTEM Supply Air

	A DUCT RUN	B SEC-TION	C ITEM		D FLOW (L/S)	E FRICTION PER METRE	F VELOCITY (M/S)	G Vp (PA)	H LOSS COEFF	I EQUIV DIAM	J RECTANGULAR SIZE	K CORR FACT	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1	Run CS (Cont)	RS	10m	Duct	500	1.7	6.8	28	–	300	–	–	17		170	
2	″	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			
6	″	S	–	VAV box	500	–	–	–	–	–	–	–	140	170		
7																
8																
9																
10																
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NOTES: *Indicates duct lining used. Sizes are interior dimensions.

of design allowed a 15 percent lower system total pressure, which results in a yearly savings of approximately $1132 based on the example in Chapter 2 where electrical energy costs were 9 cents per kW-hour. On-going costs of operating a system are extremely important, but savings in initial system costs also can conserve energy.

G EXTENDED PLENUM DUCT SIZING

1. Introduction

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; i.e., equal friction method, etc. Though there is a lack of published data concerning extended plenum use and design, extensive field testing, both in experimental form and in many actual installations throughout the country, have proven the concept. An *extended plenum* is a trunk duct of constant size, usually at the discharge of a fan, fan-coil unit, mixing box, variable air volume (VAV) box, etc., extended as a plenum to serve multiple outlets and/or branch ducts.

A *semi-extended plenum* is a trunk design *system* utilizing the concept of extended plenum incorporating a *minimum* number of size reductions due to decreasing volume.

2. 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 length of straight duct to fitting ratio.

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.

d) 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 that low velocities, which could develop, might result in excessive heat gain or loss through the duct walls.

A limiting factor to be considered when using the extended plenum method is that low velocities, which could develop, might result in excessive heat gain or loss through the duct walls.

3. 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 pascals and duct velocities up through 15 metres per second. Other specific design considerations include:

a) Branch takeoffs from the trunk duct should preferably be round duct connecting at a 45° angle. If rectangular branches are used, a 45° 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.

4. Comparison of Design Methods

Figures 8-3 and 8-4 illustrate identical medium pressure systems differing only in the trunk duct sizing techniques used. The trunk duct system shown in Figure 8-3 has been sized by the equal friction

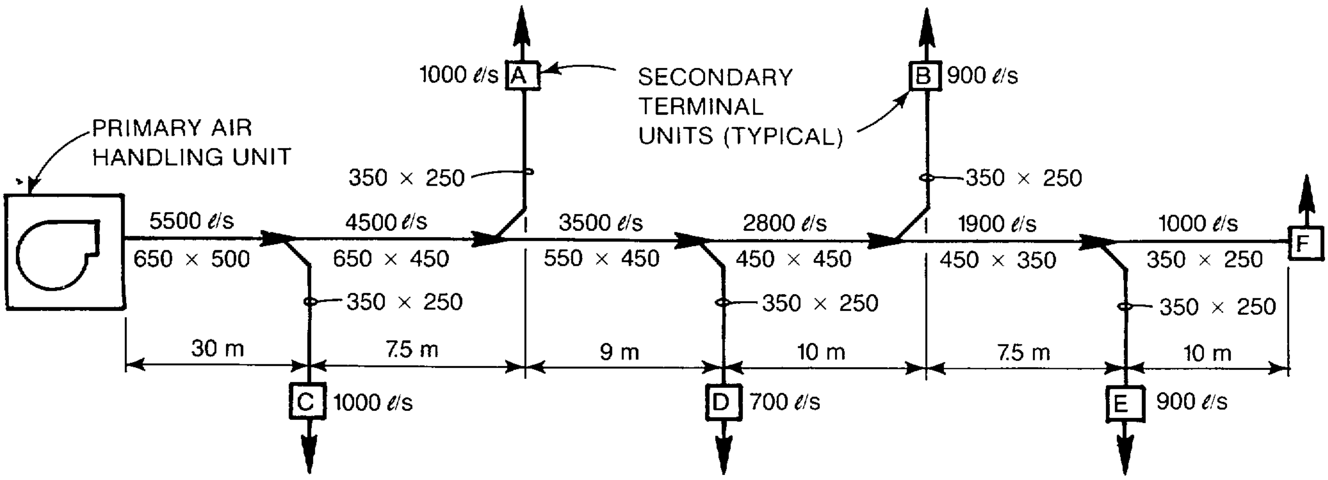

Figure 8-3 SYSTEM "A"—SIZED BY EQUAL FRICTION METHOD

method at a pressure loss of approximately 4 Pa/m. Note that reducing fittings have been used at each branch takeoff.

In Figure 8-4, 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 suzes 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 Chapter 14, the results are tabulated in Table 8-4. Ignoring branch duct and outlet losses, which are identical for both systems, the semi-extended plenum system has a 157 pascals (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, is also shown in Table 8-4. It can be seen that the semi-extended plenum design results in reduced fan power and, therefore, 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.

5. Cost Comparison

Although energy conservation holds the "spotlight," installation costs are still of primary concern to the designer, the contractor and the owner. Table 8-5 illustrates the estimated installation cost comparison between the two systems analyzed. It can be seen that 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.

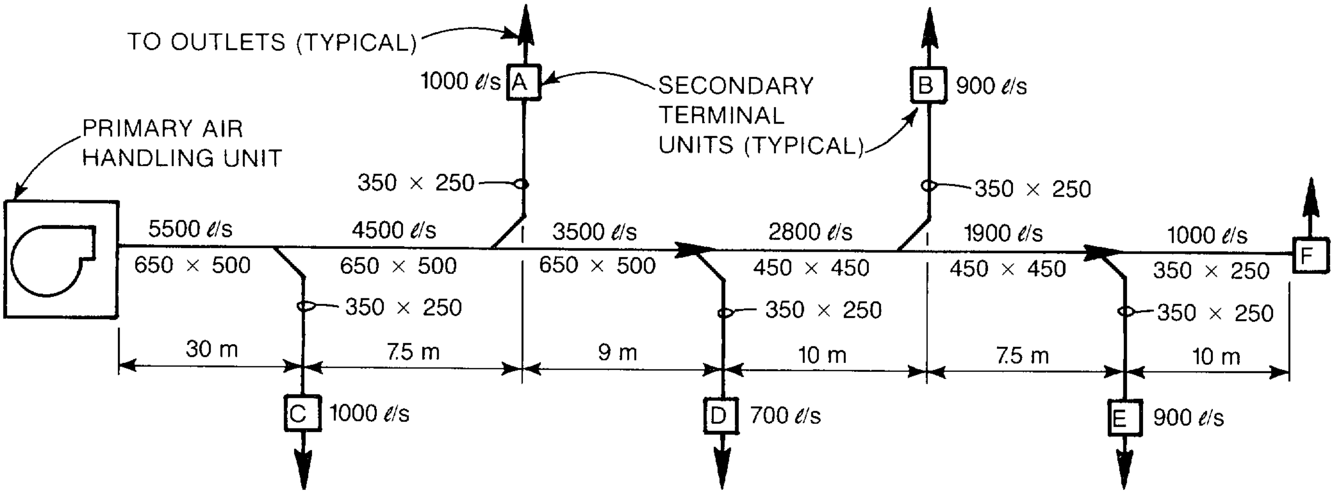

Figure 8-4 SYSTEM "B"—MODIFIED BY SEMI-EXTENDED PLENUM CONCEPT

Table 8-4 SEMI-EXTENDED PLENUM COMPARISON

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	324	3.4

Table 8-5 SEMI-EXTENDED PLENUM INSTALLATION COST 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	

DUCT SIZING WORK SHEET
(Metric Units)

DATE ______________ PAGE ______

PROJECT ______________ LOCATION ______________ SYSTEM ______________

	A DUCT RUN	B SEC-TION	C ITEM		D FLOW (L S)	E FRICTION PER METRE	F VELOCITY (M S)	G Vp (PA)	H LOSS COEFF.	I EQUIV DIAM.	J RECTANGULAR SIZE	K CORR. FACT.	L LOSS PER ITEM	M LOSS PER SECTION	N CUMULATIVE LOSS	O TOTAL LOSS
1																
2																
3																
4																
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NOTES: *Indicates duct lining used. Sizes are interior dimensions.

Figure 8-5 DUCT SIZING WORK SHEET (METRIC)

CHAPTER 9

PRESSURE LOSS OF SYSTEMS COMPONENTS

A PROCEDURE

1. Preliminary Pressure Loss Data

The pressure loss data provided in this section represents reasonable pressure loss allowances for each component to be installed in the duct system. Where a pressure loss range is provided, the designer may select the high or low figure shown or some representative average figure. A range has been provided, rather than a specific value, where 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. Where free area velocity is indicated, it must be used for calculations. *Please note the type of area used on each individual chart.* Representative free area (effective square feet) can be selected from the free area tables for louvers, which should be adequate for preliminary duct system design. All system component pressure losses should be entered on the SMACNA Duct Sizing Form during preliminary design.

2. Final Design Data

After completion of the preliminary duct system design, actual equipment selection can be made from manufacturers data; such as selection of heating coil capacity, size, flow rate of heating media, static pressure loss, etc. *PRELIMINARY PRESSURE LOSS VALUES MUST BE REPLACED WITH THE ACTUAL MANUFACTURERS' PRESSURE VALUES FOR THE SELECTED COMPONENTS.* A system static pressure check list (Figure 9-1) has been provided to aid the designer in his final review of the system design to assure that all system pressure losses have been considered.

3. Submittal Review

The designer must review all equipment submittal drawings and data to be assured that 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 and/or efficiencies of the fans, then different equipment must be selected, or the necessary 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 equipment/system first cost and the overall operating costs (life-cycle costs).

B USE OF TABLES & CHARTS

1. Filters

Table 9-1—"Filter Pressure Loss Data" contains the static pressure loss ranges of the most commonly used HVAC system filters. A "recommended design" column has been provided, although the designer should make the selection based on area conditions and his own experience.

2. 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 ¼" Metal frames have a loss coefficient of C = 0.52 (see Table 14-18, Figures E and F). These values can be compared with manufactured 36 in. × 36 in. (900mm × 900mm) volume dampers with frames (Figure 9-2) by selecting some duct velocities:

Figure 9-1 SYSTEM PRESSURE LOSS CHECK LIST

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 LOSS 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
- ☐ 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
- ☐ 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. w.g. (12 to 25 Pa) static pressure allowance for building pressurization should be adequate. The designer should determine the proper building pressurization value based upon individual system requirements. Consideration should include elevator shaft ventilation requirements, tightness of building construction, stack effect, etc.

Table 9-1 FILTER PRESSURE LOSS DATA

	Static Pressure Loss		
	Recommended		
	Clean	Design	Dirty
1. Viscous Impingement, Flat Panel Filters			
A. Replaceable Media 8-12% Efficiency	.18	.35	.50
B. High Velocity Cleanable 8-12% Efficiency	.13	.30	.50
C. Disposable 8-12% Efficiency	.08	.20	.30
2. Electrostatic Filters 80-90% Efficiency	.24	.24	.24
3. Moving Curtain, Viscous Impingement Filters			
A. Renewable Media 10-15% Efficiency	.20	.45	.55
B. Self Cleaning Panels 10-15% Efficiency	.75	.75	.75
4. Dry Media Filters			
A. Cartridge 30-35% Efficiency 9" Deep	.30	.45	.60
B. Cartridge 30-35% Efficiency 15" Deep	.55	.65	.75
C. Cartridge 35-40% Efficiency 9" Deep	.30	.65	1.0
D. Cartridge 35-40% Efficiency 15" Deep	.40	.70	1.0
E. Cartridge 40% Efficiency 19" Deep	.40	.60	.80
F. Cartridge 30% Efficiency 9" Deep	.15	.37	.60
G. Cartridge 30% Efficiency 15" Deep	.25	.42	.60
H. Cartridge 30-36% Efficiency 2" Deep	.15	.42	.70
I. Cartridge 30-36% Efficiency 4" Deep	.135	.41	.70
J. Cartridge 55% Efficiency 21" Deep	.20	.40	.60
K. Cartridge 55% Efficiency 29" Deep	.45	.62	.80
L. Cartridge 55% Efficiency 36" Deep	.65	.82	1.00
M. Cartridge 85% Efficiency 21" Deep	.35	.52	.70
N. Cartridge 85% Efficiency 29" Deep	.50	.65	.80
O. Cartridge 85% Efficiency 36" Deep	.70	.85	1.0
P. Cartridge 95% Efficiency 21" Deep	.35	.55	.75
Q. Cartridge 95% Efficiency 29" Deep	.60	.70	.80
R. Cartridge 95% Efficiency 36" Deep	.75	.87	1.00
S. High Efficiency 60% Efficiency 6" Deep	.15	.57	1.00
T. High Efficiency 60% Efficiency 12" Deep	.35	.67	1.00
U. High Efficiency 90% Efficiency 6" Deep	.35	.67	1.00
V. High Efficiency 90% Efficiency 12" Deep	.55	.77	1.00
W. High Efficiency 95% DOP	1.00	1.50	2.00
X. High Efficiency 99.97% DOP	1.00	2.00	2.00 +
5. Carbon Filters			
A. Full Flow	.35	.35	.35
B. ByPass	.15	.15	.15

U.S. UNITS

fpm	V_p	C = 0.20	C = 0.52	Figure 9-2
1000	0.06	0.012 in.w.g.	0.031 in.w.g.	0.032 in.w.g.
1500	0.14	0.028 in.w.g.	0.073 in.w.g.	0.065 in.w.g.
2000	0.25	0.050 in.w.g.	0.130 in.w.g.	0.14 in.w.g.
3000	0.56	0.112 in.w.g.	0.292 in.w.g.	0.32 in.w.g.

METRIC UNITS

m/s	V_p	C = 0.20	C = 0.52	Figure 9-2
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	27.1 Pa	70.2 Pa	79.7 Pa

Shop fabricated dampers with frames would have loss values similar to the *volume dampers* in Figure 9-2.

Volume dampers are needed to balance even the most carefully designed system. But excessive use, particularly of dampers with high pressure losses where tight shut-off is not essential, can quickly build up the duct system pressure losses and be a source of noise.

For outside air and mixed air dampers, use a minimum velocity of 1500 fpm (7.5 m/s). Also include a pressure drop of at least 0.25 in.w.g. (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 type dampers, such as *relief dampers*, must comply with code requirements or certain job conditions. Therefore, the designer must add the pressure losses of these required items to the system totals. (Figures 9-3 and 9-4.)

3. Duct System Apparatus

Figures 9-5 to 9-11 contain pressure loss data for commonly used *HEATING AND COOLING COILS.*

Data needed for determining the free area of *LOUVERS* is furnished in Tables 9-2 to 9-4, as the "free area" velocity must be used to obtain louver pressure losses in Figure 9-13. Tables for other types of entries and exits can be found in Chapter 14.

The pressure loss data for several sizes and types of *SOUND TRAPS* should be used for preliminary calculations only, with manufacturers data being entered into the final design. (Figures 9-14 to 9-18.)

Pressure loss for a *SPRAYED COIL AIR WASHER* can be obtained by combining the loss of an appro-

Table 9-2 LOUVER FREE AREA CHART
2"–45° BLADES

Square Feet (Effective)

Width (Inches)

Height (Inches)	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	0.7
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	5.7	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	19.4	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

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

Table 9-3 LOUVER FREE AREA CHART
4″–45° BLADES

Square Feet (Effective)

Height	Width (Inches)																										
(Inches)	**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	3.0	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	3.6	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	24.1	25.2	26.3	27.4	28.4
88	0.5	1.7	2.9	4.1	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

Conversions: 1 mm = 0.03937 in.; 1 m² = 10.76 ft²
1 in. = 25.4 mm; 1 ft² = 0.0929 m²

Table 9-4 LOUVER FREE AREA CHART
6″–45° BLADES

Square Feet (Effective)

Height	Width (Inches)																										
(Inches)	**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	2.8	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	4.4	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	25.6	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	12.5	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

Conversions: 1 mm = 0.03937 in.; 1 m² = 10.76 ft²
1 in. = 25.4 mm; 1 ft² = 0.0929 m²

priate cooling coil (Figures 9-9 to 9-11) with the pressure loss of the 3 bend eliminator (Figure 9-19).

Duct mounted *HUMIDIFIERS* normally offer minimal resistance to the duct air flow. Should the humidifier manifold be installed in a narrow duct where it would serve as an obstruction, Tables 14-18 G or H (Chapter 14) could be used to calculate the pressure loss, or the duct could be expanded around the manifold (use the transition pressure losses).

ENERGY RECOVERY EQUIPMENT is divided into 5 categories: Air-to-air Plate Exchangers, Single Tube Exchangers (Heat Pipe Banks), Rotary Wheel, Run-around Coil Systems, and Multiple Tower Systems. Pressure loss data charts (Figures 9-22 to 9-25) must be used for *rough* estimates only as testing and rating methods and procedures have not been standardized within the industry. Heating and/or cooling coil data can be used for run-around coil systems. The *DRY AIR EVAPORATIVE COOLER* pressure loss data (Figure 9-26) is for the type of unit where the two air streams are totally separated with no moisture interchange.

4. Room Air Terminal Devices

A WORD OF SPECIAL CAUTION 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 Tables 9-5 to 9-7, are total pressure losses. The values shown serve to 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. w.g. or 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 MANUFACTURERS' CATALOGUE DATA PRIOR TO COMPLETING THE SYSTEM DESIGN. Air terminal device total pressure requirements will vary with the terminal velocity selected (see Chapter 3).

5. Operating Conditions

The pressure drop data found in the following subsections C (Damper Charts) and D (Duct System Apparatus Charts) is for standard air (0.075 lb/cu ft, 70°F, 29.92 in. Hg at sea level or 1.2041 kg/m³, 20°C, 101.325 kPa at sea level), and needs to be corrected where necessary to operating conditions by the following equation:

Equation 9-1

$$P_a = P_s \left(\frac{d_a}{d_s}\right)$$

Where:

P_a = Actual pressure drop, in. w.g. (Pa)
P_s = Pressure drop from tables, in. w.g. (Pa)
d_a = Actual air density, lb/cu ft (kg/m³)
d_s = Standard air density, 0.075 lb/cu ft (1.2041 kg/m³)

C DAMPER CHARTS

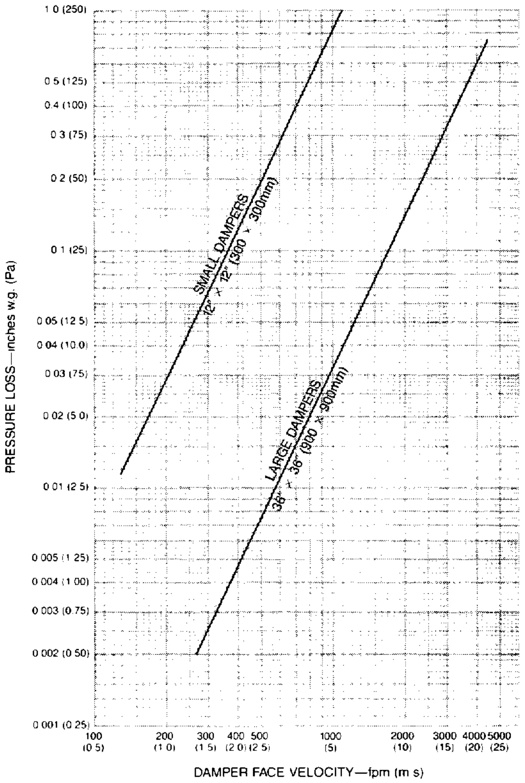
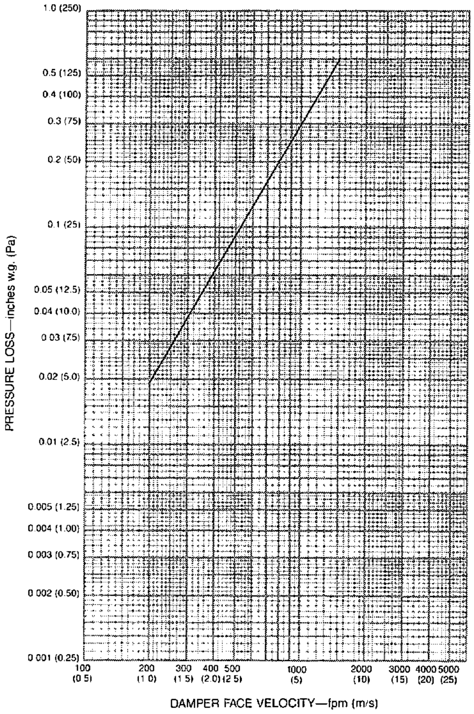
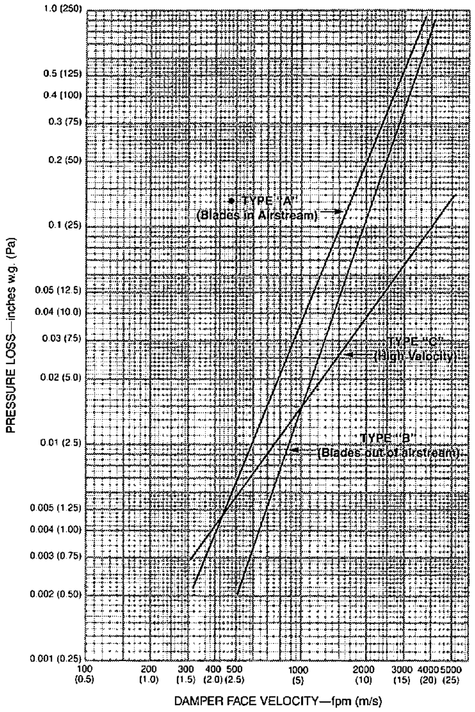


Figure 9-2 VOLUME DAMPERS (BASED UPON AMCA CERTIFIED VOLUME DAMPERS)



NOTE: 1. For dampers 8 inches (200mm) height or 1 sq. ft. (90,000mm^2) area—see manufacturers date for pressure loss.
2. Relief dampers counterbalanced to open at a specified S.P—Add opening S.P. to value found on chart.

Figure 9-3 BACKDRAFT OR RELIEF DAMPERS



NOTE: 1. For dampers under 8 inches (200 mm) height or 1 sq. ft. (90,000mm^2) area—see manufacturers data for pressure loss.
2. For 3 hr & 4 hr dampers—see manufacturers data for pressure loss.

Figure 9-4 2-HOUR FIRE & SMOKE DAMPERS (BASED ON AMCA CERTIFIED FIRE DAMPERS)

D DUCT SYSTEM APPARATUS CHARTS

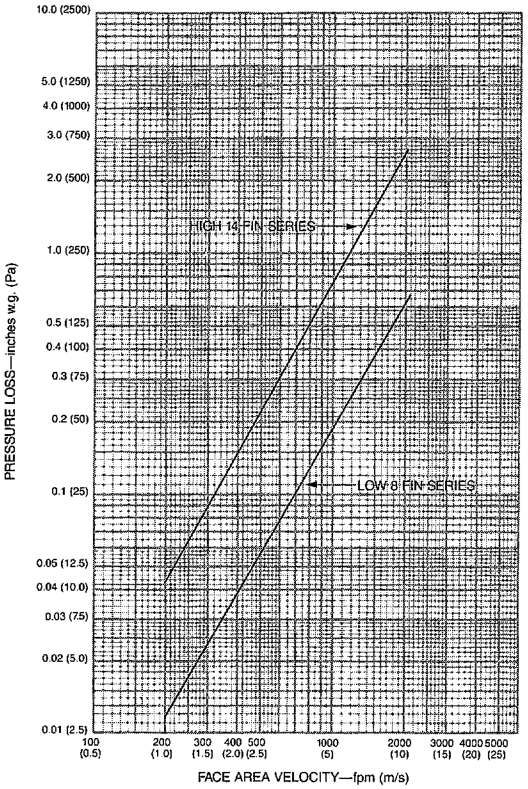


Figure 9-5 HEATING COILS—1 ROW

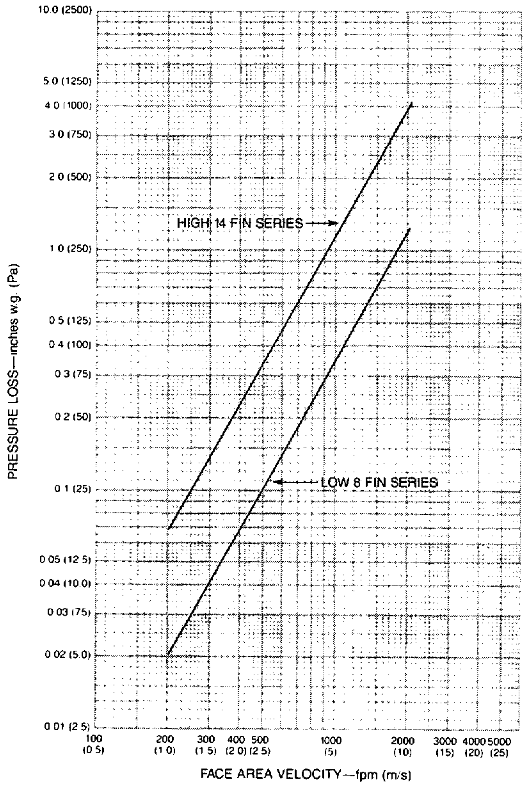


Figure 9-6 HEATING COILS—2 ROW

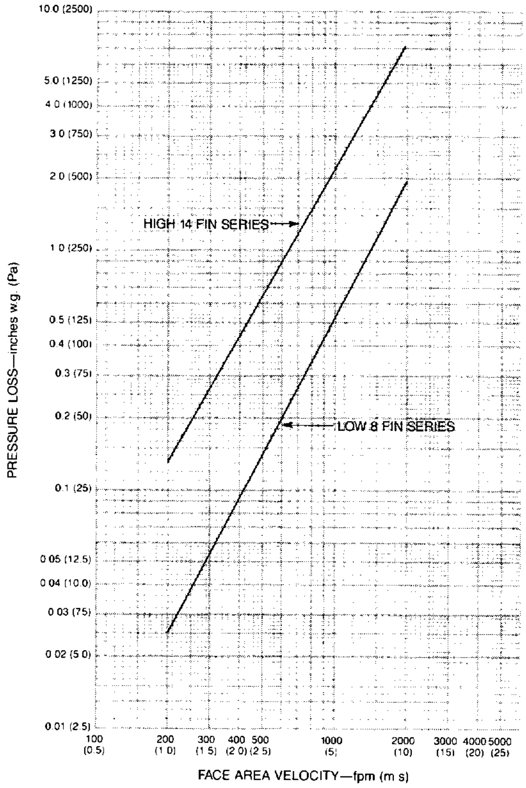


Figure 9-7 HEATING COILS—3 ROW

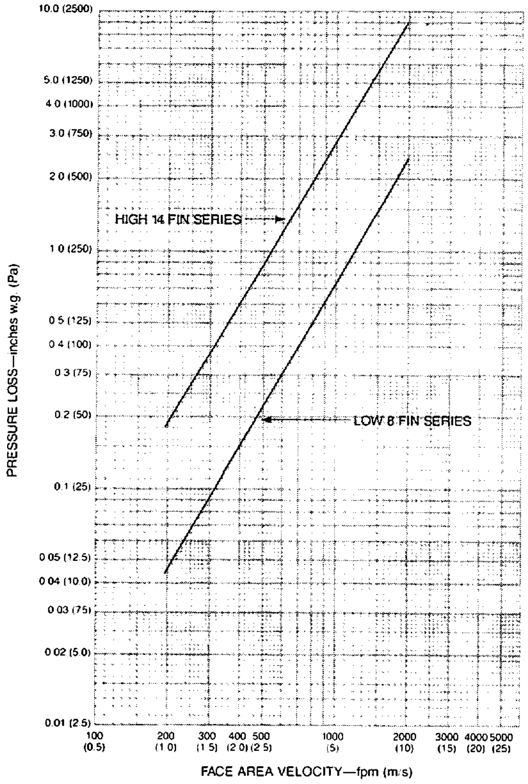


Figure 9-8 HEATING COILS—4 ROW

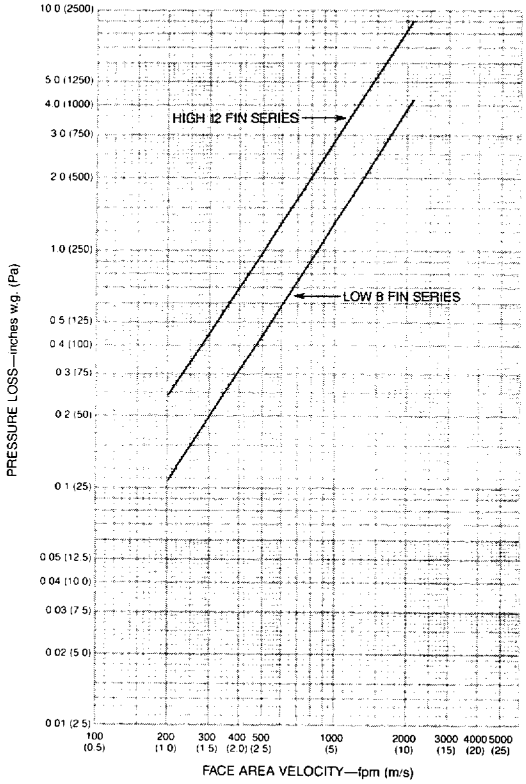


Figure 9-9 COOLING COILS (WET)—4 ROW

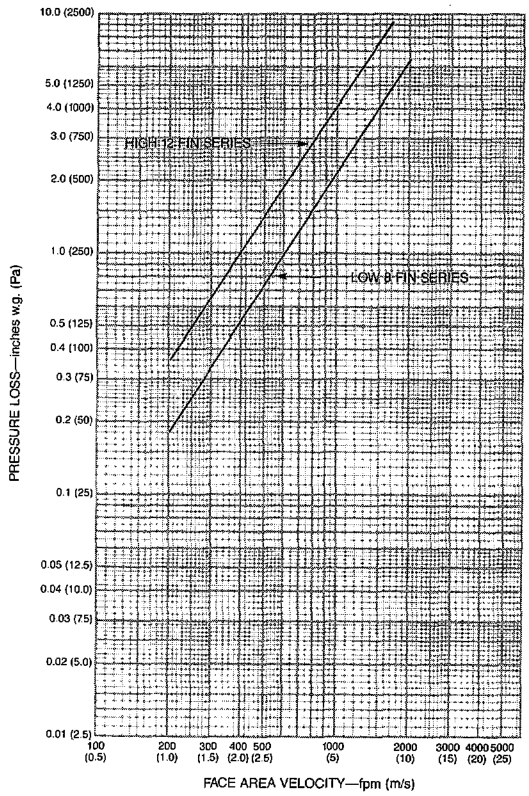


Figure 9-10 COOLING COILS (WET)—6 ROW

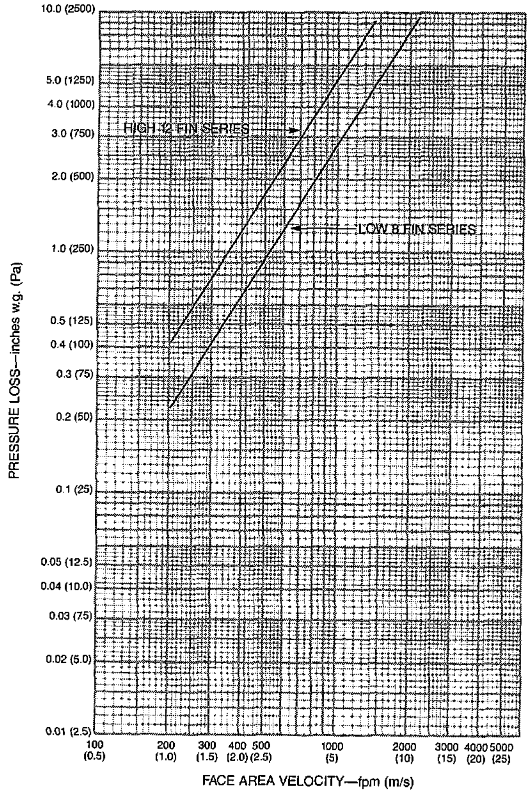


Figure 9-11 COOLING COILS (WET)—8 ROW

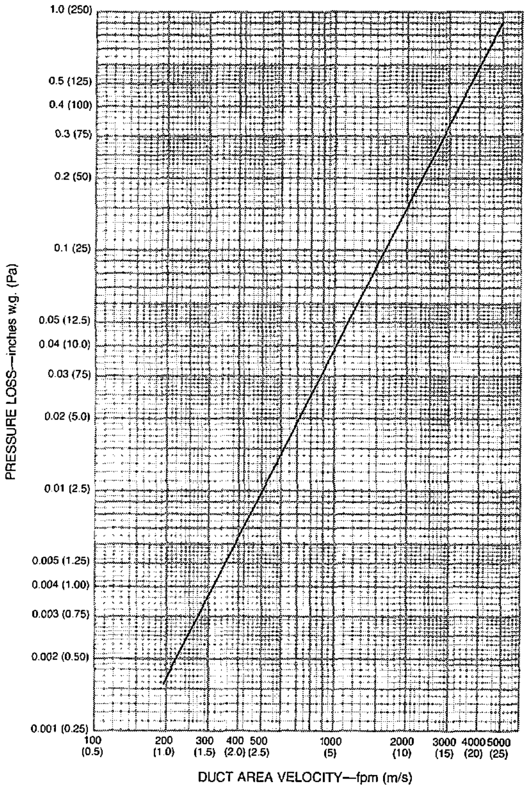
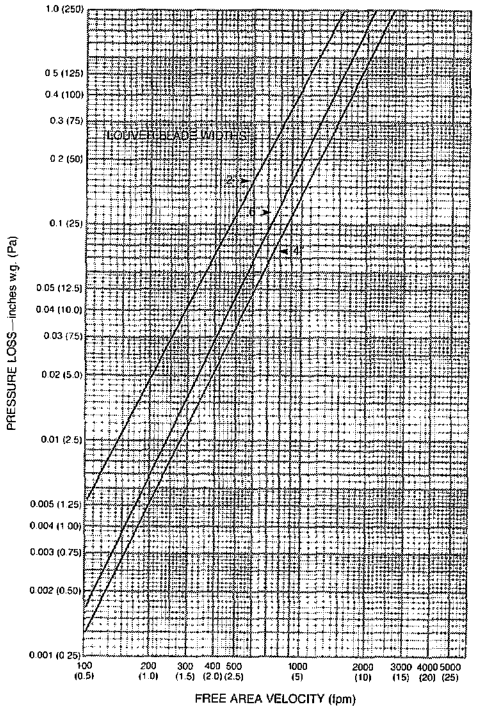


Figure 9-12 AIR MONITOR DEVICE



NOTE: Determine free area from Tables 9-2 to 9-4

$$\text{Free Area Velocity} = \frac{\text{Airflow}}{\text{Free Area}}$$

Figure 9-13 LOUVERS—45° BLADE ANGLE (BASED ON AMCA CERTIFIED LOUVERS)

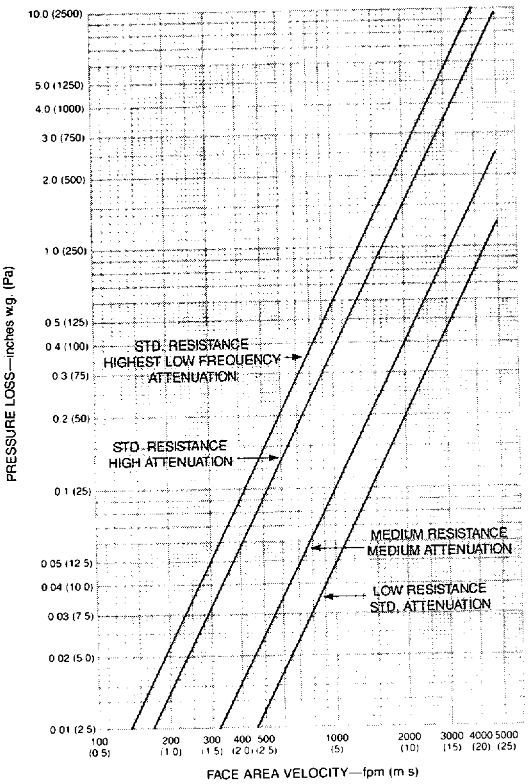


Figure 9-14 3 RECTANGULAR SOUND TRAPS— 3 FOOT (1 m)

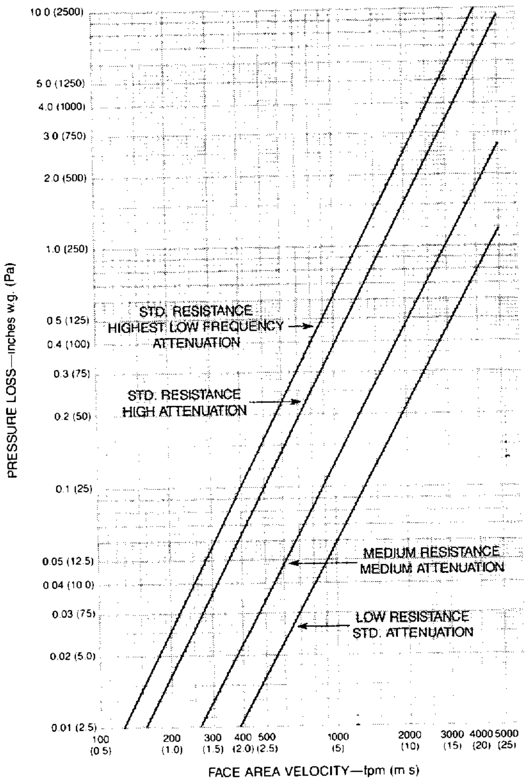


Figure 9-15 RECTANGULAR SOUND TRAPS— 5 FOOT (1.5 m)

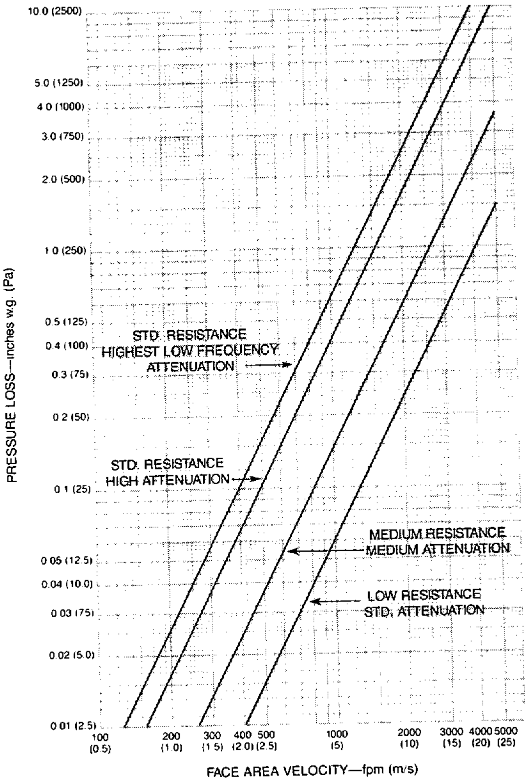


Figure 9-16 RECTANGULAR SOUND TRAPS— 7 FOOT (2 m)

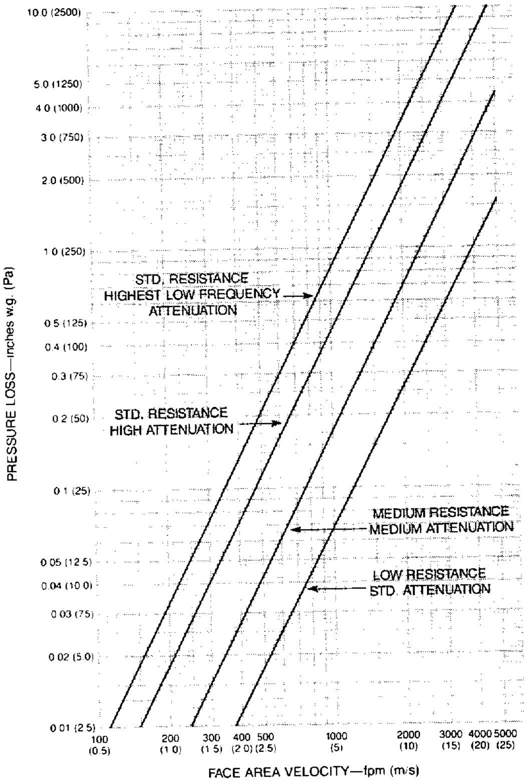


Figure 9-17 RECTANGULAR SOUND TRAPS— 10 FOOT (3 m)

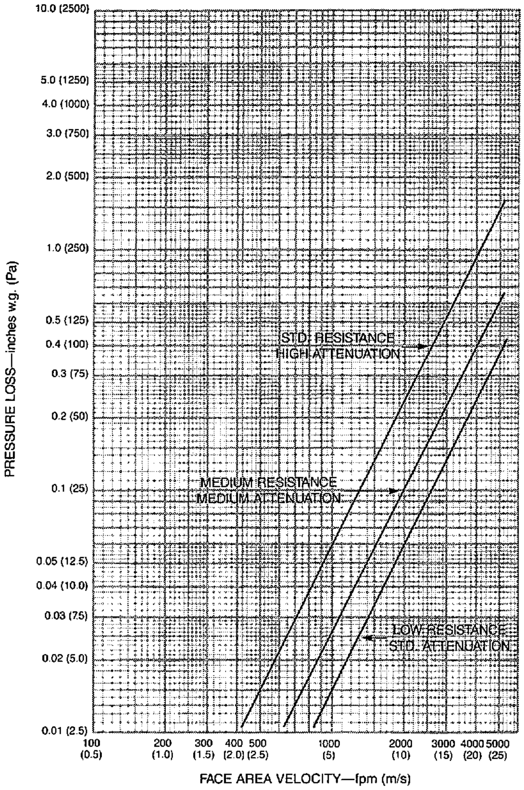


Figure 9-18 ROUND SOUND TRAPS

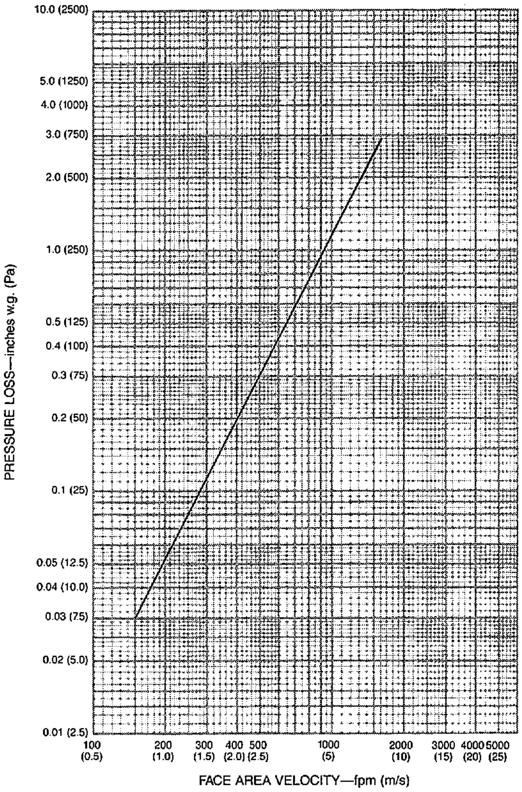


Figure 9-19 ELIMINATORS—THREE BEND

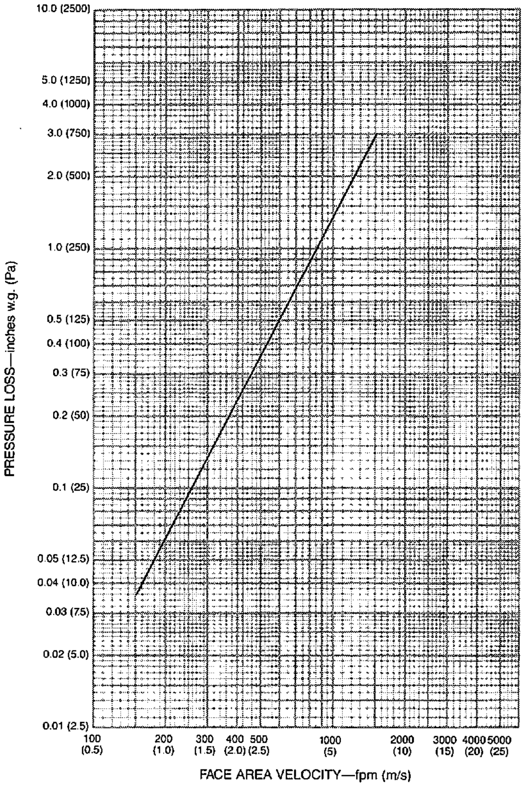
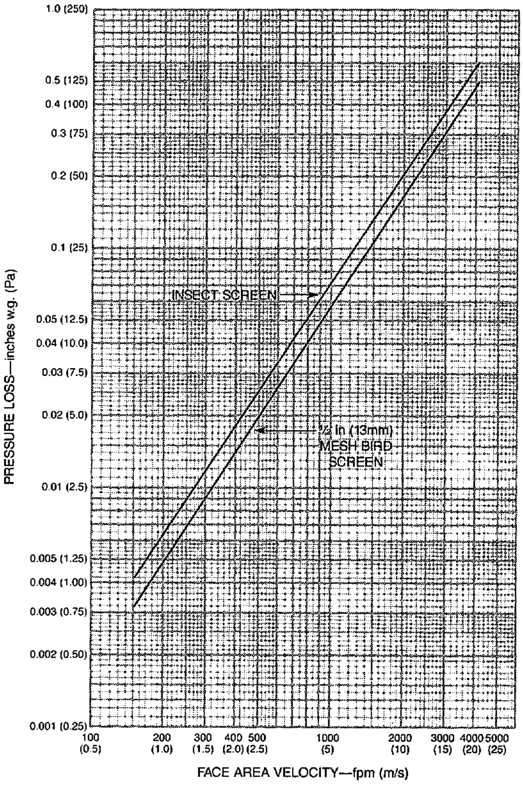


Figure 9-20 AIR WASHER



NOTE: See Table 14-17, Figure A for other types.

Figure 9-21 SCREENS

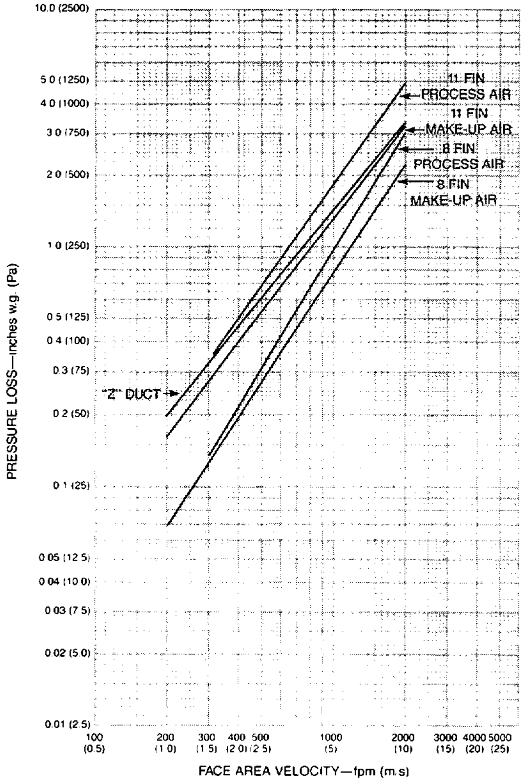


Figure 9-22 AIR-TO-AIR PLATE EXCHANGERS (Modular)

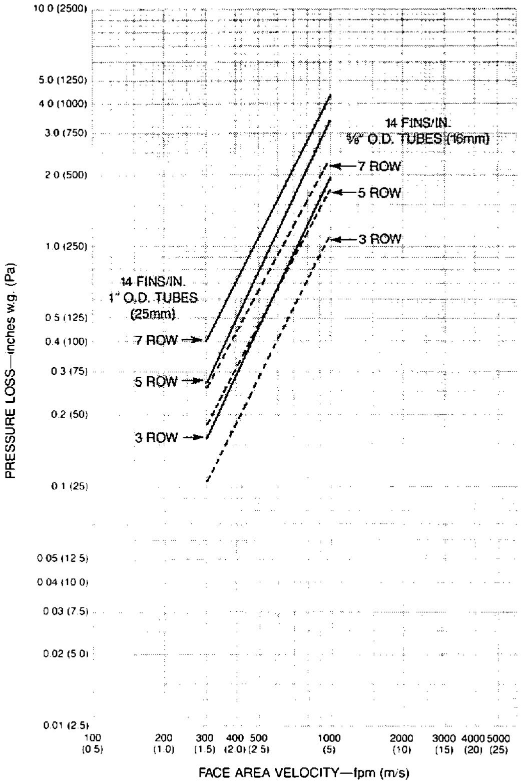


Figure 9-23 AIR-TO-AIR SINGLE TUBE EXCHANGERS

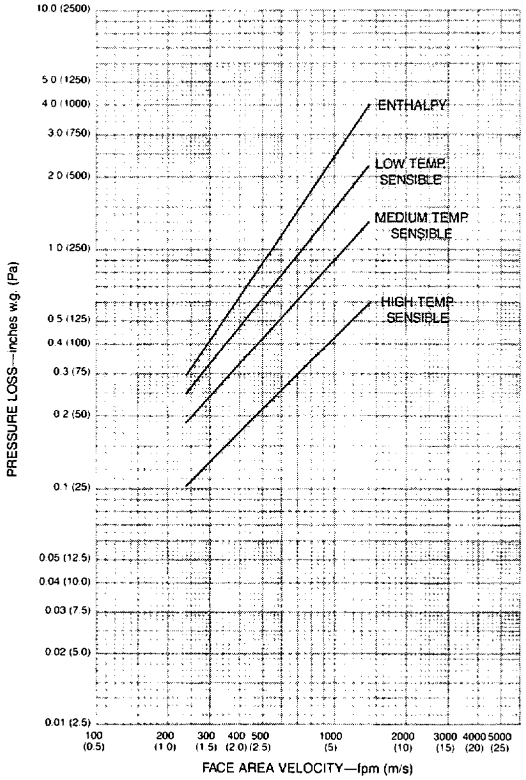


Figure 9-24 ROTARY WHEEL EXCHANGER

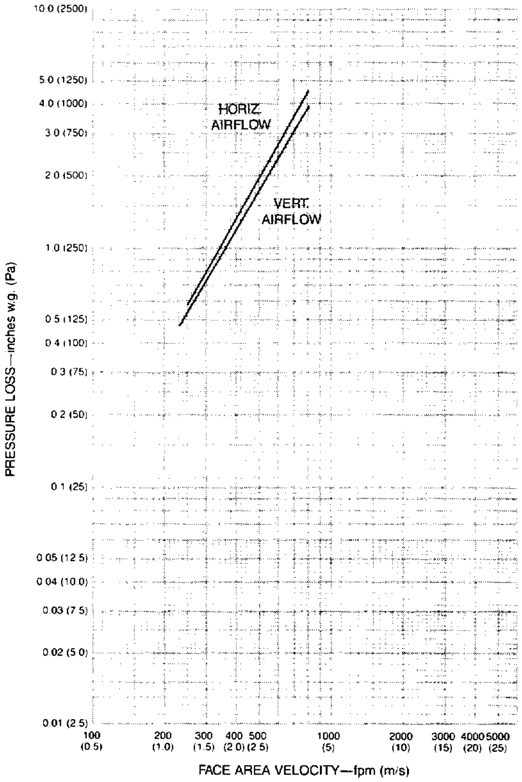


Figure 9-25 MULTIPLE TOWER ENERGY EXCHANGERS

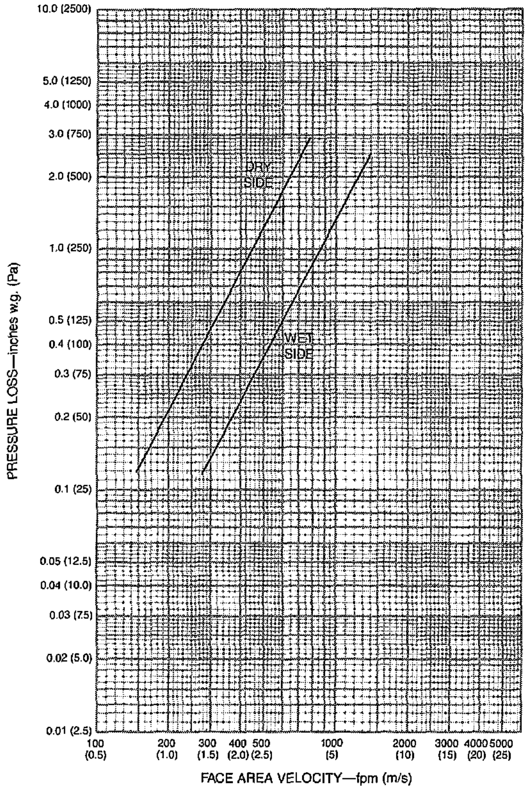


Figure 9-26 DRY AIR EVAPORATIVE COOLER

E ROOM AIR TERMINAL DEVICES

Table 9-5 AIR OUTLETS & DIFFUSERS—TOTAL PRESSURE LOSS AVERAGE

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						
¼″ (6mm) Bar Spacing*	.011	.015	.024	.030	.044	.069
Linear Slot Diffuser*	.051	.079	.110	.150	.200	—

*Velocity Thru Face Open Area

Table 9-6 SUPPLY REGISTERS—TOTAL PRESSURE LOSS AVERAGE

Velocity—fpm m/s	300 1.5	400 2.0	500 2.5	600 3.0	800 4.0	1000 5.0
0° Deflection	.010	.017	.028	.038	.069	.107
22½° Deflection	.011	.019	.031	.043	.078	.120
45° Deflection	.016	.029	.047	.064	.117	.181

Table 9-7 RETURN REGISTERS—TOTAL PRESSURE LOSS AVERAGE

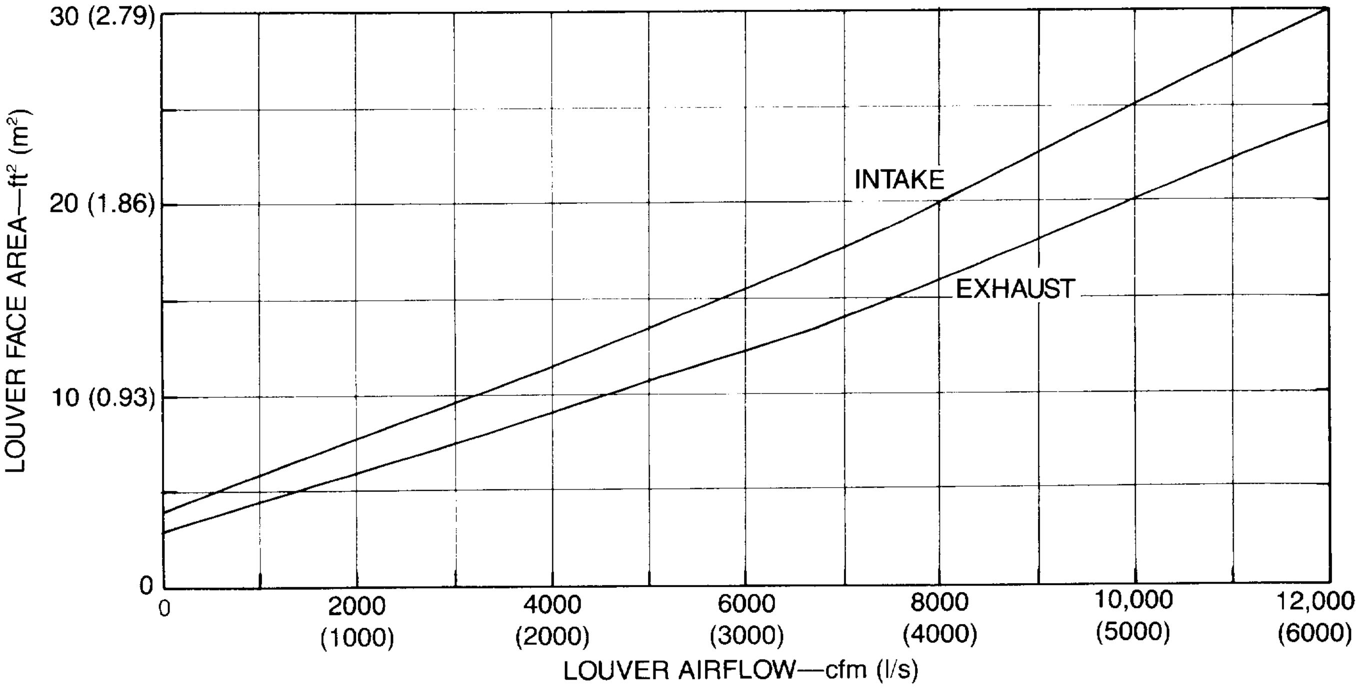
Velocity—fpm m/s	300 1.5	400 2.0	500 2.5	600 3.0	800 4.0	900 4.5
12″ Square Diffuser (300mm)	.033	.060	.092	.134	.238	.302
24″ Square Diffuser (600mm)	.068	.122	.187	.272	.483	.614
21″ × 12″ Rectangular Diffuser (300 × 525mm)	.055	.098	.152	.222	.390	.496
Perforated Return Diffuser (Neck Velocity	.025	.060	.080	.100	.180	.230
Register, 0° Deflection	.012	.020	.032	.046	.080	.102
Register, 30° Deflection	.033	.055	.088	.126	.220	.275
Register, 45° Deflection	.054	.090	.144	.207	.360	—
Register, Perforated Face	.042	.070	.112	.161	.280	.350

Table 9-8 TYPICAL DESIGN VELOCITIES

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.)
	1,500 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	Up to 750	(Up to 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)

DUCT ELEMENT	FACE VELOCITY fpm	(m/s)
FILTERS (Cont.)		
C. Electronic Air Cleaners:		
1. Ionizing Plate-Type	300-500	(1.5-2.5)
LOUVERS		
A. Intake:		
1. 7,000 cfm and greater	400	(2.0)
1. Less than 7,000 cfm	See Fig. 9.27	
B. Exhaust:		
1. 5,000 cfm and greater	500	(2.5)
2. Less than 5,00 cfm	See Fig. 9-27	
SOUND TRAPS		
A. Rectangular:		
1. High attentuation	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)

F LOUVER DESIGN DATA



Parameters Used Above	Intake Louver	Exhaust Louver
Minimum Free Area—48-Inch Square Test Section (1.5 m^2)	45%	45%
Water Penetration, Oz/ (ft^2-15 min.)	Negligible (less than 0.2)	Not Applicable
Maximum Static Pressure Drop, in. w.g. (Pa)	0.15 (38)	0.25 (63)

Figure 9-27 RECOMMENDED CRITERIA FOR LOUVER SIZING

CHAPTER 10

PROVISIONS FOR TESTING, ADJUSTING AND BALANCING

The need for accurate balancing of air and water systems is essential in today's construction industry. With continuing emphasis on energy conservation, the heating, ventilating and air conditioning engineer has been called upon to design more efficient systems.

The mystical "10 percent Safety Factor" to cover all errors in both design and installation can no longer be afforded as either a first cost or an operating cost. Therefore, systems now must be designed with minimum air quantities and fluid flow that will ensure the proper distribution of air and water to meet the design loads. Comfort can then be attained by balancing the systems to these design criteria.

While systems will vary considerably in design, size or extent of duct distribution, the same general procedure for testing, adjusting and balancing should be employed on most projects.

A TAB DESIGN CONSIDERATIONS

The system air must flow to the occupied space with the least possible losses from leakage and resistance with proper mixing of tempered air, and with a minimal temperature change from heat gain or loss. Once having arrived, the air must be distributed in the most efficient, draftless, noiseless manner available for each job requirement.

The means to accomplish these requirements are 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 and compromises are sometimes made which make the design and installation a matter of some ingenuity.

The designer should give *special consideration* to the balancing and adjusting process during the design. The TAB technician must be able to test and analyze the particular installation so that he can properly balance it with the least effort and yet obtain the greatest system efficiency and comfort level. Therefore, it is necessary that the balancing capability be designed into the system initially. The following are some considerations to use when designing duct systems.

1. Ductwork to and from air conditioning equipment should be designed very carefully so that stratified air will be mixed properly before entering branch ducts or equipment.
2. Splitter-type dampers offer little or no control of air volume in ducts. They should be regarded as air diverters only, with maximum effectiveness when present on duct systems exhibiting low resistance to air flow.
3. Manually operated, opposed blade or single blade, quadrant-type volume dampers should be installed in each branch duct takeoff after leaving the main duct to control the amount of air entering or leaving the branch (see Figure 10-1).
4. Turning vanes should be installed so that the air leaving the vanes is parallel to the downstream duct walls. Turning vanes should be utilized in all rectangular elbows (return systems as well as supply and exhaust systems—see Figures 5-14 and 5-15).
5. Manual volume dampers should be provided in branch duct takeoffs to control the total air to the face dampers of the registers or diffusers. The use of extractors is not recommended because they can cause turbulence in the main trunk duct thereby increasing the system total pressure and affecting the performance of other branch outlets downstream. Register or diffuser dampers cannot be used for reducing high air volumes without inducing objectionable air noise levels (see Figure 10-2).
6. Do not use extractors at branch or main duct takeoffs to provide volume control. Branch duct tap-in fittings with a 45° entry throat provide the most efficient airflow of all tap-in type fittings.
7. The application of 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.
8. 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.

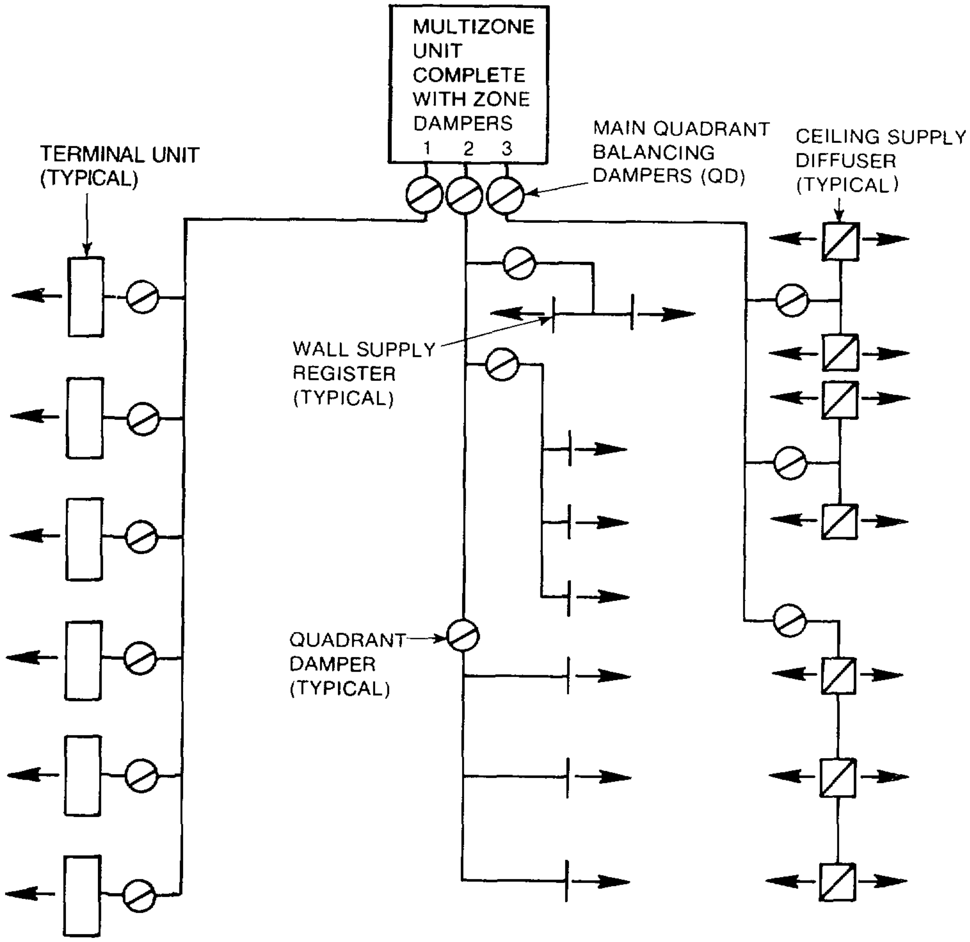
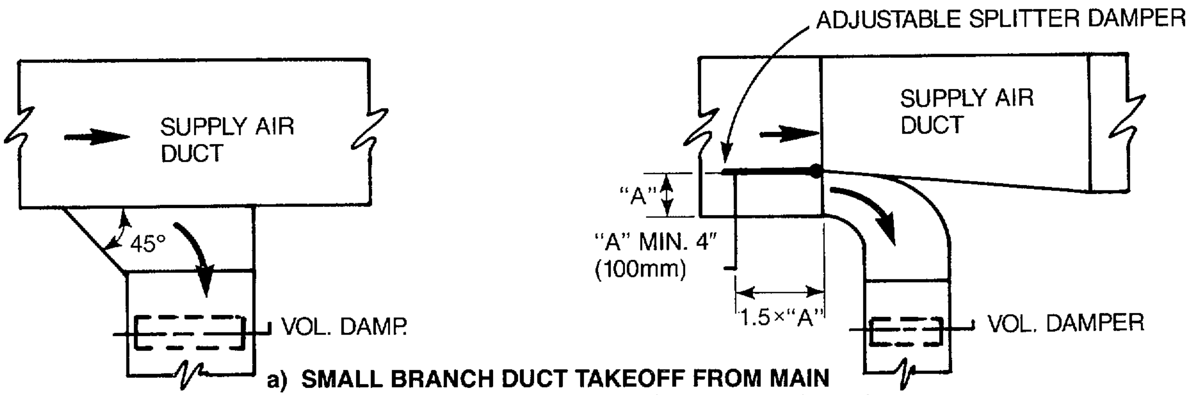
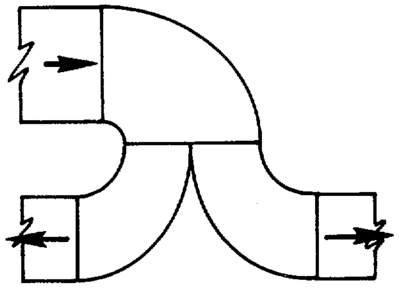


Figure 10-1 DESIGN CONSIDERATIONS FOR DIFFUSER LAYOUTS AND BALANCING DAMPER LOCATIONS

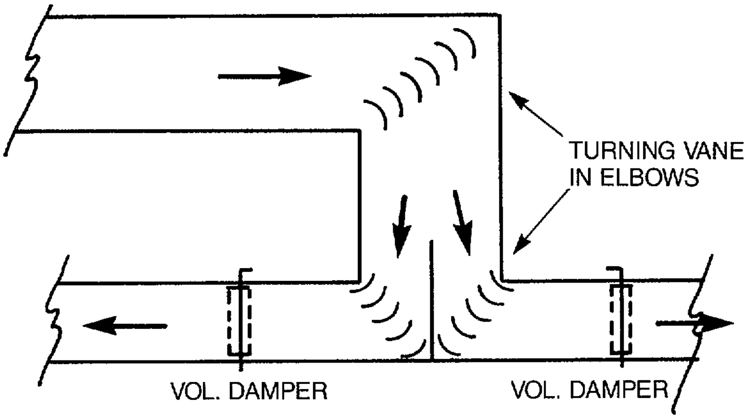
9. To minimize generated duct noise at volume dampers, indicate damper locations at least two diameters from a fitting and as far as possible from an outlet.
10. All portions of the main return air duct system require manual balancing dampers at each branch duct inlet.
11. Avoid placing a return air opening directly in or adjacent to the return air plenum. Lining of the duct behind the opening normally will not reduce the transmitted noise to acceptable levels (see Chapter 11).
12. Terminal boxes or volume control assemblies should be located so that the discharge ductwork will minimize air turbulence and stratification (see Figure 10-3).
13. Provide the necessary space around components of the duct system to allow the TAB technician to take proper readings. Allow straight duct sections of 7-1/2 duct diameters from fan outlets, elbows, or open duct ends for accurate traverse readings. (See Figure 6-2 for velocity profiles at fan discharges.)
14. Adequate size access doors should be installed within a normal working distance of all volume dampers, fire dampers, pressure reducing valves, reheat coils, volume control assemblies (boxes), blenders, constant volume regulators, etc., that require adjustments *within* the ductwork. Coordinate locations with the architect.
15. Provide for test wells, plugged openings, etc., normally used in TAB procedures.



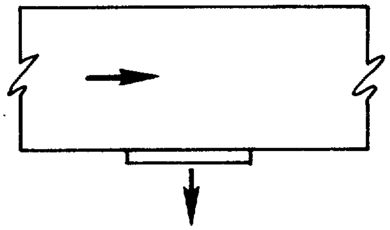
a) SMALL BRANCH DUCT TAKEOFF FROM MAIN DUCT WITH VELOCITIES UP TO 1800 FPM* (9 m/s).



b) POOR ELBOW & SPLIT DESIGN RESULTING IN UNADJUSTABLE AIR BUILDUP IN RIGHT BRANCH DUCT.



c) PREFERRED MAIN DUCT ELBOW AND DIVIDED FLOW FITTING TO BRANCH DUCTS

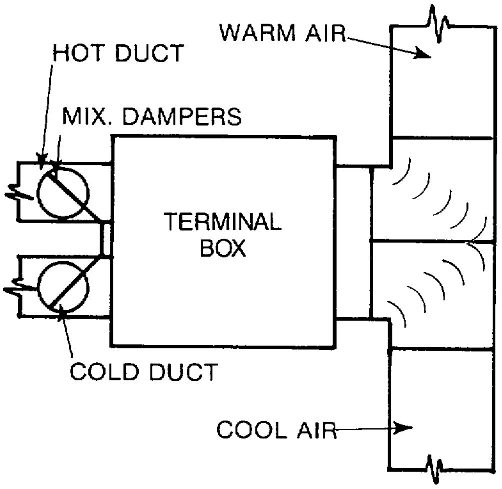


d) SHORT REGISTER BOOT CONNECTION WILL CREATE AIR NOISE AND CAUSE EXTREMELY UNEVEN AIR FLOW ACROSS REGISTER FACE.

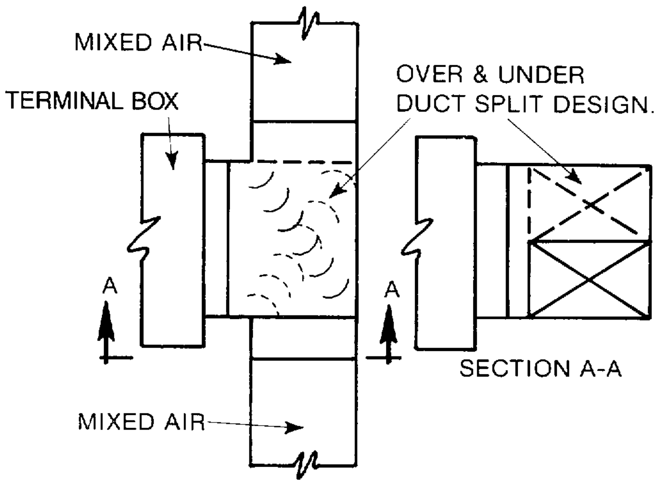
Figure 10-2 DUCT DESIGN CONSIDERATIONS FOR SUGGESTED BALANCING DAMPER LOCATIONS

B AIR MEASUREMENT DEVICES

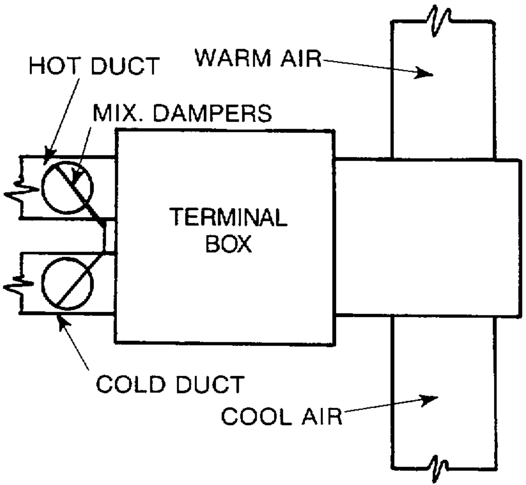
Before 1960, there was no established procedure and few attempts were made to measure airflow in HVAC systems. Total volumetric airflow measurements were attempted by making traverses with anemometers at the central station equipment. Airflow volumes at air terminals were determined by using instruments that measured jet velocities of the discharge or intake air pattern and then applying laboratory developed empirical area factors published by the terminal manufacturer.



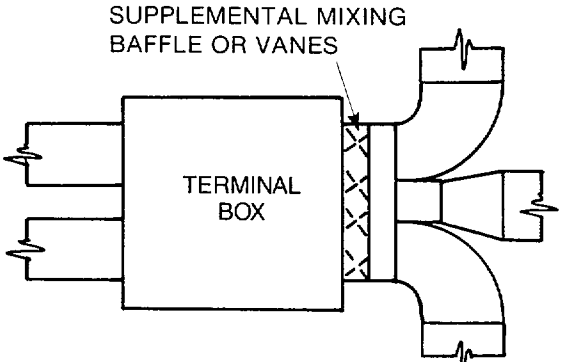
a) POOR DISCHARGE DESIGN



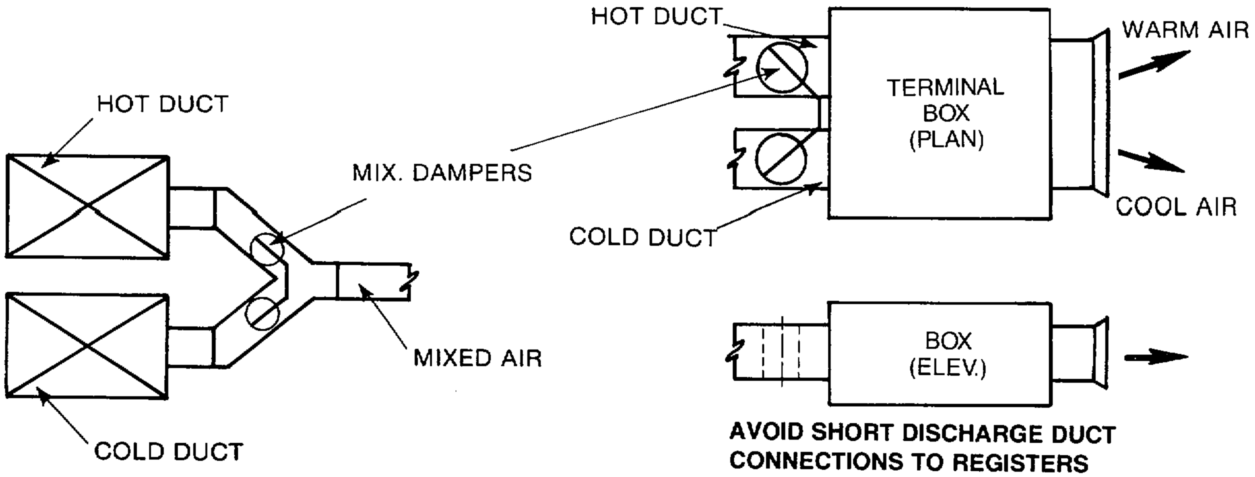
b) GOOD DISCHARGE DESIGN



c) POSSIBLE DISCHARGE AIR TEMPERATURE STRATIFICATION



d) UNIFORM DISCHARGE AIR TEMPERATURE TO ALL BRANCH DUCTS



AVOID SHORT DISCHARGE DUCT CONNECTIONS TO REGISTERS

Figure 10-3 DESIGN CONSIDERATIONS TO MINIMIZE AIRFLOW TURBULENCE & TO AVOID STRATIFICATION FROM TERMINAL BOXES

In recent years, a test procedure involving Pitot tube traverses as the primary means of determining volumetric flows through air distribution systems came into wide useage. Consequently, the systems approached designed performance. Today, we have a selection of factory fabricated volumetric airflow measuring and control devices which may be used in areas requiring critical air control. A complete list of these instruments, their accuracy and use may be found in the National Environmental Balancing Bureau (NEBB) "Procedural Standards for Testing Adjusting and Balancing of Environmental Systems" or the SMACNA "HVAC Systems—Testing, Adjusting and Balancing" manual.

C BALANCING WITH ORIFICES

The use of sharp-edged orifice plates to balance airflow to outlets or branches induces a high level of accuracy, but loses 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 14-17B 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, 1.400 in., 2.625 in. and 4,900 in. (36.6mm, 66.7mm and 124.5mm) 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 and Balancing" manual. Orifices for larger ducts can be sized using data found in Chapter 2 of the Eighth Edition of the "Fan Engineering" handbook published by the Buffalo Forge Company.

The orifice can be calibrated with a standard Pitot tube. A micromanometer 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.

D PROVISIONS FOR TAB IN SYSTEM DESIGN

1. General Procedures

In Chapter 4, a suggestion was made that a schematic diagram of each duct system be prepared in order to test and balance the systems after the installation work has been completed. It would also help the system designer to develop these schematic diagrams when designing the systems in order to determine if all necessary balancing devices have been included.

Where there is more than one system, make a separate diagram for each system. All dampers, regulating devices, terminal units, outlets and inlets should be indicated. Also, 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, and return air and relief air ducts and louvers where applicable. For rapid identification and reporting purposes, number all outlets. Add general notes indicating thermostat locations, i.e. room thermostat, thermostat integral with unit or in ductwork, etc.

2. "HVAC Systems—Testing, Adjusting and Balancing" Manual

The SMACNA "HVAC Systems—Testing, Adjusting and Balancing" manual presents the basic fundamentals, methods, and procedures, including the necessary tables and charts, that a SMACNA Contractor, with a reasonable technical background in HVAC systems, could use to adequately balance most HVAC systems that the firm installs.

In addition to the fundamentals and procedures for balancing air systems, this manual includes hydronic piping system balancing fundamentals and procedures, because many SMACNA Contractors do install the complete mechanical system. Even if only duct systems are installed and balanced, it is necessary for the SMACNA Contractor to know how to balance the hydronic portion of the system. Additional information on testing, adjusting and balancing can be found in other SMACNA and NEBB manuals listed on the publications page of this manual.

If a SMACNA Contractor wants to become more proficient and more involved in the testing, adjusting and balancing of environmental or HVAC systems, it is recommended that consideration be given to becoming a Certified TAB Contractor of the National Environmental Balancing Bureau (NEBB). NEBB has a comprehensive home study course designed to educate qualified personnel, particularly those in management positions, to direct and be responsible for the TAB operations of the firm. Information about the study course and NEBB membership can be obtained from the SMACNA or NEBB National Offices or local chapters.

CHAPTER 11

NOISE CONTROL

A INTRODUCTION

Duct systems, unless adequately designed, will act as large "speaker tubes" and will transmit noise throughout the building. The direction of the airflow has little to do with the transmission of the noise. When confined in a duct, sound transmits just as effectively upstream in a return air duct as it does downstream in a supply air duct.

Adequate noise control in a duct system is not difficult to achieve during the design of the system, providing the basic noise control principles are understood. This chapter provides the principles, terms and design data required by the designer of a duct system. Additional information relating to noise control and acoustical principles can be found in books and technical papers listed under "References" at the front of the manual. The National Environmental Balancing Bureau (NEBB) also has two publications and a home study course on Sound and Vibration.

It is recommended that those not familiar with terms used in duct noise control design study Section B—"Definitions" before proceeding further. It is suggested that those with past experience in this type of work also read Section B to become acquainted with the new terms.

Mechanical equipment noise is one of the major sources of unwanted noise in a building. The primary considerations given to the selection and use of mechanical equipment in buildings have generally been only those directly related to the intended use of the equipment. However, with the trend towards light weight building structures and variable-volume air distribution systems, the noise generated by mechanical equipment and its effects on the over all acoustical environment in a building must also be considered. Thus, the selection of mechanical equipment and the design of equipment spaces should not only be undertaken with an emphasis on the intended uses of the equipment, but also with a desire to provide acceptable noise and vibration levels in the occupied spaces of the building in which the equipment is located.

Over the past 15 years ASHRAE Technical Committee TC 2.6, Sound and Vibration Control, has sponsored research that has greatly expanded the available technical data associated with HVAC acoustics. These data, all of which have been included in this chapter on noise control, have greatly expanded the ability of designers to make more accurate calculations related to the acoustical characteristics of HVAC systems.

B DEFINITIONS

Absorption Coefficient: For a surface, the ratio of the sound energy absorbed by a surface of a medium (or material) exposed to a sound field (or to sound radiation) divided by the sound energy incident on the surface. The stated values of this are to hold for an infinite area of the surface. The conditions under which measurements of absorption coefficients are made must be stated explicitly. The absorption coefficient is a function of both angle of incidence and frequency. Tables of absorption coefficients usually list the absorption coefficients at various frequencies, the values being those obtained by averaging over all angles of incidence.

Aerodynamic Noise: also called generated noise, self-generated noise; is noise of aerodynamic origin in a moving fluid arising from flow instabilities. In duct systems, aerodynamic noise is caused by airflow through elbows, dampers, branch wyes, pressure reduction devices, silencers and other duct components.

Airborne Noise: Noise which reaches the observer by transmission through air.

Attenuation: The transmission loss or reduction in magnitude of a signal between two points in a transmission system.

Background Noise: Sound other than the signal wanted. In room acoustics, it is the irreducible noise level measured in the absence of any building occupants when all of the known sound sources have been turned off.

Breakout Noise: The transmission or radiation of noise through some part of the duct system to an occupied space in the building.

Decibel (dB): The unit "bel" is used in telecommunication engineering as a dimensionless unit for the logarithmic ratio of two power quantities. The decibel is one-tenth of a bel. Therefore:

$$L = 10 \log_{10} \left[\frac{\text{sound power}}{\text{reference power}} \right]$$

The referenced power for sound power level is 10^{-12} watts.

In noise control work, the decibel notation is used to indicate the magnitude of sound pressure and sound power.

Combining Decibels: In sound survey work, it is frequently necessary to combine sound pressure level readings. An example would be to evaluate the effect of adding a noise source in a room where the noise level is already considered borderline. Since the decibel scale is logarithmic, decibel values cannot be added directly. The correct procedure is to convert the decibels to intensity ratios, add the intensity ratios, and reconvert this sum into decibels.

Directivity Factor: The ratio of the sound pressure squared at some fixed distance and direction divided by the mean-squared sound pressure at the same distance averaged over all directions from the source.

Dynamic Insertion Loss: The dynamic insertion loss of a silencer, duct lining, or other attenuating device is the performance measured in accordance with ASTM E 477 when handling the rated airflow. It is the reduction in sound pressure level, expressed in decibels, due solely to the placement of the sound attenuating device in the duct system.

End Reflection: When a duct system opens abruptly into a large room, some of the acoustic energy at the exit of the duct is reflected upstream with the result that the amount of the acoustic energy radiated into the room is reduced. This decrease in radiated energy increases as the frequency decreases.

First Acoustically Critical Room: Most duct systems service a number of rooms. The room that has the shortest duct run from the fan is usually exposed to more fan noise than rooms further away from the fan. If this "first" room has the same noise criterion (NC) or a lower NC value than rooms further away from the fan, it may be assumed that, if the acoustical attenuation of the duct system from the fan to this "first" room satisfies the requirements for this "first" room, it also satisfies the acoustical requirements for rooms further away from the fan.

Flanking (Sound) Transmission: The transmission of sound between two rooms by any indirect path of sound transmission.

Forward Flow: Forward flow occurs when air flows and noise propagates in the same direction, as in an air conditioning supply system or in a fan discharge.

Frequency: The number of vibrations or waves or cycles of any periodic phenomenon per second. In noise control of duct systems, our interest lies in the audible frequency range of 20 to 20,000 cycles per second. The United States has adopted the international designation of "hertz" (Hz) for cycles per second.

Frequency Spectrum: A representation of a complex noise which has been resolved into frequency components. The most commonly used components are 1/1 octave bands and 1/3 octave bands.

Level: The logarithm of the ratio, expressed in decibels, of two quantities proportional to power or energy. The quantity which is the denominator of the ratio is the standard reference quantity.

Mass Law: The law relating to the transmission loss of sound barriers which says that in part of the frequency range, the magnitude of the loss is controlled entirely by the mass per unit area of the barrier. The law also says that the transmission loss increases 6 decibels for each doubling of the frequency or for each doubling of the barrier mass per unit area.

Noise Criterion (NC) Curves: Established 1/1 octave band noise spectra for rating the amount of noise of an occupied space with a single number.

Noise: Sound which is unpleasant or unwanted by the recipient.

1/1 Octave Band: A range of frequencies where the highest frequency of the band is double the lowest frequency in the band. The band is specified by the center frequency. The preferred octave bands are designated by the following center frequencies: 31.5, 63, 125, 250, 500, 1000,2000, 4000, 8000, 16,000.

Preferred Noise Criterion (PNC) Curves: The PNC curves are a proposed modification of the older NC curves. These PNC curves have values that are about 1 dB lower than the NC curves in the four octave bands at 125, 250, 500, and 1000 Hz for the same curve rating numbers. In the 63 Hz band, the permissible levels are 4 or 5 dB lower; in the highest three bands, they are 4 or 5 dB lower.

Reverse Flow: Occurs when noise propagates and air flows in opposing direction, as in a typical return-air system.

Room Absorption: The product of average absorption coefficients inside a room and the total surface area. This is usually expressed in sabins.

Room Criterion (RC) Curves: The RC curves are similar to NC or PNC curves. However, they have a slightly different shape to approximate a well balanced, bland-sounding spectrum whenever the space requirements dictate that a certain amount of background noise be maintained for masking or other purposes.

Room Effect: The difference between the sound power level discharged by a duct (through a diffuser or other termination device) and the sound pressure level heard by an occupant of a room is called the Room Effect. The Magnitude of the Room Effect depends upon the amount of the sound absorption in the room (sabins), the distance between the termination duct and the nearest observer and the directivity factor of the source.

Sabin: The unit of acoustic absorption. One sabin is one square foot of perfect sound absorbing material.

Sone: One sone is defined as the loudness of a 1000 Hz tone having a sound pressure level of 40 dB. Two sones is twice as loud as the 40 dB reference sound of one sone, etc.

Sound Power Level (L_w): the fundamental characteristic of an acoustic source (fan, etc.) is its ability to radiate power. Sound power level cannot be measured directly; it must be calculated from sound pressure level measurements. The sound power level of a source (L_w) is the ratio, expressed in decibels, of its sound power divided by the reference sound power which is 10^{-12} watts.

A considerable amount of confusion exists in the relative use of sound power level and sound pressure level. An analogy may be made in that the measurement of sound pressure level is comparable to the measurement of temperature in a room; whereas, the sound power level is comparable to the cooling capacity of the equipment conditioning the room. The resulting temperature is a function of the cooling capacity of the equipment and the heat gains and losses of the room. In exactly the same way, the resulting sound pressure level would be a function of the sound power output of the equipment together with the sound reflective and sound absorptive properties of the room.

Given the total sound power output of a sound source and knowing the acoustical properties and dimensions of a room, it is possible to calculate the resulting sound pressure levels.

Sound Pressure: Sound pressure is an alternating pressure superimposed on the barometric pressure by sound. It can be measured or expressed in several ways such as maximum sound pressure or instantaneous sound pressure. Unless such a qualifying word is used, it is the effective of root-mean-square pressure which is meant.

Sound Pressure Level (L_p): A measure of the air pressure change caused by a sound wave expressed on a decibel scale reference to a reference sound pressure of 2×10^{-5} Pa or 0.0002 microbar.

Sound Transmission Class: Sound transmission class is the preferred single figure rating designed to give a preliminary estimate of the sound insulating properties of a barrier.

Structure-Borne Noise: A condition when the sound waves are being carried by a portion of the building structure. Sound waves in this state are inaudible to the human ear since they cannot carry energy to it. Airborne sound can be created from the radiation of the structure-borne sound into the air.

C BASICS OF SOUND

1. Sound Levels

The most common parameter which is used to give an indication of "loudness" is the sound pressure level, L_p. Sound pressure level, L_p (dB), is defined as:

Equation 11-1

$$L_p = 10 \log_{10} \left[\frac{P^2_{rms}}{P^2_{ref}} \right]$$

where p_{rms} is the root-mean-square value (rms) of acoustic pressure (Pa). p_{ref} is the reference sound pressure and has a value of 2×10^{-5} Pa or 0.0002 microbar. This amplitude was selected because it is the amplitude of the sound pressure that roughly corresponds to the threshold of hearing at a frequency of 1000 Hz.

Intensity level, L_I (dB), is defined as:

Equation 11-2

$$L_I = 10 \log_{10} \left[\frac{I}{I_{ref}} \right]$$

where I is acoustic intensity (watts/m^2). I_{ref} is the reference intensity and has a value of 10^{-12} watts/m^2.

Sound power level, L_W (dB), is defined as:

Equation 11-3

$$L_W = 10 \log_{10} \left[\frac{W}{W_{ref}} \right]$$

where W is sound power (watts). W_{ref} is the reference sound power and has a value of 10^{-12} watts.

Noise reduction, NR (db), with respect to HVAC systems, is:

Equation 11-4

$$NR = L_{p(1)} - L_{p(2)}$$

where $L_{p(1)}$ is the sound pressure level (dB) of the sound entering a duct element and $L_{p(2)}$ is the sound pressure level (dB) of the sound coming out of the element.

Insertion loss, IL (dB), is:

Equation 11-5

$$IL = L_{p(w/o)} - L_{p(w)}$$

where $L_{p(w/o)}$ is the sound pressure level (dB) at a point without a specific duct element inserted and $L_{p(w)}$ is the sound pressure level (dB) at the same point with the duct element inserted.

Transmission loss, TL (dB), is:

Equation 11-6

$$TL = -10 \log_{10} \left[\frac{W_{out}}{W_{in}} \right]$$

where W_{in} is the sound power (watts) of sound entering a duct element and W_{out} is the sound power (watts) of the sound exiting the duct element.

Often it is necessary to add the sound pressure levels at a point in a room from several sound sources or to add the sound power levels at a specific point in a duct system associated with different duct elements. When adding sound power or sound pressure levels, the total level, L_T (dB), is:

Equation 11-7

$$L_T = 10 \log_{10} [SUM]$$

where when adding sound pressure levels:

Equation 11-8

$$SUM = 10^{[L_{p1}/10]} + 10^{[L_{p2}/10]} + 10^{[L_{p3}/10]} + \ldots$$

and when adding sound power levels:

Equation 11-9

$$SUM = 10^{[L_{W1}/10]} + 10^{[L_{W2}/10]} + 10^{[L_{W3}/10]} + \ldots$$

Figure 11-1 is a nomogram that can be used to add two sound pressure levels or two sound power levels.

When examining the sound propagation in a HVAC system, it is necessary to subtract noise reduction, insertion loss, or transmission loss values from given sound power levels at different points in the system. When this is done,

Equation 11-10

$$L_{W2} = L_{W1} - NR \quad \text{or} \quad L_{W2} = L_{W1} - IL \quad \text{or}$$
$$L_{W2} = L_{W1} - TL$$

where L_{W1} is the sound power level before a duct element and L_{W2} is the sound power level after the element.

Example 11-1

The following sound power levels exist at a point in a duct system and the IL values are associated with a duct element that exists at the point.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
L_{W1}	84	87	90	89	91	88	87	84
L_{W2}	74	95	93	98	99	92	88	87
L_{W3}	70	78	86	86	73	67	59	57
IL	6	11	17	22	28	32	24	20

Determine the sound power levels that exist after the duct element.

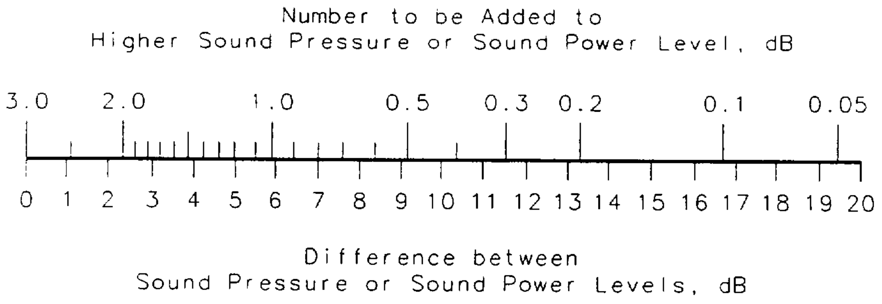


Figure 11-1 NOMOGRAM FOR ADDING TWO SOUND PRESSURE OR SOUND POWER LEVELS

Solution

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
$(W_1/W_{ref}) \times 10^{-8}$	2.51	5.01	10.00	7.94	12.59	6.31	5.012	2.512
$(W_2/W_{ref}) \times 10^{-8}$	0.25	31.62	19.95	63.10	79.43	15.85	6.310	5.012
$(W_3/W_{ref}) \times 10^{-8}$	0.10	0.63	3.98	3.98	0.20	0.50	0.008	0.005
$SUM \times 10^{-8}$	2.86	37.26	33.93	75.02	92.22	22.21	11.33	7.529
L_{WT}, dB	84.6	95.7	95.3	98.8	99.6	93.5	90.5	88.8
IL, dB	−6.0	−11.0	−17.0	−22.0	−28.0	−32.0	−24.0	−20.0
L_W, dB	78.6	84.7	78.3	76.8	71.6	61.5	66.5	68.8

Example 11-2

The following sound pressure levels are given at a specified point within a room.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
L_{p1}	74	78	77	77	75	77	75	66
L_{p2}	73	83	86	80	76	78	79	71
L_{p3}	84	89	88	86	89	88	85	78

Determine the total sound pressure level.

Solution

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
$(p_{rms1}/p_{ref}) \times 10^{-7}$	2.51	6.31	5.01	5.01	3.16	5.01	3.16	0.40
$(p_{rms2}/p_{ref}) \times 10^{-7}$	2.00	19.95	39.81	10.00	4.00	6.31	7.94	1.26
$(p_{rms3}/p_{ref}) \times 10^{-7}$	25.12	79.43	63.10	39.81	79.43	63.10	31.62	6.31
$SUM \times 10^{-7}$	29.63	105.7	107.9	54.82	86.59	74.42	42.72	7.97
L_{pl}, dB	84.7	90.2	90.3	87.4	89.4	88.7	86.3	79.0

2. Noise Criterion Curves

Noise criterion curves are shown in Figure 11-2. These curves apply to steady noise and specify the maximum noise levels permitted in each 1/1 octave band for a specified NC curve. For example, if the noise requirements for an activity area call for a NC 20 rating, the sound pressure levels in all eight 1/1 octave frequency bands must be less than or equal to the corresponding values for the NC 20 curve. Conversely the NC rating of a given noise equals the highest penetration of any of the 1/1 octave band sound pressure levels into the curves. If the farthest penetration falls between two curves, the NC rating is the interpolated value between the two curves.

Table 14-35 in Chapter 14 shows the recommended NC levels for several activity areas. The lower NC levels in the table should be used in buildings where high quality acoustical environments are desired. The upper levels should be used only for situations where economics or other conditions make use of the lower values impractical. Table 14-36 shows telephone use and listening conditions as a function of NC levels.

Example 11-3

The following 1/1 octave band sound pressure levels were measured in a laboratory work area. What is the NC rating of the noise in the work area?

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
L_p, dB	50	55	58	58	55	50	45	39

Solution

Figure 11-3 shows a plot of the above data relative to the NC curves. Since the 1/1 octave band sound pressure level in the 500 Hz 1/1 octave band penetrates to the NC 55 curve, the NC rating of the work area is NC 55.

3. Room Criterion Curves

HVAC noise is often the primary type of background noise that exists in many indoor areas. Experience has indicated that when HVAC background noise is present the use of NC levels has often resulted in a poor correlation between the calculated NC levels and an individual's subjective response to the corresponding background noise. As a means of overcoming this, the room criterion (RC) curves shown in Figure 11-4 can be used. There are four factors that should be considered when assessing HVAC system background noise: (1) level, (2) spectrum shape or balance, (3) tonal content, and (4) temporal fluctuations.

There are two parts to determining the RC noise rating associated with HVAC background noise. The first is the calculation of a number which corresponds to the speech communication or masking properties of the noise. The second is designating the quality or character of the background noise. The procedure for determining the RC rating is:

1. Calculate the arithmetic average of the 1/1 octave band sound pressure levels in the 500 Hz, 1,000 Hz and 2,000 Hz 1/1 octave frequency

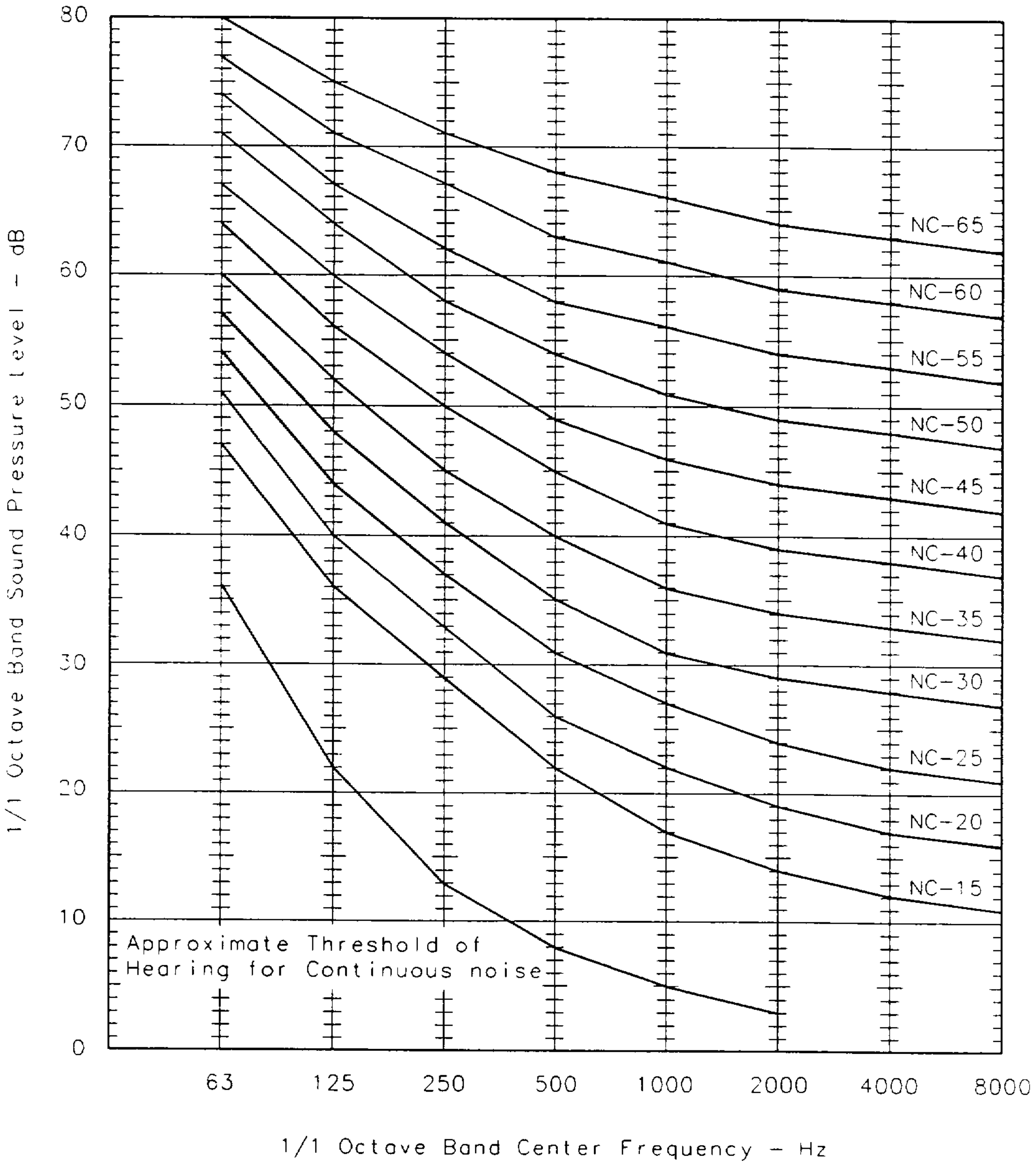


Figure 11-2 NOISE CRITERION CURVES

bands. Round off to the nearest integer. This is the RC level associated with the background noise.

2. Draw a line which has a −5 dB/octave slope which passes through the calculated RC level at 1,000 Hz. For example, if the RC level is RC 32, the line will pass through a value of 32 dB at the 1,000 Hz 1/1 octave band. This value may not be equal to the value of the 1/1 octave band sound pressure level of the background noise in the 1,000 Hz 1/1 octave band.
3. Determine the subjective quality or character of the background noise.

The subjective rating of background noise associated with the RC level can be classified as follows:

1. **Neutral**: Noise that is classified as neutral has no particular identity with frequency, It is usually

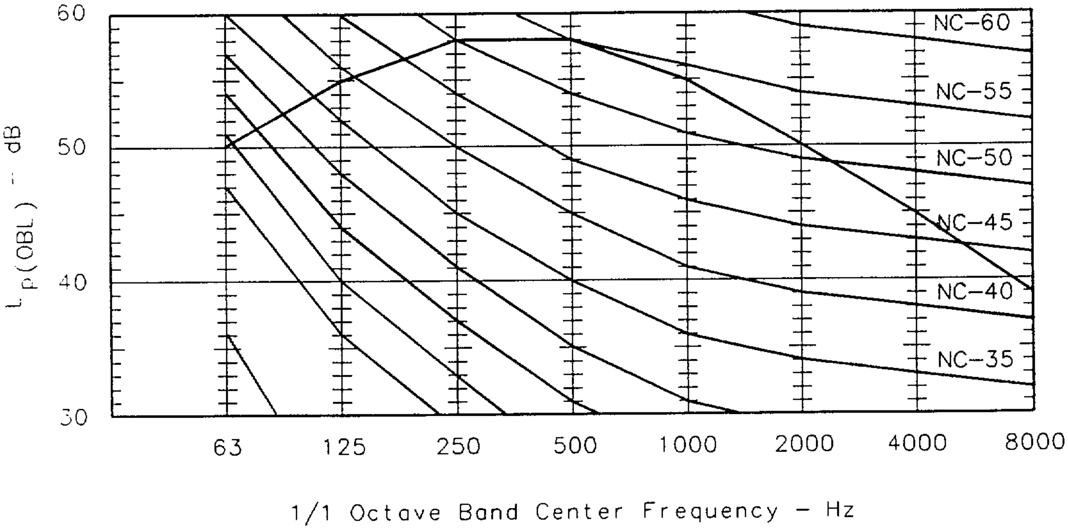


Figure 11-3 NC LEVEL FOR EXAMPLE 11-3

bland and unobtrusive. Background noise which is neutral usually has a 1/1 octave band spectrum shape similar to the RC curves in Figure 11-4. If the 1/1 octave band data do not exceed the RC curve by 5 dB the background noise is neutral and a "(N)" can be placed after the RC level.

2. **Rumble**: Noise that has a rumble has an excess of low-frequency sound energy. If any of the 1/1 octave band sound pressure levels below the 500 Hz 1/1 octave band are more than 5 dB above the RC curve associated with the background noise in the room, the noise will be judged to have a "rumbly" quality or character. If the background noise has a rumbly quality, place a "(R)" after the RC level.
3. **Hiss**: Noise that has an excess of high-frequency sound energy will have a "hissy" quality. If any of the 1/1 octave band sound pressure levels above the 500 Hz 1/1 octave band are more than 3 dB above the RC curve, the noise will be judged to have a hissy quality. If the background noise has a hiss quality, place an "(H)" after the RC level.
4. **Tonal**: Noise that has a tonal character usually contains a humming, buzzing, whining, or whistling sound. When a background sound has a tonal quality, it will generally have one 1/1 octave band in which the sound pressure level is noticeably higher than the other 1/1 octave bands. If the background noise has a tonal character, place a "(T)" after the RC level.

Background noise which has a 1/1 octave band spectrum that falls within the limiting boundaries identified with rumble and hiss and which has no tonal components is classified as neutral.

It is desirable to have background noise that has a 1/1 octave band spectrum that has a neutral character or quality. If the noise spectrum is such that it has a rumble, hiss or tonal character, it will generally be judged to be objectionable. If the background noise has a neutral quality, the NC levels specified in Tables 14-35 and 14-36 can be used to indicate the desired RC levels in different indoor activity areas.

Example 11-4

The 1/1 octave band sound pressure levels of background noise in an office area are given below:

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
L_p, dB	70	62	54	46	40	33	27	20

Determine the RC level and the corresponding character of the noise.

Solution

The RC level is determined by obtaining the arithmetic average of the 1/1 octave band sound pressure

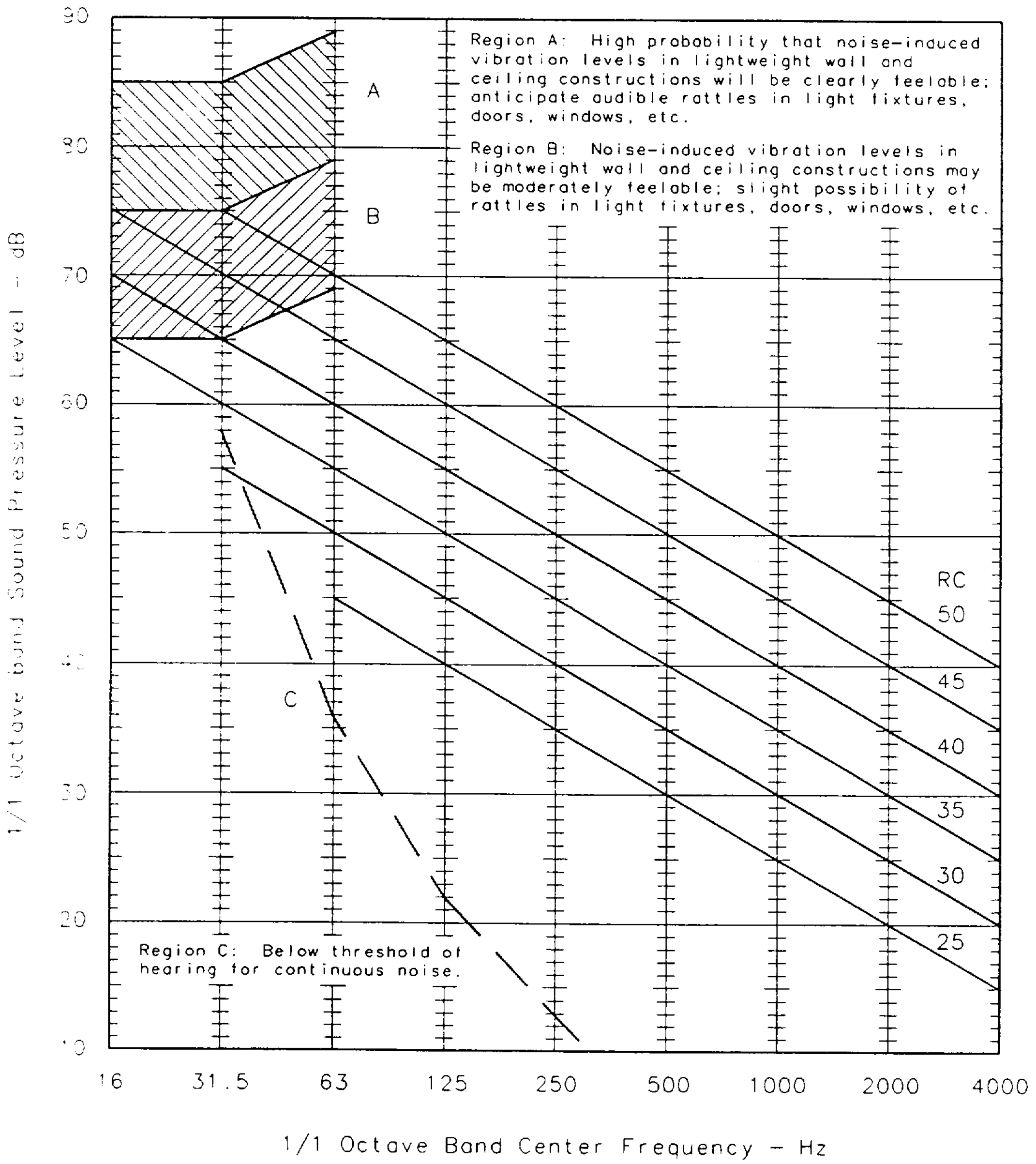


Figure 11-4 ROOM CRITERION CURVES

levels in the 500 Hz, 1,000 Hz, and 2,000 Hz 1/1 octave bands, or

$$RC = \left[\frac{40 + 33 + 27}{3}\right] \text{ or } RC = 33$$

Thus, the RC level is RC 33. The 1/1 octave band sound pressure levels for the background noise are plotted in Figure 11-5. The RC 33 curve (level in 1,000 Hz 1/1 octave band is 33 dB) is shown in the figure. A dashed line 5 dB above the RC 33 curve for frequencies below 500 Hz and a dashed line 3 dB above the RC 33 curve for frequencies above 500 Hz are also shown in the figure. An examination of the figure indicates that at frequencies below the 250 Hz 1/1 octave band, the 1/1 octave band sound pressure levels of the background noise are 5 dB or more above the RC 33 curve. Thus, the background noise has a rumble character. The 1/1 octave band sound pressure levels above 500 Hz are equal to or below the RC 33 curve, so there is no problem at these frequencies. The RC rating of the background noise is RC 33(R).

D GENERAL INFORMATION ON THE DESIGN OF HVAC SYSTEMS

Several general factors should be considered when selecting fans and other related equipment and when designing air distribution systems to minimize the noise transmitted from different components of the system to the occupied spaces which it serves. They include:

1. Air distribution systems should be designed to minimize flow resistance and turbulence. High flow resistance increases the required fan pressure, which results in higher noise being generated by the fan. Turbulence increases the flow noise generated by duct fittings and dampers in the air distribution system.
2. A fan should be selected to operate as near as possible to its rated peak efficiency when handling the required quantity of air and static pressure. Also, a fan should be selected which generates the lowest possible noise but still meets the required design conditions for which it is selected. Oversized or undersized fans which do not operate at or near rated peak efficiencies result in substantially higher noise levels.
3. Duct connections at both the fan inlet and outlet should be designed for uniform and straight air flow. Failure to do this can result in severe turbulence at the fan inlet and outlet and in flow separation at the fan blades. Both of these can significantly increase the noise generated by the fan.
4. Care should be exercised when selecting duct silencers to attenuate supply or return air noise. Duct silencers can significantly increase the required fan static pressure. When a rectangular duct silencer is used, it may be necessary to line the duct for a distance of at least ten feet beyond the silencer with a minimum one inch thick fiberglass duct lining to reduce high frequency regenerated noise associated

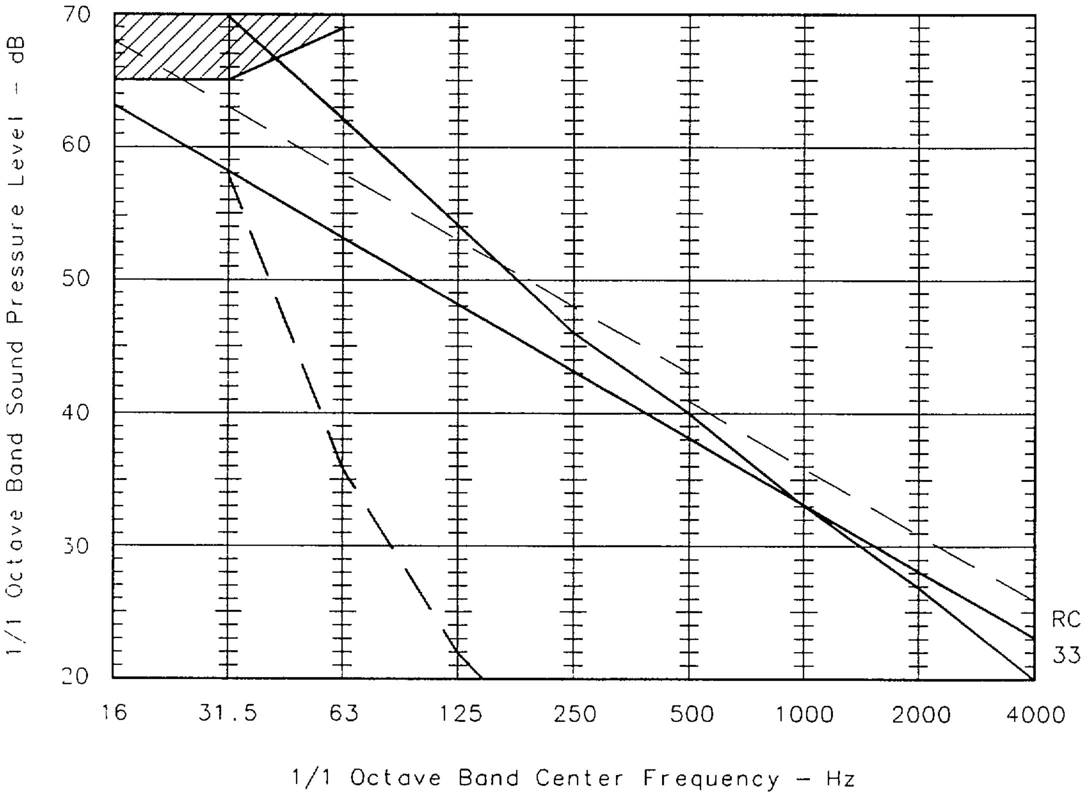


Figure 11-5 RC LEVEL FOR EXAMPLE 11-4

with the silencer. For some applications, acoustically lined sound plenums may be used in the place of duct silencers.

5. Fan-powered mixing boxes associated with variable-volume air distribution systems should not be placed over or near noise-sensitive areas.
6. Air flowing by or through elbows or duct branch take-offs generate turbulence. To minimize the flow noise associated with this turbulence, whenever possible, elbows and duct branch take-offs should be located 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 ten duct diameters in critical noise areas.
7. Near critical noise areas, it may be desirable to expand the duct cross-section area to keep the air flow velocity in the duct as low as possible. This will reduce potential flow noise associated with turbulence in these areas.
8. Turning vanes should be used in large 90 degree rectangular elbows. This provides a smoother transition in which the air can change flow direction, thus reducing turbulence.
9. Grilles, diffusers and registers should be placed as far as possible from elbows and branch take-offs.
10. Dampers in grilles, diffusers and registers should not be used for balancing.

Table 14-37 lists several common sound sources associated with mechanical equipment noise. Anticipated sound transmission paths and recommended noise reduction methods are also listed in the table. Airborne and/or structure-borne noise can follow any or all of the transmission paths associated with a specified sound source.

With respect to the quality of sound associated with HVAC system noise in an occupied space, fan noise generally contributes to the sound levels in the 63 Hz through 250 Hz 1/1 octave frequency bands. This is shown in Figure 11-6 as curve A. Diffuser noise usually contributes to the overall HVAC noise in the 250 Hz through 8,000 Hz 1/1 octave frequency bands. This is shown as curve B in Figure 11-6. The overall sound pressure levels associated with both the fan and diffuser noise is shown as curve D. The RC level of the overall noise is RC 36. The RC 36 curve is superimposed over curve D. As can be seen by comparing the RC curve with curve D, the classification of the overall noise is neutral. Curve D represents what would be considered acceptable and desirable 1/1 octave band sound pressure levels in many occupied spaces.

In order to effectively deal with each of the different sound sources and related sound paths associated with a HVAC system, the following design procedures are suggested:

1. Determine the design goal for HVAC system noise for each critical area according to its use and construction. Use Table 14-35 to specify the desirable NC or RC levels.
2. Relative to equipment that radiates sound directly into a room, select equipment that will be quiet enough to meet the desired design goal.
3. If central or roof-mounted mechanical equipment is used, complete an initial design and layout of the HVAC system, using acoustical treatment where it appears appropriate.
4. Starting at the fan, appropriately 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 possible duct sound breakout when central fans are adjacent to the room of interest or roof-mounted fans are above the room of interest.
5. If the mechanical equipment room is adjacent to the room of interest, determine the sound pressure levels in the room associated with sound transmitted through the mechanical equipment room wall.
6. Add the sound pressure levels in the room of interest that are associated with all of the sound paths between the mechanical equipment room or roof-mounted unit and the room of interest.
7. Determine the corresponding NC or RC level associated with the calculated total sound pressure levels in the room of interest.
8. If the NC or RC level exceeds the design goal,

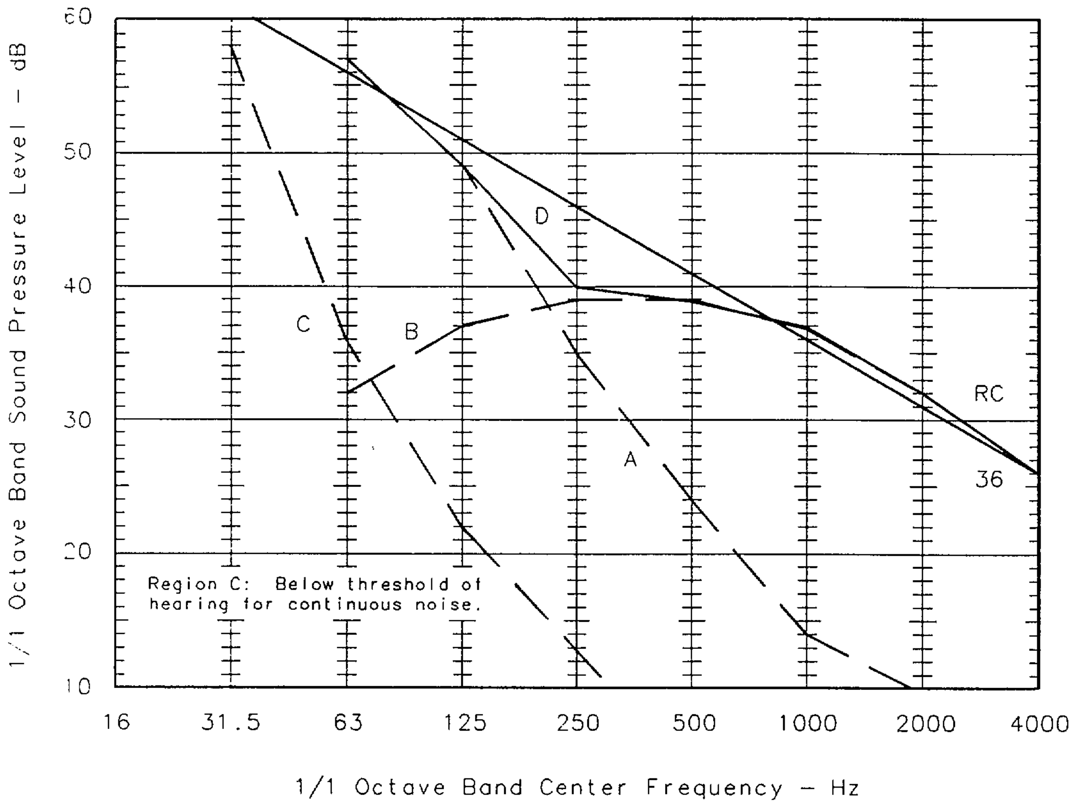


Figure 11-6 ILLUSTRATION OF WELL-BALANCED HVAC SOUND SPECTRUM FOR OCCUPIED SPACES

determine the 1/1 octave frequency bands in which the corresponding sound pressure levels are exceeded and the sound paths that are associated with these 1/1 octave frequency bands.

9. Redesign the system, adding additional sound attenuation to the paths which contribute to the excessive sound pressure levels in the room of interest.
10. Repeat Steps 4 through 9 until the desired design goal is achieved.
11. Steps 3 through 10 must be repeated for every room that is to be analyzed.
12. Make sure that noise radiated by outdoor equipment will not disturb adjacent properties.
13. With respect to outdoor equipment, use barriers when noise associated with the equipment will disturb adjacent properties.
14. If mechanical equipment is located on upper floors or is roof-mounted, vibration isolate all reciprocating and rotating equipment. It may be necessary to vibration isolate mechanical equipment that is located in the basement of a building.
15. If possible, use flexible connectors between rotating and reciprocating equipment and pipes and ducts that are connected to the equipment.
16. If it is not possible to use flexible connectors between rotating and reciprocating equipment and pipes and ducts connected to the equipment, use spring or neoprene hangers to vibration isolate the ducts and pipes within the first twenty feet of the equipment.
17. Use either spring or neoprene hangers. Do not use both.
18. Use flexible conduit between rigid electrical conduit and reciprocating and rotating equipment.

E FANS

The sound power generation of a given fan performing a specific task is best obtained from the fan manufacturers test data. Manufacturers' test data should be obtained from either AMCA Standard 300-85, Reverberant Room Method for Sound Testing of Fans, or ANSI/ASHRAE Standard 68-1986/ANSI/AMCA Standard 330-86, Laboratory Method of Testing In-Duct Sound Power Measurement Procedure for Fans. When such data are not available, the 1/1 octave band sound power levels for various fans can be estimated by the procedures outlined below.

While the size divisions of the fans shown in Table 14-38 are somewhat arbitrary, these divisions are practical for estimating fan noise. Fans generate a tone at the blade passage frequency. To account for this, the sound power level in the 1/1 octave band in which the blade passage frequency occurs is increased by a specified amount. The number of decibels to be added to this 1/1 octave band is called the blade frequency increment (B_f). Table 14-39 gives an estimate of the 1/1 octave band for different types of fans in which the blade passage frequency occurs and the corresponding blade frequency increment. For a more accurate estimate of the blade passage frequency, B_f, the following equation can be used:

Equation 11-11

$$B_f = \left(\frac{RPM}{60}\right) \times \text{no. of blades}$$

where RPM is the rotational speed of the fan in revolutions per minute.

The specific sound power levels associated with fan total sound power given in Table 14-38 in Chapter 14 are for fans operating at a point of operation where the volume flow rate equals 1 cfm (0.5 l/s) and the static pressure is 1 in. w.g. (250 Pa). Equation 11-12 is used to calculate the fan total sound power levels corresponding to a specific point of operation.

Equation 11-12

$$L_W = K_W + 10 \log_{10}\left[\frac{Q}{Q_1}\right] + 20 \log_{10}\left[\frac{P}{P_1}\right] + C$$

where L_W is the estimated sound power level of the fan in dB; K_w is the specific sound power level in dB from Table 14-38; Q is the flow rate in cfm; Q_1 is 1 cfm, P is the pressure drop in inches w.g.; P_1 is 1 in. w.g., C is the correction factor in dB for the case where the point of fan operation is other than the point of peak efficiency. Values for C are obtained from Table 14-40.

Example 11-5

A forward curved fan supplies 10,000 cfm of air at a static pressure of 1.5 in. w.g. It has 24 blades and operates at 1,175 rpm. The fan has a peak efficiency of 85%. The fan horsepower is 3 HP. Determine the outlet fan sound power levels.

Solution

$$L_W = K_w + 10 \log_{10} [Q] + 20 \log_{10} [P] + C$$

operating efficiency (E1)

$$= \frac{\text{flow volume} \times \text{static pressure}}{6356 \times \text{Hp}} \times 100$$

$$E1 = \frac{10000 \times 1.5}{6356 \times 3} \times 100 = 79\%$$

Peak efficiency E2 = 85%.

$$\%\text{ of peak efficiency} = \frac{E1}{E2} \times 100$$

$$= \frac{79}{85} \times 100 = 93\%$$

From Table 14-40, the correction for off peak efficiency operation is 0 dB. Thus,

$$L_W = K_W + 10 \log_{10}[10000] + 20 \log_{10}[1.5] + 0$$

$$= K_W + 44$$

$$B_f = \frac{1175}{60} \times 24 = 470 \text{ Hz}$$

470 Hz is in the 500 Hz 1/1 octave frequency band. From Table 14-40, the blade frequency increment is 2 dB.

The results are tabulated below. For metric units, convert the metric data to its U.S. unit equivalents and calculate as above, using the equivalents in Chapter 14, Section F.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
K_W—Table 14-38	47	43	39	36	34	32	28	
Equation (11-12)	44	44	44	44	44	44	44	
Table 14-39				2				
$L_{W(OBL)}$ (dB)	91	87	83	82	78	76	72	

F AERODYNAMIC NOISE

Aerodynamic noise is generated when airflow in the duct becomes turbulent as it passes through sharp bends, sudden enlargements or contractions, and most devices that cause substantial pressure drops. Aerodynamic noise is usually of no importance when the velocity of airflow is below 2000 feet per minute (10 m/s) in the main ducts; below 1500 fpm (7.5 m/s) in branch ducts; and below 800 fpm (4 m/s) in ducts serving room terminal devices. When the duct system velocities are in excess of the above or when the duct does not follow good airflow design principles, aerodynamic noise can become a major problem.

Aerodynamic noise is predominantly low frequency in spectrum (31.5 through 500 Hz 1/1 octave band center frequencies). Low frequency energy is transmitted readily, with little loss, through the light gauge walls of ducts and through suspended acoustic ceilings.

The duct elements covered in this section include: dampers, elbows with turning vanes, elbows without turning vanes, junctions, and 90 degree branch takeoffs.

1. Dampers

The 1/1 octave band sound power level of the noise generated by single or multi-blade dampers can be predicted by Equation 11-13.

Equation 11-13

$$L_W(f_O) = K_D + 10 \log_{10} \left[\frac{f_O}{63}\right] + 50 \log_{10} [U_C] + 10 \log_{10} [S] + 10 \log_{10} [DH]$$

where f_o is the 1/1 octave band center frequency (Hz), U_c is the flow velocity (ft/sec) in the constricted part of the flow field determined according to Equation 11-16, S is the cross-section area (sq. ft.) of the duct, DH is the duct height (ft) normal to the damper axis, and K_D is the characteristic spectrum (Figure 11-7). Figure 11-8 shows a schematic of a single-blade damper. The regenerated sound power levels associated with dampers are obtained as follows:

Step 1: Determine the total pressure loss coefficient, C.

Equation 11-14

$$C = 15.9 \times 10^6 \frac{\Delta P}{(Q/S)^2}$$

where Q is the volume flow rate (cfm), ΔP is the total pressure loss (inches w.g.) across the damper, and S is the duct cross-section area (sq. ft.).

Step 2: Determine the blockage factor, BF.

For multi-blade dampers:

Equation 11-15a

$$BF = \frac{(\sqrt{C} - 1)}{(C - 1)} \qquad \text{If } C = 1, \text{ then } BF = 0.50.$$

For single-blade dampers:

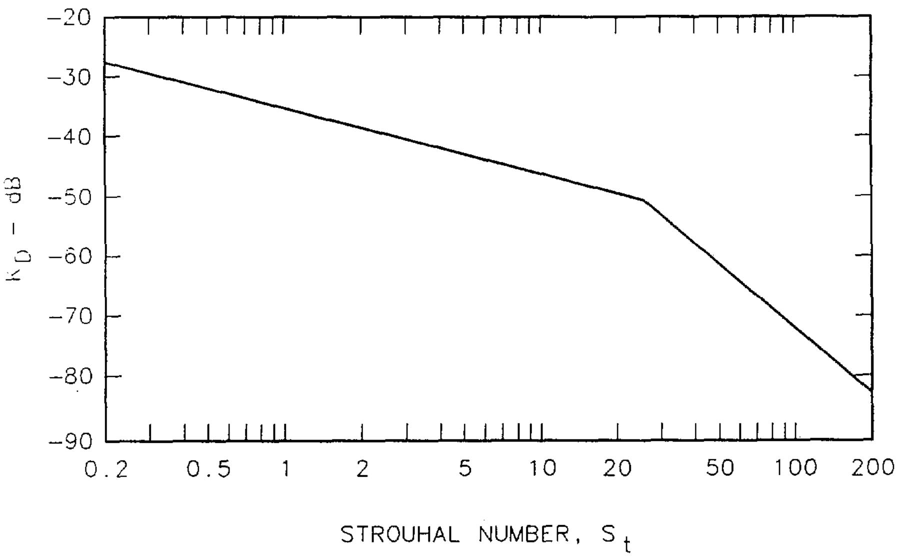


Figure 11-7 CHARACTERISTIC SPECTRUM, K_D, FOR DAMPERS

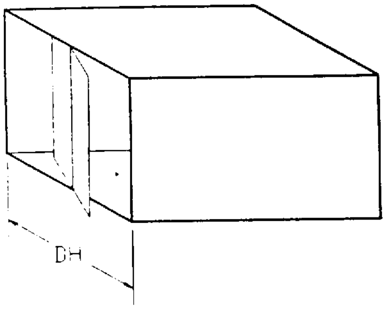


Figure 11-8 DAMPER

Equation 11-15b

$$BF = \frac{(\sqrt{C} - 1)}{(C - 1)} \qquad \text{for } C < 4$$

$$BF = 0.68\ C^{-0.15} - 0.22 \qquad \text{for } C > 4$$

Step 3: Determine the flow velocity, U_c (ft/sec), in the damper constriction.

Equation 11-16

$$U_C = 0.0167 \times \frac{Q}{S \times BF}$$

Step 4: Determine the Strouhal number, S_t. The Strouhal number which corresponds to the 1/1 octave band center frequencies is given by

Equation 11-17

$$S_t = \frac{f_O \times DH}{U_C}$$

Determine the Characteristic Spectrum, K_D.

The characteristic spectrum is the same for all dampers and duct sizes if plotted as a function of the Strouhal frequency. The characteristic spectrum, K_D, is obtained from Figure 11-11 or from

Equation 11-18

$$K_D = -36.6 - 10.7 \log_{10} [S_t] \qquad \text{for } S_t \le 25$$

$$K_D = -1.1 - 35.9 \log_{10} [S_t] \qquad \text{for } S_t > 25$$

All the required information is now available for calculating the 1/1 octave band sound power levels predicted by Equation 11-13.

Example 11-6

Determine the 1/1 octave band sound power levels associated with a multi-blade damper positioned in a 12 in. x 12 in. duct. The pressure drop across the damper is 0.5 in. w.g. and the volume flow rate in the duct is 4,000 cfm.

Solution

From the given data: Q = 4,000 cfm; P = 0.5 inches w.g.; S = 1 sq. ft.; DH = 1 ft.

Step 1: Total pressure loss coefficient, C.

$$C = 15.9 \times 10^6 \times \frac{0.5}{(4000/1)^2} = 0.5$$

Step 2: Blockage factor, BF

$$BF = \frac{(\sqrt{0.5} - 1)}{(0.5 - 1)} = 0.585$$

Step 3: Constricted flow velocity, U_c.

$$U_c = 0.0167 \times \frac{4000}{1.0 \times 0.585} = 114 \text{ (ft/sec)}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
S_t	0.55	1.1	2.2	4.4	8.8	17.6	35.1	70.2
K_T, dB	−33.5	−36.7	−40.0	−43.2	−46.4	−49.6	−56.6	−67.4
10 $Log_{10}[f_o/63]$, dB	0.0	3.0	6.0	9.0	12.0	15.0	18.0	21.0
50 $Log_{10}[U_c]$, dB	102.8	102.8	102.8	102.8	102.8	102.8	102.8	102.8
10 $Log_{10}[S]$, dB	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
10 $Log_{10}[CD]$, dB	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
$L_W(f_O)$, dB	69.3	69.1	68.8	68.6	68.4	68.2	64.2	56.4

2. Elbows Fitted With Turning Vanes

The 1/1 octave band sound power levels associated with the noise generated by elbows fitted with turning vanes can be predicted if the total pressure drop across the blades is known or can be estimated. The method that is presented applies to any elbow that has an angle between 60 degrees and 120 degrees. The 1/1 octave band sound power levels generated by elbows with turning vanes is given by

Equation 11-19

$$L_W(f_O) = K_T + 10 \log_{10} \left[\frac{f_O}{63}\right] + 50 \log_{10} [U_C] + 10 \log_{10} [S] + 10 \log_{10} [CD] + 10 \log_{10} [n]$$

where f_o is the 1/1 octave band center frequency (Hz), U_c is the flow velocity (ft/sec) in the constricted part of the flow field between the blades determined from Equation 11-22, S is the cross-section area (sq. ft.) of the duct, CD is the cord length (in.) of a typical vane, n is the number of turning vanes, and K_T is the characteristic spectrum (Figure 11-9). In addition to

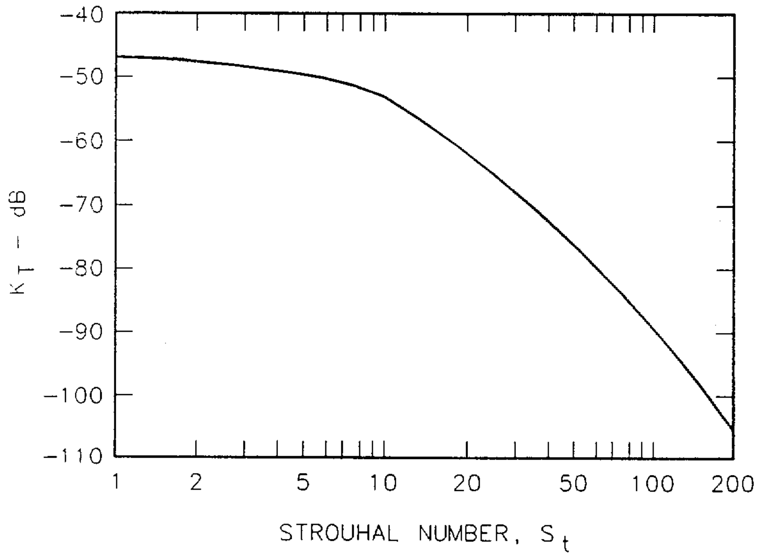


Figure 11-9 CHARACTERISTIC SPECTRUM, K, FOR ELBOWS FITTED WITH TURNING VANES

the above parameters, it is also necessary to know the duct height DH (ft) normal to the turning vane length (Figure 11-10). The regenerated sound power levels associated with elbows with turning vanes are obtained as follows:

Step 1: Determine the total pressure loss coefficient, C using Equation 11-14:

$$C = 15.9 \times 10^6 \times \frac{\Delta P}{(Q/S)^2}$$

Step 2: Determine the blockage factor, BF using Equation 11-15a:

$$BF = \frac{(\sqrt{C} - 1)}{(C - 1)}$$

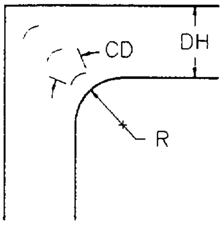


Figure 11-10 90° ELBOW WITH TURNING VANES

Step 3: Determine the flow velocity, U_c (ft/sec), in the turning vane constriction using Equation 11-16:

$$U_C = 0.0167 \times \frac{Q}{S \times BF}$$

Step 4: Determine the Strouhal number, S_t using Equation 11-17:

$$S_t = \frac{f_O \times DH}{U_C}$$

Step 5: Determine the characteristic spectrum, K_T.

Equation 11-20

$$K_T = -47.5 - 7.69\,[\log_{10}[S_t]]^{2.5}$$

The characteristic spectrum is the same for any elbow fitted with turning vanes if plotted as a function of the Strouhal number. The characteristic spectrum is obtained from Figure 11-9.

All the required information is now available for calculating the 1/1 octave band sound power levels predicted by Equation 11-19.

Example 11-7

A 90° elbow of a 20 in. × 20 in. duct is fitted with 5 turning vanes that have a cord length of 7.9 inches. The volume flow rate is 8,500 cfm and the corresponding pressure loss across the turning vanes is 0.16 inch in. w.g. Determine the resulting 1/1 octave band sound power levels.

Solution

From the given data: $Q = 8{,}500$ cfm; $\Delta P = 0.16$ inch in. w.g.; $S = 2.78$ sq. ft.; $DH = 1.64$ ft; $CD = 7.9$ inches; $n = 5$.

Step 1: Total pressure loss coefficient, C

$$C = 15.9 \times 10^6 \times \frac{0.16}{(8500/2.78)^2} = 0.27$$

Step 2: Blockage factor, BF

$$BF = \frac{(\sqrt{0.27} - 1)}{(0.27 - 1)} = 0.66$$

Step 3: Constricted flow velocity, U_c

$$U_C = 0.0167 \times \frac{8500}{2.78 \times 0.66} = 77.4 \text{ (ft/sec)}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
S_t	1.3	2.6	5.3	10.6	21.2	42.4	84.8	167.5
K_T, dB	−47.5	−48.4	−50.9	−55.7	−63.1	−73.5	−87.0	−104.2
10 $\text{Log}^{10}[f_0/63]$,dB	0.0	3.0	6.0	9.0	12.0	15.0	18.0	21.0
50 $\text{Log}_{10}[U_C]$, dB	94.4	94.4	94.4	94.4	94.4	94.4	94.4	94.4
10 $\text{Log}_{10}[S]$, dB	4.4	4.4	4.4	4.4	4.4	4.4	4.4	4.4
10 $\text{Log}_{10}[CD]$, dB	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0
10 $\text{Log}_{10}[n]$, dB	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0
$L_W(f_0)$, dB	67.3	69.4	69.9	68.1	63.7	56.3	45.8	31.6

3. Junctions and Turns

Equation 11-21 has been developed as a means to predict the regenerated sound power levels in a branch duct associated with air flowing in duct turns and junctions. Equation 11-21 applies to 90 degree elbows without turning vanes, X-junctions, T-junctions, and 90 degree branch takeoffs (Figure 11-11).

Equation 11-21

$$L_W(f_O)_b = L_b(f_O) + \Delta r + \Delta T$$

$L_b(f_O)$ is given by

Equation 11-22

$$L_b(f_O) = K_J + 10 \log_{10}\left[\frac{f_O}{63}\right] + 50 \log_{10} [U_B] + 10 \log_{10} [S_B] + 10 \log_{10} [D_B]$$

where f_o is the 1/1 octave band center frequency (Hz), D_B is the equivalent diameter (ft) of the branch duct, U_B is the flow velocity (ft/sec) in the branch duct, S_B is the cross-section area (sq. ft.) of the branch duct, and K_J is the characteristic spectrum (Figure 11-12). If the branch duct is circular, D_B is the duct diameter. If the branch duct is rectangular, D_B is obtained from

Equation 11-23

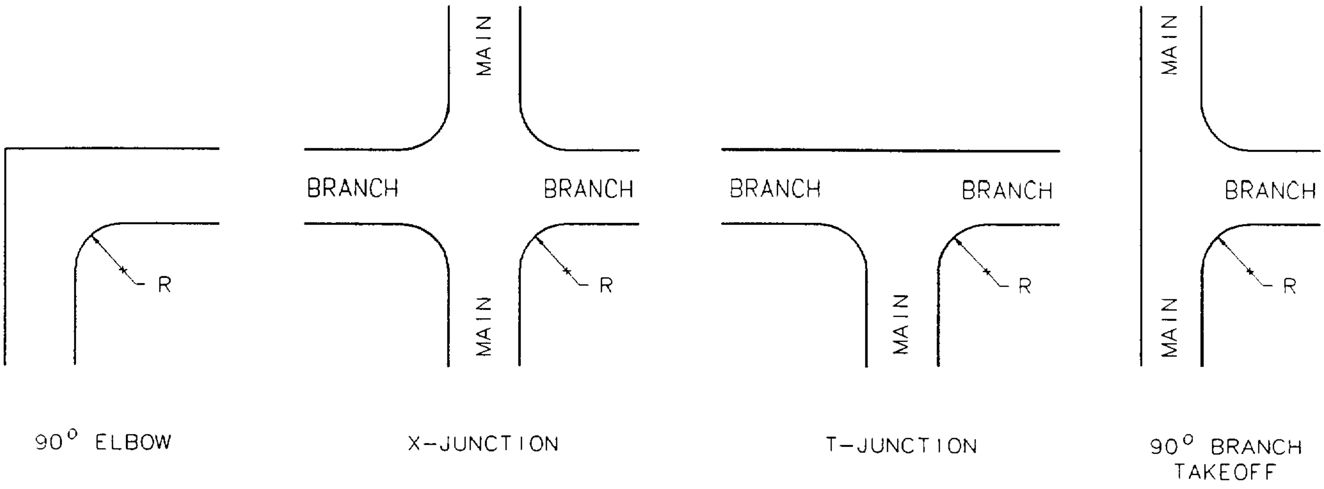
$$D_B = \left[\frac{4\ S_B}{\pi}\right]^{1/2}$$

The corresponding flow velocity (ft/sec), U_B, is given by

Equation 11-24

$$U_B = \frac{Q_B}{60\ S_B}$$

where Q_B is the volume flow rate (cfm) in the branch. D_M (ft) and U_M (ft/sec) for the main duct are obtained in a manner similar to those implied by Equations 11-23 and 11-24.



90° ELBOW X-JUNCTION T-JUNCTION 90° BRANCH TAKEOFF

Figure 11-11 ELBOWS, JUNCTIONS, AND BRANCH TAKEOFFS

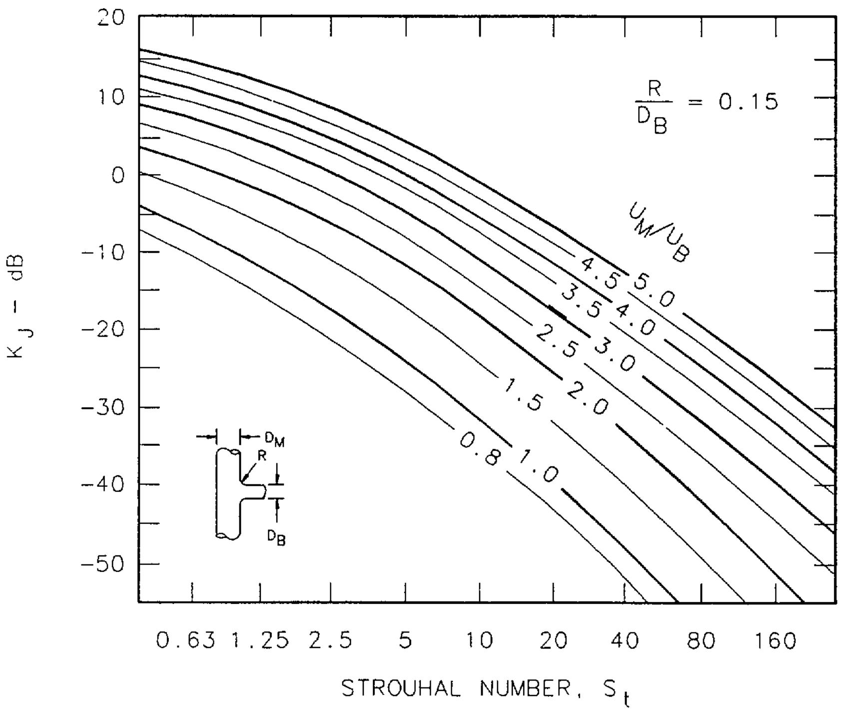


Figure 11-12 CHARACTERISTICS SPECTRUM, K_J, FOR JUNCTIONS

In Equation 11-21, Δr is the correction term that quantifies the effect of the size of the radius of the bend or elbow associated with the turn or junction. r is obtained from Figure 11-13(a) or from

Equation 11-25

$$\Delta r = \left[1.0 - \frac{RD}{0.15}\right] \times [6.793 - 1.86 \log_{10}(S_t)]$$

where RD is the rounding parameter and S_t is the Strouhal number. RD is specified by

Equation 11-26

$$RD = \frac{R}{12\ D_B}$$

where R is the radius (in) of the bend or elbow associated with the turn or junction and D_B is defined above. The Strouhal number is given by

Equation 11-27

$$S_t = \frac{f_O \times D_B}{U_B}$$

In Equation 11-21, ΔT is a correction factor for upstream turbulence. This correction is only applied when there are dampers, elbows or branch takeoffs upstream within five main duct diameters of the turn or junction being examined. ΔT is obtained from Figure 11-13(b) or from

Equation 11-28

$$\Delta T = -1.667 + 1.8\ m - 0.133\ m^2$$

where m is the velocity ratio that is specified by

Equation 11-29

$$m = \frac{U_m}{U_B}$$

U_m is the flow velocity in the main duct before the turn or junction and U_B is the flow velocity in the branch duct after the turn or junction.

The characteristic spectrum, K_J, in Equation 11-30 is obtained from Figure 11-12 or from

Equation 11-30

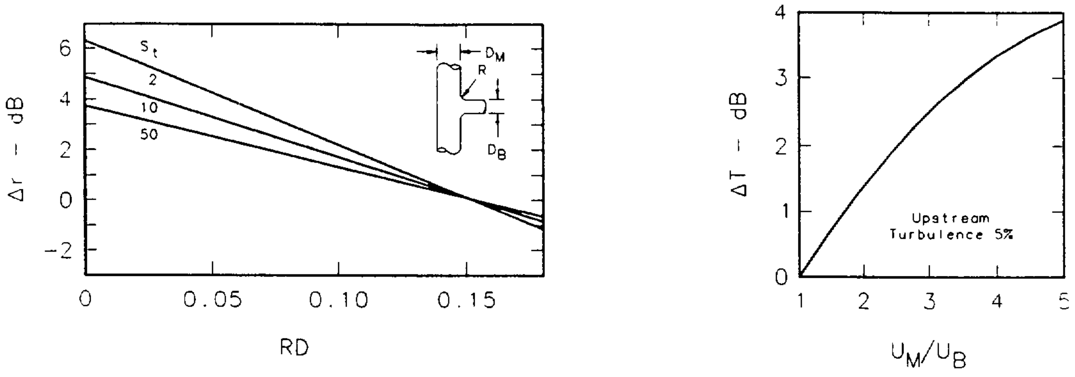
$$K_J = -21.61 + 12.388\ m^{0.673} - 16.482\ m^{-0.303} \log_{10}[S_r] - 5.047\ m^{-0.254} [\log_{10}[S_t]]^2$$

The regenerated sound power levels in a branch duct and the continuation of the main duct that are associated with a turn or junction are obtained as follows:

Step 1: Obtain or determine the values of D_B and D_M.

Step 2: Determine the values of U_B and U_M.

Step 3: Determine the ratios, D_M/D_B and m.



(a) Corner Rounding Correction

(b) Correction for Upstream Turbulence

Figure 11-13 CORRECTION FACTORS FOR CORNER ROUNDING AND FOR UPSTREAM TURBULENCE

Step 4: Determine the rounding parameter, RD.

Step 5: Determine the Strouhal number, S_t.

Step 6: Determine the value of Δr.

Step 7: If turbulence is present, determine the value of Δt.

Step 8: Determine the characteristic spectrum, K_J.

Step 9: Determine the value of the branch sound power levels, $L_w(f_o)_b$.

Step 10: Specify the type of junction and determine the main duct sound power levels, $L_w(f_o)_m$, using Equations 11-31, 11-32, 11-33, or 11-34.

Equation 11-31

X-Junction:

$$L_W(f_o)_m = L_W(f_o)_b + 20 \log_{10}\left[\frac{D_M}{D_B}\right] + 3$$

Equation 11-32

T-Junction:

$$L_W(f_o)_m = L_W(f_o) + 3$$

Equation 11-33

90° Elbow without Turning Vanes:

$$L_W(f_o)_m = L_W(f_o)_b$$

Equation 11-34

90° Branch Takeoff:

$$L_W(f_o)_m = L_W(f_o)_b + 20 \log_{10}\left[\frac{D_M}{D_B}\right]$$

Example 11-8

Determine the regenerated sound power levels associated with a X-junction that exist in the branch and main ducts given the following information:

Main Duct: Rectangular—12 in. x 36 in., Volume flow rate—12,000 cfm

Branch Duct: Rectangular—10 in. x 10 in., Volume flow rate—1,200 cfm

Radius of bend or elbow: 0.0

No dampers, elbows or branch takeoffs are within five main duct diameters of junction.

Solution

Step 1: Determine the values of D_B and D_M:

$$D_M = \left[\frac{4 \times 12 \times 36}{\pi \times 144}\right]^{1/2} = 1.95 \text{ ft}$$

$$D_B = \left[\frac{4 \times 10 \times 10}{\pi \times 144}\right]^{1/2} = 0.94 \text{ ft}$$

Step 2: Determine the values of U_B and U_M:

$$U_M = \frac{1200 \times 144}{12 \times 36 \times 60} = 66.67 \text{ ft/sec}$$

$$U_B = \frac{1200 \times 144}{10 \times 10 \times 60} = 28.80 \text{ ft/sec}$$

Step 3: Determine the ratios, D_m/D_B and m:

$$\frac{D_M}{D_B} = \frac{1.95}{0.94} = 2.06$$

$$m = \frac{66.67}{28.80} = 2.31$$

Step 4: Determine the rounding parameter, RD:

$$RD = \frac{0}{12 \times 0.95} = 0$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
S_t	2.0	4.1	8.2	16.3	32.6	65.3	130.6	261.2
K_T, dB	−4.2	−9.1	−14.9	−21.3	−28.5	−36.4	−45.1	−54.5
$10 \log_{10}[f_0/63]$, dB	0.0	3.0	6.0	9.0	12.0	15.0	18.0	21.0
$50 \log_{10}[U_B]$, dB	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0
$10 \log_{10}[S_B]$, dB	−1.6	−1.6	−1.6	−1.6	−1.6	−1.6	−1.6	−1.6
$10 \log_{10}[D_B]$, dB	−0.3	−0.3	−0.3	−0.3	−0.3	−0.3		
Δr, dB	6.2	5.7	5.1	4.5	4.0	3.4	2.9	2.3
ΔT, dB	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
$L_W(f_o)_b$, dB	73.2	70.6	67.4	63.4	58.6	53.1	46.9	39.9
$20 \log_{10}(D_M/D_B)$, dB	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2
	3.0	3.0	3.0	3.0	3.0	3.0	3.0	
$L_W(f_o)_{,m}$, dB	82.4	79.8	76.6	72.6	67.8	62.3	56.1	49.1

Example 11-9

Determine the regenerated sound power levels associated with a T-junction that exist in the branch and main ducts given the following information:

Main Duct: Rectangular—12 in. x 36 in., Volume flow rate—12,000 CFM

Branch Duct: Rectangular—12 in. x 18 in., Volume flow rate—6,000 CFM

Radius of bend or elbow: 0.0 in.

No dampers, elbows or branch takeoffs within five main duct diameters of junction.

Step 1: Determine the values of D_B and D_M:

$$D_M = \left[\frac{4 \times 12 \times 36}{\pi \times 144}\right]^{1/2} = 1.95 \text{ ft}$$

$$D_B = \left[\frac{4 \times 12 \times 18}{\pi \times 144}\right]^{1/2} = 1.38 \text{ ft}$$

Step 2: Determine the values of U_B and U_M:

$$U_M = \frac{12{,}000 \times 144}{12 \times 36 \times 60} = 66.67 \text{ ft/sec}$$

$$U_B = \frac{6000 \times 144}{12 \times 18 \times 60} = 66.67 \text{ ft/sec}$$

Step 3: Determine the ratios, D_m/D_B and m:

$$\frac{D_M}{D_B} = \frac{1.95}{1.38} = 1.41$$

$$m = \frac{66.67}{66.67} = 1.00$$

Step 4: Determine the rounding parameter, RD:

$$RD = \frac{0}{12 \times 1.38} = 0$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
S_t	1.3	2.6	5.2	10.4	20.7	41.5	82.9	165.8
K_J, db)	−11.1	−16.9	−23.6	−31.2	−49.7	−49.1	−59.4	−70.7
$10 \text{ Log}_{10}[f_0/63]$, dB	0.0	3.0	6.0	9.0	12.0	15.0	18.0	21.0
$50 \text{ Log}_{10}[U_B]$, dB	91.2	91.2	91.2	91.2	91.2	91.2	91.2	91.2
$10 \text{ Log}_{10}[S_B]$, dB	1.8	1.8	1.8	1.8	1.8	1.8	1.8	1.8
$10 \text{ Log}_{10}[D_B]$, dB	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4
Δr, dB	6.6	6.0	5.5	4.9	4.3	3.8	3.2	2.7
ΔT, dB	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
$L_W(f_o)$, dB	89.8	86.5	82.3	77.1	71.1	64.1	56.2	47.4
	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0
$L_W(f_o)$, dB	92.8	89.5	85.3	80.1	74.1	67.1	59.2	50.4

Example 11-10

Determine the regenerated sound power levels associated with a 90° elbow without turning vanes given the following information:

Main Duct: Rectangular—12 in. x 36 in., Volume flow rate—12,000 CFM

Branch Duct: Rectangular—12 in. x 36 in., Volume flow rate—12,000 CFM

Radius of bend or elbow:—0.0 in.
No dampers, elbows or branch takeoffs within five main duct diameters of elbow.

Solution

Step 1: Determine the values of D_B and D_M:

$$D_M = \left[\frac{4 \times 12 \times 36}{\pi \times 144}\right]^{1/2} = 1.95 \text{ ft}$$

$$D_B = \left[\frac{4 \times 12 \times 36}{\pi \times 144}\right] = 1.95 \text{ ft}$$

Step 2: Determine the values of U_B and U_M:

$$U_M = \frac{12{,}000 \times 144}{12 \times 36 \times 60} = 66.67 \text{ ft/sec}$$

$$U_B = \frac{12{,}000 \times 144}{12 \times 36 \times 60} 66.67 \text{ ft/sec}$$

Step 3: Determine the ratios, D_M/D_B and m:

$$\frac{D_M}{D_B} = \frac{1.95}{1.95} = 1.00$$

$$m = \frac{66.67}{66.67} = 1.00$$

Step 4: Determine the rounding parameter, RD:

$$RD = \frac{0}{12 \times 1.95} = 0$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
S_t	1.8	3.7	7.3	14.7	29.3	58.6	1117.3	234.8
K_J, dB	−13.9	−20.1	−27.2	−35.3	−44.3	−54.1	−64.9	−76.7
10 $Log_{10}[f_o/63]$, dB	0.00	3.0	6.0	9.0	12.0	15.0	18.0	21.0
50 $Log_{10}[U_B]$, dB	91.2	91.2	91.2	91.2	91.2	91.2	91.2	91.2
10 $Log_{10}[S_B]$, dB	4.8	4.8	4.8	4.8	4.8	4.8	4.8	4.8
10 $Log_{10}[D_B]$, dB	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9
Δr, dB	6.3	5.7	5.2	4.6	4.1	3.5	2.9	2.4
ΔT, dB	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
$L_W(f_o)_b$, dB	91.3	87.5	82.5	77.2	70.7	63.3	54.9	45.6
	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
$L_W(f_o)_m$, dB	91.3	87.5	82.5	77.2	70.7	63.3	54.9	45.6

Example 11-11

Determine the regenerated sound power levels associated with a 90° branch takeoff that exist in the branch and main ducts given the following information:

Main Duct: Rectangular—12 in. × 36 in., Volume flow rate—12,000 CFM

Branch Duct: Rectangular—10 in. × 10 in., Volume flow rate—1,200 CFM

Radius of bend or elbow: 0.0 in.

No dampers, elbows or branch takeoffs within five main duct diameters of takeoff.

Solution

Step 1: Determine the values of D_B and D_M:

$$D_M = \left[\frac{4 \times 12 \times 36}{\pi \times 144}\right]^{1\,2} = 1.95 \text{ ft}$$

$$D_B = \left[\frac{4 \times 10 \times 10}{\pi \times 144}\right]^{1\,2} = 0.94 \text{ ft}$$

Step 2: Determine the values of U_B and U_M:

$$U_M = \frac{12{,}000 \times 144}{12 \times 36 \times 60} = 66.67 \text{ ft/sec}$$

$$U_B = \frac{1200 \times 144}{10 \times 10 \times 60} = 28.80 \text{ ft/sec}$$

Step 3: Determine the ratios, D_m/D_B and m:

$$\frac{D_M}{D_B} = \frac{1.95}{0.95} = 2.06$$

$$m = \frac{66.67}{28.80} = 2.31$$

Step 4: Determine the rounding parameter, RD:

$$RD = \frac{0}{12 \times 0.94} = 0$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
S_t	2.0	4.1	8.2	16.3	32.6	65.3	130.6	261.2
K_J, dB	−4.2	−9.1	−14.9	−21.3	−28.5	−36.4	−45.1	−54.5
10 $Log_{10}[f_0/63]$, dB	0.0	3.0	6.0	9.0	12.0	15.0	18.0	21.0
50 $Log_{10}[U_B]$, dB	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0
10 $Log_{10}[S_B]$, dB	−1.6	−1.6	−1.6	−1.6	−1.6	−1.6	−1.6	−1.6
10 $Log_{10}[D_B]$, dB	−0.3	−0.3	−0.3	−0.3	−0.3	−0.3	−0.3	−0.3
Δr, dB	6.2	5.7	5.1	4.5	4.0	3.4	2.9	2.3
ΔT, dB	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
$L_W(f_o)_b$, dB	73.2	70.6	67.4	63.4	58.6	53.1	46.9	39.9
20 $Log_{10}[D_M/D_B]$	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2
$L_W(f_o)_m$, dB	79.4	76.8	73.6	69.6	64.8	59.3	53.1	46.1

G DUCT TERMINAL DEVICES

Pressure reducing valves in mixing and variable volume boxes usually have published noise ratings indicating the sound power levels that are discharged from the low pressure end of the box. The manufacturer may also indicate the requirements, if any, for the sound attenuation materials to be installed in the low pressure duct between the box and outlet.

Some of the box manufacturers also test the noise radiated from the exterior of the box, however this data is not usually published. If the box is located away from critical areas (such as in a storeroom or corridor), the noise radiating from the box may be of no concern. If, however, the box is located above a critical space and separated from the space by a suspended acoustical ceiling which has little or no transmission loss at low frequencies, the noise radiated from the box may exceed the noise criterion for the room below. For this case it may be necessary to relocate the box to a non-critical area or to enclose it with a construction having a high transmission loss.

Room air terminal devices such as diffusers, grilles, air handling light fixtures and air handling suspension bars are always rated for noise generation. The test data is obtained in accordance with the Air Research Institute (ARI) Standard 880-87, Industry Standard for Air Terminals. The room air terminal unit should be selected to meet the noise criterion required or specified for the room, bearing in mind that the manufacturer's sound power rating is obtained with a uniform velocity distribution throughout the diffuser neck or grille collar. If a duct turn precedes the entrance to the diffuser or if a balancing damper is installed immediately before the diffuser, the air flow will be turbulent and the noise generated by the device will be substantially higher than the manufacturer's published data. This turbulence can be substantially reduced by specifying an equalizer grid to be placed in the neck of the diffuser. The equalizer grid provides a uniform velocity gradient within the neck of the diffuser and the sound power will be close to that listed in the manufacturer's catalog. If the equalizer grid is omitted, the sound power level of the diffuser can be increased by as much as 12 dB.

A flexible duct connection between the diffuser and the supply duct provides a convenient means to align the diffuser with respect to the ceiling grid. A misalignment in this connection that exceeds 1/4 of the diffuser diameter over a length of two times the diffuser diameter can cause a significant increase in the diffuser sound power levels relative to the levels specified by the manufacturer. If the diffuser offset is less than 1/8 of the length of the connection, there will be no appreciable increase in the sound power levels. If the offset is equal to or greater than the diffuser diameter over a connection length equal to two times the diffuser diameter, the sound power levels associated with the diffuser can be increased by as much as 12 dB.

Sound radiation associated with air flow through diffusers and diffusers with porous plates that terminate air conditioning ducts is similar to sound radiation associated with air flowing over a spoiler. The interaction of the airflow and diffuser guide vanes behaves as an acoustic dipole. Thus, the associated sound power is proportional to the sixth power of flow velocity and the third power of pressure. The pressure drop across a diffuser can be specified by the normalized pressure drop coefficient, ξ which is given by

Equation 11-35

$$\xi = 334.9 \frac{\Delta P}{\rho\, u^2}$$

where ΔP is the pressure drop across a diffuser (in. w.g.), ρ is the density of air (lb_m/ft^3), u is the mean flow velocity (ft/sec) of the air in the duct prior to the diffuser. For most situations, $\rho = 0.075\ lb_m/ft^3$, and u is obtained from:

Equation 11-36

$$u = \frac{Q}{60\ S}$$

where Q is the flow volume (cfm) and S is the duct cross-section area (ft^2) prior to the diffuser. The overall sound power level, $L_{w(overall)}$ (dB), associated with a diffuser is given by

Equation 11-37

$$L_{W(overall)} = 10 \log_{10}[S] + 30 \log_{10}[\xi] + 60 \log_{10}[u] - 31.3$$

where ξ, u, and S are as defined before.

The peak frequency, f_p (Hz), associated with sound generated by diffusers can be approximated by

Equation 11-38

$$f_p = 48.8\ u$$

where u is as defined above. The shape of the 1/1 octave band sound spectrum for a diffuser is similar to that shown in Figure 11-14. If the diffusers are generic rectangular, round, and square perforated face (with round inlet) diffusers, the equation for the curve in Figure 11-14 is given by

Equation 11-39

$$C = -5.82 - 0.15\ A - 1.13\ A^2$$

for generic round diffusers and by

Equation 11-40

$$C = -11.82 - 0.15\ A - 1.13\ A^2$$

for generic rectangular and square perforated face (with round inlet) diffusers where

Equation 11-41

$$A = I - II;$$

I = 1 for 63 Hz, 2 for 125 Hz, 3 for 250 Hz, etc.; and II is dependent upon peak frequency and is specified by:

$0 \le f_p < 44$ Hz $\quad II = 0$
$44 \le f_p < 88$ Hz $\quad II = 1$
$88 \le f_p < 177$ Hz $\quad II = 2$
$177 \le f_p < 355$ Hz $\quad II = 3$
$355 \le f_p < 710$ Hz $\quad II = 4$
$710 \le f_p < 1420$ Hz $\quad II = 5$
$1420 \le f_p < 2840$ Hz $\quad II = 6$
$2840 \le f_p < 5680$ Hz $\quad II = 7$
$5680 \le f_p < 11360$ Hz $\quad II = 8$

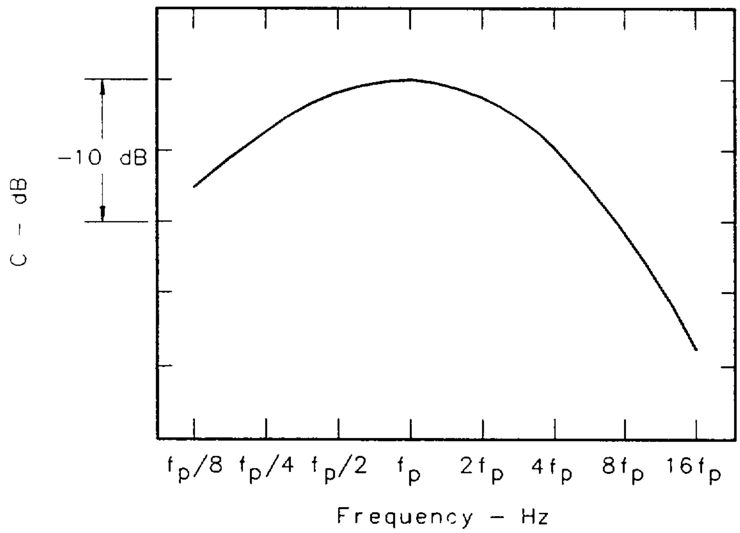


Figure 11-14 GENERALIZED 1/1 OCTAVE BAND SPECTRUM SHAPE ASSOCIATED WITH DIFFUSER NOISE

Equation 11-40 can also be used for generic slot diffusers that do not have special plenum or damper systems. For rectangular slot diffusers, S and u in Equation 11-37 are the cross-section area and flow velocity just prior to the slots. The 1/1 octave band sound power levels associated with generic diffusers are given by

Equation 11-42

$$L_W = 10 \log_{10}[S] + 30 \log_{10}[\xi] + 60 \log_{10}[u] - 31.3 + C$$

The sound power levels predicted by Equation 11-42 usually yield NC levels that are within 5 points of corresponding levels that are published by manufacturers when an 8 to 10 dB room correction is applied to each 1/1 octave band to convert from sound power levels to corresponding sound pressure levels in the room.

The method for determining the sound power levels associated with generic diffusers described above does not apply to diffusers that have specially designed plenum and damper systems. When this is the case, the sound power levels of a diffuser can be estimated by using the manufacturer's published NC levels for a specified diffuser system and the related pressure drop, ΔP, and flow velocity, u, associated with the point of operation of the diffuser. The flow velocity, u, and corresponding peak frequency, f_p, are determined as described above. The curve in Figure 11-14 is shifted such that f_p corresponds to the 1/1 octave frequency band which contains f_p. Position the curve such that it is tangent to the NC curve that corresponds to the NC level published by the manufacturer for the specified point of operation. Read the related 1/1 octave band sound pressure levels. Finally, add 10 dB to all of the 1/1 octave band sound pressure levels to obtain the 1/1 octave band sound power levels of the diffuser.

Example 11-12

A rectangular diffuser has the following duct dimensions prior to the diffuser: 12 in. × 16 in. The volume flow rate is Q = 1200 ft^3/min and the pressure drop across the diffuser is $\Delta P = 0.3$ in H_2O. Determine the 1/1 octave band sound power levels associated with the diffuser.

Solution

The cross-section area, S, and flow velocity, u, are

$$S = \frac{12 \times 16}{144} = 1.33 \text{ ft}^2$$

$$u = \frac{1200}{60 \times 1.33} = 15 \text{ ft/sec}$$

The normalized pressure drop coefficient, ξ, is

$$\xi = \frac{334.9 \times 0.3}{0.0749 \times 15^2} = 5.96$$

The overall sound power level is

$$L_{W(overall)} = 10\log_{10}[1.33] + 30\log_{10}[5.96] + 60\log_{10}[15] - 31.3 = 68.1$$

The frequency f_p is

$$f_p = 48.8 \times 15 = 732 \text{ Hz}$$

732 Hz is between 710 Hz and 1420 Hz. Thus, II = 5. The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
A	−4	−3	−2	−1	0	1	2	3
$L_{W(overall)}$, dB	63.8	63.8	63.8	63.8	63.8	63.8	63.8	63.8
C, dB (Eq. 11-40)	−29.3	−21.5	−16.0	−12.8	−11.8	−13.1	−16.6	−22.4
L_W, dB	34.5	42.3	47.8	51.0	52.0	50.7	47.2	41.4

H DUCT SOUND BREAKOUT AND BREAKIN

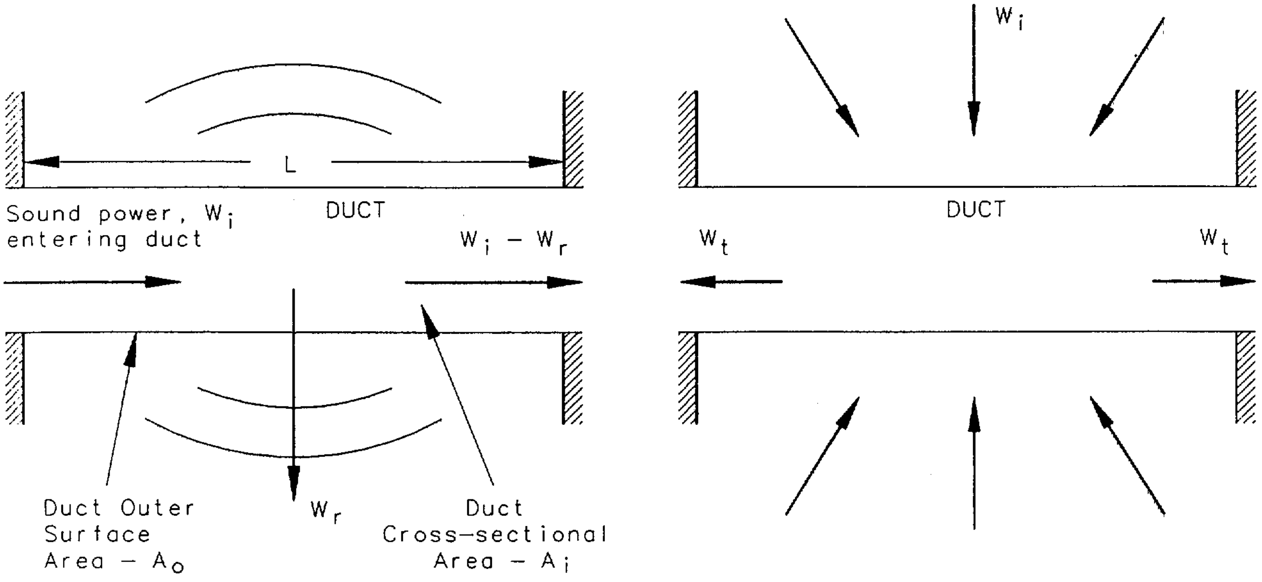
The analytical procedures discussed in this section include: sound breakout and breakin of rectangular ducts, sound breakout and breakin of circular ducts, sound breakout and breakin of flat-oval ducts, and insertion loss of externally lagged rectangular ducts.

1. Sound Breakout and Breakin

Noise that is generated within a duct and then transmitted through the duct wall into the surrounding area is called "breakout" [Figure 11-15(a)]. This phenomenon is often referred to as low-frequency duct rumble. There are two possible sources for duct breakout. One is associated with noise that is generated within the duct, usually by a fan. This noise, designated W_i in Figure 11-15(a), is transmitted down the duct and then through the duct walls into surrounding spaces. The transmitted sound is designated W_r in Figure 11-15(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 into the surrounding spaces. In many situations, particularly near fan discharge sections, duct breakout may be associated with both of these sources.

Noise that is transmitted into a duct from the surrounding area and then transmitted within the duct is called "breakin" [Figure 11-15(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.

The breakout transmission loss, TL_{out} (dB), of a duct is given by



(a) Duct Breakout

(b) Duct Breakin

Figure 11-15 DUCT SOUND BREAKOUT AND BREAKIN

Equation 11-43

$$TL_{out} = 10 \log_{10} \left[\frac{W_i}{A_i} \frac{A_O}{W_r} \right]$$

where W_i is the sound power (watts) in the duct, W_r is the sound power (watts) radiated from the duct, A_i is the cross sectional area (in^2) of the inside of the duct, and A_o is the sound radiation surface area (in^2) of the outside of the duct. Rearranging Equation 11-43 yields

Equation 11-44

$$L_{wr} = L_{wi} + 10 \log_{10} \left[\frac{A_O}{A_I} \right] - TL_{out}$$

where L_{wr} (dB) and L_{wi} (dB) are given by

Equation 11-45

$$L_{wr} = 10 \log_{10} \left[\frac{W_r}{10^{-12}} \right]$$

Equation 11-46

$$L_{wi} = 10 \log_{10} \left[\frac{W_i}{10^{-12}} \right]$$

The breakin transmission loss, TL_{in} (dB), associated with ducts is given by

Equation 11-47

$$TL_{in} = 10 \log_{10} \left[\frac{W_i}{2\,W_t} \right]$$

where W_i is the incident sound power (watts) on the duct from the surrounding space and W_t is the sound power (watts) that travels along the duct both upstream and downstream from the point where the sound enters the duct. The sound power level of the sound transmitted into the duct is obtained by rearranging Equation 11-47, or

Equation 11-48

$$L_{wt} = L_{wi} - TL_{in} - 3$$

where L_{wi} is given by equation 11-46 and L_{wt} is given by

Equation 11-49

$$L_{wt} = 10 \log_{10} \left[\frac{W_t}{10^{-12}} \right]$$

2. Rectangular Ducts

If the duct is a rectangular duct, A_i and A_o in Equations 11-43 and 11-44 are given by

Equation 11-50

$$A_i = a \times b$$

Equation 11-51

$$A_o = 24 \times L \times (a + b)$$

where a is the larger duct cross-section dimension (in), b is the smaller duct cross-section dimension (in), and L is the exposed length (ft) of the duct [Figure 11-15(a)]. For rectangular ducts, the breakout transmission loss curve shown in Figure 11-16 can be divided into two regions: (1) a region where plane mode transmission within the duct is dominant and (2) a region where multi-mode transmission is dominant. The frequency, f_L, that divides these two regions is given by

Equation 11-52

$$f_L = \frac{24134}{\sqrt{a \times b}}$$

If $f < f_L$, the plane mode predominates and TL_{out} is given by

Equation 11-53

$$TL_{out} = 10 \log_{10} \left[\frac{f \times q^2}{a + b} \right] + 17$$

where f is frequency (Hz), q is the mass/unit area (lb/ft^2) of the duct walls, and a and b are as described above. If $f \geq f_L$, multi-mode transmission predominates and TL_{out} is calculated from

Equation 11-54

$$TL_{out} = 20 \log_{10} [a \times b] - 31$$

where q and f are as specified above. The minimum value of TL_{out} occurs when $W_i = W_r$ and is specified by

Equation 11-55

$$TL_{out}\,(min) = 10 \log_{10} \left[24 \times L \times \left(\frac{1}{a} + \frac{1}{b} \right) \right]$$

Table 14-41 in Chapter 14 shows some values of TL_{out} calculated using the above equations.

The breakin transmission loss can be divided into two regions which are separated by a cutoff frequency f_1. The cutoff frequency is the frequency for the lowest acoustic cross-mode in the duct. It is given by

Equation 11-56

$$f_1 = \frac{6764}{a}$$

If $f \leq f_1$,

Equation 11-57a and 11-57b

$$TL_{in} = \text{larger of} \begin{cases} TL_{out} - 4 - 10 \log_{10} \left[\frac{a}{b} \right] + 20 \log_{10} \left[\frac{f}{f_1} \right] \\ 10 \log_{10} \left[12 \times L \times \left(\frac{1}{a} + \frac{1}{b} \right) \right] \end{cases}$$

If $f > f_1$

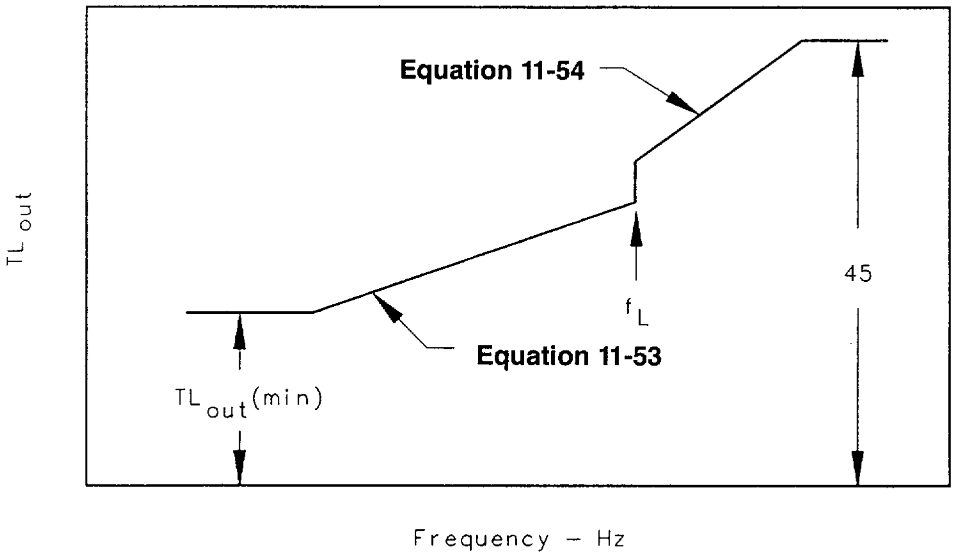


Figure 11-16 TL_{out} ASSOCIATED WITH RECTANGULAR DUCTS

Equation 11-58

$$TL_{in} = TL_{out} - 3$$

Table 14-42 shows some values of TL_{in} calculated using the above equations.

Example 11-13

Determine the breakout and breakin sound power for a duct with the following dimensions: smaller duct dimension—12 inches; larger duct dimension—24 inches; duct length—20 feet. The duct is constructed of 24 gauge sheet metal. q = mass/unit area of 24 gauge sheet metal = 1.0 lb/ft².

Solution

Sound breakout:

$$A_i = 24 \times 12 = 288 \text{ in}^2$$

$$A_o = 24 \times 20 \times (24 + 12) = 17{,}280 \text{ in}^2$$

$$f_L = \frac{24134}{\sqrt{24 \times 12}} = 1{,}422 \text{ Hz}$$

$$TL_{out}\,(\min) = 10 \log_{10} \left[24 \times 20 \times \left(\frac{1}{24} + \frac{1}{12}\right)\right] = 17.8 \text{ dB}$$

$$10 \log_{10}\left[\frac{A_o}{A_i}\right] = 10 \log_{10}\left[\frac{17280}{288}\right] = 17.8 \text{ dB}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
Eq. 11-53	19.4	22.4	25.4	28.4	31.4			
Eq. 11-54						35.0	41.0	45.0
TL_{out}, dB	19.4	22.4	25.4	28.4	31.4	35.0	41.0	45.0
10 $Log_{10}[A_o/A_i]$, dB	17.8	17.8	17.8	17.8	17.8	17.8	17.8	17.8
$L_{wr} - L_{wi}$, dB	−1.6	−4.6	−7.6	−10.6	−13.6	−17.2	−23.2	−27.2

Sound breakin:

$$f_1 = \frac{6764}{24} = 282 \text{ Hz}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
TL_{out}, dB	19.2	22.4	25.4	28.4	31.4	35.0	41.0	45.0
Eq. 11-57a	−1.0	8.3	17.3					
Eq. 11-57b	14.8	14.8	14.8					
	14.8	14.8	17.3					
				−3.0	−3.0	−3.0	−3.0	−3.0
TL_{in}, dB	14.8	14.8	17.3	25.4	28.4	32.0	38.0	42.0
	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0
$L_{wt} - L_{wi}$, dB	−17.8	−17.8	−20.3	−28.4	−31.4	−35.0	−41.0	−45.0

3. Round Ducts

If the duct is round, A_i and A_o in Equations 11-43 and 11-44 are given by

Equation 11-59

$$A_i = \frac{\pi d^2}{4}$$

Equation 11-60

$$A_o = 12 \pi \times d \times L$$

where d is the duct diameter (inches) and L is the exposed length (feet) of the duct. Narrow band and 1/3 octave band breakout transmission loss values for round ducts are very hard to predict and no simple prediction techniques are available. However, if the analysis is limited to 1/1 octave frequency bands, TL_{out} associated with round ducts can be approximated by a curve similar to the one shown in Figure 11-17. Table 14-43 shows experimentally obtained TL_{out} data for round ducts. If the breakout analysis is limited to 1/1 octave band values, Equations 11-61 and 11-62 can be used to approximate the data in Table 14-43.

Equation 11-61

$$TL_1 = 17.6 \log_{10}[q] - 49.8 \log_{10}[f] - 55.3 \log_{10}[d] + C_o$$

Equation 11-62

$$TL_2 = 17.6 \log_{10}[q] - 6.6 \log_{10}[f] - 36.9 \log_{10}[d] + 97.4$$

where q is the mass/unit area (lb/ft^2) of the duct wall, f is frequency (Hz), d is the inside duct diameter (inches), and

C_o = 230.4 for long seam ducts
C_o = 232.9 for spiral wound ducts

Equation 11-63

$$TL_{out} = \text{the larger of } TL_{1,2}$$

The above equations yield good results except when the diameter of the duct is equal to or greater than 26 inches and the 1/1 octave band center frequency is equal to 4000 Hz. For this special case TL_{out} is given by

Equation 11-64

$$TL_{out} = 17.6 \log_{10}[q] - 36.9 \log_{10}[d] + 90.6$$

The maximum allowable value for TL_{out} is 50 dB. Thus, if the value for TL_1 obtained from equation 11-63 exceeds 50 dB, the value should be set equal to 50 dB. Table 14-44 lists the calculated values for TL_{out}.

For calculating the breakin transmission loss for round ducts, the cut-off frequency for the lowest acoustic cross-mode is given by

Equation 11-65

$$f_1 = \frac{7929}{d}$$

If $f \leq f_1$,

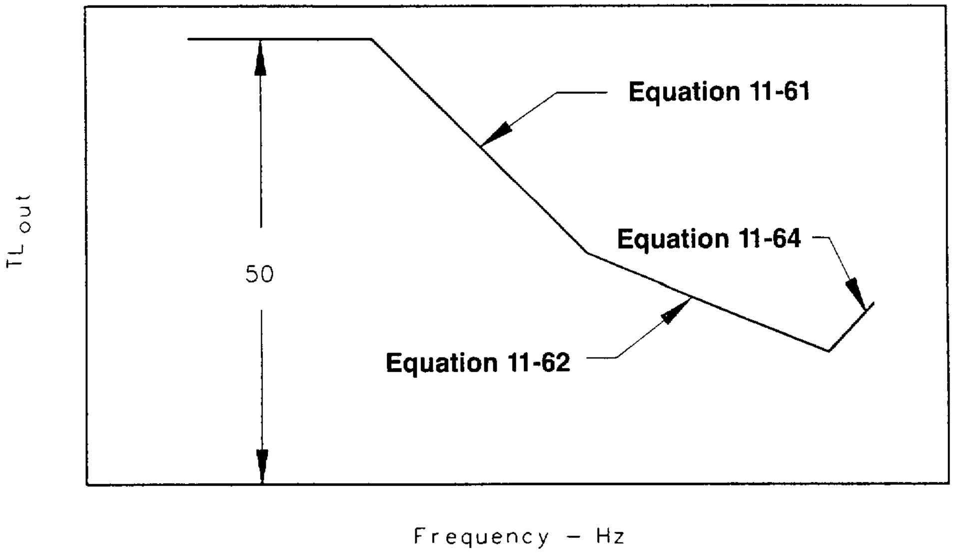


Figure 11-17 TL_{out} ASSOCIATED WITH ROUND DUCTS

Equation 11-66a and 11-66b

$$TL_{in} = \text{larger of} \begin{cases} TL_{out} - 4 + 20 \log_{10}\left[\dfrac{f}{f_1}\right] \\ 10 \log_{10}\left[\dfrac{2\,L}{d}\right] \end{cases}$$

If f > f_1, the breakin transmission loss is defined by

Equation 11-67

$$TL_{in} = TL_{out} - 3$$

Table 14-45 in Chapter 14 gives values for the breakin transmission loss for various duct sizes obtained from experimental data. Table 14-46 gives the corresponding values calculated using the above equations.

Example 11-14

Determine the breakout and breakin sound power of a long seam round duct given the following information: diameter—14 inches; length—15 feet. The duct is constructed of 24 gauge sheet metal.

Solution

q = mass/unit area of 24 ga sheet metal
$= 1.0\ \text{lb}_m/\text{ft}^2$

Sound breakout:

$$A_i = \pi \frac{14^2}{4} = 153.9\ \text{in}^2$$

$$A_o = 12 \times 15 \times \pi \times 14 = 7{,}916.8\ \text{in}^2$$

$$10 \log_{10}\left[\frac{A_o}{A_i}\right] = 10 \log_{10}\left[\frac{7{,}916.8}{153.9}\right] = 17.1\ \text{dB}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
TL_1, dB	77.6	62.6	47.6	32.6	17.6	2.6	−12.4	
TL_2, dB	43.3	41.3	39.3	37.3	36.3	33.3	31.3	
TL_{max}, dB	50.0	50.0						
TL_{out}, dB	50.0	50.0	47.6	37.3	35.3	33.3	31.3	
10 $Log_{10}[A_0/A_i]$, dB	17.1	17.1	17.1	17.1	17.1	17.1	17.1	
$L_{wr} - L_{wi}$, dB	−32.9	−32.9	−30.5	−20.2	−18.2	−16.2	−14.2	

Sound breakin:

$$f_1 = \frac{7929}{14} = 566.4\ \text{Hz}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
TL_{out}, dB	45.0	45.0	45.6	37.3	35.3	33.3	31.3	
Eq. 11-66a	26.9	32.9	36.5	32.2				
Eq. 11-66b	14.1	14.1	14.1	14.1				
	26.9	32.9	36.5	32.2	−3.0	−3.0	−3.0	
TL_{in}, dB	26.9	32.9	36.5	32.2	32.3	30.3	28.3	
	3.0	3.0	3.0	3.0	3.0	3.0	3.0	
$L_{wt} - L_w$, dB	−29.9	35.9	−39.5	−35.2	−35.3	−33.3	−31.3	

4. Flat Oval Ducts

If the duct is a flat oval duct, A_i and A_o in Equations 11-43 and 11-44 are given by

Equation 11-68

$$A_i = b \times (a - b) + \frac{\pi b^2}{4}$$

Equation 11-69

$$A_o = 12 \times L \times [2 \times (a - b) + \pi\, b]$$

Equation 11-70

$$P = 2 \times (a - b) + \pi\, b$$

where a is the length (inches) of the major duct axis, b is the length (inches) of the minor duct axis, L is the duct length (feet), A_i is the cross-section area (in^2), A_o is the surface area of the outside of the duct (in^2), and P is the perimeter of the duct in inches (Figure 11-18). The fraction of the perimeter taken up by the flat sides, σ, is given by

Equation 11-71

$$\sigma = \frac{1}{\left[1 + \dfrac{\pi\, b}{2\,(a - b)}\right]}$$

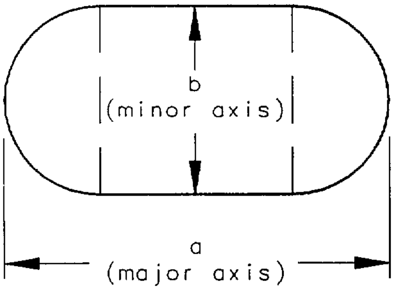


Figure 11-18 FLAT OVAL DUCT

The minimum breakout transmission loss, TL_{out} (min) (dB), or flat oval ducts is given by

Equation 11-72

$$TL_{out} = 10 \log_{10} \left[\frac{A_o}{A_i} \right]$$

The low-to-mid frequency transmission loss, TL_{out} (dB), associated with flat oval ducts is specified by

Equation 11-73

$$TL_{out} = 10 \log_{10} \left[\frac{q^2 \times f}{\sigma^2 \times p} \right] + 20 \text{ dB}$$

The upper frequency limit, f_L (Hz), of applicability of Equation 11-73 is

Equation 11-74

$$f_L = \frac{8115}{b}$$

Table 14-47 in Chapter 14 gives some values of TL_{out} for flat oval ducts of various sizes.

As was the case with rectangular and circular ducts, TL_{in} can be written in terms of TL_{out}. While there are no exact solutions for the cut-off frequency for the lowest acoustic cross-mode in flat oval ducts, Equation 11-75 gives an approximate solution.

Equation 11-75

$$f_1 = \frac{6764}{(a - b) \times \left[1 + \frac{\pi b}{2 (a - b)} \right]^{1/2}}$$

where a and b are in inches. This equation is valid when $a/b \geq 2$. When $a/b < 2$, the accuracy of Equation 11-75 deteriorates progressively as a/b approaches unity. When $f \leq f_1$, TL_{in} is given by

Equation 11-76a and 11-76b

$$TL_{in} = \text{the larger of} \begin{cases} TL_{out} + 10 \log_{10}[f^2 \times A_i] - 81 \\ 10 \log_{10} \left[\frac{6 P \times L}{A_i} \right] \end{cases}$$

When $f > f_1$, TL_{in} is given by

Equation 11-77

$$TL_{in} = TL_{out} - 3$$

Table 14-48 gives TL_{in} values for the duct sizes listed in Table 14-47.

Example 11-15

Determine the breakout and breakin sound power of a flat oval duct given the following information: major axis—24 inches; minor axis—6 inches; length—20 feet. The duct is constructed of 24 gauge sheet metal.

Solution

q = mass/unit area of 24 ga. sheet metal
= 1.0 lb_m/ft^2

Sound breakout:

$$A_i = 6 \times (24 - 6) + \frac{\pi 6^2}{4} = 136.3 \text{ in}^2$$

$$A_o = 12 \times 20 \times [2 \times (24 - 6) + \pi 6] = 13{,}163.9 \text{ in}^2$$

$$P = 2 \times (24 - 6) + \pi 6 = 54.9 \text{ in}$$

$$\sigma = \frac{1}{1 + \frac{\pi 6}{2 \times (24 - 6)}} = 0.656$$

$$TL_{out} \text{ (min)} = 10 \log_{10} \left[\frac{13163.9}{136.3} \right] = 19.8 \text{ dB}$$

$$f_L = \frac{8115}{6} = 1{,}352.5 \text{ Hz}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
TL_{out}, dB	24.2	27.2	30.2	33.3	36.3	—	—	—
10 $Log_{10}[A_o/A_i]$, dB	19.8	19.8	19.8	19.8	19.8			
$L_{wr} - L_{wi}$, dB	−4.4	−7.4	−10.4	−13.4	−16.5	—	—	—

Sound breakin:

$$f_1 = \frac{6764}{(24 - 6) \times \left[1 + \frac{\pi 6}{2 \times (24 - 6)} \right]^{1/2}} = 304.4 \text{ Hz}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
TL_{out}, dB	24.2	27.2	30.2	33.3	36.3	—	—	—
Eq. 11-76a	0.5	9.5	18.5	—	—	—	—	—
Eq. 11-76b	16.8	16.8	16.8	—	—	—	—	—
	16.8	16.8	16.8	−3.0	−3.0	—	—	—
TL_{in}, dB	16.8	16.8	18.5	30.3	33.3	—	—	—
	3.0	3.0	3.0	3.0	3.0	—	—	—
$L_{wt} - L_{wi}$, dB	−19.8	−19.8	−21.5	−33.3	−36.3	—	—	—

5. Insertion Loss of External Duct Lagging

External acoustic lagging is often applied to rectangular ductwork 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 (Figure 11-19). 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. For both techniques, it is necessary to determine the low frequency insertion loss, IL(lf) (dB). It is given by

Equation 11-78

$$IL(lf) = 20\log_{10}\left[1 + \frac{M_2}{M_1}\frac{P_1}{P_2}\right]$$

where P_1 is the perimeter of the duct (inches), P_2 is the perimeter of the outer covering (inches), M_1 is the mass per unit area of the duct (lb/ft^2), and M_2 is the mass per unit area of the outer covering (lb/ft^2). P_1 and P_2 are specified by

Equation 11-79

$$P_1 = 2\,(a + b)$$

Equation 11-80

$$P_2 = 2\,(a + b + 4\,h)$$

where a is the duct width (inches), b is the duct height (inches), and h is the thickness (inches) of the soft, flexible, porous material between the duct wall and the outer covering.

If a rigid outer covering is used, it is necessary to determine the resonance frequency, f_r (Hz), associated with the interaction between the duct wall and outer covering. f_r is given by

Equation 11-81

$$f_r = 156\left[\left(\frac{P_2}{P_1} + \frac{M_2}{M_1}\right) \times \frac{P_1}{M_2} \times S\right]^{1/2}$$

where M_1, M_2, P_1, and P_2 are as previously defined. S is the cross-section area (in^2) of the absorbent material and is given by

Equation 11-82

$$S = 2\,h \times (a + b + 2\,h)$$

The following procedures for determining the insertion loss for external duct lagging should be used for rigid and limp outer coverings.

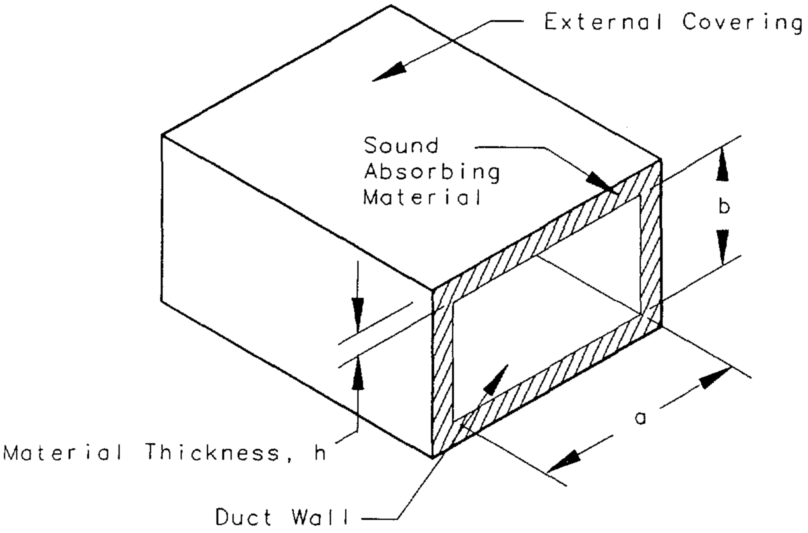


Figure 11-19 EXTERNAL DUCT LAGGING ON RECTANGULAR DUCTS

a. RIGID COVERING MATERIALS

If 1/3 octave band values are desired, draw a line from point B (0.71 f_r) to point A (f_r) on Figure 11-20(a). The difference in IL (dB) between points B and A is 10 dB. The equation for this line is

Equation 11-83

$$IL = IL(lf) - 67.23 \log_{10}\left[\frac{f}{0.71\ f_r}\right]$$

Next draw a line from point A (f_r) to point C (1.41 f_r) on Figure 11-20(a). The equation for this line is

Equation 11-84

$$IL = IL(lf) - 10 + 67 \log_{10}\left[\frac{f}{f_r}\right]$$

From point C (1.41 f_r), draw a line with a slope of 9 dB/octave. The equation for this line is

Equation 11-85

$$IL = IL(lf) + 29.9 \log_{10}\left[\frac{f}{1.41\ f_r}\right]$$

If 1/1 octave band values are desired, use Equation 11-78 for the 1/1 octave bands below the one that contains f_r. For the 1/1 octave band that contains f_r, subtract 5 dB from IL(lf) obtained from Equation 11-78. For the 1/1 octave bands above the one that contains f_r, use Equation 11-85.

b. LIMP COVERING MATERIALS

Since there is no pronounced resonance with limp covering materials, the low frequency insertion loss, IL(lf), is valid up to f_r, after which the insertion loss increases at a rate of 9 dB per octave [Figure 11-20(b)]. For frequencies above f_r, the equation for insertion loss is

Equation 11-86

$$IL = IL(lf) + 29.90 \log_{10}\left[\frac{f}{f_r}\right]$$

The insertion loss of duct lagging probably does not exceed 25 dB.

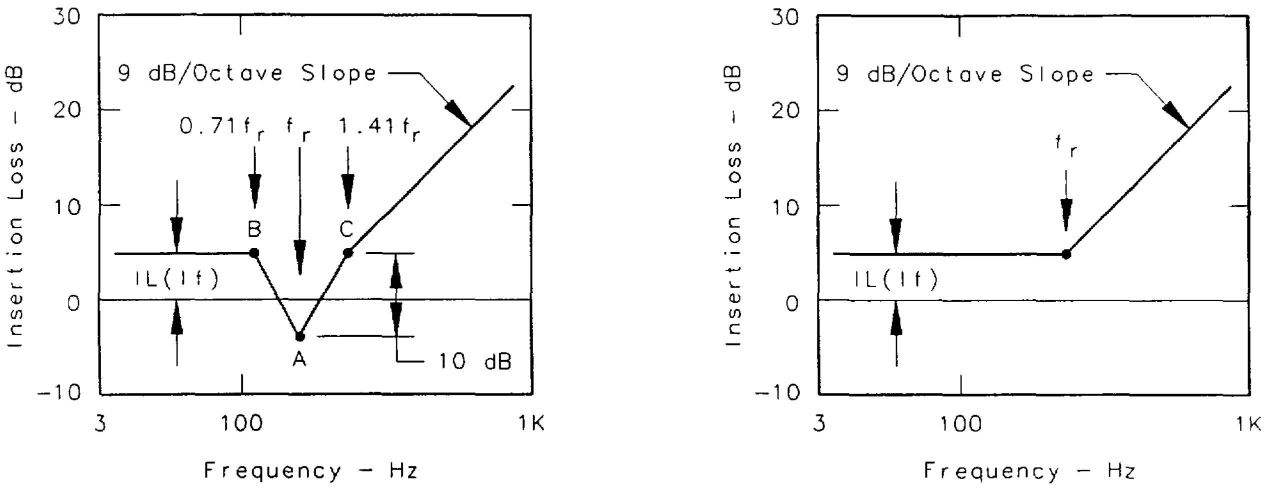
The insertion loss predictions using the procedures described above should be fairly accurate up to about 1,000 Hz for most ducts. Duct lagging may not be a particularly effective method for reducing low frequency (<100 Hz) duct sound breakout. A more effective method for reducing duct breakout is the use of round ductwork, which has a high transmission loss at low frequencies.

Example 11-16

Determine the 1/1 octave band insertion loss associated with the external lagging of a rectangular sheet metal duct with the following characteristics: duct dimensions—8 in × 8 in; duct constructed of 18 gauge sheet metal; thickness of absorbent material—1 inch; outer covering—1/2 inch gypsum board.

Solution

M_1 = mass/unit area of 18 gauge sheet metal
= 2.0 lb/ft^2



(a) Rigid Outer Covering (b) Limp Outer Covering

Figure 11-20 INSERTION LOSS ASSOCIATED WITH RECTANGULAR EXTERNAL DUCT LAGGING

M_2 = mass/using area of one sheet of 1/2 inch gypsum board = 2.1 lb/ft^2

$P_1 = 2(8 + 8) = 32$ inches

$P_2 = 2(8 + 8 + 4 \times 1) = 40$ inches

Thus,

$$IL(If) = 20 \log_{10}\left[1 + \frac{2.1}{2.0}\frac{32}{40}\right] = 5.3 \text{ dB}$$

$$S = 2 \times 1 \times (8 + 8 + 2 \times 1) = 36 \text{ in}^2$$

The resonance frequency is

$$f_r = 156\left[\left(\frac{40}{32} + \frac{2.1}{2.0}\right) \times \frac{32}{2.1} \times 36\right] = 154 \text{ Hz}$$

154 Hz is in the 125 Hz 1/1 octave band. Equation 11-86 can be written

$$IL = IL(If) + 29.9 \log_{10}\left[\frac{f}{1.41 \times 154}\right]$$

The results are summarized below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
IL(If), dB	5.3	5.3 −5.0						
IL (9dB/octave), dB			7.1	16.1	25.1			
Max IL value, dB					25.0	25.0	25.0	25.0
IL, dB	5.3	0.3	7.1	16.1	25.0	25.0	25.0	25.0

J DUCT ELEMENT SOUND ATTENUATION

The duct elements covered in this section include: sound plenums, unlined rectangular ducts, acoustically lined rectangular ducts, unlined circular ducts, acoustically lined circular ducts, elbows, acoustically lined circular radiused elbows, duct silencers, duct branch power division, and duct end reflection loss.

1. Plenum Chambers

The plenum chamber is usually placed between the discharge section of a fan and the main duct of the air distribution system. These chambers are usually lined with acoustically absorbent material to reduce fan and other types of noise. Plenum chambers are usually large rectangular enclosures with an inlet and one or more outlet sections. The transmission loss associated with a plenum chamber can be expressed as

Equation 11-87

$$TL = -10 \log_{10}\left[S_{out}\left(\frac{Q \cos \Theta}{4 \pi r^2} + \frac{1 - \alpha}{S \alpha_A}\right)\right]$$

Referring to Figure 11-21, S_{out} is the area (ft^2) of the output section of the plenum, S is the total inside surface area (ft^2) of the plenum minus the inlet and outlet areas, r is the distance (feet) between the centers of the inlet and outlet sections of the plenum, and α_A is the average absorption coefficient of the plenum lining. α_A is given by

Equation 11-88

$$\alpha_A = \frac{S_1\alpha_1 + S_2\alpha_2}{S}$$

where α_1 and S_1 are the sound absorption coefficient and corresponding surface area (ft^2) of any bare or unlined inside surfaces of the plenum chamber and α_2 and S_2 are the sound absorption coefficient and corresponding surface area (ft^2) of the acoustically lined inside surfaces of the plenum chamber. In many situations, 100 percent of the inside surfaces of a plenum chamber are lined with a sound absorbing material. For these situations, $\alpha_A = \alpha_2$.

Q in Equation 11-87 is the directivity factor which equals 2 if the inlet section is near the center of the side of the plenum on which it is located. This corresponds to the situation where sound from the inlet section of the plenum chamber is radiating into half space. Q equals 4 if the inlet section is located in the corner where two sides of the plenum come together. This corresponds to the situation where sound from the inlet section is radiating into quarter space.

Θ in Equation 11-87 is the angle of the vector representing r relative to the horizontal plane. cos Θ and r can be written

Equation 11-89

$$r = \sqrt{rh^2 + rv^2}$$

Equation 11-90

$$\cos \Theta = \frac{rh}{r}$$

where rh and rv are the horizontal and vertical distances (ft), respectively, between the inlet and outlet sections of the plenum (Figure 11-21).

Equation 11-87 treats a plenum as if it is a large enclosure. Thus, Equation 11-87 is valid only for the

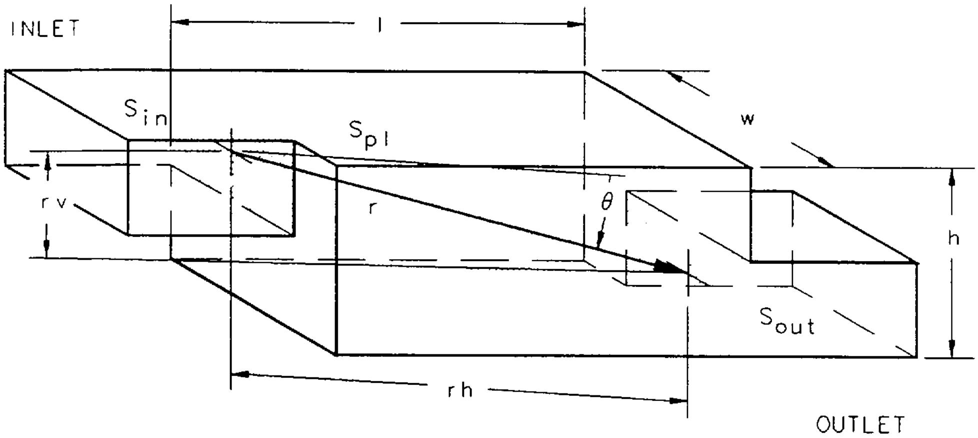


Figure 11-21 SCHEMATIC OF A PLENUM CHAMBER

case where the wavelength of sound is small compared to the characteristic dimensions of the plenum. For frequencies which correspond to plane wave propagation in the duct, the results predicted by Equation 11-87 are usually not valid. Plane wave propagation in a duct exists at frequencies below

Equation 11-91

$$f_{co} = \frac{c_o}{2a}$$

where c_o is the speed of sound in air (ft/sec) and a is the larger cross-section dimension (feet) of a rectangular duct, or below

Equation 11-92

$$f_{co} = 0.586 \frac{c_o}{d}$$

where d is the diameter (feet) of a circular duct. The cutoff frequency, f_{co}, is the frequency above which plane waves no longer propagate in a duct. At these higher frequencies the waves that propagate in the duct are referred to as cross or spinning modes. At frequencies below f_{co}, the plenum chamber can be treated as an acoustically lined expansion chamber. The equation for the transmission loss of an acoustically lined expansion chamber is

Equation 11-93

$$TL = 10 \log_{10} \left[\left(\cosh\left[\frac{\sigma l}{2}\right] + \frac{1}{2}\left[m + \frac{1}{m}\right] \sinh\left[\frac{\sigma l}{2}\right] \right)^2 \times \cos^2\left(\frac{2\pi \times f \times l}{c_o}\right) + \left(\sinh\left[\frac{\sigma l}{2}\right] + \frac{1}{2}\left(m + \frac{1}{m}\right) \cosh\left[\frac{\sigma l}{2}\right] \right)^2 \times \sin^2\left(\frac{2\pi \times f \times l}{c_o}\right) \right]$$

where σ is sound attenuation per unit length in the chamber (dB/ft), l is the horizontal length of the plenum chamber (feet), c_o is the speed of sound in air (ft/sec), f is frequency (Hz), and m is the ratio of the cross-sectional area of the plenum divided by the cross-sectional area of the inlet section of the plenum. m is given by

Equation 11-94

$$m = \frac{S_{pl}}{S_{in}} \text{ (refer to Figure 11-21).}$$

For frequencies less than f_{co}, the transmission loss of a plenum is given by Equation 11-93. For frequencies greater than or equal to f_{co}, the transmission loss of a plenum is given by Equation 11-87. f_{co} associated with Equations 11-91 and 11-92 is calculated on the bases of the inlet section of the plenum. Table 14-49 gives the absorption coefficients of typical plenum materials.

Equations for σl for the 1/1 octave frequency bands from 63 Hz to 500 Hz are:

Equation 11-95

$$63 \text{ Hz: } \sigma l = [0.00306 \times (P/A)^{1.959} \times t^{0.917}] \times l$$

Equation 11-96

$$125 \text{ Hz: } \sigma l = [0.01323 \times (P/A)^{1.410} \times t^{0.941}] \times l$$

Equation 11-97

$$250 \text{ Hz: } \sigma l = [0.06244 \times (P/A)^{0.824} \times t^{1.079}] \times l$$

Equation 11-98

$$500 \text{ Hz: } \sigma l = [0.23380 \times (P/A)^{0.500} \times t^{1.087}] \times l$$

where P/A is the perimeter (P) of the cross-section of the plenum chamber (feet) divided by the area (A or S_{pl}) of the cross-section of the plenum chamber (ft^2), t is the thickness of the fiberglass insulation (inches) used to line the inside surfaces of the plenum, and l is the length (feet) of the plenum chamber. Equation 11-87 will nearly always apply at frequencies of 1,000 Hz and above.

Example 11-17

A plenum chamber is 6 feet high, 4 feet wide, and 10 feet long. The configuration of the plenum is similar to that shown in Figure 11-21. The inlet is 36 inches wide by 24 inches high. The outlet is 36 inches wide by 24 inches high. The horizontal distance between centers of the plenum inlet and outlet is 10 feet. The vertical distance is 4 feet. The plenum is lined with 1 inch thick 3.0 lb/ft³ density fiberglass insulation board. 100% of the inside surfaces of the plenum are lined with the fiberglass insulation. Determine the transmission loss associated with this plenum. For this example, assume Q = 4.

The areas of the inlet section, outlet section, and plenum cross section are:

$$S_{in} = \frac{24 \times 36}{144} = 6 \text{ ft}^2$$

$$S_{out} = \frac{24 \times 36}{144} = 6 \text{ ft}^2$$

$$S_{pl} = 4 \times 6 = 24 \text{ ft}^2$$

The values of r and cos θ are:

$$r = \sqrt{6^2 + 4^2} = 7.75 \text{ ft}$$

$$\cos \theta = \frac{6}{7.75} = 0.77$$

The total inside surface area of the plenum is:

$$S = 2\,(4 \times 6) + 2\,(4 \times 6) + 2\,(6 \times 6) - 12$$
$$= 156 \text{ ft}^2$$

The values of P/A, m, and f_{co} are:

$$P/A = \frac{2\,(4 + 6)}{4 \times 6} = 0.83$$

$$m = \frac{24}{6} = 4$$

$$f_{co} = \frac{1125}{2 \times 3} = 187.5 \text{ Hz}$$

Thus, Equation 11-95 is used for the 63 Hz and 125 Hz 1/1 octave bands and Equation 11-87 is used for the 250 Hz through 4,000 Hz 1/1 octave bands. The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz						
	63	**125**	**250**	**500**	**1000**	**2000**	**4000**
σl	**0.0128**	**0.0614**					
m	**4**	**4**					
$2\,\pi \cdot f \cdot l/c_o$	2.111	4.222					
$Q\cos \cdot /4\,\pi\, r^2\ (\times 10^3)$			5.09	5.09	5.09	5.09	5.09
$(1 - \alpha_A)/S\,\alpha_A\ (\times 10^3)$			22.7	2.87	0.634	0.267	0.0647
TL, dB	5.6	5.8	7.8	13.2	14.6	14.9	15.1

2. Unlined Rectangular Ducts

Straight unlined rectangular sheet metal ducts provide a small amount of sound attenuation. At low frequencies, the attenuation is significant and it tends to decrease as frequency increases. The attenuation in unlined ducts in the 1/1 octave frequency bands from 63 Hz to 250 Hz can be approximated by

Equation 11-99

$$ATTN = 17.0 \times \left(\frac{P}{A}\right)^{-0.25} \times FREQ^{-0.85} \times L$$

$$\text{for } \frac{P}{A} \geq 3$$

Equation 11-100

$$ATTN = 1.64 \times \left(\frac{P}{A}\right)^{0.73} \times FREQ^{-0.58} \times L$$

$$\text{for } \frac{P}{A} < 3$$

where ATTN is the total attenuation (dB) in the unlined rectangular duct, P is the length of the duct perimeter (feet), A is the duct cross-sectional area (ft^2), FREQ is the 1/1 octave band center frequency (Hz), and L is the duct length (feet).

At frequencies above 250 Hz the attenuation can be approximated by

Equation 11-101

$$ATTN = 0.02 \times \left(\frac{P}{A}\right)^{0.8} \times L$$

Table 14-50 shows the tabulated results that correspond to Equations 11-99 through 11-101. If the rectangular duct is externally lined with fiberglass, multiply the results associated with Equation 11-99 or 11-100 by a factor of 2.

The attenuations values shown in Table 14-50 and the corresponding attenuation values predicted by Equations 11-99 through 11-101 apply only to rectangular sheet metal ducts that have gauge thicknesses that are selected according to SMACNA HVAC duct construction standards.

Example 11-18

A straight section of unlined rectangular duct has the following dimensions: Height = 18 inches, width = 12 inches, and length = 20 feet. Determine the total sound attenuation in dB.

Solution

$$\frac{P}{A} = \frac{2\,(12 + 18) \times 12}{(12 \times 18)} = 3.333 \text{ in/ft}$$

The tabulated results are shown below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
Eq. 11-99, db/ft	0.37	0.21	0.12					
Eq. 11-100, db/ft				0.05	0.05	0.05	0.05	0.05
Duct length ft	×20	×20	×20	×20	×20	×20	×20	×20
Total Atten, dB	7.4	4.2	2.4	1.0	1.0	1.0	1.0	1.0

3. Acoustically Lined Rectangular Ducts

Fiberglass internal duct lining for rectangular sheet metal ducts can be used to attenuate sound in ducts and to thermally insulate ducts. The thickness of duct linings associated with thermal insulation usually varies from 0.5 inches to 2.0 inches. For fiberglass duct lining to be effective for attenuating sound, it must have a minimum thickness of 1.0 inch.

The regression equation for insertion loss in acoustically lined rectangular ducts is

Equation 11-102

$$IL = B \times \left(\frac{P}{A}\right)^{C} \times t^{D} \times L$$

where IL is the insertion loss (dB); P/A is the perimeter divided by the cross-sectional area of the free area inside the duct (1/ft); B, C and D are regression constants that are a function of the 1/1 octave band center frequency; t is lining thickness (inches); and L is duct length (feet). The values for B, C and D are given in Table 14-51 for 1/1 octave band center frequencies from 63 Hz to 8,000 Hz. Tables 14-52 and 14-53 give tabulated values of selected rectangular sheet metal ducts for 1 inch and 2 inch duct lining respectively.

With respect to Equation 11-102, the P/A values in unit of 1/ft of the ducts tested ranged from 1.1667 to 6; the thickness of the fiberglass duct lining was either 1 inch or 2 inches, and the density of the fiberglass duct liner ranged from 1.5 to 3.0 lb/ft^3. Caution must be exercised when extrapolating the values of insertion loss beyond the range of the parameters associated with the data used to obtain Equation 11-102. The insertion loss values predicted by Equation 11-102 are valid only for 1/1 octave frequency bands. The regression analyses indicated that for the samples tested, the insertion loss of acoustically lined rectangular sheet metal ducts is not a function of the density of the fiberglass lining when the density of the material is between 1.5 and 3.0 lb/ft^3. At 1/1 octave band center frequencies of 1,000 Hz and above, the insertion loss is not a function of lining thickness.

The insertion loss described by Equation 11-102 is the difference in the sound pressure level measured in a reverberation chamber with sound propagating through an unlined section of rectangular duct minus the corresponding sound pressure level that is measured when the unlined section of rectangular duct is replaced with a similar section of acoustically lined rectangular duct. As was mentioned in the section on unlined rectangular ducts, the sound attenuation associated with unlined rectangular duct can be significant at low frequencies. This attenuation is, in effect, subtracted out during the process of calculating the insertion loss from measured data. Even though it is not known for certain at this time, it is believed that this attenuation should be added to the insertion loss of correspondingly sized acoustically lined rectangular ducts to obtain the total sound attenuation of acoustically lined rectangular ducts. The sound attenuation, ATTN, in unlined rectangular ducts for the 1/1 octave band center frequencies from 63 Hz to 250 Hz is given by Equations 11-99 and 11-100. For 1/1 octave band center frequencies above 250 Hz, the sound attenuation, ATTN, is given by Equation 11-101. The total sound attenuation, ATTN(T), in acoustically lined rectangular ducts is obtained from

Equation 11-103

$$ATTN(T) = ATTN + IL$$

Because of structure-borne sound that is transmitted in and through the duct wall, the total sound attenuation in lined rectangular sheet metal ducts usually does not exceed 40 dB. Thus, the maximum allowable sound attenuation in Equation 11-103 is 40 dB. Insertion loss and attenuation values obtained from Equations 11-102 and 11-103 apply only to rectangular sheet metal ducts that have gauge thicknesses that are selected according to SMACNA "HVAC Duct Construction Standards."

Example 11-19

A straight section of acoustically lined rectangular duct has the following free inside dimensions: height = 24 inches, width = 36 inches, length = 10 feet. The duct is lined with 1 inch thick 1.5 lb/ft^3 fiberglass duct liner. Determine the total sound attenuation in the 10 foot section of acoustically lined rectangular duct.

Solution

$$\frac{P}{A} = \frac{2\ (24 + 36) \times 12}{(24 \times 36)} = 1.667\ 1/\text{ft}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
Eq. 11-102, dB	0.4	1.2	4.1	13.1	25.2	21.0	19.1	17.7
Eq. 11-100, dB	2.2	1.4	0.9					
Eq. 11-101, dB				0.3	0.3	0.3	0.3	0.3
ATTN(T), dB	2.6	2.6	5.0	13.4	25.5	21.3	19.4	18.0

4. Unlined Round Ducts

As with unlined rectangular ducts, unlined round ducts provide some sound attenuation which should be taken into account when designing a duct system. In contrast with rectangular ducts, round ducts are much more rigid and, therefore, do not resonate or absorb as much sound energy. Because of this, round ducts will only provide about 1/10th the sound attenuation at low frequencies as compared to the sound attenuation associated with rectangular ducts. Table 14-54 lists sound attenuation values for unlined round ducts.

Example 11-20

A straight unlined round duct has the following dimensions: diameter = 12 inches; length = 20 feet. Determine the total attenuation in dB.

Solution

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
Table 11-19, dB/ft Length, feet	0.03 ×20	0.03 ×20	0.03 ×20	0.05 ×20	0.07 ×20	0.07 ×20	0.07 ×20	
Total Atten., dB	0.6	0.6	0.6	1.0	1.4	1.4	1.4	

5. Acoustically Lined Round Ducts

There are very little data available in the literature with regard to the insertion loss of acoustically lined round ducts. The data that are available are usually manufacturer's product data. A regression equation was developed using measured insertion loss data for round ducts. The data were obtained for spiral dual-wall round ducts. The acoustical lining was a 0.75 lb/ft^3 density fiberglass blanket which ranged in thickness from one to three inches. The fiberglass was covered with an internal liner of perforated galvanized steel that had an open area of 25 percent. The inside duct diameters tested ranged from 6 to 60 inches. The equation is

Equation 11-104

$$IL = [A + (B \times t) + (C \times t^2) + (D \times d) + (E \times d^2) + (F \times d^3)] \times L$$

where IL is insertion loss (dB), t is the lining thickness (inches), d is the inside duct diameter (inches), and L is the duct length (feet). The coefficients for Equation 11-104 for each of the 1/1 octave frequency bands are given in Table 14-55. Equation 11-104 should not be extrapolated beyond the range of the data used to develop the equation. At frequencies between 63 Hz and 500 Hz, the insertion loss is a function of both duct diameter and lining thickness. At frequencies of 1,000 Hz and above, the insertion loss is a function of only duct diameter. The sound attenuation of unlined circular ducts is generally negligible. Thus, it is not necessary to include it when calculating the total sound attenuation of lined circular ducts. Because of structure-borne sound that is transmitted through the duct wall, the total sound attenuation of lined circular ducts usually does not exceed 40 dB. Tables 14-56, 14-57, and 14-58 give the insertion loss values for dual-wall circular sheet metal ducts with 1 inch, 2 inch and 3 inch acoustical lining respectively.

Example 11-21

Determine the sound attenuation in dB through a circular duct that has an inside diameter of 24 inches and a one inch thick fiberglass lining. Assume the duct lining has a density of 0.75 lb/ft^3. The fiberglass lining is covered with an internal perforated galvanized steel liner that has an open area of 25 percent. The duct is 10 feet long.

Solution

The insertion loss is calculated using Equation 11-

104 and the corresponding coefficients in Table 14-54. For example, for the 63 Hz 1/1 octave band Equation 11-104 will have the following form:

$$IL = [0.2825 + (0.3447 \times t) - (5.251 \times 10^{-2} \times t^2) - (0.03837 \times d) + (9.1331 \times 10^{-4} \times d^2) - (8.294 \times 10^{-6} \times d^3)] \times L$$

where t is 1 inch, d is 24 inches, and L is 10 feet. Substituting in the values for t and d and reducing yields:

$$IL = 0.065 \text{ dB/ft}$$

The results for the 1/1 octave frequency bands between 63 Hz and 8000 Hz are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
IL, dB/ft Length, feet	0.065 ×10	0.25 ×10	0.57 ×10	1.28 ×10	1.71 ×10	1.24 ×10	0.85 ×10	0.80 ×10
IL, dB	0.65	2.5	5.7	12.8	17.1	12.4	8.5	8.0

6. Rectangular Duct Elbows

Table 14-59 in Chapter 14 displays insertion loss values for unlined and lined square elbows without turning vanes. For lined square elbows, the duct lining must extend at least two duct widths, w, beyond the elbow and the thickness of the total lining thickness should be at least 10 percent of the duct width, w. Table 14-59 applies only for the solution where the duct is lined before and after the elbow. Table 14-60 gives the insertion loss values associated with round elbows. Table 14-61 gives the insertion loss values for unlined and lined square elbows with turning vanes. In Tables 14-59 through 14-61, "f × w" is the center frequency of the 1/1 octave frequency band (kHz) times the width of elbow (in.) (Figure 11-22).

Example 11-22

Determine the insertion loss (dB) of a 24 inch acoustically lined square elbow without turning vanes.

Solution

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
fxw	1.5	3	6	12	24	48	96	192
IL (Table 14-59), dB	0	1	6	11	10	10	10	10

Example 11-23

Determine the insertion loss (dB) of a round elbow constructed of a 12 inch diameter unlined circular duct.

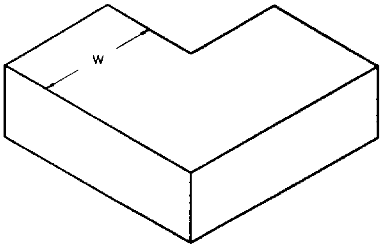
Solution

The results are tabulated below.

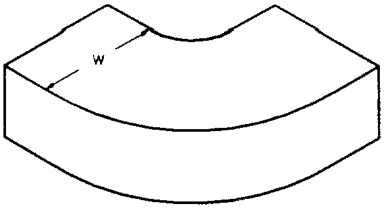
	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
fxw	0.76	1.5	3	6	12	24	48	96
IL (Table 14-60), dB	0	0	1	2	3	3	3	3

7. Acoustically Lined Round Radius Elbows

There are very little data available in the literature with regard to the insertion loss of acoustically lined radius round elbows. A regression equation was developed using measured insertion loss data for round radius elbows. The data were obtained for spiral dual-wall circular ducts. The acoustical lining was a 0.75 lb/ft^3 density fiberglass blanket which ranged in



Square Elbow



Radius Elbow

Figure 11-22 RECTANGULAR DUCT ELBOWS

thickness from one to three inches. The fiberglass was covered with an internal liner of perforated galvanized steel that had an open area of 25 percent. The inside diameter of the elbows tested ranged from 6 to 60 inches. For elbows where $6 \le d \le 18$ inches,

Equation 11-106

$$\begin{aligned} IL \times \left(\frac{d}{r}\right)^2 &= 0.485 + 2.094 \log_{10}(f \times d) \\ &+ 3.172\,[\log_{10}(f \times d)]^2 \\ &- 1.578\,[\log_{10}(f \times d)]^4 \\ &+ 0.085\,[\log_{10}(f \times d)]^7 \end{aligned}$$

and for elbows where $18 < d \le 60$ inches,

Equation 11-107

$$\begin{aligned} IL \times \left(\frac{d}{r}\right)^2 &= -1.493 + 0.538 \times t \\ &+ 1.406 \log_{10}(f \times d) \\ &+ 2.779\,[\log_{10}(f \times d)]^2 \\ &- 0.662\,[\log_{10}(f \times d)]^4 \\ &+ 0.016\,[\log_{10}(f \times d)]^7 \end{aligned}$$

where f is the 1/1 octave band center frequency (Hz), d is the duct diameter (inches), r is the radius of the elbow to the center line of the duct (inches), and t is the thickness (inches), of the acoustical duct liner. Equations 11-106 or 11-107 are seventh order polynomials. Thus, the equations should not be extrapolated beyond the specified limits for each equation. If the value for IL $(d/r)^2$ is negative in either Equation 11-106 or 11-107, set the value equal to zero. The relation that existed between r, d, and t for the elbows that were tested is

Equation 11-108

$$r = 1.5\,d + 3\,t$$

Example 11-24

Determine the insertion loss (dB) of a 24 inch diameter acoustically lined circular elbow with a lining thickness of 2 inches.

Solution

$$r = 1.5 \times 24 + 3 \times 2 = 42 \text{ inches}$$

$$\left(\frac{d}{r}\right)^2 = \left(\frac{24}{42}\right)^2 = 0.327$$

For a diameter of 24 inches, use equation 11-107:

$$\begin{aligned} IL \times \left(\frac{d}{r}\right)^2 &= -1.493 + 0.538 \times 2 \\ &+ 1.406 \log_{10}(f \times 24) \\ &+ 2.779\,[\log_{10}(f \times 24)]^2 \\ &- 0.662\,[\log_{10}(f \times 24)]^4 \\ &+ 0.016\,[\log_{10}(f \times 24)]^7 \end{aligned}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
$IL \cdot [d/r]^2$	0.0	0.85	2.1	3.5	4.6	5.1	5.0	4.5
$x(r/d)^2$	3.06	3.06	3.06	3.06	3.06	3.06	3.06	3.06
IL, dB	0.0	2.6	6.4	10.7	14.1	15.6	15.3	13.8

8. Duct Silencers

Duct silencers (or sound traps) are often used as a means to attenuate unwanted noise in heating, ventilating and air conditioning systems. When duct silencers are used, the following parameters should be considered:

Insertion Loss—The difference between two sound power levels when measured at the same point before and after the silencer is installed.

Airflow Regenerated Noise—The sound power level generated by air flowing through a silencer.

Static Pressure Drop—The airflow pressure loss.

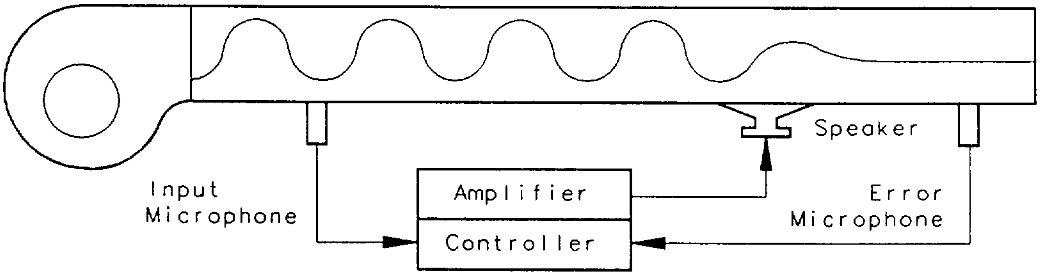
Forward or Reverse Flow—Silencers have different acoustic and aerodynamic characteristics for forward and reverse flow directions.

a. ACTIVE DUCT SILENCERS

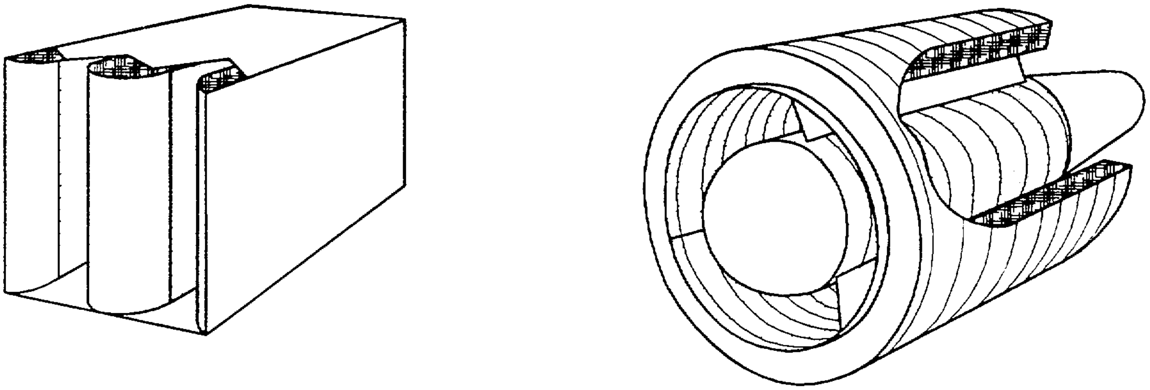
There are two basic types of HVAC duct silencers: active and dissipative. Active duct silencers systems are rather new. They are very effective in attenuating low-frequency, pure-tone noise in a duct. They are also effective in attenuating low-frequency, broadband noise. Active duct silencers consist of a microprocessor, two microphones placed a specified distance apart in a duct and a speaker placed between the microphones, which is mounted external to the duct but radiates sound into the duct [Figure 11-23(a)].

The microphone closest to a sound source that generates objectional low-frequency noise senses the noise. The microphone signal is processed by the microprocessor which generates a signal that is out-of-phase with the objectional noise and transmitted to the speaker. The speaker noise destructively interferes with the objective noise, effectively attenuating it. The second microphone downstream of the speaker senses the attenuated noise and sends a corresponding feedback signal to the microprocessor, so the speaker signal can be adjusted, if necessary.

Active duct silencer systems have no components that are located within the duct. Thus, they can be



(a) Active Duct Silencer



(b) Rectangular Dissipative Duct Silencer

(c) Circular Dissipative Duct Silencer

Figure 11-23 ACTIVE AND DISSIPATIVE DUCT SILENCERS

used to attenuate objectional noise without introducing a pressure loss or regenerated noise into a duct. At present, not enough application data is available to develop an active silencer performance prediction algorithm.

b. DISSIPATIVE DUCT SILENCERS

Dissipative silencers are effective in attenuating broad-band noise. However, they introduce a pressure drop and regenerated noise into a duct. These should always be examined when considering the use of a dissipative silencer. Dissipative silencers can have a rectangular or circular cross section [Figure 11-23(b) and (c)]. Rectangular silencers are available in several different cross-section dimensional configurations and in 3 foot, 5 foot, 7 foot, and 10 foot lengths. Rectangular silencers have parallel sound absorbing surfaces. These surfaces are usually perforated sheet metal surfaces that cover cavities filled with either fiberglass or mineral wool.

Round silencers come in several different open-face diameters and usually have lengths that are a function of the open face diameter. All round silencers have a center body similar to the one shown in Figure 11-23(c). This body is a cylindrical body with perforated sheet metal surface and filled with either fiberglass or mineral wool. The outside shell of a round silencer can be either single- or double-wall construction. For single-wall construction, the outside shell is a solid cylindrical sheet metal shell that has a diameter equal to the open face diameter of the silencer. For double-wall construction, the outside shell consists of two concentric cylindrical sheet metal shells. The outside shell is solid sheet metal. The inner shell is perforated sheet metal and it has a diameter equal to the open face diameter of the silencer. The space between the two shells is filled with fiberglass or mineral wool. The round silencer in Figure 11-23(c) has a center body and double-wall outer shell.

Both rectangular and circular dissipative silencers come in several different pressure drop configurations. The insertion loss, regenerated noise and pressure drop of dissipative duct silencers are functions of silencer design and the location of the silencer in the duct system. These data are experimentally measured and are presented as part of manufacturers' data associated with their product lines. The data should be obtained in a manner consistent with the procedures outlined in ASTM Standard E477-84, Standard Method of Testing Duct Liner Materials and Prefabricated Silencers for Acoustical and Airflow Performance.

Active and dissipative silencers complement each other. Active silencers are usually effective between the 16 Hz to 250 Hz 1/1 octave frequency bands. Dissipative silencers are effective from 63 Hz to 8000 Hz 1/1 octave frequency bands. The general insertion loss or attenuation characteristics of active and dissipative duct silencers are shown in Figure 11-24.

It is not practical to present data for a complete range of rectangular and round duct silencers. This data is highly dependent on manufacturer's design and will be different for each manufacturer. When possible, the silencer's manufacturer data should be used. If it is not available, the typical data presented for rectangular and round, high and low pressure drop, dissipative silencers can be used to estimate the insertion loss, regenerated sound power, and pressure drop associated with selected rectangular and round dissipative duct silencers. The data include insertion loss and regenerated sound power values for sound traveling with (+) and against (−) the airflow. Equations are presented which can be used to calculate the pressure loss across typical silencers and to calculate the silencer face area correction associated with regenerated sound power. Table 14-62 gives typical insertion loss and regenerated sound power levels for rectangular, high pressure drop duct silencers. Table 14-63 gives the same information for rectangular, low pressure drop silencers. The face area correction, FAC, for rectangular duct silencers is given by

Equation 11-109

$$FAC = 10 \log_{10}[FA] - 6$$

where FA is the face area (ft^2) of the silencer. Table 14-64 gives typical insertion loss and regenerated sound power levels for round, high pressure drop duct silencers. Table 14-65 gives the same information for round, low pressure drop silencers. The face area correction, FAC, for round duct silencers is given by

Equaltion 11-110

$$FAC = 10 \log_{10}[FA] - 4.76$$

The silencer face velocity, V (fpm), is given by

Equation 11-111

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

where Q is the volume flow rate (cfm) and FA is defined as before.

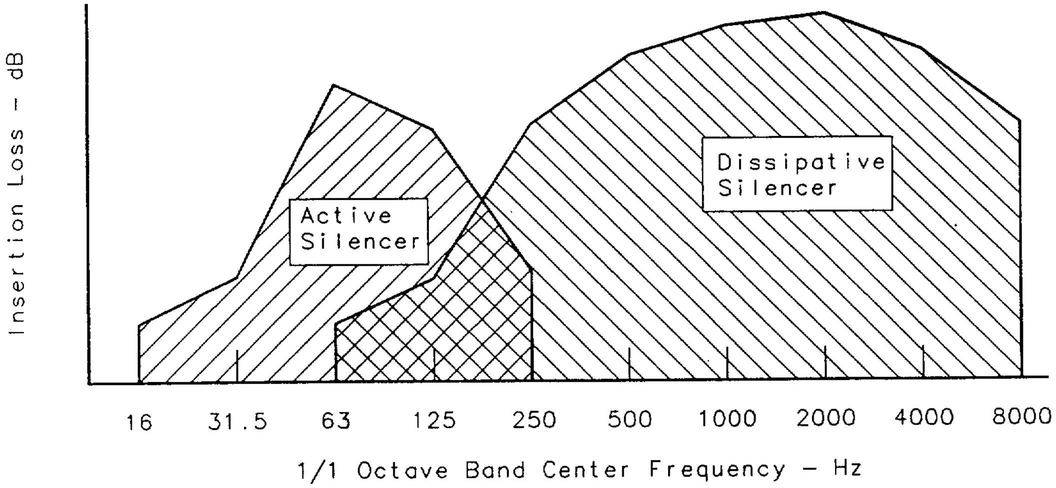


Figure 11-24 INSERTION LOSS OF ACTIVE AND DISSIPATIVE DUCT SILENCERS

The static pressure drop, ΔP (in. w.g.), across a rectangular duct silencer is obtained from

Equation 11-112

$$\Delta P = C_1 \times L^{C_2} \times C_3 \times V^{C_4}$$

where L is the silencer length (feet) and V is the silencer face velocity (fpm). The static pressure drop, ΔP (in. w.g.), across a circular duct silencer is obtained from

Equation 11-113

$$\Delta P = C_3 \times \left(\frac{Q}{C_1 \times d^{C_2}}\right)^{C_4}$$

where Q is volume flow rate (cfm) through the silencer and d is the silencer face diameter (inches). The values of the coefficients, C_1, C_2, C_3, and C_4 are given in Table 14-66.

The pressure drops for dissipative duct silencers specified by Equations 11-112 and 11-113 are for the case where there are no system component effects associated with duct elements, such as fan discharge or return sections, elbows, branch take-offs, etc., upstream or downstream of a duct silencer. When system components effects must be taken into account, a correction factor must be added to the pressure drop specified by Equations 11-112 and 11-113. The pressure drop, ΔP_s (in. w.g.), taking into account system component effects, is given by

Equation 11-114

$$\Delta P_s = \Delta P \times C_5$$

where C_5 is obtained from the coefficients specified in Table 14-67 and is given by

Equation 11-115

$$C_5 = C(up) \times C(down)$$

The equivalent duct diameter for round ducts is the duct diameter. For rectangular ducts, the equivalent duct diameter, D_{eq} (inches), is

Equation 11-116

$$D_{eq} = \sqrt{\frac{4W \times H}{\pi}}$$

where W is the width (inches) of the rectangular duct and H is the height (inches) of the rectangular duct.

When determining the effectiveness of a duct silencer, it is necessary to take into account both the insertion loss and the regenerated sound power levels of the silencer. If L_{W1} is the sound power level (dB) that exists before the sound enters the silencer, the sound power level, L_{W2} (dB), at the exit of the silencer associated with the silencer insertion loss, IL (dB), is given by

Equation 11-117

$$L_{W2} = L_{W1} - IL$$

The regenerated sound power levels, L_{W3}, associated with air flowing through a silencer are equal to the sound power levels, L_{Wr}, given in Tables 14-64 through 14-68, plus the face area correction, FAC, specified by Equation 11-109, 11-110, or

Equation 11-118

$$L_{W3} = L_{Wr} + FAC$$

The regenerated sound power level, L_{W3} (dB), must be added to L_{W2} to obtain the total sound power level, L_{W4} (dB), at the exit of the duct silencer. Because sound power levels are being added, L_{W2} and L_{W3} must be added logarithmically, or

Equation 11-119

$$L_{W4} = 10 \log_{10}[10^{(L_{W2}\ 10)} + 10^{(L_{W3}\ 10)}]$$

Example 11-25

A fan has the following sound power levels:

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
Fan L_{W1}, dB	91	87	83	82	78	76	72	70

The volume flow rate for the fan is 10,000 cfm and the fan has a total static pressure of 1.5 in. w.g. If a low pressure drop rectangular duct silencer is used that has face dimensions of 30 inches × 24 inches and a length of 7 feet, determine the sound power level on the exit side of the duct silencer.

Solution

$$FA = \frac{30 \times 24}{144} = 5 \text{ ft}^2$$

$$V = \frac{10{,}000}{5} = 2{,}000 \text{ ft/min}$$

The static pressure drop from Equation 11-112 is

$$\Delta P = C_1 \times L^{C_2} \times C_3 \times V^{C_4}$$

where $C_1 = 0.6015$, $C_2 = 0.4627$, $C_3 = 9.802 \times 10^{\ 8}$, $C_4 = 2.011$. Thus,

$$\begin{aligned}\Delta P &= 0.6016 \times 7^{0.4627} \times 9.802 \\ &\quad \times 10^{\ 8} \times 2000^{2.011} \\ &= 0.63 \text{ in. w.g.}\end{aligned}$$

From Table 14-67, C(up) equals 1 and C(down) equals 1.4. Thus, C_5 equals 1.4 and

$$\Delta P_s = 1.4 \times 0.63 = 0.88 \text{ in. w.g.}$$

The face area adjustment factor (Equation 11-109) is

$$FAC = 10 \log_{10}[5] - 6 = 1.0 \text{ dB}$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
Fan L_{W1}, dB	91	87	83	82	78	76	72	70
Silencer IL, dB	−4	−12	−26	−43	−47	−48	−47	−30
L_{W2}, dB	87	75	57	39	31	28	25	40
Flow L_{Wr}, dB	73	65	64	63	55	56	57	56
FAC, dB	1	1	1	1	1	1	1	1
L_{W3}, dB	74	66	65	64	56	57	58	57
L_{W4}, dB	87	76	66	64	56	57	58	57

9. Duct Branch Sound Power Division

When sound traveling in a duct encounters a junction, the sound power contained in the incident sound waves in the main duct is distributed between the branches associated with the junction. This division of sound power is referred to as the branch sound power division. The corresponding attenuation of sound power that is transmitted down each branch of the junction is comprised of two components. The first is associated with the reflection of the incident sound wave if the sum of the cross-sectional areas of the individual branches, ΣS_{Bi}, differs from the cross-sectional area, S_M, of the main duct. The second component is associated with the ratio of the cross-sectional area, S_{Bi}, of an individual branch divided by the sum of the cross-sectional areas of the individual branches, ΣS_{Bi}.

The attenuation of sound power, ΔL_{Bi}, at a junction that is related to the sound power transmitted down an individual branch of the junction is given by

Equation 11-120

$$\Delta L_{Bi} = 10 \log_{10}\left[1 - \left(\frac{\frac{\Sigma S_{Bi}}{S_m} - 1}{\frac{\Sigma S_{Bi}}{S_m} + 1}\right)^2\right] + 10 \log_{10}\left[\frac{S_{Bi}}{\Sigma S_{Bi}}\right]$$

where S_{Bi} is the cross-sectional area (in^2) of branch i, ΣS_{Bi} is the total cross-sectional area (in^2) of the individual branches that continue from the main feeder duct, and S_M is the cross-sectional area (in^2) of the main duct. The first term in Equation 11-120 is related to the reflection of the incident wave when the area of the branches differs from the area of the main feeder duct. It is present only when the sound waves propagating in the main feeder duct are plane waves and when ΣS_{Bi} is not equal to S_M. Plane wave propagation in a duct exists at frequencies below

Equation 11-121

$$f_{co} = \frac{c_o}{a}$$

where c_o is the speed of sound in air (ft/sec) and a is the larger cross-section dimension (feet) of a rectangular duct, or below

Equation 11-122

$$f_{co} = 0.586 \frac{c_o}{d}$$

where d is the diameter (feet) of a circular duct. The cutoff frequency, f_{co}, is the frequency above which plane waves no longer propagate in a duct. At these higher frequencies the waves that propagate in the duct are referred to as cross or spinning modes. The second term in Equation 11-120 is associated with the division of the remaining incident sound power at the junction between the individual branches. If the total cross-sectional area of the branches after the junction is equal to the cross-sectional area of the main duct or if the frequencies of interest are above the cutoff frequency, Equation 11-120 reduces to

Equation 11-123

$$\Delta L_{Bi} = 10 \log_{10}\left[\frac{S_{Bi}}{\Sigma S_{Bi}}\right]$$

Example 11-26

An 18 inch diameter main feeder duct terminates into a junction that has a 12 inch diameter branch (continuation of the main duct) and a 6 inch diameter 90° branch takeoff. Determine the attenuation (dB) of the sound power transmitted into the 90° branch takeoff.

Solution

$$S_M = \frac{\pi \times 18^2}{4} = 254.5 \text{ in}^2 \text{ Main duct}$$

$$S_{B1} = \frac{\pi \times 12^2}{4} = 113.1 \text{ in}^2 \text{ Continuation branch}$$

$$S_{B2} = \frac{\pi \times 6^2}{4} = 28.3 \text{ in}^2\ 90^\circ \text{ Branch takeoff}$$

$$\Sigma S_{Bi} = 113.1 + 28.3 = 141.4 \text{ in}^2$$

$$f_{co} = \frac{0.586 \times 1125 \times 12}{18} = 439.5 \text{ Hz}$$

Using Equation 11-120, the branch power division associated with branch 2 can be determined.

$$\frac{S_{B2}}{\Sigma\, S_{Bi}} = \frac{28.3}{141.4} = 0.2$$

$$1 - \left[\frac{\dfrac{\Sigma\, S_{Bi}}{S_m} - 1}{\dfrac{\Sigma\, S_{Bi}}{S_m} + 1}\right]^2 = 1 - \left[\frac{\dfrac{141.4}{254.5} - 1}{\dfrac{141.4}{254.5} + 1}\right]^2 = 0.918$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
10 Log_{10}[0.918], dB	0.4	0.4	0.4					
10 Log_{10}[0.2], dB	7.0	7.0	7.0	7.0	7.0	7.0	7.0	7.0
ΔL_{B2}, dB	7.4	7.4	7.4	7.0	7.0	7.0	7.0	7.0

10. Duct End Reflection Loss

When low frequency plane sound waves interact with a small diffuser that discharges into a large room, a significant amount of the sound energy incident on this interface is reflected back into the duct. The sound attenuation, ΔL, associated with duct end reflection losses can be approximated by

Equation 11-124

$$\Delta L = 10 \log_{10}\left[1 + \left(\frac{c_o}{\pi \times f \times D}\right)^{1.88}\right]$$

for ducts terminated in free space and by

Equation 11-125

$$\Delta L = 10 \log_{10}\left[1 + \left(\frac{0.8\, c_o}{\pi \times f \times D}\right)^{1.88}\right]$$

for ducts terminated flush with a wall. f is frequency (Hz), c_o is the speed of sound in air (ft/sec), and D is the diameter (feet) of a round duct or the effective diameter of a rectangular duct. If the duct is rectangular, D is

Equation 11-126

$$D = \left[\frac{4 \times \text{Area}}{\pi}\right]^{1/2}$$

where Area is the area (ft^2) of the rectangular duct. D can have the unit of inches if c_o has the units of in/sec.

There are some limitations associated with Equations 11-124 and 11-125. The tests on which these equations are based were conducted with straight sections of round ducts. These ducts directly terminated into a reverberation chamber with no restriction on the end of the duct or with a circular orifice constriction placed over the end of the duct. Diffusers can be either round or rectangular. They usually have a restriction associated with them which may either be a damper, guide vanes to direct airflow, a perforated metal facing, or a combination of these elements. Currently, there is no data which indicate the effects of these elements. It is not known whether these elements react similar to the orifices used in the above-described tests. As a result, the effects of an orifice placed over the end of a duct are not included in Equations 11-124 and 11-125.

One can assume that using Equation 11-123 to calculate D will yield reasonable results with diffusers that have low aspect ratios (length/width). However, many types of diffusers (particularly slot diffusers) have high aspect ratios. It is currently not known whether Equations 11-124 and 11-125 can be accurately used with these diffusers.

Finally, many diffusers do not have long straight sections (greater than three duct diameters) before they terminate into a room. Many duct sections between a main feed branch and a diffuser may be curved or may be short, stubby sections. The effects of these configurations on the duct end reflection loss are currently not known. It is felt that Equations 11-124 and 11-125 can be used with reasonable accuracy for many diffuser configurations. However, some caution should be exercised when a diffuser configuration differs quite drastically from the test conditions used to derive these equations.

Example 11-27

Determine the duct end reflection loss associated with a circular diffuser that has a diameter of 12 inches. Assume the diffuser terminates in free space.

Solution

Use Equation 11-124 for the calculations associated with this example. The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
$c_o/(\pi \cdot f \cdot D)$	5.72	2.86	1.43	0.72	0.36	0.18	0.09	0.04
ΔL, dB	14.4	9.1	4.7	1.9	0.6	0.2	0.0	0.0

K SOUND TRANSMISSION IN INDOOR SPACES

1. Sound Transmission Through Ceiling Systems

When mechanical equipment is located in the ceiling plenum above an occupied room, noise transmission through the ceiling can be high enough to cause excessive noise levels in that room. Since there are no standard tests for determining the transmission loss through ceiling construction, data are limited. Table 14-68 gives single-pass transmission loss values for various ceiling materials.

The single-pass transmission loss values in Table 14-68 are for ceilings in which there are no penetrations for acoustical flanking. The acoustical integrity of ceilings can be greatly compromised by these factors. When leaks and/or flanking paths are present, the transmission loss of a ceiling can be significantly reduced. Equation 11-127 gives the corrected transmission loss, TL (dB), taking into account flanking transmission and acoustical leaks.

$$TL = -10 \log_{10}[(l - \tau) \times 10^{(-TL/10)} + \tau]$$

where τ is the correction coefficient for type of ceiling. Table 14-69 gives approximate values for τ for various types of ceilings. The values were selected because they yield transmission loss values that agree reasonably well with expected values.

Example 11-28

Determine the transmission loss of a ceiling with 1 layer of 1/2 inch gypsum board which has a surface weight of 2.1 lb/ft^2. The ceiling has few diffusers and the penetrations are well sealed.

Solution

The following transmission loss values are obtained from Table 14-68:

	1/1 Octave Band Center Frequency—Hz						
	63	125	250	500	1000	2000	4000
TL, dB	9	14	20	24	30	31	27

The correction factor for a suspended ceiling with few ceiling diffusers and well sealed penetrations is 0.001. Thus,

$$TL = -10 \log_{10}[(1 - 0.001) \times 10^{(TL/10)} + 0.001]$$

The results are tabulated below.

	1/1 Octave Band Center Frequency—Hz						
	63	125	250	500	1000	2000	4000
TL, dB	9.0	13.9	19.6	23.0	27.0	27.5	25.2

2. Receiver Room Sound Corrections

The sound pressure levels associated with a sound source that occur at a given point in a room depend on the source strength, the acoustical characteristics of the room (surface treatments, furnishings, etc.), the room volume, and the distance of the sound source from the point of observation. There are two types of sound sources associated with HVAC systems that can exist in a room: point source and line source. The point source is usually associated with sound radiated from supply and return air diffusers, equipment items, such as, fan-powered terminal units above a lay-in ceiling, and other similar items. The line source is associated with duct breakout noise. Two equations are proposed for use with point sound sources: Thompson's equation and Schultz's equation.

Thompson's equation is based on experimental results, and it is an empirical modification of the classical diffuse room equation. The equation is

Equation 11-128
(Thompson)

$$L_p = L_w + 10 \log_{10}\left[\frac{Qe^{-mr}}{4\pi r^2} + \frac{MFP}{r}\frac{4}{R}\right] + 10 \log_{10}[N] + 10.5$$

where Q is the directivity factor associated with the sound source in the direction of the receiver, r is the distance (feet) between the sound source and receiver, MFP is the mean free path (feet), R is the room constant (ft^2), m is the air absorption coefficient (1/ft), and N is the number of point sound sources. Q equals 1 for whole space, 2 for half space, 4 for quarter space, and 8 for eighth space. For most problems of interest to a HVAC designer, sound from a sound source generally radiates into half space; thus, Q will usually equal 2. The mean free path, MFP, is given by

Equation 11-129

$$MFP = \frac{4\ V}{S}$$

where S is the total surface area of the room (ft^2) and V is the total room volume (ft^3). The room constant is specified by

Equation 11-130

$$R = \frac{S\alpha_T}{1 - \alpha_T}$$

where α_T is the average room absorption coefficient. α_T is given by

Equation 11-131

$$\alpha_T = \alpha_a + \frac{4mV}{S}$$

where α_a is the average room absorption coefficient associated with the acoustical characteristics of the room surfaces and 4mV/S is the contribution to room sound absorption associated with air absorption. Table 14-70 gives typical values of α for rooms with the specified acoustical characteristics. The values of α In Table 14-70 were calculated assuming a 20 foot by 40 foot by 8 foot high room. The walls of the room were constructed of a single layer of 5/8 inch gypsum wallboard. The ceiling, floor, furnishings, and occupancy of the room were varied as indicated in Table 14-70, and the corresponding values for average sound absorption coefficients were calculated. Table 14-71 gives typical values of m. Air absorption associated with values of m need only be considered in large rooms, such as, auditoriums and large open office areas.

Schultz's equation for converting from L_w to L_p in a receiver room is an empirical equation based solely on experimental results. Schultz found that the sound pressure levels in a receiver room are functions of the sound power level of the sound source, the room volume, and 1/1 octave band center frequency. For individual sound sources, the sound pressure level, L_p (dB), in a receiver room is given by

Equation 11-132
(Schultz)

$$L_p = L_w - 10\log_{10}[r] - 5\log_{10}[V] - 3\log_{10}[f] + 10\log_{10}[N] + 25$$

where L_W is the sound power level (dB) of the sound source, r is the distance (feet) between the sound source and receiver, V is the room volume (ft^3), f is the 1/1 octave band center frequency (Hz), and N is the number of sound sources. For an array of distributed ceiling diffusers, Equation 11-129 can be expressed

Equation 11-133

$$L_p(5\ ft) = L_{ws} - 27.6\log_{10}[h] - 5\log_{10}[X] - 3\log_{10}[f] + 1.3\log_{10}[N] + 30$$

where L_p(5 feet) is the sound pressure level (dB) at a distance 5 feet above the floor, L_{ws} is the sound power level (dB) associated with a single diffuser in the array, h is the ceiling height, N is the number of ceiling diffusers, and X is the ratio of the floor area served by each diffuser divided by the square of the ceiling height (X = 1 if the area served equals h^2).

Caution should be exercised in using both Thompson's Equation 11-128 and Schultz's Equation 11-132. With regard to Thompson's equation, it is necessary to make an assumption as to the acoustical characteristics of the room being analyzed. Most rooms of interest to the HVAC designer will be acoustically medium dead. Thus, the average absorption coefficients in Table 14-70 for a medium dead room should be selected for most calculations. It is generally understood that the conversion from L_w to L_p in a receiving room is a function of room sound absorption, typically described by the average absorption coefficient of the receiving room.

In Schultz's equations, the effects of the room absorption are contained in the room volume term. Equations 11-132 and 11-133 apply primarily to rooms that are acoustically medium dead. These equations will give reasonable results for rooms that have acoustical characteristics that range from average to dead. The lowest frequency band for which data were presented by Thompson was the 250 Hz 1/1 octave band. For Schultz, the lowest frequency band was the 125 Hz 1/1 octave band. Also, with regard Schultz's data, there are some questions relative to the value of the calibrated sound power level in the 125 Hz 1/1 octave band of the reference sound source used to generate the data that were utilized in developing his equation. Equation 11-132 is a completely empirical equation. As such, there are currently no data available that can be used to justify extrapolating this equation or Equation 11-132 to the 63 Hz 1/1 octave band.

Since Thompson's equation is a modification of the classical diffuse room equation, there is some justification for extrapolating Equation 11-128 to the 63 Hz and 125 Hz 1/1 octave bands. However, when this is done, a great deal of caution should be exercised when interpeting the results in these bands. In the 1/1 octave frequency bands from 125 Hz to 8,000 Hz and for medium dead rooms, currently Schultz's equation gives the most accurate and dependable results.

When considering the sound pressure levels in a room that are associated with duct breakout, the sound source is the duct, which must be considered as a line source. For converting from duct breakout sound power levels to the corresponding sound pressure levels in the room, the equation for a line sound source should be used. It is

Equation 11-134

$$L_p = L_W + 10\log_{10}\left[\frac{Q}{4\pi rL} + \frac{4}{R}\right] + 10.5$$

where L_W and L_p are defined as before, Q is as specified before, r is the distance (feet) from the line source to the receiver, L is the length of the line source, and R is defined by Equation 11-130. This is the classical diffuse room equation for a line sound source. There currently is not other information for converting from L_W to L_p for line sound sources.

Example 11-29

Determine the values for converting from L_W to L_p for a room that is 15 feet long, 10 feet wide, and 8 feet high. The sound source is a single diffuser located in the ceiling. The distance between the sound source and receiver is 8 feet.

Solution

Use Equation 11-132:

$V = 15 \times 10 \times 8 = 1200\ \text{ft}^3$

The results are tabulated below

	1/1 Octave Band Center Frequency—Hz							
	63	125	250	500	1000	2000	4000	8000
$-10\ \text{Log}_{10}[8]$	−9.0	−9.0	−9.0	−9.0	−9.0	−9.0	−9.0	−9.0
$-5\ \text{Log}_{10}[1200]$	−15.4	−15.4	−15.4	−15.4	−15.4	−15.4	−15.4	−15.4
$-3\ \text{Log}_{10}[f]$	−5.4	−6.3	−7.2	−8.1	−9.0	−9.9	−10.8	−11.7
	25.0	25.0	25.0	25.0	25.0	25.0	25.0	25.0
$L_p - L_W$, dB	−4.8	−5.7	−6.6	−7.5	−8.4	−9.3	−10.2	−11.1

L SYSTEM EXAMPLE

Individual examples have been given in the preceding sections which demonstrate how to calculate equipment and regenerated sound power levels and sound attenuation values associated with the system elements of HVAC air distribution systems. It is now worth while to examine a complete HVAC system example to see how the information that has been presented can be tied together to determine the sound pressure levels associated with a specific HVAC system that will exist in a specified space. Complete calculations for each system element will not be given. Only a summary of the tabulated results will be listed.

Air is supplied to the HVAC system in this example by the rooftop unit shown in Figure 11-25. The receiver room is a room that is directly below the unit. The room has the following dimensions: length—20 feet, width—20 feet; and height—9 feet. For this example, it is assumed that the roof penetrations associated with the supply and return air ducts are well sealed and that there are no other roof penetrations associated with the unit. The supply side of the rooftop unit is ducted to a VAV control unit which serves the room in question. A return air grill conducts air to a common ceiling return air plenum. The return air is then directed to the rooftop unit through a short rectangular return air duct.

Three sound paths should be examined. They are:

Path 1. Fan airborne supply air sound that enters the room from the supply air system through the ceiling diffuser.

Path 2. Fan airborne supply air sound that breaks out through the duct wall of the main supply air duct into the plenum space above the room.

Path 3. Fan airborne return air sound that enters the room from the inlet of the return air duct.

The sound power levels associated with the fan in the rooftop unit are specified by the manufacturer to be:

	1/1 Octave Band Center Frequency—Hz						
	63	125	250	500	1000	2000	4000
Rooftop Sup. Air; CFM 7000; SP 2.5 in.w.g.	92	86	80	78	78	74	71
Rooftop Ret. Air; CFM 7000; SP 2.5 in.w.g.	82	79	73	69	69	67	59

Paths 1 and 2 are associated with the supply air side of the system. Figure 11-26 shows a layout of the part of the supply air system that is associated with the receiver room. The main duct is a 22 inch diameter, 26 gauge, unlined, circular sheet metal duct. The flow volume in the main duct is 7,000 cfm. The silencer after the radiused elbow is a 22 inch diameter by 44 inch long, high pressure, circular, duct silencer. The branch junction that occurs 8 feet from the silencer is a 45 degree wye. The branch duct between the main duct and the VAV control unit is a 10 inch diameter,

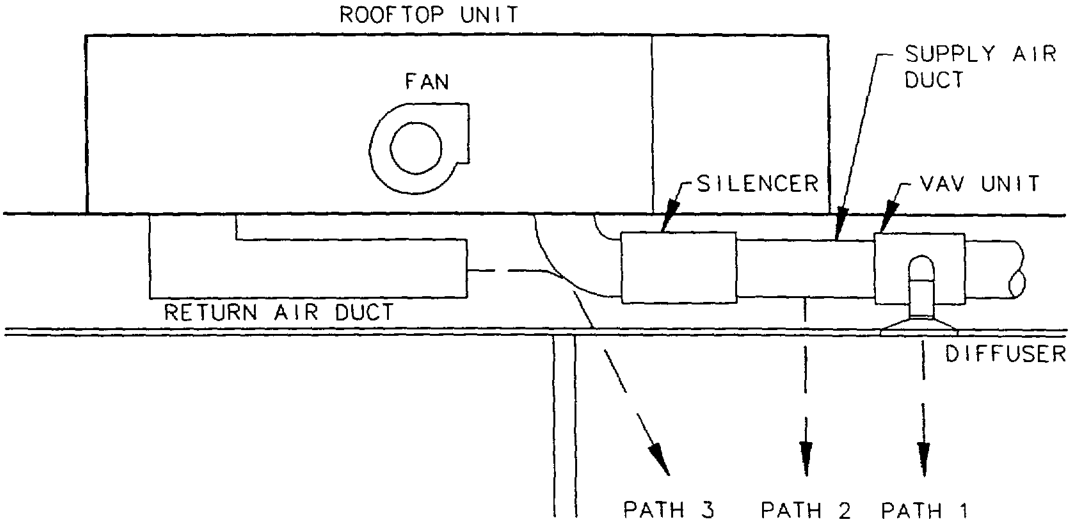


Figure 11-25 PATHS FOR SYSTEM EXAMPLE

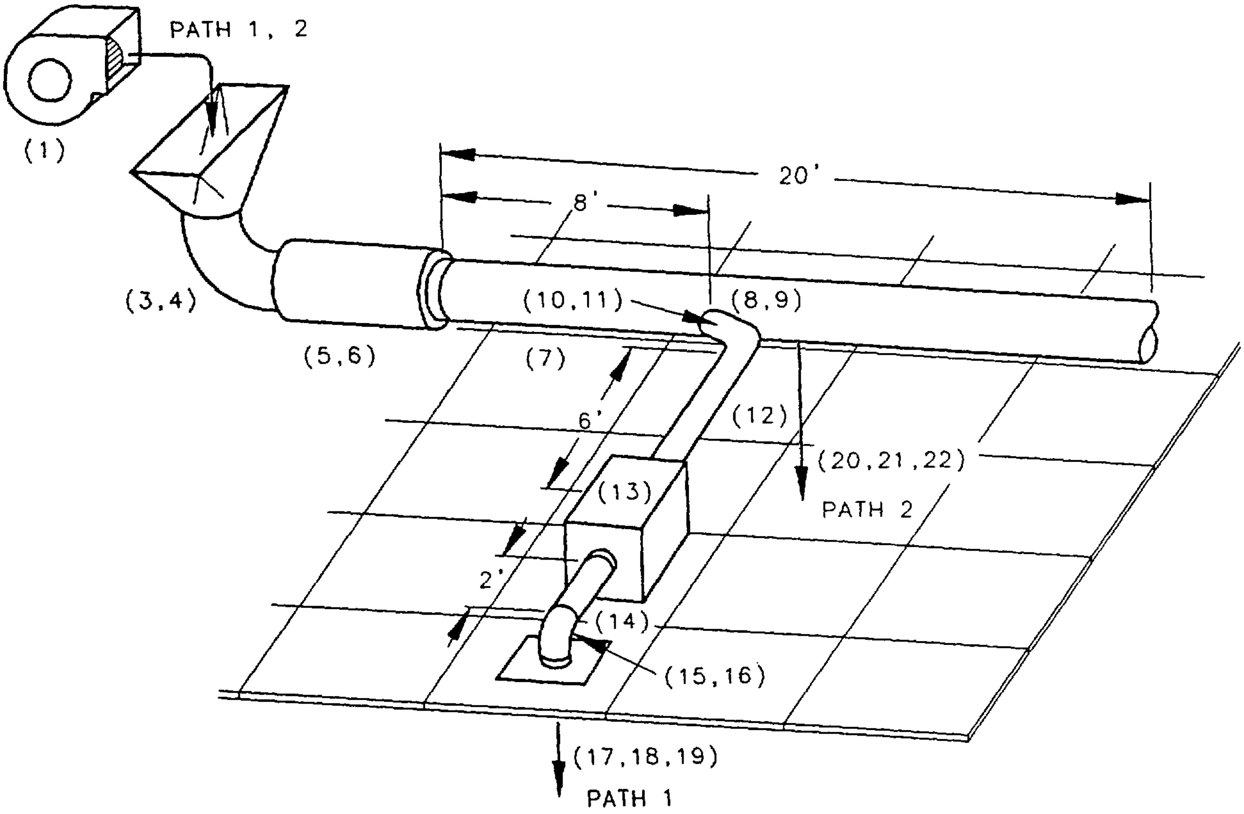


Figure 11-26 SUPPLY AIR PORTION OF SYSTEM EXAMPLE

unlined, circular sheet metal duct. The flow volume in the branch duct is 800 cfm. The straight section of duct between the VAV control unit and the diffuser is a 10 inch diameter, unlined circular sheet metal duct. The diffuser is a 15 inch by 15 inch square diffuser. Assume a typical distance between the diffuser and a listener in the room is 5 feet. With regard to the duct breakout sound associated with the main duct, the length of the duct that runs over the room is 20 feet. The ceiling of the room is comprised of 2 ft × 4 ft × ⅝ in. lay-in ceiling tiles that have a surface weight of 0.6-0.7 lb/sq. ft. The ceiling has integrated lighting and diffusers. Path 3 is associated with the return air side of the system. Figure 11-27 shows a layout of the part of the return air system that is associated with the receiver room. The rectangular return air duct is lined with 2 inch thick 3 lb/ft^3 density fiberglass duct liner. For the return air path, assume the typical distance between the inlet of the return air duct and a listener is 10 feet.

The analysis associated with each path begins at the rooftop unit (fan) and proceeds progressively through the different system elements to the receiver room. The system element numbers in the tables correspond to the element numbers contained in brackets in Figures 11-26 and 11-27.

The first table is associated with Path 1. The first entry in the table is the manufacturer's values for supply air fan sound power levels (1). The second entry is the sound attenuation associated with the 22 inch diameter unlined radius elbow (3). Since the next entry is associated with the regenerated sound power levels associated with the elbow (4), it is necessary to tabulate the results associated with the elbow attenuation to determine the sound power levels at the exit of the elbow. These sound power levels and the elbow regenerated sound power levels are then added logarithmically. In a like fashion, the dynamic insertion loss values of the duct silencer (5) and the silencer regenerated sound power levels (6) are included in the table and tabulated. Next, the attenuation associated with the 8 foot section of 22 inch diameter duct (7) and the branch power division (10) associated with sound propagation in the 10 inch diameter branch duct are included in the table. After element 10, the sound power levels that exist in the

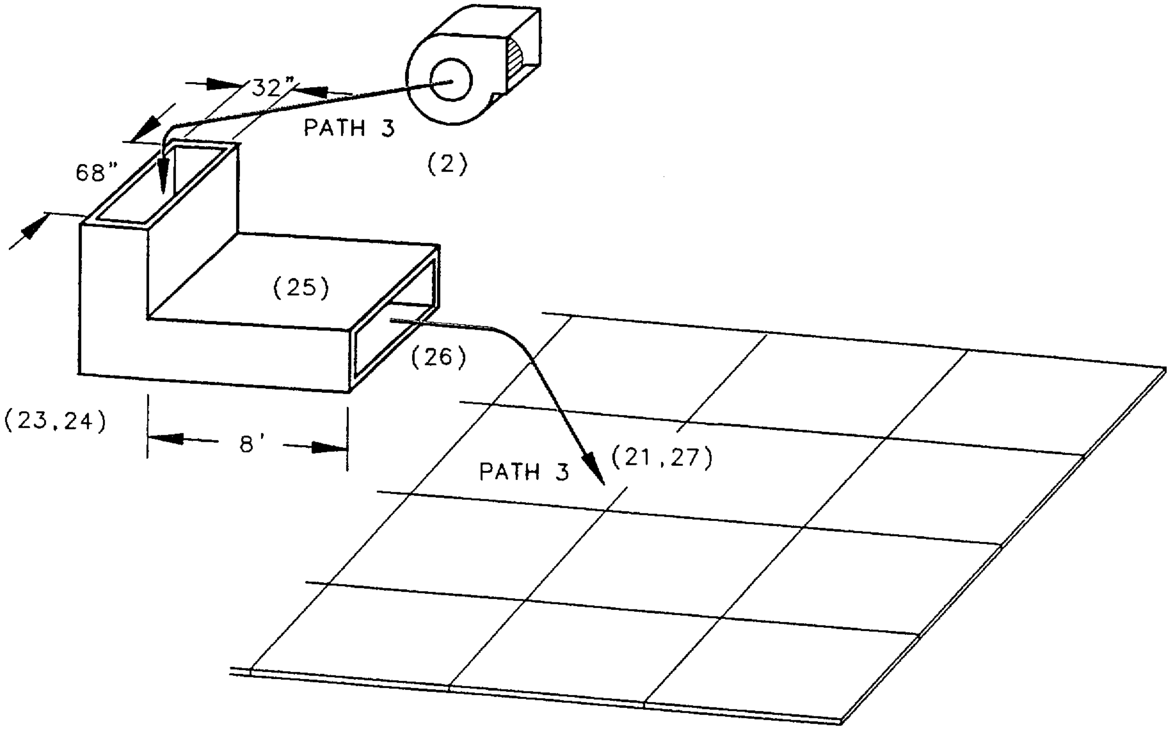


Figure 11-27 RETURN AIR PORTION OF SYSTEM EXAMPLE

branch duct after the branch take off are calculated so that the regenerated sound power levels (11) in the branch duct associated with the branch take-off can be logarithmically added to the results.

Next, the sound attenuation values associated with the 6 foot section of 10 inch diameter unlined duct (12), the terminal volume regulation unit (13), the 2 foot section of 10 inch diameter unlined duct (14), and 10 inch diameter radius elbow (15) are included in the table. The sound power levels that exist at the exit of the elbow are then calculated so that the regenerated sound power levels (16) associated with the elbow can be logarithmically added to the results. The diffuser end reflection loss (17) and the diffuser regenerated sound power levels (18) are appropriately included in the table. The sound power levels that are tabulated after element 18 are the sound power levels that exist at the diffuser in the receiver room. The final entry in the table is the "room correction" which converts the sound power levels at the diffuser to their corresponding sound pressure levels at the point of interest in the receiver room. The NC, RC, and dBA levels associated with the sound pressure levels from Path 1 are listed as the last line in the table.

PATH 1

	1/1 Octave Band Center Freq-Hz						
DESCRIPTION	63	125	250	500	1K	2K	4K
(1) FAN - SUPPLY AIR; CFM 7000; SP 2.5 IN.	92	86	80	78	78	74	71
(3) 22 IN. WIDE (DIA) UNLINED RADIUS ELBOW	0	−1	−2	−3	−3	−3	−3
SUM WITH NOISE REDUCTION VALUES	92	85	78	75	75	71	68
(4) 90 DEG BEND W/O TURNING VANES; 12 IN. RAD	56	54	51	47	42	37	29
SUM SOUND POWER LEVELS	92	85	78	75	75	71	68
(5) 22 IN. DIAM × 44 IN. HIGH PRESS SILENCER	−4	−7	−19	−31	−38	−38	−27
SUM WITH NOISE REDUCTION VALUES	88	78	59	44	37	33	41
(6) REG NOISE FROM ABOVE SILENCER	68	79	69	60	59	59	55
SUM SOUND POWER LEVELS	88	82	69	60	59	59	55
(7) 22 IN. DIA × 8 FT. UNLINED CIR DUCT	0	0	0	0	0	0	0
(10) BR PWL DIV; M-22 IN. DIA; B-10 IN. DIA	−8	−8	−8	−8	−8	−8	−8
SUM WITH NOISE REDUCTION VALUES	80	74	61	52	51	51	47
(11) DUCT 90 DEG BRANCH TAKEOFF; 2 IN. RADIUS	56	53	50	47	43	37	31
SUM SOUND POWER LEVELS	80	74	61	53	52	51	47
(12) 10 IN. DIA × 6 FT. UNLINED CIR DUCT	0	0	0	0	0	0	0
(13) TERMINAL VOLUME REG UNIT (GEN ATTN)	0	−5	−10	−15	−15	−15	−15
(14) 10 IN. DIA × 2 FT. UNLINED CIR DUCT	0	0	0	0	0	0	0
(15) 10 IN. WIDE (DIA) UNLINED RADIUS ELBOW	0	0	−1	−2	−3	−3	−3
SUM WITH NOISE REDUCTION VALUES	80	69	50	36	34	33	29
(16) 90 DEG BEND W/O TURNING VANES; 2 IN. RAD	49	45	41	37	31	24	16
SUM SOUND POWER LEVELS	80	69	51	40	36	34	29
(17) 10 IN. DIA DIFFUSER END REF LOSS	−16	10	−6	−2	−1	0	0
SUM WITH NOISE REDUCTION VALUES	64	59	45	38	35	34	29
(18) 15 IN. × 15 IN. RECTANGULAR DIFFUSER	31	36	39	40	39	36	30
SUM SOUND POWER LEVELS	64	59	46	42	40	38	33
(19) ASHRAE ROOM CORR - 1 IND SOUND SOURCE	−5	−6	−7	−8	−9	−10	−11
SOUND PRESS LEVELS - RECEIVER ROOM	59	53	39	34	31	28	22

NC: NC = 36 RC: RC = 31(R) DBA: 41 DBA

Elements 1 through 7 in Path 2 are the same as Path 1. Elements 8 and 9 are associated with the branch power division (8) and the corresponding regenerated sound power levels (9) associated with sound that propagates down the main duct beyond the duct branch. The next three entries in the table are the sound transmission loss associated with the duct breakout sound (20), the sound transmission loss

PATH 2

DESCRIPTION	1/1 Octave Band Center Freq-Hz						
	63	125	250	500	1K	2K	4K
(1) FAN - SUPPLY AIR; CFM 7000; SP 2.5 IN.	92	86	80	78	78	74	71
(3) 22 IN. WIDE (DIA) UNLINED RADIUS ELBOW	0	−1	−2	−3	−3	−3	−3
SUM WITH NOISE REDUCTION VALUES	92	85	78	75	75	71	68
(4) 90 DEG BEND W/O TURNING VANES; 12 IN. RAD	56	54	51	47	42	37	29
SUM SOUND POWER LEVELS	92	85	78	75	75	71	68
(5) 22 IN. DIA × 44 IN. HIGH PRESS SILENCER	−4	−7	−19	−31	−38	−38	−27
SUM WITH NOISE REDUCTION VALUES	88	78	59	44	37	33	41
(6) REG NOISE FROM ABOVE SILENCER	68	79	69	60	59	59	55
SUM SOUND POWER LEVELS	88	82	69	60	59	59	55
(7) 22 IN. DIA × 8 FT. UNLINED CIR DUCT	0	0	0	0	0	0	0
(8) BR.PWL DIV; M-22 IN. DIA; B-22 IN. DIA	−1	−1	−1	−1	−1	−1	−1
SUM WITH NOISE REDUCTION VALUES	87	81	68	59	58	58	54
(9) DUCT 90 DEG BRANCH TAKEOFF; 2 IN. RADIUS	63	60	57	54	50	44	34
SUM SOUND POWER LEVELS	87	81	68	60	59	58	54
(20) 22 IN. DIA × 20 FT. 26 GA DUCT BREAKOUT	−29	−29	−21	−11	−9	−7	−5
(21) 2×4×⅝ LAY-IN CEILING	−4	−8	−8	−12	−14	−15	−15
(22) LINE SOURCE-MED DEAD ROOM	−6	−5	−4	−6	−7	−8	−9
SOUND PRESS LEVELS - RECEIVER ROOM	48	39	35	31	29	28	25

NC: NC = 29 RC: RC = 29(H) DBA: 35 DBA

PATH 3

DESCRIPTION	1/1 Octave Band Center Freq-Hz						
	63	125	250	500	1K	2K	4K
(1) FAN - RETURN AIR; CFM 7000; SP 2.5 IN.	82	79	80	78	78	74	71
(23) 32 IN. WIDE LINED SQ ELBOW W/O TURN VANES	−1	−6	−11	−10	−10	−10	−10
SUM WITH NOISE REDUCTION VALUES	81	73	69	68	68	64	61
(24) 90 DEG BEND W/O TURNING VANES; 0.5 IN. RAD	77	73	68	62	55	48	38
SUM SOUND POWER LEVELS	82	76	72	69	68	64	61
(25) 32 IN. × 68 IN. × 8 FT. LINED DUCT	−2	−2	−6	−18	−15	−12	−13
(26) 32 IN. × 68 IN. DIFFUSER END REF LOSS	−4	−2	0	0	0	0	0
(21) 2×4×⅝ LAY-IN CEILING	−4	−8	−8	−12	−14	−15	−15
(27) ASHRAE ROOM CORR −1 IND SOUND SOURCE	−8	−9	−10	−11	−12	−13	−14
SOUND PRESS LEVELS - RECEIVER ROOM	64	55	48	28	27	24	19

NC: NC = 40 RC: RC = 26(R) DBA: 44 DBA

associated with the ceiling (21), and the "room correction" (22) converting the sound power levels at the ceiling to corresponding sound pressure levels in the room.

The first element in Path 3 is the manufacturer's values for return air fan sound power levels (2). The next two elements are the sound attenuation associated with a 32 inch wide lined square elbow without turning vanes (23) and the regenerated sound power levels associated with the square elbow (24). The final four elements are the insertion loss associated with a 32 inch × 68 inch × 8 foot rectangular sheet metal duct lined with 2 inch thick 3 lb/ft^3 fiberglass duct lining (26), the diffuser end reflection loss (27), the transmission loss through the ceiling (21), and the "room correction" (27) converting the sound power levels at the ceiling to corresponding sound pressure levels in the room.

The total sound pressure levels in the receiver room from the three paths are obtained by logarithmically adding the individual sound pressure levels associated with each path. From the total sound pressure levels for all three paths, the NC level in the room is NC 40; the RC level is RC 33(R-H) and the A-weighted sound pressure level is 44 dBA.

TOTAL SOUND PRESSURE LEVELS—ALL PATHS

	1/1 Octave Band Center Freq-Hz						
DESCRIPTION	**63**	**125**	**250**	**500**	**1K**	**2K**	**4K**
SOUND PRESSURE LEVELS PATH NO. 1	59	53	39	34	31	28	22
SOUND PRESSURE LEVELS PATH NO. 2	48	39	35	31	29	28	25
SOUND PRESSURE LEVELS PATH NO. 3	65	55	48	28	27	24	19
TOTAL SOUND PRESSURE LEVELS—ALL PATHS	65	57	49	37	34	32	28

NC: NC = 40 RC: RC = 26(R) DBA: 44 DBA

CHAPTER 12

DUCT SYSTEM CONSTRUCTION

A INTRODUCTION

A variety of materials have been used in the construction of ducts. There has been a tendency to emphasize (or deemphasize) certain of the general requirements for all ducts, depending on the particular character of the application. Selection of the materials used throughout the duct system should follow the same careful consideration as the other system components. The different materials used in duct systems can substantially affect the overall performance of the systems, as the listed "advantages" should be evaluated with the "limiting characteristics" prior to the material selection.

Some materials used for ducts include: galvanized steel, black carbon steel, aluminum, stainless steel, copper, fiberglas reinforced plastic (FRP), polyvinyl chloride (PVC), polyvinyl steel (PVS), concrete, fibrous glass (duct board), and gypsum board. Information will be given on 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 14-3 and Table 14-1 of Chapter 14 gives correction factors used to adjust duct friction losses for materials other than galvanized steel (multiply duct friction loss by factor in Table). This higher friction loss can be one of the most important items to consider when selecting a different duct material.

Consideration must also be given to selection of duct construction components other than those materials used for the duct walls. Such items as flexible ducts, duct liner, pressure sensitive tapes, sealants, adhesives, reinforcements, hangers, etc., are appropriately described in individual SMACNA Manuals as well as many other publications.

It is emphasized that special material selection and construction could be necessary when designing systems serving nuclear projects, earthquake prone areas and projects with other unusual requirements.

B DUCT SYSTEM SPECIFICATION CHECK LIST

In addition to SMACNA duct construction standards, the specification and/or detail drawings should include the following detailed duct system requirements:

a) Local code requirements
b) Duct system static pressure classifications (SMACNA standard flag designation).
c) Duct material selection.
d) Allowable duct leakage (specify sealing system classification).
e) Insulation requirement (external and/or liner).
f) Sound control devices and methods.
g) Outlet and inlet performance.
h) Filters.
i) Dampers (fire, smoke, and volume control).
j) Duct mounted apparatus.
k) Duct mounted equipment.
l) Special duct suspension system.

A complete SMACNA duct design specification will include all of the above specification requirements in sufficient detail to indicate performance standards, materials and design methods for all ducts and duct components required for the total HVAC system.

C DUCT CONSTRUCTION MATERIALS

1. Galvanized Steel

a. APPLICATIONS

Widely used as a duct material for most air handling systems; not recommended for corrosive product handling or temperatures above 400°F (200°C).

b. ADVANTAGES

High strength, rigidity, durability, rust resistance, availability, non-porous, workability, and weldability.

c. LIMITING CHARACTERISTICS

Weldability, paintability, weight, corrosion resistance.

d. REMARKS

Galvanized steel sheet is customarily available in commercial quality, lock forming quality, drawing quality, drawing quality special killed and physical (structural) quality. The most common material used for ductwork is lock forming quality. Table 12-1 shows the chemical requirements of carbon steel prior to galvanizing. Galvanized steel sheet is produced to various zinc-coating designations to give the service life required (see Table 12-2). 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 of *both* surfaces. Galvanized sheets with the surface treated for painting by phosphatizing are commonly used.

Table 12-2 shows information on various galvanizing coatings. SMACNA originally had a specification within its duct standards calling for a 1.25 oz./sq. ft. commercial coating class. Such coating corresponds with the G90 coating designation within ASTM A525, Standard Specification for "Steel Sheet Zinc Coated (Galvanized) by the Hot-dip Process." A lighter coating (Designation G60) may be used in interior applications. Although SMACNA generally recommends G90, G60 may be considered when the duct is free from exposure to industrial pollutants, marine atmospheres or continuous contact with moisture. (See the SMACNA "Special Study Report on Galvanized Coating Thickness" for more detailed information.)

Table 12-1 CARBON STEEL CHEMICAL REQUIREMENTS (Prior to Galvanizing)

Elements	Percent
Carbon (Maximum)	0.15
Manganese (Maximum)	0.60
Phosphorous (Maximum)	0.035
Sulfur (Maximum)	0.040
Copper (when copper bearing steel is specified, minimum	0.20

2. Carbon Steel (Black Iron)

a. APPLICATIONS

Breechings, flues, stacks, hoods, other high temperature duct systems, kitchen exhaust systems, ducts requiring paint or special coating.

b. ADVANTAGES

High strength, rigidity, durability, availability, paintability, weldability, non-porous.

c. LIMITING CHARACTERISTICS

Corrosion resistance, weight.

d. 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. Obviously, 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. The chemical requirements for carbon steel, commercial quality, are shown in Table 12-1.

3. Aluminum

a. APPLICATIONS

Duct systems for moisture-laden air, louvers, special exhaust systems, ornamental duct systems. Often substituted for galvanized steel in HVAC duct systems.

b. ADVANTAGES

Weight, resistance to moisture corrosion (salt free), availability.

c. LIMITING CHARACTERISTICS

Low strength, material cost, weldability, thermal expansion.

Table 12-2 GALVANIZED STEEL SHEETS OR COIL-WEIGHT OF COATING (Total Both Sides)

			Min. Check Limit Triple-Spot Test		Min. Check Limit Single Spot Test	
Type	Coating Designation	Previous Coating Class	oz/ft^2 (of sheet)	g/m^2	oz/ft^2 (of sheet)	g/m^2
Regular	G235	2.75	2.35	717	2.00	610
	G210	2.50	2.10	640	1.80	549
	G185	2.25	1.85	564	1.60	488
	G165	2.00	1.65	503	1.40	427
	G140	1.75	1.40	427	1.20	366
	G115	1.50	1.15	351	1.00	305
	G90	1.25 Commercial	0.90	275	0.80	244
	G60	Light Commercial	0.60	183	0.50	152
	G01	. . . .	No Min . . .		No Min . . .	

NOTE: The coating designation number is the term by which this product is specified. The weight of coating in ounces per square foot of sheet refers to the total coating on both surfaces. Because of the many variables and changing conditions that are characteristic of continuous galvanizing, the weight of zinc coating is not always evenly divided between the two surfaces of a galvanized sheet; neither is the zinc coating evenly distributed from edge to edge. However, it can normally be expected that not less than 40 percent of the single-spot check limit will be found on either surface.

d. 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.

4. Stainless Steel

a. APPLICATIONS

Duct systems for kitchen exhaust, moisture laden air, fume exhaust.

b. ADVANTAGES

High resistance to corrosion from moisture and most chemicals, ability to take a high polish.

c. LIMITING CHARACTERISTICS

Labor and material cost, workability, availability.

d. REMARKS

Available in many different alloy combinations (see Table 12-3), type 304 and 316 are most commonly used. Stainless is usually available in the following finishes (as listed by ASTM):

Mill Rolled No. 1: Produced on hand sheet mills by hot rolling to specified thickness followed by annealing and descaling. 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 descaling. The dull, non-reflective finish may result from the descaling 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 which may be polished after fabrication.

Mill Rolled No. 2B: Commonly produced the same as 2D, except that the annealed and descaled 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.

Table 12-3 STAINLESS STEEL

Compositions, Mechanical and Physical Properties	AISI Type					
	302	304	304L	316	316L	410
Nominal Composition %						
Chromium	17-19	18-20	18-20	16-18	16-18	11.5-13.5
Nickel	8-10	8-10.5	8-11	10-14	10-14	—
Manganese	2.00 Max	2.0 Max	2.0 Max	2.0 Max	2.0 Max	1.0 Max
Molybdenum	—	—	—	2-3	2-3	—
Carbon	0.15 Max	0.08 Max	0.03 Max	0.06 Max	0.03 Max	0.15 Max
Representative Mechanical Properties (sheet & strip annealed)						
Tensile 1000 psi (MPa)	75(515)	75(515)	70(485)	94(650)	75(515)	65(450)
Yield 1000psi (MPa) (2.0% Offset)	30(205)	30(205)	25(170)	44(300)	30(205)	30(205)
Elongation in 2"%	40	40	40	35	40	20
Modulus of Elasticity, psi × 10^6 (GPa)	29.0(200)	29.0(200)	29(200)	29.0(200)	29(200)	29.0(200)
Hardness, Rockwell B	92	92	88	96	95	95
Physical Properties						
Weight, lb./cu.in. (kg/m^3)	0.29(8060)	0.29(8060)	0.29(8060)	0.29(8060)	0.29(8060)	0.28(7780)
Thermal Conductivity at 212°F (100°C) btu/sq.ft./hr deg F/ft (W/m . K)	9.4(0.113)	9.4(0.113)	9.4(0.113)	9.4(0.113)	9.4(0.113)	14.4(0.174)
Coefficient of Thermal Expansion in/in/deg F × 10^{-6} 32 to 212° F (cm/cm/deg C × 10^{-6} 0 to 100° C)	9.2(16.6)	9.2(16.6)	9.2(16.6)	9.2(16.6)	9.2(16.6)	5.5(9.9)

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 mesh.

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 exten-

Table 12-4 SHEET METAL PROPERTIES

		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 in)	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
Aluminum	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

sively 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.

5. Copper

a. APPLICATIONS

Duct systems for exposure to outside elements and moisture laden air, certain chemical exhaust, ornamental ductwork.

b. ADVANTAGES

Accepts solder readily, durable, resists corrosion, non-magnetic.

c. LIMITING CHARACTERISTICS

Cost, electrolysis, thermal expansion, stains.

d. REMARKS

Commonly used for ornamental systems and hoods. The various brown to green color shades (patina) formed by oxidation and exposure to moisture is found to be a desirable characteristic.

6. Fiberglass Reinforced Plastic (FRP)

a. APPLICATIONS

Chemical exhaust, scrubbers, underground duct systems.

b. ADVANTAGES

Resistance to corrosion, ease of modification.

c. LIMITING CHARACTERISTICS

Cost, weight, range of chemical and physical properties, brittleness, fabrication (necessity of molds and expertise in mixing basic materials), code acceptance.

7. Polyvinyl Chloride (PVC)

a. APPLICATIONS

Exhaust systems for chemical fumes and hospitals, underground duct systems.

b. ADVANTAGES

Resistance to corrosion, weight, weldability, ease of modification.

c. LIMITING CHARACTERISTICS

Cost, fabrication, code acceptance, thermal shock, weight.

8. Polyvinyl Steel (PVS)

a. APPLICATIONS

Underground duct systems, moisture laden air, and corrosive air systems.

b. ADVANTAGES

Resistance to corrosion, weight, workability fabrication, rigidity.

c. LIMITING CHARACTERISTICS

Susceptible to coating damage, temperature limitations (250°F or 120°C Max.), weldability, code acceptance.

d. REMARKS

Polyvinyl steel is a polyvinyl chloride plastic coating heat fused to galvanized steel. 2 mil and 4 mil coating thicknesses usually are standard, with steel gauges (US standard) available from 26 gauge thru and including 14 gauge. This product is most popular in spiral formed pipe and is available in flat sheets and coil stock of lockforming quality.

9. Concrete

a. APPLICATIONS

Underground ducts, air shafts.

b. ADVANTAGES

Compressive strength, corrosion resistance.

c. LIMITING CHARACTERISTICS

Cost, weight, porous, fabrication (requires forming processes).

10. Rigid Fibrous Glass

a. APPLICATIONS

Interior HVAC low pressure duct systems.

b. ADVANTAGES

Weight, thermal insulation and vapor barrier, acoustical qualities, ease of modification, inexpensive tooling for fabrication.

c. LIMITING CHARACTERISTICS

Cost, susceptible to damage, system pressure, code acceptance.

d. REMARKS

Joints must be properly taped (see SMACNA Fibrous Glass Duct Construction Standards).

11. Gypsum Board

a. APPLICATIONS

Ceiling plenums, corridor ducts, airshafts.

b. ADVANTAGES

Cost, availability.

c. LIMITING CHARACTERISTICS

Weight, code acceptance, leakage, deterioration when damp.

d. REMARKS

Must be sealed. Water resistance gypsum board should be used for all ductwork.

D ASTM STANDARDS

The American Society for Testing and Materials 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 which are defined in one of the 48 parts (each bound in its own volume) of ASTM. The last two digits of the number designate the year of revision, such as the "86" in A525-86. These "yearly" designations have been dropped from this manual, but the complete list can be found in "Part 48—Annual Book of ASME Standards."

The following is a partial list of ASTM specifications construction of ductwork and is intended as an informational guide only. For a complete description of these materials, the designer should refer to the appropriate ASTM Manual or other material sources.

ASTM No.	Standard Title
A 167	Stainless and Heat-Resisting—Nickel Steel Plate Sheet and Strip
A 176	Stainless and Heat-Resisting Chromium Steel Plate Sheet and Strip
A 177	High Strength Stainless & Heat Resisting Chromium-Nickel Steel Sheet and Strip
A 263	Corrosion—Resisting Chromium Steel Clad Plate, Sheet and Strip
A 264	Stainless Chromium—Nickel Steel Clad Plate, Sheet and Strip
A 308	Sheet Steel, Cold-Rolled Long Terne Coated.
A 361	Sheet Steel, Zinc Coated (Galvanized) by the Hot-Dip Process for Roofing.
A 366	Steel Carbon, Cold-Rolled Sheet, Commercial Quality.
A 412	Stainless and Heat-resisting Chromium-Nickel Manganese Steel Plate, Sheet and Strip

A 424	Steel Sheets for Porcelain Enameling.
A 446	Steel Sheet, Zinc Coated (Galvanized) by the Hot-Dip Process, Physical (Structural) Quality
A 463	Steel Sheet, Cold-Rolled, Aluminum-Coated Type 1.
A 480	Flat-Rolled Stainless & Heat-Resisting Steel Plate, Sheet and Strip.
A 505	Hot-Rolled & Cold-Rolled Steel Sheet and Strip Alloy.
A 506	Steel Sheet and Strip, Alloy, H.R. and C.R., Regular Quality.
A 525	Steel Sheet, Zinc-Coated (Galvanized) by the Hot-Dip Process.
A 526	Steel Sheet, Zinc-Coated (Galvanized) by the Hot-Dip Process, Commercial Quality.
A 527	Steel Sheet, Zinc-Coated (Galvanized) by the Hot-Dip Process, Lock Forming Quality.
A 568 A 568M	General Requirements for Steel, Carbon & Hi-strength, Low Alloy, Hot-Rolled Sheet, H.R. Strip and Cold-Rolled Sheet. (M = metric)
A 569	Steel, Carbon (0.15 Max. Percent) Hot-Rolled Sheet and Strip, Commercial Quality.
A 570	H.R. Carbon Steel Sheet and Strip, Structural Quality.
A 591	Steel Sheets, Cold-Rolled, Electrolytic Zinc-Coated.
A 599	Steel Sheet, Cold-Rolled, Tin Coated by Electrodeposition.
A 606	Steel Sheet and Strip, Hot-Rolled and Cold-Rolled, High Strength, Low-Alloy with Improved Corrosion Resistance.
A 607	Steel Sheet and Strip, H.R. & C.R., High Strength, Low-Alloy Columbium and/or Vanadium.
A 611	Steel, Cold-Rolled Sheet, Carbon, Structural.
B 36	Brass Plate, Sheet, Strip and Rolled Bar.
B 101	Lead-Coated Copper Sheets.
B 152	Copper Sheet, Strip, Plate and Rolled Bar.
B 209 B 209M	Aluminum-Alloy Sheet and Plate. (M = metric)
B 370	Copper Sheet & Strip for Building Construction.
B 506	Copper Clad, Stainless Steel Sheet and Strip for Building Construction.
C 14	Concrete Sewer, Storm Drain and Culvert Pipe.
C 94	Ready-Mixed Concrete.
C 700	Vitrified Clay Pipe, Extra Strength, Standard Strength and Perforated.
D 1927	Rigid Poly (Vinyl Chloride) (PVC) Plastic Sheet
D 2123	Rigid Poly (Vinyl chloride-Vinyl Acetate) Plastic Sheet
D 2241	Poly (Vinyl Chloride) (PVC) Plastic Pipe (SDRPR).

CHAPTER 13

SPECIAL DUCT SYSTEMS

Ductwork requiring special attention is sometimes encountered by the designer. It is of utmost importance that the duct designer be aware of the different requirements existing in regards to the geographical area of the installation.

Before design of special systems, the designer must acquaint himself thoroughly with local practices and concerned governing authorities.

Industrial process or material handling systems are appropriately covered in other SMACNA publications and will not be considered herein. This section contains a general description of some of the special duct systems frequently encountered in HVAC work.

A KITCHEN AND MOISTURE LADEN SYSTEMS

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. The temperature of the vapor may be excessively high and, therefore, may require the use of duct insulation or other treatment. All duct penetrations should be avoided.

2. Range and Grease Hood Exhaust Ducts

Vapors from cooking equipment must be exclusively handled through ducts designed specifically for that purpose. Care must be taken to assure that these ducts will contain fire and smoke. Materials used must be heavier than standard and are usually 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 such that grease cannot be trapped and the duct should be sloped toward the hood or a grease reservoir. Ducts within the building should lead as directly as possible to the exterior. Where ducts pass through combustible walls, partitions, etc., adequate clearance or protection must be provided. In the event of a fire, temperatures in excess of 2000°F (1100°C) may be experienced. Fire extinguishing systems may be required by local codes. Long, straight runs of duct should have a means 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 96.

B SYSTEMS HANDLING SPECIAL GASES

1. Corrosive Vapors and Noxious Gases

Ducts which convey these gases should be fabricated from materials impervious to all the gases that may be handled, and must be sealed air tight. They must terminate outside the building, maintaining adequate 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 conditions.

2. Flammable Vapors

Ducts conveying these vapors must be sealed air tight and terminate outside the building, maintaining adequate clearances from building construction and other objects. Nonflammable materials must be used for the ducts and duct supports.

C SOLAR SYSTEMS

1. Solar System Sizing

Successful application of solar heating systems requires careful selection and sizing of components. Collectors, heat storage units, fans and pumps, controls, heat exchangers, and auxiliary heaters must be effectively integrated. Unlike the selection of a furnace or boiler, a solar space heating system may be sized to provide a selected portion of the annual heating load. Generally 30 to 70 percent is reasonable. The size of the solar system basically depends on the collector area. The collector area then determines the quantity of solar heat delivery or the amount of fossil fuel savings.

Guidelines for sizing components of integrated air-based solar systems for space and potable water heating are listed in Table 13-1. A typical arrangement for which the guidelines apply is shown in Figure 13-1.

2. Duct System Layout

A layout of the duct distribution system should be prepared and sizing of all ductwork should be accomplished using the method that the designer is most comfortable with for the air volume required. However, the designer shall be responsible for correctly sizing the duct system so that its total external static pressure (ESP) shall not exceed the manufacturer's ESP rating for the air handling equipment.

Ducts connecting solar air collector inlets and outlets shall be sized to meet the air quantities that are required by the airflow characteristics of the collector. Review the collector manufacturer's literature to determine the correct flow rates. Connections to the collectors shall be in accordance with the manufacturer's recommendations.

When auxiliary heating equipment is used, the airflow volume of the duct distribution system must provide an air temperature rise through the equipment that is below the maximum temperature rise noted on the equipment nameplate.

Table 13-1 GUIDELINES FOR SIZING COMPONENTS OF AIR-BASED SOLAR SYSTEMS FOR SPACE AND POTABLE WATER HEATING

COMPONENT	RECOMMENDED VALUE	RANGE
Collector		
Slope*	Lat + 15°	Lat to Lat + 30°
Orientation	South	30° SE to 30° SW
Air Flow Rate per ft.^{2} (m^2) of collector	2 cfm (11 l/s)	1.5 to 3 cfm (8 to 15 l/s)
Pebble-Bed Storage:**		
Volume per ft.^{2} (m^2) of collector	0.75 ft.^{3} (0.021m^3)	0.5 to 1.0 ft.^{3} (0.014 to 0.028 m^3)
Depth (in flow direction)	6 ft. (2 m)	4 to 8 ft. (1.2 to 2.4 m)
Pebble Size	0.75 to 1.5 in. (19 to 38mm)	0.5 to 1.0 in.; 1.0 to 2.0 in. (12 to 25mm; 25 to 50mm)
Primary Solar Duct System		
Air Velocity	600 fpm (3 m/s)	500 to 900 fpm (2.5 to 4.5 m/s)
Pressure Drops:		
Collectors—12-14 ft. (3.6-4.2 m) Lengths	0.2 in. w.g. to 0.3 in. w.g. (50 Pa to 75 Pa)	—
Collectors—18-20 ft. (5.5-6.1 m) Lengths	0.3 in. w.g. to 0.5 in. w.g. (75 Pa to 125 Pa)	—
Pebble Bed	0.1 in. w.g. to 0.3 in. w.g. (25 Pa to 75 Pa)	—
Ducts	0.08 in. w.g./100 ft. length (0.65 Pa/m)	—

*For potable water heating *only* the collector slope should be at latitude angle, and the recommended range is Lat −5° to Lat +5°

**For potable water heating *only* systems, pebble bed storage is not required.

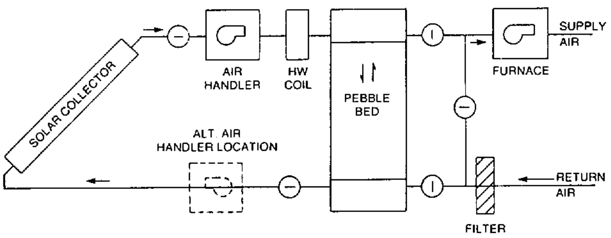


Figure 13-1 TYPICAL AIR-HEATING SYSTEM

3. Solar Collecting Systems

The duct system between the solar collectors and the thermal storage containers, and the ductwork connecting to the space distribution system shall be known as the *primary solar duct system* (PSDA). The PSDA shall be designed using the criteria described above. Care shall be used to assure balanced airflow in the PSDS for the various operational modes of the system.

All ducts and duct linings composing the PSDS shall be installed in strict conformance with the SMACNA HVAC Duct Construction Standards and Fibrous Glass Duct Construction Standards. All materials used in the PSDS shall be able to withstand temperatures up to 250°F (121°C) without degradation or release of odor-causing or noxious gasses.

Air leakage from PSDS should not exceed 5 percent. It is not the intent of these Standards to test the PSDS for compliance with the 5 percent duct leakage requirement, but simply to assure construction standards which will essentially provide the required degree of air tightness in the PSDS.

Ducts may be sealed using mastic, or mastic plus tape or gasketing as appropriate. The selection of the most appropriate sealant depends on joint configuration, clearances, surface conditions, temperature, the direction of pressure and preassembly or post assembly placement. Tapes should not be applied to bare metal nor to dry sealant. Foil tapes are not suitable. Liquids and mastics should be used in well-ventilated areas and the precautions of manufacturers followed. Oil base caulking and glazing compounds should not be used. Gasketing should be material with long life and suitable for the service.

4. Solar System Dampers

a. CONTROL DAMPERS (Motorized)

Dampers that open or close to divert, direct, or shut-off airflow in the Primary Solar Duct System shall have "sealing" edges on the blades with a suitable material such as felt, rubber, etc., to insure tight cutoff of the air stream when closed.

b. SHUT-OFF DAMPERS (Not Motorized)

Shut-off dampers installed to prevent air flow, as in the summer by-pass duct in the Primary Solar Duct System, shall be sealed tightly to prevent air flow when pressurized from either side of the damper. Slide dampers shall have suitable seals on the guides to prevent leakage around the blade and through the guide.

Table 13-2 SOLAR AIR DISTRIBUTION SYSTEMS

	Recommended Velocity		Maximum Velocity	
Designation	fpm	m/s	fpm	m/s
Main Trunk Ducts	700-900	3.5-4.5	1000	5.0
Branch Ducts	600	3.0	800	4.0
Branch Riser	500	2.5	800	4.0
Outdoor Air Intake	500	2.5	600	3.0
Return Air Ducts	700	3.5	900	4.5
Air Collector Manifold Ducts	700-900	3.5-4.5	1000	5.0
Air Collector Riser Ducts ..	800	4.0	1000	5.0

c. VOLUME DAMPERS

Volume control of balancing dampers shall be installed in each branch or zone duct. Single leaf dampers which are a part of a manufactured air grille do not meet the requirements of the SMACNA solar installation standards found in the SMACNA "Installation Standards for Residential Heating and Air Conditioning Systems." Opposed blade dampers which are a part of a manufactured air grille meet the requirements of the Standards if sufficient space is provided behind the grille face for proper operation of the damper.

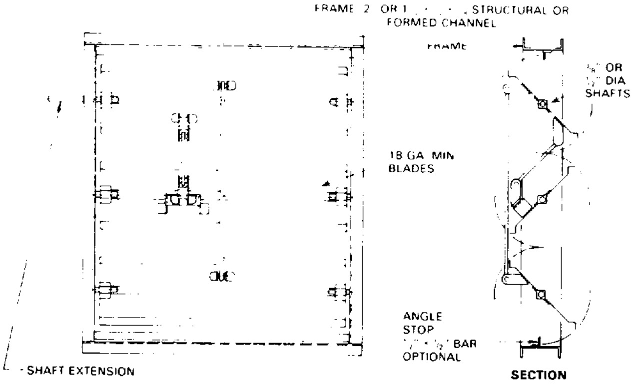


FIG. A
OPPOSED ACTION

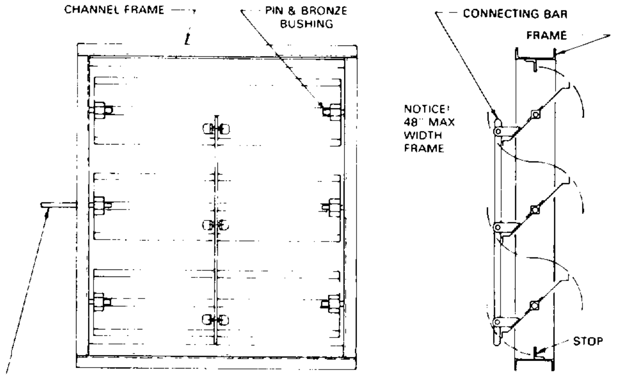


FIG. B
PARALLEL ACTION

SEE TEXT ON VOLUME DAMPERS

SECTION
FIG. C

Figure 13-2 MULTI-BLADE VOLUME DAMPERS

Where space prohibits the use of an opposed blade damper behind the grille face, an opposed blade damper may be installed in the register stack at a location where it is accessible from the grille opening.

Volume dampers installed in branch ducts where the total estimated system static pressure is less than 0.5 in. w.g. (125 Pa) should be of a single leaf type. Volume dampers installed in ductwork where the total estimated system static pressure exceeds 0.5 in. w.g. (125 Pa) shall be manufactured in accordance with Figure 13-2.

d. BACK-DRAFT DAMPERS

Back draft dampers shall be installed to close under the action of gravitational force when there is no air flow, and open when there is a drop in pressure across the damper in the direction of desired air flow. Multi-bladed back-draft dampers shall have suitable seals on the blade edges, and appropriate seals along the sides. Light-weight rubberized fabric dampers of the type shown in Figure 13-3 shall be tilted sufficiently to ensure closure when there is no airflow. Single blade dampers shall have seals along the seat and the pivot shall be off-center and horizontal to ensure closure when there is no airflow.

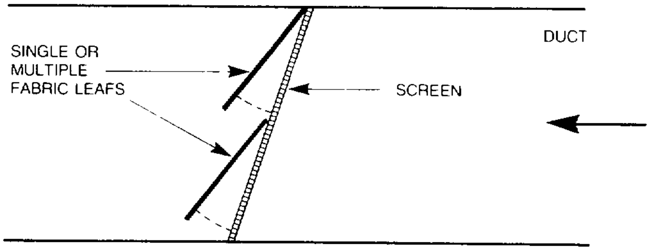


Figure 13-3 RUBBERIZED FABRIC BACK-DRAFT DAMPER

CHAPTER 14
DUCT DESIGN TABLES AND CHARTS

I INTRODUCTION

This chapter of the "HVAC Systems—Duct Design" manual combines all of the recognized basic duct system design tables and charts into an easy-to-use, single source. 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.

New in this revised edition, are many duct fitting loss coefficient tables and a new duct friction loss chart developed as a result of research data funded by SMACNA in an extensive research program at the ETL Testing Laboratories, Inc. in Cortland, New York and by ASHRAE research, partially funded by SMACNA.

Chapter 5—"Duct Design Fundamentals", Chapter 7—"Duct Sizing Procedures (U.S. Units)", and Chapter 8— "Duct Sizing Procedures (Metric Units)" show how to use the various duct design tables and charts. The step-by-step examples use the prescribed procedures for designing basic duct systems.

Some of the tables come from unconfirmed sources. A world-wide 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 coefficient tables (recently corrected with newer data) conflict with existing tables used in segments of the industry, the consensus of members of the SMACNA and ASHRAE Duct Design Committees is that the data is reasonable to use for the design of HVAC duct systems. Therefore these tables, which are contained in this chapter and in the ASHRAE 1989 "Fundamentals" Handbook, should continue to be used until current and future research programs can validate the figures or establish new verified data. Some of the most recent duct fitting loss coefficient data from limited SMACNA research also may be found in Chapter 5, Section H—"SMACNA Duct Research."

The duct fitting loss coefficients used to calculate the resistance to flow are in terms of *total pressure.* When static regain occurs, it need not be addressed separately because it is included in the fitting loss coefficient and therefore in the calculation. When these values are added to the calculated friction loss of the straight duct sections, the total system resistance (pressure drop) will be in terms of *Total Pressure.*

W. David Bevirt, P.E.

Numbers in parentheses, 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.

II TABLE OF CONTENTS (CHAPTER 14)

A Duct Friction Loss—Tables and Charts

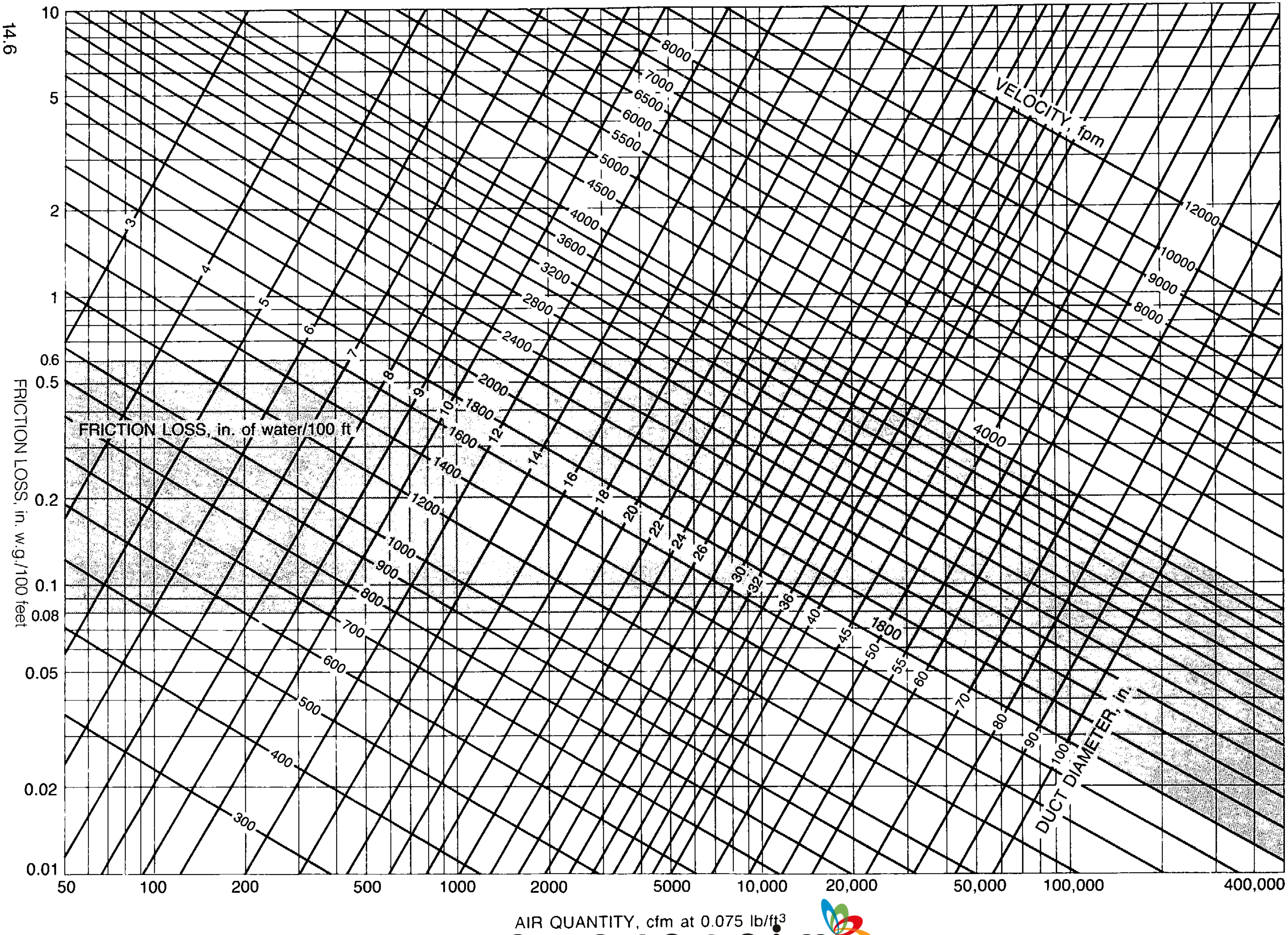


Figure 14-1 DUCT FRICTION LOSS CHART (U.S. Units) (2)

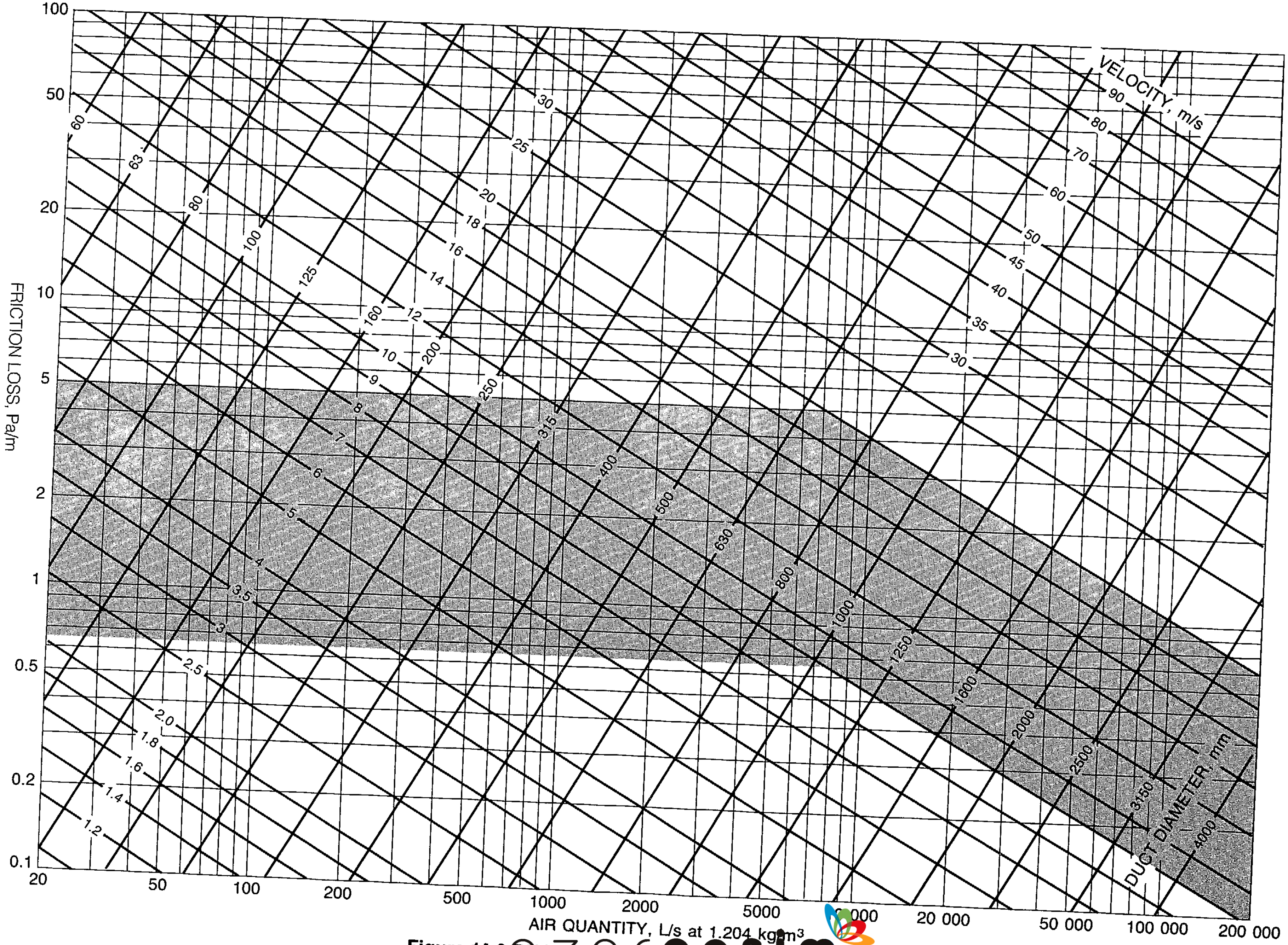


Figure 14-2 DUCT FRICTION LOSS CHART (Metric Units) (2)

Table 14-1 DUCT MATERIAL ROUGHNESS FACTORS

		Absolute Roughness ε,	
Duct Material	Roughness Category	ft	mm
Uncoated carbon steel, clean (Moody 1944) (0.00015 ft) (0.05 mm)	Smooth	0.0001	0.03
PVC plastic pipe (Swim 1982) (0.0003 to 0.00015 ft) (0.01 to 0.05 mm)			
Aluminum (Hutchinson 1953) (0.00015 to 0.0002 ft) (0.04 to 0.06 mm)			
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 3 ribs, 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, 2.5 ft (760 mm) joints (Wright 1945) (0.0005 ft) (0.15 mm)	Old Average	0.0005	0.15
Fibrous glass duct, rigid	Medium rough	0.003	0.9
Fibrous glass duct liner, air side with facing material (Swim 1978) (0.005 ft) (1.5 mm)			
Fibrous glass duct liner, air side spray coated (Swim 1978) (0.015 ft) (4.5 mm)	Rough	0.01	3.0
Flexible duct, metallic, (0.004 to 0.007 ft (1.2 to 2.1 mm) when fully extended)			
Flexible duct, all types of fabric and wire (0.0035 to 0.015 ft (1.0 to 4.6 mm) when fully extended)			
Concrete (Moody 1944) (0.001 to 0.01 ft) (0.3 to 3.0 mm)			

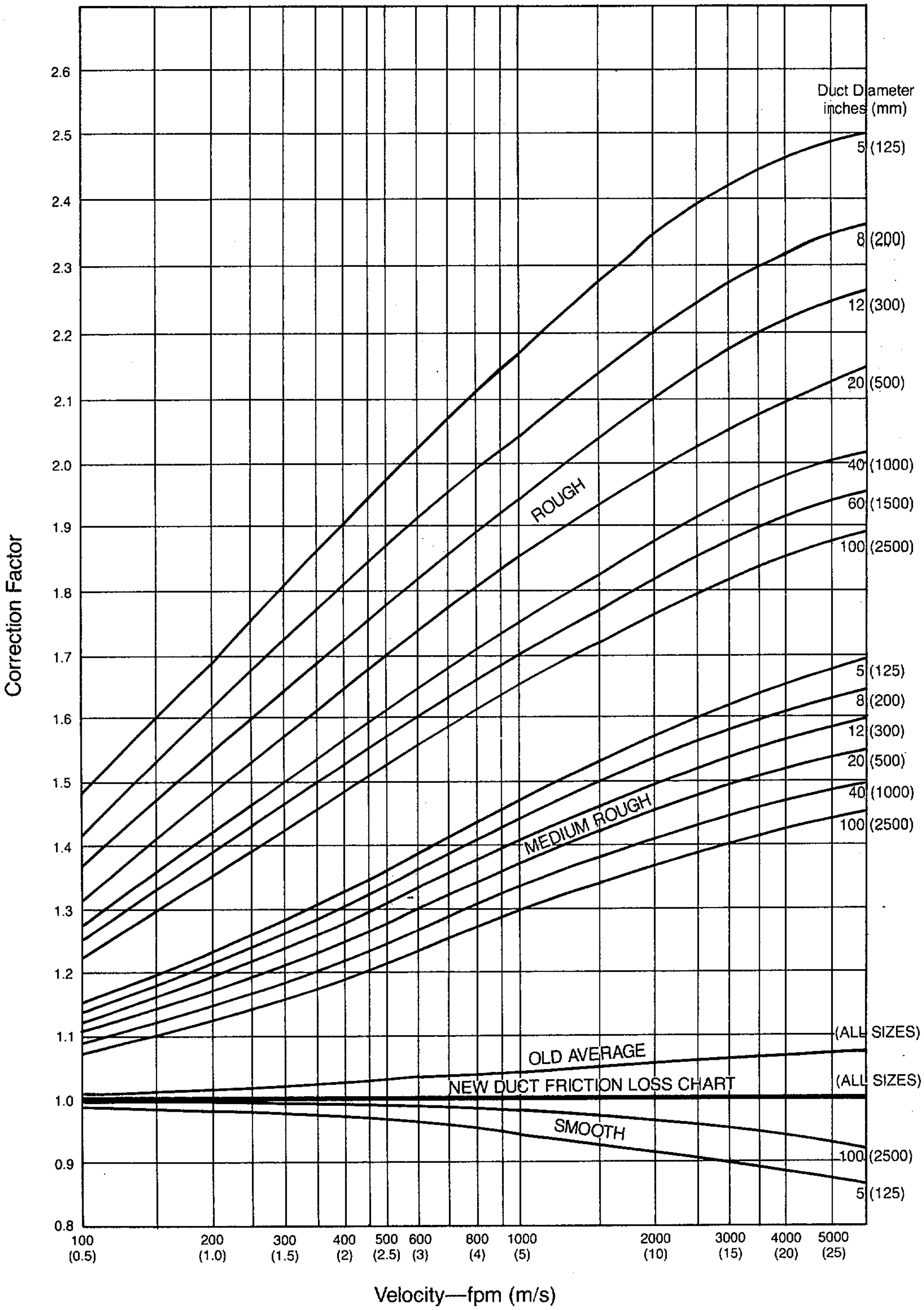


Figure 14-3 DUCT FRICTION LOSS CORRECTION FACTORS

Table 14-2 CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY (U.S. Units) (2)

Dimensions in Inches

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	21.4	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

Cont. Table 14-2 CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY (U.S. Units) (2)

Dimensions in Inches

Side Rectangular Duct	32	34	36	38	40	42	44	46	48	50	52	56	60	64	68	72	76	80	84	88	Side Rectangular 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	53.9	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	58.4	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	98.1	100.5	96

Equation for Circular Equivalent of a Rectangular Duct:

$$D_e = 1.30\,[(ab)^{0.625}/(a+b)^{0.250}]$$

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.

Table 14-3 CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY (Metric Units) (2)

Dimensions in mm

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	944	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	1107	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

Cont. Table 14-3 CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY (Metric Units) (2)

Dimensions in mm

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	1146	1202																			1100
1200	1196	1256	1312																		1200
1300	1244	1306	1365	1421																	1300
1400	1289	1354	1416	1475	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

Equation for Circular Equivalent of a Rectangular Duct:

$$D_e = 1.30 \, [(ab)^{0.625}/(a + b)^{0.250}]$$

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.

Table 14-4 SPIRAL FLAT-OVAL DUCT (Nominal Sizes—U.S. Units)
(Diameter of the round duct which will have the capacity and friction equivalent to the actual duct size.)

	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	14.7			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						19.1					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						16.6					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											25.3			30.3			34.3					39.8		
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			46.8			50.4
60								24.4			29.6	31.8		36.1	37.8		41.3		44.5	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			38.0		42.0			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 14-5 SPIRAL FLAT-OVAL DUCT (Nominal Sizes—Metric Units)
(Diameter of 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						485					650									
725								450				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			1189			1280
1500								620			752	808		917	960		1049		1130	1166		1232	1262	
1550										699			876			1018		1105			1222			1316
1600								638			775	830		942	988		1079		1168			1275		
1650								645		719		846	899		1006			1138		1219	1262		1331	
1700										726	792			965		1067			1196			1318		1384
1750								660			800	866	919		1031		1128	1168		1255	1293		1374	
1800								668		742									1234		1311	1379		1427

Table 14-6 VELOCITIES/VELOCITY PRESSURES (U.S. Units)

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

$$\text{Velocity} = 4005\sqrt{V_p}\ \text{(or)}\ V_p = \left(\frac{\text{Velocity}}{4005}\right)^2$$

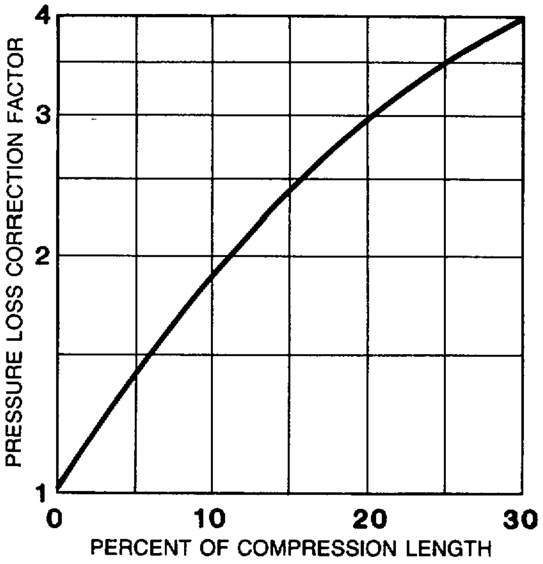


Figure 14-4 CORRECTION FACTOR FOR UNEXTENDED FLEXIBLE DUCT (2)

Table 14-7 VELOCITIES/VELOCITY PRESSURES (Metric Units)

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 14-8 ANGULAR CONVERSION

Degrees	Radians
10°	0.175
20°	0.349
30°	0.524
40°	0.698
50°	0.873
60°	1.05
70°	1.22
80°	1.40
90°	1.57 ($\pi/2$)
135°	2.36
180°	3.14 (π)
360°	6.28 (2π)

Table 14-9 LOSS COEFFICIENTS FOR STRAIGHT-THROUGH FLOW

$\frac{V_s}{V_c}$	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 (System) Velocity
V_c = Upstream (System) Velocity

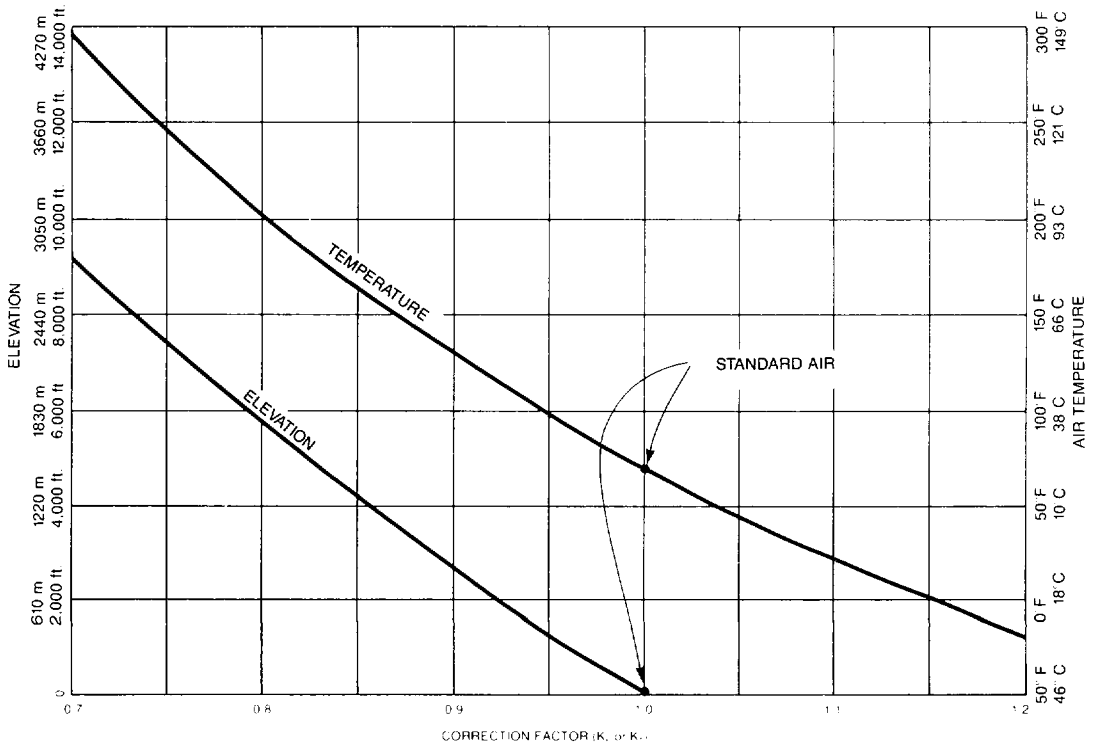


Figure 14-5 AIR DENSITY FRICTION CHART CORRECTION FACTORS

When an air distribution system is designed to operate above 2000 feet (610 m) altitude, below 32°F (0°C), or above 120°F (49°C) temperature, the duct friction loss obtained must be corrected for the air density. The actual airflow (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.

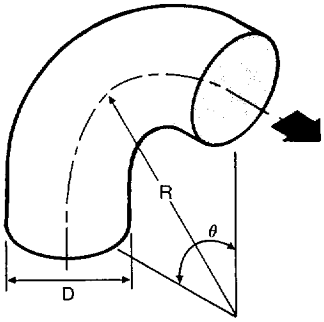
B FITTING LOSS COEFFICIENT TABLES

Duct Cross Section to which Coefficient "C" is referenced is at the top of each table. Negative numbers indicate that the static regain exceeds the dynamic pressure loss of the fitting.

Table 14-10 LOSS COEFFICIENTS, ELBOWS

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

A. Elbow, Smooth Radius (Die Stamped), Round (2)



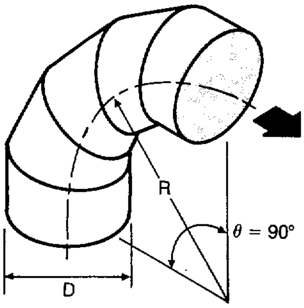
Coefficients for 90° Elbows: (See Note 1)

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:

θ	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

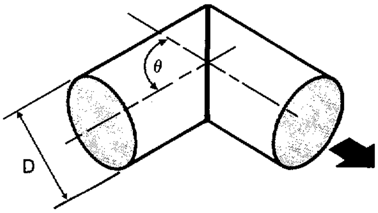
B. Elbow, Round, 3 to 5 pc — 90° (2)



Coefficient C

No. of Pieces	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 (15)



Coefficient C (See Note 2)

θ	20°	30°	45°	60°	75°	90°
C	0.08	0.16	0.34	0.55	0.81	1.2

Note 2: Correction factor for Reynolds number — K_{Re}

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
For Standard Air:
$R_e = 8.56$ DV

(Metric Units)
($R_e = 66.4$ DV)

where:
D = duct diameter, inches (mm)
V = duct velocity, fpm (m/s)

For Rectangular Ducts:

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

Reynolds Number Correction Factor Chart

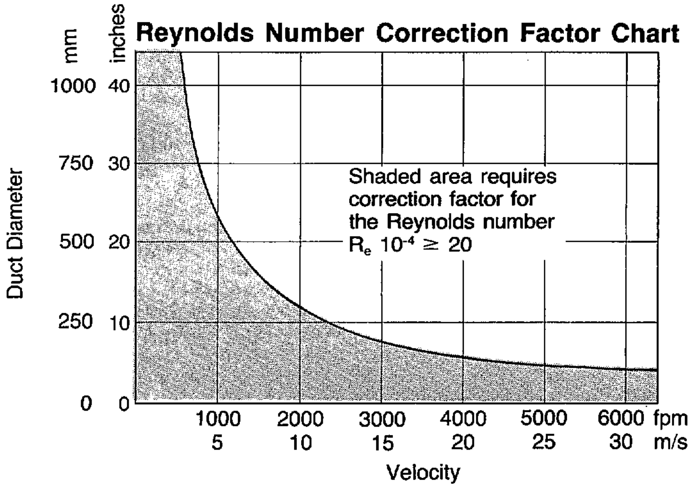
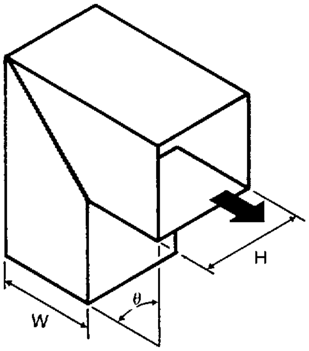


Table 14-10 LOSS COEFFICIENTS, ELBOWS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

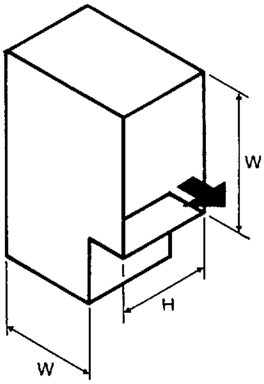
D. Elbow, Rectangular, Mitered (15)



Coefficient C (See Note 2—Page 14.19)

θ	H/W										
	0.25	0.5	0.75	1.0	1.5	2.0	3.0	4.0	5.0	6.0	8.0
20°	0.08	0.08	0.08	0.07	0.07	0.07	0.06	0.06	0.05	0.05	0.05
30°	0.18	0.17	0.17	0.16	0.15	0.15	0.13	0.13	0.12	0.12	0.11
45°	0.38	0.37	0.36	0.34	0.33	0.31	0.28	0.27	0.26	0.25	0.24
60°	0.60	0.59	0.57	0.55	0.52	0.49	0.46	0.43	0.41	0.39	0.38
75°	0.89	0.87	0.84	0.81	0.77	0.73	0.67	0.63	0.61	0.58	0.57
90°	1.3	1.3	1.2	1.2	1.1	1.1	0.98	0.92	0.89	0.85	0.83

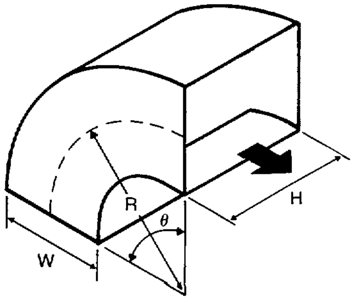
E. Elbow, Rectangular, Mitered with Converging or Diverging Flow (15)



Coefficient C (See Note 2—Page 14.19)

H/W	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	1.4	1.0	0.95	0.90	0.84
4.0	1.5	1.1	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 (15)



Coefficients for 90° elbows: (See Note 1)

Coefficient C (See Note 3)

R/W	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

Note 3: Correction Factor for Reynolds number — K_{Re}

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

U.S. Units
For Standard Air:
$R_e = 8.56\ DV$

(Metric Units)
($R_e = 66.4\ DV$)

where:
D = duct diameter, inches (mm)
V = duct velocity, fpm (m/s)

For Rectangular Ducts:

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

Table 14-10 LOSS COEFFICIENTS, ELBOWS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

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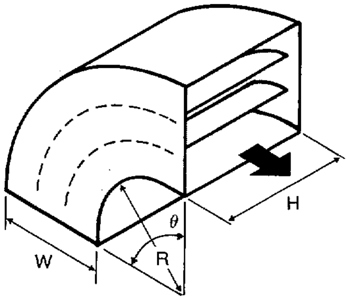
G. Elbow, Rectangular, Smooth Radius with Splitter Vanes (2)



NOTES FOR THIS FIGURE ONLY:

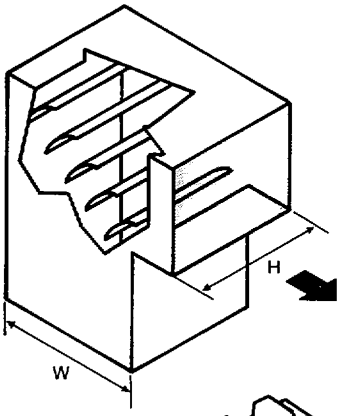
A) See Page 5.14 to calculate splitter vane spacing.
B) CR = Curve Ratio
C) Use correction factors in Note 1 on Page 14.19 for elbows other than 90°.

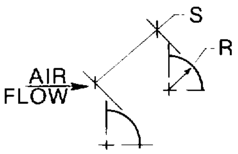
			Coefficient C										
	R/W	CR	H/W										
			0.25	0.5	1.0	1.5	2.0	3.0	4.0	5.0	6.0	7.0	8.0
1 splitter vane	0.05	0.218	0.52	0.40	0.43	0.49	0.55	0.66	0.75	0.84	0.93	1.0	1.1
	0.10	0.302	0.36	0.27	0.25	0.28	0.30	0.35	0.39	0.42	0.46	0.49	0.52
	0.15	0.361	0.28	0.21	0.18	0.19	0.20	0.22	0.25	0.26	0.28	0.30	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	0.447	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	0.02	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

Table 14-10 LOSS COEFFICIENTS, ELBOWS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

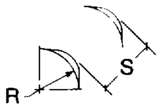
H. Elbow, Rectangular, Mitered with Turning Vanes (See Chapter 5, Sections E and H for additional information and data.)

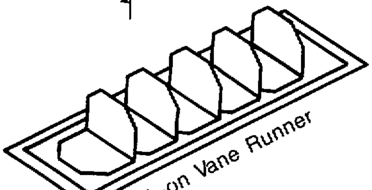


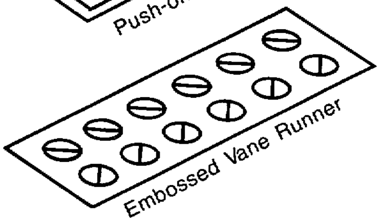


Loss Coefficients (C) for Single Thickness Vanes

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



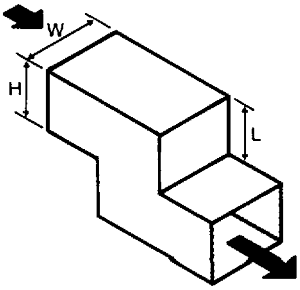




Loss Coefficients (C) for Double Thickness Vanes

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. Elbows, 90°, Rectangular, Z-Shaped (15)



(NO 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	∝
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	1.26	1.19	1.14	1.09	1.06	1.04	1.0

U.S. Units

For Standard Air: **(Metric Units)**

R_e = 8.56 D V (R_e = 66.4 DV)

where:

D = duct diameter, inches (mm)

V = duct velocity, fpm (m/s)

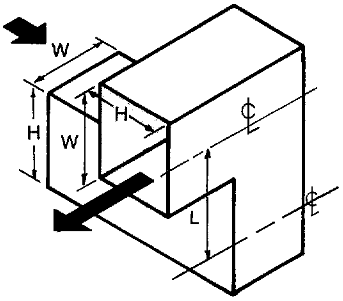
For Rectangular Ducts:

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

Table 14-10 LOSS COEFFICIENTS, ELBOWS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

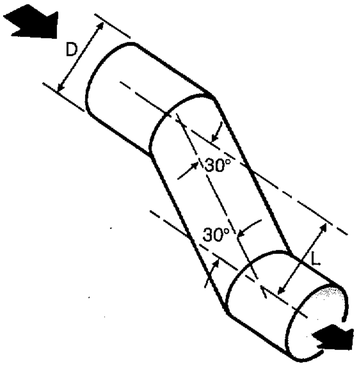
J. Elbows, 90°, Rectangular in Different Planes (15)
(See Chapter 5, Section H for new data on spin-in fittings)



Coefficients for H/W = 1.0: (See Notes 4 & 5—Page 14.22)

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	∞
C	3.2	3.2	3.2	3.0	2.9	2.8	2.7	2.5	2.4	2.3

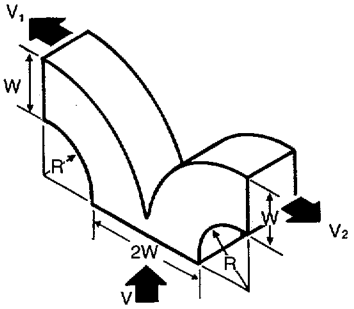
K. Elbows, 30°, Round, Offset (15)

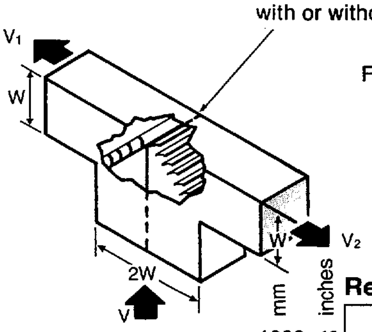


Coefficients C (See Note 5—Page 14.22)

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





Flow can be diverging or converging

Use value of C for a similar elbow, provided V = V_1 = V_2 (Velocities)

Reynolds Number Correction Factor Chart

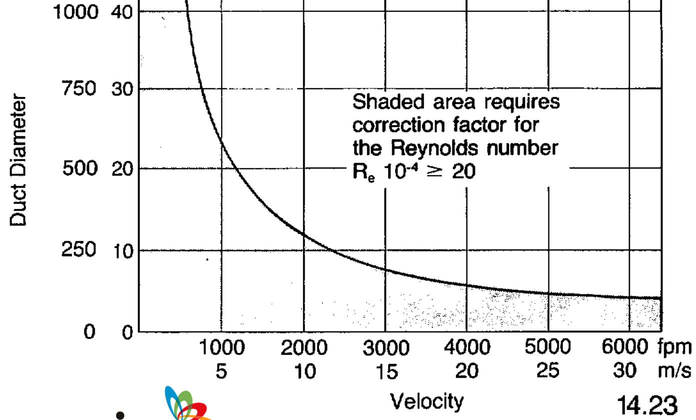
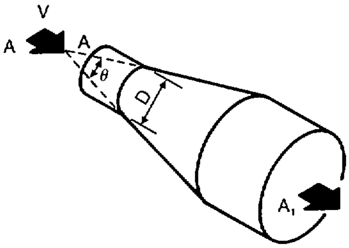
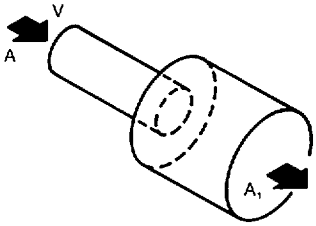


Table 14-11 LOSS COEFFICIENTS, TRANSITIONS (Diverging Flow)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

A. Transition, Round, Conical (15)





When: θ = 180°

U.S. Units R_e = 8.56 DV

(Metric Units) (R_e = 66.4 DV)

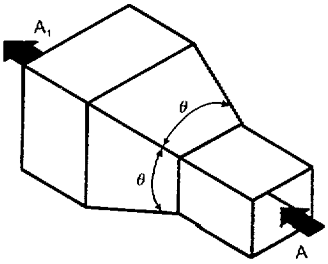
where:

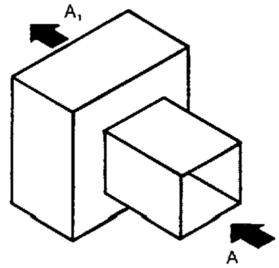
D = Upstream Diameter, inches (mm)
V = Upstream Velocity, fpm (m/s)

Coefficient C (See Note 6)

R_e	A_1/A	θ							
		16°	20°	30°	45°	60°	90°	120°	180°
0.5 × 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 × 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	0.87
≥6 × 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 (15)





When θ = 180°

Coefficient C (See Note 6)

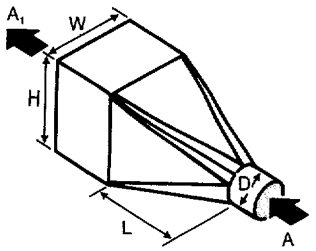
A_1/A	θ							
	16°	20°	30°	45°	60°	90°	120°	180°
2	0.18	0.22	0.25	0.29	0.31	0.32	0.33	0.30
4	0.36	0.43	0.50	0.56	0.61	0.63	0.63	0.63
6	0.42	0.47	0.58	0.68	0.72	0.76	0.76	0.75
≥10	0.42	0.49	0.59	0.70	0.80	0.87	0.85	0.86

Note 6: A = Area (Entering airstream), A_1 = Area (Leaving airstream)

Table 14-11 LOSS COEFFICIENTS, TRANSITIONS (Diverging Flow) (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

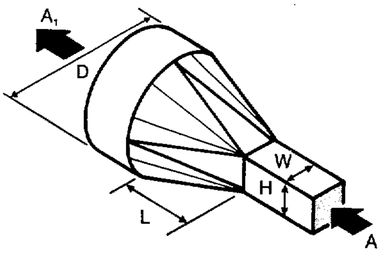
C. Transition, Round to Rectangular (15)



$$\tan\left(\frac{\theta}{2}\right) = \frac{1.13\sqrt{HW} - D}{2L}$$

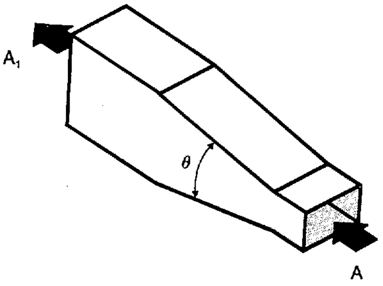
D. Transition, Rectangular to Round (15)

For Figures C and D, compute θ using the equations and find the coefficient C from Table 14-11B, Transition, Rectangular, Pyramidal.



$$\tan\left(\frac{\theta}{2}\right) = \frac{D - 1.13\sqrt{HW}}{2L}$$

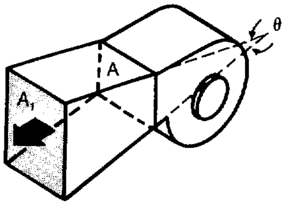
E. Transition, Rectangular, Sides Straight (15)



Coefficient C (See Note 6)

A_1/A	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 (15)



Coefficient C (See Note 6)

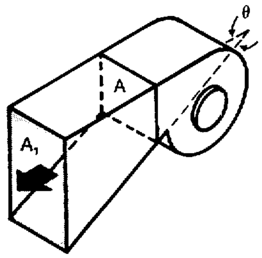
A_1/A	1.5	2.0	2.5	3.0	3.5	4.0
θ						
10°	0.05	0.07	0.09	0.10	0.11	0.11
15°	0.06	0.09	0.11	0.13	0.13	0.14
20°	0.07	0.10	0.13	0.15	0.16	0.16
25°	0.08	0.13	0.16	0.19	0.21	0.23
30°	0.16	0.24	0.29	0.32	0.34	0.35
35°	0.24	0.34	0.39	0.44	0.48	0.50

Note 6: A = Area (Entering airstream), A_1 = Area (Leaving airstream)

Table 14-11 LOSS COEFFICIENTS, TRANSITIONS (Diverging Flow) (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

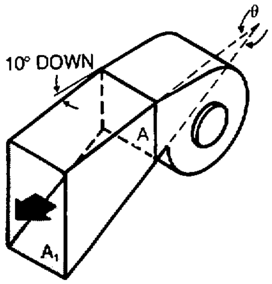
G. Transition, Asymmetric at Fan With Duct Sides Straight, Top Level (15)



Coefficient C (See Note 6)

A1/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

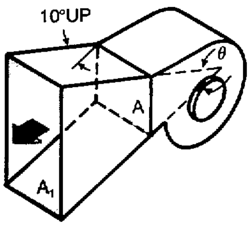
H. Transition, Asymmetric at Fan With Duct Sides Straight, Top 10° Down (15)



Coefficient C (See Note 6)

A1/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

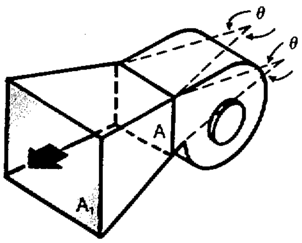
I. Transition, Asymmetric at Fan With Duct Sides Straight, Top 10° Up (15)



Coefficient C (See Note 6)

A1/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 (15)



Coefficient C (See Note 6)

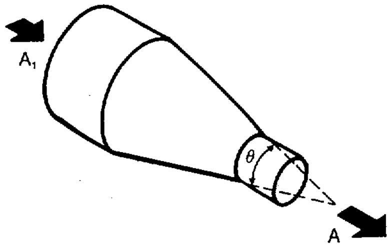
A1/A	1.5	2.0	2.5	3.0	3.5	4.0
θ						
10°	0.10	0.18	0.21	0.23	0.24	0.25
15°	0.23	0.33	0.38	0.40	0.42	0.44
20°	0.31	0.43	0.48	0.53	0.56	0.58
25°	0.36	0.49	0.55	0.58	0.62	0.64
30°	0.42	0.53	0.59	0.64	0.67	0.69

Note 6: A = Area (Entering airstream), A_1 = Area (Leaving airstream)

Table 14-12 LOSS COEFFICIENTS, TRANSITIONS (Converging Flow)

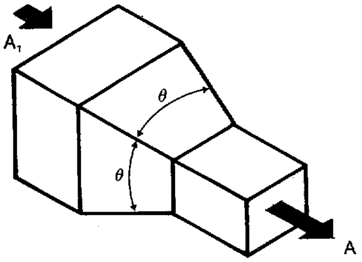
Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

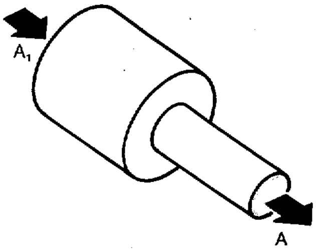
A. Contraction, Round and Rectangular, Gradual to Abrupt (15)



Coefficient C (See Note 7)

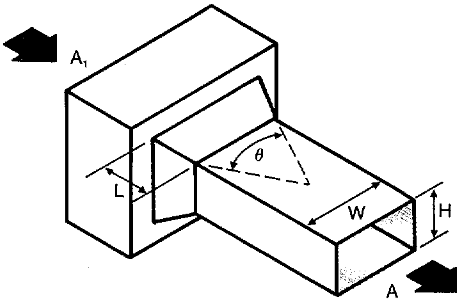
A_1/A	θ						
	10°	15°-40°	50°-60°	90°	120°	150°	180°
2	0.05	0.05	0.06	0.12	0.18	0.24	0.26
4	0.05	0.04	0.07	0.17	0.27	0.35	0.41
6	0.05	0.04	0.07	0.18	0.28	0.36	0.42
10	0.05	0.05	0.08	0.19	0.29	0.37	0.43





When θ = 180°

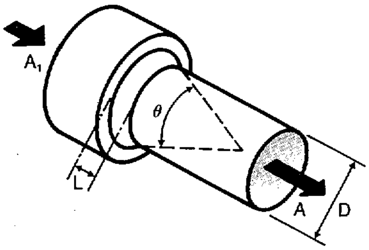
B. Contraction, Conical, Round and Rectangular (15)



For Rectangular: $D = \frac{2\,HW}{H + W}$

Coefficient C = K C_o

θ is major angle for rectangular transition



(See Note 7)

A/A_1	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

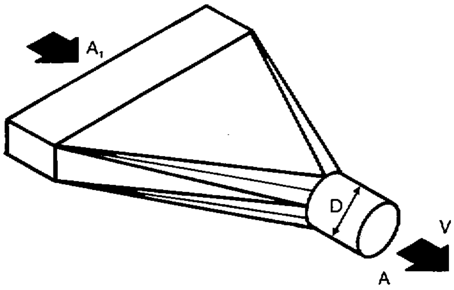
L/D	θ								
	0°	10°	20°	30°	40°	60°	100°	140°	180°
0.025	0.50	0.47	0.45	0.43	0.41	0.40	0.42	0.45	0.50
0.05	0.50	0.45	0.41	0.36	0.33	0.30	0.35	0.42	0.50
0.075	0.50	0.42	0.35	0.30	0.26	0.23	0.30	0.40	0.50
0.10	0.50	0.39	0.32	0.25	0.22	0.18	0.27	0.38	0.50
0.15	0.50	0.37	0.27	0.20	0.16	0.15	0.25	0.37	0.50
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)

Table 14-12 LOSS COEFFICIENTS, TRANSITIONS (Converging Flow) (Cont.)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

C. Contraction, Rectangular Slot to Round (15)



$A < A_1$

R_e = 8.56 DV (U.S. Units)
R_e = 66.4 DV (Metric Units)

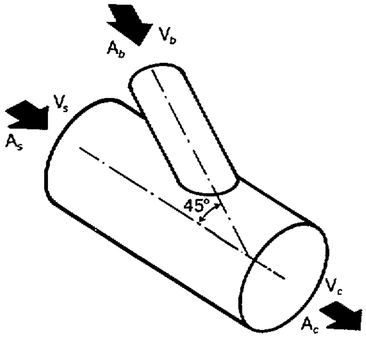
Coefficient C (See Note 7)

R_e 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

Table 14-13 LOSS COEFFICIENTS, CONVERGING JUNCTIONS (Tees, Wyes)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

A. Converging Wye, Round (2)



Branch, Coefficient C (See Note 8)

$\frac{V_b}{V_c}$	A_b/A_c						
	0.1	0.2	0.3	0.4	0.6	0.8	1.0
0.4	-.56	-.44	-.35	-.28	-.15	-.04	0.05
0.5	-.48	-.37	-.28	-.21	-.09	0.02	0.11
0.6	-.38	-.27	-.19	-.12	0	0.10	0.18
0.7	-.26	-.16	-.08	-.01	0.10	0.20	0.28
0.8	-.21	-.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	0.53
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

$\frac{V_s}{V_c}$	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	-.97	-.58	-.34
0.2	-6.7	-3.1	-1.9	-1.3	-.67	-.36	-.18
0.3	-5.0	-2.2	-1.3	-.88	-.42	-.19	-.05
0.4	-3.5	-1.5	-.88	-.55	-.21	-.05	0.05
0.5	-2.3	-.95	-.51	-.28	-.06	0.06	0.13
0.6	-1.3	-.50	-.22	-.09	0.05	0.12	0.17
0.7	-.63	-.18	-.03	0.04	0.12	0.16	0.18
0.8	-.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

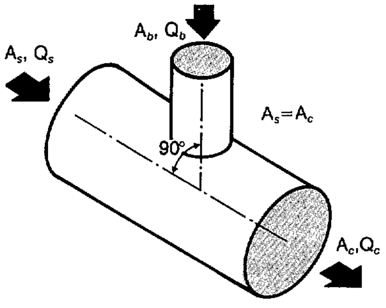
Note 7: A_1 = Area (Entering airstream), A = Area (Leaving airstream)

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-13 LOSS COEFFICIENTS, CONVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

B. Converging Tee, 90°, Round (15)



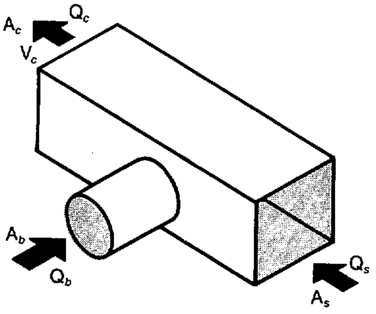
Branch, Coefficient C (See Note 8)

Q_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	-.37	-.51	-.46	-.50	-.51	-.52
0.2	3.8	0.72	0.17	-.02	-.14	-.18	-.24
0.3	9.2	2.3	1.0	0.44	0.21	0.11	-.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)

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	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



Branch, Coefficient C (See Note 8)

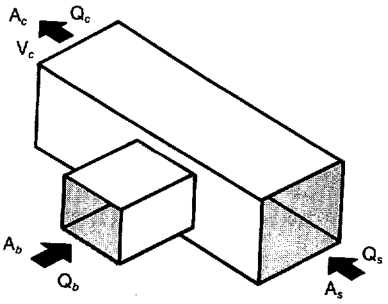
V_c fpm (m/s)	Q_b/Q_c									
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
< 1200 (6)	-.63	-.55	0.13	0.23	0.78	1.30	1.93	3.10	4.88	5.60
> 1200 (6)	-.49	-.21	0.23	0.60	1.27	2.06	2.75	3.70	4.93	5.95

For Main Loss Coefficients (C) see Fitting 14-13B

When:

A_b/A_s	A_s/A_c	A_b/A_c
0.5	1.0	0.5

D. Converging Tee, Rectangular Main and Branch



Branch, Coefficient (See Note 8)

V_c fpm (m/s)	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 (6)	-.75	-.53	-.03	0.33	1.03	1.10	2.15	2.93	4.18	4.78
> 1200 (6)	-.69	-.21	0.23	0.67	1.17	1.66	2.67	3.36	3.93	5.13

For Main Loss Coefficients (C) see Fitting 14-13B

When:

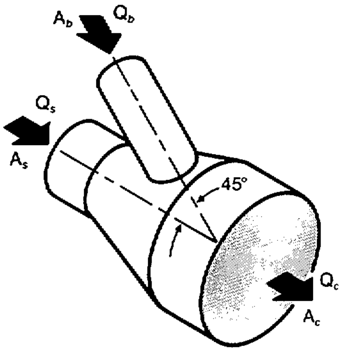
A_b/A_s	A_s/A_c	A_b/A_c
0.5	1.0	0.5

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-13 LOSS COEFFICIENTS, CONVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

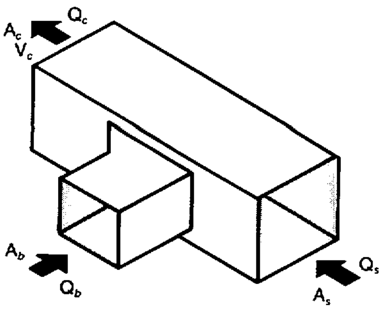
E. Converging Wye, Conical, Round (2)



Branch, 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	-2.4	-.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	-.27	0.81	1.7	2.4	3.0	3.6	4.1	4.5	4.9
	0.4	-1.8	-.72	0.07	0.66	1.1	1.5	1.8	2.1	2.3	2.5
0.5	0.2	-.46	1.5	3.3	4.9	6.4	7.7	8.8	9.9	11	12
	0.3	-.94	0.25	1.2	2.0	2.7	3.3	3.8	4.2	4.7	5.0
	0.4	-1.1	-.24	0.42	0.92	1.3	1.6	1.9	2.1	2.3	2.5
	0.5	-1.2	-.38	0.18	0.58	0.88	1.1	1.3	1.5	1.6	1.7
0.6	0.2	-.55	1.3	3.1	4.7	6.1	7.4	8.6	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	-.48	0.10	0.54	0.89	1.2	1.4	1.6	1.8	2.0
	0.5	-1.3	-.62	-.14	0.21	0.47	0.68	0.85	0.99	1.1	1.2
	0.6	-1.3	-.69	-.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	-.52	0.35	1.1	1.7	2.3	2.8	3.2	3.6	3.9	4.2
	0.4	-.67	-.05	0.43	0.80	1.1	1.4	1.6	1.8	1.9	2.1
	0.5	-.73	-.19	0.18	0.46	0.68	0.85	0.99	1.1	1.2	1.3
	0.6	-.75	-.27	0.05	0.28	0.45	0.58	0.68	0.76	0.83	0.88
	0.7	-.77	-.31	-.02	0.18	0.32	0.43	0.50	0.56	0.61	0.65
	0.8	-.78	-.34	-.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	—	.54	1.2	1.8	2.3	2.7	3.1	3.7	3.7	4.0
	0.4	—	.21	0.62	0.96	1.2	1.5	1.7	2.0	2.0	2.1
	0.5	—	.05	0.37	0.60	0.79	0.93	1.1	1.2	1.2	1.3
	0.6	—	-.02	0.23	0.42	0.55	0.66	0.73	0.80	0.85	0.89
	0.8	—	-.10	0.11	0.24	0.33	0.39	0.43	0.46	0.47	0.48
	1.0	—	-.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_b/A_c
0.5	1.0	0.5

Branch, Coefficient C (See Note 8)

V_c fpm (m/s)	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 (6)	-.83	-.68	-.30	0.28	0.55	1.03	1.50	1.93	2.50	3.03
> 1200 (6)	-.72	-.52	-.23	0.34	0.76	1.14	1.83	2.01	2.90	3.63

For Main Loss Coefficients (C) see Fitting 14-13B (Page 14.29)

Note 8: A = Area, Q = Airflow, V = Velocity

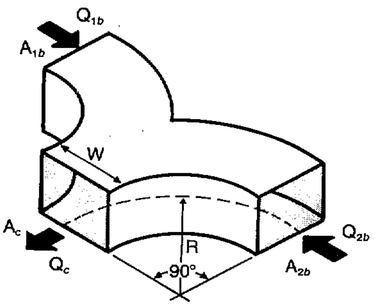
Table 14-13 LOSS COEFFICIENTS, CONVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

Main, Coefficient C (See Note 8)

$\frac{A_s}{A_c}$	$\frac{A_b}{A_c}$	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
	0.6	0.44	0.41	0.33	0.24	0.14	0.05	-.03	-.11	-.18	-.25
0.8	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 (15)



Coefficient C (See Note 8)

A_{1b}/A_c or A_{2b}/A_c	0.50	1.0
C	0.23	0.07

$$\frac{R}{W} = 1.5 \qquad \frac{Q_{1b}}{Q_c} = \frac{Q_{2b}}{Q_c} = 0.5$$

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-13 LOSS COEFFICIENTS, CONVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

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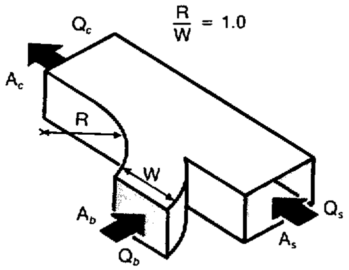
H. Converging Wye, Rectangular (15)



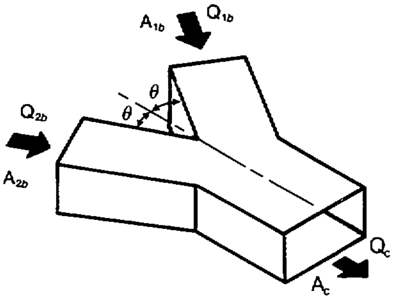
Branch, Coefficient (See Note 8)

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	-.50	0	0.50	1.2	2.2	3.7	5.8	8.4	11
0.33	0.25	-1.2	-.40	0.40	1.6	3.0	4.8	6.8	8.9	11
0.5	0.5	-.50	-.20	0	0.25	0.45	0.70	1.0	1.5	2.0
0.67	0.5	-1.0	-.60	-.20	0.10	0.30	0.60	1.0	1.5	2.0
1.0	0.5	-2.2	-1.5	-.95	-.50	0	0.40	0.80	1.3	1.9
1.0	1.0	-.60	-.30	-.10	-.04	0.13	0.21	0.29	0.36	0.42
1.33	1.0	-1.2	-.80	-.40	-.20	0	0.16	0.24	0.32	0.38
2.0	1.0	-2.1	-1.4	-.90	-.50	-.20	0	0.20	0.25	0.30

Main, Coefficient C (See Note 8)

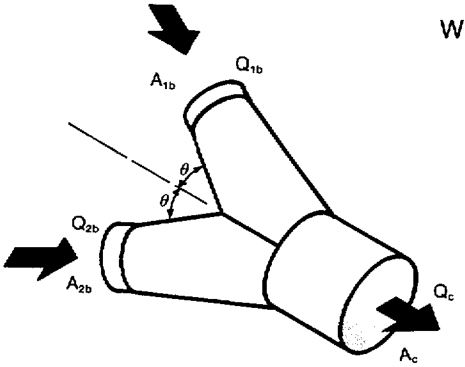
A_s/A_c	A_b/A_c	Q_b/Q_c								
		0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.75	0.25	0.30	0.30	0.20	-.10	- .45	-.92	-1.5	-.2.0	-2.6
1.0	0.5	0.17	0.16	0.10	0	-0.08	-.18	-.27	-.37	-.46
0.75	0.5	0.27	0.35	0.32	0.25	0.12	-.03	-.23	-.42	-.58
0.5	0.5	1.2	1.1	0.90	0.65	0.35	0	-.40	-.80	-1.3
1.0	1.0	0.18	0.24	0.27	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 (15)



Coefficient C (See Note 8)

θ	Q_{1b}/Q_c or Q_{2b}/Q_c										
	0	0.10	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.0
15°	-2.6	-1.9	-1.3	-.77	-.30	0.10	0.41	0.67	0.85	0.97	1.0
30°	-2.1	-1.5	-1.0	-.53	-.10	0.28	0.69	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:

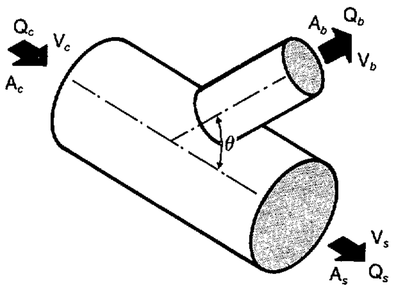
$A_{1b} = A_{2b}$
$A_c = A_{1b} + A_{2b}$

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Tees, Wyes)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

A. Tee or Wye, 30° to 90°, Round (15)



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 θ = 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	0.75	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 θ = 45°: Branch, Coefficient C (See note 8)

A_b/A_c	Q_bQ_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	—	—	—	—	—	—	—

Wye θ = 60°: 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	0.83	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	—	—	—	—	—	—	—

Tee θ = 90°: Branch, Coefficient C (See note 8)

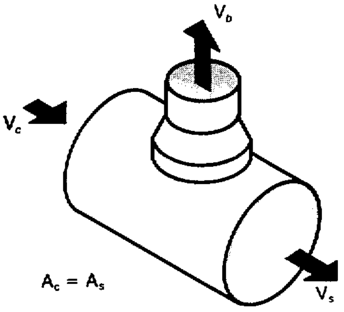
A_b/A_c	Q_b/Q_c								
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.8	0.95	0.92	0.92	0.93	0.94	0.95	1.1	1.2	1.4
0.7	0.95	0.94	0.95	0.98	1.0	1.1	1.2	1.4	1.6
0.6	0.96	0.97	1.0	1.1	1.1	1.2	1.4	1.7	2.0
0.5	0.97	1.0	1.1	1.2	1.4	1.5	1.8	2.1	2.5
0.4	0.99	1.1	1.3	1.5	1.7	2.0	2.4	—	—
0.3	1.1	1.4	1.8	2.3	—	—	—	—	—
0.2	1.3	1.9	2.9	—	—	—	—	—	—
0.1	2.1	—	—	—	—	—	—	—	—

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

B. 90° Conical Tee, Round (2)

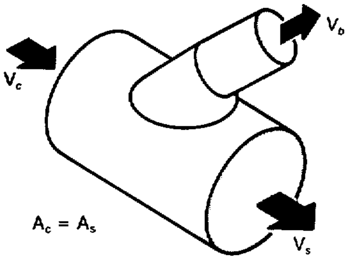


Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

C. 45° Conical Wye, Round (2)

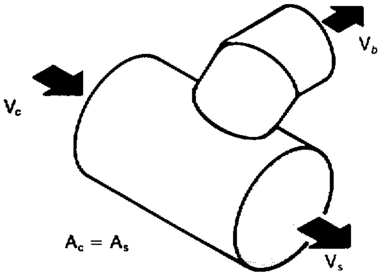


Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

D. 90° Tee, Round, Rolled 45° with 45° Elbow, Branch 90° to Main (2)

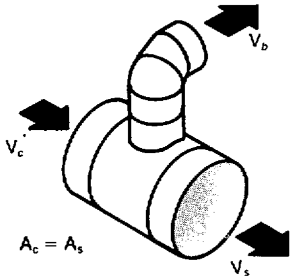


Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

E. 90° Tee, Round, with 90° Elbow, Branch 90° to Main (2)



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

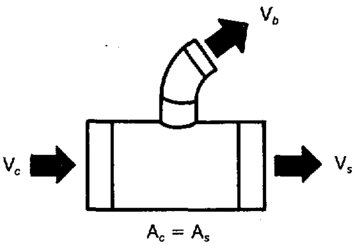
For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

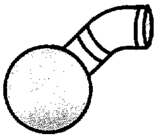
Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

F. 90° Tee, Round, Rolled 45° with 60° Elbow, Branch 45° to Main (2)



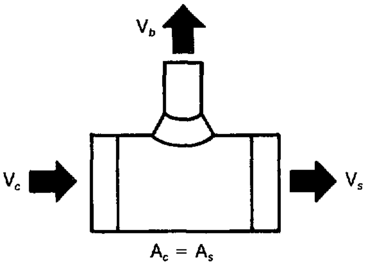


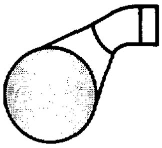
Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

G. 90° Conical Tee, Round, Rolled 45° with 45° Elbow, Branch 90° to Main (2)



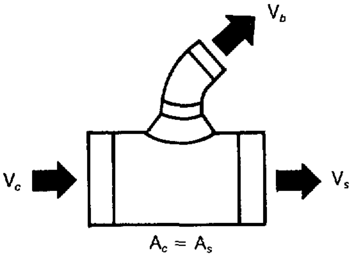


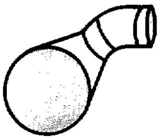
Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

H. 90° Conical Tee, Round, Rolled 45° with 60° Elbow, Branch 45° to Main (2)



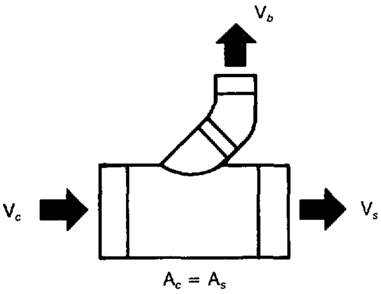


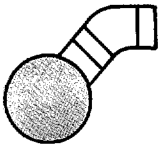
Branch, Coefficient C (See Note 8)

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	0.95	0.90	0.86	0.81	0.79	0.79	0.81	0.86	0.96	1.10

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

I. 45° Wye, Round, Rolled 45° with 60° Elbow, Branch 90° to Main (2)





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

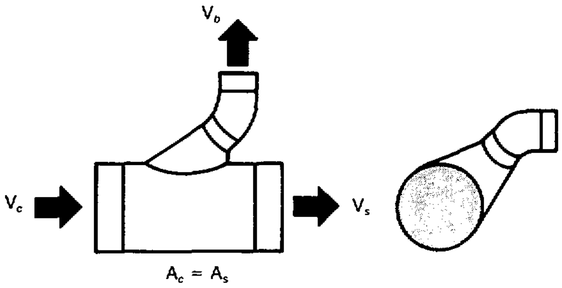
For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

J. 45° Conical Wye, Round, Rolled 45° with 60° Elbow, Branch 90° to Main (2)

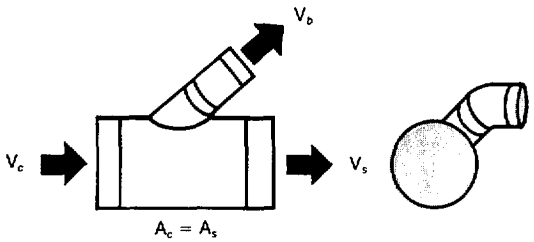


Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

K. 45° Wye, Round, Rolled 45° with 30° Elbow, Branch 45° to Main (2)

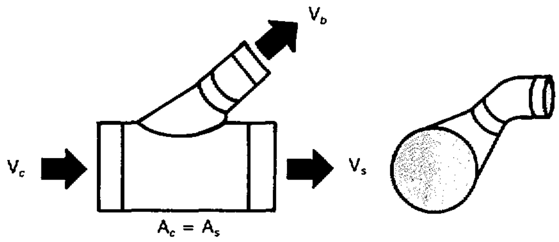


Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

L. 45° Conical Wye, Round, Rolled 45° with 30° Elbow, Branch 45° to Main (2)



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

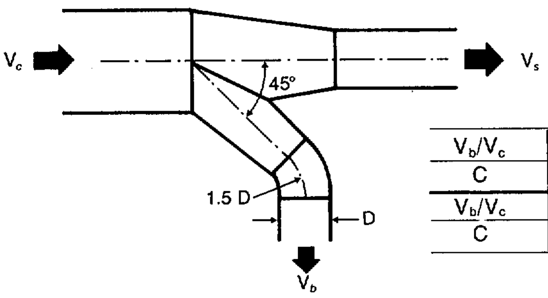
For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

M. 45° Wye, Conical Main and Branch with 45° Elbow, Branch 90° to Main (15)



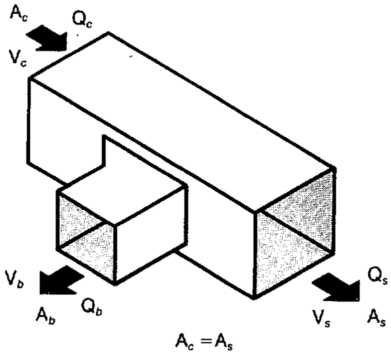
Branch, Coefficient C (See Note 8)

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	0.61	0.68
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

Main, Coefficient C (See Note 8)

V_s/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° Entry, Rectangular Main and Branch

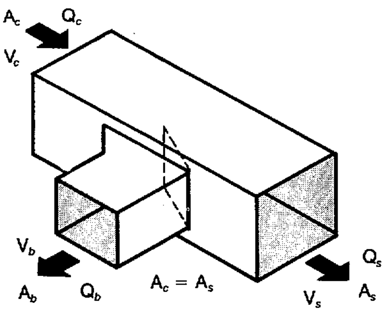


Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

P. 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

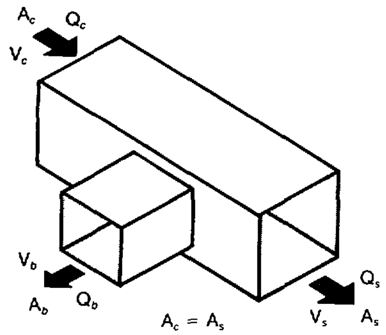
For Main Loss Coefficient (C) see Fitting 14-14S (Page 14.38)

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

Q. Tee, Rectangular Main and Branch

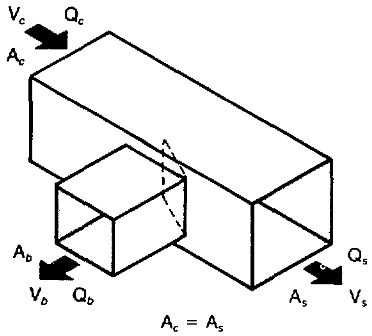


Branch, Coefficient C (See Note 8)

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	1.91	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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

R. Tee, Rectangular Main and Branch with Damper

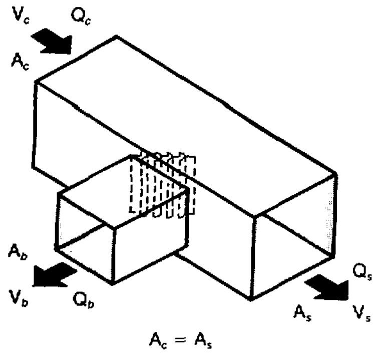


Branch, Coefficient C (See Note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14S

S. 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)

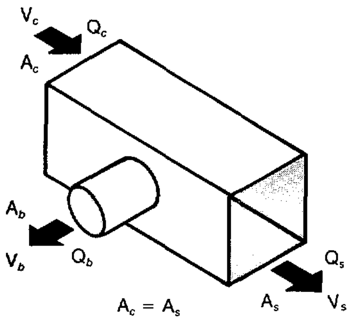
V_b/V_c	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8
C	0.03	0.04	0.07	0.12	0.13	0.14	0.27	0.30	0.25

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

T. Tee, Rectangular Main to Round Branch

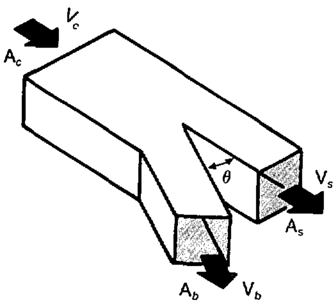


Branch, Coefficient C (See note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

U. Wye, Rectangular (15)



θ = 15° to 90°
$A_c = A_s + A_b$

Branch, Coefficient C (See Note 8)

θ	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

Main, Coefficient C (See Note 8)

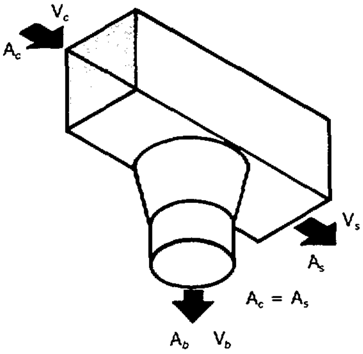
θ	15°-60°	90°				
V_s/V_c	A_s/A_c					
	0-1.0	0-0.4	0.5	0.6	0.7	≥0.8
0	1.0	1.0	1.0	1.0	1.0	1.0
0.1	0.81	0.81	0.81	0.81	0.81	0.81
0.2	0.64	0.64	0.64	0.64	0.64	0.64
0.3	0.50	0.50	0.52	0.52	0.50	0.50
0.4	0.36	0.36	0.40	0.38	0.37	0.36
0.5	0.25	0.25	0.30	0.28	0.27	0.25
0.6	0.16	0.16	0.23	0.20	0.18	0.16
0.8	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, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

V. Tee Rectangular Main to Conical Branch (2)

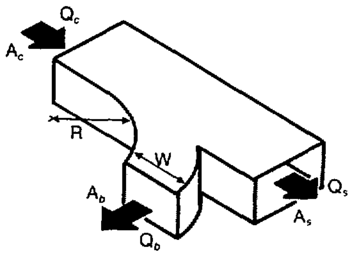


Branch, Coefficient C (See note 8)

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

For Main Loss Coefficient (C) see Fitting 14-14A (Page 14.33)

W. Wye, Rectangular (15)



$\frac{R}{W} = 1.0$

90° Branch

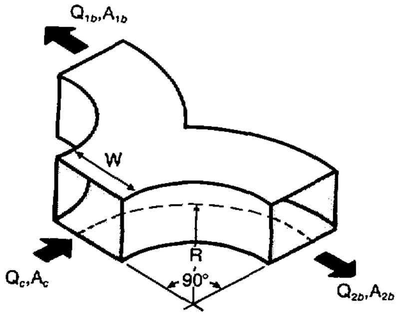
Branch, Coefficient C (See Note 8)

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	0.55	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

Main, 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	-.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

X. Symmetrical Wye, Dovetail, Rectangular (15)



Coefficient C (See Note 8)

A_{1b}/A_c or A_{2b}/A_c	0.50	1.0
C	0.30	0.25

When: R/W = 1.5

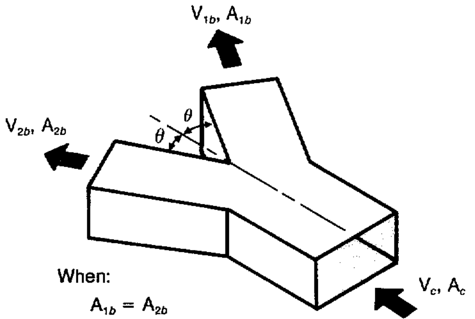
$$\frac{Q_{1b}}{Q_c} = \frac{Q_{2b}}{Q_c} = 0.5$$

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-14 LOSS COEFFICIENTS, DIVERGING JUNCTIONS (Cont.)

Use the velocity pressure (V^p) of the upstream section. Fitting loss (TP) = C × V_p

Y. Wye, Rectangular and Round (15)



When:

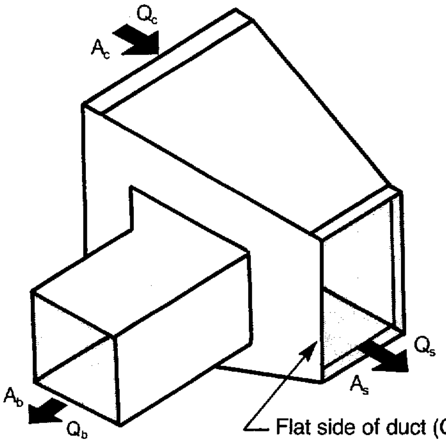
$A_{1b} = A_{2b}$
$A_c = A_{1b} + A_{2b}$

Coefficient C (See Note 8)

θ	V_{1b}/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_{1b}/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

Z. Tee, Rectangular Reducing, 45° Entry Branch (2)



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

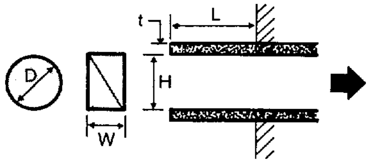
When:

A_b/A_c = 0.5 to 1.0
A_s/A_c = 0.5 to 1.0

Table 14-15 LOSS COEFFICIENTS, ENTRIES

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

A. Duct Mounted in Wall, Round and Rectangular (15)



Rectangular: $D = \frac{2\,HW}{(H + W)}$

Coefficient C

t/D	L/D						
	0	0.002	0.01	0.05	0.2	0.5	≥1.0
≈ 0	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 ($t/D_e \le 0.05$): $C_s = 1 + C_1$

b. Thick Edge ($t/D_e > 0.05$): $C_s = C + C_1$

where:

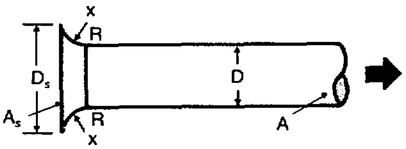
C_s is new coefficient of fitting with a screen or perforated plate at the entrance.
C is from above table
C_1 is from Table 14-17A (screen) or Table 14-17B (perforated plate)

Note 8: A = Area, Q = Airflow, V = Velocity

Table 14-15 LOSS COEFFICIENTS, ENTRIES (Cont.)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

B. Smooth Converging Bellmouth, Round, without End Wall (15)

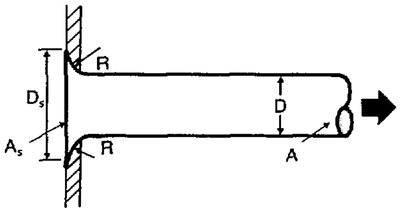


Coefficient C (See Note 9)

R/D	0	0.01	0.02	0.03	0.04	0.05
C	1.0	0.87	0.74	0.61	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 Converging Bellmouth, Round, with End Wall (15)

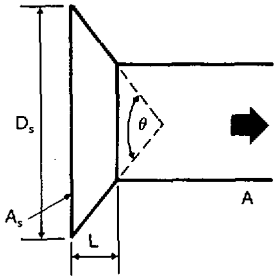


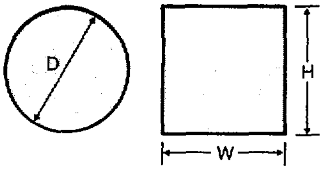
Coefficient C (See Note 9)

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 (15)





Rectangular: $D = \frac{2\,HW}{(H + W)}$

θ is major angle for rectangular entry

Coefficient C (See Note 9)

L/D	θ								
	0°	10°	20°	30°	40°	60°	100°	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

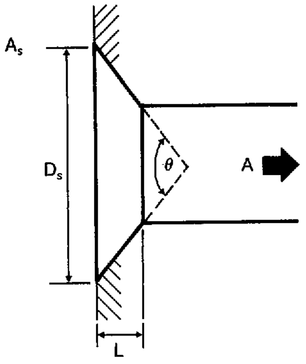
Note 9: With screen in opening at D_S, $C_S = C \text{ (from table)} + \frac{C \text{ (Screen coef. Table 14-17)}}{\left(\frac{A_S}{A}\right)^2}$

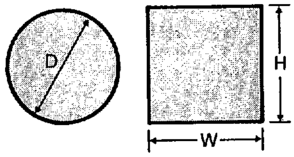
where: A= Area at D; A_S = Area at D_S

Table 14-15 LOSS COEFFICIENTS, ENTRIES (Cont.)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

E. Conical, Converging Bellmouth, Round and Rectangular, with End Wall (15)





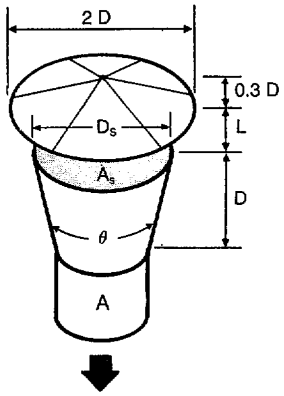
Rectangular: $D = \frac{2\,HW}{(H + W)}$

θ is major angle for rectangular entry

Coefficient C (See Note 9)

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

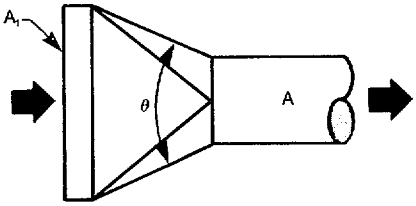
F. Intake Hood (15)



Coefficient C (See Note 9)

θ	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, Flanged or Unflanged (2)



$A_1 \geq 2A$

θ is major angle for rectangular hoods

Coefficient C

θ	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 screen in opening at D_S, $C_S = C \text{ (from table)} + \frac{C \text{ (Screen coef. Table 14-17)}}{\left(\frac{A_S}{A}\right)^2}$

where: A= Area at D; A_S = Area at D_S

Table 14-15 LOSS COEFFICIENTS, ENTRIES (Cont.)

Use the velocity pressure (V_p) of the downstream section. Fitting loss (TP) = C × V_p

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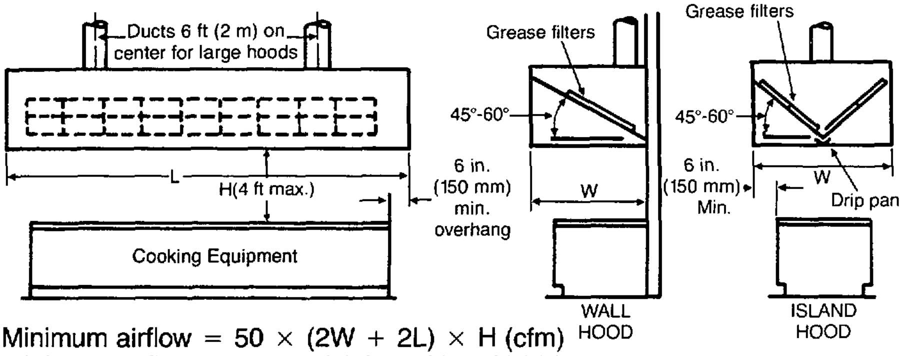
H. Hood, Canopy Island or Range

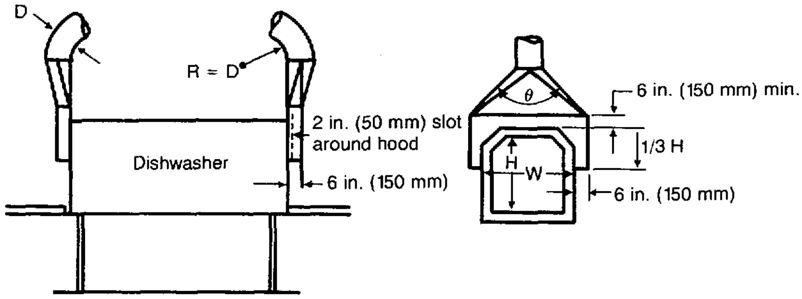


C = 0.5

Total Entry Loss = C × V_p + clean filter resistance + 0.1 in. w.g. (25 Pa)

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 metres (metric)

I. Hood, Slot (Dishwasher)



Use "C" loss coefficient from above Table 14-15 G. "Hood—Tapered, Flanged or Unflanged". (Total Entry Loss = 1.0 × Slot V_p + C × Duct V_p)

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)

Table 14-16 LOSS COEFFICIENTS, EXITS

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

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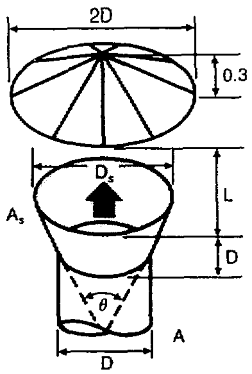
A. Exhaust Hood (15)



Coefficient C (See Note 9)

θ	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

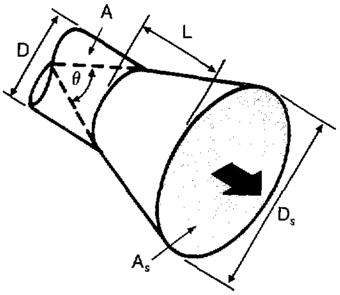
Note 9: With screen in opening at D_S, C_S = C (from table) + $\frac{\text{C (Screen coef. Table 14-17)}}{\left(\frac{A_S}{A}\right)^2}$

where: A= Area at D; A_S = Area at D_S

Table 14-16 LOSS COEFFICIENTS, EXITS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

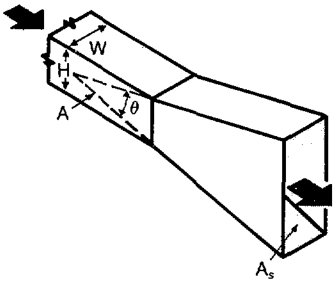
B. Exit, Conical, Round, with or without a Wall (15)



Coefficient C (See Note 9)

A_s/A	θ						
	14°	16°	20°	30°	45°	60°	≥90°
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 (15)

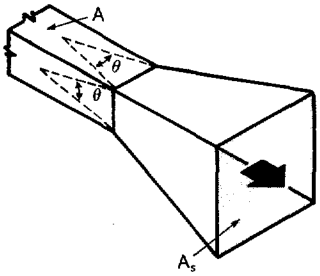


Coefficient C (See Note 10)

A_s/A	θ					
	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

When: 0.5 ≤ H/W ≤ 2.0

D. Exit, Pyramidal Diffuser, Rectangular, with or without a Wall (15)



Coefficient C (See Note 10)

A_s/A	θ					
	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 10: With screen in opening at A_s, C_s = C (from table) + $\dfrac{\text{C (Screen coef. Table 14-17)}}{\left(\dfrac{A_s}{A}\right)^2}$

Table 14-16 LOSS COEFFICIENTS, EXITS (Cont.)

Use the velocity pressure (V^p) of the upstream section. Fitting loss (TP) = C × V_p

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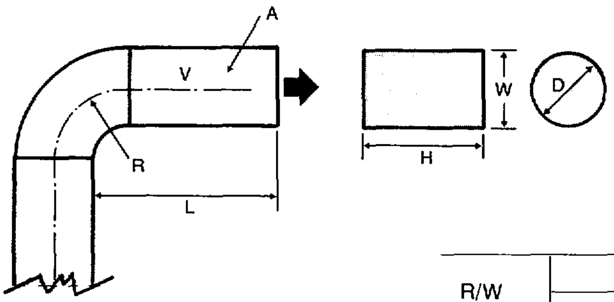
E. Exit, Discharge to Atmosphere from a 90° Elbow, Round and Rectangular (15)



Note:
Elbow loss included
in loss coefficient.

RECTANGULAR: Coefficient C (See Note 11)

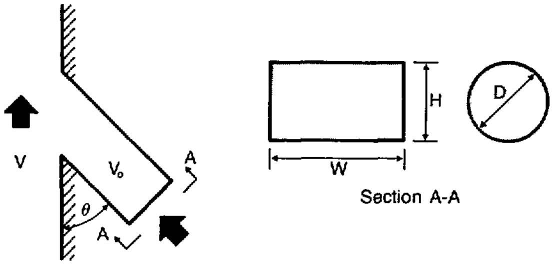
R/W	L/W									
	0	0.5	1.0	1.5	2.0	3.0	4.0	6.0	8.0	12.0
0	3.0	3.1	3.2	3.0	2.7	2.4	2.2	2.1	2.1	2.0
0.75	2.2	2.2	2.1	1.8	1.7	1.6	1.6	1.5	1.5	1.5
1.0	1.8	1.5	1.4	1.4	1.3	1.3	1.2	1.2	1.2	1.2
1.5	1.5	1.2	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1
2.5	1.2	1.1	1.1	1.0	1.0	1.0	1.0	1.0	1.0	1.0

ROUND: Coefficient C (See Note 10)

L/D	0.9	1.3
C	1.5	1.4

When:
R/D = 1.0 (Round)

F. Exit, Duct Flush with Wall, Flow along Wall (15)



Section A-A

RECTANGULAR:

Coefficient C (See Note 11)

Aspect Ratio (H/W)	θ	V/V$_o$				
		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)

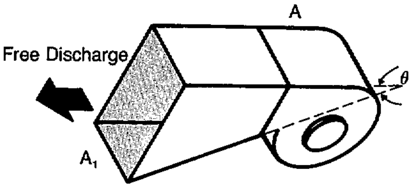
θ	V/V$_o$				
	0	0.5	1.0	1.5	2.0
30°-45°	1.0	1.0	1.1	1.3	1.6
60°	1.0	0.90	1.1	1.4	1.6
90°	1.0	0.80	0.95	1.4	1.7
120°	1.0	0.80	0.95	1.3	1.7
150°	1.0	0.82	0.83	1.0	1.3

NOTE 11: With screen in opening C_s = C (from table) + C (Screen Coef. from Table 14-17)

Table 14-16 LOSS COEFFICIENTS, EXITS (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = $C \times V_p$

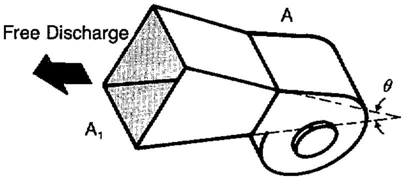
G. Plane Asymmetric Diffuser at Fan Outlet without Ductwork (15)



Coefficient C (See Note 12)

θ	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

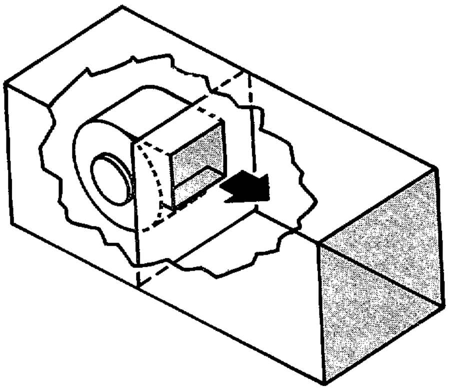
H. Pyramidal Diffuser at Fan Outlet without Ductwork (15)



Coefficient C (See Note 12)

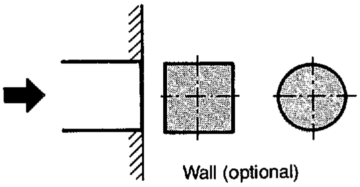
θ	A_1/A					
	1.5	2.0	2.5	3.0	3.5	4.0
10°	0.54	0.42	0.37	0.34	0.32	0.31
15°	0.67	0.58	0.53	0.51	0.50	0.51
20°	0.75	0.67	0.65	0.64	0.64	0.65
25°	0.80	0.74	0.72	0.70	0.70	0.72
30°	0.85	0.78	0.76	0.75	0.75	0.76

I. Fan, Free Discharge, Plenum



C = 1.0 (Airfoil wheels—Backwardly inclined)
C = 1.5 (Backwardly inclined wheels)
C = 1.8 (Forward curved wheels)

J. Exit, Abrupt, Round and Rectangular, with or without a Wall (15)



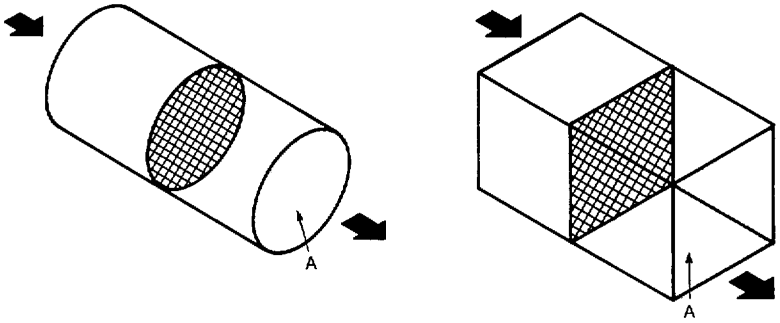
C = 1.0
With Screen: $C_s = 1 + (\text{C from Table 14-17})$

NOTE 12: With screen in opening at A_1, $C_s = C \text{ (from table)} + \dfrac{C \text{ (Screen coef. Table 14-17)}}{\left(\frac{A_1}{A}\right)^2}$

Table 14-17 LOSS COEFFICIENTS, SCREENS AND PLATES

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

A. Screen in Duct, Round and Rectangular (15)



n = A_s/A

Where:

n = Free area ratio of screen

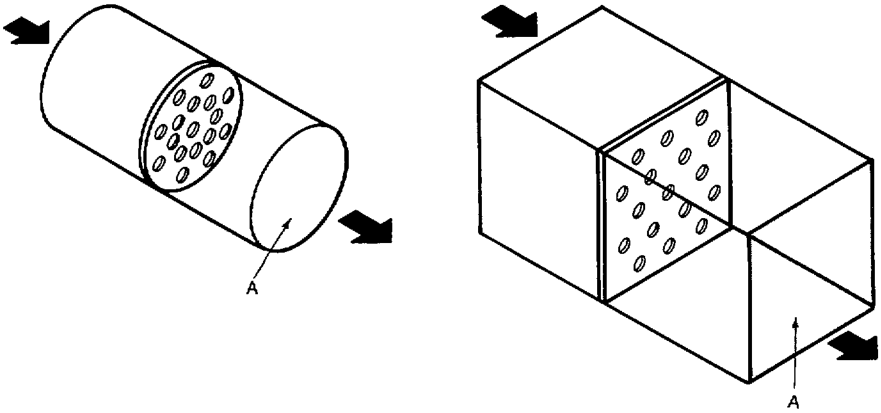
A_s = Total flow area of screen

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 (15)



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 = diameter of perforated holes

n = free area ratio of plate

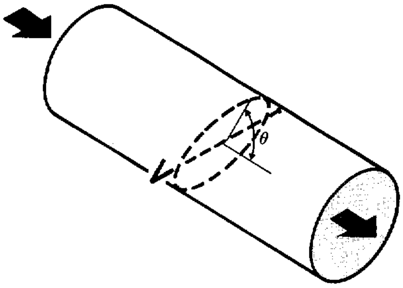
A_p = total flow area of perforated plate

A = area of duct

Table 14-18 LOSS COEFFICIENTS, OBSTRUCTIONS (Constant Velocities)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

A. Damper, Butterfly, Thin Plate, Round (15)

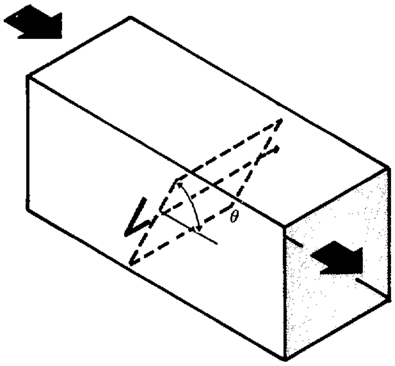


Coefficient C

θ	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 (15)

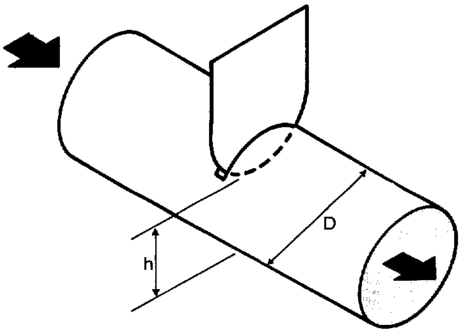


Coefficient C

θ	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 (15)



Coefficient C

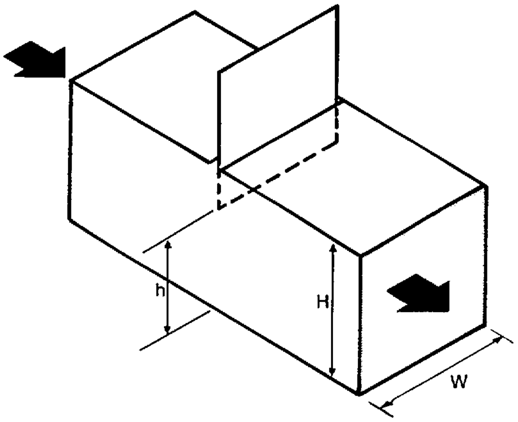
h/D	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
A_h/A	0.25	0.38	0.50	0.61	0.71	0.81	0.90	0.96
C	35	10	4.6	2.1	0.98	0.44	0.17	0.06

A_h = Free Area
A = Area of Duct

Table 14-18 LOSS COEFFICIENTS, OBSTRUCTIONS (Constant Velocities) (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

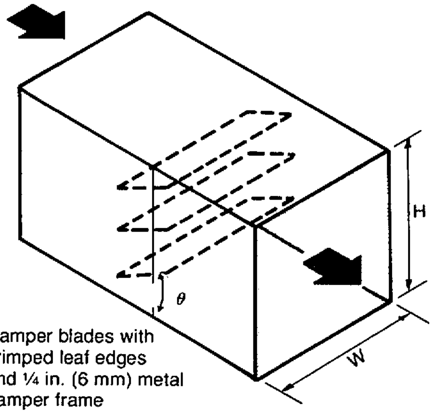
D. Damper, Gate, Rectangular (15)



Coefficient C

H/W	h/H						
	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 (2)



Damper blades with crimped leaf edges and ¼ in. (6 mm) metal damper frame

Coefficient C

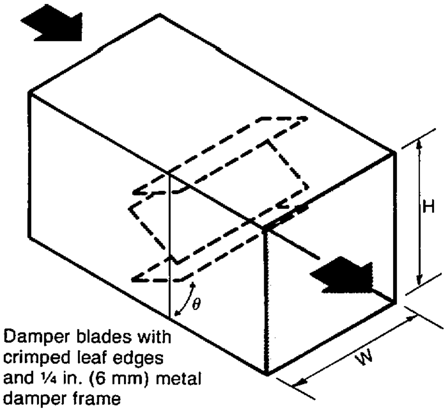
L/R	θ								
	80°	70°	60°	50°	40°	30°	20°	10°	0° Fully open
0.3	116	32	14	9.0	5.0	2.3	1.4	0.79	0.52
0.4	152	38	16	9.0	6.0	2.4	1.5	0.85	0.52
0.5	188	45	18	9.0	6.0	2.4	1.5	0.92	0.52
0.6	245	45	21	9.0	5.4	2.4	1.5	0.92	0.52
0.8	284	55	22	9.0	5.4	2.5	1.5	0.92	0.52
1.0	361	65	24	10	5.4	2.6	1.6	1.0	0.52
1.5	576	102	28	10	5.4	2.7	1.6	1.0	0.52

$$\frac{L}{R} = \frac{N\ W}{2\,(H + W)}$$

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 perpendicular to blade axis

F. Damper, Rectangular, Opposed Blades (2)



Damper blades with crimped leaf edges and ¼ in. (6 mm) metal damper frame

Coefficient C

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

$$\frac{L}{R} = \frac{N\ W}{2\,(H + W)}$$

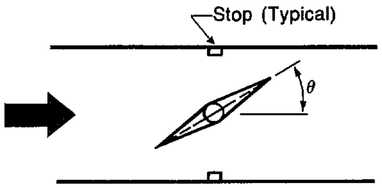
where:

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 perpendicular to blade axis

Table 14-18 LOSS COEFFICIENTS, OBSTRUCTIONS (Constant Velocities) (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

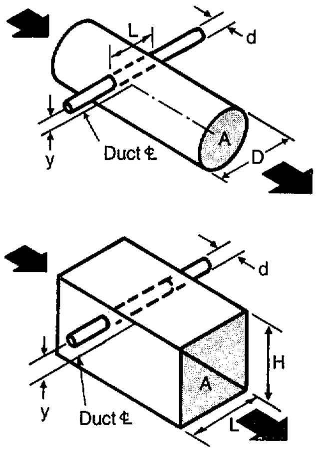
G. Damper, Butterfly, Airfoil Blade, Rectangular (15)



Coefficient C

θ	0°	10°	20°	30°	40°	50°	60°
C	0.50	0.65	1.6	4.0	9.4	24	67

H. Obstruction, Smooth Cylinder in Round and Rectangular Ducts (15)



Coefficient C_o

R_e	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
(0.5-200) × 10^3	0.07	0.15	0.24	0.33
3 × 10^5	0.07	0.16	0.26	0.35
4 × 10^5	0.05	0.11	0.19	0.25
5 × 10^5	0.04	0.09	0.14	0.19
(6-10) × 10^5	0.02	0.05	0.07	0.10

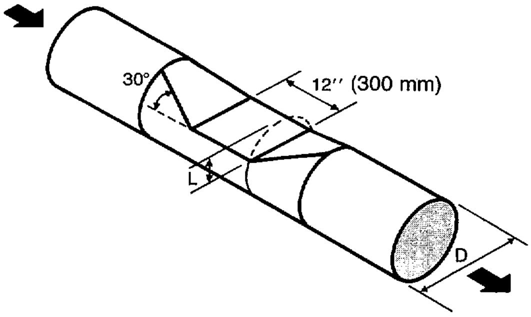
where:

$S_m = d\,L$
$R_e = 8.56\ DV$ (Std. Air—U.S.)
$R_e = 66.4\ DV$ (Std. Air-Metric)
$S_m/A < 0.3$
$C = K\,C_o$
A = Area of duct

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

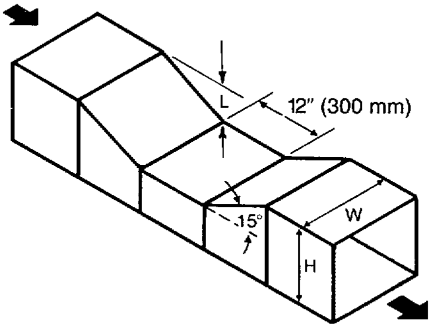


When: L/D = 0.33
C = 0.24

Table 14-18 LOSS COEFFICIENTS, OBSTRUCTIONS (Constant Velocities) (Cont.)

Use the velocity pressure (V_p) of the upstream section. Fitting loss (TP) = C × V_p

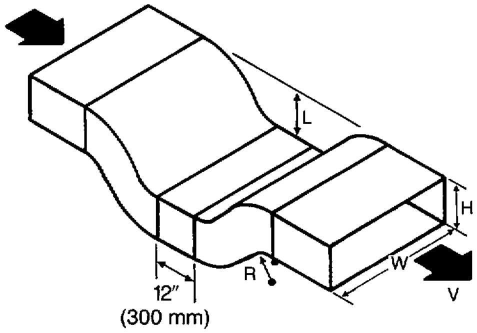
J. Rectangular Duct, Depressed to Avoid an Obstruction



Coefficient C

W/H	L/H			
	0.125	0.15	0.25	0.30
1.0	0.26	0.30	0.33	0.35
4.0	0.10	0.14	0.22	0.30

K. Rectangular Duct with 4-45° Smooth Radius Ells to Avoid an Obstruction



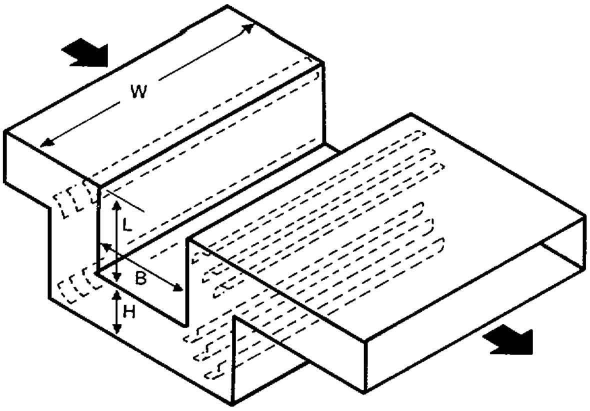
Coefficient C

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 an Obstruction

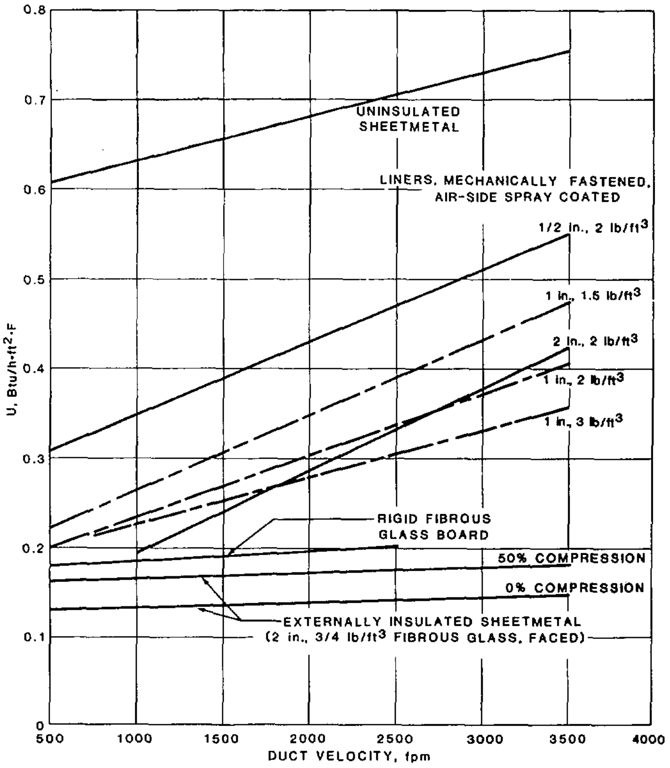


Coefficient C

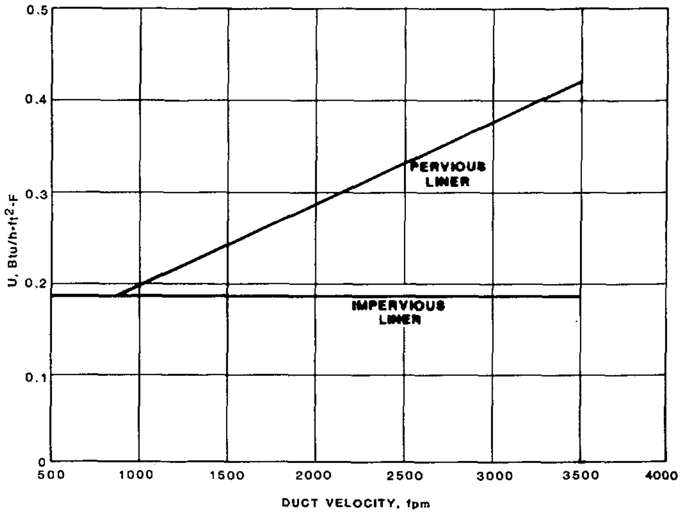
L/H Ratio	0.5	1.0	1.5	2
Single Blade Turning Vanes	—	0.86	0.83	0.77
Double Blade Turning Vanes	—	1.85	2.84	2.91
"S" type Splitter Vanes	0.61	0.65	—	—
No Vanes—Up to 1200 fpm (6 m/s)	0.88	5.26	6.92	7.56
No Vanes—Over 1200 fpm (6 m/s)	1.26	6.22	8.82	9.24

Where: W/H = 1.0 to 3.0
B = 12" to 24"

C HEAT TRANSFER COEFFICIENTS

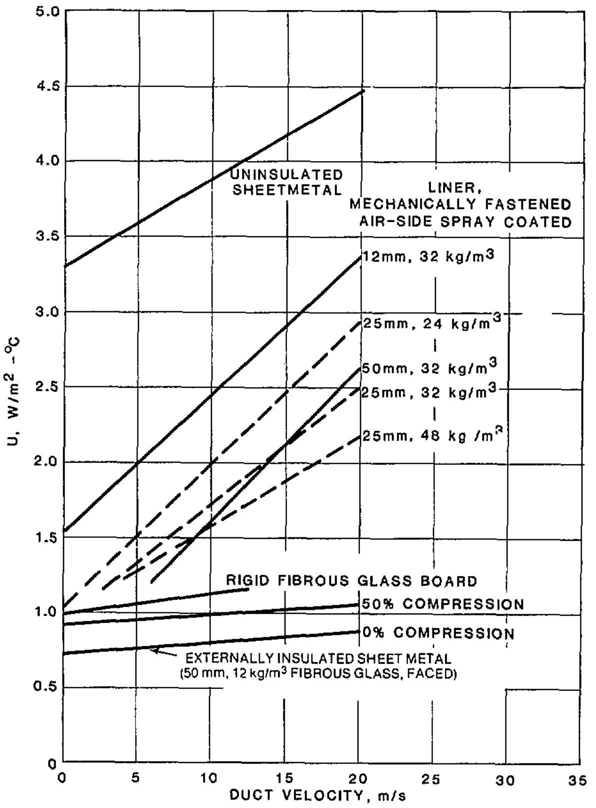


a. Rigid Ducts

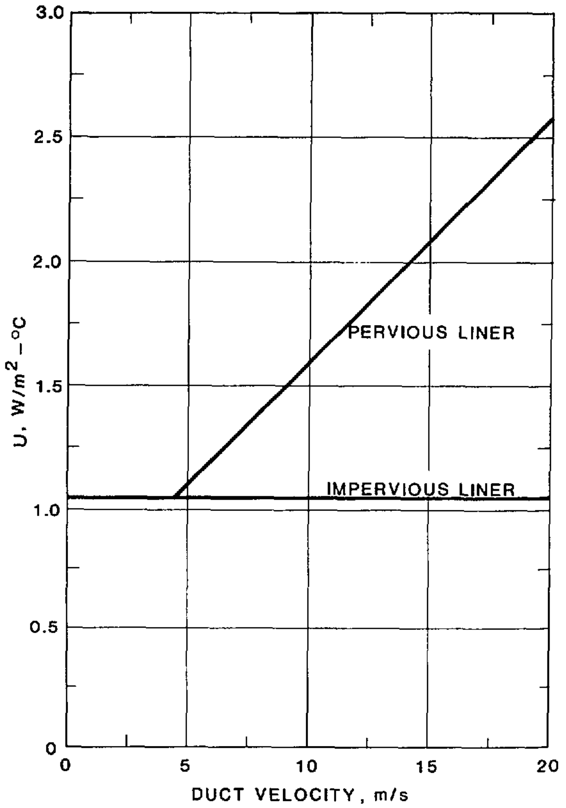


b. Flexible Ducts

Figure 14-6 DUCT HEAT TRANSFER COEFFICIENTS (U.S. Units) (2)



a. Rigid Ducts



b. Flexible Ducts

Figure 14-7 DUCT HEAT TRANSFER COEFFICIENTS (Metric Units) (2)

D HVAC EQUATIONS (U.S. UNITS)

Table 14-19 AIR EQUATIONS

a) $V = 1096 \sqrt{\frac{V_p}{d}}$

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. w.g.)

d = Density (lb/cu ft)

P_b = Absolute Static Pressure (in. Hg)
(Barometric pressure + static pressure)

T = Absolute Temp. (460° + °F)

---

b) $Q\text{ (sens.)} = 60 \times C_p \times d \times \text{cfm} \times \Delta t$

or for standard air (C_p = 0.24 Btu/lb − °F):

$Q\text{ (sens.)} = 1.08 \times \text{cfm} \times \Delta t$

c) $Q\text{ (lat.)} = 4750 \times \text{cfm} \times \Delta W\text{ (lb.)}$

or

$Q\text{ (lat.)} = 0.67 \times \text{cfm} \times \Delta W\text{ (gr.)}$

d) $Q\text{ (total)} = 4.5 \times \text{cfm} \times \Delta h$

e) $Q = A \times U \times \Delta t$

f) $R = \frac{1}{U}$

g) $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = RM$

Q = Heat Flow (Btu/hr)

C_p = Specific Heat (Btu/lb • °F)

d = Density (lb/cu ft)

Δt = Temperature Difference (°F)

ΔW = Humidity Ratio (lb or gr H_2O/lb dry air)

Δh = Enthalpy Diff. (Btu/lb dry air)

A = Area of Surface (sq ft)

U = Heat Transfer Coefficient (Btu/sq ft • hr • °F)

R = Sum of Thermal Resistances (sq ft · hr · °F/Btu)

P = Absolute Pressure (lb/sq ft)

V = Total Volume (cu ft)

T = Absolute Temp. (460° + °F = °R)

R = Gas Constant (ft/°R)

M = Mass (lb)

---

h) $TP = V_p + SP$

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

j) $V = V_m \left[\frac{d\text{ (other than standard)}}{0.075\text{ (d = std. air)}}\right]$

k) $\text{cfm} = A \times V$

l) $TP = C \times V_p$

TP = Total Pressure (in. w.g.)

V_p = Velocity Pressure (in. w.g.)

SP = Static Pressure (in. w.g.)

V = Velocity (fpm)

V_m = Measured Velocity (fpm)

d = Density (lb/cu ft)

A = Area of duct cross section (sq ft)

C = Duct Fitting Loss Coefficient

Table 14-20 FAN EQUATIONS

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) $\frac{\text{rpm (fan)}}{\text{rpm (motor)}} = \frac{\text{Pitch diam. motor pulley}}{\text{Pitch diam. fan pulley}}$

cfm = Cubic feet per minute
rpm = Revolutions per minute
P = Static or Total Pressure (in. w.g.)
bhp = Brake horsepower
d = Density (lb/cu ft)

Table 14-21 PUMP EQUATIONS

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) $\frac{bhp_2}{bhp_1} = \left(\frac{D_2}{D_1}\right)^3$

gpm = Gallons per minute
rpm = Revolutions per minute
D = Impeller diameter
H = Head (ft. w.g.)
bhp = Brake horsepower

Table 14-22 HYDRONIC EQUIVALENTS

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 ft. of mercury (Hg) = 5.89 psi
f. One cu.ft. of water = 62.4 lb = 7.49 gal.
g. One in. of mercury (Hg) = 13.6 in.w.g. = 1.13 ft. w.g.
h. Atmospheric Pressure = 29.92 in.Hg = 14.696 psi
i. One psi = 2.31 ft. w.g. = 2.04 in.Hg

Table 14-23 HYDRONIC EQUATIONS

a) $Q = 500 \times gpm \times \Delta t$

b) $\frac{\Delta P_2}{\Delta P_1} = \left(\frac{gpm_2}{gpm_1}\right)^2$

c) $\Delta P = \left(\frac{gpm}{C_v}\right)^2$

gpm = Gallons per minute

Q = Heat flow (Btu/hr)

Δt = Temperature diff. (°F)

ΔP = Pressure diff. (psi)

C_v = Valve constant (dimensionless)

---

d) $whp = \frac{gpm \times H \times Sp.\,Gr.}{3960}$

e) $bph = \frac{gpm \times H \times Sp.\,Gr.}{3960 \times E_p} = \frac{whp}{E_p}$

f) $E_p = \frac{whp \times 100}{bhp}$ (in percent)

whp = Water horsepower

gpm = Gallons per minute

bhp = Brake horsepower

H = Head (ft w.g.)

Sp. Gr. = Specific gravity (use 1.0 for water)

E_p = Efficiency of pump

---

g) $NPSHA = P_a \pm P_s + \frac{V^2}{2g} - P_{vp}$

h) $h = f \times \frac{L}{D} \times \frac{V^2}{2g}$

NPSHA = Net positive suction head available

P_a = Atm. press. (use 34 ft w.g.)

P_s = Pressure at pump centerline (ft w.g.)

$\frac{V^2}{2g}$ = Velocity head at point P_s (ft w.g.)

P_{vp} = Absolute vapor pressure (ft w.g.)

g = Gravity acceleration (32.2 ft/sec^2)

h = Head loss (ft)

f = Friction factor (dimensionless)

L = Length of pipe (ft)

D = Internal diameter (ft)

V = Velocity (ft/sec)

Table 14-24 CONVERTING PRESSURE IN INCHES OF MERCURY TO FEET OF WATER AT VARIOUS WATER TEMPERATURES

Water Temperature degrees F	60°	150°	200°	250°	300°	340°
Ft. head differential per in. Hg. differential	1.046	1.07	1.09	1.11	1.15	1.165

Table 14-25 ELECTRIC EQUATIONS

a) $\text{Bhp} = \dfrac{\text{I x E x P.F. x Eff.}}{746}$ (Single Phase)

b) $\text{Bhp} = \dfrac{\text{I x E x P.F. x Eff. x 1.73}}{746}$ (Three Phase)

c) $E = IR$

d) $P = EI$

e) $\dfrac{\text{F.L. Amps* x Voltage*}}{\text{Actual Voltage}} = \text{Actual F.L. Amps}$

*Nameplate ratings

I = Amps (A)
E = Volts (V)
P.F. = Power factor
R = ohms (Ω)
P = watts (W)
Bhp = Brake horsepower

Table 14-26 AIR DENSITY CORRECTION FACTORS (U.S. Units) (13)

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.w.g.)	407.5	392.8	378.6	365.0	351.7	338.9	326.4	314.3	302.1	291.1	280.1
Air Temp.	-40°	1.26	1.22	1.17	1.13	1.09	1.05	1.01	0.97	0.93	0.90	0.87
°F	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 = 0.075 lb/cu ft at 29.92 in. Hg

E HVAC EQUATIONS (METRIC UNITS)

Table 14-27 AIR EQUATIONS

a) $V = 1.414 \sqrt{\frac{V_p}{d}}$

or for standard air (d = 1.204 kg/m^3):

$V = \sqrt{1.66\ V_p}$

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/m^3)

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)

$Q\ (sens.) = 1.23 \times l/s \times \Delta t$

c) $Q\ (lat.) = 3.0 \times l/s \times \Delta W$

d) $Q\ (total\ heat) = 1.20 \times l/s \times \Delta h$

e) $Q = A \times U \times \Delta t$

f) $R = \frac{1}{U}$

g) $\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} = RM$

Q = Heat Flow (watts or kW)

C_p = Specific Heat (kJ/kg • °C)

d = Density (kg/m^3)

Δt = Temperature Difference (°C)

ΔW = Humidity Ratio (g H_2O/kg dry air)

Δh = Enthalpy Diff. (kJ/kg dry air)

A = Area of Surface (m^2)

U = Heat Transfer Coefficient (W/m^2 • °C)

R = Sum of Thermal Resistances (m^2 · °C/W)

P = Absolute Pressure (kPa)

V = Total Volume (m^3)

T = Absolute Temperature (273° + °C = °K)

R = Gas Constant (kJ/kg • °C)

M = Mass (kg)

---

h) $TP = V_p + SP$

i) $V_p = \frac{d}{2} \times V^2 = 0.602\ V^2$

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) $TP = C \times V_p$

TP = Total Pressure (Pa)

V_p = Velocity Pressure (Pa)

SP = Static Pressure (Pa)

V = Velocity (m/s)

V_m = Measured Velocity (m/s)

d = Density (kg/m^3)

A = Area of duct cross section (m^2)

C = Duct Fitting Loss Coefficient

Table 14-28 FAN EQUATIONS

a) $$\frac{l/s_2}{l/s_1} = \frac{m^3/s_2}{m^3/s_1} = \frac{rad/s_2}{rad/s_1}$$

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) $$\frac{\text{rad/s (fan)}}{\text{rad/s (motor)}} = \frac{\text{Pitch diam. motor pulley}}{\text{Pitch diam. fan pulley}}$$

l/s = Litres per second
m³/s = Cubic metres per second
rad/s = Radians per second

P = Static or Total Pressure (Pa)
kW = Kilowatts
d = Density (kg/m³)

Table 14-29 PUMP EQUATIONS

a) $$\frac{l/s_2}{l/s_1} = \frac{m^3/s_2}{m^3/s_1} = \frac{rad/s_2}{rad/s_1}$$

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) $$\frac{BP_2}{BP_1} = \left(\frac{rad/s_2}{rad/s_1}\right)^3$$

f) $$\frac{BP_2}{BP_1} = \left(\frac{D_2}{D_1}\right)^3$$

l/s = Litres per second

m³/s = Cubic metres per second

rad/s = Radians per second

D = Impeller diameter

H = Head (kPa)

BP = Brake horsepower

Table 14-30 HYDRONIC EQUATIONS

a) $Q = 4190 \times m^3/s \times \Delta t$
$= 4.19 \times l/s \times \Delta t$

b) $\frac{\Delta P_2}{\Delta P_1} = \left(\frac{m^3/s_2}{m^3/s_1}\right)^2 = \left(\frac{l/s_2}{l/s_1}\right)^2$

c) $\Delta P = \left(\frac{m^3/s}{C_v}\right)^2 = \left(\frac{l/s}{C_v}\right)^2$

Q = Heat flow (kilowatts)

Δt = Temperature difference (°C)

m^3/s (used for large volumes) = Cubic metres per second

l/s = Litres per second

ΔP = Pressure diff. (Pa or kPa)

C_v = Valve constant (dimensionless)

---

d) $WP(kW) = 9.81 \times m^3/s \times H(m) \times Sp.\ Gr.$

$$WP(W) = \frac{l/s \times H(Pa) \times Sp.\ Gr.}{1002}$$

e) $BP = \frac{WP}{E_p}$

f) $E_p = \frac{WP \times 100}{BP}$ (in percent)

WP = Water power (kW) or (W)

m^3/s = Cubic metres per second

l/s = Litres per second

Sp. Gr. = Specific gravity (use 1.0 for water)

BP = Brake power (kW)

E_p = Efficiency of Pump

H = Head (Pa) or (m)

---

g) $NPSHA = P_a \pm P_s + \frac{V^2}{2g} - P_{vp}$

h) $h = f \times \frac{L}{D} \times \frac{V^2}{2g}$

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)

g = Gravity acceleration (9.807 m/s^2)

h = Head loss (m)

f = Friction factor (dimensionless)

L = Length of pipe (m)

D = Internal diameter (m)

V = Velocity (m/s)

Table 14-31 ELECTRIC EQUATIONS

a) $\text{kW} = \frac{\text{I x E x P.F. x Eff.}}{1000}$ (Single Phase)

b) $\text{kW} = \frac{\text{I x E x P.F. x Eff. x 1.73}}{1000}$ (Three Phase)

c) $E = IR$

d) $P = EI$

e) $\frac{\text{F.L. Amps}^* \text{ x Voltage}^*}{\text{Actual Voltage}}$ = Actual F.L. Amps

*Nameplate ratings

kW = Kilowatts
I = Amps (A)
E = Volts (V)
P.F. = Power factor
R = ohms (Ω)
P. = watts (W)

Table 14-32 AIR DENSITY CORRECTION FACTORS (Metric Units)

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	0.49	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

Standard Air Density, Sea Level, 20°C = 1.2041 kg/m³ at 101.325 kPa

F METRIC UNITS AND EQUIVALENTS

Table 14-33 **METRIC UNITS (Basic & Derived)** (13)

Unit	Symbol	Quantity	Equivalent or Relationship
ampere	A	Electric current	Same as U.S.
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 U.S.
farad	F	Electric capacitance	Same as U.S.
henry	H	Electric inductance	Same as U.S.
hertz	Hz	Frequency	Same as cycles per second
joule	J	Energy, work, heat	1 J = 0.7376 ft-lb = 0.000948 Btu
kelvin	K	Thermodynamic temperature	°K = °C + 273.15° $= \frac{°F + 459.67}{1.8}$
kilogram	kg	Mass	1 kg = 2.2046 lb
litre	l	Liquid volume	1 l = 1.056qt = 0.264 gal
lumens	lm	Luminous flux	1 lm/m^2 = 0.0929 ft candles
lux	lx	Illuminance	1 lx = 0.0929 ft candles
metre	m	Length	1 m= 3.281 ft
mole	mol	Amount of substance	—
newton	N	Force	1 N = $kg \cdot m/s^2$ = 0.2248 lb (force)
ohm	Ω	Electrical resistance	Same as U.S.
pascal	Pa	Pressure, stress	1 Pa = N/m^2 = 0.000145 psi = 0.004022 in. w.g.
radian	rad	Plane angle	1 rad = 57.29°
second	s	Time	Same as U.S.
siemens	S	Electric conductance	—
steradian	sr	Solid angle	—
volt	V	Electric potential	Same as U.S.
watt	W	Power, heat flow	1 W = J/s = 3.4122 Btu/hr 1 W = 0.000284 tons of refrig.

Table 14-34 METRIC EQUIVALENTS (13)

Quantity	Symbol	Unit	U.S. Relationship
acceleration	m/s^2	metres 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
area	m^2	square metre	1 m^2 = 10.76 sq ft
atmospheric pressure	—	101.325 kPa	29.92 in Hg = 14.696 psi
density	kg/m^3	kilograms per cubic metre	1 kg/m^3 = 0.0624 lb/cu ft
density, air	—	1.2 kg/m^3	0.075 lb/cu ft
density, water	—	1000 kg/m^3	62.4 lb/cu ft
duct friction loss	Pa/m	pascals per metre	1 Pa/m = 0.1224 in.w.g./100′
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	metre	1 m = 3.281 ft = 39.37 in.
linear velocity	m/s	metres 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•m^2	kilograms x square metre	1 kg. • m^2 = 23.73 lb • sq ft
power	W	watt	1 W = 0.00134 hp′
pressure	kPa Pa	kilopascal (1000 pascals) pascal	1 kPa = 0.296 in Hg = 0.145 psi 1 Pa = 0.004015 in.w.g.
specific heat-air (C_p)		1000 J/kg • °C	1000 J/kg·°C = 1 kJ/kg • °C = 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 metres per kilogram	1 m^3/kg = 16.019 cu ft/lb
thermal conductivity	W • mm/m^2 • °C	watt millimetre per square metre °C	1 W • mm/m^2 • °C = 0.0069 Btu • in/ft^2 • hr • °F
volume flow rate	m^3/s l/s	cubic metres per second litres per second 1m^3/s = 1000 l/s 1 ml = litres/1000	1 m^3/s = 2118.88 cfm (air). 1 l/s = 2.12 cfm (air) 1 m^3/s = 15,850 gpm (water) 1 ml/s = 1.05 gph (water)

Table 14-35 RECOMMENDED NC-RC LEVELS FOR DIFFERENT INDOOR ACTIVITY AREAS

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 Rm	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, wating 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	30-35
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)	35-45	40-50
Department stores (main floors)	40-50	45-55
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	35-45	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 14-36 LISTENING CONDITIONS AND TELEPHONE USE AS A FUNCTION OF NC-RC LEVELS

NC-RC 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 14-37 SOUND SOURCES, TRANSMISSION PATHS, AND RECOMMENDED NOISE REDUCTION METHODS

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 Reflected sound from walls, ceiling, and walls	Direct sound can be controlled only by selecting quiet equipment 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

Table 14-37 SOUND SOURCES, TRANSMISSION PATHS, AND RECOMMENDED NOISE REDUCTION METHODS (Cont.)

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 14-38 SPECIFIC SOUND POWER LEVELS, K_w, FOR FAN TOTAL SOUND POWER

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 (inches)								
	> 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.w.g.)								
Material Wheel	4-10	56	47	43	39	37	32	29	26
Medium Pressure	6-15	58	54	45	42	38	33	29	26
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 (inches)								
	> 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 14-39 BLADE FREQUENCY INCREMENTS (BFI)

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
Vaneaxial	125 Hz	6
Tubeaxial	63 Hz	7
Propeller		
Cooling Tower	3 Hz	5

Table 14-40 CORRECTION FACTOR, C, FOR OFF-PEAK OPERATION

Static Efficiency % of Peak	Correction Factor dB
90 to 100	0
85 to 89	3
75 to 85	6
65 to 74	9
55 to 64	12
50 to 54	15
below 50	16

Table 14-41 TL_{out} vs. FREQUENCY FOR VARIOUS RECTANGULAR DUCTS

Duct Size		1/1 Octave Band Center Frequency—Hz							
(in. × in.)	Gauge	63	125	250	500	1000	2000	4000	8000
12 × 12	24 ga.	21	24	27	30	33	36	41	45
12 × 24	24 ga.	19	22	25	28	31	35	41	45
12 × 48	22 ga.	19	22	25	28	31	37	43	45
24 × 24	22 ga.	20	23	26	29	32	37	43	45
24 × 48	20 ga.	20	23	26	29	31	39	45	45
48 × 48	18 ga.	21	24	27	30	35	41	45	45
48 × 96	18 ga.	19	22	25	29	35	41	45	45

Data are for duct lengths of 20 feet, but values may be used for the cross section shown regardless of length.

Table 14-42 TL_{in} vs. FREQUENCY FOR VARIOUS RECTANGULAR DUCTS

Duct Size		1/1 Octave Band Center Frequency—Hz							
(in. × in.)	Gauge	63	125	250	500	1000	2000	4000	8000
12 × 12	24 ga.	16	16	16	25	30	33	38	42
12 × 24	24 ga.	15	15	17	25	28	32	38	42
12 × 48	22 ga.	14	14	22	25	28	34	40	42
24 × 24	22 ga.	13	13	21	26	29	34	40	42
24 × 48	20 ga.	12	15	23	26	28	36	42	42
48 × 48	18 ga.	10	19	24	27	32	38	42	42
48 × 96	18 ga.	11	19	22	26	32	38	42	42

Data are for duct lengths of 20 feet, but values may be used for the cross section shown regardless of length.

Table 14-43 EXPERIMENTALLY MEASURED TL_{out} vs. FREQUENCY FOR ROUND DUCTS

Duct Size			1/1 Octave Band Center Frequency—Hz							
Diam.	Length	Gauge	63	125	250	500	1000	2000	4000	8000
Long Seam Ducts										
8 in.	15 ft	26 ga.	45	53	55	52	44	35	34	26
14 in.	15 ft	24 ga.	50	60	54	36	34	31	25	38
22 in.	15 ft	22 ga.	47	53	37	33	33	27	25	43
32 in.	15 ft	22 ga.	51	46	26	26	24	22	38	43
Spiral Wound Ducts										
8 in.	10 ft	26 ga.	48	64	75	72	56	56	46	29
14 in.	10 ft	26 ga.	43	53	55	33	34	35	25	40
26 in.	10 ft	24 ga.	45	50	26	26	25	22	36	43
26 in.	10 ft	16 ga.	48	53	36	32	32	28	41	36
32 in.	10 ft	22 ga.	43	42	28	25	26	24	40	35

Table 14-44 CALCULATED TL_{out} vs. FREQUENCY FOR ROUND DUCTS

Duct Size Diam.	Length	Gauge	1/1 Octave Band Center Frequency—Hz 63	125	250	500	1000	2000	4000
Long Seam Ducts									
8 in.	15 ft	26 ga.	50	50	50	44	42	40	38
14 in.	15 ft	24 ga.	50	50	48	37	35	33	31
22 in.	15 ft	22 ga.	50	50	38	32	30	28	26
32 in.	15 ft	22 ga.	50	44	29	26	24	22	37
Spiral Wound Ducts									
8 in.	10 ft	26 ga.	50	50	50	46	42	40	38
14 in.	10 ft	26 ga.	50	50	48	35	33	31	29
26 in.	10 ft	24 ga.	45	45	35	27	25	23	38
26 in.	10 ft	16 ga.	50	50	42	34	32	30	45
32 in.	10 ft	22 ga.	50	47	32	26	24	22	37

Table 14-45 EXPERIMENTALLY DETERMINED TL_{in} vs. FREQUENCY FOR ROUND DUCTS

Duct Size Diam.	Length	Gauge	1/1 Octave Band Center Frequency—Hz 63	125	250	500	1000	2000	4000	8000
Long Seam Ducts										
8 in.	15 ft	26 ga.	17	31	39	42	41	32	31	23
14 in.	15 ft	24 ga.	27	43	43	31	31	28	22	35
22 in.	15 ft	22 ga.	28	40	30	30	30	24	22	40
32 in.	15 ft	22 ga.	35	36	23	23	21	19	35	40
Spiral Wound Ducts										
8 in.	10 ft	26 ga.	20	42	59	62	53	43	26	26
14 in.	10 ft	26 ga.	20	36	44	28	31	32	22	37
26 in.	10 ft	24 ga.	27	38	20	23	22	19	33	40
26 in.	10 ft	16 ga.	30	41	30	29	29	25	38	33
32 in.	10 ft	22 ga.	27	32	25	22	23	21	37	42

Table 14-46 CALCULATED TL_{in} vs. FREQUENCY FOR ROUND DUCTS

Duct Size Diam.	Length	Gauge	1/1 Octave Band Center Frequency—Hz 63	125	250	500	1000	2000	4000
Long Seam Ducts									
8 in.	15 ft	26 ga.	17	23	29	34	39	37	35
14 in.	15 ft	24 ga.	22	28	34	32	32	30	28
22 in.	15 ft	22 ga.	26	32	31	29	27	25	23
32 in.	15 ft	22 ga.	29	34	26	23	21	19	34
Spiral Wound Ducts									
8 in.	10 ft	26 ga.	17	23	29	35	39	37	35
14 in.	10 ft	26 ga.	27	38	37	30	30	28	26
26 in.	10 ft	24 ga.	27	33	29	24	22	20	35
26 in.	10 ft	16 ga.	27	33	36	31	29	27	42
32 in.	10 ft	22 ga.	29	35	29	23	21	19	34

Table 14-47 TL_{out} vs. FREQUENCY FOR VARIOUS FLAT-OVAL DUCTS

Duct Size (in. × in.)	Gauge	1/1 Octave Band Center Frequency—Hz 63	125	250	500	1000	2000	4000	8000
12 × 6	24 ga.	31	34	37	40	43	—	—	
24 × 6	24 ga.	24	27	30	33	36	—	—	—
24 × 12	24 ga.	28	31	34	37	—	—	—	—
48 × 12	22 ga.	23	26	29	32	—	—	—	—
48 × 24	22 ga.	27	30	33	—	—	—	—	—
96 × 24	20 ga.	22	25	28	—	—	—	—	—
96 × 48	18 ga.	28	31	—	—	—	—	—	—

Table 14-48 TL_{in} vs. FREQUENCY FOR VARIOUS FLAT-OVAL DUCTS

Duct Size (in. × in.)	Gauge	1/1 Octave Band Center Frequency—Hz 63	125	250	500	1000	2000	4000	8000
12 × 6	24 ga.	18	18	22	31	40	—	—	—
24 × 6	24 ga.	17	17	18	30	33	—	—	—
24 × 12	24 ga.	15	16	25	34	—	—	—	—
48 × 12	22 ga.	14	14	26	29	—	—	—	—
48 × 24	22 ga.	12	21	30	—	—	—	—	—
96 × 24	20 ga.	11	22	25	—	—	—	—	—
96 × 48	18 ga.	19	28	—	—	—	—	—	—

Table 14-49 ABSORPTION COEFFICIENTS FOR SELECTED PLENUM MATERIALS

	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 14-50 SOUND ATTENUATION IN UNLINED RECTANGULAR SHEET METAL DUCTS

		Attenuation—dB/ft 1/1 Octave Band Center Freq.—Hz			
Duct Size in. × in.	P/A 1/ft	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

If duct is externally lined, multiply results associated with 63 Hz, 125 Hz and 250 Hz by 2.

Table 14-51 COEFFICIENTS FOR EQUATION 11-102 IN CHAPTER 11

1/1 Octave Band Center Freq.—Hz	B	C	D
63	0.0133	1.959	0.917
125	0.0574	1.410	0.941
250	0.2710	0.824	1.079
500	1.0147	0.500	1.087
1,000	1.7700	0.695	0.000
2,000	1.3920	0.802	0.000
4,000	1.5180	0.451	0.000
8,000	1.5810	0.219	0.000

Table 14-52 INSERTION LOSS FOR RECTANGULAR DUCTS WITH 1″ OF FIBERGLASS LINING

	Insertion Loss—dB/ft 1/1 Octave Band Center Frequency—Hz							
DIMENSIONS in. × in.	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

Table 14-52 INSERTION LOSS FOR RECTANGULAR DUCTS WITH 1″ OF FIBERGLASS LINING (Cont.)

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	01.79	1.41	1.54	1.60
72	0.14	0.13	0.29	0.94	1.58	1.22	1.42	1.54

Table 14-53 INSERTION LOSS FOR RECTANGULAR DUCTS WITH 2″ OF FIBERGLASS LINING

	Insertion Loss—dB/ft 1/1 Octave Band Center Frequency—Hz							
DIMENSIONS in. × in.	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	5.36	6.23	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

Table 14-53 INSERTION LOSS FOR RECTANGULAR DUCTS WITH 2″ OF FIBERGLASS LINING (Cont.)

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 × 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 14-54 SOUND ATTENUATION IN STRAIGHT ROUND DUCTS

	Attenuation—dB/ft 1/1 Octave Band Center Frequency—Hz						
Diameter—in.	63	125	250	500	1000	2000	4000
$D \leq 7$	0.03	0.03	0.05	0.05	0.10	0.10	0.10
$7 < D \leq 15$	0.03	0.03	0.03	0.05	0.07	0.07	0.07
$15 < D \leq 30$	0.02	0.02	0.02	0.03	0.05	0.05	0.05
$30 < D \leq 60$	0.01	0.01	0.01	0.02	0.02	0.02	0.02

Table 14-55 CONSTANTS FOR USE IN EQUATION 11-104 IN CHAPTER 11

1/1 Octave Band Center Freq.-Hz	A	B	C	D	E	F
63	0.2825	0.3447	−5.251E-2	−3.837E-2	9.132E-4	−8.294E-6
125	0.5237	0.2234	−4.936E-3	−2.724E-2	3.377E-4	−2.490E-4
250	0.3652	0.7900	−0.1157	−1.834E-2	−1.211E-4	2.681E-4
500	0.1333	1.8450	−0.3735	−1.293E-2	8.624E-5	−4.986E-6
1000	1.933	0.0	0.0	6.135E-2	−3.891E-3	3.934E-5
2000	2.730	0.0	0.0	−7.341E-2	4.428E-4	1.006E-6
4000	2.800	0.0	0.0	−0.1467	3.404E-3	-2.851E-5
8000	1.545	0.0	0.0	−5.452E-2	1.290E-3	−1.318E-5

Table 14-56 INSERTION LOSS FOR ACOUSTICALLY LINED ROUND DUCTS—1 INCH LINING

Diameter (inches)	Insertion Loss—dB/ft 1/1 Octave Band Center Frequency—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	0.11	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 14-57 INSERTION LOSS FOR ACOUSTICALLY LINED ROUND DUCTS—2 INCH LINING

Diameter (inches)	Insertion Loss—dB/ft 1/1 Octave Band Center Frequency—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 14-58 INSERTION LOSS FOR ACOUSTICALLY LINED ROUND DUCTS—3 INCH LINING

Diameter (inches)	Insertion Loss—dB/ft 1/1 Octave Band Center Frequency—Hz 63	125	250	500	1000	2000	4000	8000
6	0.64	1.00	1.58	2.23	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 14-59 INSERTION LOSS OF UNLINED AND LINED SQUARE ELBOWS WITHOUT TURNING VANES

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

$fw = f \times w$ where f is the VI octave center frequency (kHz) and w is the width of the elbow (inches).

Table 14-60 INSERTION LOSS OF ROUND ELBOWS

fw	Insertion loss—dB
fw < 1.9	0
1.9 < fw < 3.8	1
3.8 < fw < 7.5	2
fw > 7.5	3

fw = f × w where f is the Vl octave center frequency (kHz) and w is the width of the elbow (inches).

Table 14-61 INSERTION LOSS OF UNLINED AND LINED SQUARE ELBOWS WITH TURNING VANES

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	4	4
7.5 < fw < 15	6	7
fw > 15	4	7

fw = f × w where f is the Vl octave center frequency (kHz) and w is the width of the elbow (inches).

Table 14-62 7 FOOT, RECTANGULAR, STANDARD PRESSURE DROP DUCT SILENCERS

Face Velocity fpm	1/1 Octave Band Center Frequency—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 14-63 7 FOOT, RECTANGULAR, LOW PRESSURE DROP DUCT SILENCERS

Face Velocity fpm	1/1 Octave Band Center Frequency—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 14-64 ROUND, HIGH PRESSURE DROP DUCT SILENCERS

Face Velocity fpm	1/1 Octave Band Center Frequency—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 14-65 ROUND, LOW PRESSURE DROP DUCT SILENCERS

Face Velocity fpm	1/1 Octave Band Center Frequency—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 14-66 COEFFICIENTS FOR DETERMINING STATIC PRESSURE DROP ACROSS DUCT SILENCERS

Rectangular	C1	C2	C3	C4
High Pressure Drop	0.6464	0.3971	2.637×10^{7}	2.012
Low Pressure Drop	0.6015	0.4627	9.802×10^{-8}	2.011
Circular				
High Pressure Drop	5.108×10^{-3}	1.999	4.007×10^{-8}	2.002
Low Pressure Drop	5.097×10^{-3}	2.000	1.104×10^{-8}	2.022

Table 14-67 COEFFICIENT FOR SYSTEM COMPONENT EFFECT ON 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 14-68 TRANSMISSION LOSS VALUES FOR CEILING MATERIALS

		Transmission Loss—dB 1/1 Octave Band Center Frequency—Hz						
Thickness (inches)	Surface Weight (lb/ft^2)	63	125	250	500	1000	2000	4000
Gypsum Board								
3/8	1.6	6	11	17	22	28	32	24
1/2	2.1	9	14	20	24	30	31	27
5/8	2.7	10	15	22	26	31	28	30
1	4.6	13	18	26	30	30	29	37

		Transmission Loss—dB 1/1 Octave Band Center Frequency—Hz						
Dimensions (ft. × ft. × in.)	Surface Weight (lb/ft^2)	63	125	250	500	1000	2000	4000
Acoustical Ceiling Tile—Exposed T-Bar Grid Suspended Lay-in-Ceilings								
2 × 4 × 5/8	0.6–0.7	4	9	9	14	19	24	26
2 × 4 × 1-1/2	0.9	4	9	10	11	15	20	25
2 × 4 × 5/8	0.95–1.1	5	11	13	15	21	24	28
2 × 2 × 5/8	1.2–1.3	5	10	11	15	19	22	24
Acoustical Ceiling Tile—Concealed Spline Suspended Ceiling								
1 × 1 × 5/8	1.2	6	14	14	18	22	27	30

Table 14-69 CORRECTION COEFFICIENT "τ" FOR DIFFERENT TYPES OF CEILINGS

Ceiling Configuration	τ
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 14-70 AVERAGE SOUND ABSORPTION COEFFICIENTS, α, FOR TYPICAL RECEIVING ROOMS

	Room Absorption Coefficient 1/1 Octave Band Center Frequency—Hz						
Type of Room	63	125	250	500	1000	2000	4000
Dead Acoustical ceiling, plush carpet, soft furnishings, people	0.26	0.30	0.35	0.4	0.43	0.46	0.52
Medium Dead Acoustical ceiling, commercial carpet, people	0.24	0.22	0.18	0.25	0.30	0.36	0.42
Average Acoustical ceiling or commercial carpet, people	0.25	0.23	0.17	0.20	0.24	0.29	0.34
Medium Live Some acoustical material, people	0.25	0.23	0.15	0.15	0.17	0.20	0.23
Live People only	0.26	0.24	0.12	0.10	0.09	0.11	0.13

Table 14-71 AIR ABSORPTION COEFFICIENTS

	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 14-72 DECIBEL EQUIVALENTS OF NUMBERS (N)

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	12	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 14-73 FIVE PLACE LOGARITHMS

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	1.86923	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	1,000	3.00000
.33	-1.51851	.82	-1.91381	33	1.51851	82	1.91381	2,000	3.30103
.34	-1.53148	.83	-1.91908	34	1.53148	83	1.91908	3,000	3.47712
.35	-1.54407	.84	-1.92428	35	1.54407	84	1.92428	4,000	3.60206
.36	-1.55630	.85	-1.92942	36	1.55630	85	1.92942	5,000	3.69897
.37	-1.56820	.86	-1.93450	37	1.56820	86	1.93450	6,000	3.77815
.38	-1.57978	.87	-1.93952	38	1.57978	87	1.93952	7,000	3.84510
.39	-1.59106	.88	-1.94448	39	1.59106	88	1.94448	8,000	3.90309
.40	-1.60206	.89	-1.94939	40	1.60206	89	1.94939	9,000	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

CHAPTER 15
GLOSSARY

A

A-Scale: A filtering system that has characteristics which roughly match the response characteristics of the human ear at low sound levels (below 55 dB Sound Pressure Level, but frequently used to gauge levels to 85 dB). A-scale measurements are often referred to as dB(A).

Absolute Filter: Obsolete term (See HEPA filter)

Absolute Pressure: Air at standard conditions (70°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 gauge reads zero. So the *absolute pressure* of a system is the gauge pressure in pounds per square inch added to the atmospheric pressure of 14.696 psi (use 14.7 psi in *environmental system work*) and the symbol is "psia."

Absorbent: A material which, due to an affinity for certain substances, extracts one or more such substances from a liquid or gaseous medium with which it contacts and which changes physically or chemically, or both, during the process. Calcium chloride is an example of a solid absorbent, while solutions of lithium chloride, lithium bromide, and ethylene glycols are liquid absorbents.

Absorber Surface: The surface of the collector plate which absorbs solar energy and transfers it to the collector plate.

Absorptance: The ratio of the amount of radiation absorbed by a surface to the amount of radiation incident upon it.

Absorption: A process whereby a material extracts one or more substances present in an atmosphere or mixture of gases or liquids accompanied by the material's physical and/or chemical changes.

Absorption Coefficient: For a surface, the ratio of the sound energy absorbed by a surface of a medium (or material) exposed to a sound field (or to sound radiation) divided by the sound energy incident on the surface. The stated values of this are to hold for an infinite area of the surface. The conditions under which measurements of absorption coefficients are made must be stated explicitly. The absorption coefficient is a function of both angle of incidence and frequency. Tables of absorption coefficients usually list the absorption coefficients at various frequencies, the values being those obtained by averaging over all angles of incidence.

Absorption Unit: Is a factory tested assembly of component parts producing refrigeration for comfort cooling by the application of heat. This definition shall apply to those absorption units which also produce comfort heating.

Acceleration: The time rate of change of velocity; i.e., the derivative of velocity; with respect to time.

Acceleration Due to Gravity: The rate of increase in velocity of a body falling freely in a vacuum. Its value varies with latitude and elevation. The International Standard is 32.174 ft. per second per second.

Acceptance Test: A test made upon completion of fabrication, receipt, installation or modification of a component unit or system to verify that it meets the requirements specified.

Accuracy: The extent to which the value of a quantity indicated by an instrument under test agrees with an accepted value of the quantity.

Actuator: A controlled motor, relay or solenoid in which the electric energy is converted into a rotary, linear, or switching action. An actuator can effect a change in the controlled variable by operating the final control elements a number of times. Valves and dampers are examples of mechanisms which can be controlled by actuators.

Adiabatic Process: A thermodynamic process during which no heat is added to, or taken from, a substance or system.

Adjustable Differential: A means of changing the difference between the control cut-in and cutout points.

Adsorbent: A material which has the ability to cause molecules of gases, liquids, or solids to adhere to its internal surfaces without changing the adsorbent physically or chemically. Certain solid materials, such as silica gel and activated alumina, have this property.

Adsorption: The action, associated with the surface adherence, of a material in extracting one or more substances present in an atmosphere or mixture of gases and liquids, unaccompanied by physical or chemical change. Commercial adsorbent materials have enormous internal surfaces.

Aerodynamic Noise: Also called generated noise, self-generated noise; is noise of aerodynamic origin in a moving fluid arising from flow instabilities. In duct systems, aerodynamic noise is caused by airflow through elbows, dampers, branch wyes, pressure reduction devices, silencers and other duct components.

Air, Ambient: Generally speaking, the air surrounding an object.

Airborne Noise: Noise which reaches the observer by transmission through air.

Air, Dry: Air without contained water vapor; air only.

Air, Outdoor: Air taken from outdoors and, therefore, not previously circulated through the system.

Air, Outside: External air, atmosphere exterior to refrigerated or conditioned space; ambient (surrounding) air.

Air, Recirculated: Return air passed through the conditioner before being again supplied to the conditioned space.

Air, Reheating of: In an air conditioning system, the final step in treatment, in the event the temperature is too low.

Air, Return: Air returned from conditioned or refrigerated space.

Air, Saturated: Moist air in which the partial pressure of the water vapor is equal to the vapor pressure of water at the existing temperature. This occurs when dry air and saturated water vapor coexist at the same dry-bulb temperature.

Air, Standard: Dry air at a pressure of 29.92 in. Hg at 69.8°F temperature and with a specific volume of 13.33 ft.^{3}/lb.

Air Change Rate: The number of times the total air volume of a defined space is replaced in a given unit of time. Ordinarily computed by dividing the total volume of the subject space (in cubic feet) into the total volume of air exhausted from the space per unit of time.

Air Changes: A method of expressing the amount of air leakage into or out of a building or room in terms of the number of building volumes or room volumes exchanged.

Air Conditioner, Unitary: An evaporator, compressor, and condenser combination; designed in one or more assemblies, the separate parts designed to be assembled together.

Air Conditioning, Comfort: The process of treating air so as to control simultaneously its temperature, humidity, cleanliness and distribution to meet the comfort requirements of the occupants of the conditioned space.

Air Conditioning Unit: An assembly of equipment for the treatment of air so as to control, simultaneously, its temperature, humidity, cleanliness and distribution to meet the requirements of a conditioned space.

Air Cooler: A factory-encased assembly of elements whereby the temperature of air passing through the device is reduced.

Air Diffuser: A circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of pimary air with secondary room air.

Air Gap: An air gap in a potable water distribution system is the unobstructed vertical distance through the free atmosphere between the lowest opening from any pipe or faucet supplying water to a tank, plumbing fixture or other device and the floor level rim of the receptacle.

Air Shower: A relatively small, isolated "chamber" normally located at the main entrance of a cleanroom. Designed allegedly to remove particulate from personnel and garments by high velocity air.

Air, Supply: That air delivered to the conditioned space and used for ventilation, heating, cooling, humidification or dehumidification.

Air, Transfer: The movement of indoor air from one space to another.

Air, Ventilation: That portion of supply air which is outdoor air plus any recirculated air that has been treated for the purpose of maintaining acceptable indoor air quality.

Air Washer: A water spray system or device for cleaning, humidifying, or dehumidifying the air.

Airborne Sound: Sound which reaches the point of interest by radiation through the air.

Airlock: An area between the entrance to the cleanroom and the entry from an outside area. The airlock receives the same clean, filtered air as the cleanroom, and is designed to prevent contaminated air in the outside area from flowing into the cleanroom. (Also referred to as "Ante-Room.")

Algae: A minute fresh water plant growth which forms a scum on the surfaces of recirculated water apparatus, interfering with fluid flow and heat transfer.

Alternating Current (AC): A source of power for an electrical circuit which periodically reverses the polarity of its charge.

Ambient: The existing surrounding environmental conditions (Temperature, Relative Humidity, Pressure, etc. . . .) of a particular area of consideration.

Ampacity: A wire's ability to carry current safely, without undue heating. The term formerly used to describe this characteristic was current-capacity of the wire.

Amperage: The flow of current in an electrical circuit measured in "amperes," abbreviated "amps" (A).

Amplitude of Ground Surface Temperature Variation: Peak Annual fluctuation of ground surface temperature about a mean value.

Anemometer: An instrument for measuring the velocity of a fluid.

Anemometer, Shielded Hot-Wire: An instrument for measuring air velocities based on the convective cooling effect of airflow on a heated wire. Instruments of this type are specifically designed for low air speeds, ranging from about 25 to 300 feet per minute (fpm).

Anticipating Control: One which, by artificial means, is activated sooner than it would be without such means, to produce a smaller differential of the controlled property. Heat and cool anticipators are commonly used in thermostats.

Anticipators: A small heater element in two-position temperature contollers which deliberately cause false indications of temperature in the controller in an attempt to minimize the override of the differential and smooth out the temperature variation in the controlled space.

Approach: In an evaporative cooling device, the difference between the average temperature of the circulating water leaving the device and the average wet-bulb temperature of the entering air. In a conduction heat exchanger device, the temperature difference between the leaving treated fluid and the entering working fluid.

Area: Generally used to designate a portion of a building at a given level of protection or contamination control, as set off from adjoining portions of different contamination levels. Used somewhat interchangeably with "SPACE" or "ZONE."

As-Built Facility: A cleanroom which is complete and operating, with all services connected and functioning, but has no production equipment or operating personnel within the facility.

As-Found Data: Data comparing the response of an instrument to known standards as determined without adjustment after the instrument is made operational.

Aspect Ratio: In air distribution outlets, the ratio of the length of the core opening of a grille, face, or register to the width. In rectangular ducts, the ratio of the width to the depth.

Aspiration: Production of movement in a fluid by suction created by fluid velocity.

At-Rest Facility: A cleanroom which is complete and has the production equipment installed, but has no personnel within the facility.

Attenuation: The transmission loss or reduction in magnitude of a signal between two points in a transmission system.

Autumnal Equinox (See Also Vernal Equinox): The position of the sun midway between its lowest and highest altitude during the autumn; it occurs September 21.

Auxiliary Contacts: A set of contacts that perform a secondary function, usually in relation to the operation of a set of primary contacts.

Averaging Element: A thermostat sensing element which will respond to the average duct temperature.

Azimuth Angle (Solar): The angular direction of the sun with respect to true south.

B

Backflow: The unintentional reversal of flow in a potable water distribution system which may result in the transport of foreign materials or substances into the other branches of the distribution system.

Background Noise: Sound other than the signal wanted. In room acoustics, it is the irreducible noise level measured in the absence of any building occupants when all of the known sound sources have been turned off.

Barometer: Instrument for measuring atmospheric pressure.

Basic Principles: Essential theory and understanding of operation.

Bimetallic Element: One formed of two metals having different coefficients of thermal expansion such as are used in temperature indicating and controlling devices.

Boiling Point: The temperature at which the vapor pressure of a liquid equals the absolute external pressure at the liquid-vapor interface.

Branch Circuit: Wiring between the last overcurrent device and the branch circuit outlets.

Breakout Noise: The transmission or radiation of noise through some part of the duct system to an occupied space in the building.

British Thermal Unit (Btu): The Btu is defined as the heat required to raise the temperature of a pound of water from 59° to 60°F.

Btuh: Number of Btu's transferred during a period of one hour.

Bulb: The name given to the temperature sensing device located in the fluid for which control or indication is provided. The bulb may be liquid-filled, gas-filled, or gas-and-liquid filled. Changes in temperature produce pressure changes within the bulb which are transmitted to the controller.

Building Envelope: The elements of a building which enclose conditioned spaces through which energy may be transferred to or from the exterior.

Bus Bar: A heavy, rigid metallic conductor which carries a large current and makes a common connection between several circuits. Bus bars are usually uninsulated and located where the electrical service enters a building; that is, in the main distribution cabinet.

Bus Duct: An assembly of heavy bars of copper or aluminum that acts as a conductor of large capacity.

Bypass: A pipe or duct, usually controlled by valve or damper, for conveying a fluid around an element of a system.

C

Calibration: Comparison of a measurement standard or instrument of unknown accuracy with another standard or instrument of known accuracy to detect, correlate, report, or eliminate by adjustment, any variation in the accuracy of the unknown standard or instrument.

Calibration, Field: Calibration test performed in the field in accordance with the manufacturer's recommendation and/or accepted industry practices.

Calibration, On-Line: Calibration performed using the reference system built into the instrument, in accordance with manufacturer's recommendations and/or accepted industry standards.

Capacitance: The property of an electric current that permits the storage of electrical energy in an electrostatic field and the release of that energy at a later time.

Capacitor (condenser): An electrical device that will store an electric charge used to produce a power factor change.

Capacity, Latent: The available refrigerating capacity of an air conditioner for removing latent heat from the space to be conditioned.

Capillary: The name given to the thin tube attached to the bulb which transmits the bulb pressure changes to the controller or indicator. The cross sectional area of the capillary is extremely small compared to the cross section of the bulb so that the capillary, which is usually outside of the controlled fluid, will introduce the smallest possible error in the signal being transmitted from the bulb.

Capillary Tube: The capillary tube is a metering device made from a thin tube approximately 2 to 20 feet long and from 0.025 to 0.090 inches in diameter which feeds liquid directly to the evaporator. Usually limited to systems of 1 ton or less, it performs all of the functions of the thermal expansion valve when properly sized.

Cathodic Protection: The process of providing corrosion protection against electrolytic reactions that could be deleterious to the performance of the protected material or component.

Ceiling Outlet: A round, square, rectangular, or linear air diffuser located in the ceiling which provides a horizontal distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the occupied zone.

Celsius (Formerly Centigrade): A thermometric scale in which the freezing point of water is called 0°C and its boiling point 100°C at normal atmospheric pressure (14.696 psi).

Certificate of Compliance (Conformance): A written statement, signed by a qualified party, attesting that the items or services are in accordance with specified requirements, and accompanied by additional information to substantiate the statement.

Certification: The process of validation required to obtain a certificate of compliance.

Certification Agency: A company providing on-site, field certification services for profit or gain.

Change of State: Change from one phase, such as solid, liquid or gas, to another.

Changeover: The process of switching an air conditioning system from heating to cooling, or vice versa.

Channel: Term used to describe output of a load management system. Usually corresponds to a specific relay.

Chemical Compatibility: The ability of materials and components in contact with each other to resist mutual chemical degradation, such as that caused by electrolytic action.

Circuit: An electrical arrangement requiring a source of voltage, a closed loop of wiring, an electric load and some means for opening and closing it.

Circuit Breaker: A switch-type mechanism that opens automatically when it senses an overload (excess current).

Cleanroom: A specially constructed room in which the air supply, air distribution, filtration of air supply, materials of construction, and operating procedures are regulated to control airborne particle concentrations to meet appropriate cleanliness levels as defined by Federal Standard 209E.

Clean Zone: A defined space in which the concentration of airborne particles is controlled to specified limits.

Clearing a Fault: Eliminating a fault condition by some means. Generally taken to mean operation of the over-circuit device that opens the circuit and clears the fault.

Clo Value: A numerical representation of a clothing ensemble's thermal resistance. 1 Clo = 0.88 sq. ft. hr.°F/Btu.

Coanda Effect: The diversion of the normal fluid flow path from a jet by its attachment to an adjacent surface (wall or ceiling) caused by a low pressure region between the fluid flow path and the surface.

Coefficient of Discharge: For an air diffuser, the ratio of net area or effective area at vena contracta of an orificed airstream to the free area of the opening.

Coefficient of Expansion: The change in length per unit length or the change in volume per unit volume, per deg. change in temperature.

Coefficient of Performance (COP), Heat Pump: The ratio of the compressor heating effect (heat pump) to the rate of energy input to the shaft of the compressor, in consistent units, in a complete heat pump, under designated operating conditions.

Coil: A cooling or heating element made of pipe or tubing.

Cold Deck: The cooling section of a mixed air zoning system.

Collector Azimuth: The horizontal angle between true south and a line which is perpendicular to the plane of the collector that is projected on a horizontal plane.

Collector Plate: The component of a solar collector which transfers the heat from solar energy to a circulating fluid.

Collector (Solar): An assembly of components intended to capture usable solar energy.

Combustion: The act or process of burning.

Comfort Chart: A chart showing effective temperatures with dry-bulb temperatures and humidities (and sometimes air motion) by which the effects of various air conditions on human comfort may be compared.

Comfort Cooling: Refrigeration for comfort as opposed to refrigeration for storage or manufacture.

Comfort Zone: (Average) the range of effective temperatures over which the majority (50 percent or more) of adults feels comfortable; (extreme) the range of effective temperatures over which one or more adults feel comfortable.

Common Neutral: A neutral conductor that is common to, or serves, more than one circuit.

Compressibility: The ease which a fluid may be reduced in volume by the application of pressure, depends upon the state of the fluid as well as the type of fluid itself. In TAB work, consider that water may not be compressed. Air is a compressible gas, but that factor is usually not considered during normal testing and balancing procedures.

Compressor: The pump which provides the pressure differential to cause fluid to flow and in the pumping process increases pressure of the refrigerant to the high side condition. The compressor is the separation between low side and high side.

Concentration: The quantity of one constituent dispersed in a defined amount of another.

Concentrator: A reflective surface or refracting lens for directing insolation onto the absorber surface.

Condensate: The liquid formed by condensation of a vapor. In steam heating, water condensed from steam; in air conditioning, water extracted from air, as by condensation on the cooling coil of a refrigeration machine.

Condensation: Process of changing a vapor into liquid by extracting heat. Condensation of steam or water vapor is effected in either steam condensers or dehumidifying coils, and the resulting water is called condensate.

Condenser: The heat exchanger in which the heat absorbed by the evaporator and some of the heat of compression introduced by the compressor are removed from the system. The gaseous refrigerant changes to a liquid, again taking advantage of the relatively large heat transfer by the change of state in the condensing process.

Condenser: Electrical—see "capacitor."

Condensing Unit, Refrigerant: An assembly of refrigerating components designed to compress and liquify a specific refrigerant, consisting of one or more refrigerant compressors, refrigerant condensers, liquid receivers (when required) and regularly furnished accessories.

Conditioned Space: Space within a building which is provided with heated and/or cooled air or surfaces and, where required, with humidification or dehumidification means so as to maintain a space condition falling within the "comfort zone."

Conditions, Standard: A set of physical, chemical, or other parameters of a substance or system which defines an accepted reference state or forms a basis for comparison.

Conductance, Electrical: The reciprocal (opposite) of resistance and is the current carrying ability of any wire or electrical component. Resistance is the ability to oppose the flow of current.

Conductance, Surface Film: Time rate of heat flow per unit area under steady conditions between a surface and a fluid for unit temperature difference between the surface and fluid.

Conductance, Thermal: Time rate of heat flow through a body (frequently per unit area) from one of its bounding surfaces to the other for a unit temperature difference between the two surfaces, under steady conditions.

Conductivity, Thermal: The time rate of heat flow through unit area and unit thickness of a homogeneous material under steady conditions when a unit temperature gradient is maintained in the direction perpendicular to area. Materials are considered homogeneous when the value of the thermal conductivity is not affected by variation in thickness or in size of sample within the range normally used in construction.

Conductor, Thermal: A material which readily transmits heat by means of conduction.

Conduit: A round cross-section electrical raceway, of metal or plastic.

Connected Load: The sum of all loads on a circuit.

Connection in Parallel: System whereby flow is divided among two or more channels from a common starting point or header.

Connection in Series: System whereby flow through two or more channels is in a single path entering each succeeding channel only after leaving the first or previous channel.

Contamination: The presence of any unwanted substance, material or energy which adversely affects a product or procedure in a cleanroom.

Contactor: Electromagnetic switching device.

Contaminant: An unwanted airborne constituent that may reduced acceptability of the air.

Control: A device for regulation of a system or component in normal operation, manual or automatic. If automatic, the implication is that it is responsive to changes of pressure, temperature or other property whose magnitude is to be regulated.

Control Diagram (ladder diagram): A diagram that shows the control scheme only. Power wiring is not shown. The control items are shown between two vertical lines; hence, the name—ladder diagram.

Control Point: The value of the controlled variable which the controller operates to maintain.

Controlled Area: An air conditioned work space or room in which the particle concentration is lower than normal air conditioned spaces. A controlled area is not to be classified as a cleanroom, but some special filtration is required.

Controlled Device: One which receives the converted signal from the transmission system and translates it into the appropriate action in the environmental system. For example: a valve opens or closes to regulate fluid flow in the system.

Controller: An instrument which receives the signal from the sensing device and translates that signal into the appropriate corrective measure. The correction is then sent to the system controlled devices through the transmission system.

Convection: Transfer of heat by movement of fluid.

Convection, Forced: Convection resulting from forced circulation of a fluid, as by a fan, jet or pump.

Convection, Natural: Circulation of gas or liquid (usually air or water) due to differences in density resulting from temperature changes.

Conventional Flow (Nonlaminar Flow) Cleanroom: A cleanroom with non-uniform or mixed airflow patterns and velocities.

Cooling, Evaporative: Involves the adiabatic exchange of heat between air and water spray or wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.

Cooling, Regenerative: Process of utilizing heat which must be rejected or absorbed in one part of the cycle to function usefully in another part of the cycle by heat transfer.

Cooling Coil: An arrangement of pipe or tubing which transfers heat from air to a refrigerant or brine.

Cooling Effect, Sensible: Difference between the total cooling effect and the dehumidifying effect, usually in watts (Btuh).

Cooling Effect, Total: Difference between the total enthalpy of the dry air and water vapor mixture entering the cooler per hour and the total enthalpy of the dry air and water vapor mixture leaving the cooler per hour, expressed in watts (Btuh).

Cooling Range: In a water cooling device, the difference between the average temperatures of the water entering and leaving the device.

Core Area: The total plane area of that portion of a grille, included within lines tangent to the outer edges of the openings through which air can pass.

Corresponding Values: Simultaneous values of various properties of a fluid, such as pressure, volume, temperature, etc., for a given condition of fluid.

Corrosive: Having chemically destructive effect on metals (occasionally on other materials).

Counterflow: In heat exchange between two fluids, opposite direction of flow, coldest portion of one meeting coldest portion of the other.

Critical Surface: The surface of the work part to be protected from particulate contamination.

Critical Velocity: The velocity above which fluid flow is turbulent.

Cross Connection: Any physical connection or arrangement between two otherwise separate piping systems, one of which contains potable water and the other either water of unknown or questionable safety or steam, gas, chemicals, or other substances whereby there may be a flow from one system to the other, the direction of flow depending on the pressure differential between the two systems.

Crossflow: Horizontal airflow.

Crystal Formation, Zone of Maximum: Temperature range in freezing in which most freezing takes place, i.e., about 25°F to 30°F for water.

Curb Box: Access to an underground valve at the street curb. It controls water service to a house or building.

Current (I): The electric flow in an electric circuit, which is expressed in *amperes* (amps).

Cycle: A complete course of operation of working fluid back to a starting point, measured in thermodynamic terms (functions). Also in general for any repeated process on any system.

Cycle, Reversible: Theoretical thermodynamic cycle, composed of a series of reversible processes, which can be completely reversed.

D

DWV: Drainage, waste and vent.

Dalton's Law of Partial Pressure: Each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.

Damper: A device used to vary the volume of air passing through an air outlet, air inlet or duct.

Deadband: In HVAC, a temperature range in which neither heating nor cooling is turned on; in load management, a kilowatt range in which loads are neither shed nor restored.

Decay Rate: The rate at which the sound pressure level in an enclosed space decreases after the sound source has stopped. It is measured in decibels per second.

Decibel (dB): The unit "bel" is used in telecommunication engineering as a dimensionless unit for the logarithmic ratio of two power quantities. The decibel is one-tenth of a bel. Therefore:

$$L = 10 \log_{10} \left[\frac{\text{sound power}}{\text{reference power}}\right]$$

The referenced power for sound power level is 10^{-12} watts.

In noise control work, the decibel notation is used to indicate the magnitude of sound pressure and sound power.

Combining Decibels: In sound survey work, it is frequently necessary to combine sound pressure level readings. An example would be to evaluate the effect of adding a noise source in a room where the noise level is already considered borderline. Since the decibel scale is logarithmic, decibel values cannot be added directly. The correct procedure is to convert the decibels to intensity ratios, add the intensity ratios, and reconvert this sum into decibels.

Degree Day: A unit, based upon temperature difference and time, used in estimating fuel consumption and specifying nominal heating load of a building in winter. For any one day, when the mean temperature is less than 65°F, there exist as many degree days as there are Fahrenheit degrees difference in temperature between the mean temperature for the day and 65°F.

Dehumidification: The condensation of water vapor from air by cooling below the dewpoint or removal of water vapor from air by chemical or physical methods.

Dehumidifier: (1) An air cooler or washer used for lowering the moisture content of the air passing through it; (2) An absorption or adsorption device for removing moisture from air.

Dehydration: (1) removal of water vapor from air by the use of absorbing or adsorbing materials; (2) removal of water from stored goods.

Delta Service: An arrangement of the utility transformers. Commonly shown "Δ."

Demand: The probable maximum rate of water flow as determined by the number of water supply fixture units.

Demand Charge: That part of an electric bill based on kW demand and the demand interval, expressed in dollars per kilowatt. Demand charges offset construction and maintenance of a utility's need for a large generating capacity.

Demand Control: A device which controls the kW demand level by shedding loads when the kW demand exceeds a predetermined set point.

Demand Interval: The period of time during which kW demand is monitored by a utility service, usually 15 or 30 minutes long.

Demand Load: The actual amount of load on a circuit at any time. The sum of all the loads which are ON. Equal to the connected load minus the loads that are OFF.

Demand Reading: Highest or maximum demand for electricity an individual customer registers in a given interval, example, 15 minute interval. The metered demand reading sets the demand charge for the month.

Density: The ratio of the mass of a specimen of a substance to the volume of the specimen. The mass of a unit volume of a substance. When weight can be used without confusion, as synonymous with mass, density is the weight per unit volume.

Desiccant: Any absorbent or adsorbent, liquid or solid, that will remove water or water vapor from a material. In a refrigeration circuit, the desiccant should be insoluble in the refrigerant.

Design Working Pressure: The maximum allowable working pressure for which a specific part of a system is designed.

Dewpoint, Apparatus: That temperature which would result if the psychrometric process occurring in a dehumidifier, humidifier or surface-cooler were carried to the saturation condition of the leaving air while maintaining the same ratio of sensible to total heat load in the process.

Dew Point Depression: The difference between dry bulb and dew point temperatures (°F DB – °F DP).

Dew Point Temperature: (t_{dp}) The temperature at which moist air becomes saturated (100% relative humidity) with water vapor when cooled at constant pressure.

Dielectric Fitting: An insulating or nonconducting fitting used to isolate electrochemically dissimilar materials.

Differential: The difference between the points where a controller turns "on" and "off." If a thermostat turns a furnace on at 68° and the differential is 3°, the burner will be turned off at 71°.

Diffuse Sound Field: A diffuse sound field is a space in which at every point the flow of sound energy in all directions is equally probable. (It is often assumed that in a diffuse field, the sound pressure level, averaged through time, is everywhere the same.)

Diffuser: A circular, square, or rectangular air distribution outlet, generally located in the ceiling and comprised of deflecting members discharging supply air in various directions and planes, and arranged to promote mixing of primary air with secondary room air.

Direct Acting: Instruments that increase control pressure as the controlled variable (such as temperature or pressure) increases; while *reverse acting* instruments increase control pressure as the controlled variable decreases.

Direct Current (DC): A source of power for an electrical circuit which does not reverse the polarity of its charge.

Direct Field: The sound in a region in which all or most of the sound arrives directly from the source without reflection.

Directivity Factor: The ratio of the sound pressure squared at some fixed distance and direction divided by the mean-squared sound pressure at the same distance averaged over all directions from the source.

Discharge Stop Valve: The manual service valve at the leaving connection of the compressor.

Discrete Logic: Electronic circuitry composed of standard transistors, resistors, capacitors, etc., as compared to microprocessor circuits where the logic is condensed on a single chip (integrated circuit).

Domestic Hot Water: Potable hot water as distinguished from hot water used for house heating.

D.O.P. (Dioctyl Phthalate): An aerosol generated by blowing air through liquid dioctyl phthalate. *Thermally generated D.O.P.* is an aerosol generated by condensing vapor that has been evaporated from liquid (D.O.P.) by heat. The aerosol mean particle diameter is between 0.2 and 0.4 micron with a maximum geometric standard deviation of 1.3.

D.O.P. Aerosol Generator, Air Operated: A device for producing a D.O.P. aerosol, operated by compressed air at room temperature, equipped with Laskin nozzles to produce a heterogeneous D.O.P. test aerosol.

D.O.P. Aerosol Generator, Pressurized Gas-Thermal: A device for producing D.O.P. aerosol, operated by pressurized gas and equipped with heating means.

Downflow: Vertical airflow (from ceiling to floor).

Draft: a) A current of air, when referring to the pressure difference which causes a current of air or gases to flow through a flue, chimney, heater, or space; or b) to a localized effect caused by one or more factors of high air velocity, low ambient temperature, or direction of air flow, whereby more heat is withdrawn from a person's skin than is normally dissipated.

Drier: A manufactured device containing a desiccant placed in the refrigerant circuit. Its primary purpose is to collect and hold within the desiccant, all water in the system in excess of the amount which can be tolerated in the circulating refrigerant.

Drift: Term used to describe the difference between the set point and the actual operating or control point.

Drip: A pipe, or a steam trap and a pipe considered as a unit, which conducts condensation from the steam side of a piping system to the water or return side of the system.

Droop: Terms used to describe the difference between the set point and the actual operating or control point.
Drop: The vertical distance that the lower edge of a horizontally projected airstream drops between the outlet and the end of its throw.
Dry bulb, Room: The dry bulb (dewpoint, etc.) temperature of the conditioned room or space.
Dry Bulb Temperature: The temperature registered by an ordinary thermometer. The dry bulb temperature represents the measure of sensible heat, or the intensity of heat.
Dry Bulb Temperature, Adjusted (t_{adb}): The average of the air temperature (t_a) and the mean radiant temperature (t_r) at a given location. The adjusted dry bulb temperature (t_{adb}) is approximately equivalent to operative temperature (t_o) at air motions less than 80 fpm when t_r is less than 120°F.
Duct: A passageway made of sheet metal or other suitable material, not necessarily leaktight, used for conveying air or other gas at low pressures.
Dust: An air suspension (aerosol) or particles of any solid material, usually with particle size less than 100 microns.
Dynamic Discharge Head: Static discharge head plus friction head plus velocity head.
Dynamic Insertion Loss: The dynamic insertion loss of a silencer, duct lining, or other attenuating device is the performance measured in accordance with ASTM E 477 when handling the rated airflow. It is the reduction in sound pressure level, expressed in decibels, due solely to the placement of the sound attenuating device in the duct system.
Dynamic Suction Head: Positive static suction head minus friction head and minus velocity head.
Dynamic Suction Lift: The sum of suction lift and velocity head at the pump suction when the source is below pump centerline.

E

Economizer: A system of dampers, temperature and humidity sensors, and motors which maximizes the use of outdoor air for cooling.
Effect, Humidifying: Latent heat of water vaporization at the average evaporating temperature times the number of pounds of water evaporated per hour in Btuh.
Effect, Sun: Solar energy transmitted into space through windows and building materials.
Effect, Total Cooling: The difference between the total enthalpy of the dry air and water vapor mixture entering a unit per hour and the total enthalpy of the dry air and water vapor (and water) mixture leaving the unit per hour, expressed in Btu per hour.
Effective Area: The net area of an outlet or inlet device through which air can pass, equal to the free area times the coefficient of discharge.
Effectiveness (Efficiency): The ratio of the actual amount of heat transferred by a heat recovery device to the maximum heat transfer possible between the airstreams (sensible heat/sensible heat, sensible heat/total heat, or total heat/total heat).
Elasticity of Demand: The change in quantity of electricity (or other commodity) purchased as a result of a change in its price. Demand for electricity is "elastic" when it increases or decreases in response to decreases or increases, respectively, in the price for the electricity.
Electrical Circuit: A power supply, a load, and a path for current flow are the minimum requirements for an electrical circuit.
Electromechanical: Converting electrical input into mechanical action. A relay is an electromechanical switch.
Electro-Pneumatic (EP) Switches: Switches that open or close an air line valve from an electrical impulse.
Electrostatic Discharge (E.S.D.): A transfer of electrostatic charge between objects at different electrostatic potentials caused by direct contact or induced by electrostatic field.
Emissivity: The property of a surface that determines its ability to give off radiant energy.
Emittance: The ratio of the radiant energy emitted by a body to the energy emitted by a black body at the same temperature.
End Reflection: When a duct system opens abruptly into a large room, some of the acoustic energy at the exit of the duct is reflected upstream with the result that the amount of the acoustic energy radiated into the room is reduced. This decrease in radiated energy increases as the frequency decreases.
Energy: Expressed in *kilowatt-hours* (kWh) or *watt-hours* (Wh), and is equal to the product of power and time.

energy = power × time
kilowatt-hours = kilowatts × hours
watt-hours = watts × hours

Energy (Consumption) Charge: That part of an electric bill based on kWh consumption (expressed in cents per kWh). Energy charge covers cost of utility fuel, general operating costs, and part of the amortization of the utility's equipment.
Energy Efficiency Ratio (EER), Cooling: The ratio of net cooling capacity in Btuh to total electric input in watts under designated operating conditions.
Engine: Prime mover; device for transforming fuel or heat energy into mechanical energy.
Enthalpy: The total quantity of heat energy contained in a substance, also called *total heat;* the thermodynamic property of a substance defined as the sum of its internal energy plus the quantity Pv J, where P = pressure of the substance, v = its volume, and J = the mechanical equivalent of heat.
Enthalpy, Specific: A term sometimes applied to enthalpy per unit weight.
Entrainment: The capture of part of the surrounding air by the airstream discharged from an outlet (sometimes called secondary air motion).
Entropy: The ratio of the heat added to a substance to the absolute temperature at which it is added.
Entropy, Specific: A term sometimes applied to entropy per unit weight.
Equal Friction Method: A method of duct sizing wherein the selected duct friction loss value is used constantly throughout the design of a low pressure duct system.
Equivalent Duct Diameter: The equivalent duct diameter for a rectangular duct with sides of dimensions a and b is 4 abπ.
Evaporation: Change of state from liquid to vapor.
Evaporative Cooling: The adiabatic exchange of heat between air and a water spray or wetted surface. The water approaches the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.
Evaporator: The heat exchanger in which the medium being cooled, usually air or water, gives up heat to the refrigerant through the exchanger transfer surface. The liquid refrigerant boils into a gas in the process of the heat absorption.
Exfiltration: Air leakage outward through cracks and interstices and through ceilings, floors and walls of a space or building.
Extended Surface: Heat transfer surface, one or both sides of which are increased in area by the addition of fins, discs, or other means.

F

Face Area: The total plane area of the portion of a grille, coil, or other items bounded by a line tangent to the outer edges of the openings through which air can pass.
Face Velocity: The velocity obtained by dividing the air quantity by the component face area.
Fahrenheit: A thermometric scale in which 32 (°F) denotes freezing and 212 (°F) the boiling point of water under normal pressure at sea level (14.696 psi).

Fail Safe: In load management, returning all loads to conventional control during a power failure. Accomplished by a relay whose contacts are normally closed.

Fan, Centrifugal: A fan rotor or wheel within a scroll type housing and including driving mechanism supports for either belt drive or direct connection.

Fan Performance Curve: Fan performance curve refers to the constant speed performance curve. This is a graphical presentation of static or total pressure and power input over a range of air volume flow rate at a stated inlet density and fan speed. It may include static and mechanical efficiency curves. The range of air volume flow rate which is covered generally extends from shutoff (zero air volume flow rate) to free delivery (zero fan static pressure). The pressure curves are generally referred to as the pressure-volume curves.

Fan Propeller: A propeller or disc type wheel within a mounting ring or plate and including driving mechanism supports for either belt drive or direct connection.

Fan, Tubeaxial: A propeller or disc type wheel within a cylinder and including driving mechanism supports for either belt drive or direct connection.

Fan, Vaneaxial: A disc type wheel within a cylinder, a set of air guide vanes located either before or after the wheel and including driving mechanism supports for either belt drive or direct connection.

Fault: A short circuit—either line to line, or line to ground.

Feed Line: A pipe that supplies water to items such as a boiler or a domestic hot water tank.

Filter: A device to remove solid material from a fluid.

Filter-Drier: A combination device used as a strainer and moisture remover.

Fin: An extended surface to increase the heat transfer area, as metal sheets attached to tubes.

Final Filter: The last stage of filtration before the airstream enters the clean space. The performance grade of this filter determines the air quality entering the clean space.

First Acoustically Critical Room: Most duct systems service a number of rooms. The room that has the shortest duct run from the fan is usually exposed to more fan noise than rooms further away from the fan. If this "first" room has the same noise criterion (NC) or a lower NC value than rooms further away from the fan, it may be assumed that, if the acoustical attenuation of the duct system from the fan to this "first" room satisfies the requirements for this "first" room, it also satisfies the acoustical requirements for rooms further away from the fan.

Fire Damper: A device, installed in an air distribution system, designed to close automatically upon detection of heat, to interrupt migratory airflow, and to restrict the passage of flame. A combination fire and smoke damper shall meet the requirements of both.

Fire Resistance Rating: The time, in minutes or hours, that materials or assemblies have withstood a fire exposure as established in accordance with the test procedures of NFPA 251, *Standard Methods of Fire Tests of Building Construction and Materials.*

Fire Wall: A wall having adequate fire resistance and structural stability under fire conditions to accomplish the purpose of subdividing buildings to restrict the spread of fire.

First Air: The air which issues directly from the HEPA filter before it passes over any work location.

First Work Location: The work location nearest the downstream side of the HEPA filters in a laminar airflow device or cleanroom.

Fixed Collector: A permanently oriented collector that has no provision for seasonal adjustment or tracking of the sun.

Flame Spread Rating: The flame spread rating of a material refers to a number or classification of material obtained according to NFPA 255, *Method of Test of Surface Burning Characteristics of Building Materials.*

Flanking (Sound) Transmission: The transmission of sound between two rooms by any indirect path of sound transmission.

Flat-Plate Collector: A collector without external concentrators or focusing devices, usually consisting of an absorber plate, cover plates, back and side insulation and a container.

Floating Action Controllers: Essentially two position type controllers which vary the position of the controlled devices but which are arranged to stop before reaching a maximum or minimum position.

Flow, Laminar or Streamline: Fluid flow in which each fluid particle moves in a smooth path substantially parallel to the paths followed by all other particles.

Flow, Turbulent: Fluid flow in which the fluid moves transversely as well as in the direction of the tube or pipe axis, as opposed to streamline or viscous flow.

Flue: A special enclosure incorporated into a building for the removal of products of combustion to the out-of-doors.

Type "A": A flue listed for use with oil, gas, or coal burning equipment.

Type "B": A manufactured flue listed for use with gas burning equipment.

Fluid: Gas, vapor, or liquid.

Fluid Head: The static pressure of fluid expressed in terms of the height of a column of the fluid, or of some manometric fluid, which it would support.

Fluid, Heat Transfer: Any gas, vapor, or liquid used to absorb heat from a source at a high temperature and reject it to a lower temperature substance.

Fluid Dynamics: Fluid Dynamics is used to describe the condition of motion of a fluid within a system. The velocity of a fluid is based upon the cross-sectional area and the volume of a fluid passing through it. The importance of this property is that volume may be determined for air or water systems when the area and velocity are known.

Fluid Statics: Fluid Statics as applied to TAB work, refers to a condition of a quantity of fluid at rest. It is the direct result of gravity and weight. Static pressure is used in both air and water testing to determine the potential for the movement of fluid within a system. Pressures in air systems are normally measured in units of inches of water (in.w.g.). A pressure unit of one inch of water is equivalent to the static pressure found at the base of a column of water one inch high. Pressures in water systems are normally measured in pounds per square inch (psi), but are converted to feet of water (ft. w.g.) for the purpose of evaluating pump and equipment performance.

Focusing Collector: A collector using some type of focusing device (parabolic mirror, fresnel lens, etc.) to concentrate the insolation on an absorbing element.

Force: The action on a body which tends to change its relative condition as to rest or motion.

Forced Circulation: Circulation of heat transfer fluid by a pump or fan.

Forward Flow: Forward flow occurs when air flows and noise propagates in the same direction, as in an air conditioning supply system or in a fan discharge.

Free Area: The total minimum area of the openings in the air outlet or inlet through which air can pass.

Free Delivery-Type Unit: A device which takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

Free Sound Field (Free Field): A free sound field is a field in a homogeneous, isotropic medium free from boundaries. In practice, it is a field in which the effects of the boundaries are negligible over the region of interest. In the free field, the sound pressure level decreases 6dB for a doubling of distance from a point source.

Freezing Point: Temperature at which a given liquid substance will solidify or freeze on removal of heat. Freezing point of water is 32°F.

Frequency: The number of vibrations or waves or cycles of any periodic phenomenon per second. In noise control of duct systems, our interest lies in the audible frequency range of 20 to 20,000 cycles per second. The United States has adopted the international designation of "hertz" (Hz.) for cycles per second.

Frequency Spectrum: A representation of a complex noise which has been resolved into frequency components. The most commonly used components are 1/1 octave bands and 1/3 octave bands.

Friction: Friction is the resistance found at the duct and piping walls. Resistance creates a static pressure loss in systems. The primary purpose of a fan or pump is to produce a design volume of fluid at a pressure equal to the frictional resistance of the system and the other dynamic pressure losses of the components.

Friction Head: The pressure in psi or feet of the liquid pumped which represents system resistance that must be overcome.

Full Load Current: See Running Current.

Fumes: Solid particles commonly formed by the condensation of vapors from normally solid materials such as molten metals. Fumes may also be formed by sublimation, distillation, calcination, or chemical reaction wherever such processes create airborne particles predominantly below one micron in size. Such solid particles sometimes serve as condensation nuclei for water vapor to form smog.

G

GFI, GFCI: Ground fault (circuit) interrupter—a device that senses ground faults and reacts by opening the circuit.

Gang: One wiring device position in a box.

Gas: Usually a highly superheated vapor which, within acceptable limits of accuracy, satisfies the perfect gas laws.

Gas, Inert: A gas that neither experiences nor causes chemical reaction nor undergoes a change of state in a system or process; e.g., nitrogen or helium mixed with a volatile refrigerant.

Gas Constant: The coefficient "R" in the perfect gas equation: $PV = MRT$.

Gradual Switches: Manual adjustment devices which proportion the control condition in accordance with the position of the switch.

Grains of Moisture: The unit of measurement of actual moisture contained in a sample of air. (7000 grains = one pound of water).

Gravity, Specific: Density compared to density of standard material; reference usually to water or to air.

Grille: A louvered or perforated covering for an air passage opening which can be located on a wall, ceiling or floor.

Ground: Zero voltage, or any point connected to the earth or "ground."

Ground Bus: A busbar in a panel or elsewhere, deliberately connected to ground.

Ground Conductor: Conductor run in an electrical system, which is deliberately connected to the ground electrode. Purpose is to provide a ground point throughout the system. Insulation color—*green*. Also called "green ground."

Ground Fault: An unintentional connection to ground.

H

Head, Dynamic or Total: In flowing fluid, the sum of the static and velocity heads at the point of measurement.

Head, Static: The static pressure of fluid expressed in terms of the height of a column of the fluid, or of some manometric fluid, which it would support.

Head, Velocity: In a flowing fluid, the height of the fluid or of some manometric fluid equivalent to its velocity pressure.

Heat: The form of energy that is transferred by virtue of a temperature difference.

Heat, Latent: Change of enthalpy during a change of state, usually expressed in Btu per lb. With pure substances, latent heat is absorbed or rejected at a constant temperature.

Heat, Sensible: Heat which is associated with a change in temperature; specific heat exchange of temperature; in contrast to a heat interchange in which a change of state (latent heat) occurs.

Heat, Specific: The ratio of the quantity of heat required to raise the temperature of a given mass of any substance one degree to the quantity required to raise the temperature of an equal mass of a standard substance (usually water at 59° F) one degree.

Heat, Total (Enthalpy): The sum of sensible heat and latent heat between an arbitrary datum point and the temperature and state under consideration.

Heat Capacity: The amount of heat necessary to raise the temperature of a given mass one degree. Numerically, the mass multiplied by the specific heat.

Heat Conductor: A material capable of readily conducting heat. The opposite of an insulator or insulation.

Heat Exchanger: A device specifically designed to transfer heat between two physically separated fluids.

Heat of Fusion: Latent heat involved in changing between the solid and the liquid states.

Heat of Vaporization: Latent heat involved in the change between liquid and vapor states.

Heat Pump: A refrigerating system employed to transfer heat into a space or substance. The condenser provides the heat while the evaporator is arranged to pick up heat from air, water, etc. By shifting the flow of air or other fluid, a heat pump system may also be used to cool the space.

Heat Transfer Medium: A fluid used in the transport of thermal energy.

Heat Transmission: Any time-rate of heat flow; usually refers to conduction, convection and radiation combined.

Heat Transmission Coefficient: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures.

Heating, Regenerative (or Cooling): Process of utilizing heat, which must be rejected or absorbed in one part of the cycle, to perform a useful function in another part of the cycle by heat transfer.

HEPA Filter (High Efficiency Particulate Air Filter): A throwaway extended media, dry-type filter in a rigid frame having a minimum particle-collection efficiency of 99.97 percent for 0.3 micron thermally-generated dioctyl phthalate (D.O.P.) or acceptable alternative particles, and a maximum clean-filter pressure drop of 1.0 inch water gauge, when tested at rated airflow capacity.

Heterogeneous Dioctyl Phthalate (D.O.P.): An aerosol having the approximate light-scattering mean droplet-size distribution as follows:

99 percent less than 3.0 micron
50 percent less than 0.7 micron
10 percent less than 0.4 micron

Hidden Demand Charge: Electric bill charges that are based on cents per kWh per kW demand contain a hidden demand charge. A low load factor for a building then penalizes the energy user through this "hidden" charge.

High Limit Control: A device which normally monitors the condition of the controlled medium and interrupts system operation if the monitored condition becomes excessive.

High Pressure Cutout: A pressure actuated switch to protect the compressor from pressure often caused by high condenser temperatures and pressure due to fouling and lack of water or air.

High Side: Parts of the refrigerating system subjected to condenser pressure or higher; the system from the compression side of the compressor through the condenser to the expansion point of the evaporator.

Horsepower: Unit of power in foot-pound-second system; work done at the rate of 550 ft-lb per sec, or 33,000 ft-lb per min.

Hot Deck: The heating section of a multizone system.

Hot Gas Bypass: The piping and manual, but more often automatic, valve used to introduce compressor discharge gas directly into the evaporator. This type of arrangement will maintain compressor operation at light loads down to zero by falsely loading the evaporator and compressor.

Hot Gas Piping: The compressor discharge piping which carries the hot refrigerant gas from the compressor to the condenser. Velocities must be high enough to carry entrained oil.

Humidifier: A device to add moisture to air.

Humidifying Effect: The latent heat of vaporization of water at the average evaporating temperature times the weight of water evaporated per unit of time.

Humidistat: A regulatory device, actuated by changes in humidity, used for the automatic control of relative humidity.

Humidity: Water vapor within a given space.

Humidity, Absolute: The weight of water vapor per unit volume.

Humidity, Percentage: The ratio of the specific humidity of humid air to that of saturated air at the same temperature and pressure, usually expressed as a percentage (degree of saturation; saturation ratio).

Humidity Ratio: The ratio of the mass of the water vapor to the mass of dry air contained in the sample.

Humidity, Relative: The ratio of the mol fraction of water vapor present in the air, to the mol fraction of water vapor present in saturated air at the same temperature and barometric pressure; approximately, it equals the ratio of the partial pressure or density of the water vapor in the air, to the saturation pressure or density, respectively, of water vapor at the same temperature.

Humidity, Specific: Weight of water vapor (steam) associated with 1 lb. weight of dry air, also called humidity.

Hunting: A condition which occurs when the desired condition cannot be maintained. The controller, controlled device and system, individually or collectively, continuously override or "overshoot" the control point with a resulting fluctuation and loss of control of the condition to be maintained.

Hydrostatic Pressure: The pressure at any point in a liquid at rest; equal to the depth of the liquid multiplied by its density.

Hygroscopic: Absorptive of moisture, readily absorbing and retaining moisture.

I

Impedance (Z): The quantity in an AC circuit that is equivalent to resistance in a DC circuit, inasmuch as it relates current and voltage. It is composed of *resistance* plus a purely AC concept called *reactance* and is expressed, like resistance, in *ohms*.

Impervious: Not air porous.

"In" Contacts: Those relay contacts which complete circuits when the relay armature is energized. Also referred to as Normally Open Contacts.

Inch of Water (in. w.g.): A unit of pressure equal to the pressure exerted by a column of liquid water 1 inch high at a temperature of 39.2°F.

Incidence, Angle of: The angle at which insolation strikes a surface.

Indicator: A term used to describe any device such as a thermometer or pressure gauge which is used to indicate the condition at a point in the system but which does not provide any controlling action or effect on the system operation.

Inductance: The process when a second conductor is placed next to a conductor carrying AC current (but not touching it), the ever-changing magnetic field will induce a current in the second conductor.

Induction: The capture of part of the ambient air by the jet action of the primary airstream discharging from a controlled device.

Inductive Loads: Loads whose voltage and current are out-of-phase. True power consumption for inductive loads is calculated by multiplying its voltage, current, and the power factor of the load.

Infiltration: Air flowing into a building as through a wall, crack, etc.

Input Override Relay: A relay that allows the duty cycle to be inhibited on specific channels because of inputs from outdoor temperature, space temperature, case temperature, time-of-day, etc. Sometimes called "duty cycle control relay."

Inrush Current: The current that flows the instant after the switch controlling current flow to a load is closed. Also called "locked rotor current."

Insertion Loss: The insertion loss of an element of an acoustic transmission system is the positive or negative change in acoustic power transmission that results when the element is introduced.

Insolation: The total amount of solar energy reaching a surface per unit of time.

Instantaneous Rate: Method for determining when load shedding should occur. Actual energy usage is measured and compared to a present kilowatt level. If the actual kilowatt level exceeds a designated set point, loads will be shed until the actual rate drops below the set point.

Insulation, Thermal: A material having a relatively high resistance to heat flow and used principally to retard heat flow.

Interstage Differential: In a multistage HVAC system, the change in temperature at the thermostat needed to turn additional heating or cooling equipment on.

Isentropic: An adjective describing a reversible adiabatic process; a change taking place at constant entropy.

Isobaric: An adjective used to indicate a change taking place at constant pressure.

Isokinetic Sampling: Any technique for collecting airborne particulate matter in which the collection is so designed that the airstream entering it has a velocity equal to that of the air passing around and outside the collector.

Isothermal: An adjective used to indicate a change taking place at constant temperature.

J

Junction Box: Metal box in which tap to circuit conductors is made. Junction box is not an outlet, since no load is fed from it directly.

K

Kilovolt Ampere: Product of the voltage times the current. Different from kilowatts because of inductive loads in an electrical system. Abbreviated: kVA kilowatts is equal to KVA times power factor.

Kilowatt: 1000 watts. Abbreviated: kW.

Kilowatt-Hour: A measure of electrical energy consumption. 1000 watts being consumed per hour. Abbreviated: kWh.

kW Demand: The maximum rate of electric power usage required to operate a facility during a period of time, usually a month or billing period. Often called "demand".

kWh Consumption: The amount of electric energy used over a period of time; the number of kWh used per month. Often called "consumption".

L

Lag: A delay in the effect of a changed condition at one point in the system, on some other condition to which it is related. Also, the delay in action of the sensing element of a control, due to the time required for the sensing element to reach equilibrium with the property being controlled; i.e., temperature lag, flow lag, etc.

Laminar (Unidirectional) Airflow: Airflow in which the entire body of air within a confined area moves with uniform velocity along parallel flow lines.

Laminar Airflow Cleanroom: A cleanroom in which the filtered air entering the room makes a single pass through the work area in a parallel airflow pattern, with a minimum of turbulent flow areas. Laminar airflow rooms have HEPA filter coverage of at least 80 percent of the ceiling (Vertical Flow) or one wall (Horizontal Flow), producing a uniform and parallel airflow. (Net filter medium face area versus gross area = 0.80.)

Laminar Flow Clean Air Device: A clean bench, clean work station, wall or ceiling hung module, or other device (except a cleanroom) which incorporates a HEPA filter(s) and motor-blower(s) for the purpose of supplying laminar flow clean air to a controlled work space.

Langley: Standard unit of insolation measurements, 1 langley = 1 cal/sq.cm. (1 langley/min. = 221 Btuh/sq.ft.).

Laskin Nozzle: A nozzle used for the generation of a heterogeneous D.O.P. aerosol by compressed gas (as defined in Air Generated D.O.P.).

Latent Heat of Fusion: The heat required to change a solid to a liquid at the same temperature; i.e. ice to water.

Latent Heat of Vaporization: The amount of heat necessary to change a quantity of water to water vapor without changing either temperature or pressure. When water is vaporized and passes into the air, the latent heat of vaporization passes into the air along with the vapor. Likewise, latent heat is removed when water vapor is condensed.

Law of Partial Pressure, Dalton's: Each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constitutents equals the total pressure of the mixture.

Level: The logarithm of the ratio, expressed in decibels, of two quantities proportional to power or energy. The quantity which is the denominator of the ratio is the standard reference quantity.

Light Emitting Diode: A low current and voltage light used as an indicator on load management equipment. Abbreviated: LED.

Limit: Control applied in the line or low voltage control circuit to break the circuit of conditions move outside a preset range. In a motor, a switch which cuts off power to the motor windings when the motor reaches its full open position.

Limit Control: A temperature, pressure, humidity, dew point or other control that overrides the demand control and/or duty cycler to prevent any affect on the business operation from load management, malfunction, or abnormal conditions. Also called "load override".

Limited Combustible Material: A building construction material not complying with the definition of noncombustible material, which, in the form in which it is used, has a potential heat value not exceeding 3500 Btu/lb (8141 kj/kg) (*see NFPA 259, Standard Test Method for Potential Heat of Building Materials*) and complies with one of the following paragraphs (a) or (b). Materials subject to increase in combustibility or flame spread rating beyond the limits herein established through the effects of age, moisture, or other atmospheric condition shall be considered combustible.

(a) Materials having a structural base of noncombustible material, with a surfacing not exceeding a thickness of ⅛ in. (3.2 mm) which has a flame spread rating not greater than 50.

(b) Materials, in the form and thickness used, other than as described in (a), having neither a flame spread rating greater than 25 nor evidence of continued progressive combustion, and of such composition that surfaces that would be exposed by cutting through the material on any plane would have neither a flame spread rating greater than 25 nor evidence of continued progressive combustion.

Line Slide: The side of a device electrically closest to the source of current.

Line Voltage: In the control industry, the normal electric supply voltages, which are usually 120 or 240 volts.

Liquefaction: A change of state to liquid; generally used instead of condensation in cases of substances ordinarily gaseous.

Liquid Sight Glass: The glass ported fitting in the liquid line used to indicate adequate refrigerant charge. When bubbles appear in the glass, there is insufficient refrigerant in the system.

Liquid Solenoid Valve: The electrically operated automatic shutoff valve in the liquid piping which closes on system shutdown to close off receiver discharge when used in pump down cycle and which prevents refrigerant migration in any system.

Load: The amount of heat per unit time imposed on a refrigerant system, or the required rate of heat removal.

Load Factor: This is a ratio expressing a customer's average actual use of the utility's capacity provided versus the maximum amount used.

Load Management: The control of electrical loads to reduce kW demand and kWh consumption.

Load Programmer: Any device which turns loads on and off on a real time, time interval, or kW demand basis.

Load Side: The side of a device electrically farthest from the current source.

Locked Rotor Current: See "Inrush Current".

Loudness: The subjective human definition of the intensity of a sound. Human reaction to sound is highly dependent on the sound pressure and frequency.

Loudness Level: A subjective method of rating loudness in which a 1000 Hz tone is varied in intensity until it is judged by listeners to be equally as loud as a given sound sample. The loudness level in "phons" is taken as the sound pressure level, in decibels, of the 1000 Hz tone.

Louver: An assembly of sloping vanes intended to permit air to pass through and to inhibit transfer of water droplets.

Low Limit Control: A device which normally monitors the condition of the controlled medium and interrupts system operation if the monitored condition drops below the desired minimum value.

Low Side: The refrigerating system from the expansion point to the point where the refrigerant vapor is compressed; where the system is at or below evaporated pressure.

Low Temperature Cutout: A pressure or temperature actuated device with sensing element in the evaporator, which will shut the system down at its control setting to prevent freezing chilled water or to prevent coil frosting. Direct expansion equipment may not use this device.

Low Voltage: In the control industry, a power supply of 25 volts or less.

M

MCM: Thousand circular mil—used to describe large wire sizes.

Magnahelic Gauge: An instrument used to measure (in inches of water gauge) differential air pressure between two spaces. (Trade name of Dwyer Instruments, Inc.)

Manometer: An instrument for measuring pressures: especially a U-tube partially filled with a liquid, usually water, mercury, or a light oil, so constructed that the amount of displacement of the liquid indicates the pressure being exerted on the instrument.

Mass: The quantity of matter in a body as measured by the ratio of the force required to produce given acceleration, to the acceleration.

Mass Law (Sound): The law relating to the transmission loss of sound barriers which says that in a part of the frequency range, the magnitude of the loss is controlled entirely by the mass per unit area of the barrier. The law also says that the transmission loss increases 6 decibels for each doubling of frequency or each doubling of the barrier mass per unit area.

Master (Central) Control: Control of all outlets from one point.

Maximum "No-Flow" Temperature: The maximum temperature that will be obtained in a component when the heat transfer fluid is not flowing through the system.

Media: The heat transfer material used in rotary heat exchangers, also referred to as *matrix*.

Melting Point: For a given pressure, the temperature at which the solid and liquid phases of the subsance are in equilbrium.

Microbar: A unit of pressure equal to 1 dyne/cm^2 (one millionth of the pressure of the atmosphere).

Micro-Organism: A microscopic organism, especially a bacterium, fungus, or a protozoan.

Micron: A unit of measurement equal to one-millionth of a meter or approximately 0.00003937 inch. (25 microns are approximately 0.001 inch).

Microprocessor: A small computer used in load management to analyze energy demand and consumption such that loads are turned on and off according to a predetermined program.

Mixed Airflow Cleanroom: A hybrid cleanroom consisting of a combination of laminar airflow and turbulent airflow within the same enclosure.

Modulation: Of a control, tending to adjust by increments and decrements.

Modulating Control: A mode of automatic control in which the action of the final control element is proportional to the deviation, from set point, of the controlled medium.

Modulating Controllers: Constantly reposition themselves in proportion to the requirements of the system, theoretically being able to maintain an accurately constant condition.

Motor Control Center: A single metal enclosed assembly containing a number of motor controllers and possibly other devices such as switches and control devices.

Multidirectional Airflow: Airflow in which the air within a confined area moves in a non-uniform or turbulent flow.

Multipole: Connects to more than 1 pole such as a 2-pole circuit breaker.

Multistage Thermostat: A thermostat which controls auxiliary equipment for heating or cooling in response to a greater demand for heating or cooling.

N

Natural Ventilation: The movement of outdoor air into a space through intentionally provided openings, such as windows and doors, or through nonpowered ventilators or by infiltration.

N.C.: *Normally closed* contacts of a relay. Contacts are close-circuited when the relay is de-energized.

N.O.: *Normally open* contacts of a relay. Contacts are open-circuited when relay is deenergized.

Neutral: The circuit conductor that is normally grounded or at zero voltage difference to the ground.

Nocturnal Radiation: Loss of energy by radiation to the night sky.

Noise: Sound which is unpleasant or unwanted by the recipient.

Noise Criteria Curves (NC Curves): Curves that define the limits which the octave band spectrum of a noise source must not exceed if a certain level of occupant acceptance is to be achieved.

Noise Criterion (NC) Curves: Established 1/1 octave band noise spectra for rating the the amount of noise of an occupied space with a single number.

Noncombustible Material: A material which, in the form in which it is used and under the conditions anticipated, will not ignite, burn, support combustion, or release flammable vapors when subjected to fire or heat. Materials reported as noncombustible, when tested in accordance with ASTM E136, *Standard Method of Test for Noncombustibility of Elementary Materials,* shall be considered noncombustible materials.

Normally open (or Normally closed): The position of a valve, damper, relay contacts, or switch when external power or pressure is *not* being applied to the device. Valves and dampers usually are returned to a "normal" position by a spring.

O

Occupied Zone: The region within an occupied space between planes 3 and 72 inches (75 and 1800 mm) above the floor and more than 2 feet (600 mm) from the walls or fixed air conditioning equipment (see ASHERAE Standard 55-1981).(1)

Octave Band (O.B.): A range of frequency where the highest frequency of the band is double the lowest frequency of the band. The band is usually specified by the center frequency.

1/1 Octave Band:: A range of frequencies where the highest frequency of the band is double the lowest frequency band. The band is specified by the center frequency. The preferred octave bands are designated by the following center frequencies: 31.5, 63, 125, 250, 500, 1000,2000, 4000, 8000, 16,000.

Odor: A quality of gases, liquids or particles that stimulates the olfactory organ.

Offset: Term used to describe the difference between the set point and the actual operating or control point.

Ohm (R) or (V): A measure of pure resistance in an electrical circuit.

Ohm's Law: The relationship between current and voltage in a circuit. It states that current is proportional to voltage and inversely proportional to resistance. Expressed algebraically, in DC circuits $I = E/R$; in AC circuits $I = E/Z$.

"On-off" Control: A two position action which allows operation at either maximum or minimum condition, or on or off, depending on the position of the controller.

On-Site Field Certification: Certification at the location of usage.

Opaque: Not permitting transmission of radiant energy.

Open Circuit: The condition when either deliberately or accidentally, an electrical conductor or connection is broken or open with a switch.

Operating Point: The value of the controlled condition at which the controller actually operates. Also called control point.

Operational Facility: A cleanroom in normal operation.

Optimum Operative Temperature: Temperature that satisfies the greatest possible number of people at a given clothing and activity level.

Out Contacts: Those relay contacts which complete circuits when the relay coil is deenergized. Also referred to as "normally closed contacts."

Outgassing: The emission of gases by materials and components, usually during exposure to elevated temperature, or reduced pressure.

Outlet, Ceiling: A round, square, rectangular, or linear air diffuser located in the ceiling, which provides a horizontal distribution pattern of primary and secondary air over the occupied zone and induces low velocity secondary air motion through the occupied zone.

Outlet, Slotted: A long, narrow air distribution outlet, comprised of deflecting members, located in the ceiling sidewall, or sill, with an aspect ratio greater than 10, designed to distribute supply air in varying directions and planes, and arranged to promote mixing of primary air and secondary room air.

Outlet, Vaned: A register or grille equipped with vertical and/or horizontal adjustable vanes.

Outlet Velocity: The average velocity of air emerging from an opening, fan or outlet, measured in the plane of the opening.

Output: Capacity, duty, performance, net refrigeration produced by system.

Outside Air Opening: Any opening used as an entry for air from outdoors.

Overall Coefficient of Heat Transfer (thermal transmittance): The time rate of heat flow through a body per unit area, under steady conditions, for a unit temperature difference between the fluids on the two sides of the body.

Overcurrent Device: A device such as a fuse or a circuit breaker designed to protect a circuit against excessive current by opening the circuit.

Overload: A condition of excess current; more current flowing than the circuit was designed to carry.

Override: A manual or automatic action taken to bypass the normal operation of a device or system.

Oxidation: A reaction in which oxygen combines with another substance.

P

Package A/C unit: Consists of a factory-made assembly which normally includes an indoor conditioning coil, compressor(s), condensing coil, and may include a heating function as well.

Parallel Airflow: Unidirectional airflow, as demonstrated by introduction of an isokinetic smoke stream which exhibits a measured dispersion of not more than 14° from straight line flow.

Parallel Circuit: One where all the elements are connected across the voltage source. Therefore, the voltage on each element is the same but the current through each may be different.

Particulate Matter: A state of matter in which solid or liquid substances exist in the form of aggregated molecules or particles. Airborne particulate matter is typically in the size range of 0.01 to 100 micrometers.

Particle: A very small discrete mass of solid or liquid matter, usually measured in microns.

Particle Count: Concentration expressed in terms of the number of particles per unit volume of air or other gas.

Particle Counter: A light-scattering instrument with display or recording means to count and size discrete particles in air.

Particle Size: An expression for the size of solid or liquid particles expressed as the apparent maximum linear dimension or diameter of the particle.

Particulate Matter: A general term applied to miniature particles of material suspended in gases or liquids.

Passive Solar System: An assembly of collectors, thermal storage device(s), and transfer media which converts solar energy into thermal energy and in which no energy in addition to solar is used to accomplish the transfer of thermal energy. The prime element in a passive solar system is usually some form of thermal capacitance.

Pass-Through-Box: A double-doored chamber arranged to permit transfer of material and/or equipment between two confined spaces of different contamination levels.

Peak Demand: The greatest amount of kilowatts needed during a demand interval.

Peak Load Pricing: A pricing principle that charges more for purchases that contribute to the peak demand and, thereby, cause the expansion of productive capacity when the peak demand exceeds the peak capacity (less minimum excess capacity). In the electric power industry, this means charging more for electricity bought on or near the seasonal peak of the utility or on or near the daily peak of the utility. The latter requires special meters; the former does not.

Performance Factor: Ratio of the useful output capacity of a system to the input required to obtain it. Units of capacity and input need not be consistent.

Pert: Program, evaluation and review technique; a system of planning, scheduling, controlling and reviewing a series of interdependent events in order to follow a proper sequence and complete a project as quickly and inexpensively as possible.

Pervious: Air porous.

Phase: Part of an AC voltage cycle. Residential electrical service is 2-phase; commercial facilities are usually 3-phase AC voltage.

Phon: A measurement of loudness level. The loudness level in *phons* of any sound is the sound pressure level of the 1000-Hz reference tone which is equally loud to the sound being rated. The loudness of 1 sone corresponds to a loudness level of 40 phons in accordance with the definition of the sone; a two fold change of loudness in sones is associated with a 10-phon change in loudness level.

Photovoltaic Conversion: Use of semiconductor or other photovoltaic devices that convert solar radiation directly to electricity.

Pilot Duty Relay: A relay used for switching loads such as another relay or solenoid valve coils. The pilot duty relay contacts are located in a second control circuit. Pilot duty relays are rated in volt-amperes (VA).

Pitch: The pitch of a sound depends primarily on its frequency. In music, sounds of higher frequencies are referred to as treble notes, while those of lower frequencies are referred to as bass notes.

Planting Screen: Bushes or other planting that hides a refrigerant compressor.

Plenum: An air compartment connected to one or more distributing ducts.

Plug-in Bus Duct: Bus duct with built-in power tap-off points. Tap-off is made with a plug-in switch, circuit breaker, or other fitting.

Pneumatic: Operated by air pressure.

Pneumatic-Electric (PE) Switches: Device that operates an electric switch from a change of air pressure.

Point, Critical: Of a substance, state point at which liquid and vapor have identical properties; critical temperature, critical pressure, and critical volume are the terms given to the temperature, pressure, and volume at the critical point. Above the critical temperature or pressure, there is no demarcation line between liquid and gaseous phases.

Point of Duty: A statement of air volume flow rate and static or total pressure at a stated density and is used to specify the point on the system curve at which a fan is to operate.

Point of Operation: Used to designate the single set fan performance values which correspond to the point of intersection of the system curve and the fan pressure-volume curve.

Point of Rating: A statement of fan performance values which correspond to one specific point on the fan pressure-volume curve.

Polarity: The direction of current flow in a DC circuit. By convention, current flows from plus to minus. Electron flow is actually in the opposite direction.

Pole: An electrical connection point. In a panel, the point of connection. On a device, the terminal that connects to the power.

Potable Water: Water that is safe to drink.

Potential Transformer: A voltage transformer. The voltage supplied to a primary coil induces a voltage in a secondary coil according to the ratio of the wire windings in each of the coils.

Potentiometer: An electromechanical device having a terminal connected to each and to the resistive element, and a third terminal connected to the wiper contact. The electrical input is divided as the contact moves over the element, thus making it possible to mechanically change the resistance.

Power (P): Expressed in *watts* (W) or *kilowatts* (kW), and is equal to:

in DC circuit, $P = EI$ and $P = I^2R$
in AC circuit, $P = EI \times$ *Power factor*

Power Factor (pf): A quantity that relates the volt-amperes of an AC circuit to the wattage (power = volt-amperes × power factor). Power factor also is the ratio of the circuit resistance (R) to the impedance (Z) expressed as a decimal between zero and one (p.f. = R/Z). When the power factor equals one, all consumed power produces useful work.

Power Factor Charge: A utility charge for "poor" power factor. It is more expensive to provide power to a facility with a poor power factor (usually less than 0.8).

Power Factor Correction: Installing capacitors on the utility service's supply line to improve the power factor of the building.

Power Supply: The voltage and current source for an electrical circuit. A battery, a utility service, and a transformer are power supplies.

Predicting Method: Method for determining when load shedding should occur. A formula is used to arrive at a preset kilowatt limit. Then the actual amount of energy accumulated during the utility's demand intervals is measured. A projection is made of the actual rate of energy usage during the rest of the interval. If the predicted value exceeds the preset limit, loads will be shed.

Preferred Noise Criterion (PNC) Curves: The PNC curves are a proposed modification of the older NC curves. These PNC curves have values that are about 1 dB lower than the NC curves in the four octave bands at 125, 250, 500, and 1000 Hz for the same curve rating numbers. In the 63 Hz band, the permissible levels are 4 or 5 dB lower; in the highest three bands, they are 4 or 5 dB lower.

Prefilter: A filter, usually of lower performance grade than the final filter, that precedes the final filter.

Preheating: In air conditioning, to heat the air ahead of other processes.

Pressure: The normal force exerted by a homogeneous liquid or gas, per unit of area, on the wall of its container.

Pressure, Absolute: Pressure referred to that of a perfect vacuum. It is the sum of gauge pressure and atmospheric pressure.

Pressure, Atmospheric: It is the pressure indicated by a barometer. Standard atmosphere is the pressure equivalent to 14.696 psi or 29.921 in. of mercury at 32°F.

Pressure, Critical: Vapor pressure corresponding to the substance's critical state at which the liquid and vapor have identical properties.

Pressure, Gauge: Pressure above atmospheric.

Pressure, Hydrostatic: The normal force per unit area that would be exerted by a moving fluid on an infinitesimally small body immersed in it if the body were carried along with the fluid.

Pressure, Partial: Portion of total gas pressure of a mixture attributable to one component.

Pressure, Saturation: The saturation pressure for a pure substance for any given temperature is that pressure at which vapor and liquid, or vapor and solid, can coexist in stable equilibrium.

Pressure, Static (SP): The normal force per unit area that would be exerted by a moving fluid on a small body immersed in it if the body were carried along with the fluid. Practically, it is the normal force per unit are at a small hole in a wall of the duct through which the fluid flows (piezometer) or on the surface of a stationary tube at a point where the disturbances, created by inserting the tube, cancel. It is supposed that the thermodynamic properties of a moving fluid depend on static pressure in exactly the same manner as those of the same fluid at rest depend upon its uniform hydrostatic pressure.

Pressure, Total (TP): In the theory of the flow of fluids, the sum of the static pressure and the velocity pressure at the point of measurement. Also called dynamic pressure.

Pressure, Vapor: The partial pressure exerted by the water vapor contained in air.

Pressure, Velocity (V_p): In moving fluid, the pressure capable of causing an equivalent velocity, if applied to move the same fluid through an orifice such that all pressure energy expended is converted into kinetic energy.

Pressure Drop: Pressure loss in fluid pressure, as from one end of a duct to the other, due to friction, dynamic losses, and changes in velocity pressure.

Pressure Regulator: Automatic valve between the evaporator outlet and compressor inlet that is responsive to pressure or temperature; it functions to throttle the vapor flow when necessary to prevent the evaporator pressure from falling below a preset level.

Primary Air: The initial airstream to an air outlet or terminal device being supplied by a fan or supply duct prior to any entrainment of ambient air.

Primary Control: A device which directly or indirectly controls the control agent in response to needs indicated by the controller. Typically a motor, valve, relay, etc.

Primary Element: The portion of the controller which first uses energy derived from the controlled medium to produce a condition representing the value of the controlled variable; for example, a thermostat bimetal.

Primary Service: High voltage service, above 600 volts.

"Process" Hot Water: Hot water needed for manufacturing processes over and above the "domestic hot water" that is for the personal use of industrial workers.

Properties, Thermodynamic: Basic qualities used in defining the condition of a substance, such as temperature, pressure, volume, enthalpy, entropy.

Proportional Band: The range of values of a proportional positioning controller through which the controlled variable must pass to move the final control element through its full operating range. Commonly used equivalents are "throttling range" and "modulating range."

Proportional Control: See Modulating Control.

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere.

Psychrometric Chart: A graphical representation of the thermodynamic properties of moist air.

Pull Box: A metal cabinet inserted into a conduit run for the purpose of providing a cable pulling point. Cable may be spliced in these boxes.

Pulsing Demand Meter: A meter which generates a pulse in correspondence with each revolution of a kWh meter. Pulses are recorded on paper or magnetic tape. Pulse can also be the signal to demand control equipment.

Pure Tone: A pure tone sound that has a unique pitch and is characterized by a sinusoidal variation in sound pressure with time. The frequency spectrum of a pure tone shows a single line at a discrete frequency.

Pyranometer: A measurement device to determine local values of total (direct and diffuse) isolation.

Pyrometer: An instrument for measuring high temperatures.

Pyrheliometer: A measurement device to determine local values of direct insolation.

R

Raceway: Any support system, open or closed, for carrying electric wires.

Radiation (Acoustic): The process of turning structure-borne noise into airborne (or some other fluid-borne) noise.

Radiation, Thermal: The transmission of heat through space by wave motion; the passage of heat from one object to another without warming the space between.

Radius of Diffusion: The horizontal axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level; e.g., 200, 150 or 100 fpm.

Random Incidence: If an object is in a diffuse sound field, the sound is said to strike the object at random incidence.

Real Time: Time measured according to the time of day (1 PM, 2 PM, etc.). Different from the "electronic time" of a cycling device.

Receiver: An auxiliary storage receptacle for refrigerant when the system is pumped down and shut down.

Reflected Isolation: The portion of the total solar energy reaching a surface (window, wall, collector) which has been reflected by an adjoining surface.

Reflectivity: The property of a material that determines its ability to reflect radiant energy.

Refrigerant: The fluid used for heat transfer in a refrigerating system, which absorbs heat at a low temperature and a low pressure of the fluid and rejects heat at a higher temperature and a higher pressure of the fluid, usually involving changes of state of the fluid.

Regenerated Noise: Duct noise, which is generated by air turbulence in the duct or fittings.

Register: A grille equipped with an integral damper or control valve.

Relative Humidity (RH): The ratio of water vapor in the air as compared to the maximum amount of water vapor that may be contained.

Relay: An electromechanical switch that opens or closes contacts in response to some controlled action. Relay contacts *can* be normally open (N.O.) and or normally closed (N.C.). Relays may be electric, pneumatic, or a combination of both. PE and EP switches are relays.
Relay, Thermal: A switching relay in which a small heater warms a bimetal element which bends to provide the switching force.
Remote Temperature Set Point: Ability to set a temperature control point for a space from outside the space. Often used in public areas.
Reset: A process of automatically adjusting the control point of a given controller to compensate for changes in outdoor temperature. The hot deck control point is normally reset upward as the outdoor temperature drops. The cold deck control point is normally reset downward as the outdoor temperature increases.
Reset Controllers: Two controllers operating together; one sensing a condition other than that of the controlled space and changing the set point of the second controller, which is directly responsible for the result in the controlled space. The resetting controller is commonly called the master, and the controller being reset is commonly called the submaster (slave).
Reset Ratio: The ratio of change in outdoor temperature to the change in control point temperature. For example, a 2:1 reset ratio means that the control point will increase 1 degree for every 2 degrees change in outdoor temperature.
Resistance (Ω): The opposition which limits the amount of current that can be produced by an applied voltage in an electrical circuit, measured in ohms.
Resistance, Thermal: The reciprocal of thermal conductance.
Resistive Loads: Electrical loads whose power factor is one. Usually contain heating elements.
Resistivity, Thermal: The reciprocal of thermal conductivity.
Respirable Particles: Respirables particles are those that penetrate into and are deposited in the non-ciliated portion of the lung. Particles greater than 10 micrometers aerodynamic diameter are not respirable.
Return Air: Air returned from conditioned or refrigerated space.
Reverberant Sound Field: A space in which sound persists because of continuous reflections. A reverberant field is not necessarily diffuse.
Reverberation: The persistence of sound in an enclosed space after the sound source has stopped. In a reverberation room, it is characterized by the decay or dying away of the sound.
Reverberation Room: A highly sound reflective room in which special care has been taken to make the sound field as diffuse as possible.
Reverberation Time: The reverberation time of an enclosed space is the number of seconds required, or that would be required were the decay rate to remain constant, for the sound pressure level to decrease by 60 decibels.
Reverse Flow: Occurs when noise propagates and air flows in opposing direction, as in a typical return-air system.
Riser Diagram: Electrical block-type diagram showing connection of major items of equipment. It is also applied to signal equipment connections. Also generally applied to multistory buildings for vertical hydronic piping and ductwork.
Riser Shaft: A vertical shaft designed to house electric cables, piping and ductwork.
Room Absorption: The product of average absorption coefficients inside a room and the total surface area. This is usually expressed in sabins.
Room Criterion (RC) Curves: The RC similar to NC or PNC curves. However, they have a slightly different shape to approximate a well balanced, bland-sounding spectrum whenever the space requirements dictate that a certain amount of background noise be maintained for masking or other purposes.
Room Dry Bulb: The actual temperature of the conditioned room or space as measured with an accurate thermometer.
Room Effect:: The difference between the sound power level discharged by a duct (through a diffuser or other termination device) and the sound pressure level heard by an occupant of a room is called the Room Effect. The Magnitude of the Room Effect depends upon the amount of the sound absorption in the room (Sabins), the distance between the termination duct and the nearest observer and the directivity factor of the source.
Room Velocity: The residual air velocity level in the occupied zone of the conditioned space (e.g., 65, 50, 35 fpm).
Running Current: The current that flows through a load after inrush current. Usually called "full load current."

S

Sabin: The unit of acoustic absorption. One sabin is one square foot of perfect sound-absorbing material.
Saturation, Degree of: The ratio of the weight of water vapor associated with a pound of dry air to the weight of water vapor associated with a pound of dry air saturated at the same temperature.
Seasonal Energy Efficiency Ratio (SEER): The total quantity of heat delivered or removed by heating or cooling equipment in Btu divided by the total electrical energy input in kilowatt hours over an entire season.
Seasonal Performance Factor: The ratio of the total quantity of heat delivered by a heat pump, (including supplemental resistance heaters) to the total quantity of energy input (including supplementary resistance heaters) for the total annual heating hours below 65 F.
Secondary Air: The air surrounding an outlet that is captured or entrained by the initial outlet discharge airstream (furnished by a supply duct or fan).
Secondary Service: Voltage service up to 600 volts.
Seismic: Subject to or caused by an earthquake.
Semi-Extended Plenum: A trunk duct that is extended as a plenum from a fan or HVAC unit to serve multiple outlets and or branch ducts.
Sensible Heat: Sensible heat is any heat transfer that causes a change in temperature. Heating and cooling of air and water that may be measured with a thermometer is sensible heat. Heating or cooling coils that simply increase or decrease the air temperature without a change in moisture content are examples of sensible heat.
Sensible Heat Factor: The ratio of sensible heat to total heat.
Sensible Heat Ratio, Air Cooler: The ratio of sensible cooling effect to total cooling effect of an air cooler.
Sensing Device: A device that keeps track of the measured condition and its fluctuations so that when sufficient variation occurs it will originate the signal to revise the operation of the system and offset the change. Example: a thermostat "bulb." A sensing device may be an integral part of a controller.
Sensing Element: The first system element or group of elements. The sensing element performs the initial measurement operation.
Sensitivity: The ability of a control instrument to measure and act upon variations of the measured condition.
Sensor: A sensing element.
Sequencer: A mechanical or electrical device that may be set to initiate a series of events and to make the events follow in sequence.
Sequencing Control: A control which energizes successive stages of heating or cooling equipment as its sensor detects the need for increased heating or cooling capacity. May be electronic or electromechanical.
Series Circuit: One with all the elements connected end to end. The current is the same throughout but the voltage can be different across each element.
Service Drop: The overhead service wires that serve a building.
Service Switch: One to six disconnect switches or circuit breakers. Purpose is to completely disconnect the building from the electric service.

Set Point: The value of the controlled condition at which the instrument is set to operate. The set point in the example in "differential" might be 69½°, the mid point of the differential.
Shading Loss: The loss of collector efficiency caused by the shading of the absorber plate by collector edges or components. The shading loss usually varies with the angle of incidence of the isolation.
Shall or Will: Where shall or will is used as a provision specified, that provision is mandatory if compliance with the standard is claimed.
Shed: To de-energize a load in order to maintain a kW demand set point.
Shed Mode: A method of demand control that reduces kW demand through shedding and restoring loads.
Shielded Cable: Special cable used with equipment that generates a low voltage output. Used to minimize the effects of frequency "noise" on the output signal.
Short Circuit: An electric circuit with zero load; an electrical fault.
Short Cycling: Unit runs and then stops at short intervals; generally this excessive cycling rate is hard on the system equipment.
Should or It Is Recommended: Term used to indicate provisions which are not mandatory but which are desirable as good practice.
Single Particle Counter: An instrument for continuous counting of individual airborne particles larger than a given threshold size(s). The sensing means may be optical, electrical, aerodynamic, etc.
Single-phasing: The condition when one phase of a multiphase (polyphase) motor circuit is broken or opened. Motors running when this occurs may continue to run but with lower power output and overheating.
Smoke: The airborne solid and liquid particles and gases that evolve when a material undergoes pyrolysis or combustion. Note: chemical smoke is excluded from this definition.
Smoke Barrier: A continuous membrane, either vertical or horizontal, such as a wall, floor, or ceiling assembly, that is designed and constructed to restrict the movement of smoke.
Smoke Control System: A system that utilizes fans to produce pressure differences to manage smoke movement.
Smoke Damper: A device to resist the passage of smoke which:

(a) Is arranged to operate automatically, and/or
(b) Is controlled by a smoke detector, and/or
(c) May be capable of being positioned manually from a remote command station.

A smoke damper may be a fire damper or a damper serving other functions, if its location lends itself to the multiple functions. A combination fire and smoke damper shall meet the requirements of both.
Smoke Detector: A device which senses visible or invisible particles of combustion.
Smoke Developed Rating: A smoke developed rating of a material refers to a number or classification of a material obtained according to NFPA 255, *Method of Test of Surface Burning Characteristics of Building Materials,* which measures visible smoke.
Solar Altitude: The angular elevation of the sun above the horizon.
Solar Azimuth: Angle between true south and projection of earth-sun line on a horizontal plane.
Solar Collector: Any device which collects solar energy and transforms it to another usable form of energy.
Solar Energy: The photon energy originating from the sun's radiation in the wavelength region from 0.3 to 2.4 micrometers; the radiant energy of the sun, whether it be direct, diffuse or reflected radiation.
Solar Time: The time of day based on the relative position of the sun with respect to a position on the earth's surface. Solar noon is that instant on any day at which the sun reaches its maximum altitude for that day.
Solenoid Air Valves: EP switches with an electromagnetic coil in the valve topworks that opens or closes the valve from normal position. A spring returns the valve to the normal position when the coil is de-energized.
Sone: One sone is defined as the loudness of a 1000 Hz tone having a sound pressure of 40 dB. Two sones is twice as loud as the 40 dB reference sound of one sone, etc.
Sorbent: See absorbent.
Sound Absorption: (1) The process of dissipating or removing sound energy. (2) The property possessed by materials, objects, and structures, such as rooms, of absorbing sound energy. (3) The measure of the magnitude of the absorptive property of a material, an object, or a structure, such as a room.
Sound Attenuator: A device or equipment that prevents, reduces, or absorbs sound.
Sound Power Level (L_w): the fundamental characteristic of an acoustic source (fan, etc.) is its ability to radiate power. Sound power level cannot be measured directly; it must be calculated from sound pressure level measurements. The sound power level of a source (L_w) is the ratio, expressed in decibels, of its sound power divided by the reference sound power which is 10^{-12} watts.

A considerable amount of confusion exists in the relative use of sound power level and sound pressure level. An analogy may be made in that the measurement of sound pressure level is comparable to the measurement of temperature in a room; whereas, the sound power level is comparable to the cooling capacity of the equipment conditioning the room. The resulting temperature is a function of the cooling capacity of the equipment and the heat gains and losses of the room. In exactly the same way, the resulting sound pressure level would be a function of the sound power output of the equipment together with the sound reflective and sound absorptive properties of the room.

Given the total sound power output of sound source and knowing the acoustical properties and dimensions of a room, it is possible to calculate the resulting sound pressure levels.
Sound Power Level of a Source(L_w): The ratio, expressed in decibels, of its sound power to the reference sound power which, by divided agreement, is either 10^{-13} or 10^{-12} watts. The reference power should always be stated.
Sound Power of a Source (W): The rate at which sound energy is radiated by the source. Without qualification, overall sound power is meant but often sound power in a specific frequency band is indicated.
Sound Pressure: Sound pressure is an alternating pressure superimposed on the barometric pressure by sound. It can be measured or expressed in several ways such as maximum sound pressure or instantaneous sound pressure. Unless such a qualifying word is used, it is the effective of root-mean-square pressure which is meant.
Sound Pressure Level (L_p): A measure of the air pressure change caused by a sound wave expressed on a decibel scale reference to a reference sound pressure of 2×10^{-5} Pa or 0.0002 microbar.
Sound Transmission Class: Sound transmission class is the preferred single figure rating designed to give a preliminary estimate of the sound insulating properties of a barrier.
Sound Transmission Coefficient: The sound transmission coefficient of a partition is the fraction of the incident sound power transmitted through it when the sound fields on both sides of the partition are diffuse.
Sound Transmission Loss of a Partition (TL): The ratio, expressed in decibels, of the incident sound power on the source side of the specimen to the transmitted sound power on the receiving side when the sound fields on both sides of the specimen are diffuse.

When the sound fields are not diffuse, a qualifying word is necessary, such as normal incidence sound transmission loss, or field transmission loss.
Specific Heat: Specific heat (C_p) is the amount of heat energy in Btu's required to raise the temperature of one pound of substance one degree Fahrenheit. The following are specific heat values at standard conditions:

water—C_p = 1.00 Btu/lb/°F
air—C_p = 0.24 Btu/lb/°F

Using these values in simple equations, gallons per minute or cubic feet per minute may be determined in a system if the Btu per hour and the temperature difference are known.

Specification Design: A concise document defining technical requirements in sufficient detail to form the basis for a product or process. It indicates, when appropriate, the procedure that determines whether or not the given requirements are satisfied.

Specification Performance: A concise document which details the performance requirement for a product. The performance specification includes procedures and/or references for testing and certification of the product.

Specific Volume: The reciprocal of density and is used to determine the cubic feet of volume, if the pounds of weight are known. Both density and specific volume are affected by temperature and pressure. The specific volume of air under standard conditions is 13.33 cubic feet per pound and the specific volume of water at standard conditions is 0.016 cubic feet per pound.

Spread: The divergence of the airstream in a horizontal or vertical plane after it leaves the outlet.

Stage Differential: Change in temperature at the thermostat needed to turn heating or cooling equipment off once it is turned on.

Staging Interval: The minimum time period for shedding or restoring two sequential loads.

Stagnant Air Area: An area within a space where the air velocity is less than 25 fpm.

Standard Air Density (d): Standard air density has been set at 0.075 lb/cu. ft. This corresponds approximately to dry air at 70°F and 29.92 in. Hg. In metric units, the standard air density is 1.2041 kg/m^3 at 20°C and at 101.325 kPa.

Standard Conditions: The standard conditions referred to in environmental system work for air are: dry air at 70°F, and at an atmospheric pressure of 29.92 inches mercury (in.Hg.). For water, standard conditions are 68°F at the same barometric pressure. At these standard conditions, the density of air is 0.075 pounds per cubic feet and the density of water is 62.4 pounds per cubic foot.

Standard Rating: A ring based on tests performed at Standard Rating Conditions

Starter: Basic contactor with motor overloads, etc., added—a motor starter is an adaptation of the basic contactor which includes overload relays. Starters for large motors may include reactors, step resistors, disconnects, or other features required in a more sophisticated starter package.

State: Refers to the form of a fluid, either liquid, gas or solid. Liquids used in environmental systems are water, thermal fluids such as ethylene glycol solutions, 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 three states. A simple example is water, which may be solid (ice), liquid (water), or gas (steam or water vapor).

Static Head: The pressure due to the weight of a fluid above the point of measurement.

Static Regain Method: A method of duct sizing wherein the duct velocities are systmatically reduced, allowing a portion of the velocity pressure to convert to static pressure offsetting the duct friction losses.

Static Suction Head: The positive vertical height in feet from the pump centerline to the top of the level of the liquid source.

Static Suction Lift: The distance in feet between the pump centerline and the source of liquid below the pump centerline.

Step Controller: See Sequencer.

Stratified Air: Unmixed air that is in thermal layers that have temperature variations of approximately five degrees or more.

Structure-Borne Noise: A condition when the sound waves are being carried by a portion of the building structure. Sound waves in this state are inaudible to the human ear since they cannot carry energy to it. Airborne sound can be created from the radiation of the structure-borne sound into the air.

Subcooling: The difference between the temperature of a pure condensable fluid below saturation and the temperature at the liquid saturated state, at the same pressure.

Subcooling Specific: The difference between specific enthalpies of a pure condensable fluid between liquid at a given temperature below saturation and liquid at saturation, at the same pressure.

Sublimation: A change of state directly from solid to gas without appearance of liquid.

Suction Head: The positive pressure on the pump inlet when the source of liquid supply is above the pump centerline.

Suction Lift: The combination of static suction lift and friction head in the suction piping when the source of liquid is below the pump centerline.

Suction Piping: The Piping which returns gaseous refrigerant to the compressor. Sizes must be large enough to maintain minimum friction to prevent reduced compressor and system capacity but must be small enough to produce adequate velocity to return oil to the compressor.

Sun Effect: Solar energy transmitted into space through windows and building materials.

Sun Rights (Solar Rights): A legal question concerning the rights to non-interrupted use of the sun's radiant energy, such as shading of solar collectors by a neighbor's building.

Superheat: The difference between the temperature of a pure condensable fluid above saturation and the temperature at the dry saturated state, at the same pressure.

Superheat, Specific: The difference between specific enthalpies of a pure condensable fluid between vapor at a given temperature above saturation and vapor at dry saturation, at the same pressure.

Surface, Heating: The exterior surface of a heating unit. *Extended heating surface* (or *extended surface*), consisting of fins, pins, or ribs which receive heat by conduction from the prime surface. *Prime surface:* heating surface having the heating medium on one side and air (or extended surface) on the other.

Surge Suppressor: A device that reduces harmonic distortion in line voltage circuits by clipping off transient voltages which are fed through the power lines from operating equipment.

Switching Relays: Relays are devices which operate by a variation in the conditions of one electrical circuit to affect the operation of devices in the same or another circuit. General purpose switching relays are used to increase switching capability and isolate electrical circuits, such as in systems where the heating and cooling equipment have separate power supplies, and provide electrical interlocks within the system.

System: A series of ducts, conduits, elbows, branch piping, etc. designed to guide the flow of air, gas or vapor to and from one or more locations. A fan provides the necessary energy to overcome the resistance to flow of the system and causes air or gas flow through the system. Some components of a typical system are louvers, grilles, diffusers, filters, heating and cooling coils, air pollution control devices, burner assemblies, volume flow control dampers, mixing boxes, sound attenuators, the ductwork and related fittings.

System, Central Fan: A mechanical, indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and conveyed to and from the rooms by means of a fan and a system of distributing ducts.

System, Closed: A heating or refrigerating piping system in which circulating water or brine is completely enclosed, under pressure above atmospheric, and shut off from the atmosphere except for an expansion tank.

System, Duct: A series of ducts, conduits, elbows, branch piping, etc. designed to guide the flow of air, gas or vapor to and from one or more locations. A fan provides the necessary energy to overcome the resistance to flow of the system and causes air or gas to flow through the system. Some components of a typical system are louvers, grilles, diffusers, filters, heating and cooling coils, energy recovery devices, burner assemblies, volume dampers, mixing boxes, sound attenuators, the ductwork and related fittings.

System, Flooded: A system in which only part of the refrigerant passing over the heat transfer surface is evaporated, and the portion not evaporated is separated from the vapor and recirculated.
System, Gravity Circulation: A heating or refrigerating system in which the heating or cooling fluid circulation is effected by the motive head due to difference in densities of cooler and warmer fluids in the two sides of the system.
System, Run-around: A regenerative-type, closed, secondary system in which continuously circulated fluid abstracts heat from the primary system fluid at one place, returning this heat to the primary system fluid at another place.
System, Unitary: A complete, factory-assembled and factory-tested refrigerating system comprising one or more assemblies which may be shipped as one unit or separately but which are designed to be used together.
System Curve: A graphic presentation of the pressure vs. volume flow rate characteristics of a particular system.
System Effect Factor: A pressure loss factor which recognizes the effect of fan inlet restrictions, fan outlet restrictions, or other conditions influencing fan performance when installed in the system.

T

Temperature, Absolute Zero: The zero point on the absolute temperature scale, 459.69 degrees below the zero of the Fahrenheit scale, 273.16 degrees below the zero of the Celsius scale.
Temperature, Critical: The saturation temperature corresponding to the critical state of the substance at which the properties of the liquid and vapor are identical.
Temperature, Dewpoint: The temperature at which the condensation of water vapor in a space begins for a given state of humidity and pressure as the temperature of the vapor is reduced. The temperature corresponding to saturation (100 percent relative humidity) for a given absolute humidity at constant pressure.
Temperature, Drybulb: The temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.
Temperature, Effective: An arbitrary index which combines into a single value the effect of temperature, humidity, and air movement on the sensation of warmth or cold felt by the human body. The numerical value is that of the temperature of still, saturated air which would induce an identical sensation.
Temperature, Mean Radiant (MRT): The temperature of a uniform black enclosure in which a solid body or occupant would exchange the same amount of radiant heat as in the existing nonuniform environment.
Temperature, Saturation: The temperature at which no further moisture can be added to the air-water vapor mixture. Equals dew point temperature.
Temperature, Wet-bulb: Thermodynamic wet bulb temperature is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. Wet bulb temperature (without qualification) is the temperature indicated by a wet bulb psychorometer constructed and used according to specifications.
Temperature, Wet Bulb Depression: Difference between dry bulb and wet bulb temperatures.
Temperature Difference, Mean: Mean of difference between temperatures of a fluid receiving and a fluid yielding heat.
Terminal Velocity: The maximum airstream velocity at the end of the throw.
Therm: Measurement used by gas utilities for billing purposes. 1 Therm = 100 cubic feet of gas = 100,000 Btu.
Thermal Capacity: The ability of a medium to hold heat.
Thermal Expansion Valve: The metering device or flow control which regulates the amount of liquid refrigerant which is allowed to enter the evaporator.
Thermistor: Semiconductor material that responds to temperature changes by changing its resistance.
Thermocouple: Device for measuring temperature utilzing the fact that an electromotive force is generated whenever two junctions of two dissimilar metals in an electric circuit are at different temperature levels.
Thermodynamics, Laws of: Two laws upon which rest the classical theory of thermodynamics. These laws have been stated in many different, but equivalent ways.

> *The First Law:* (1) When work is expanded in generating heat, the quantity of heat produced is proportional to the work expended; and, conversely, when heat is employed in the performance of work, the quantity of heat which disappears is proportional to the work done (Joule); (2) If a system is caused to change from an initial state to a final state by adiabatic means only, the work done is the same for all adiabatic paths connecting the two states (Zemansky); (3) In any power cycle or refrigeration cycle, the net heat absorbed by the working substance is exactly equal to the net work done.
> *The Second Law:* (1) It is impossible for a self-acting machine, unaided by any external agency, to convey heat from a body of lower temperature to one of higher temperature (Clausius); (2) It is impossible to derive mechanical work from heat taken from a body unless there is available a body of lower temperature into which the residue not so used may be discharged (Kelvin); (3) It is impossible to construct an engine that, operating in a cycle, will produce no effect other than the extraction of heat from a reservoir and the performance of an equivalent amount of work (Zemansky).

Thermodynamic Wet Bulb Temperature: The temperature at which water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature. The wet bulb temperature measured with an appropriate psychrometer can approach the thermodynamic wet bulb temperature (also called the Adiabatic Saturation Temperature).
Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls temperature.
Thermosyphon: Circulation of a fluid by making use of the change in density of a material when it is heated and cooled. Also called Natural Circulation.
Throttling Range: The amount of change in the variable being controlled to make the controlled device more through the full length of its stroke.
Throw: The horizontal or vertical axial distance an airstream travels after leaving an air outlet before the maximum stream velocity is reduced to a specified terminal level; e.g., 200, 150, 100 or 50 fpm.
Tic Mark: Hatch mark on drawing raceway symbol, showing number of wires.
Ton (of Refrigeration): A useful refrigerating effect equal to 12,000 Btuh; 200 Btu/min.
Total Dynamic Head: Dynamic discharge head (static discharge head, plus friction head, plus velocity head) plus dynamic suction lift, or dynamic discharge head minus dynamic suction head.
Total Heat (Enthalpy): Total heat is the sum of the sensible heat and latent heat in an exchange process. In many cases, the addition or subtraction of latent and sensible heat at terminal coils appears simultaneously. Total heat also is called *enthalpy*, both of which can be defined as the quantity of heat energy contained in that substance.
Total Pressure Method: A method of duct sizing which allows the designer to determine all friction and dynamic losses in each section of a duct system (including the total system).
Total Suspended Particulate Matter: The mass of particles suspended in a unit of volume of air when collected by a high volume air sampler.

Toxic Fluids: Gases or liquids which are poisonous, irritating and/or suffocting.
Tracking Collector: A solar energy collector that constantly positions itself perpendicular to the sun as the earth rotates.
Transducer: The means by which the controller converts the signal from the sensing device into the means necessary to have the appropriate effect on the controlled device. For example, a change in air pressure in the pneumatic transmission piping.
Transformer: The system power supply—a transformer is an inductive stationary device which transfers electrical energy from one circuit to another. The transformer has two windings, primary and secondary. A changing voltage applied to one of these, usually the primary, induces a current to flow in the other winding. A coupling transformer transfers energy at the same voltage; a step-down transformer transfers energy at a lower voltage, and a step-up transformer transfers energy at a higher voltage.
Transmission: The means by which a signal is moved from one point of origin to the point of action.
Transmission, coefficient of Heat: Any one of a number of coefficients used in the calculation of heat transmission by conduction, convection, and radiation, through various materials and structures.
Transmittance, Thermal (U factor): The time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids.
Turbulent Airflow Cleanroom: A cleanroom in which the filtered air enters the room in a non-uniform velocity or turbulent flow. Such rooms exhibit non-uniform or random airflow patterns throughout the enclosure.

U

Utility Service: The utility company. Also, the amount and configuration of voltage supplied by a utility company. There are four main types of commercial utility sevices: 208V AC wye, 480V AC wye, 240V AC delta, and 480V AC delta.
Utility Transformer: Primary and secondary coils of wire which reduce (step down) the utility supply voltage for use within a facility.
Uniform Airflow: Unidirectional airflow pattern in which the point-to-point readings are within ± 20 percent of the average airflow velocity for the total area of the laminar flow work zone.
Unitary System: A room unit which performs part or all of the air conditioning functions. It may or may not be used with a central fan system.
Unloader: A device on or in a compressor for equalizing the high and low side pressures for a brief period during starting, in order to decrease the starting load on the motor; also a device for controlling compressor capacity by rendering one or more cylinders ineffective.

V

Vacuum: Any pressure less than that exerted by the atmosphere.
Vacuum Breaker: A device to prevent a suction in a water pipe.
Validation: Establishing documented evidence that a system does what it purports to do.
Valve, Modulating: A valve which can be positioned anywhere between fully on and fully off to proportion the rate of flow in response to a modulating controller (see modulating control).
Valve, Two-Position: A valve which is either fully on or fully off with no positions between. Also called an "on-off valve."
Vane Ratio: In air distributing devices, the ratio of depth of vane to shortest opening width between two adjacent grille bars.
Vapor: A gas, particularly one near to equilibrium with the liquid phase of the substance and which does not follow the gas laws. Usually used instead of gas for a refrigerant, and, in general, for any gas below the critical temperature.
Vapor, Saturated: Vapor in equilibrium with its liquid; i.e., when the numbers per unit time of molecules passing in two directions through the surface dividing the two phases are equal.
Vapor, Superheaded: Vapor at a temperature which is higher than the saturation temperature (i.e., boiling point) at the existing pressure.
Vapor, Water: Used commonly in air conditioning parlance to refer to steam in the atmosphere.
Vapor Barrier: A moisture-impervious layer applied to the surfaces enclosing a humid space to prevent moisture travel to a point where it may condense due to lower temperature.
Vapor Pressure: Vapor pressure denotes the lowest absolute pressure that a given liquid at a given temperature will remain liquid before evaporating into its gaseous form or state.
Velocity: A vector quantity which denotes, at once, the time rate and the direction of a linear motion.
Velocity Head: The pressure needed to accelerate the fluid being pumped.
Velocity Reduction Method: A method of duct sizing where arbitrary reductions are made in velocity after each branch or outlet.
Velocity, Outlet: The average discharge velocity of primary air being discharged from the outlet, normally measured in the plane of the opening.
Velocity, Room: The average, sustained, residual air velocity level in the occupied zone of the conditioned space: e.g., 65, 50, 35 fpm.
Velocity, Terminal: The highest sustained airstream velocity existing in the mixed air path at the end of the throw.
Vena Contracta: At an orifice, the fluid flow will converge from the cross-sectional area of the duct or conduit to the smaller opening area of the orifice. After leaving the orifice, the fluid flow will continue to converge somewhat before diverging to the downstream cross-sectional area of the duct or conduit. The point of the smallest cross-sectional area of the fluid flow is called *vena contracta*. This point also has the highest fluid flow velocity.
Ventilation: The process of supplying or removing air, by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.
Vernal Equinox: The position of the sun midway between its lowest and highest altitude; during the spring it occurs March 21. The sunlit period is approximately the same length as the autumnal equinox.

(see also AUTUMNAL EQUINOX)
Viscosity: That property of semifluids, fluids, and gases by virtue of which they resist an instantaneous change of shape or arrangement of parts. It is the cause of fluid friction whenever adjacent layers of fluid move with relation to each other.
Volatility: Volatilty, surface tension and capillary action of a fluid are incidental to environmental systems. *Volatility* is the rapidity with which liquids evaporates extremely rapidly and therefore is highly volatile.
Voltage (E): The electromotive force in an electrical circuit. The difference in potential between two unlike charges in an electrical circuit is its voltage measured in "volts" (V).
Voltage Drop: The voltage drop around a circuit including wiring and loads must equal the supply voltage.
Volume: Cubic feet per pound of dry air in the air-water vapor mixture as used in psychrometrics.
Volume, Specific: The volume of a substance per unit mass; the reciprocal of density.

W

Water Hammer: Banging of pipes caused by the shock of closing valves (faucets).

Watt (W): A measure of electric power equal to a current flow of one ampere under one volt of pressure; or one joule per second in SI units.
Watt Transducer: A device which converts a current signal into a proportional millivolt signal. Used to interface between current transformers and a load management panel.
Wavelength: The distance between two similar and successive points on an alternating wave. The wavelength is equal to the velocity of the propagation divided by the frequency of the alternations.
Weight: The amount of force a substance exerts under pull by the earth's gravitational field and that force is measured in pounds in the United States.
Wet Bulb Temperature (WB): The temperature registered by a thermometer whose bulb is covered by a saturated wick and exposed to a current of rapidly moving air. The wet bulb temperature also represents the dew point temperature of the air, where the moisture of the air condenses on a cold surface.
Wet bulb Depression: Difference between dry bulb and wet bulb temperatures.
White Room: A room designed to be free of dust and other contaminants, but not controlled to the same level as a cleanroom.
Winter Solstice: The shortest sunlit day of the year at which the sun is at the lowest altitude; it occurs December 21.
Work Station: An open or enclosed work surface with direct filtered air supply.
Work Zone: That volume within the cleanroom which is designated for clean work. The volume shall be identified by an entrance and exit plane normal to the airflow (where there is laminar airflow).
Wye Service: An arrangement of the utility transformers. Abbreviated: Y.

Z

Zoning: The practice of dividing a building into small sections for heating and cooling control. Each section is selected so that one thermostat can be used to determine its requirements.

CHAPTER 16
INDEX

SMACNA

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