Manual D
Residential Duct Systems
Third Edition, Version 2.00

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The Third Edition of ANSI/ACCA ***Manual D*** is the Air Conditioning Contractors of America procedure for sizing residential duct systems. This procedure uses ***Manual J*** (ANSI/ACCA, Eighth Edition) heating and cooling loads to determine space air delivery requirements. This procedure matches duct system resistance (pressure drop) to blower performance (as defined by manufacture's blower performance tables). This assures that appropriate air flow is delivered to all rooms and spaces; and that system air flow is compatible with the operating range of primary equipment. The capabilities and sensitivities of this procedure are compatible with single-zone systems and multi-zone systems. The primary equipment may feature a multi-speed blower or a variable-speed blower.

Edition Three of ***Manual D*** (D3) updates the capabilities and sensitivities of the previous ANSI Standard.

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The *Manual D* Standard

Sections 1 through 13 and Appendices 1 through 5 are the standard. They contain all requirements necessary for conformance to the standard.

Appendices 6 through 18 and all ancillary pages are not part of the standard. This material is informative (or a publishing formality), and do not contain requirements necessary for conformance to the standard.

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Dedication

Professional Dedication

The author offers a personal thank you to Paul Stalnecht, President and CEO, ACCA for his support and to Glenn Hourahan, Vice President, Research and Technology, ACCA for valuable suggestions and for managing the Version 3.00 project. A robust thank you is extended to all the members of the ***Manual D*** committee and to the ACCA staff. The guidance and direction provided by this collection of industry leaders is embodied in ***Manual D***.

Personal Dedication

The author dedicates this work to historical ACCA leadership. This collection of board members and staff created and consistently supported ACCA's educational mission (which dates back to the 1950's). This work also is dedicated to the teachers and instructors that provide training for the industry; and to contractors, system designers, practitioners and code officials that produce quality installations.

Overview of this Manual

The Third Edition of ***Manual D***®© (D3) designs duct systems for low-rise, residential-use buildings. This set excludes dwellings classified as Residential Group R (per Section 310 of the 2006 International Building Code), low-rise apartment buildings, and residential structures converted to commercial use. Use commercial procedures for institutional and high-rise construction.

Organization

Thirteen sections provide information, procedures, calculation tools, and examples relevant to the standard. The first five appendices provide charts, tables, look-up values, supporting detail, guidance and procedures relevant to the standard. The last thirteen appendices provide informative guidance, detail and worksheets that are not part of the standard.

Obligatory Sections and Appendices

Obligatory sections and appendices are required reading (part of the ANSI Standard). Obligatory appendices provide requirements and calculation procedures for sizing residential duct system airways. Obligatory appendices provide implementation examples, explanations and related procedures.

Section 1 — Basic Duct Sizing Principles
Provides a brief summary of the ***Manual D*** procedure.

Section 2 — System Operating Point
Introduces the concept of the system operating point, as it applies to a multi-speed blower operating at one speed and at alternative speed settings.

Section 3 — Blowers
Expanded discussion of blower performance (variable-speed, variable Cfm design; variable-speed, constant Cfm design; altitude and temperature effects; inlet and discharge effects and noise).

Section 4 — System Performance Issues
Discusses the friction rate and pressure drop for straight duct, fitting performance, the standard of care for design and installation, and the pressure drop produced by air-side devices and equipment.

Section 5 — Air Distribution System Design
Summarizes tasks that must be completed prior to using the ***Manual D*** procedure and their relationship to the duct sizing procedure.

Section 6 — Duct Sizing Calculations
General discussion of the components and sequence of the ***Manual D*** procedure.

Section 7 — Sizing Rigid Constant Cfm Duct Systems
Application examples (constant Cfm, rigid duct material, rectangular and radial geometries).

Section 8 — Sizing Flexible Constant Cfm Duct Systems
Application examples (constant Cfm, flexible duct material, rectangular and radial geometries).

Section 9 — Air-Zoned Systems
General discussion of variable Cfm system airway sizing issues and duct sizing procedures Reference to ***Manual Zr*** procedures.

Section 10 — Sizing Rigid Air-Zoned Duct Systems
Application examples (air-zoned system with a bypass duct, air-zoned system with distributed relief).

Section 11 — Sizing Flexible Air-Zoned Duct Systems
Application example (air-zoned system with flexible wire helix duct material, junction box geometry and distributed relief).

Section 12 — Sizing Two Zone Bi-level Duct Systems
Application example (two-zone dampers in trunk ducts adjust the supply air Cfm delivered to the lower level and upper level to compensate for the buoyancy of warm air. The maximum change in the flow rate to each floor is limited by damper stops to ± 20 percent of the single zone Cfm for the level.)

Section 13 — Zone Damper Retrofi
Application example (zone dampers added to existing single-zone constant Cfm system).

Appendix 1 — Tables and Equations
Summary of commonly used equations; air velocity limits.

Appendix 2 — Friction charts, Duct Slide Rules and Equivalency Tables
Friction charts summarize the performance of common duct materials; duct slide rules duplicate friction chart data; tables convert round airway size to rectangular airway size and vice versa.

Appendix 3 — Fitting Equivalent Lengths
Default equivalent length values for duct fittings (fitting pressure drops for 900 Fpm air velocity and a 0.08 IWC/100 Ft friction rate are converted to feet of straight duct that has equivalent air flow resistance).

Appendix 4 — Fitting Equivalent Length Adjustments
Alternative equivalent length values for duct fittings (fitting pressure drops for air velocities less than 900 Fpm and/or friction rates other than 0.08 IWC/100 Ft are converted to feet of straight duct that has equivalent air flow resistance).

Appendix 5 — Terminology
Definitions on terms used by ***Manual D.***

Informative Appendices

Informative appendices are recommended reading (not part of the ACCA/ANSI Standard). Informative appendices provide references to related good practice and/or statutory requirements not covered by this standard (standard and codes that deal with design issues other than airway sizing, installation issues and commissioning issues). Informative appendices also provide general discussion of duct system issues, information and detail concerning certain aspects of this standard, guidance and procedures pertaining to performance issues that are related to air delivery, but which do not affect air delivery.

Appendix 6 — Duct Construction Standards
Summary of codes and standards for fabricating and installing residential duct systems.

Appendix 7 — Standard of Care and Continuity
Recommended standard of care for installing duct systems and for making airway sizing calculations.

Appendix 8 — Residential Air Distribution Systems
Summary of zoning issues (a primary consideration for a successful design); overview of various types of duct systems and duct materials; guidance for selecting the type of duct system; reviews issues that affect duct location and duct fabrication.

Appendix 9 — Equipment and Air-Side Components
Introduces the types of primary equipment, secondary equipment and air-side devices used with residential duct systems.

Appendix 10 — Duct System Efficiency
Duct system efficiency (affect of airway shape, aspect ratio, duct material, heat transmission across duct walls and duct leakage); duct insulation issues, duct leakage issues, moisture and condensation issues, effect on applied cooling and heating efficiency ratings (EER and COP), affect on utility demand load.

Appendix 11 — Duct Leakage and System Interactions
Space pressure, infiltration, exfiltration, supply Cfm, return Cfm, combustion air and vent performance are affected by the tightness of the thermal envelope, the performance of the primary air moving equipment, the use of exhaust equipment, and by supply duct leakage and return duct leakage. Synergistic effects may produce unsafe or undesirable conditions in the occupied space.

Appendix 12 — Air Quality Issues
Brief discussion of indoor air quality issues.

Appendix 13 — Noise
Sources of noise and strategies for reducing or attenuating noise.

Appendix 14 — Testing and Balancing
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Duct System Efficiency

Introduction

Residential duct systems have a direct and significant effect on equipment size, equipment efficiency, equipment malfunctions, envelope infiltration, operating cost, utility demand loads, vent performance, exhaust system performance, indoor air quality, ambient noise, occupant comfort and owner satisfaction. Therefore, the duct system must be carefully designed and properly installed or the benefits of an efficient structure and high-efficiency equipment will not materialize.

Manual D procedures shall be used to design residential duct systems. The subject material includes information about constant Cfm and air-zoned systems, system performance characteristics, duct materials, blower performance, air-side components and airway sizing procedures. The informative part of the manual includes information about duct system efficiency and the synergistic interactions between the duct system, the building envelope, the HVAC equipment, the vents and the household appliances. Indoor air quality, noise control, testing and balancing also are discussed. ***Manual D*** procedures depend on, or interact with other comfort system design procedures, including ***Manual J*** (load calculation), ***Manual S*** (equipment selection) and ***Manual T*** (air distribution).

It is important to emphasize that the procedures documented in this manual shall not be used to design commercial duct systems. This limitation is necessary because the design process for residential is different than commercial:

- For the residential problem, equipment manufacturer's blower data establishes the duct sizing criterion (i.e., available static pressure and total effective length determine the design friction rate value).
- For the commercial problem, airways are sized for a selected friction rate, then duct system performance (Cfm and pressure drop information) determines blower RPM and motor horsepower.
- Other incompatibilities relate to maximum air flow velocities (air velocities are limited to 900 Fpm for dwellings, they can be much larger for commercial buildings); and fitting loss calculations (equivalent lengths are used for residential, pressure drops are used for commercial).

This manual does not have to be read from cover to cover, but there is material that cannot be ignored and material that should not be ignored.

- Sections 5 and 6 are mandatory reading because they define the ***Manual D*** procedure (how ***Manual D*** fits into the overall design process and step-by-step instructions for sizing duct airways).
- Familiarity with the material in Appendices 1, 2 and 3 is mandatory (component performance data, limiting values used by the ***Manual D*** procedure and factors used by the ***Manual D*** procedure).
- Sections 1 through 4 and Appendix 16 are recommended reading. Sections 1 though 4 deal with fundamental concepts and issues relevant to proper use of the airway sizing procedure. Appendix 16 deals with flexible wire helix duct systems that are characterized by a deficient standard of care.
- Appendix 6 is a mandatory reference and Appendix 7 is recommended reading (***Manual D*** endorses the methods and materials requirements found in standards and codes).
- Section 7 provides application examples for various types of constant Cfm systems.
- Section 8 is mandatory reading and Appendix 16 is recommended reading for practitioners that install variable Cfm systems.
- Sections 9 through 13 provide application examples for various types of variable Cfm systems.
- Appendices 8 and 9 provide useful information about air distribution systems and components.
- Appendix 5 defines terminology (which is part of the standard).
- Appendices 10 and 11 are recommended reading (duct system efficiency and duct leakage).
- Appendices 12 and 13 discuss indoor air quality and noise control, and Appendix 14 summarizes testing and balancing requirements.
- Appendix 15 discusses issues that affect air velocity guidance.
- Appendix 4 provides a method to produce alternative equivalent length values for fittings.

Prerequisites and Learned Skills

Manual D procedures process information produced by other calculation tools and information provided by manufacturer's performance data. ***Manual D*** procedures assume the practitioner is familiar with these tools and sources of information. The prerequisites for using ***Manual D*** procedures are summarized here:

Load Calculations Determine Airflow Requirements

In order to correctly size duct system airways, the practitioner shall determine the system airflow requirement (blower Cfm) and room airflow requirements (supply air Cfm for each conditioned room or space).

- For each piece of central heating-cooling equipment, the practitioner shall produce a block load (and zone loads for zoned systems), and a set of room and/or space loads.
- The ANSI procedure for load calculations is provided by ***Manual J***, Eighth Edition Version 2.10 (or later).
- The requirements for producing accurate load calculations are provided by ***Manual J***, Eighth Edition Version 2.10 (or later), Section 2.

Equipment Performance Data Determines Blower Performance

Manual D calculations shall be based on blower performance (Cfm vs. external static pressure) published by the equipment manufacturer.

- Calculated heating and cooling loads, manufacturer's heating performance data and expanded cooling performance data shall be used to select equipment.
- The manufacturer's blower table for the selected equipment determines blower performance.
- The ANSI procedure for selecting heating and cooling equipment is provided by ***Manual S***.

Understanding Duct Performance

The practitioner shall be familiar with duct run attributes, which are sectional shape (round or rectangular, for example), cross-sectional dimensions (diameter or length and width), airflow rate (Cfm), airflow velocity (Fpm), and a friction rate (pressure drop in Inches Water Column per 100 feet of duct length).

- For a given operating condition, duct performance is summarized by a set of variables, which are, airway size (diameter or equivalent diameter), airflow rate (Cfm), airflow velocity (Fpm) and friction rate (IWC/100 Ft).
- These variables are interdependent and their relationship is summarized by a friction chart or a duct sizing slide rule (if two items are known, the other two are read from the chart or slide rule).
- These relationships depend on the type of duct material (some materials produce more resistance to airflow than others).
- The practitioner must be able to read friction charts and how to use duct sizing slide rules before attempting to apply ***Manual D*** procedures.
- Instructions for using the ACCA Duct Sizing Slide Rule are provided with the tool.
- The ACCA publication tiled ***Understanding the Friction Chart*** provides instructions for using a friction chart.

Evaluate Equivalent Length and Effective Length

The observed length of a duct run is determined by the measured centerline length of the runs. The effective (flow-resistance) length of a duct run is much longer than the observed length because of the resistance produced by duct fittings. The practitioner shall understand these concepts and master these skills.

- Duct fittings have a flow-resistance length. This length is the length of straight duct (feet) that would produce the same resistance as the fitting. This length is called *fitting equivalent length*.
- The practitioner shall be able to calculate the total effective length of a duct run (the sum of the duct run lengths, plus the sum of relevant fitting equivalent lengths.)
- Instructions for making effective length calculations are provided by ***Manual D*** (latest edition).

Evaluate Pressure Drop for External Components and Devices

Blower pressure is used to move a flow through the resistance produced by duct runs, by duct fittings and by components or devices installed in the flow path. Pressure used to move air through components and devices is not available for duct runs and fittings. The practitioner shall understand these concepts and master these skills.

- External components and devices are items that were not in place when blower data was collected at a test stand (components and devices that were in place during the blower test are listed in the blower table notes).

- Blower power used to move air though an external component or device is not available to move air through a flow path (duct runs plus fittings).
- An external component or device may be a cooling coil, a water coil, an electric coil, a filter or filter option, a supply grille, a return grille, a balancing damper, an automatic flow control damper, etc.
- Component or device pressure drop is read from manufacturer's performance data, except for hand dampers, supply air grilles and return air grilles.
- For ***Manual D***, the default pressure drop for a hand damper, supply air grille or return air grille is 0.03 IWC.
- Instructions for determining component or device pressure drop are provided by ***Manual D*** (latest edition).

Evaluate Available Pressure

The pressure drops for external components and devices are subtracted from external static pressure (the blower table pressure). The result is the available static pressure. The practitioner shall understand these concepts and master these skills.

- External static pressure is the pressure read from the blower table, which already has been adjusted for the components and devices that were in place during the blower test (as listed in the blower table footnotes).
- The pressured drop for components and devices not in place when the blower was tested are subtracted from the blower table pressure. The result is the available static pressure.
- Duct airway sizing calculations are based on the pressure that is available to move air through the duct runs and fittings.
- Instructions for determining available static pressure are provided by ***Manual D*** (latest edition).

Use the Design Friction Rate to Size Airways

For ducted supply flow, the positive static pressure is a maximum at the duct entrance (at the blower), it diminishes along the length of the path (because of friction), and is zero as it enters the space. For ducted return flow, the static pressure is zero as it enters the return grill, it diminishes (becomes more negative) along the length of the path (because of friction), and reaches its maximum negative value as it enters the blower. This behavior is summarized by a duct friction rate. The practitioner shall understand these concepts and master these skills.

- The pressure drop for a supply duct run equals the positive pressure at the entrance (at the blower discharge collar).
- The pressure drop for a return duct run equals the negative pressure at the exit (at the blower return collar).
- The pressure drop for a complete circulation path equals the positive supply pressure plus the magnitude of the negative return pressure (supply pressure value—negative return pressure value).
- The rate at which pressure is dissipated along the circulation path is a friction rate, which depends on the effective length of the path.
- Friction charts and duct sizing slide rules only apply to ducts that are 100 feet long, so these tools summarize duct performance for a special case (the pressure drop for 100 feet of duct).
- The circulation path for a real duct system may have (and usually does have) a total effective length that is less than or greater than 100 feet. Friction charts and duct slide rules do not model the performance of real duct systems (except for those that have 100 feet of effective length).
- Available pressure and total effective path length shall be converted to a design friction rate (IWC/100 Ft) before using a friction chart or duct slide rule.
- Airway size is read from a friction chart or duct slide rule for the design friction rate (IWC/100 Ft) and the local airflow rate (Cfm).
- Instructions for determining the design friction rate and local airflow rate are provided by ***Manual D*** (latest edition).

Observe Limitations Concerning Noise

Noise is an occupant comfort and satisfaction issue. Moving air generates noise. Duct airway sizes that are adequate for air delivery may generate unwanted noise. The practitioner shall understand these concepts and master these skills.

- Airways are sized for the air delivery requirement (based on the design friction rate and Cfm). The resulting airway size and Cfm determines air velocity.
- Air velocity is compared to the velocity for an acceptable noise level.
- Airway size is increased if air velocity exceeds the velocity limit for noise.
- Air velocity limits are provided by ***Manual D*** (latest edition).

Section 1

Basic Duct Sizing Principles

Poor heating and cooling performance is commonly attributed to inadequate equipment size when the actual problem is a restrictive or deficient duct system. Air-side design is critical. Poor air-side design causes inadequate heating and/or cooling in some or all rooms. This section introduces the basic principles for sizing duct runs. These principles are the basis of the ***Manual D*** duct sizing procedure.

1-1 Pressure Units

The pressures for residential air distribution systems are quite small, typically less than 0.025 pounds per square inch (positive or negative). Because these pressures values are so small, it is more convenient to use inches of water column (IWC) for the pressure unit (27.7 inches water column equals 1.0 pound per square inch.)

For the USA, Inches Water Column is the unit of choice for air distribution system design work and for summarizing blower performance. However, Pascals (Pa) are used for blower door testing and duct blaster testing (1.0 IWC = 249 Pa).

1-2 Blower Performance

Blowers move air through duct systems. The flow rate (Cfm) delivered by a blower depends on the external resistance (pressure) the blower has to work against. This behavior is summarized by blower data, which may be a table (Figure 1-1) or a graph (Figure 1-2). Notice that the air flow rate decreases as resistance increases.

Blower Data for One Wheel RPM	
Cfm	**Resistance (IWC)**
1,300	—
1,350	0.69
1,400	0.62
1,450	0.55
1,500	0.47
1,550	0.39
1,600	0.31
1,650	0.23
1,700	0.14
1,750	0.04
1,800	—

Figure 1-1

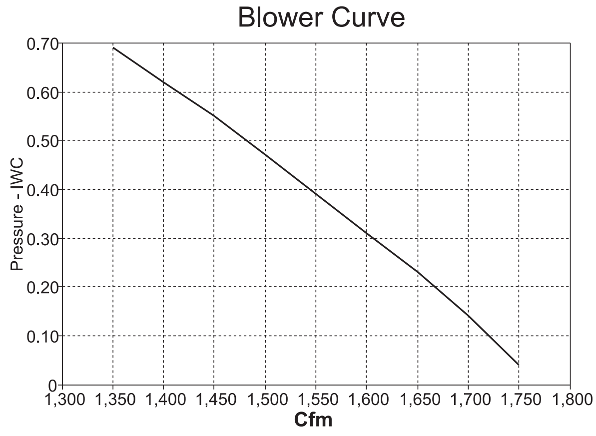


Figure 1-2

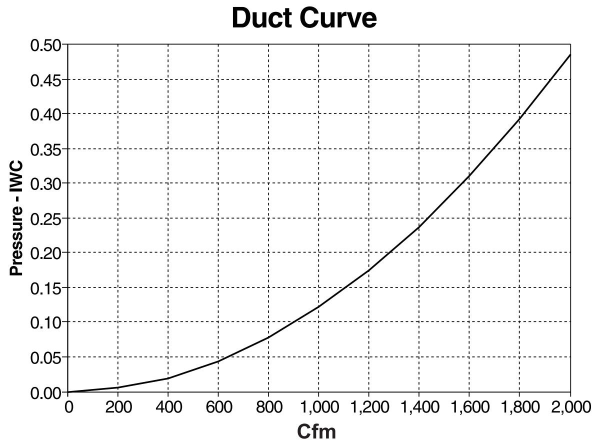


Figure 1-3

1-3 Duct Performance

Resistance is created when air is forced through a duct system. This resistance is caused by friction. Figure 1-3 provides an example of duct system performance. Notice that resistance increases rapidly as more air is forced through the duct.

1-4 System Operating Point

If a blower (Figure 1-2) is connected to a duct system (Figure 1-3) there is only one possible operating point. Since this point must be compatible with blower performance, it must fall on the blower curve. And since this point must

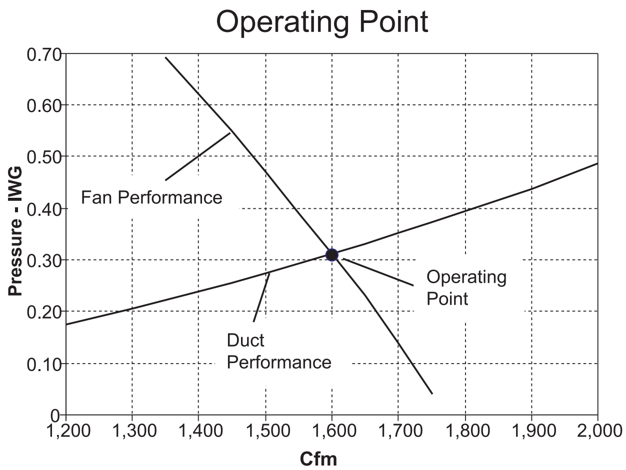


Figure 1-4

be compatible with duct performance, it must fall on the system curve. This can happen only at the intersection of the two performance curves, as demonstrated by Figure 1-4.

For this example the system delivers 1,600 Cfm to the conditioned space. If the practitioner is not satisfied with this air flow rate, blower performance may be adjusted or duct system performance may be modified. Since the blower performance curve (Figure 1-2) depends on a fixed blower geometry (per the manufacturer's design) and blower wheel speed, the only way to change the blower performance curve is to change wheel speed. And, since the duct system performance (Figure 1-3) depends on the duct geometry, duct fittings, and the duct material, the only way to change the duct system curve is to alter duct geometry, use different fittings or change duct material.

1-5 Objective of the Residential Sizing Procedure

Residential equipment manufacturers provide a blower with the equipment package (furnace or air handler). The basic objective of the ***Manual D*** procedure is to design a duct system that it will work with the blower that is supplied with the HVAC equipment.

To meet this objective, the air flow resistance produced by the duct system (duct runs, duct fittings and air-side components), in terms of static pressure drop, shall match the external static pressure (ESP) produced by the blower package (furnace or air handler) when the blower delivers the desired Cfm.

This concept is demonstrated by Figure 1-5. In this case the blower delivers 1,000 Cfm when it works against a resistance of 0.20 IWC. Therefore, the only acceptable duct size is the size that produces a 0.20 IWC of resistance when the flow rate is 1,000 Cfm. Ultimately, the required duct size is determined by a friction chart or duct slide rule, but first it is necessary to distinguish between a pressure drop and a friction rate.

1-6 Pressure Drop and Friction Rate

A pressure drop (PD) is the pressure loss between any two points in a duct system. For example, in Figure 1-5, the pressure drop for 300 feet of duct is 0.20 IWC. (Note that IWC units are used for pressure drop values.)

A friction rate (FR) is the pressure drop between two points in a duct system that are separated by a specific distance. Friction charts and duct slide rules use 100 feet for the reference distance (see Appendix 2). Therefore, before using a friction chart or duct slide rule to size a duct run, the system pressure drop value must be converted to a friction rate value for 100 feet of duct. (Friction rate units are inches water column per 100 feet of duct, or for convenience, IWC/100.)

This equation converts a pressure drop (PD) value to a friction rate (FR) value. Where TEL represents the total effective length of the duct run. (TEL is explained in the following sections.)

$$FR = \frac{PD \times 100}{TEL}$$

For example, in Figure 1-5, the friction rate for the duct run is equal to 0.067 IWC/100. This value is determined as follows:

$$FR = \frac{0.20 \times 100}{300} = 0.067 \ \ IWC / 100$$

Now that the friction rate is known, duct size is determined by using a friction chart or a duct slide rule. Figure 1-6)next page) shows that a 15-inch diameter sheet metal

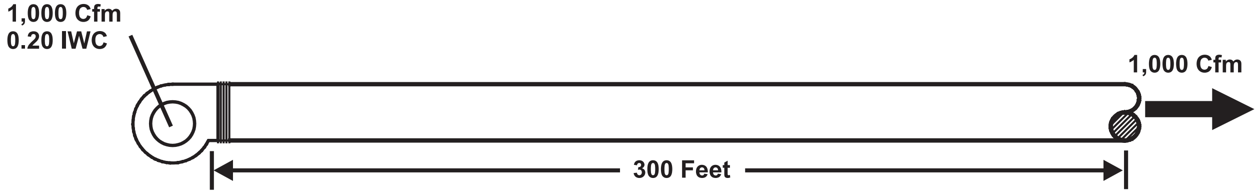


Figure 1-5

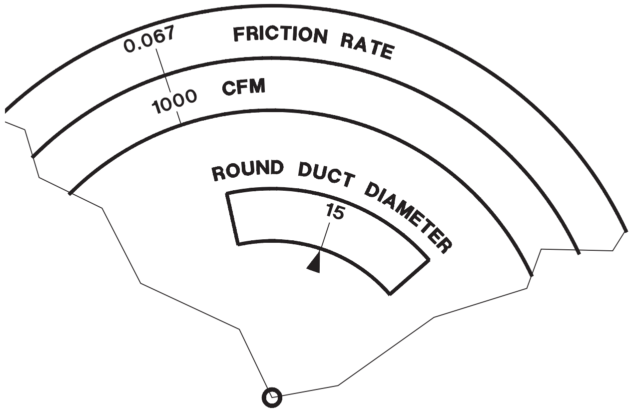


Figure 1-6

duct is required for 1,000 Cfm and a 0.067 IWC/100 friction rate. (See Appendix 2 for more information about using friction charts and duct slide rules.)

1-7 Effective Length

Duct runs have straight sections and fittings; and pressure losses are produced by these elements. Therefore, the total pressure drop for a duct run equals the pressure loss for all straight sections plus the pressure loss produced by each and every fitting in the duct run.

It is not unusual for a fitting pressure loss to be equal, to or greater than, the pressure loss for a fairly long section of straight duct. For example, a fitting could produce the same air flow resistance as a 60-foot section of straight duct. In this case the fitting is said to have an equivalent length (EL) of 60 feet.

Fitting equivalent lengths are a convenient way to account for fitting pressure losses because fitting length values are simply added to the straight run lengths. The resulting total effective length (TEL) represents the total air flow resistance of the duct run. The corresponding pressure drop (PD), depends on the friction rate (FR/100) and duct length, as explained by Section 1-6.

For example, adding a few jogs to the Figure 1-5 duct system may increase the effective length of the run by 80 feet (see Figure 1-7). For this scenario, the blower performance is the same and the Cfm is the same, but the effective duct length is 80 feet longer (TEL = 380 feet). Since a longer duct produces more air flow resistance, a larger duct section is required.

As explained in Section 1-5, the only acceptable duct size is the size that produces a resistance of 0.20 IWC when the flow rate is 1,000 Cfm. For this example the friction rate for the sizing calculation is 0.053 IWC/100, as demonstrated here:

$$FR = \frac{0.20 \times 100}{380} = 0.053 \quad IWC / 100$$

Now that Cfm and friction rate are known, duct size is determined by using a friction chart or a duct slide rule. This produces a 16-inch diameter for a metal duct.

- The flow rate is 1,000 Cfm
- The friction rate is 0.053 IWC/100
- The metal duct diameter is 16 Inches

1-8 Ducted Return

In the two previous examples there was no return duct, so all the blower pressure is used to move the air through the supply duct. If a return duct is added to the system (Figure 1-8, next page), some pressure is used to move air through the return duct, so less pressure is available for the supply-side of the system.

Procedurally, this example is not any different than the previous examples, except the return duct adds 100 feet of effective length to the system. Blower performance is still the same, and Cfm is still the same, but now the effective length is 480 feet. Since the total effective length produces more resistance, a larger duct diameter is required. When the sizing calculation is based on 480 feet, the design friction rate is 0.042 IWC/100, as demonstrated here:

$$FR = \frac{0.20 \times 100}{480} = 0.042 \quad IWC / 100$$

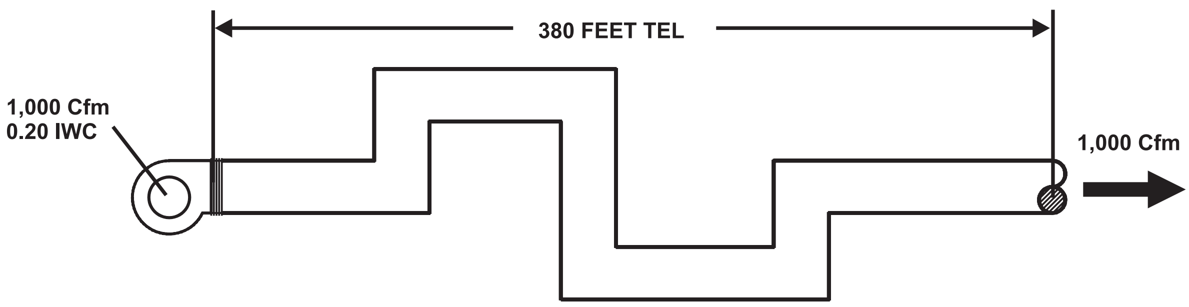


Figure 1-7

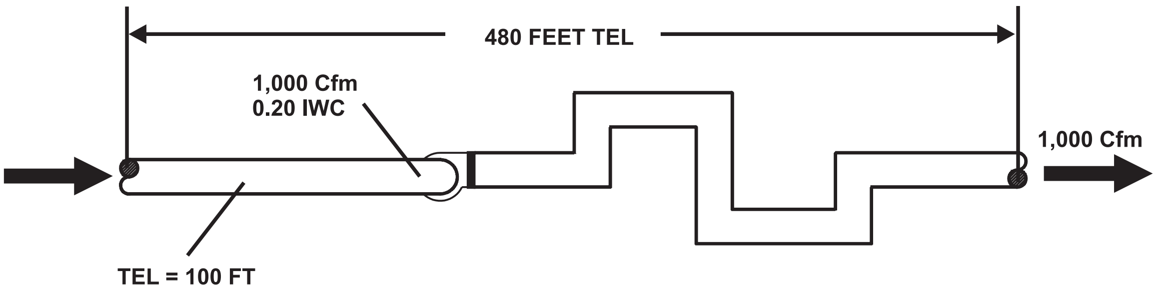


Figure 1-8

Now that Cfm and friction rate are known, duct size is determined by using a friction chart or a duct slide rule. One size is used for both sides of the system.

- The flow rate is 1,000 Cfm
- The friction rate is 0.042 IWC/100
- The metal duct diameter is 16.6 Inches

1-9 Branch Ducts

Supply air systems normally have more than one outlet and many return air systems have more than one inlet. For example, Figure 1-9 shows a system that has three outlets and two inlets. In this case the system has six effective lengths (circulation paths).

- Air flows into R2 and out of S1 or S2 or S3.
- Air flows into R1 and out of S1 or S2 or S3.

The procedure for sizing these duct runs is not any different than used for previous example. In this case, the appropriate sizes are based on the largest effective length value, which is 480 feet (380 feet for the supply-side plus 100 feet for the return-side).

If the blower can deliver the required flow to the outlet that requires the most blower pressure and capture the required flow from the inlet that requires the most blower pressure, it will certainly satisfy the air flow requirement at all other supply and return openings.

As demonstrated by the previous example, the design friction rate is 0.042 IWC/100 for a 480 foot length. This friction rate is used to size all trunk ducts and all branch runs. This friction rate, the Figure 1-9 Cfm values and the duct slide rule provide the round duct sizes, as demonstrated by Figure 1-10 (next page).

1-10 Pressure Drop for Air-Side Components

A resistance is created when air is forced through equipment or a device that is installed in the air stream; a filter, coil, damper, supply outlet or return grille, for example. This resistance translates to a pressure drop across the component. The size of this pressure drop depends on the flow (Cfm) through the component. Figure 1-11 (next page) provides an example of the air-side performance of an electric heating coil. Notice that the pressure drop increases rapidly as more and more air is forced through the coil.

1-11 Available Static Pressure (ASP)

Component pressure drops are very important because the pressure dissipated by one or more items must be subtracted from the external static pressure value from the

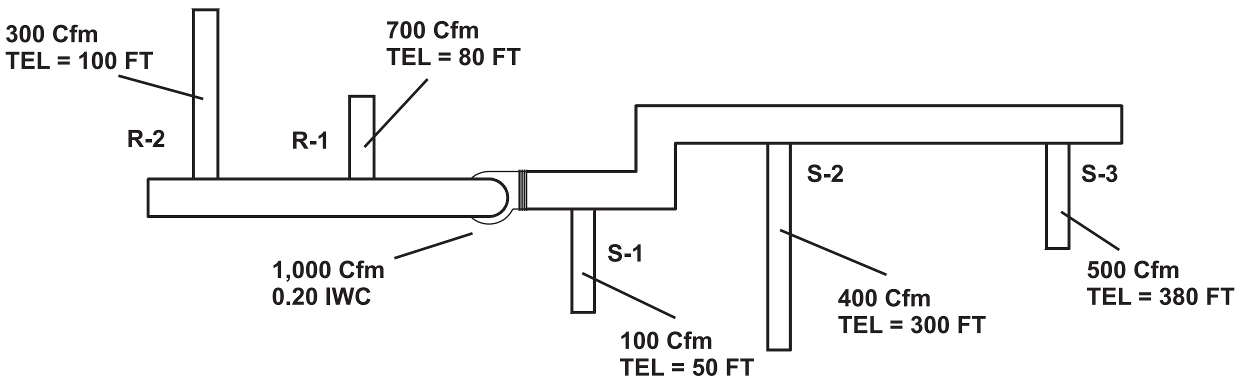


Figure 1-9

Trunks	Cfm	FR	Diameter
Fan to S1	1,000	0.042	17"
S1 to S2	900	0.042	16"
S2 to S3	500	0.042	13"
Fan to R-1	1,000	0.042	17"
R1 to R2	300	0.042	11"
Runouts	**Cfm**	**FR**	**Diameter**
S1	100	0.042	7"
S2	400	0.042	12"
S3	500	0.042	13"
R1	700	0.042	15"
R2	300	0.042	11"

Sheet metal duct diameters rounded up to eliminate fractional dimensions.

Figure 1-10

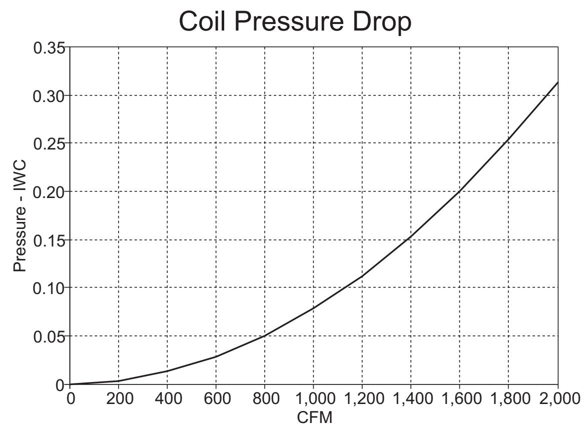


Figure 1-11

OEM's blower table. The resulting value is the pressure that is available to move the air through the straight runs and fittings of a circulation path.

- Duct sizes are not based on the amount of pressure that the blower produces (i.e, the external pressure for the furnace or air handler), but on the net pressure that is available to move the air through the circulation path that produces the most resistance to airflow (i.e., the critical path).
- For a given amount of blower pressure, duct sizes have to be increased to compensate for the pressure dissipated by components and devices located in the critical circulation path of the duct system.

This concept is demonstrated by Figure 1-12. The blower still delivers 1,000 Cfm when working against 0.20 IWC of resistance, but 0.08 IWC of pressure is dissipated by the coil. Therefore, the only acceptable duct size is the size that produces a resistance of 0.12 IWC when the flow is 1,000 Cfm. Based on 0.12 IWC, the design friction rate is 0.025 IWC/100, and the size from the duct calculator is 18.6 inches. Notice that if no coil was installed, the duct size would have been based on 0.20 IWC of pressure, which would have resulted in a 0.042 IWC/100 friction rate and a 16.6 inch duct (see Section 1-8).

1-12 Velocity Limits

The friction rate procedure described above always produces a design that delivers adequate air flow (Cfm) at each supply and return. However, this is not the only design criterion. If the velocity in an airway gets too high, it may produce turbulence and objectionable noise. When this happens, the duct airway size that satisfies the friction rate requirement is increased to comply with air velocity limitations specified by Table A1-1.

Once the design value for air velocity is known, a friction chart or duct sizing slide rule is used to determine the duct size for the desired velocity. For example,

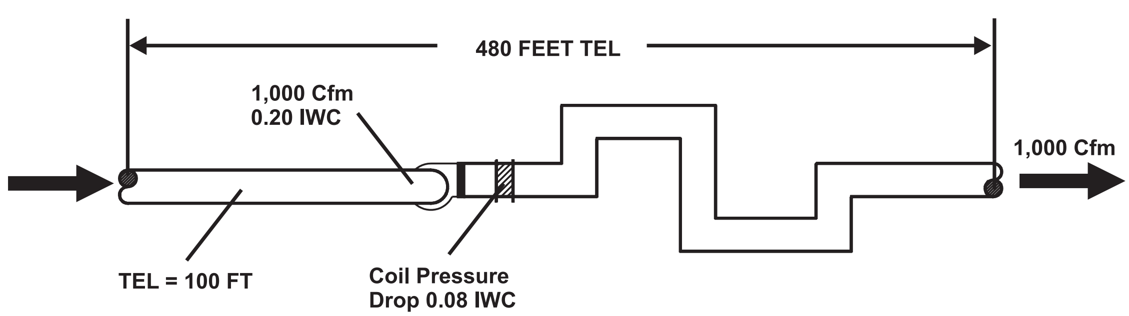


Figure 1-12

Air Velocity for Noise Control Subject to Notes 1, 2 and 8								
Component	**Supply Side (Fpm)**				**Return-Side (Fpm)**			
	Conservative		**Maximum**		**Conservative**		**Maximum**	
	Rigid	**Flex**	**Rigid**	**Flex**	**Rigid**	**Flex**	**Rigid**	**Flex**
Trunk Ducts	700	700	900	900	600	600	700	700
Branch Ducts	600	700	900	900	500	600	700	700
Supply Outlet Face Velocity	Size for Throw		700 Note 7		—		—	
Return Grille Face Velocity	—		—		—		500	
Filter Grille Face Velocity	—		—		—		300	

1) The design friction rate is affected if air velocity exceeds 900 Fpm (fitting equivalent lengths are for 900 Fpm or less).
2) System resistance considerations supercede velocity considerations (minimum acceptable airway size shall be based on the local Cfm value and the design friction rate). Air way size shall be increased if the local air velocity exceeds the maximum limit.
3) This table applies to metal duct with transverse seams and metal fittings (duct runs and fittings not lined or wrapped with insulating material).
4) This table applies to flexible wire helix duct with duct board junction box fittings.
5) Maximum velocities may be exceeded when construction has less surface irregularities (no transverse seams or less irregularity at transverse seams, and very efficient fittings); and has a sound absorbing attribute (duct board or duct liner).
6) Authoritative guidance concerning velocity limits for aerodynamically efficient and/or sound absorbing designs is not available at this time.
7) The velocity limit for a supply outlet may be ignored if the noise criteria (NC) value for a grille, register or diffuser is 30 or less over the range of Cfm values that will flow through the device (or combination of devices, if a damper is involved), during any mode of system operation.
8) Air velocity limits are superceded by measured noise criteria (NC) values for low rise dwellings (Notes 1 and 2 still apply).
· NC values measured by sound meter in middle of the room when normal human ear perceives maximum HVAC system noise.
· Measured NC equals or exceeds 30 with comfort system off; measured NC shall not increase by more than 3 with comfort system on.
· Measured NC less than 30 with comfort system off; measured NC shall not exceed 33 with comfort system on.

Copy of Table A1-1

Figure 1-13 shows that a 14.5 inch duct is required for a 1,000 Cfm flow rate and a 900 Fpm (maximum) velocity.

Figure 1-14 (next page) shows that the available pressure is 0.60 IWC and the total effective length is 480 feet, so the design friction rate is 0.125 IWC/100. Duct airways are sized for this friction rate, provided air flow velocities do not exceed limits.

Figure 1-15 (next page) shows how velocity limits affect final duct size. In this case a sheet metal duct system has a blower that produces a relatively large amount of pressure when it moves 1,000 Cfm through the system.

Figure 1-15 summarizes the design calculations for this system. In this case, most of the duct sizes that satisfy the friction rate procedure are too small to satisfy the Table A1-1 velocity requirement. When this happens, the final duct size is the size that is compatible with the velocity limit.

So, there are two sizes for every duct section; the size that satisfies the air flow requirement (per the friction rate procedure) and the size that satisfies the velocity limit. The final design always uses the larger of these two sizes.

If the airway size is dictated by a velocity limit, the duct run produces less air flow resistance than a smaller size that satisfies the friction rate procedure. The reduction in path resistance causes increased air flow for the path, which will exceed the desired value. Excessive path Cfm problems are resolved by installing and adjusting balancing dampers in appropriate locations.

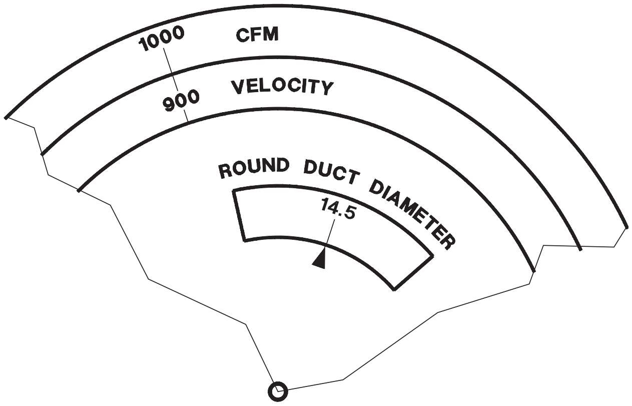


Figure 1-13

1-13 Psychometric Air Flow Calculation

The air flow rate (Cfm) that is required for a conditioned space, depends on the sensible cooling load (Btuh) or the heating load (Btuh) for the space, and on the temperature difference (TD) between the supply air and the room air.

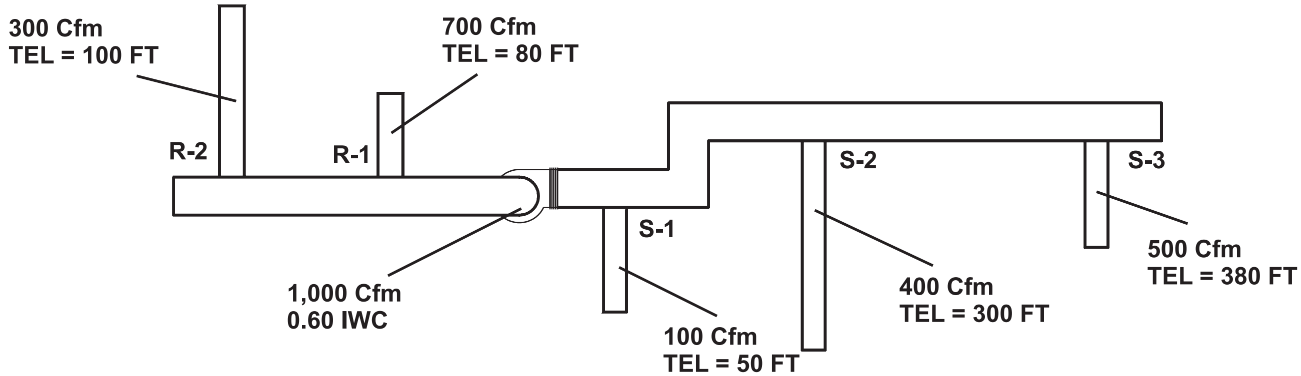


Figure 1-14

This relationship is summarized by the sensible heat equation, which is adjusted for altitude (ACF is the altitude correction factor from ***Manual J***, Table 10A).

$$Cfm = \frac{Load}{1.1 \times ACF \times TD}$$

For example, at sea level (ACF = 1.0), the value for supply air temperature is 56°F, and the temperature of the air in the conditioned space is 75°F, so the TD value is 19°F. If the ***Manual J*** calculation for the sensible cooling load for a room or space is 4,500 Btuh , the supply air flow rate for cooling is 237 Cfm.

$$Cfm = \frac{4{,}500}{1.1 \times 1.0 \times 19} = 237$$

This principle applies to any room or space, and to the entire conditioned space. For example, if the Manual J calculation for the sensible cooling load for the entire conditioned space is 30,000 Btuh , the design air flow rate for cooling is 1,435 Cfm.

However, the heating load and sensible cooling load on the equipment may be larger than the load for the conditioned space. This difference is typically caused by an engineered ventilation load, and/or a blower heat load. Therefore, the ***Manual J*** load value for a psychrometric airflow calculation equals the sum of the line 14, line 15

Trunk Section	Cfm	FR	Diameter for Air flow	Velocity (Fpm)	Design Velocity (Fpm)	Diameter for Velocity	Design Diameter
Fan to S1	1,000	0.125	13.5"	1,050	900	14.5"	15"
S1 to S2	900	0.125	13.0"	1,000	900	13.8"	14"
S2 to S3	500	0.125	10.3"	890	900	10.2"	11"
Fan to R1	1,000	0.125	13.5"	1,050	700	16.5"	17"
R1 to R2	300	0.125	8.5"	780	700	9.0"	9"
Runouts	**Cfm**	**FR**	**Diameter for Air flow**	**Velocity (Fpm)**	**Design Velocity (Fpm**	**Diameter for Velocity**	**Design Diameter**
S1	100	0.125	5.5"	600	900	4.5"	6"
S2	400	0.125	9.5"	840	900	9.1"	10"
S3	500	0.125	10.3"	880	900	10.2"	11"
R1	700	0.125	11.7"	960	600	14.8"	15"
R2	300	0.125	8.5"	780	600	9.6"	10"
Sheet metal duct diameters rounded up to eliminate fractional dimensions.							

Figure 1-15

and line 20 values on Manual J, Form J1, as explained by Section 6-9.

A psychometric air flow calculation is correct in principle, but it is not used in practice. Per ***Manual S***, the practitioner uses total load values from ***Manual J***, Form J1, line21 and OEM expanded performance data to find equipment that has the correct output capacity for the line 21 loads. Since this data correlates output capacity values with blower Cfm values, the design value for blower Cfm is read from the OEM's equipment capacity table.

1-14 Blower Cfm and Room Cfm

The design value for blower Cfm shall be available before ***Manual D*** duct sizing calculations begin. As previously noted, this value is obtained when primary equipment is selected by use of ***Manual S*** procedures (which require ***Manual J*** solutions for the heating and cooling loads).

Once the design value for the blower Cfm is known, the air flow (Cfm) for each room is estimated by multiplying the blower Cfm by the room load and dividing by the ***Manual J*** equipment sizing load for the entire space served by heating-cooling the equipment (total heating load or total sensible cooling load from line 21 of Form J1).

$$\text{Room Cfm} = \frac{\text{Blower Cfm x Room Load}}{\textbf{Equipment Sizing Load}}$$

Normally, two calculations are required for each room. One for the heating and one for sensible cooling. This work is expedited by using a heating factor (HF) and a cooling factor (CF), as defined here:

$$HF = \frac{\text{Blower Cfm for Heating}}{\textbf{Sizing Load for Heating}}$$

$$CF = \frac{\text{Blower Cfm for Cooling}}{\textbf{Sizing Load for Sensible Cooling}}$$

*Note: The blower Cfm for heating may be equal to, or different than, the blower Cfm for cooling. When different blower Cfm values are used, it is normally because the cooling Cfm is not compatible with a furnace temperature rise limitation. This is determined when equipment is selected (per **Manual S** procedures).*

With heating and cooling factors in hand, the heating Cfm and cooling Cfm for each room or space is estimated by multiplying the room load (line 21, Form J1 or Line 20 on the $MJ8_{AE}$ spreadsheet) by the corresponding flow factor.

Room Heating Cfm = HF x Room Heating Load
Room Cooling Cfm = CF x Room Sensible Load

However, only the larger of the two Cfm values is used to size the duct run.

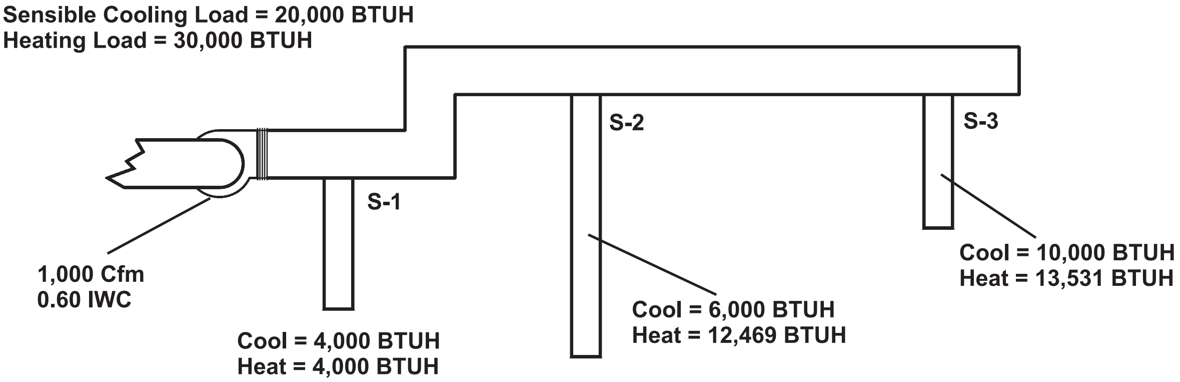


Figure 1-16

Room	Cooling			Heating			Design Cfm
	Load	CF	Cfm	Load	HF	Cfm	
S1	4,000	0.050	200	4,000	0.033	133	200
S2	6,000	0.050	300	12,469	0.033	415	415
S3	10,000	0.050	500	13,531	0.033	450	500
Blower	*CF = 1,000 / 20,000 = 0.050*			*HF = 1,000 / 30,000 = 0.033*			1,000
For this example, the same blower speed used for heating and cooling.							

Figure 1-17

Design Cfm = Larger of the two room Cfm values

Sometimes there is not much difference between the room heating Cfm and the room cooling Cfm. Sometimes there is a significant difference. ***Manual D*** sizes duct runs for the worst-case condition, which may be for heating or cooling. In any case, airway sizes are compatible with maximum air flow requirement and more than adequate for a lesser requirement. In other words, the correct air flow for cooling is often incorrect for heating, so duct airway are sized for the worst case (largest) Cfm value, as explained by the sidebar on the this page.

For example, Figure 1-16 (previous page) shows a system that provides cooling and heating to three large rooms. ***Manual J*** and ***Manual S*** procedures were used to generate block load and room load values, to select equipment and to obtain a blower Cfm value. This is all that is required to determine the design Cfm for each room.

Figure 1-17 (previous page) summarizes the room Cfm calculations for the example system. Note that the larger of the two room Cfm values determines airway sizes (the duct slide rule converts a Cfm value and a friction rate value to a duct diameter or equivalent rectangular shape).

1-15 Trunk Cfm Values

The Cfm flowing through any point along a supply trunk equals the sum of the Cfm values flowing through the downstream supply air outlets. Two calculations are required, one for summing cooling Cfm values, and one for summing heating Cfm values. Depending on circumstances, the two sums may be equal or different. When they are different, the larger Cfm value is used for supply trunk sizing.

The Cfm flowing through any point along a return trunk equals the sum of the Cfm values flowing through the upstream return grilles. Two calculations are required, one for summing cooling Cfm values, and one for summing heating Cfm values. Depending on circumstances, the two sums may be equal or different. When they are different, the larger Cfm value is used for return trunk sizing.

- If 100% of the blower Cfm enters a supply trunk or leaves a return trunk, simply use the blower Cfm value for trunk airway sizing because this is the maximum Cfm that can flow through the trunk.
- Calculations similar to the Figure 7-8 example apply when the blower feeds two supply trunks.
- Calculations similar to the Figure 7-13 and Figure 8-1 example apply when one long supply trunk reduces in size downstream from a set of branch runouts.
- The Figure 7-18 example has one section of supply trunk that carries 100% of the blower Cfm, three secondary supply trunks that carry a portion of the blower Cfm, and two return trunks that carry a portion of the blower Cfm.
- The Figure 8-8 example has four supply trunks that carry a portion of the blower Cfm, and two return trunks that carry a portion of the blower Cfm.

System Air Balance

Manual D procedures (Figure 1-10 duct sizes, for example) do not produce a perfectly balanced system because airway sizes are based on a worst case friction rate. Therefore, the airway sizes for the runs in the critical circulation path are correct, but the airway sizes for the runs in the other circulation paths are larger than needed. This causes excessive (i.e., somewhat more than the design Cfm value) air flow through the shorter circulation paths. Also note that:

- ***Manual D*** sizes duct runs for the most demanding operating condition, which may be heating or cooling. So, airway sizes are for the maximum seasonal air flow requirement.
- Airways are first sized for correct airflow resistance, with no regard for velocity or standard size consequences. System balance is affected when an airway size is increased to comply with an air velocity limit, and when a duct slide rule size is rounded to a standard size.
- Excessive air flow problems are resolved by installing balancing dampers in the branch runout ducts. Once the balancing dampers are adjusted, the total effective length of all the circulation paths are approximately equal, and each supply air run will deliver the desired Cfm.
- A system balanced for the desired heating Cfm values is usually not correct for the cooling Cfm values, and vice versa. Hand dampers optimize the performance of single-zone systems and zone damper systems
- See Section 6-2 and Section 6-3. For zone damper systems, see Appendix 9 in ***Manual Zr***)

Section 2

System Operating Point

The air distribution system operating point is defined in Section 1-4. This section shows how speed adjustments and system resistance changes affect the system operating point. This section also shows how to determine the operating point, how to adjust the operating point and how to use this information during field tests.

2-1 System Operating Point

Blower performance is summarized by a pressure vs. flow rate curve and duct system performance is characterized by a resistance-flow relationship. These behaviors may be summarized by two tables or one graph. The advantage of using a graph is that the operating point is obvious, as shown by Figure 2-1.

The operating point is the only possible condition that can occur when a particular blower (operating at a given speed) is attached to a duct system. Therefore, the practitioner shall be sure the Cfm value at the operating point equals (or is close to) the design Cfm value. (The design Cfm is determined when ***Manual J*** loads and ***Manual S*** procedures are used to select and size heating and cooling equipment.)

The practitioner has little control over the size and power of the blower because it is (normally) a standard component in a residential furnace or air handler. This means that the practitioner must match the air distribution system's performance to the blower. If this is done correctly, the system curve will intersect the blower curve at (or near) the design Cfm value.

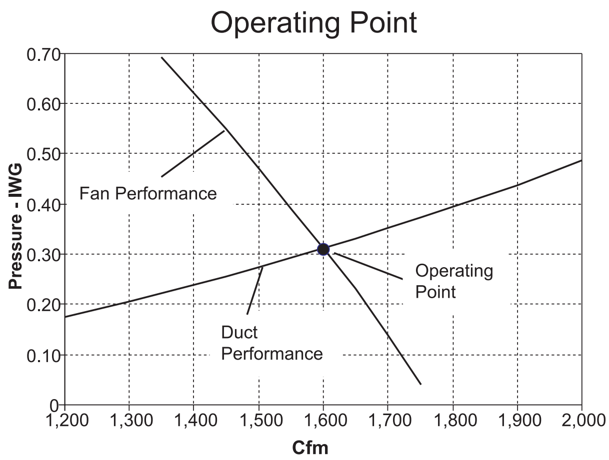


Figure 2-1

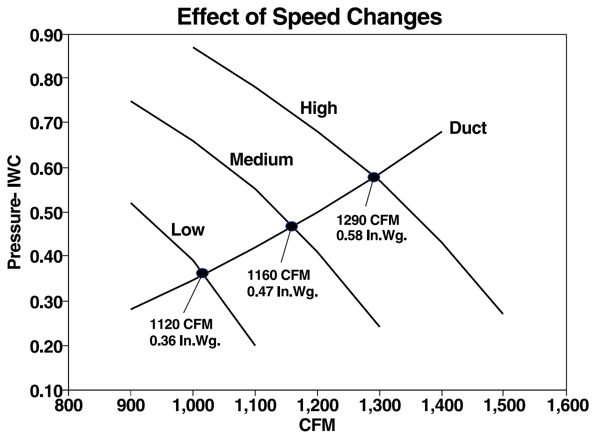


Figure 2-2

2-2 Changing Blower Wheel Speed

If the system curve does not intersect the blower curve at the desired Cfm, the operating point may be adjusted by changing wheel speed. Figure 2-2 shows that a system with a three speed blower has three operating points. (Operating points only occur at the intersection of the system curve and the blower curves for each wheel speed.)

Note that when wheel speed is changed, it is not possible to know what the new flow rate will be by just looking at a blower table. The only thing that can be said about the consequence of a speed change is that more RPM produces more air flow and less RPM produces less air flow.

- A speed change produces a relatively small Cfm change if the blower curves are steep.
- Steep blower curves characterize the performance of wheels that have forward curved vanes.
- The forward curved design is commonly used for residential equipment.

As already explained, blower Cfm depends on how the blower interacts with the resistance produced by the air distribution system. A balance point diagram, such as Figure 2-2, shall be used to evaluate the consequences of a wheel speed change.

2-3 Changing System Resistance

If the system curve does not intersect the blower curve at the desired Cfm, the operating point may be adjusted by changing system resistance. Such changes may be intentional or unintentional. Intentional adjustments are normally made by opening or closing balancing dampers.

- If balancing damper positions decrease or increase system resistance, the new operating point occurs at the intersection of the blower curve and a new system curve.
- System resistance and operating point will unintentionally change if equipment and/or a device is added to or removed from the duct system (a pleated filter retrofit by a homeowner or contractor, for example).
- System resistance and operating point may be modified by replacing inefficient fittings with more efficient fittings, or vice versa.
- The resistance produced by any component or fitting is unintentionally modified when it gets fouled by dirt, debris or biological growth.

Figure 2-3 shows how a change in system resistance affects the system curve and the operating point. Note that when flow path resistance is changed, the new operating point occurs at the intersection of the blower curve and the new system resistance curve.

A system resistance adjustment, by itself, cannot be used to evaluate the effect of a system modification because system Cfm also depends on the blower performance. The only thing that can be said is that increased resistance produces less air flow and decreased resistance allows more air flow. Therefore, a system balance point diagram shall be used to predict the consequences of a system resistance change (see Figure 2-3).

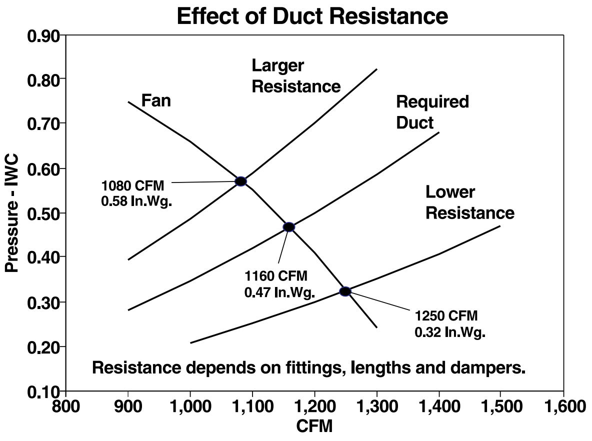


Figure 2-3

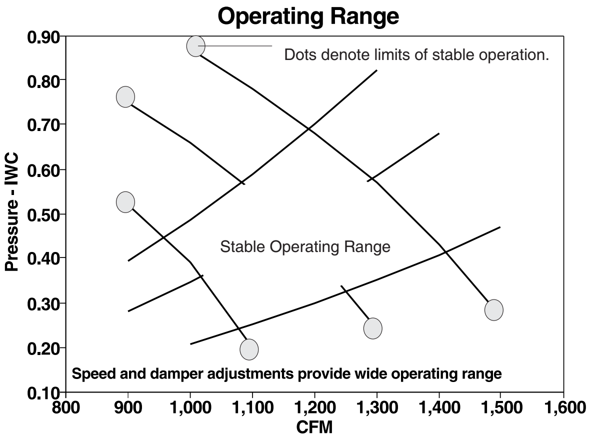


Figure 2-4

2-4 Operating Envelope

Hand dampers and speed adjustments produce a large set of operating points. Blower wheel speed shall be adjusted first. Ideally, blower Cfm should slightly exceed the design Cfm when all the registers, balancing dampers and control dampers are in the wide open position. After wheel speed is set, branch runout dampers are adjusted so that each outlet provides the desired air flow. Figure 2-4 provides an example of a stable operating envelope produced by combinations of speed changes and damper adjustments.

- Blowers are designed to operate over a certain range of flow rates and pressures.
- For a given blower wheel speed, stable operation is defined by the range of Cfm and pressure values in the manufacturer's blower table (see the blower curves on Figure 2-4).
- System air flow may be deficient, erratic or negligible if system resistance causes the blower to operate outside its stable operating range.

2-5 Drawing the System Curve

A system resistance curve can be drawn if a single performance point (P_1, Cfm_1) is known. This point may be the desired operating point or it could be a point that was measured during a balancing test.

This equation associates system resistance (P_x) with air flow (Cfm_x) for any point on the duct system curve. Figure 2-5 (next page) shows the system resistance curve for P_1 = 0.216 IWC and Cfm_1 = 1,200.

$$P_x = P_1 x \left[Cfm_x / Cfm_1 \right]^2$$

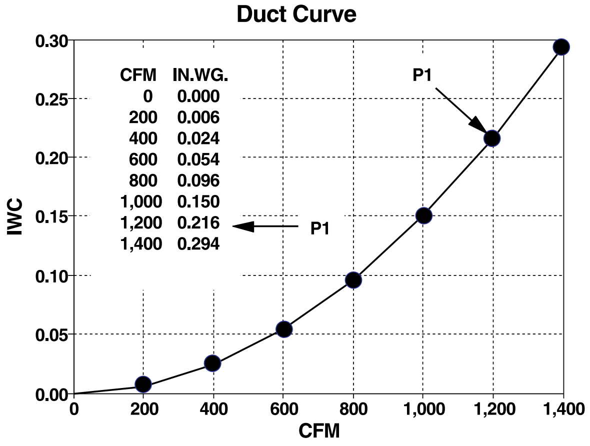


Figure 2-5

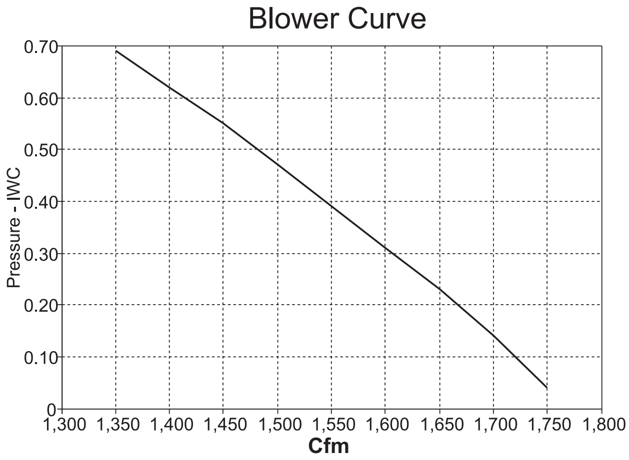


Figure 2-6

Blower Data for One Wheel RPM	
Cfm	**IWC**
1,300	—
1,350	0.69
1,400	0.62
1,450	0.55
1,500	0.47
1,550	0.39
1,600	0.31
1,650	0.23
1,700	0.14
1,750	0.04
1,800	—

Figure 2-7

2-6 Drawing the Blower Curve

Some manufacturers publish blower performance data in graphical form, which makes the blower curve immediately available. Figure 2-6 provides an example of this method of presentation.

If blower performance is not presented as a graph, it will be defined by a table that correlates flow (Cfm) and pressure (IWC). In this case, the blower curve is graphed from data points in the manufacturer's blower data table. Figure 2-7 shows the blower table for the Figure 2-6 blower curve.

2-7 Establishing the Operating Point

The operating point is determined by drawing the blower curve and the system curve on the same graph. This produces a balance point diagram and the operating point is at the intersection of the two curves (see Figure 2-8, next page).

Note that a balance point diagram is not used to design a residential duct system. In practice, the operating point (Cfm value and resistance value) is read from the equipment manufacturer's blower table.

- Use ***Manual J*** to evaluate block load.
- Use manufacture's expanded performance data and ***Manual S*** procedures to select equipment.
- The design Cfm is the Cfm used for equipment selection.
- Inspect the manufacturer's blower table for the heating-cooling equipment.
- Use the design Cfm to find a set of resistance values for the listed blower wheel speeds.
- Select a wheel speed and note the corresponding resistance value (IWC external static pressure).
- The design Cfm and the selected resistance value define the system operating point for ***Manual D*** calculations.

For example, suppose 1,350 Cfm is obtained from the equipment selection procedure and blower performance is summarized by Figure 2-9 (next page). In this case, there are two possible operating points. The external static pressure value is 0.25 IWC if the blower operates at medium speed or 0.69 IWC if the blower operates at high speed. (The blower cannot deliver 1,350 Cfm at low speed.)

After a blower wheel speed and external static pressure value are selected, duct runs are sized so that system resistance is equal to, or less than, the available pressure value. In other words, if the system resistance curve was

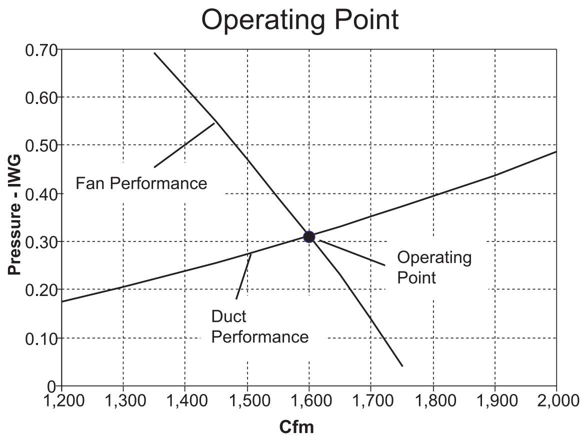


Figure 2-8

External Static Pressure (IWC)			
Cfm	**Blower Wheel Speed**		
	High	**Medium**	**Low**
1,150			0.45
1,200			0.30
1,250		0.49	0.05
1,300		0.37	
1,350	0.69	0.25	
1,400	0.62	0.14	
1,450	0.55	0.04	
1,500	0.47		
1,550	0.39		
1,600	0.31		
1,650	0.23		

Figure 2-9

superimposed on the blower curve, it would intersect the blower curve at the correct operating point.

2-8 Wheel Speed Design Value

If possible, calculations are made for a mid-range (medium) wheel speed because this provides the most flexibility for field adjustment. However, high speed is compatible with a relatively large system resistance and low speed is compatible with a relatively low system resistance. In any case, the system operating point shall be compatible with the Cfm value that was used to select equipment.

For example, if the equipment selection procedure was based on 1,350 Cfm, Figure 2-9 shows that one operating point is compatible with 0.25 IWC of system resistance and the other is compatible with 0.69 IWC of system resistance. In this case, the medium speed setting is desired, but is not arbitrarily used for duct sizing calculations.

- The medium speed is appropriate if the ***Manual D*** resistance calculation for the critical circulation path (longest supply-side plus longest return-side) is approximately 0.25 IWC.
- The high speed setting is appropriate if the ***Manual D*** resistance calculation for the critical circulation path is significantly larger than 0.25 IWC.
- Balancing dampers dissipate excess pressure if the resistance of the critical circulation path is less than the available static pressure.

2-9 Balance Point Diagram — Application

After the air distribution system is designed and installed, air flow rates shall be verified by field tests. The primary task is to measure total air flow (Cfm) delivered by the blower when all dampers and registers in the wide open position. A blower wheel speed adjustment is required if this test shows that system air flow is excessive or deficient. However, the consequences of a speed change cannot be predicted by looking at the blower table.

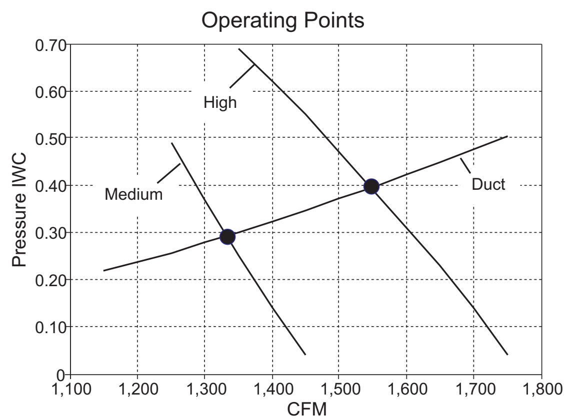


Figure 2-10

For example, the desired flow rate for an existing air distribution system is 1,600 Cfm, Figure 2-10 defines blower performance and system flow rate is measured at medium speed. But system air flow and pressure measurements indicate the blower delivers 1,330 Cfm against 0.29 IWC of system resistance. Since the air flow is less than desired, performance at high speed setting is evaluated.

Figure 2-10 shows that at high speed, the blower will deliver 1,600 Cfm against 0.31 IWC of resistance; therefore 1,600 Cfm will not flow through the existing duct system because system resistance at 1,600 Cfm is larger than 0.31 IWC (Figure 2-10 shows 0.43 IWC for 1,600 Cfm). Therefore, a balance point diagram similar to Figure 2-10 must be used to determine the benefit of the speed change.

- If blower speed in increased from medium to high, the new operating point will fall at the intersection of the system curve and the high-speed blower curve.
- Figure 2-10 shows the system flow rate will increase to 1,550 Cfm at 0.40 IWC at the high speed setting, which is less than the desired value (1,600 Cfm).
- This is the best the blower can do when operating at high speed. If a larger flow rate is required, the resistance of the critical circulation path must be reduced (clean the filter and coil, replace inefficient fittings with aerodynamically efficient fittings, measure the pressure drop across air-side equipment and devices and substitute components that have smaller pressure drops.)

Section 3

Blowers

A centrifugal fan (blower) moves air against the resistance produced by the air distribution system (return grille, return duct work, filter, casing, heat transfer equipment, supply duct work, balancing dampers and supply air grille, for example). This section looks at the types of blowers found in residential equipment packages.

3-1 Blower Selection

Normally, the practitioner does not specify blower size or performance because the blower is a standard, factory-installed component of the equipment package. Therefore, one part of the ***Manual D*** procedure is used to see if blower performance is roughly compatible with the air flow and pressure requirements of the proposed air distribution system (see the Friction Rate Worksheet, Appendix 19). After verifying blower capability, duct airways are sized so that system air flow resistance matches the external static pressure produced by the blower when it delivers the desired Cfm (see Section 1).

3-2 Multi-Speed Operating Point Blower

At a selected blower wheel speed, an operating point blower has a unique static pressure value for each Cfm value. In this case, blower performance data appears in manufacturer's literature as a table or graph that correlates delivered flow (Cfm) with the airflow resistance (IWC) that the blower works against.

For example, Figure 3-1 shows tabular data for a three speed blower. Note that for each blower-wheel speed, pressure values are listed for a few Cfm values. These Cfm values define the operating range for a given speed. This range defines the upper and lower limits of the blower's air delivery capability at a given speed.

- For a given speed there is a unique static pressure value for each Cfm value.
- The largest Cfm value corresponds to the flow rate delivered against a smallest system resistance.
- The smallest Cfm value corresponds to the flow rate delivered against a large system resistance.
- The lowest Cfm value is compatible with the aerodynamic stability of the fan blades.

If blower Cfm is too low, the fan blades will stall and cause erratic performance (possibly little or no flow). Therefore, for a given wheel speed, system resistance shall not exceed the value that causes stall. For example, Figure 3-1 shows that the maximum system resistance that will not cause stall is 0.69 IWC at high speed, 0.49 IWC at medium speed and 0.45 IWC at low speed.

Operating point blowers may be driven directly by a multi-speed (PSC) motor or indirectly by a belt drive. The performance characteristics for such designs are summarized by tables that are similar to Figure 3-1 or performance curves that are similar to Figure 3-2.

External Static Pressure (IWC)			
Cfm	**Blower Wheel Speed**		
	High	**Medium**	**Low**
1,150			0.45
1,200			0.30
1,250		0.49	0.05
1,300		0.37	
1,350	0.69	0.25	
1,400	0.62	0.14	
1,450	0.55	0.04	
1,500	0.47		
1,550	0.39		
1,600	0.31		
1,650	0.23		

Figure 3-1

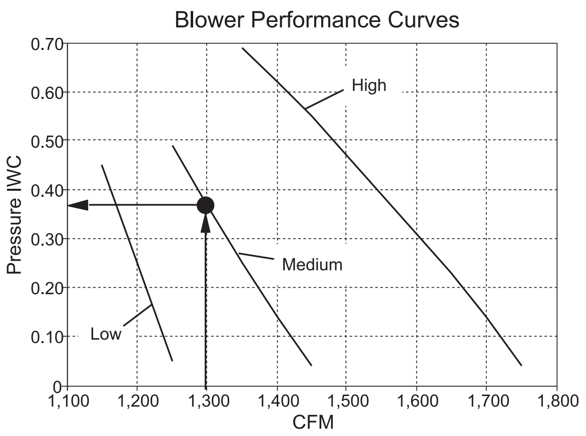


Figure 3-2

Note that there is one performance curve for each wheel speed. Also note that blower Cfm decreases as system resistance increases. Therefore, to obtain the desired system flow rate (blower Cfm), system resistance shall be equal to, or somewhat less than, the external pressure value from a blower table or blower curve.

For example, if the blower operates at medium speed and the desired flow rate is 1,300 Cfm, Figures 3-1 and 3-2 show that the air distribution system must be designed and balanced so system resistance is 0.37 IWC when system air flow is 1,300 Cfm.

3-3 Variable-Speed, Operating Range Blower

Blower wheel speed may be continuously adjusted when driven by a variable-speed motor or drive. In this case, an unlimited number of blower curves summarize blower performance.

Figure 3-3 shows just a few of the blower curves for a wheel that operates between 300 and 1,400 RPM. Note that speed control can accommodate any operating point (Cfm value and resistance value) that is within the envelope of the blower curves.

Figure 3-3 is not the only way to summarize variable-speed performance. Equipment manufacturers may provide performance data in tabular form, but there is no standard format for presenting performance data.

Figure 3-4 provides an example of tabular data. In this case, the table correlates an ECM motor speed setting and a Cfm value with a range of external static pressure (system resistance) values.

For example, if SR-5 setting is selected, the blower will deliver 1,600 Cfm when it operates against a system resistance that ranges between 0.15 IWC and 0.80 IWC. Note

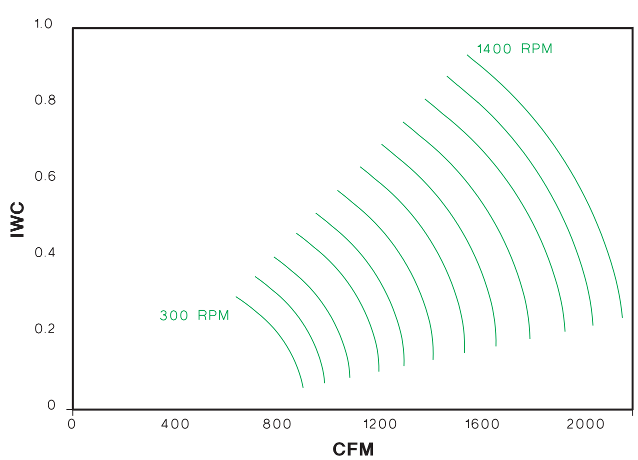


Figure 3-3

ECM Blower Performance

Setting	Max. RPM	Cfm	ESP (IWC)
SR-1	500	800	0.22 – 0.40
SR-2	700	1,000	0.24 – 0.50
SR-3	1,000	1,200	0.23 – 0.70
SR-4	1,200	1,400	0.19 – 0.80
SR-5	1,300	1,600	0.15 – 0.80
SR-6	1,400	1,800	0.11 – 0.75
SR-7	1,400	2,000	0.08 – 0.50

1) ECM motor control logic.
2) Wheel speed range = 300 to 1,400 RPM.

Figure 3-4

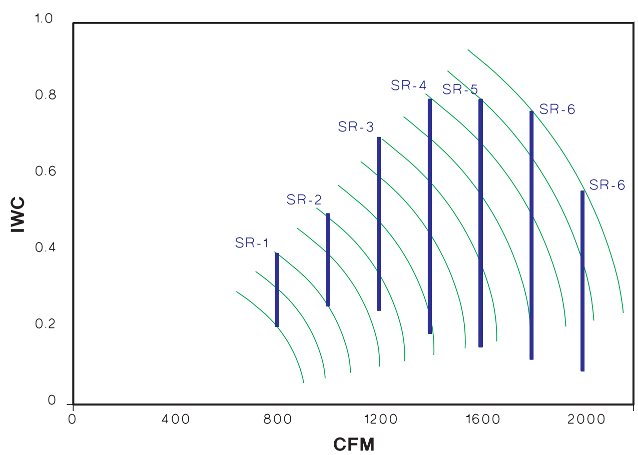


Figure 3-5

that wheel speed increases from a low RPM to 1,300 RPM as system resistance increases from 0.15 IWC to 0.80 IWC.

Figure 3-5 shows the relationship between the tabular data (Figure 3-4) and the graphical data (Figure 3-3). In this diagram, the tabular data corresponds to a set of vertical lines. Each of these lines correlates a static pressure range with the Cfm value for the selected motor speed setting.

Note that, for a pure variable-speed blower, there are an unlimited number of pressure-Cfm lines (one for each Cfm value from the OEM's minimum allowable Cfm value to the maximum allowable Cfm value), as demonstrated by Figures 3-3. For an ECM blower, the OEM's motor control logic provides a defined set of Cfm lines, as demonstrated by Figures 3-4 and 3-5.

Figure 3-6 (next page) shows how a variable-speed, operating range blower that has an ECM motor interacts with an air distribution system. Per Figure 3-4, the SR-4 speed

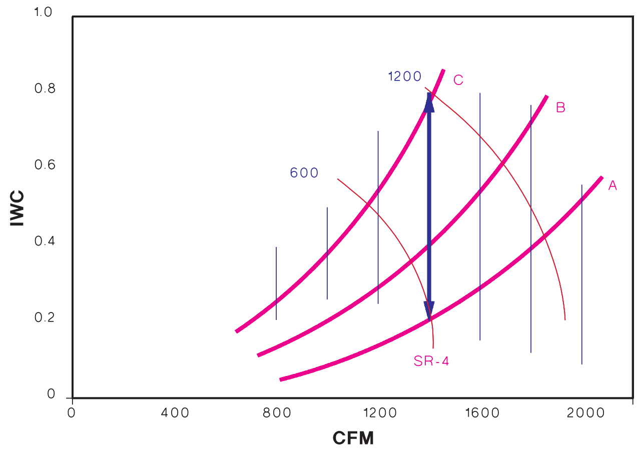


Figure 3-6

setting, provides 1,400 Cfm for 0.19 IWC to 0.80 IWC of external airflow resistance. Figure 3-6 graph shows that as the system resistance varies (curves A, B and C), system Cfm holds constant as wheel speed modulates between a minimum value and 1,200 RPM.

Figure 3-6 shows that operating range blowers are more forgiving than operating point blowers because system airflow resistance does not have to be carefully matched to a specific external static pressure value — it just has to fall within a range of resistance values.

For example, if the desired flow rate is 1,600 Cfm and if the blower motor is operated on the SR-5 speed setting (per Figure 3-4), the resistance air distribution system resistance may vary between 0.15 and 0.80 IWC.

A more common example of operating range data is provided by Figure 3-7. In this case, performance is tabulated for various air flow set point options. The selected air flow set point (heating Cfm or cooling Cfm) will be maintained, or nearly maintained, over a range of external static pressures. Related issues:

- Design values for cooling Cfm and heating Cfm are determined when primary equipment is selected per ***Manual S*** procedures.
- For furnaces and electric heating coils, the heating Cfm shall be compatible with ***Manual J*** loads and the equipment manufacturer's limits for minimum and maximum temperature rise (heating Cfm may not equal cooling Cfm).
- Jumper cables or switches, or pins or taps are set for the desired heating Cfm and cooling Cfm.
- The equipment's control package sensors monitor motor RPM and motor power draw or external static pressure.
- Blower performance data is mapped into control package software, or a memory chip. The four variables are Cfm, IWC pressure, RPM and Watts. If two variables are known, the other two variables are uniquely determined.
- Computer logic software (or chip) compares instantaneous RPM and Watts (or external static pressure) with the embedded blower performance map and determines Cfm.
- If the instantaneous Cfm value is not equal to the Cfm set point, the software increases or reduces motor speed, and the blower delivers the desired Cfm.

<table>
<tr><th colspan="13">ECM Blower Performance (Cfm Vs. Resistance)</th></tr>
<tr><th rowspan="2">Function</th><th rowspan="2">Cfm Set Point</th><th rowspan="2">ESP (IWC) Range</th><th colspan="10">External Static Pressure (IWC)</th></tr>
<tr><th>0.10</th><th>0.20</th><th>0.30</th><th>0.40</th><th>0.50</th><th>0.60</th><th>0.70</th><th>0.80</th><th>0.90</th><th>1.00</th></tr>
<tr><td>Low Heat</td><td>735</td><td>0.0 – 0.5</td><td colspan="4">735</td><td>725</td><td colspan="5">~</td></tr>
<tr><td>High Heat</td><td>1,180</td><td>0.0 – 1.0</td><td>1,160</td><td>1,165</td><td>1,175</td><td colspan="6">1,180</td><td>1,175</td></tr>
<tr><td rowspan="5">Cooling Set point Options</td><td>525</td><td>0.0 – 0.5</td><td colspan="4">525</td><td>510</td><td colspan="5">~</td></tr>
<tr><td>700</td><td>0.0 – 0.5</td><td colspan="3">700</td><td>695</td><td>685</td><td colspan="5">~</td></tr>
<tr><td>875</td><td>0.0 – 1.0</td><td colspan="6">875</td><td>865</td><td>855</td><td>845</td><td>840</td></tr>
<tr><td>1,050</td><td>0.0 – 1.0</td><td colspan="8">1,050</td><td>1,045</td><td>1,035</td></tr>
<tr><td>1,225</td><td>0.0 – 1.0</td><td>1,205</td><td>1,215</td><td colspan="7">1,225</td><td>1,210</td></tr>
<tr><td>Maximum</td><td>1,400</td><td>0.0 – 1.0</td><td>1,395</td><td colspan="6">1,400</td><td>1,385</td><td>1,360</td><td>1,310</td></tr>
<tr><td colspan="13">No notes concerning air-side components in place when blower was tested.</td></tr>
</table>

Figure 3-7

Cooling Cfm Set Point for Blower Driven by an ECM Motor

Equipment manufacturer's ECM blower data may correlate Cfm set point with cooling capacity. For example, an OEM's version of Figure 3-7 may suggest something like this:

- 1.5 Tons set for 525 Cfm
- 2.0 Tons set for 700 Cfm
- 3.0 Tons, set for 875 Cfm
- Etc.

The practitioner shall investigate the consequences of using suggested Cfm set points based on cooling Tons. For example, the preceding suggestions reduce to 350 Cfm per Ton, which is incorrect for most USA dwellings.

- The ***Manual J*** sensible heat ratio equals the sensible load divided by the total (sensible plus latent) load.
- 350 Cfm per Ton is appropriate for dwellings that have a relatively large latent load, compared to the sensible load (***Manual J*** sensible heat ratio about 0.70).
- 400 Cfm per Ton is appropriate for dwellings that have a significant latent load, compared to the sensible load (***Manual J*** sensible heat ratio about 0.80).
- 450 to more than 500 Cfm per Ton is appropriate for dwellings that have a small latent load, compared to the sensible load, or no latent load (***Manual J*** sensible heat ratio 0.90 to 1.00).

The preceding bullet items just summarize a concept. The procedure for determining cooling Cfm and cooling Cfm per Ton for a particular dwelling is provided here:

- ***Manual J*** shall be used to calculate sensible and latent loads for a particular dwelling for a particular location.
- The total, sensible and latent capacities of the cooling equipment (operating at ***Manual J*** design conditions) shall conform to the sizing limits provided by ***Manual S***, Second Edition, Section N2.
- ***Manual S*** procedures and expanded cooling performance data for condenser-evaporator combinations, as published in manufacturer's engineering literature, shall be used to determine cooling equipment size (sensible and latent capacity) and the design blower Cfm for cooling.
- The design Cfm for cooling shall be a value allowed by the OEM's expanded performance data for cooling capacities.
- Manufacturer's data may list total cooling capacity; or total capacity equals the sum of listed sensible capacity and listed latent capacity.
- Cooling Tons equals total capacity divided by 12,000.
- The correct Cfm per Ton for a particular dwelling equals the design Cfm for cooling divided by cooling Tons.

After cooling equipment has been selected for a specific indoor coil Cfm, go to the corresponding blower data table and search for a Cfm set point that matches the

ECM Blower Performance (Cfm Vs. Resistance)

Nominal (ARI) Condensing Unit Tons	Cfm / Ton Options	Blower Speed Switch Setting					External Static Pressure (IWC)				
		S1	S2	S3	S4		0.10	0.30	0.50	0.70	0.90
3.5	350	open	close	open	close	Cfm	1,210	1,210	1,220	1,230	1,230
	400	open	close	open	open		1,400	1,440	1,450	1,450	1,410
	450	open	close	close	open		1,590	1,600	1,610	1,600	1,380
4.0	350	close	open	open	open	Cfm	1,390	1,400	1,430	1,440	1,420
	400	close	open	close	open		1,620	1,650	1,670	1,640	1,480
	450	close	open	open	close		1,840	1,830	1,820	1,670	1,490
5.0	350	open	open	open	close	Cfm	1,800	1,780	1,780	1,700	1,530
	400	open	open	open	open		2,050	2,010	1,960	1,710	1,530
	450	open	open	close	open		2,160	2,040	1,920	1,790	1,620

1) No deduction required for standard throw-away filter (blower unit tested with filter in place).
2) Deduct pressure drop for wet refrigerant coil.
3) The 350 Cfm / Ton option is for comfort in a very humid climate; the 400 Cfm / Ton option is for most climates that produce a latent load; the 450 Cfm / Ton option is the dry climate setting.

Figure 3-8

design value for cooling Cfm. Some equipment manufacturer's provide blower data for more than one Cfm per Ton setting (see Figure 3-8, previous page).

The preferred scenario would be to find a Cfm set point option that matches the Cfm/Ton requirement and the design cooling Cfm value. Since set point Cfm options are limited, the desired cooling Cfm value may not be available. In this case, the practitioner shall select the Cfm set point that exceeds the design cooling Cfm value by the least amount. For example, if the design cooling Cfm value is 1,150, the Figure 3-7 set point choice would be 1,225 Cfm, or 1,220 for Figure 3-8.

Using more than the design cooling Cfm value to size duct airways assures that duct system performance will be adequate for the larger Cfm set point. However, the larger Cfm value may not be the set point Cfm value after the equipment is installed and commissioned. The operating Cfm set point depends on what Cfm is best for controlling indoor humidity.

- The design value for indoor humidity is specified before making ***Manual J*** calculations.
- The cooling Cfm that will satisfy the sensible and latent loads is determined when ***Manual S*** procedure are used to select cooling equipment.
- Indoor humidity will be somewhat higher than the design value if the operating Cfm exceeds the design Cfm.
- Indoor humidity will be somewhat lower than the design value if the operating Cfm is less than the design Cfm.
- If the design value for indoor relative humidity is 50% or less, the indoor humidity can drift to 55%, or even close to 60%, and still be in the comfort zone.
- The larger blower Cfm value may be the operating Cfm if the indoor humidity is in the comfort zone, and if the occupants are satisfied.
- If the indoor humidity is too high at the larger blower Cfm value, a lower blower Cfm set point determines the operating Cfm
- Duct system airway sizes are compatible with either Cfm set point.

Pressure Data for Operating Range Blowers

Figure 3-7 is similar to furnace blower data published by a major equipment manufacturer. Note that the heading at the top of the pressure values says "External Static Pressure." Also note that the table has no footnotes concerning air-side devices in place when the blower was tested. Therefore, the table implies that a common air-side component, such as a standard throw-away filter is an external device (an item for Step-2 on the Friction Rate Worksheet).

Maximum Static Pressure (IWC)		
Model	**Discharge Cfm**	**Available Static Pressure**
Air-air heat pump with operating range blower (ECM motor).	700	1.00
	875	1.00
	1,050	1.00
	1,225	1.00
	1,400	0.80
No notes concerning air-side components in place when blower was tested.		

Figure 3-9

Figure 3-8 is another exhibit of manufacturer's blower data. In this case, the heading at the top of the pressure values says "External Static Pressure." Also note that this table does have footnotes concerning air-side devices in place when the blower was tested.

Figure 3-9 shows heat pump blower data published by the Figure 3-7 equipment manufacturer. Note that the table has no footnotes concerning air-side devices in place when the blower was tested. Therefore, the blower table implies that common, necessary air-side components, such as a refrigerant coil, electric resistance heater and standard throw-away filter are external devices (items for Step-2 on the Friction Rate Worksheet).

Also note that the header at the top of the pressure column of Figure 3-9 says "Available Static Pressure." The manufacturer's use of this term is not compatible with ***Manual D*** (or with their own furnace blower table), but equipment manufacturer's have no obligation to use ***Manual D*** terminology.

- For ***Manual D***, the Available Static Pressure from Figure 3-9 actually means the External Static Pressure for the duct runs, fittings and all air-side devices not in place when the blower was tested (for Line 1 on the Friction Rate Worksheet).
- For ***Manual D***, the Available Static Pressure (Line 3 on the Friction Rate Worksheet) is the pressure from Figure 3-9 minus the pressure drop for all air-side devices not in place when the blower was tested.

Determining external static pressure for Line 1 on the Friction Rate Worksheet is an important issue. Figure 3-9 shows that the blower produces a relatively large amount of static pressure, but after subtracting the pressure drop for a refrigerant coil, electric resistance heating coil and standard filter, the available static pressure for everything else in the duct system could be less than 0.50 IWC, which isn't much different than a common operating point blower (see the Section 7-5 example).

Each equipment manufacturer has their own format for presenting blower performance data. This data may, or may not, be discounted for factory installed components and devices. When there are no blower table notes pertaining to items in place when the blower was tested, the practitioner shall scrutinize all the manufacturer's data and guidance to determine what External Static Pressure or Available Static Pressure actually means. If this effort fails to produce a definitive answer, the practitioner shall have the equipment manufacturer, or manufacturer's representative, provide a definitive answer (in writing).

Implications for Constant Cfm Systems

The design friction rate for airway sizing depends on an external static pressure value from a blower table (see Sections 1-2 and 3-5). When an ECM motor maintains a constant Cfm (Figure 3-7 or 3-8), there is no external static pressure value for a given Cfm. Instead, there is an external static pressure range. This seems to imply that any external static pressure in the range could be the design value for external static pressure, but this is not the case. The adaptability of ECM blowers is no excuse for not making detailed airway sizing calculations. There are good reasons for keeping external static pressure as low as possible.

- System efficiency decreases and blower motor power increases as external static pressure increases.
- Unnecessary use of blower motor power increases occupant utility bills and adds an unnecessary demand load on the electric power grid.
- The useful range of the blower motor speed depends on external static pressure. For example, if the blower's external static pressure range is 0.0 to 1.0 IWC, and if an inefficient duct system produces 0.90 IWC of resistance for the design Cfm, the effective blower adjustment range is 0.10 IWC. In other words, avoid designs that operate in the top third of the blower pressure range.

Implications for Variable Cfm Systems

After the heating and cooling loads for a zoned damper system or a true variable air volume (VAV) system are determined, the basic procedure for sizing airways is the same as used for a constant Cfm system. This means that the procedure for selecting a design value for external static pressure for an air-zoned system is the same as for a constant Cfm system (as discussed above). However, the design values for air-way Cfm at various point in the duct system may be different than the values used for a constant Cfm system that has the same duct geometry. In addition, an air-zoning system may have a by-pass air duct (see Sections 9-5 and 9-6 of this manual).

3-4 Blower Wheel Speed Vs. System Type

Normally, one blower wheel speed is adequate for single-zone, constant Cfm systems; but in some cases, two-speeds settings are required. Speed control (multi-speed or variable-speed) is very desirable when zoning is provided by zone dampers, and/or when the primary equipment has compressor capacity control, and/or burner capacity control.

Single-Speed Operation

Many comfort systems (a furnace, with or without a cooling coil, a cooling-only unit or heat pump) can operate at one blower wheel speed without degrading the year-round comfort. But if one speed setting is used, there shall be adequate air flow through a furnace heat exchanger, electric coil or refrigerant coil during any operating condition.

- Minimum and maximum air flow requirements for heat exchangers and coils are specified by the equipment manufacturer.
- Cooling coil Cfm has an effect on total cooling capacity, and how total cooling capacity is split between sensible capacity and latent capacity. Therefore, blower wheel speed shall be compatible with the sensible and latent cooling loads.
- As far as air distribution is concerned, single- speed operation is desirable because the supply air grilles and registers are (typically) constant Cfm devices.

Two-Speed Operation

In some cases heating-cooling equipment must accommodate a mismatch between the size of the design cooling load and the size of the design heating load. For example, a relatively large supply Cfm for cooling may not be compatible with the minimum acceptable temperature rise across a furnace heat exchanger. In this case a higher blower wheel speed is appropriate for cooling and a lower speed is used for heating (wheel speed is determined by the heating-cooling switch and has nothing to do with capacity control).

Two blower speeds may be used to tune the performance of single-zone equipment. For example, two-stage furnaces and cooling units or heat pumps that have two stages of compressor capacity, benefit from two blower wheel speeds. This arrangement produces a better match between equipment load, equipment capacity, and heat exchanger or refrigerant coil air flow during part-load operation.

Two-speed operation may be used with a zone damper system. This way, blower Cfm may be reduced when automatic control dampers significantly throttle system Cfm at part-load.

Two-speed operation may cause air distribution problems. If supply grille sizes are based on the maximum air flow requirement, they may be too large to provide adequate mixing at reduced flow.

Variable-Speed Blowers

Variable blower wheel speed is a refinement of the two-speed option (see above). Variable-speed blowers resolve heating-cooling change-over problems and improve the performance of equipment that has staged or modulated heating capacity and/or cooling capacity.

Variable-speed blowers are desirable for zone damper systems, particularly if the capacity of the heating and cooling equipment can be staged or modulated. This way, the system air flow rate is continuously monitored and adjusted as automatic control dampers regulate the heating or cooling capacity delivered to each zone.

- A minimum flow of air through heat exchangers, electric coils and refrigerant coils shall be maintained during any operating condition.
- Variable-speed blowers may cause air distribution problems when blower Cfm is reduced. If constant Cfm grilles, registers or diffusers are used, the throw distance, mixing ability, and generated noise may not be compatible with the range of air flow rates produced by the maximum and minimum wheel speeds.
- Variable volume diffusers provide adequate performance for a range of Cfm values, per the OEM's engineering data.

3-5 External Static Pressure

Blower table data is based on laboratory tests that document the performance of a specific equipment configuration. This data does not apply to any other configuration.

For example, if a furnace is tested with a filter in place, the blower table applies to a heating-only furnace that is equipped with a similar filter, but does not apply to a furnace that is equipped with a cooling coil. In this case, it is necessary to subtract the pressure drop for the cooling coil from the external static pressure value listed in the blower table.

Always read the footnotes for the blower table. These notes may list the components that were installed when the blower test was conducted and/or may list equipment and devices that are not accounted for.

For example, for an air-source heat pump, the Figure 3-10 notes indicate that the pressure drop for a wet cooling coil and a standard filter are accounted for, but the pressure drop for an electric resistance heating coil is not accounted for. In this case, the pressure drop for an electric resistance coil must be subtracted from the external static pressure listed in the blower table.

External Static Pressure (IWC)			
Cfm	**High**	**Medium**	**Low**
1,200			0.45
1,250		0.49	0.30
1,300		0.37	0.08
1,350		0.25	
1,400	0.62	0.14	
1,450	0.55	0.04	
1,500	0.47		
1,550	0.39		
1,600	0.31		
Tested with wet coil and filter in place. Subtract pressure drop for a resistance heating coil.			

Figure 3-10

3-6 Air Density Correction

Unless stated otherwise, blower table data is for standard air. Standard air has a specific weight of 0.075 pounds per cubic foot, which is the specific volume of 70°F air at sea level. Other temperatures and altitudes are non-standard conditions.

As far as the performance of the blower and the air distribution system are concerned, air density effects can be ignored for elevations less than 2,500 feet and air temperatures between 40°F and 110°F. Therefore, an air density adjustment is not required for a large percentage of comfort systems. Altitude effects shall be evaluated when the elevation exceeds 2,500 feet.

Figure 3-11 (next page) shows how altitude affects the performance of the blower, the resistance of the air distribution system and the system operating point. The diagram shows that there is no change in volume flow (blower Cfm) at altitude. However:

- Mass flow (pounds of air per minute) is reduced at altitude because the air is less dense.
- There is less system resistance (IWC) and blower pressure (IWC) at altitude, and both pressure values are reduced by the same amount.
- The system operating point at altitude occurs at the intersection of the sea level Cfm value and a pressure value that is less than the sea level value.

- For a given Cfm, less blower power is required at altitude because mass flow system resistance is reduced.

Since altitude does not affect the Cfm for the system operating point, standard (sea level) duct sizing slide rules (or friction charts) are used to size duct runs. In other words, for a given Cfm, there is no altitude correction for airway sizing calculations.

- At altitude, more supply air Cfm is required to duplicate sea level heating and cooling capacity (i.e, mass flow at altitude must equal mass flow at sea level).
- If sea level Cfm is used at altitude, the heating and cooling capacities of supply air are derated for altitude.
- If sea level capacity is required at altitude, Cfm at altitude is increased to duplicate sea level mass flow.
- Airways are sized for the desired Cfm value.

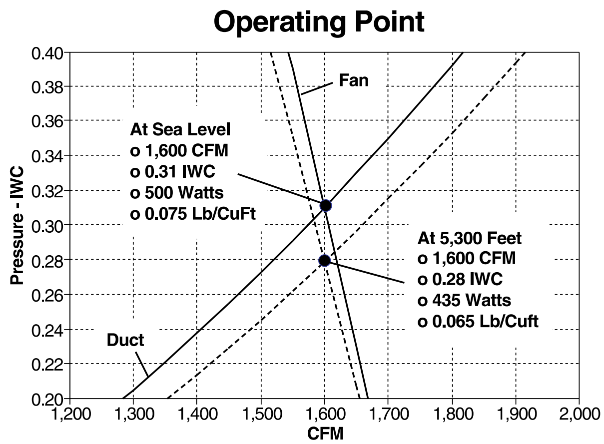


Figure 3-11

Equipment Capacity at Altitude

Published HVAC equipment performance values for heating and cooling capacity are based on sea-level conditions. At altitude, the heating and cooling capacity of HVAC equipment is reduced. Therefore, a slightly larger piece of equipment and/or a slightly larger blower Cfm compensates for reduced capacity. After the design value for blower Cfm is known (when the equipment is selected and sized), the duct sizing procedure for altitude is identical to the sea level procedure. ACCA ***Manual S***, Second Edition (2014) , Appendix 5 provides equipment selection, at altitude, guidance.

3-7 Inlet and Discharge Conditions

Normally, residential air moving equipment has the blower and heat exchanger, or cooling coil, in the same cabinet (i.e., furnace or air handler). When such equipment is tested; straight, full-size duct sections are fitted to the inlet collar and discharge collar. This produces a uniform velocity profile as air flow approaches and leaves the cabinet. Any deviation from this ideal condition degrades blower performance.

When the equipment is installed, the conditions at the inlet and discharge openings seldom duplicate the test stand conditions. Occasionally, elbows and tees are placed close to, or may even be attached to, the cabinet. This practice will degrade the performance of the blower. This loss of performance is accounted for when the total equivalent length of the duct system is calculated. Appendix 3 provides equivalent length values for fittings that are attached to air handling equipment.

3-8 Blower Noise

A blower must be capable of moving the required airflow against the resistance produced by the air distribution system. When possible, a medium wheel speed is used to select equipment because high speed operation produces more noise. But if maximum blower pressure is required to overcome system resistance, equipment selection may be based on a high wheel speed. In any case, air moving equipment in should be located in a room that is physically and acoustically isolated from the occupied space. If the equipment must be installed in or near an occupied space, special measures should be taken to reduce noise. Appendix 13 provides more information about noise control.

Section 4

System Performance Issues

Residential duct system components fall in four categories: straight duct sections, fittings, air-side equipment and air distribution devices. This section provides information about the pressure losses produced by these components.

4-1 Straight Section Pressure Drop

When air flows through straight duct, friction causes the static pressure in the duct to decrease in the direction of flow. Figure 4-1 demonstrates this behavior and the following equation computes the pressure drop (PD) for a section of duct. For this equation, FR is the friction rate (IWC pressure drop per 100 feet of straight duct length), which is read from a duct slide rule or a friction chart; and L is the length (in feet) of a straight section of duct.

$$PD = \frac{FR \times L}{100}$$

4-2 Duct Material

The laboratory friction rate depends on the surface properties of the airway material. Smooth materials have smaller friction rates than rough materials. For example, an aluminum duct has a significantly smaller friction rate than a fiberglass duct.

Manual D uses generic friction charts as a default reference and advises practitioners to use specific friction data provided by the product manufacturer.

- Material construction differences affect friction loss. This applies to all types of duct materials.

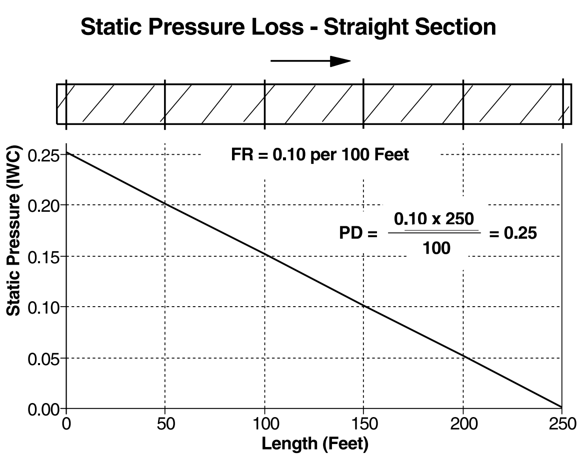


Figure 4-1

- Metal duct friction depends on seam spacing, seam construction and is marginally affected by surface roughness.
- Various types of flexible wire helix duct have coil spacing differences, wall lining differences (rough materials, smoother materials), inner core diameter differences, material flaps, no material flaps, in-ward protruding wires, encapsulated wires, and more.
- Duct board and duct liner is affected by seam spacing, seam construction, fasteners and the roughness of the facing material.

4-3 Required Standard of Care for Installing Flexible Wire Helix Duct

The pressure drop for a section of flexible wire helix duct is significantly affected by the way it is tested and installed. This is the ***Manual D*** standard of care.

- Duct sections cut to length (0% to 4% longer than the straight line span), for a maximum of 4% compression along a straight line.
- Duct centerline relatively straight, no significant sag or snaking (2.5 inches sag per 5 feet of span, or less).
- The radius of a bend (R) or turn shall not be less than the diameter (D) of the airway (R/D shall be 1.0 or greater).
- No crimping or crushing at any point in a duct run.
- Friction charts and duct slide rules shall model friction rate performance in accordance with ***Air Diffusion Council (ADC)Test Code FD 72-R1***, or ASHRAE Standard 120. Use compliant information provided by the product manufacture, if available. Appendix 2 of this manual provides a default friction chart for flexible wire helix duct.
- Installation methods, materials and accessories shall conform to guidance provided by Section 4 and Section 5 of the ***ADC Flexible Duct Performance and Installation Standard, 5th edition, 2010***.
- Appendix 7 provides related guidance.

4-4 Excess Length for Flexible Wire Helix Duct

Excess length is the difference between the fully stretched cut length and the measured, straight-line, entrance-to-exit span length. Excess length increases duct run pressure drop.

- Excess length may not cause compression in the duct section (the run may have sag or bends, but the duct inner core material is fairly taught).
- Excess length may cause compression in the duct section (the run is reasonably straight or has some sag, and the duct inner core material is not taught).
- Excess length may act like a fitting (the sag or bend does not have a long, sweeping radius).
- The default ***Manual D*** procedure is for 0% to 4% excess length (duct length defaults to span length).
- The ***Manual D*** duct sizing calculations are based on the industry-recognized recommendations of 4% or less axial compression (refer to the flexible duct documents listed in Appendix 6).
- Appendix 16 provides a procedure for evaluating the consequences of excess length.

4-5 Panned Joist Space and Stud Space

Panned joist space and stud space are commonly used as airways — usually as part of a return air system. ***Manual D*** does not endorse this practice.

The following comments pertain to potential space pressure problems, air quality problems, duct system efficiency problems and air delivery problems caused by panned construction. If the practitioner chooses to use this type of construction, the practitioner assumes full responsibility for all unintended consequences.

- Leakage is an important issue for all airways, regardless of construction material and detail. Leakage affects, or may affect, indoor air quality, ambient pressure in a room(s) or space(s), duct system efficiency, energy use and operating cost.
- Panned airways that are primarily in an unconditioned space, or which significantly interface with an unconditioned space shall be sealed to prevent significant leakage from an unconditioned space. The recommended minimum sealing requirement for panned airways is 0.09 Cfm of leakage per square foot of airway surface area with the airway pressurized to 0.10 Inches Water Column.
- Local building codes and/or fire codes may prohibit all, or specific types of, structural cavity airways.
- Local duct wall insulation codes and/or a similar good practice standards apply to structural cavity airways.
- Friction rate information for panned airways and stud spaces is not available, but it is reasonable to assume that the roughness index for panned joists and stud spaces is larger than the value for galvanized steel duct.
- Since the surface irregularities for framing and sheathing materials are smaller than the surface irregularities for flexible wire helix duct, it is reasonable to assume that the roughness index for panned joists and stud spaces is smaller than the value for flexible duct that has negligible compression and negligible sag.
- For lack of certified information, assume the friction rate information for duct liner applies to panned airways and stud spaces.
- There is no procedure for adjusting effective airway size if there are obstructions in the airway (bracing, piping, wiring or anything that reduces the size of the framed or panned airway).
- There is no procedure for adjusting effective airway size if the airway size of entrance and exit openings cut into structural framing is less than the framed or panned airway size.

4-6 Fitting Pressure Loss

Duct fittings fall in two categories — those that have different entering and leaving air flow velocities and those that have a constant "flow-through" velocity. For scenario one, it is not correct to use an equivalent length value to simulate the pressure loss across the fitting. For scenario two, fitting pressure drop can be simulated by an equivalent length value.

Velocity Change Across the Fitting

When velocities at the entrance and exit of the fitting are different, some static pressure is used to increase air velocity or some static pressure is recovered if velocity decreases. Some examples of this type of fitting include:

- Transitions between straight duct sections of different airway sizes.
- Elbows with different entrance and exit areas.
- Tees with different entrance and exit velocities.
- Inlets and outlets.

For these fittings, use of an equivalent length value is not precisely correct because this model does not account for static pressure difference caused by velocity change. However, for residential designs, this technicality may be ignored because air flow velocities, velocity changes and subsequent pressure conversions are relatively small.

Constant Velocity Fittings

There is no velocity pressure effect when air flow velocities at the entrance and exit of the fitting are equal. For these fittings, pressure drop is entirely due to friction and aerodynamic turbulence, so an equivalent length value does simulate fitting pressure drop.

Equivalent Length Information

As explained in Section 1-7, duct run resistance depends on the total effective length of the run. This length is easily calculated by summing straight section lengths and fitting equivalent lengths for a given duct run.

- Appendix 3 provides a comprehensive list of equivalent length values for fittings that are commonly used for residential duct systems.
- These equivalent length values are based on a specific reference velocity and a specific friction rate.
- These defaults are the "worst case" scenario (900 Fpm velocity and a 0.08 friction rate).
- Adjustment for other velocities and/or friction rates may be made, but this refinement is not necessary because ***Manual D*** equivalent length values are conservative.
- Overestimating the air flow resistance for a duct run is not a problem when balancing dampers are used to compensate for the error.

Fitting and Duct Run Pressure Drop Calculations

Usually, there is no need to calculate the pressure drop for a single fitting. Normally, the practitioner is interested in the pressure drop for an entire duct run. As explained in Section 1-6, the pressure drop (PD) across a single fitting, or an entire duct run, depends on the fitting equivalent length (EL), or path total effective length (TEL), and a friction rate (FR) value. This equation summarizes this relationship.

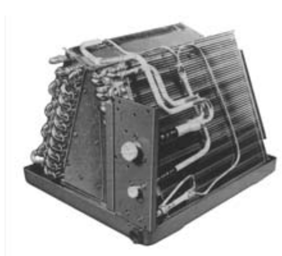
$$PD\ (IWC) = \frac{FR \times EL}{100} \quad or \quad \frac{FR \times TEL}{100}$$

4-7 Pressure Drop for Air-Side Components

When equipment or a device is installed in a duct system airway, it produces resistance to the air flow, so there is a pressure drop across the component. Examples of items that are commonly installed in residential duct systems include filters (low to high effectiveness, new, and dirty), refrigerant coils (dry, wet, clean to dirty), hot water and electric heating coils (clean to dirty), air flow control dampers (open-close or modulating), and humidifiers.

Information about the pressure drop produced by a particular component (when new) is obtained from engineering data published by the equipment manufacturer. Examples of the pressure drop information for a refrigerant coil, an electronic filter and an electric resistance heating coil are provided by Figure 4-2.

Figure 4-2 shows that the pressure drop (Px) across a piece of air-side equipment increases exponentially as air flow (Cfm) through the equipment increases linearly. This behavior is summarized by this equation:



Coil Resistance (IWC)		
Cfm	Dry	Wet
1,000	0.11	0.18
1,200	0.15	0.26
1,400	0.22	0.35
1,600	0.28	0.46

Electronic Filter Resistance	
Cfm	IWC
1,000	0.06
1,200	0.08
1,400	0.12
1,600	0.15

Heater Resistance	
Cfm	IWC
1,000	0.09
1,200	0.13
1,400	0.18
1,600	0.23

Figure 4-2

$$Px\ (IWC) = Resistance_x = P_1 \times (Cfm_x / Cfm_1)^2$$

Use this equation when pressure drop and Cfm are measured. For example, if the measured pressure drop across a wet refrigerant coil is 0.26 IWC when 1,200 Cfm flows through the coil, the estimated pressure drop is 0.46 IWC when the flow rate is 1,600 Cfm.

$$P_x\ (IWC) = 0.46 = 0.26 \times (1{,}600 / 1{,}200)^2$$

4-8 Supply Outlet, Return Grille and Hand Damper Pressure Drops

The pressure drop produced by supply air grilles, registers and diffusers, return air grilles and manual balancing dampers (wide open) is small, typically 0.03 IWC or less. For ***Manual D***, 0.03 IWC is the default value for any type of air distribution device (grille, register, diffuser, etc.). If more accurate information is desired, it is provided by manufacturer's performance data, but this refinement generally does not have a significant effect on airway sizes.

4-9 Low-Resistance Return Path

An engineered, low-resistance return path shall be provided for every room or space that receives supply air. These methods may be used for rooms and spaces that have a privacy door installed in an interior partition.

- The path from a return grille in an particular room or space to the return-side of the blower may be through a dedicated return air duct.
- An isolated room or space may have a transfer duct to a space that has a central return.
- A privacy door or partition wall may have an opening fitted with two return grilles so air flows from an isolated room or space to a central return. Provide a sealed sleeve for wall openings.
- Appendix 3, Fitting Group 14 provides guidance for designing return air transfer paths.

A door undercut does not provide a reasonable solution to the return air problem because the required gap is objectionably large. If a door undercut is used as a return path, the gap shall not be less than the Table A1-2 value.

- Adequate door cuts create appearances issues.
- Adequate door cuts create privacy issues (a significant amount of noise is transmitted though a small crack, a door cut proves negligible attenuation).
- Air flow under doors soils carpets.

4-10 Grille Air Velocity

The air velocity through return grilles shall not exceed 350 Fpm, based on the area enclosed by the grille frame (i.e., the nominal size listed in product performance data). For example, the maximum return cfm for 10 Inch x 8 Inch return grill is 195 Cfm.

Maximum Return Grille Cfm = 350 x Frame Area (SqFt)

Maximum Cfm = 350 x (10 x 8) / 144 = 194

Minimum Door Cut Height for Return Air

Cfm Under Door	Door Width (Inches)						
	24	30	36	42	48	54	60
	Clearance (Inches) to Floor or Top of Carpet						
100	2.0	1.6	1.3	1.1	1.0	0.9	0.8
200	4.0	3.2	2.7	2.3	2.0	1.8	1.6
300	6.0	4.8	4.0	3.4	3.0	2.7	2.4
400	8.0	6.4	5.3	4.6	4.0	3.6	3.2
500	10.0	8.0	6.7	5.7	5.0	4.4	4.0
600	12.0	9.6	8.0	6.9	6.0	5.3	4.8
700	14.0	11.2	9.3	8.0	7.0	6.2	5.6
800	16.0	12.8	10.7	9.1	8.0	7.1	6.4
900	18.0	14.4	12.0	10.3	9.0	8.0	7.2
1,000	20.0	16.0	13.3	11.4	10.0	8.9	8.0
1,200	24.0	19.2	16.0	13.7	12.0	10.7	9.6
1,400	28.0	22.4	18.7	16.0	14.0	12.4	11.2
1,600	32.0	25.6	21.3	18.3	16.0	14.2	12.8

Copy of Table A1-2

Section 5

Air Distribution System Design

Duct airway sizing is one step in the design process. Other mandatory tasks precede the ***Manual D*** procedure. Figure 5-1 summarizes this relationship. Commentary regarding various aspects of the work is provided here:

5-1 System Selection

Everything begins with a concept. Decisions regarding the type of heating and cooling equipment and type of air distribution system shall be made prior to performing design calculations.

- Scrutinize codes and regulations.
- Consider the local climate, the architectural features of the dwelling, the seasonal and hourly characteristics of the heating and cooling loads for the various rooms and spaces.
- Consider zoning requirements and select the type or types of primary heating and cooling equipment.
- Consider the number of supply and return air openings that will be required, the location of primary equipment, the space that is available for the duct runs, the location of the supply outlets, the location of the return grilles.
- Select supply air and return air systems that are compatible with the heating and cooling equipment, the structural features of the dwelling and project cost constraints.
- Consider duct system efficiency (conduction and leakage losses), and airborne noise.
- Consider the advantages and disadvantages of using various types of duct materials and construction techniques.

The supply air system may be a trunk and branch system, a radial system, a perimeter loop system or a plenum system. The return air system might feature a single central return with transfer paths, multiple returns (with transfer paths, as required), or a return in every room. Appendix 8 provides a comprehensive discussion of residential air distribution systems.

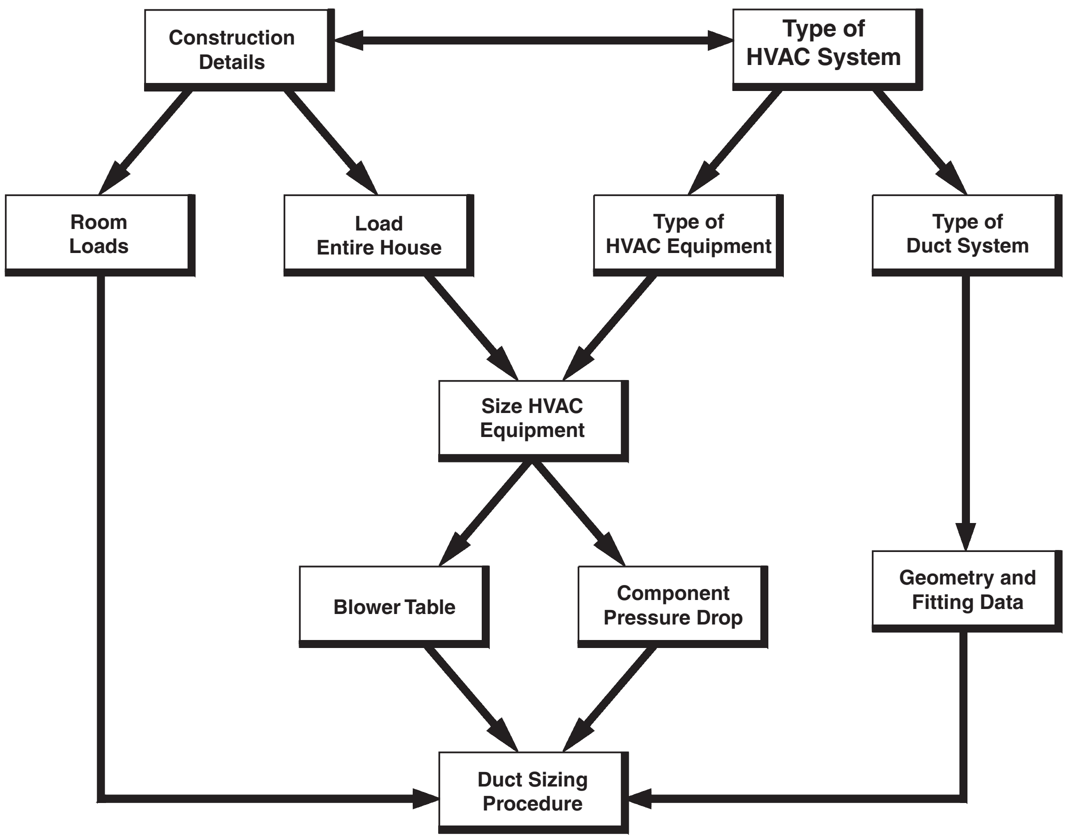


Figure 5-1

5-2 Load Calculations

Duct system design work shall be based on information provided by the ***Manual J*** load calculation procedure. Calculations for block heating-cooling loads and room heating-cooling loads are required.

- If one heating-cooling unit is used, a set of block loads for the entire dwelling is required.
- If multiple units are used, a block load calculation for the space served by each unit is required.
- Per MJ8, Figure A11-1: The standard procedure is used to calculate average room and space loads for single-zone systems; and the optional procedure is used to calculate peak zone, room and space loads for zoned systems.
- Block heating-cooling loads are used to select and size the primary heating and cooling equipment.
- The heating-cooling loads for each conditioned room or space determine the room or space air flow (Cfm) requirement.

5-3 Equipment Sizing

Heating-cooling equipment sizing, per ***Manual S*** procedures shall precede duct system design work because manufacturer's blower performance data is essential to ***Manual D*** procedures.

- Blower data is provided with the manufacturer’s equipment performance data, usually in tabular form, or as a graph.
- In some cases, blower table data has to be adjusted for the pressure drop produced by components and devices that were not in place when the blower was tested. This adjustment is made by subtracting the component or device pressure drop from the external static pressure listed in the blower table (refer to Section 1-10).
- Information about the pressure drop produced by a particular component or device (filter, refrigerant coil, electric resistance heater, humidifier, for example) is provided by equipment manufacturer’s performance data.

5-4 Fittings

Duct fittings should be efficient (fitting equivalent length typically produces a much larger path pressure drop than the straight run length of the associated path).

- Appendix 3 shall be used to compare fitting equivalent length values. Use the fittings that have smaller equivalent length values.
- The airway sizing calculations are invalid if the fittings that are actually installed in the duct system are not identical to the fittings used for ***Manual D*** calculations (design, then verify).

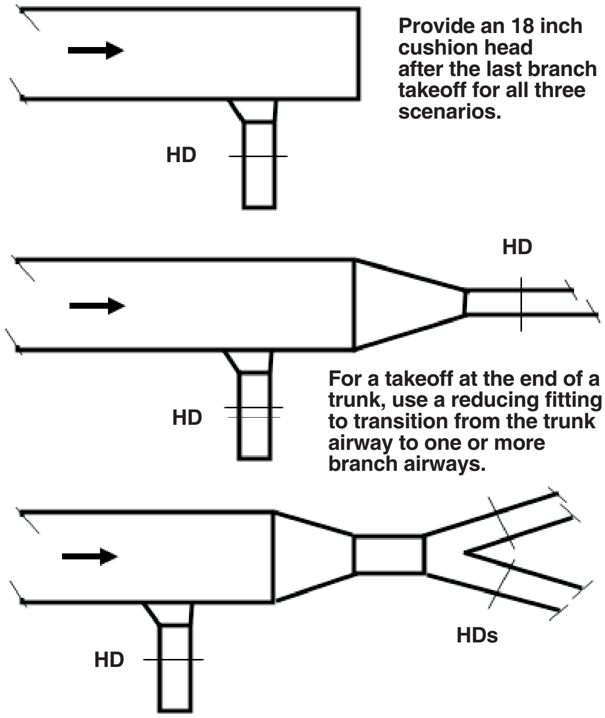


Figure 5-2

5-5 Geometry

The lengths of straight duct sections and the equivalent lengths of the fittings shall be used to calculate the effective length of duct runs. Therefore, a scale drawing or rough sketch that includes dimensional information is required. This drawing (or sketch) should show the location of the:

- Air handling equipment.
- Supply air outlets, with the heating load, cooling load and design Cfm for each supply outlet.
- Return air grilles, with the Cfm for each return.
- All duct runs, with the lengths of the straight sections.
- Fittings, with fitting identification numbers and corresponding equivalent length values (see Appendix 3).

5-6 Takeoff at End of Supply Trunk

The closing plate at the end of a trunk duct causes a turbulent zone in the trunk airway just upstream from the plate. A branch takeoff fitting shall not be installed in this zone. Figure 5-2 shows that the centerline of a branch takeoff shall be at least 18 inches upstream from a trunk end plate, or trunk reducing fitting. Figure 5-2 also shows an acceptable way to install one or two branch runs at the end of a trunk duct.

5-7 Duct Material and Duct Fabrication

For new construction, specify duct materials and fabrication methods before using the ***Manual D*** procedure. For existing construction, inspect the job site and collect information about duct materials and fabrication methods before using the ***Manual D*** procedure.

- Fabrication methods and materials and retrofit methods and materials shall conform to industry standards and good practice guidelines (see Appendix 6 and Appendix 7).
- The friction chart or duct slide rule used to size duct airways shall be representative of the performance of the duct material used to fabricate the duct run. This information should be obtained from the manufacturer of the duct material (if available).
- The airway sizing calculations may be invalid if the methods and materials used to fabricate the actual duct system are not the same as the methods and materials used for ***Manual D*** calculations (design, then verify).

5-8 Perform the Duct Sizing Calculations

Manual D duct sizing calculations are based on the heating-cooling loads for each room or space, the blower data, the pressure drop values for various types of air-side components and effective length information. The details of the ***Manual D*** duct sizing procedure are presented in the next section. Sections 7, 8, 10 , 11, 12 and 13 provide examples that show how to apply the ***Manual D*** procedure to various types of trunk and branch systems, radial systems, flexible duct systems and variable Cfm systems.

Section 6

Duct Sizing Calculations

If noise was not a consideration, the minimum airway size for every duct run could be based on one friction rate value that depends on the available static pressure, the total effective length of the longest circulation path and Cfm values at various points in the duct system. However, noise is an important consideration, so airway sizes based on this friction rate are tentative.

The practitioner shall verify that all airway sizes are compatible with the velocity limits for the supply and return-sides of the duct system. If a velocity limit is exceeded, the offending section of duct shall be resized. In these cases, the final airway size is based on the maximum allowable air velocity and the Cfm that flows through the section of duct.

Worksheets and a reference chart are used for airway sizing calculations. This section reviews these procedures and calculations. This formalizes the procedure that was introduced by Section 1.

6-1 Basis for the Sizing Procedure

Calculations for minimum airway size shall be based on blower performance data (including relevant footnotes) and air-side component pressure drop data provided by the equipment manufacturer (as applicable). This information correlates the design value for system air flow (Cfm) with the design value for available static pressure.

- The practitioner shall verify that the pressure drop for the critical circulation path (the longest combination of a supply path and its related return path) does not exceed the available static pressure.
- The practitioner shall verify that the air velocity through any section of duct does not exceed the ***Manual D*** limit.
- The practitioner shall verify that the discharge or intake velocity of air leaving or approaching grilles, registers and diffusers does not exceed the ***Manual D*** limit.
- See Table A1-1 for conservative velocities and maximum velocities.

6-2 Balancing Damper Function

A duct system could be self-balancing if duct sections are sized to ensure the pressure drop for each circulation path is exactly equal to the available static pressure. In this case, these rules apply:

- Trunk ducts common to multiple circulation paths must be sized for the path that has the longest effective length.
- Since the trunk sizes will be too large for shorter circulation paths, runout ducts must be sized to compensate for trunk sections that do not provide enough air flow resistance.
- Room supply air Cfm is typically different for heating and cooling, so room Cfm design values may be based on the average of the heating Cfm and cooling Cfm (see Section 6-23).

However, a self balancing design is not practical and the system could be noisy. These comments apply:

- The calculations become more complex and time consuming.
- Non-standard airway sizes are required to obtain the desired pressure drops for self balancing.
- Air velocity at points in the shorter circulation paths could exceed the maximum limit.
- Room supply air Cfm is typically different for heating and cooling, so seasonal performance is compromised by a self-balancing design.

The problems created by the desire for a self-balancing design cannot be reconciled. These problems are not an issue if a balancing damper is installed in each runout duct.

- For airway sizing, design Cfm values are maximum values for heating or cooling.
- Duct section airway sizes are compatible with the maximum air flow requirement.
- The size of any duct section is a compromise between the standard airway size that provides the desired amount of resistance and the standard airway size required for a quiet system.
- For constant Cfm systems, air flow should be balanced for the heating season and re-balanced for the cooling season. If seasonal balancing is pragmatically unacceptable, air flow values for balancing may be based on the average of the heating Cfm and cooling Cfm (see Section 6-23).
- For zone damper systems, air flow values for rough manual-damper balancing are the maximum of the heating Cfm or cooling Cfm. System air flow (design blower Cfm) shall be in the range

of ± 10% to ± 20% when all automatic control dampers are wide open).

- For air zoning systems, thermostatically controlled dampers fine tune air flow for season change and variation in hourly load.
- Appendix 9, ***Manual Zr*** provide guidance for balancing zone damper systems.

6-3 Balancing Damper Requirement

For ***Manual D***, the recommended standard of care requires balancing dampers at critical points in the duct system. The location of the dampers depends on the complexity of the system geometry.

- For the supply-side of the system, install branch run dampers near the trunk duct take-off. These dampers may be used for gross balancing and final balancing.
- If there are main supply air trunks and secondary supply air trunks, dampers in the secondary trunks are useful for gross balancing.
- For the return-side of the system, install branch run dampers near the trunk entrance fitting. These dampers may be used for gross balancing and final balancing.
- If there are main return air trunks and secondary return air trunks, dampers in the secondary trunks are useful for gross balancing.
- Balancing dampers shall not be installed in the turbulent wake of an upstream fitting or obstruction (install dampers where the air flow is well-ordered and uniform).
- To avoid or reduce undesirable noise, balancing dampers should be installed as far from a supply air outlet or return air grille as possible.
- Balancing dampers shall have a locking device.
- Aerodynamic fittings and soft airways (duct liner or duct board) located downstream from dampers tend to attenuate damper generated noise.

Access to a balancing damper is a problem if the duct run space is enclosed by construction materials (a basement ceiling, for example). When this is the case, practitioners tend to not install balancing dampers in affected duct runs. ***Manual D*** neither endorses nor condemns this practice.

The following comments pertain to potential air balancing problems, comfort problems and noise problems caused by missing or inappropriately placed balancing dampers. If the practitioner chooses to use this type of construction, the practitioner assumes full responsibility for all unintended consequences.

- Room or space comfort will not be adequate if room air flow is excessive or deficient.
- Single blade dampers at outlets and returns are inadequate balancing devices. Use opposed blade dampers that have a gang of blades.
- Objectionable noise may be produced by a throttling device located near a supply air out or return grille (access and adjustment may still be a problem).
- Some of the operating range of a modulating zone damper may be used to limit maximum air flow (less range for controlling space comfort).
- Balancing dampers could be installed at critical points, then roughly adjusted and locked before being isolated by structural panels.
- If all stakeholders agree (after knowing the consequences of not having balancing dampers), structural panels should have covered access holes.

6-4 Design Value for Blower Cfm

The design value for blower Cfm is determined when manufacturer's expanded performance data is used to select equipment per ***Manual S*** procedures. The blower Cfm used for ***Manual D*** calculations depends on the type of blower.

- There may be a blower Cfm for heating and a blower Cfm for cooling, or one Cfm may be used for heating and cooling.
- For an operating point blower, the design value for blower Cfm is the Cfm from the ***Manual S*** procedure. If this Cfm value is not explicitly listed in the blower table, this value and two listed blower Cfm values are used to produce an interpolated value for static pressure.
- For an operating range blower, the design value for blower Cfm is the Cfm from the ***Manual S*** procedure. If this Cfm value is not explicitly listed in the blower table, select the next highest value (blower Cfm set point) listed in the blower table. This Cfm set point determines blower pressure range (minimum and maximum static pressure values).

6-5 Available Static Pressure

It is absolutely essential for the practitioner to verify how much static pressure is available to move the air through the circulation path that produces the most airflow resistance. Steps 1, 2 and 3 on the Friction Rate Worksheet shall be used for this purpose (see Figure 6-1, next page).

Friction Rate Worksheet

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = ______ IWC   Cfm = __________

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil ______
Electric resistance heating coil ______
Hot water coil ______
Heat exchanger ______
Low efficiency filter ______
High or mid-efficiency filter ______
Electronic filter ______
Humidifier, UV lights, other ______
Supply outlet ______
Return grille ______
Balancing damper ______
Zone damper (full open) ______

Total component losses (CPL) ______ IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (______ - _______) = _______ IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (_______ + _______) = _______ Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = ______ IWC/100

See Appendix 18 for blank form.

$FR = \frac{ASP \times 100}{TEL}$

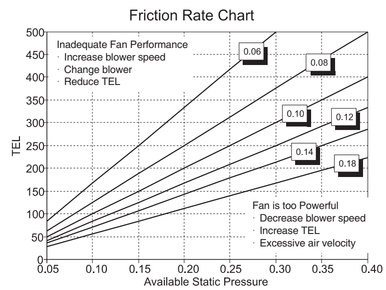


Figure 6-1

Multi-Speed (PSC) Blower Data			
	External Static (IWC) vs. Speed		
Cfm	**High**	**Med**	**Low**
1,200			0.45
1,250		0.49	0.30
1,300		0.37	0.08
1,350		0.25	
1,400	0.62	0.14	
1,450	0.55	0.04	
1,500	0.47		
1,550	0.39		
1,600	0.31		

1) Tested with wet coil and filter in place.
2) Subtract pressure drop for a resistance heating coil.

Figure 6-2

Step-1 - Blower Cfm and External Static Pressure

Use the manufacturer's blower data to determine how much external static pressure (ESP) is available when the blower delivers the design Cfm.

- Duct airways are sized for the maximum system air flow rate. When equipment capacity controls and/or zoning controls adjust blower performance for changing circumstances, blower Cfm at part-load is not relevant to the airway sizing procedure.
- Determine the system design values for heating Cfm and cooling Cfm, then use the larger of the two values for airway sizing.
- The values for the design Cfm for heating and cooling are obtained during the equipment selection process. As explained in ***Manual S***, heating and cooling equipment shall be selected and sized to satisfy the ***Manual J*** loads when the dwelling is subjected to ***Manual J*** design conditions.
- Blower Cfm values also must be compatible with the OEM operating limits for entering and leaving air temperature, and furnace heat exchanger temperature rise.
- Per ***Manual Zr***, Section A8-2: The design value for blower Cfm is determined when the heating-cooling equipment is selected and sized per ***Manual S*** procedures applied to the block load for the space served by the equipment. The design Cfm for zone damper systems is the same as the design Cfm for single-zone systems because the ***Manual J*** block load is the same for both types of systems.

Step-1 for an Operating Point Blower

If the equipment has a multi-speed, operating point blower (see Section 3-2), it is prudent (but not absolutely necessary) to base the duct sizing calculations on medium speed. This provides the ability to adjust the blower performance after the equipment has been installed.

For example, if 1,250 Cfm is required for an application, Figure 6-2 indicates that, at medium speed, the blower will deliver 1,250 Cfm when it operates against 0.49 IWC of system resistance. Therefore, the operating point for Step-1 of the Friction Rate Worksheet is 1,250 Cfm and 0.49 IWC of external static pressure.

Step-1 for an Operating Range Blower

An operating range blower (see Section 3-3) delivers a selected Cfm value over a range of external static pressures. In this case, speed taps or switches determine blower Cfm and this Cfm setting determines the blower pressure range, per the OEM's blower table.

The design value for external static pressure must fall within the blower table range, but this value should be significantly lower than the maximum pressure value listed in the blower table (the issues are blower KW load and KWH energy use, blower motor abuse, and to allow a margin for design/installation error for air distribution). Therefore, use the system design Cfm value and 70% (0.70 factor) of the maximum external static pressure value from the OEM's blower table as the operating point for Step-1 of the Friction Rate Worksheet.

For example, for the 1,225 Cfm setting, the Figure 3-7 blower table shows an external static pressure range of 0.0 to 1.0 IWC. Per the preceding guidance, the external static pressure value for Step-1 of the Friction Rate Worksheet is 0.70 IWC. (In regard to the 1,225 Cfm setting, the data shows this may be somewhat more or less than 1,225 Cfm for very low and high static pressure values, but the difference is small, and may not be relevant to the static pressure behavior of the installed system, so this nuance may be ignored.)

Step-2 Component Pressure Loss

Evaluate the component pressure loss (CPL) for air-side items that will be installed in the critical circulation path (other than those that were in place when the equipment manufacturer produced the blower table). This is important because pressure dissipated by cabinet components that are not relevant to the OEM's blower table, and pressure dissipated by external equipment and devices, is not available to force air through the straight duct runs and fittings.

Examples of pressure dissipating components that may not be relevant to published blower performance data include refrigerant coils, electric resistance heating coils, heat exchanges, hot water coils, filters and humidifiers. Refer to the footnotes below the manufacturer's blower table for information about the items that were in place when the blower was tested, and/or not relevant to the blower data.

For example, referring to Figure 6-2, the notes at the bottom of this heat pump blower table indicate that external static pressure values are discounted for the pressure dissipated by a wet refrigerant coil and a standard low efficiency air filter. These notes also advise that there is no adjustment for the resistance produced by a supplemental electrical resistance heating coil.

Bower Table Notes

Blower table footnotes may not exist, or may not provide adequate information for blower table use. For example, Figure 3-7 shows an operating range blower table that says 1.0 IWC is the maximum external static pressure value, but there is no footnote guidance to explain what this actually means.

If footnote information is missing, or incomplete, search the OEM's engineering literature for the missing information, or personally ask the OEM for the information. For the Figure 3-7 example, it turns out that the blower data is for no cabinet components other than the blower, and the equipment mounting racks. So, if the pressure drop for a wet cooling coil, an electric heating coil and a standard filter add up to something like 0.50 IWC (or more), the maximum external static pressure for moving air through external components, duct runs and fittings is only 0.50 IWC, not 1.00 IWC.

Heater Resistance	
Cfm	**IWC**
1,000	0.09
1,200	0.13
1,400	0.18
1,600	0.25
1,800	0.34

1,250

0.14

Figure 6-3

Component Pressure Drop

Exact values for component pressure drops are normally provided by equipment manufacturer's engineering information. As noted above, component pressure drop for cabinet components that are not relevant to the blower table data, and external components specified by the practitioner shall be subtracted from the external static pressure obtained from the blower table.

For example, Figure 6-3 provides pressure drop data for an electric resistance heating coil. This table shows that the coil will dissipate 0.14 IWC of pressure (approximate) when 1,250 Cfm flows through the coil. Figure 4-2 provides examples of pressure drop data for a cooling and a filter.

Component Substitution

Sometimes an optional component is substituted for the similar standard component that was in place when the blower was tested. In this case, the difference between the pressure drops produced by the two components shall be subtracted from (or added to) the available static pressure. For example:

- A high efficiency media filter (a pleated filter, for example) usually has a much larger pressure drop than a standard filter.
- An electronic filter protected by a pre-filter may have a larger pressure drop than a standard filter.
- If a standard filter is used as a pre-filter for an electrostatic grid, the additional pressure loss is produced by the electrostatic grid.

Air Distribution and Balancing Devices

Air distribution devices such as supply air grilles, diffusers and registers, return air grilles and filter-grilles, and manual balancing dampers, dissipate pressure. The exact value for the pressure drop across these devices is normally provided by manufacturer's performance data. However, it is an acceptable and typically conservative practice to use a default value (0.03 IWC) for the pressure drop across any of these devices. For example, the total pressure loss for one supply outlet, one return and a balancing damper is 0.09 IWC.

Total Component Pressure Loss

Sum the pressure drop values for the items listed for Step-2 of the Friction Rate Worksheet.

Step-3 Available Static Pressure

Determine how much pressure is available to move air through straight duct sections and fittings. To calculate the available static pressure (ASP) value, subtract the total component pressure loss (CPL) from the external static pressure (ESP).

For example, Figure 6-3 shows a blower operating at 1,250 Cfm and 0.49 IWC of external static pressure. The pressure loss for an auxiliary heating coil is 0.14 IWC (per Figure 6-3), plus the loss for one supply outlet (0.03 IWC), one return (0.03 IWC), and one open balancing damper (0.03). This adds up to 0.23 IWC. This is subtracted from the blower table pressure, so the pressure that is available to move air though the straight runs and fittings in the supply-return path that produces the most resistance to airflow (the critical circulation path) is 0.26 IWC.

External static pressure = 0.49 IWC
Component pressure drop = 0.14 + 0.03 + 0.03 + 0.03
= 0.23 IWC
Available pressure for air way sizing = 0.49 - 0.23
= 0.26 IWC

For most applications, the available static pressure (ASP) for straight runs and fittings should be about 0.15 to 0.35 IWC. Values less than 0.15 IWC may not be acceptable unless the total effective length of the critical circulation path is relatively short. Values greater than 0.35 IWC may be too large unless the total effective length of the critical circulation path is unusually long. If the available static pressure is deficient or excessive, the practitioner may use one or more of these options:

- Use more efficient fittings to reduce the total effective length of the critical circulation path (see Section 6-6).
- Use one or more air-path components that have a lower pressure drop. Eliminate an air-path component that may be desired, but not absolutely necessary.
- For a PSC blower, use a different blower speed to determine the Step-1 operating point.
- For an operating point blower, increase the 0.70 limit factor for Step-1 external static pressure. However, values of 0.85 and higher are not recommended, use some other solution.
- If a blower adjustment this does not produce a usable operating point for Step-1, find an alternative product that has a blower that will provide a suitable operating point.

Effective Length Worksheet

Element	Supply Run ID Number				Element	Return Run ID Number			
Trunk Length					Trunk Length				
Trunk Length					Trunk Length				
Trunk Length					Trunk Length				
Runout Length					Runout Length				
Group 1					Group 5				
Group 2					Group 6				
Group 3					Group 7				
Group 4					Group 8				
Group 8					Group 10				
Group 9					Group 11				
Group 11					Group 12				
Group 12					Group 13				
Group 13					Other				
Other					Other				
Total Length					Total Length				

See Appendix 18 for blank form.

Figure 6-4

6-6 Total Effective Length

The Friction Worksheet requires a value for the total effective length of the critical circulation path. The total effective length of a particular duct run is calculated by summing the straight lengths of all duct sections and the fitting equivalent lengths for the run. This calculation is made on the effective length work sheet (see Figure 6-4).

- The length of a section of ridged duct is measured from entrance to exit.
- The base value for a length of a section of flexible duct is a straight line measurement from entrance to exit. Apply the relevant Table A16-1 and Table A16-2 equivalent length multiplier to the base-length value to find the effective length of a flexible wire-helix duct run.
- Duct length measurements and calculations are rounded to the nearest foot.
- Fitting group numbers and equivalent length values are extracted from the Appendix 3

The length of a circulation path equals the sum of a supply-path length and an associated return-path length. An estimate for the effective length of every circulation path is not required. The goal is to find the total effective length of the longest circulation path (e.g., the critical circulation path.)

Sometimes the longest runs can be identified by inspection, but do not jump to conclusions. Depending on the equivalent length of the fittings, the run that has the longest effective length may not be the run that has the longest measured length. If there is any doubt about which run is the longest, check the effective length of each likely candidate.

- Fitting equivalent lengths have more influence on total effective length than straight run lengths.
- Improperly installed flexible wire helix duct can produce a large value for duct run length (see Table A16-1 and Table A16-2).
- Runs with inefficient fittings are prime candidate for a longest run investigation.
- A run that has a branch takeoff fitting near the air handler may be the longest run (see Appendix 3, Group 2 and Group 6).
- If inefficient fittings are used for return runs, it is possible for the effective length of the longest return run to be larger than the effective length of the longest supply run.
- If there is any doubt about selecting longest-run candidates, calculate the total effective length of all runs and look for the longest circulation path (e.g., the longest combination of a supply-path length and the associated return-path length).
- Effective length calculations do not have to be perfect. A ten percent error in total effective length will not produce a significant change in the final airway sizes.

After the total effective lengths of the critical supply path and the associated return path are determined, enter these length values in the spaces provided by Step-4 of the Friction Rate Worksheet. Also enter the sum of these values on the worksheet.

6-7 Design Friction Rate Value

The purpose of the Friction Worksheet is to determine the design friction rate value (IWC per 100 feet of length) for duct airway sizing. The design friction rate (FR) value depends on the available static pressure (ASP) value and the total effective length (TEL) value. The friction rate value is produced by the equation below, or it may read from the Friction Rate Chart at the bottom of the Friction Rate Worksheet (see Figure 6-5).

$$FR = \frac{ASP \ x \ 100}{TEL} \quad (\textbf{\textit{IWC}} / \textbf{100} \ \textbf{\textit{Ft}})$$

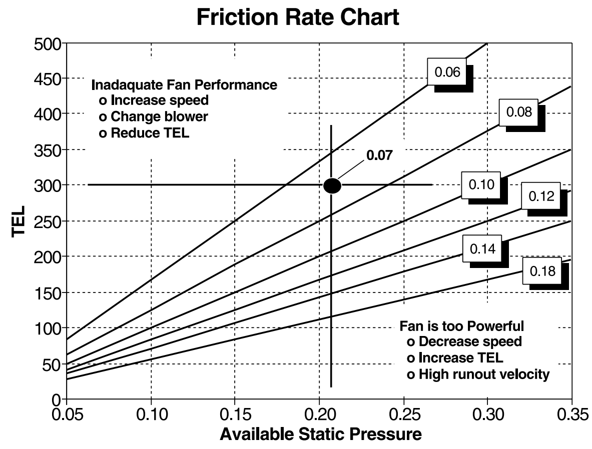


Figure 6-5

Do Not Use a Default Friction Rate

Section 6-7 and Figure 6-5 invalidate guidance printed on some duct sizing slide rules, and in some technical literature. This may be a statement, note, arrow or font-face emphasis that suggests that duct systems are designed for a default friction rate, typically 0.10 IWC per 100 Ft of duct. The fallacy of this recommendation is rooted in the fact the design friction rate value is not arbitrary.

As demonstrated on these pages, the friction rate value depends on the available static pressure and the total effective length of the longest circulation path. For example, if the length of the critical path is 200 feet, a design friction rate value of 0.10 IWC per 100 Ft of duct is only correct when the available static pressure is 0.20 IWC,, per this equation.

$$FR = \frac{ASP \ x \ 100}{TEL} \quad (\textbf{\textit{IWC}} / \textbf{100} \ \textbf{\textit{Ft}})$$

The second objection to use of a default friction rate is related to noise. Air velocity may exceed recommended limits if duct sizes are based on an arbitrary friction rate. For example, at 1,200 Cfm, the velocity in a trunk duct is excessive (1,000 Fpm) if the airway size is based on a 0.10 IWC per 100 Ft of duct friction rate.

After the design friction rate is determined, enter the value on the Friction Rate Worksheet (Step-5). This friction rate and the Cfm values at various points in the duct system are used to size duct airways.

Friction Rate Chart

The friction rate chart at the bottom of the Friction Rate Worksheet features a set of friction rate lines that slope from the lower left corner of the chart to the upper right corner. These lines provide a graphical solution for the friction rate equation.

To use the friction rate chart, draw a vertical line that represents the available static pressure (ASP) value and a horizontal line that represents the total effective length (TEL) value. The friction rate design value (FR) is at the intersection of these two lines.

If the FR value does not fall on a friction rate line, interpolate (eyeball method) between two friction rate values. For example, Figure 6-5 shows that a 0.07 friction rate correlates with 300 feet of effective length and 0.21 IWC of available static pressure.

The advantage of the chart method is that it immediately shows incompatibilities between available pressure and total effective length. Figure 6-5 shows that more blower power (increase wheel Rpm) is required if the solution point is above the 0.06 friction rate line, and less blower power (reduce wheel Rpm) is required if the solution point is below the 0.18 friction rate line.

Another benefit is that the chart method flags excessive air velocity in runout ducts (see Table A1-1 for velocity limits). For the maximum friction rate on the Friction Rate Chart (0.18 IWC/100 Ft), air velocity depends on Cfm and the type of duct material.

- For rigid supply duct, air velocity will be 900 Fpm or less if the flow rate is 300 Cfm or less.
- For rigid return duct, air velocity will be 700 Fpm or less if the flow rate is 100 Cfm or less.
- For wire helix flexible duct, air velocity will be 700 Fpm or less if the flow rate is 300 Cfm or less.

In other words, blower performance is compatible with total effective length if the solution point falls within the wedge produced by the 0.06 friction rate line and the 0.18 friction rate line. A blower adjustment, and/or component pressure drop change, and/or effective length change is required if the point of intersection does not fall in this wedge.

- Increase or decrease PSC blower wheel speed or the design static pressure value for an ECM blower (e.g., use more or less static pressure for Step-1 on the Friction Rate Worksheet), then re-evaluate the design friction rate value per Steps 2 through 5 on the Friction Rate Worksheet.)
- Revisit Step-3 of the Friction Rate Worksheet and use components that produce a smaller or larger value for total pressure drop.
- Use more efficient fittings to reduce the total effective length of the critical circulation path and re-evaluate Steps 4 and 5 on the Friction Rate Worksheet.
- If use of one or more of these options does not produce the necessary result, find some other product that has suitable blower performance.

6-8 Sensible Heating and Cooling Loads

Blower Cfm, room Cfm and space Cfm values depend on the heating loads and sensible cooling loads. There is only one type of ***Manual J*** heating load. The sensible cooling load may be an average daily load plus an AED excursion load, or an average daily load plus a full excursion load.

- Use ***Manual J*** (Eighth Edition, unabridged Version 2.10 (or later) to evaluate heating-cooling loads. Read ***Manual J*** Section 11, opening comments through Section 11-4, and study Figure A11-1.
- For any type of comfort system, cooling equipment size, and the corresponding design blower Cfm value, are based on block cooling load, as determined by the standard ***Manual J*** procedure.
- Constant Cfm systems do not have the ability to respond to hourly changes in room load. Therefore, the standard ***Manual J*** procedure determines room and local space cooling loads. This procedure adds an AED excursion load to the average daily fenestration load.
- Zone damper systems have the ability to adjust supply air Cfm to track momentary changes in room or space load. Therefore the optional ***Manual J*** procedure determines room and local space cooling loads. This procedure adds the full excursion load to the average daily fenestration load.

  For ***Manual J***, Form J1, Lines 6A and 6B hold values for the average fenestration load for windows and skylights, and Line20 holds the AED excursion value for the standard procedure, or the full excursion value for the optional (peak load) procedure.
- ***Manual J*** insists that no safety factors or allowances be applied to any part of the procedure or to the total values for heating load, sensible cooling load and latent cooling load.

6-9 Summing Room Loads

For ***Manual J*** load calculations, the sum of the room and local space heating loads for the block load space will be equal to the heating load for the block load space. For sensible cooling, the sum of the room and local space cooling loads for the block load space will, normally, not be equal to the sensible cooling load for the block load space.

Summing Room Heating Loads

There are no time of day issues for ***Manual J*** heating loads. Therefore, the total heating load for the block load space, and for each room or local space associated with the block load, equals the sum of the building envelope loads (line 14 on MJ8 Form J1), plus the duct load (line 15 on Form J1). Since there are no time of day issues, line 14 plus line 15 for the block load equals the sum of the line 14 plus line 15 values for the rooms and spaces.

Sensible Cooling Loads for Opaque Panels, Infiltration, and Engineered Ventilation

There are no time of day issues for the Manual J sensible cooling loads for opaque panels, infiltration, and engineered ventilation. Since there are no time of day issues, block load values for opaque panels, and infiltration, are equal to the sum of the values for the rooms and local spaces. The engineered ventilation load is typically a system load that does not affect room and local space loads, or exhaust fan ventilation increases infiltration loads.

- Opaque panel loads (lines 7 through 11 on Form J1) are average loads for the whole day, so they do not depend on time of day.
- Infiltration loads (line 12 on form J1) and engineered ventilation loads (line 16 on Form J1) are calculated for the local summer design condition that occurs at about 4 pm DST, so these loads do not depend on time of day.

Sensible Cooling Loads for Occupants and Appliances

The Form J1, line 13 occupant loads and internal loads can be a time of day issue, because load distribution among the rooms and local spaces, and load assignment to a particular room or space, depends on the time of day for the load calculation. However, this has no effect on block load vs. the sum of the room/space loads if the number of occupants for the block load calculation and the calculations for the room/space loads are equal, and if the total internal load for the block load and the room/space loads are equal. In other words, Form J1, line 13 does not cause a time of day issue if the total occupant load and the total internal load are the same for the block load, and the room and local space loads.

Sensible Cooling Loads for Fenestration

The sensible cooling loads for windows and skylights, per lines 6a and 6b on Form J1, are average loads for the whole day, so they do not depend on time of day. However, the excursion value on line 20 of Form J1 deals with the time of day issue, as explained here:

The AED excursion value for fenestration cooling load on line 20 of Form J1 is a space load. Therefore the sensible cooling load for the block load space, and for the associated rooms and local spaces, equals the sum of the line 14, line 15, and line 20 values. However, the line 20 value for the block load is, normally, not equal to the sum of the line 20 values for the rooms/spaces.

- For single-zone systems and fenestration that faces two or more directions, the AED excursion value for the block load will not be equal to sum of the AED excursion values for the associated rooms and spaces. Therefore, the block sensible cooling load will, normally, not be equal to the sum of the room sensible cooling loads.
- For single-zone systems with all fenestration facing one direction, the fenestration loads peak during the same hour of day, so the AED excursion value for the block load will be equal to sum of the AED excursion values for the associated rooms and local spaces. Therefore, the block sensible cooling load will be equal to the sum of the room sensible cooling loads.
- For zoned systems, an AED excursion value is used for the block load, and full excursion values are used for the associated rooms and local spaces. The AED excursion value for the block load will not equal the sum of the full excursion values for the associated rooms and local spaces. Therefore, the block sensible cooling load will not be equal to the sum of the room sensible cooling loads.

Other *Manual J* Procedures

The Seventh Edition of Manual J and the abridged version of MJ8 have no hour of day sensitivity for fenestration loads (these procedures are based on the assumption that the conditioned space always has adequate exposure diversity). These simplified procedures do not apply to zoned systems, or to any single-zone system that serves a block load space that has a fenestration excursion load.

6-10 Room Cfm for Single-Zone Systems

As explained by Section 1-13 and Section 6-8, the design value for blower Cfm is determined when OEM expanded performance data is used to find equipment that is compatible with the ***Manual J*** loads that appear on line 21 of Form J1. This blower Cfm value is used to determine the supply air Cfm for each room and local space served by the equipment.

The local air flow rate (supply air Cfm) is proportional to the size of the room or local space load in comparison to the block load for the space served by the equipment. For example, if the room sensible load is 7,000 Btuh and the block sensible load for the space served by the equipment is 35,000 Btuh, the room supply Cfm is 20 percent of the blower Cfm (7,000 divided by 35,000). This equation provides room Cfm values:

$$\text{Room Cfm} = \frac{\text{Blower Cfm x Room Load}}{\textbf{Block Space Load}}$$

*Where: The blower Cfm is the **Manual D** design value for an operating point blower, or an operating range blower (see Section 6-4).*

Where: The room/space load values for heating and sensible cooling are provided by line 21 on Form J1.

The four columns at the far right of Form J1 are for room and local space loads. For heating, these columns show that the line 21 load is equal to the line 14 space load plus the line 15 duct load. For sensible cooling, these columns show that the line 21 load is equal to the line 14 space load plus, the line 15 duct load, plus the line 20 excursion load.

Room Cfm calculations are expedited by dividing the blower Cfm by the block space load. This creates a flow factor that represents the supply air Cfm requirement for one Btuh of load.

$$\text{Flow Factor} = \frac{\text{Blower Cfm}}{\textbf{Block Space Load}}$$

This way, room Cfm calculations are just a simple multiplication. As demonstrated here, room Cfm equals the product of the flow factor and the room load.

Room Cfm = Flow Factor x Room Load

However, there are two room loads, one for heating and one for cooling. Therefore, there is a flow factor for heating and flow factor for cooling.

The heating flow factor equals blower Cfm for heating divided by the block space load for heating. The cooling flow factor equals blower Cfm for cooling divided by the block space load for sensible cooling. These factors are called the heating factor (HF) and the cooling factor (CF).

$$HF = \frac{\text{Blower CFM for Heating}}{\textbf{Block Space Heating Load}}$$

$$CF = \frac{\text{Blower CFM for Cooling}}{\textbf{Block Space Cooling Load}}$$

Since there are two flow factors, there are two room Cfm values. Room air flow rates are obtained by multiplying room heating and sensible cooling loads by HF and CF.

Room Heating Cfm = HF x Room Heating Load
Room Cooling Cfm = CF x Room Sensible Load

These room Cfm values represent the maximum supply air requirements for winter and summer. One value is usually larger than the other, so the design Cfm for airway sizing is the larger value.

Duct Sizing Worksheet

HF = Blower Cfm / Manual J Heat Loss = (   ) / (   ) =
CF = Blower Cfm / Manual J Sensible Heat Gain = (   ) / (   ) =

FR Value

See Appendix 18 for blank form.

Supply - Trunk	Heating Btuh	Cooling Btuh	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
Supply-Side Runouts									
S1 —									
S2 —									
S3 —									
S4 —									
S5 —									
S6 —									
S7 —									
S8 —									
S9 —									
S10 —									
S11 —									
S12 —									
Supply-Side Trunks									
Run numbers:		S-Trunk 1							
Run numbers:		S-Trunk 2							
Run numbers:		S-Trunk 3							
Run numbers:		S-Trunk 4							

Figure 6-6

Design Cfm = Maximum [Heating Cfm : Cooling Cfm]

Design Cfm values are calculated by the Duct Sizing Worksheet. Figure 6-6 shows that HF and CF calculations are made at the top of the worksheet. Then there are columns for recording room heating loads (H-Btuh) and sensible cooling loads (C-Btuh) at the left side of the worksheet. The next two columns (H-Cfm and C-Cfm) hold the results of the heating Cfm and cooling Cfm calculations. Then the larger of the two seasonal Cfm values is entered in the Design Cfm column.

- The sum of the design Cfm (Dsn Cfm) values on the Duct Sizing Worksheet will not be equal the blower Cfm value because some values are for heating and some are for cooling.
- ***Manual D*** sizes airways for the worst-case contingency. Since there are different values for room heating Cfm and cooling Cfm, the local airway is sized for the larger value (per the design Cfm column on the Duct Sizing Worksheet). Use of the larger value to size airways provides the capability, if so desired, to deliver the worst case Cfm to the room. or local space.
- Cfm values for air-balancing are an entirely different matter. See Sections 6-23 and 6-24.

6-11 Summing of Room Cfm Values for Single-Zone Systems

The sum of the room heating Cfm values on the Duct Sizing Worksheet will be equal the blower Cfm value. This is because the sum of the room heating loads is equal to the heating load for the block load space, as explained by Section 6-9.

The sum of the room cooling Cfm values on the Duct Sizing Worksheet will normally not be equal the blower Cfm value. This is because the sum of the room sensible cooling loads will not normally be equal to the sensible cooling load for the block load space, as explained by Section 6-9.

6-12 Summing of Room Cfm Values for Zone Damper Systems

The Duct Sizing Worksheet and Section 6-10 guidance provides base-case design Cfm (Dsn Cfm) values for zone damper systems. However the design Cfm for one or more of the branch runs is increased if the zone has over blow or selective throttling. When this is the case, the sum of the room heating Cfm values on the Duct Sizing Worksheet will be equal the blower Cfm value, and for cooling the difference will be larger than for no overblow or selective throttling. See ***Manual Zr***, Section A8-6 for related guidance.

6-13 Supply Branch Flow Rates for Airway Sizing

A branch runout duct is required for each supply air outlet and at least one supply air outlet is required for each conditioned room or local space. However, some rooms/spaces may require two or more supply air outlets, depending on the design Cfm value for the room or space, and the attributes of the room or space.

- Most rooms and spaces have one supply outlet for the design Cfm value on the Duct sizing Worksheet.
- A rooms or space that requires a relatively large flow of supply air may have two or more supply air outlets, because one properly sized outlet may be objectionably large.
- When a room is relatively large, discharge air throws from two or more properly sized supply air outlets may be required to obtain adequate air mixing and motion within the occupied zone.
- In a cold climate, when a room has a relatively large amount of exposed wall and glass area, it may be desirable to use two or more properly sized supply air outlets (vertical discharge floor outlets) to reduce stratification and improve comfort in the occupied space.

There are no fixed rules regarding the number of supply air outlets for a room or space. The practitioner may minimize duct system geometry and installed cost if comfort is not adversely affected. The design also should be sensitive to aesthetics, window shading devices and probable furniture locations. Refer to ACCA ***Manual T*** for more information about selecting and sizing air distribution hardware.

Example - Two Supply Runs for One Room

Br Run	H-Btuh	C-Btuh	H-Cfm	C-Cfm	Dsn Cfm
3 —	3,040	2,100	94	105	105
4 —	5,500	4,000	170	200	100
5 —					100
6 —	4,500	2,900	139	145	145

A room requires 200 Cfm for cooling and 170 Cfm for heating. The design Cfm is 200 Cfm. Branches 4 and 5 deliver 100 Cfm each to the same room.

Figure 6-7

Examples of Primary and Secondary Trunks

Example duct system	Primary Trunks		Secondary Trunks	
	Supply	Return	Supply	Return
Figure 7-8	0	RT-1	ST-1, ST-2	0
Figure 7-12	ST-2	RT-1	ST-1	0
Figure 7-18	ST-4	0	ST-1 > ST-3	RT-1, RT-2
Figure 8-1	ST-2	RT-2	ST-1	RT-1
Figure 8-8	0	0	ST1 > ST5	RT-1, RT-2

A primary trunk is attached to the equipment cabinet, or to a plenum that is attached to the equipment cabinet. A supply trunk downstream from a reducing fitting is a secondary trunk.. A secondary trunk must serve two or more branches.

Figure 6-8

Two or more lines on the duct sizing worksheet are used for one room or space if there are two or more runouts to the room or space. For this scenario, use the first line to calculate the H-Cfm and C-Cfm values for the room, but do not transcribe the largest Cfm value to the Dsn Cfm column, because it will be portioned to two or more branch runs. After the design Cfm value is split into two or more runout Cfm values, they are entered on the form, using as many lines as necessary. A multiple runout example is provided by Figure 6-7 (lines 4 and 5). In this case, two branch ducts deliver 200 Cfm of supply air to the room.

6-14 Primary Trunks and Secondary Trunks

A primary trunk is the section of supply duct or return duct directly downstream or upstream from the blower. All of the blower Cfm enters a primary supply trunk at the blower discharge fitting or collar, and all of the blower Cfm leaves a primary return trunk at the blower intake fitting, or collar. Primary trunks may serve:

- Runout ducts.
- Secondary trunks.
- Primary trunk runouts and secondary trunks.

Only part of the blower Cfm enters a secondary supply trunk or leaves a secondary return trunk. Secondary trunks may serve:

- Runout ducts.
- Other secondary trunks.
- Secondary trunk runouts and other secondary trunks.

Section 7 and Section 8 provide duct system sketches for example duct sizing problems. Figure 6-8 (previous page) identifies the primary trunks and secondary trunks for these sketches.

6-15 Supply Trunk Flow Rates for Airway Sizing

There is a set of branch runout Cfm values for heating and a set of values for cooling. For constant Cfm, single-zone systems, the sum of the heating Cfm values will equal the blower Cfm, but this may not be true for air zoned systems. For either type of system, the sum of the cooling Cfm values will normally not be equal to the blower Cfm (see Sections 6-8, 6-9, 6-11, and 6-12).

Use these rules to determine the design Cfm for a section of supply trunk:

- For a primary supply trunk, simply use the blower Cfm to size the trunk. This is because the largest Cfm that can flow through a primary trunk is the design value for blower Cfm.
- *Manual D* sizes secondary trunks and branch runout ducts for the worst-case contingency, as explained by Section 6-10.
- The supply air Cfm entering a section of secondary supply trunk equals the sum of the Cfm values for all of the supply air outlets that are downstream from the trunk entrance.
- The trunk Cfm for heating is the sum of the downstream heating Cfm values, and the trunk Cfm for cooling is the sum of the downstream cooling Cfm values.
- A trunk section is sized for the larger sum, which is normally the sum of the cooling Cfm values.
- Figures 6-9 and 6-10 demonstrate procedure.

The preceding rules apply to zone damper systems. In this case, the branch cooling Cfm values for a zoned system tend to be larger than for a single-zone system because supply air Cfm is based on the size of the peak fenestration cooling load for the room or space served by the branch runout (see Section 6-8). In addition, when over blow or selective throttling applies, the cooling or heating Cfm that is correct for the room or space load will be increased by some amount, per Manual Zr guidance. In other words, airway sizes for a zone damper system

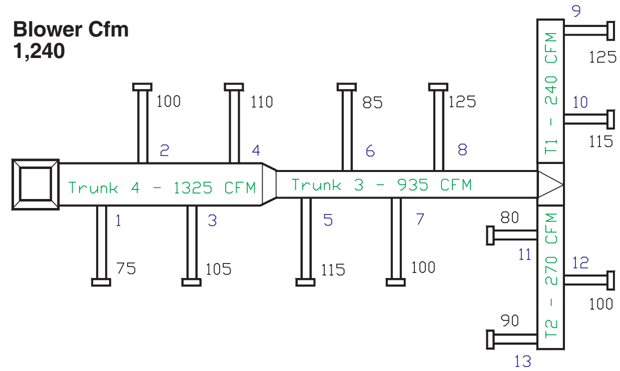


Figure 6-9

Supply Trunk Cfm				
Run - Trunk		**H-Cfm**	**C-Cfm**	**Dsn-Cfm**
1 — T-4		65	75	
2 — T-4		115	100	
3 — T-4		95	105	
4 — T-4		95	110	
5 — T-3		100	115	
6 — T-3		85	85	
7 — T-3		90	100	
8 — T-3		110	125	
9 — T-1		115	125	
10 — T-1		110	115	
11 — T-2		95	80	
12 — T-2		85	100	
13 — T-2		80	90	
Sum downstream Cfm for T1, T2 and T3; use blower Cfm for T4.	**Trunk** T1	225	240	240
	Trunk T2	260	270	270
	Trunk T3	870	935	935
	Trunk T4	1,240	1,240	1,240

Figure 6-10

tend to be larger than for the same duct system with no zone dampers. This makes the zone damper, not the airway size, the controlling authority for the amount of airflow to a room or space.

6-16 Return Branch Flow Rates for Airway Sizing

The number of branch return runs for a particular system equals the number of return air openings. For example, if return air grilles are installed in every room, there will be as many return branches as there are rooms. Or at the other extreme, there are no return branches for a system that features one central return near the air handler (located in a closet, hall or attic).

A low-resistance return path (see Section 4-9) shall be provided for isolated rooms or spaces that do not have a dedicated return. For each occurrence, the Cfm for a common return equals the total supply Cfm delivered to the rooms or areas served by the return.

The Duct Sizing Worksheet (Figure 6-11) determines the design Cfm value for a return branch. If a room has one return, the Cfm for the return run equals the supply air Cfm for the room. If two or more rooms have a common return, the Cfm for the return run equals the total supply air Cfm delivered to the rooms.

An example of a multiple return system is provided by Figure 6-12. In this case, four return grilles and four branch return ducts collect air that was distributed to four different areas of the dwelling.

The portion of the duct sizing worksheet that pertains to the return duct system is used to calculate Cfm values for branch return ducts. For example, Figure 6-13 shows the calculations for the Figure 6-12 system. In this case return branch 1 is for supply runs 9 and 10; return branch 2 is for supply runs 11, 12 and 13; return branch 3 is for supply runs 5, 6, 7 and 8; and return branch 4 is for supply runs 1, 2, 3 and 4.

6-17 Return Trunk Flow Rates for Airway Sizing

The airway size of a return trunk section depends on the Cfm entering the section. This value equals the sum of the Cfm values for the relevant upstream returns. The Duct Sizing Worksheet is used to calculate these values.

Use these rules to determine the design Cfm for a section of return trunk for a single-zone system, and for a zone damper system.

- For a primary return trunk, simply use the blower Cfm to size the trunk. This is because the largest Cfm that can flow through a primary trunk is the design value for blower Cfm.
- The Cfm entering a section of return trunk equals the sum of the Cfm values for all of the return grilles that are upstream from the trunk entrance.
- The trunk Cfm for heating is the sum of the heating Cfm values, and the trunk Cfm for cooling is the sum of the cooling Cfm values.
- A trunk section is sized for the larger heating-cooling sum, which is normally the sum of the cooling Cfm values.
- Figure 6-14 (next page) applies these rules to the Figure 6-12 system.

6-18 Branch Airway Sizing (Supply or Return)

Once design values for the friction rate and branch Cfm are known, the size of the branch duct airway is determined by a duct slide rule or friction chart. For example, Figure 6-15 (next page) shows that for metal duct, a 6.3

Return-Side Runouts								
Return - Trunk	Associated Supply Runs	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
R1 —								
R2 —								
R3 —								
R4 —								
R5 —								
R6 —								
R7 —								
R8 —								
Return-Side Trunks								
Run numbers:	R-Trunk 1							
Run numbers:	R-Trunk 2							
Run numbers:	R-Trunk 3							
Run numbers:	R-Trunk 4							

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per Manual Zr, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

See Appendix 18 for blank version.

Figure 6-11

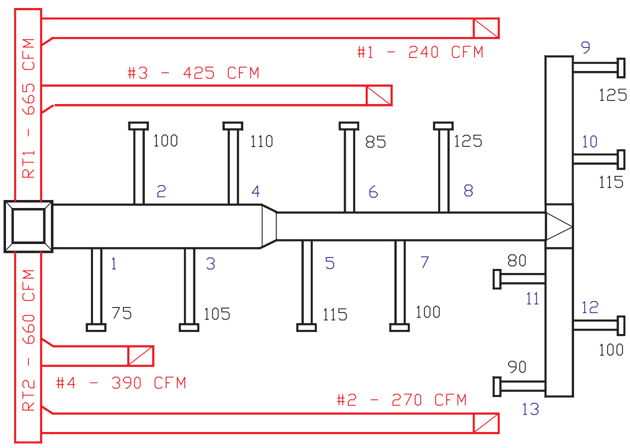


Figure 6-12

Example - Cfm Values for Multiple Returns				
Run - Trunk	**Asoc. Supply Runs**	**H-Cfm**	**C-Cfm**	**D-Cfm**
1 — R1	9, 10	225	240	240
2 — R2	11, 12, 13	260	270	270
3 — R3	5, 6, 7, 8	385	425	425
4 — R4	1, 2, 3, 4	370	390	390

Figure 6-12 shows the duct system geometry for this example.

Figure 6-13

inch diameter airway is compatible with 100 Cfm and a 0.07 IWC/100 friction rate.

Also verify that air velocities in branch runs do not exceed the recommended limit (see Table A1-1). This calculation depends on the tool that was used to find the design friction rate value.

- If the design friction rate value was produced by the friction rate equation (see Section 6-7), a duct slide rule or friction chart is used to verify that runout velocity is acceptable.

- If the design friction rate value was read from the friction rate chart (see Figure 6-5), runout velocities tend to be less than the recommended maximum, providing that the design friction rate does not exceed 0.18 IWC/100 Ft.

For the maximum friction rate on the Friction Rate Chart (0.18 IWC/100 Ft), air velocity depends on Cfm and the type of duct material.

- For rigid return duct, air velocity will be 700 Fpm or less if the flow rate is 100 Cfm or less.
- For wire helix flexible duct, air velocity will be 700 Fpm or less if the flow rate is 300 Cfm or less.

If runout velocity exceeds the maximum value, the branch duct must be resized. The new size is determined by the branch Cfm and the velocity limit. This size is read from a duct slide rule, or a friction chart. For example, Figure 6-16 shows that for metal duct, a 5.2 inch diameter airway is compatible with 130 Cfm and 900 Fpm.

The duct sizing worksheet is used to organize sizing calculations for branch ducts. Figure 6-17 (next page) provides an example that shows final metal duct sizes are based on Cfm and friction rate when air velocity is in the acceptable range.

6-19 Trunk Airway Sizing (Supply or Return)

Once the design friction rate value and trunk Cfm are known, the preliminary size for a trunk section is read from a duct slide rule or a friction chart. For example, if the friction rate is 0.10 IWC/100, a duct slide rule for metal duct shows that a 15.5 inch diameter is compatible with 1,300 Cfm of air flow.

After finding a duct size that is compatible with the design friction rate, verify that the air velocity does not exceed the maximum value (see Table A1-1). If the velocity exceeds the maximum value, the trunk shall be resized. In this case, the new size is based on airway Cfm and permitted velocity.

For example, it has already been demonstrated that a 15.5 inch (metal) duct is compatible with 1,300 Cfm and a 0.10 IWC/100 friction rate, but air velocity is excessive (the duct slide rule for metal duct indicates the air velocity is about 1,020 Fpm). Therefore, airway size is based on 1,300 Cfm and 900 Fpm. A duct slide rule for metal duct indicates the appropriate size is 16.5 inches.

The duct sizing worksheet is used to perform sizing calculations for trunk ducts. Figure 6-18 (next page) provides two examples for metal ducts. The first line shows that when velocity is acceptable, airway size is based on friction rate and Cfm. The second line shows that when air velocity is excessive, airway size depends on Cfm and maximum allowable velocity.

Example - Cfm Values for Return Trunks				
Run - Trunk	**Asoc. Supply Runs**	**H-Cfm**	**C-Cfm**	**D-Cfm**
1 — R1	9, 10	225	240	
2 — R2	11, 12, 13	260	270	
3 — R3	5, 6, 7, 8	385	425	
4 — R4	1, 2, 3, 4	370	390	
	Trunk RT-1 (R1, R3)	610	665	665
	Trunk RT-2 (R2, R4)	630	660	660
Figure 6-12 shows the duct system geometry for this example.				

Figure 6-14

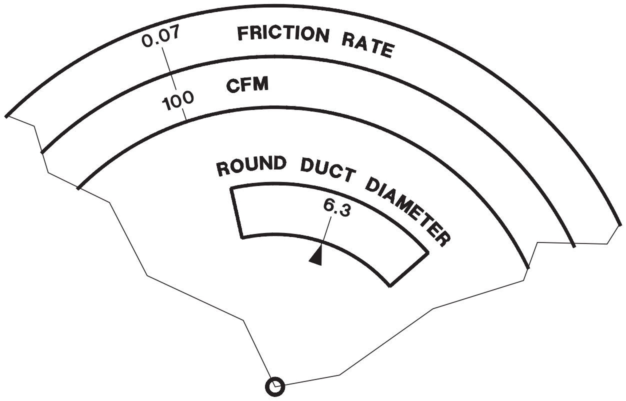


Figure 6-15

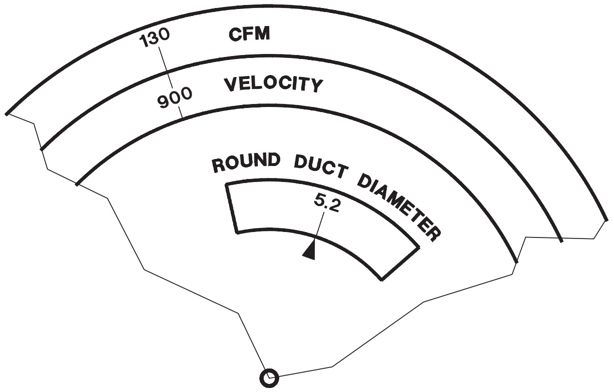


Figure 6-16

6-20 Equivalent Rectangular Sizes

If square or rectangular shapes are required, a duct slide rule, or a round-to-square conversion table, is used to find the equivalent rectangular size. This size is entered in the column on the right side of the Duct Sizing Worksheet.

- Equivalent rectangular size is the size that produces the same air flow resistance (friction rate) as the round size, but the velocity through the equivalent rectangular airway is always lower than the velocity through the round airway.
- Equivalent rectangular sizes may vary, depending on the slide rule, because the industry has two, slightly different, equations for converting round sizes to equivalent rectangular sizes.
- The velocity for a round airway is read from any duct sizing slide rule.
- The velocity for a rectangular airway is read from the "Auxiliary Calculations" side of the ACCA Duct Sizing Slide Rule.
- The velocity for any airway shape is calculated by dividing airway Cfm by square feet of cross-sectional area.

6-21 Optional Airway Sizes

The *Manual D* procedure returns the minimum acceptable airway size for duct runs that comply with good industry practice regarding design, fabrication and installation. The designer has the option to specify larger airway sizes when there is concern, or anticipation, that the actual fabrication and installation procedures and details may cause the actual total effective length of the installed circulation path to be longer than the calculated length of the best practices path. However, this acerbates the need for balancing dampers (as explained in Sections 6-2 and 6-3).

6-22 Air Distribution Hardware

After air distribution hardware has been selected, the locations and sizes of the supply air outlets and returns are recorded on a sketch of the duct system. This is useful for evaluating straight run lengths and fitting requirements.

- The air distribution devices (supply outlets and returns) shall be selected in accordance with procedures and guidance recommendations provided by *Manual T*.
- Manufacturer's performance data shall be used to select air distribution devices.

6-23 Air Balance Cfm Values for Single-Zone Systems

The air balance values for a single-zone system may be based on the average of the heating Cfm values and cooling Cfm values on the Duct Sizing Worksheet. However, the sum of a set of average values will normally be greater than the design blower Cfm value (e.g., the largest blower Cfm at the top of the Duct Sizing Worksheet). This difference is normalized by multiplying each average value by the design blower Cfm value and dividing by the sum of the average values, as demonstrated by Figure 6-19.

Sections 6-2, 6-3, 6-8, 6-9, 6-10, 6-11, and the side bar on the next explain why the sum of set of Cfm values for the rooms and spaces in a dwelling will not be equal to the blower Cfm. Balancing a single zone system is a compromise, as far as room and space Cfm values are concerned.

FR Value = 0.12 and Maximum Velocity = 900 Fpm			
Dsn Cfm	Round Size	Velocity	Final Size
120	6.0"	620 ok	6"
90	5.4" (use 6")	460	6"
140	6.4" (use 7")	540	7"

Figure 6-17

FR Value = 0.12 and Maximum Velocity = 900 Fpm			
Dsn Cfm	Round Size	Velocity	Final Size
500	10.3" (use 11')	790 ok	11"
1,100	14.0"	1,050	15"

Figure 6-18

Normalized Cfm Values for a Single-Zone System Based on 1,000 blower Cfm for air balancing				
Supply Run	Heating Cfm	Cooling Cfm	Average Cfm	Normalized Cfm
1	106	130	118	113.3
2	106	99	102.5	98.4
3	110	133	121.5	116.7
4	118	115	116.5	111.9
5	107	125	116	111.4
6	125	109	117	112.3
7	128	146	137	131.5
8	135	156	145.5	139.7
9	65	70	67.5	64.8
Totals	1,000	1,083	1,041.5	1,000

1) The Duct Sizing Work Sheet provides heating Cfm, cooling Cfm and design Cfm values, and a column for normalized Cfm values. Design Cfm values are used to size airways, normalized Cfm values are used air balancing.
2) If blower Cfm for heating and cooling are not equal, the blower Cfm for calculating normalized Cfm values is the larger of the two values.
3) A normalized Cfm value equals the average Cfm value multiplied by the design value for blower Cfm and divided by the total of the average Cfm values. LIne 1 For example:
118 x (1,000 / 1,041.5) = 113.5 Cfm

Figure 6-19

The normalized Cfm values on the Friction Rate Worksheet are a staring point.

- Based on the owner's experience with the space temperature for various rooms and spaces during winter and summer, it may be necessary to increase or reduce the Cfm for one or more spaces.
- If there is a large difference between the occupant-preferred heating and cooling Cfm for a room or space, one or more damper settings must be readjusted for winter and summer.
- If a room or apace has a very large excursion for the time-of-day fenestration load, the runout damper setting may have to be readjusted for winter and summer.
- Temperature excursions (vs. the set-point of a remote thermostat) in rooms and spaces are the unavoidable attribute of a single-zone system.

6-24 Air Balance Cfm Values for Zone Damper Systems

In principle, the ***Manual D*** guidance for single-zone systems also applies to zone damper systems, in that the Cfm values for airway sizes are maximum need values, and that the sum of these Cfm values will be greater than the blower Cfm value. However, there are differences in the air balancing procedure for the two types of systems.

- Section 6-12 of this manual explains that one or more of the zone Cfm values may have an over blow or selective throttling adjustment. This increases the Cfm value for airway sizing, and for air balancing.
- For single zone systems, the normalized Cfm column for the Duct Sizing Worksheet may be used for air balancing. The values in this column only apply to single-zone systems.
- The air balancing procedure for air-zoned systems also uses normalized Cfm values, but these are not the same as the normalized Cfm values on the Duct Sizing Worksheet. See Section 9-11 for more guidance on this issue.

6-25 Refer to Manual Zr for Related Guidance

Section 1 - Zoning Benefits
This section discuses single-zone performance vs. zoned system performance, as far as comfort and energy use are concerned.

Design Cfm vs. Air Balancing Cfm

Manual D sizes airways for the worst-case contingency. Since there are different values for room heating Cfm and cooling Cfm, the local airway is sized for the larger value (per the design Cfm column on the Duct Sizing Worksheet). Use of the larger value to size airways provides the capability, if so desired, to deliver the worst case Cfm to the room.

Balancing the system is an entirely different matter than airway sizing. The column of design Cfm values on the Duct Sizing Worksheet is a column of maximum values that cannot occur simultaneously (some are for heating and some are for cooling). Therefore, this list of values, and their sum, are not relevant to air balancing work. Refer to Section 6-23 for additional guidance.

The sum of the normalized Cfm (N-Cfm) values on the Duct Sizing Worksheet will be equal the blower Cfm value. This is explained Section 6-23.

Section 2 - Zoning Methods
This section discusses zoning issues and the type of equipment that may be used to provide zoned comfort. Figure 2-1 compares the attributes of various methods.

Section 3 - Making Zoning Decisions
This section discuses the issues that create a zone, and the issues that affect the maximum number of zones.

Section 4 - Load Calculations for Zoned Systems
This section explains Manual J procedures for zoned systems, and shows how to use AED curves to group rooms and spaces into zones.

Appendix 7 - Air Distribution
This appendix discusses air distribution hardware (diffusers, registers and grilles) for air-zoned systems.

Appendix 8 - Duct System Design
This appendix explains how ***Manual J*** and ***Manual D*** guidance applies to duct system design. for zoned systems

Appendix 9 - Balancing Zone Damper Systems
This appendix provides step-by-step guidance for balancing zone damper systems.

Sizing Rigid Constant Cfm Duct Systems

This section provides examples of airway sizing calculations for constant Cfm duct systems fabricated from rigid materials. The examples are for a simple radial system, an extended plenum system, a reducing trunk system, and a complex trunk and branch system. The ***Manual J*** procedure for single-zone comfort systems (the Standard procedure, per ***Manual J***, Figure A11-1) provides cooling load values for supply Cfm calculations (see also, Section 6-8, this manual).

7-1 Radial Duct System

Figures 7-1 and 7-2 show an 800 Cfm radial system that has below-slab supply runs and above-ceiling return runs. The supply-side ducts are plastic (for below grade use) and the return-side ducts are sheet metal.

A heat pump provides heating and cooling. Figure 7-3 (next page) shows the ***Manual J*** heating and cooling loads for calculating Cfm values for supply air outlet selection, and for duct airway sizing. Figure 7-4 (next page) provides the equipment manufacturer's blower table and component pressure drop data.

Effective Length Calculation

Considering system geometry and fitting types, the critical circulation path is defined by supply run 7 and return run R3. (Since the same fittings are used for supply runs and return runs, the longest run is determined by comparing straight run lengths.)

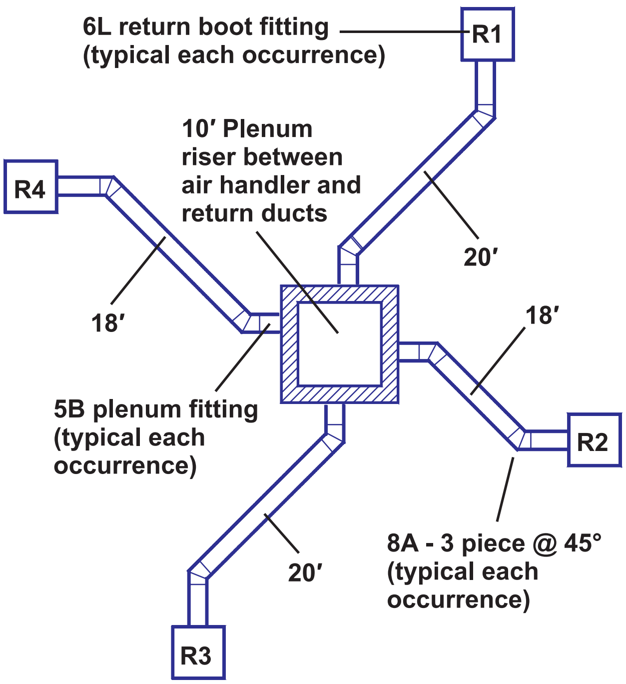


Figure 7-2

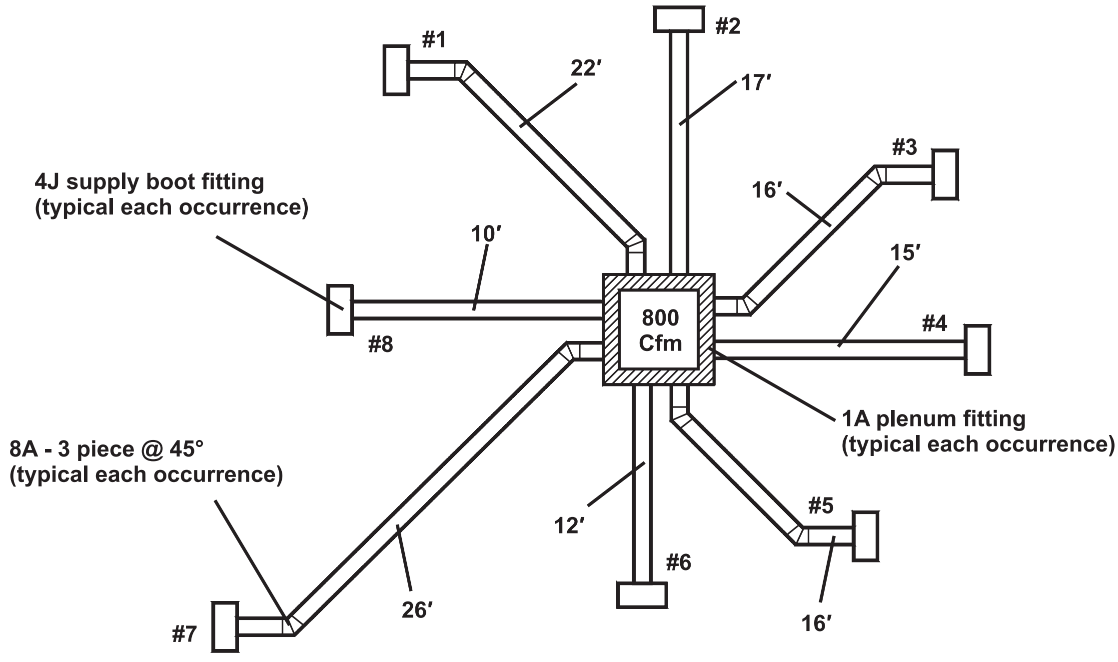


Figure 7-1

Duct Lengths and Manual J Loads for the Radial System Example			
Runout	**Length Feet**	**Heating Btuh**	**Cooling Btuh**
1	22	4,250	2,750
2	17	3,970	3,500
3	16	3,800	2,380
4	15	4,590	3,800
5	16	2,350	1,690
6	12	3,020	2,590
7	26	3,430	2,700
8	10	4,500	2,610
Total	~	29,910	22,020

1) Block loads (Btuh): Heating = 29,910 Sensible cooling = 21,120
2) For cooling, fenestration excursion loads cause the sum of the room cooling loads to be more than the block cooling load (see Sections 6-8 through 6-12).

Figure 7-3

Blower Data for the Radial System Example			
Discharge Cfm	**External Resistance (IWC) vs. Speed**		
	High	**Medium**	**Low**
600			0.48
650		0.66	0.33
725	0.67	0.51	0.17
800	0.51	0.36	
875	0.36	0.19	
950	0.17		

1) Operating point blower tested with wet coil, auxiliary heater and low efficiency filter in place
2) For an electronic filter, subtract 0.11 IWC from the pressure values listed in this table.

Figure 7-4

Figure 7-5 shows that the effective length of the critical circulation path is 221 feet. This path flows through supply run seven (111 feet), and the R3 return run (110 feet).

Design Friction Rate Calculation

Blower data for medium speed shows the blower will deliver 800 Cfm when it operates against 0.36 IWC of external resistance. An electronic filter is added to the system, so the resistance produced by this item (0.11 IWC), a supply outlet (0.03 IWC), a return (0.03 IWC) and a hand damper (0.03 IWC) is 0.20 IWC. The available static pressure is 0.16 IWC, so the design friction rate is 0.07 IWC/100, as demonstrated by Figure 7-6 (next page).

Effective Length Worksheet for the Radial System Example									
Element	**Supply Run ID Number**				**Element**	**Return Run ID Number**			
	S7	Notes				R3	Notes		
Trunk Length					**Trunk Length**	10			
Trunk Length					**Trunk Length**				
Runout Length	26				**Runout Length**	20			
Group 1 (A)	35				**Group 5** (B)	40			
Group 2					**Group 6** (L)	20			
Group 3					**Group 7**				
Group 4 (J)	30				**Group 8** (A)	20	(2 @ 10)		
Group 8 (A)	20	(2 @ 10)			**Group 10**				
Group 9					**Group 11**				
Group 11					**Group 12**				
Group 12					**Group 13**				
Group 13					**Other**				
Other					**Other**				
Total Length	111				**Total Length**	110			

Figure 7-5

Friction Rate Worksheet for the Radial System Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.36 IWC   Cfm = 800

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	______
Electric resistance heating coil	______
Hot water coil	______
Heat exchanger	______
Low efficiency filter	______
High or mid-efficiency filter	______
Electronic filter	0.11
Humidifier, UV lights, other	______
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	______

Total component losses (CPL) 0.20 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP – CPL) = (0.36 – 0.20) = 0.16 IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (111 + 110) = 221 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.07 IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

Friction Rate Chart

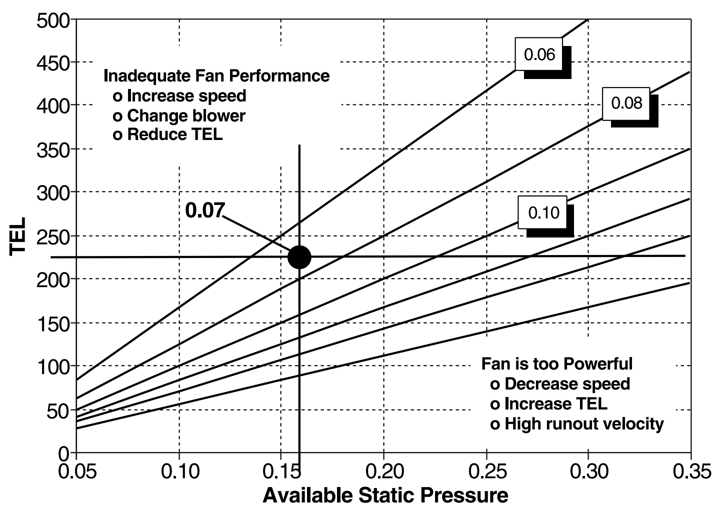


Figure 7-6

Duct Sizing Worksheet for the Radial System Example

HF = Blower Cfm / Manual J Heat Loss = 800 / 29,910 = 0.0267
CF = Blower Cfm / Manual J Sensible Heat Gain = 800 / 21,120 = 0.0379

FR Value 0.07

Supply-Side Runouts

Supply - Trunk	Heating Btuh	Cooling Btuh	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
S1 — Radial	4,250	2,750	114	104	114	7	ok	7	107
S2 — Radial	3,970	3,500	106	133	133	7	ok	7	117
S3 — Radial	3,800	2,380	102	90	102	6	ok	6	94
S4 — Radial	4,590	3,800	123	144	144	7	ok	7	131
S5 — Radial	2,350	1,690	63	64	64	6	ok	6	62
S6 — Radial	3,020	2,590	81	98	98	6	ok	6	88
S7 — Radial	3,430	2,700	92	102	102	6	ok	6	95
S8 — Radial	4,500	2,610	120	99	120	7	ok	7	107

Supply-Side Trunks

No supply trunks for a radial system.

Return-Side Runouts

Return - Trunk	Associated Supply Runs	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
R1 — Radial	S2, S3	208	223	223	9	520	9	211
R2 — Radial	S4, S5	186	208	208	8	610	8	193
R3 — Radial	S6, S7	173	200	200	8	580	8	183
R4 — Radial	S8, S1	234	203	234	9	550	9	214

Return-Side Trunks

No return trunks for a radial system.

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 7-7

Duct Sizing Calculations

Figure 7-7 summarizes the duct sizing calculations for this example. Additional detail is provided here:

- The sum of the branch cooling Cfm values is greater than the blower Cfm value (see Sections 6-8 through 6-12).
- Since the ACCA Duct Slide Rule does not provide information about plastic ducts, supply duct sizes were read from the metal duct scale. This is conservative because plastic is smoother than galvanized metal.
- All return airway sizes were read from the metal duct scale.
- Supply runs 2 and 3 are for return R1, supply runs 4 and 5 are for return R2, supply runs 6 and 7 are for return R3 and supply runs 8 and 1 are for return R4.
- Final airway size for supply runs is based on the design friction rate (0.07 IWC/100) because air velocities are less than the 900 Fpm limit.
- Final airway size of the return runs is based on the design friction rate (0.07 IWC/100) because air velocities are less than the 700 Fpm limit.

7-2 Extended Two-Way Plenum System

Figure 7-8 (next page) shows an extended plenum system with galvanized metal airways. Air handling equipment is near the center of the system between two trunk ducts. This configuration minimizes the size of the trunk ducts (each trunk carries about one-half of the required air flow). All return air is collected at a single grille located near the unit (transfer grilles provide an unrestricted return air path for each conditioned room or space).

A furnace equipped with an evaporator coil provides heating and cooling. Figure 7-8 shows the ***Manual J*** heating and cooling loads for calculating Cfm values for supply air outlet selection, and for duct airway sizing. Figure 7-8 also summarizes the equipment manufacturer's blower table and component pressure drop data.

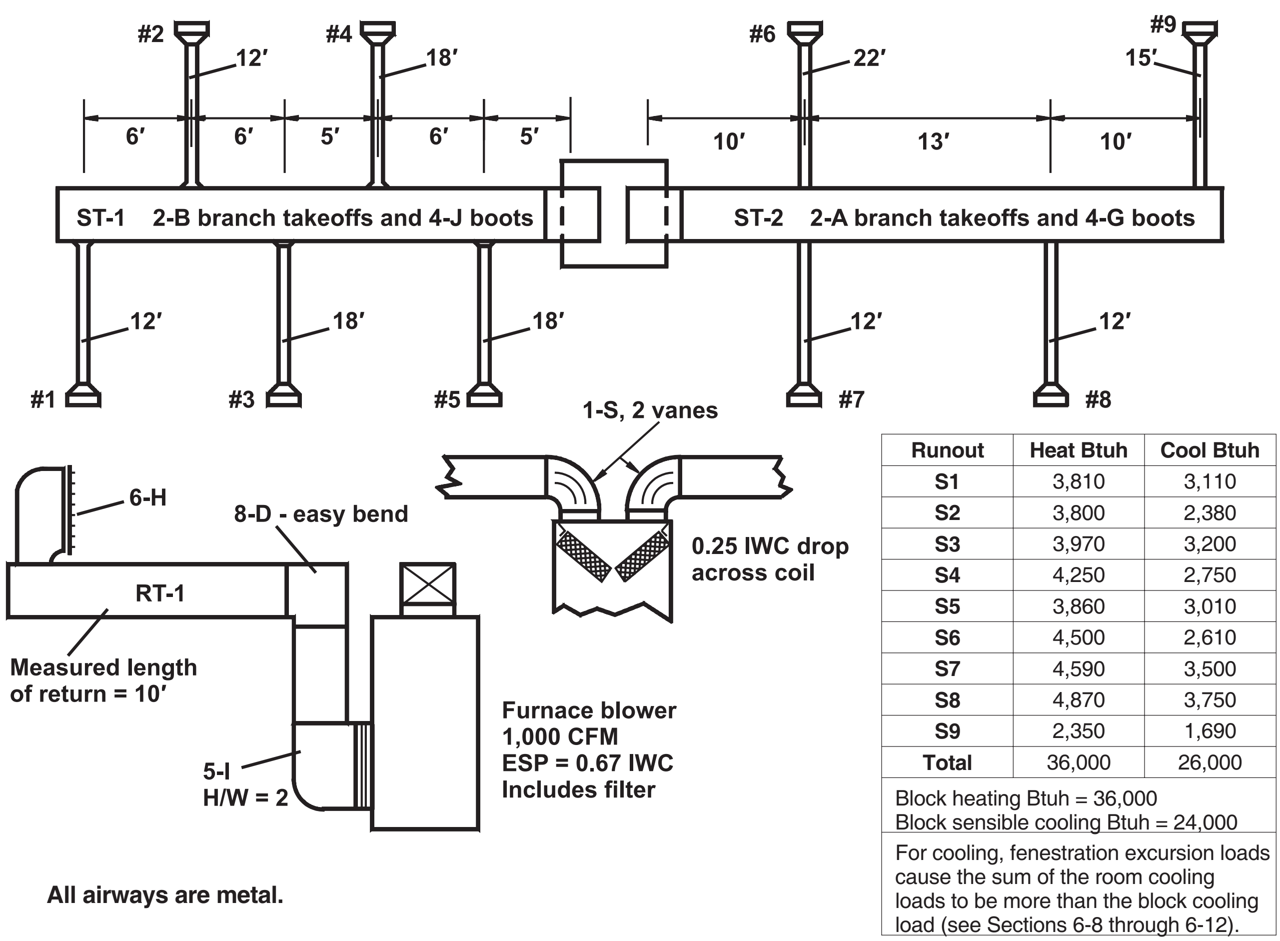
Effective Length Calculation

Effective length estimates are critical to the sizing procedure. The objective is to identify the longest supply run and the longest return run. This information is obtained by using the brute force method (calculate the effective length of each run and compare the answers), or by selecting candidates for the longest run and ignoring the other runs (calculate the effective length of a few candidate runs and compare the answers). The candidate method is preferred for hand calculations because it takes less time. For this example, runs 1, 5, 6 and 9 are candidates because:

- All the branch runouts are about the same length, so this item is discounted.
- Runs 1 and 9 are furthest from the air handler and have the longest measured lengths and the smallest branch takeoff equivalent length values.
- Runs 5 and 6 are closest to the air handler and have the shortest measured lengths and the largest branch takeoff equivalent length values.
- The other runs are ignored because the distances between the branch takeoff fittings are smaller than the differences in the branch takeoff equivalent length values.

In other words, measured lengths are compared to the equivalent lengths for branch takeoff fittings. If the differences in measured lengths are relatively small compared with the differences in branch takeoff equivalent lengths, the longest run is the run that has a branch fitting near the air handler. If the differences in measured lengths are relatively large compared with the differences in branch takeoff equivalent lengths, the longest run is the one that has a branch fitting that is far from the air handler.

It also is necessary to compare the efficiency of the fittings that are inserted in each duct run. For this example, we expect to find the longest run on the right side of the



Runout	Heat Btuh	Cool Btuh
S1	3,810	3,110
S2	3,800	2,380
S3	3,970	3,200
S4	4,250	2,750
S5	3,860	3,010
S6	4,500	2,610
S7	4,590	3,500
S8	4,870	3,750
S9	2,350	1,690
Total	36,000	26,000

Block heating Btuh = 36,000
Block sensible cooling Btuh = 24,000

For cooling, fenestration excursion loads cause the sum of the room cooling loads to be more than the block cooling load (see Sections 6-8 through 6-12).

Figure 7-8

system (run 6 or 9) because the fittings are considerably less efficient than the fittings on the left side of the system. (Normally, the same types of fittings are used on both sides of the system. Different fittings are used for this example to demonstrate a concept.)

- The equivalent length of a 4G boot is 80 feet and the equivalent length of a 4J boot is 30 feet.
- The basic equivalent length (no downstream branches) for a 2A takeoff is 35 feet and the basic equivalent length value for a 2B takeoff is 20 feet.

Based on these observations, we speculate (without making a calculation) that the run number 6 is the longest supply run. Figure 7-9 confirms this assumption. In this case the effective length of the longest supply run is 207 feet and the effective length of the return run is 120 feet. Therefore the total effective length of the critical circulation path is 327 feet.

Note: When two branch runout ducts are directly across from each other, count the opposing runout as a downstream branch. For example, refer to Figure 7-8 and note that run 6 has three downstream branches and run 7 has three downstream branches.

Design Friction Rate Calculation

For this example, the blower data shows that the blower will deliver 1,000 cooling Cfm when it operates against 0.67 IWC of external resistance. Since the resistance produced by a wet refrigerant coil (0.25 IWC), a supply outlet (0.03 IWC), a return (0.30 IWC) and a hand damper (0.03 IWC) is 0.34 IWC, the available static pressure is 0.33 IWC. Therefore, the design friction rate is based on 0.33 IWC of pressure and 327 feet of effective length. These calculations are made on Figure 7-10 (next page), which shows the friction rate for airway sizing is 0.10 IWC/100.

Duct Sizing Calculations

Figure 7-11 (ahead two pages) summarizes the duct sizing calculations, which begin with heating factor and cooling factor calculations; followed by supply runout sizing calculations, supply trunk sizing calculations and the return duct sizing calculations. Additional detail is provided here:

- The sum of the branch cooling Cfm values is greater than the blower Cfm value (see Sections 6-8 through 6-12.).

Effective Length Worksheet for the Extended Two-Way Plenum Example

Element	Supply Run ID Number				Element	Return Run ID Number			
	S1	S5	S6	S9		RT-1			
Trunk Length	28	5	10	33	**Trunk Length**	10			
Trunk Length					**Trunk Length**				
Runout Length	12	18	22	15	**Runout Length**				
Group 1 (S)	30	30	30	30	**Group 5** (I)	30			
Group 2 (A)			65	35	**Group 6** (H)	15			
Group 3					**Group 7**				
Group 4 (G)			80	80	**Group 8** (D)	65	(easy bend)		
Group 8					**Group 10**				
Group 9					**Group 11**				
Group 11					**Group 12**				
Group 12					**Group 13**				
Other					**Other**				
Other (2B)	20	45			**Other**				
Other (4J)	30	30			**Other**				
Total Length	120	128	207	193	**Total Length**	120			

Figure 7-9

Friction Rate Worksheet for the Extended Two-Way Plenum Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.67 IWC   Cfm = 1,000

Step 2) Component Pressure Losses (CPL) for Cooling

Direct expansion refrigerant coil	0.25
Electric resistance heating coil	
Hot water coil	
Heat exchanger	
Low efficiency filter	
High or mid-efficiency filter	
Electronic filter	
Humidifier, UV lights, other	
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	
Total component losses (CPL)	0.34 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (0.67 - 0.34) = 0.33 IWC

Step 4) Total Effective Length (TEL)

Supply-Side TEL + Return-Side TEL = (207 + 120) = 327 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.10 IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

Friction Rate Chart

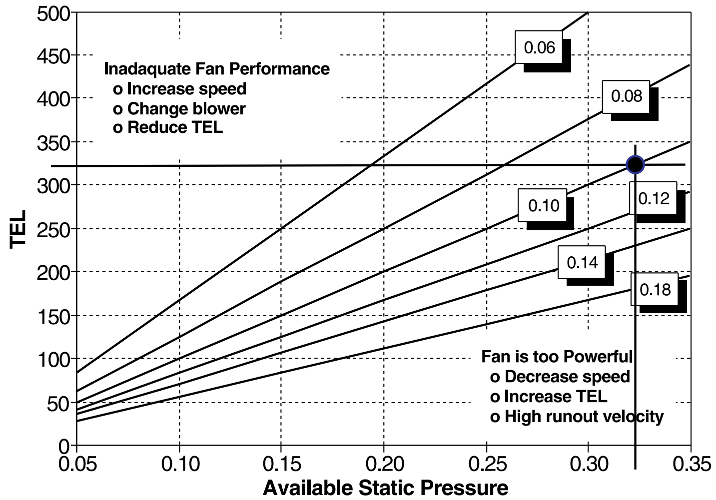


Figure 7-10

Duct Sizing Worksheet for the Extended Two-Way Plenum Example									
HF = Blower Cfm / Manual J Heat Loss = 1,000 / 36,000 = 0.278 **CF = Blower Cfm / Manual J Sensible Heat Gain =** 1,000 / 24,000 = 0.417									**FR Value** 0.10
Supply-Side Runouts									
Supply - Trunk	**Heating Btuh**	**Cooling Btuh**	**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
S1 — ST-1	3,810	3,110	106	130	130	6	ok	6	113
S2 — ST-1	3,800	2,380	106	99	106	6	ok	6	98
S3 — ST-1	3,970	3,200	110	133	133	7	ok	7	117
S4 — ST-1	4,250	2,750	118	115	118	6	ok	6	112
S5 — ST-1	3,860	3,010	107	125	125	6	ok	6	111
S6 — ST-2	4,500	2,610	125	109	125	6	ok	6	112
S7 — ST-2	4,590	3,500	128	146	146	7	ok	7	132
S8 — ST-2	4,870	3,750	135	156	156	7	ok	7	140
S9 — ST-2	2,350	1,690	65	70	70	5	ok	5	65
Supply-Side Trunks									
Run numbers: S1 through S5	**S-Trunk 1**		547	602	602	12	780	12	551
Run numbers: S6 through S9	**S-Trunk 2**		453	481	481	11	750	12	449
Return-Side Runouts One Central Return with transfer grilles, no branches.									
Return-Side Trunks									
Run numbers: Primary trunk	**R-Trunk 1**		1,000	1,000	1,000	14	950	16 std	1,000

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

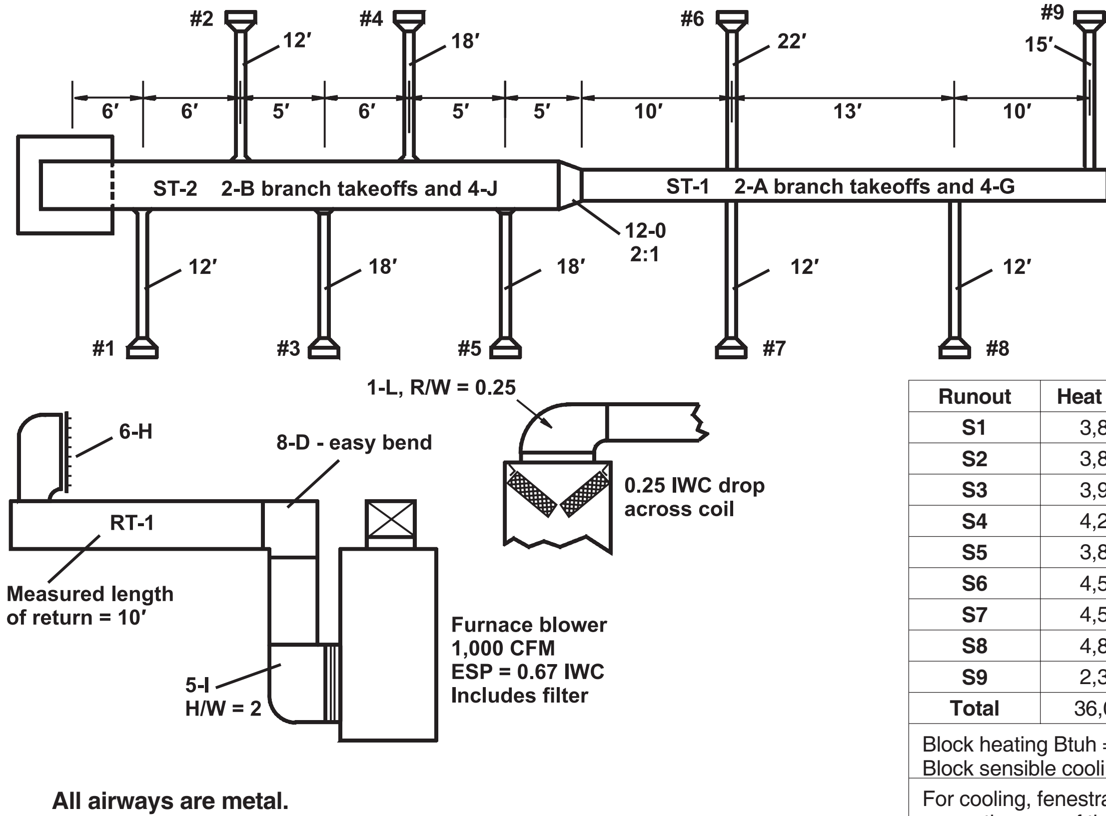
Figure 7-11

- Supply runs 1 through 5 are for trunk ST-1 and supply runs 6 through 9 are for trunk ST-2.
- The supply trunks are secondary trunks and the return trunk is a primary trunk (see Section 6-14).
- The entering Cfm value for each supply trunk equals the sum of the downstream branch Cfm values, and the return trunk Cfm value equals the blower Cfm value (see Section 6-15).
- All of the values in the “round size” column were read from the Galvanized Metal Duct scale on the ACCA Duct Sizing Slide Rule.
- The final size of supply runouts is based on the design friction rate (0.10 IWC/100) because air velocities are less than 900 Fpm.
- Runout velocities are less than 900 Fpm because the design friction rate point falls inside the 'wedge' on the friction rate chart.
- The final size of supply trunks ST-1 and ST-2 are based on the design friction rate (0.10 IWC/100) because air velocities are less than 900 Fpm.
- The final size of return trunk RT-1 is based on allowable velocity (700 Fpm) because sizing for the friction rate design value (0.10 IWC/100) produced excessive air velocity.

Equivalent Rectangular Sizes

The ACCA Duct Sizing Slide Rule converts round sizes to equivalent rectangular sizes. The rectangular size is equivalent because it produces the same air flow resistance as the round size.

However, the air velocities for the round size and equivalent rectangular size are not equal because the equivalent rectangular size has a larger cross-sectional area. This poses no problem, because the velocity for the equivalent rectangular size is always less than the velocity for the round size.



Runout	Heat Btuh	Cool Btuh
S1	3,810	3,110
S2	3,800	2,380
S3	3,970	3,200
S4	4,250	2,750
S5	3,860	3,010
S6	4,500	2,610
S7	4,590	3,500
S8	4,870	3,750
S9	2,350	1,690
Total	36,000	26,000
Block heating Btuh = 36,000 Block sensible cooling Btuh = 24,000		
For cooling, fenestration excursion loads cause the sum of the room cooling loads to be more than the block cooling load (see Sections 6-8 through 6-12).		

Figure 7-12

7-3 Reducing Plenum System

Figure 7-12 provides an example of a simple reducing plenum system that has galvanized metal airways. This example is very similar to the Figure 7-8 example. The only difference is that the equipment has been moved to one end of the duct system and an electronic filter has been added to the system.

- This arrangement requires a different type of plenum fitting at the air handler.
- Per Section A8-8, the size of a plenum airway shall be reduced if the air velocity just upstream from a branch duct slows to about 50 percent of the initial velocity.

A furnace equipped with an evaporator coil provides heating and cooling. Figure 7-12 shows the ***Manual J*** heating and cooling loads for calculating Cfm values for supply air outlet selection, and for duct airway sizing. Figure 7-12 also summarizes the equipment manufacturer's blower table and component pressure drop data.

Effective Length Calculation

Thoughtful observations expedite effective length circulations. Considering system geometry and fitting types runs 1, 6 and 9 are candidates for the longest supply run.

Figure 7-13 (next page) shows the effective length calculations for these supply runs and the return run. The result is that the effective length of the longest supply run is 255 feet and the effective length of the return run is 120 feet,.so the total effective length of the critical circulation path is 375 feet.

Design Friction Rate Calculation

For this example, blower data indicates that the blower will deliver 1,000 Cfm when it operates against 0.67 IWC of external resistance. Since the resistance produced by a wet refrigerant coil (0.25 IWC), electronic filter (0.10 IWC), a supply outlet (0.03 IWC), a return (0.03 IWC) and a hand damper (0.03 IWC) is 0.44 IWC, the available static pressure is 0.23 IWC. Therefore, the design friction rate is based on 0.23 IWC of pressure and 375 feet of effective length. These calculations are made on Figure 7-14 (ahead two pages), which shows the design friction rate is 0.06 IWC/100.

Effective Length Worksheet for the Reducing Plenum Example									
Element	**Supply Run ID Number**				**Element**	**Return Run ID Number**			
	S1	S6	S9			RT- 1			
Trunk Length	6	43	66		**Trunk Length**	10			
Trunk Length					**Trunk Length**				
Runout Length	12	22	15		**Runout Length**				
Group 1 (1L)	40	40	40		**Group 5** (I)	30			H/W=2
Group 2 (B, A)	45	65	35		**Group 6** (H)	15			
Group 3					**Group 7**				
Group 4 (J, G)	30	80	80		**Group 8** (D)	65	(easy bend)		
Group 8					**Group 10**				
Group 9					**Group 11**				
Group 11					**Group 12**				
Group 12 (O)		5	5		**Other**				
Other					**Other**				
Total Length	133	255	241		**Total Length**	120			

Figure 7-13

Duct Sizing Calculations

Figure 7-15 (ahead two pages) summarizes the duct sizing calculations. Additional detail is provided here:

- The sum of the branch cooling Cfm values is greater than the blower Cfm value (see Sections 6-8 through 6-12.).
- Supply runs 1 through 5 are for trunk ST-2 and supply runs 6 through 9 are for trunk ST-1.
- The supply trunk ST-1 is a secondary trunk, supply trunk ST-2 is a primary trunk and the return trunk is a primary trunk (see Section 6-14).
- The entering Cfm value for ST-1 equals the sum of the downstream branch Cfm values, the entering Cfm value for ST-2 equals the blower Cfm value, and the return trunk Cfm value equals the blower Cfm value (see Section 6-15.)
- All values in the round-size column were read from the Galvanized Metal Duct scale on the ACCA Duct Sizing Slide Rule.
- Final airway size for supply runouts is based on the design friction rate (0.06 IWC/100) because air velocities are less than 900 Fpm limit.
- Final airway size for supply trunks ST-1 and ST-2 is based on the friction rate design value (0.06 IWC/100) because air velocities are less than 900 Fpm limit.
- Final airway size of return trunk RT-1 is based on the friction rate design value (0.06 IWC/100) because the 720 Fpm air velocity is close to the allowable velocity (700 Fpm).
- The ACCA Duct Slide Rule converts round sizes to equivalent rectangular sizes.

Comments and Observations

The Figure 7-12 duct system is similar to the Figure 7-8 duct system, except for the position of the furnace and the electronic filter. These two changes reduced the design friction rate (0.10 IWC/100 to 0.06 IWC/100), increased the total effective length (375 feet versus 327 feet) and decreased in the available static pressure (0.23 IWC versus 0.33 IWC). Since airway size depends on the design friction rate, the 0.06 IWC/100 value caused a one inch increase in runout sizes and a one or two inch increase in the supply trunk sizes (in both cases, return ducts are sized to satisfy the 700 Fpm velocity limit).

Note that the 0.06 friction rate value is barely in the wedge on the friction rate chart. This friction rate could be increased by designing for a higher blower wheel speed or by reducing the total effective length of the critical circulation path. In this regard, the best approach is to change the inefficient branch takeoff and boot fittings for trunk 2. Figure 7-16 (ahead three pages) shows that when 2B branch takeoffs and 4J boots are substituted for the 2A and 4G fittings, total effective length is reduced by 75 feet (300 feet versus 375 feet).

When total effective length is decreased, there is a corresponding increase in the design friction rate. As indicated

Friction Rate Worksheet for the Reducing Plenum Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.67 IWC   Cfm = 1,000

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	0.25
Electric resistance heating coil	
Hot water coil	
Heat exchanger	
Low efficiency filter	
High or mid-efficiency filter	
Electronic filter	0.10
Humidifier, UV lights, other	
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	
Total component losses (CPL)	0.44 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (0.67 - 0.44) = 0.23 IWC

Step 4) Total Effective Length (TEL)

Supply-Side TEL + Return-Side TEL = (255 + 120) = 375 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.06 IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

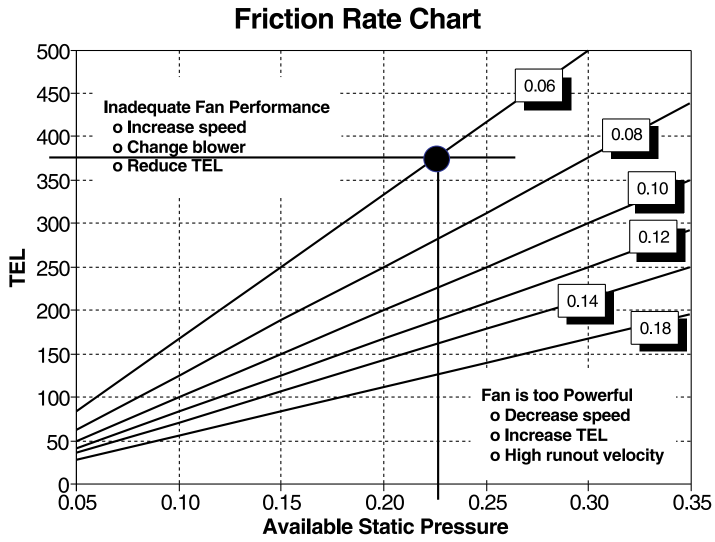


Figure 7-14

Duct Sizing Worksheet for the Reducing Plenum Example									
HF = Blower Cfm / Manual J Heat Loss = 1,000 / 36,000 = 0.278 **CF = Blower Cfm / Manual J Sensible Heat Gain =** 1,000 / 24,000 = 0.417									**FR Value** 0.06
Supply-Side Runouts									
Supply - Trunk	**Heating Btuh**	**Cooling Btuh**	**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
S1 — ST-2	3,810	3,110	106	130	130	7	ok	7	113
S2 — ST-2	3,800	2,380	106	99	106	7	ok	7	98
S3 — ST-2	3,970	3,200	110	133	133	7	ok	7	117
S4 — ST-2	4,250	2,750	118	115	118	7	ok	7	112
S5 — ST-2	3,860	3,010	107	125	125	7	ok	7	111
S6 — ST-1	4,500	2,610	125	109	125	7	ok	7	112
S7 — ST-1	4,590	3,500	128	146	146	8	ok	8	132
S8 — ST-1	4,870	3,750	135	156	156	8	ok	8	140
S9 — ST-1	2,350	1,690	65	70	70	6	ok	6	65
Supply-Side Trunks									
Run numbers: S6 through S9		**S-Trunk 1**	453	481	481	12	630	12	551
Run numbers: Primary trunk		**S-Trunk 2**	1,000	1,000	1,000	16	720	16	449
Return-Side Runouts One Central Return with transfer grilles, no branches.									
Return-Side Trunks									
Run numbers: Primary trunk		**R-Trunk 1**	1,000	1,000	1,000	16	720	16	1,000

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 7-15

Figure 7-17 (next page), the new friction rate (0.075 IWC/100 approximate) is well inside the wedge on the friction rate chart.

7-4 Primary and Secondary Trunk System with an Operating Point Blower

Figure 7-18 (ahead two pages) shows a heat pump blower delivering 1,500 Cfm to a duct system that has duct board trunks and metal runouts. The primary supply trunk has a reducing fitting to a secondary trunk, then this trunk tees to two more secondary trunks. The return path has four return branches and two return trunks.

Figure 7-19 (ahead two pages) shows the ***Manual J*** heating and cooling loads for calculating Cfm values for supply air outlet selection, and for duct airway sizing. Figure 7-20 (ahead three pages) provides the equipment manufacturer's blower table and component pressure drop data for an operating point blower.

Effective Length Calculation

Considering system geometry and fitting types, it appears that runs 1, 5, 9 and 11 are candidates for the longest supply run. The corresponding return runs are R3, R2, R1 and R4. Figure 7-21 (ahead three pages) provides effective length calculations for these runs. The Figure 7-22 (ahead three pages) summary shows that the effective length of the critical circulation path is 446 feet. This path flows through the #11 supply (187 feet) and the R4 return (259 feet).

Design Friction Rate Calculation

For this example, heating and cooling is provided by an air-to-air heat pump. The blower data for medium speed, shows the blower will deliver 1,500 Cfm when it operates against 0.32 IWC of external resistance. Since the resistance produced by the auxiliary heating coil (0.08 IWC), a supply outlet (0.03 IWC), a return (0.03 IWC) and a hand damper (0.03 IWC) is 0.17 IWC, the available static pressure is 0.15 IWC. Therefore, the design friction rate is

Alternative Effective Length Worksheet for the Reducing Plenum Example									
Element	**Supply Run ID Number**				**Element**	**Return Run ID Number**			
	S1	S6	S9			RT-1			
Trunk Length	6	43	66		**Trunk Length**	10			
Trunk Length					**Trunk Length**				
Runout Length	12	22	15		**Runout Length**				
Group 1 (L)	40	40	40		**Group 5** (I)	30			H/W=2
Group 2 (B)	45	40	20		**Group 6** (H)	15			
Group 3					**Group 7**				
Group 4 (J)	30	30	30		**Group 8** (D)	65	(easy bend)		
Group 8					**Group 10**				
Group 9					**Group 11**				
Group 11					**Group 12**				
Group 12 (J)		5	5		**Other**				
Other					**Other**				
Total Length	133	180	176		**Total Length**	120			

Figure 7-16

based on 0.15 IWC of pressure and 446 feet of effective length. These calculations are made on Figure 7-23 (ahead three pages), which shows that the blower cannot produce adequate static pressure at medium speed.

There are two ways to deal with inadequate blower performance, either increase the blower wheel speed or reduce the total effective length. The second option is preferred, but the first option is applied to this example because the fittings are reasonably efficient.

At high speed, the blower will deliver 1,500 Cfm when it operates against 0.53 IWC of external resistance. When 0.17 IWC of component pressure drop is subtracted from this value, the design friction rate is based on 0.36 IWC of pressure and 446 feet of effective length. Figure 7-23 (ahead three pages) shows that the adjusted friction rate is 0.08 IWC/100.

Duct Sizing Calculations

Figure 7-24 (ahead four pages) summarizes the duct sizing calculations for this example. Additional detail is provided here:

- Supply runs 1 through 4 are for primary trunk ST-4, supply runs 5 through 8 are for primary trunk ST-3, supply runs 11 through 13 are for secondary trunk ST-2, and supply runs 9 and 10 are for secondary trunk ST-1.
- Supply runs 9 and 10 are for return R1, supply runs 5 through 8 are for return R2, supply runs 1 through 4 are for return R3 and supply runs 11 through 13 are for return R4.

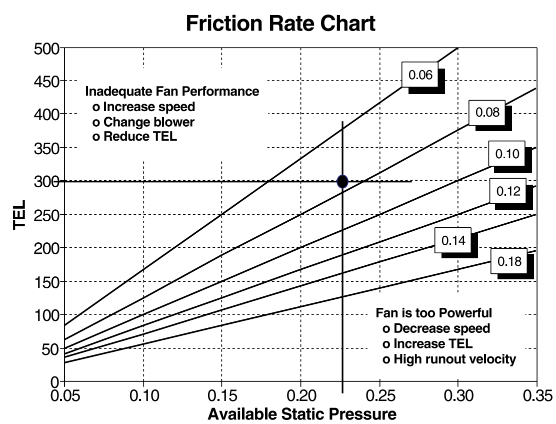


Figure 7-17

- Return runs R1 and R2 feed return trunk RT-1, and return runs R3 and R4 feed return trunk RT-2.
- The sum of the branch cooling Cfm values is greater than the blower Cfm value (see Sections 6-8 through 6-12).
- Supply trunks ST-1, ST-2 and ST-3 are secondary trunks, and supply trunk ST-4 is a primary trunk. Return trunks RT-1 and RT-2 are secondary trunks (see Section 6-14).
- The entering Cfm value for ST-1, ST-2 and ST-3 equals the sum of the downstream branch Cfm

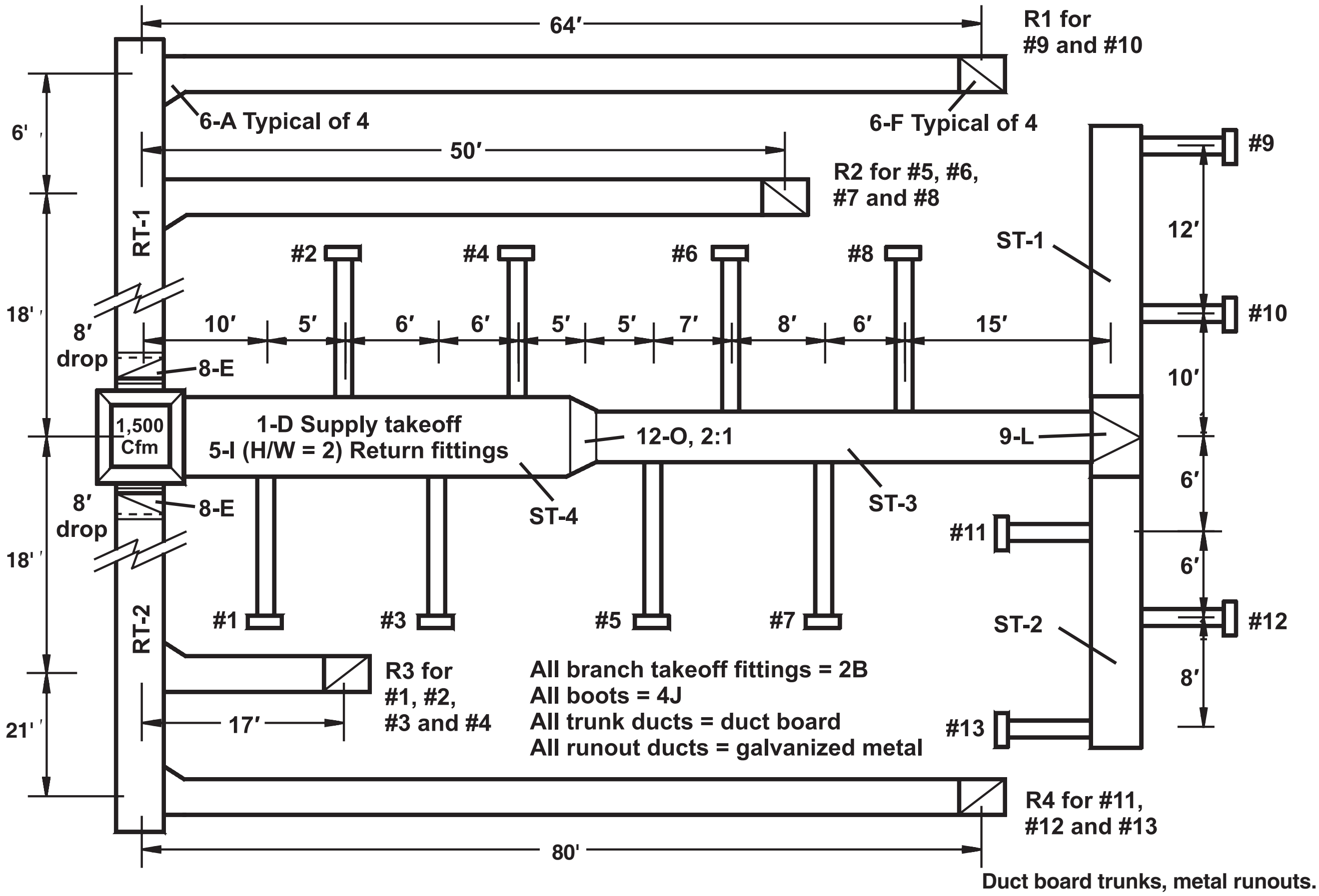


Figure 7-18

values. The entering Cfm value for ST-4 equals the blower Cfm value. The entering Cfm value for RT-1 and RT-2 equals the sum of the upstream branch Cfm values (see Section 6-15).

- All supply branch runout sizes were read from the Galvanized Metal Duct scale on the ACCA Duct Sizing Slide Rule.
- All supply trunk sizes and return duct sizes were read from the duct board scale on the ACCA Duct Sizing Slide Rule.
- Final airway size for supply runouts is based on the design friction rate (0.08 IWC/100) because air velocities are less than 900 Fpm limit.
- Final airway size for supply trunks ST-1, ST-2, ST-3 and ST-4 is based on the design friction rate (0.08 IWC/100) because air velocities are less than 900 Fpm limit.
- Final airway size for return branch ducts R1 and R4 is based on the design friction rate (0.08 IWC/100) because air velocities are less than the 700 Fpm limit.
- Final airway size for return branch ducts R3 and R5 is based on the maximum allowable velocity

Duct Lengths and *Manual J* Loads for the Primary-Secondary Example

Runout	Length - Ft	Heat Btuh	Cool Btuh
S1	16	4,250	2,750
S2	14	3,860	3,010
S3	16	3,970	3,200
S4	14	2,780	2,130
S5	17	3,800	2,380
S6	16	4,440	3,420
S7	17	4,590	3,500
S8	16	4,620	3,510
S9	12	2,350	1,690
S10	12	3,020	2,590
S11	8	3,810	3,110
S12	12	3,430	2,400
S13	8	4,500	2,610

1) Block loads (Btuh): Heat= 44,920; Sensible cooling = 32,670
2) For cooling, fenestration excursion loads cause the sum of the room cooling loads to be more than the block cooling load (see Sections 6-8 through 6-12).

Figure 7-19

Blower Data for the Primary-Secondary Example			
Discharge Cfm	External Resistance (IWC) vs. Speed		
	High	Medium	Low
1,200			0.58
1,300		0.62	0.43
1,400	0.68	0.47	0.27
1,500	0.53	0.32	0.12
1,600	0.38	0.15	
1,700	0.20		

1) Operating point blower tested with a wet refrigerant coil and low efficiency filter in place.
2) If an auxiliary heating coil is required, subtract 0.08 IWC from the values that are listed in this table.

Figure 7-20

Summary of Figure 7-21 Calculations				
Run	**S1**	**S5**	**S9**	**S11**
Supply TEL	106	139	192	187
Run	**R3**	**R2**	**R1**	**R4**
Return TEL	162	226	228	259

Figure 7-22

(700 Fpm) because sizing for the design friction rate (0.08 IWC/100) produced excessive air velocity.

- Final airway size for return trunks RT-1 and RT-2 is based on the maximum allowable velocity (700 Fpm) because sizing for the design friction rate (0.08 IWC/100) produced excessive air velocity.
- Figure 7-25 (ahead three pages) provides a summary of the Group 6A equivalent length values.
- The ACCA Duct Slide Rule converts round sizes to equivalent rectangular sizes.

Comments and Observations

This example is characterized by marginal blower performance and a large effective length value. But as demonstrated above, there was enough blower power (at the high speed setting) for the required air flow.

This example also is characterized by an absence of accessory components, which results in a practical solution for airway sizes. If a simple device, such as an electronic filter, is added to the system, the blower has marginal

Effective Length Worksheet for Primary-Secondary Example									
Element	**Supply Run ID Number**				**Element**	**Return Run ID Number**			
	S1	S5	S9	S11		R1	R2	R3	R4
Trunk Length	10	37	95	79	**Trunk Length**	24	18	18	39
Trunk Length					**Trunk Length**				
Trunk Length					**Trunk Length**				
Runout Length	16	17	12	8	**Runout Length**	64	50	17	80
Group 1 (D)	10	10	10	10	**Group 5** (I)	30	30	30	30
Group 2 (B)	40	40	20	35	**Group 6** (A br)	75	68	37	75
Group 3					**Group 7**				
Group 4 (J)	30	30	30	30	**Group 8** (E)	10	10	10	10
Group 8					**Group 10**				
Group 9 (L)			20	20	**Group 11**				
Group 11					**Group 12**				
Group 12 (O)		5	5	5	**Other** (6A main)		25	25	
Other					**Other** (6F)	25	25	25	25
Other					**Other**				
Total Length	106	139	192	187	**Total Length**	228	226	162	259

Figure 7-21

Friction Rate Worksheet for Primary-Secondary Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = ~~0.32~~ 0.53 IWC   Cfm = 1,500

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	
Electric resistance heating coil	0.08
Hot water coil	
Heat exchanger	
Low efficiency filter	
High or mid-efficiency filter	
Electronic filter	
Humidifier, UV lights, other	
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	
Total component losses (CPL)	0.17 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (~~0.32~~ 0.53 - 0.17) = ~~0.15~~ 0.36 IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (187 + 259) = 446 Feet

Step 5) Friction Rate Design Value (FR)

0.08 @ 0.36 IWC

FR value from friction rate chart = Out of range @ 0.15 IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

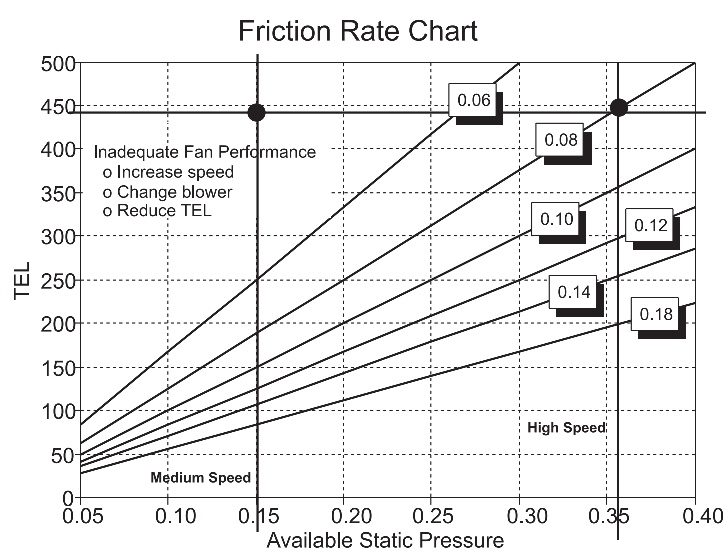


Figure 7-23

Duct Sizing Worksheet for the Primary-Secondary Example - Operating Point Blower

HF = Blower Cfm / Manual J Heat Loss = 1,500 / 49,420 = 0.0304
CF = Blower Cfm / Manual J Sensible Heat Gain = 1,500 / 32,670 = 0.0459

FR Value 0.08

Supply-Side Runouts

Supply - Trunk	Heating Btuh	Cooling Btuh	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
S1 — ST-4	4,250	2,750	129	126	129	7	ok	7	121
S2 — ST-4	3,860	3,010	117	138	138	7	ok	7	121
S3 — ST-4	3,970	3,200	120	147	147	7	ok	7	127
S4 — ST-4	2,780	2,130	84	98	98	6	ok	6	86
S5 — ST-3	3,800	2,380	115	109	115	7	ok	7	106
S6 — ST-3	4,440	3,420	135	157	157	7	ok	7	138
S7 — ST-3	4,590	3,500	139	161	161	7	ok	7	142
S8 — ST-3	4,620	3,510	140	161	161	7	ok	7	143
S9 — ST-1	2,350	1,690	71	78	78	6	ok	6	71
S10 — ST-1	3,020	2,590	92	119	119	6	ok	6	100
S11 — ST-2	3,810	3,110	116	143	143	7	ok	7	123
S12 — ST-2	3,430	2,400	104	110	110	6	ok	6	101
S13 — ST-2	4,500	2,610	137	120	137	7	ok	7	122

Supply-Side Trunks

Run numbers	Trunk	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
Run numbers: S9, S10	**S-Trunk 1**	163	197	197	8	570	8	171
Run numbers: S11, S2, S13	**S-Trunk 2**	357	373	373	10	690	10	346
Run numbers: S5 to S13	**S-Trunk 3**	1,049	1,157	1,157	16	825	16	1,046
Run numbers: Primary trunk	**S-Trunk 4**	1,500	1,500	1,500	18	850	18	1,500

Return-Side Runouts

Return - Trunk	Associated Supply Runs	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
R1 — RT-1	S9, S10	163	197	197	9	462	9	171
R2 — RT-1	S5, S6, S7, S8	529	588	588	12	775	14 std	529
R3 — RT-2	S1, S2, S3, S4	450	509	509	11	800	12	455
R4 — RT-2	S11, S12, S13	357	373	373	10	680	10	346

Return-Side Trunks

Run numbers	Trunk	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
Run numbers: R1 and R2	**R-Trunk 1**	692	785	785	14	750	16	700
Run numbers: R3 and R4	**R-Trunk 2**	807	882	882	14	840	16	801

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 7-24

ability (even at the high speed setting) to deliver the required air flow.

For example, if an electronic filter adds an additional 0.10 IWC to the component pressure loss calculation, the available static pressure is 0.26 IWC instead of 0.36 IWC. Under these circumstances, Figure 7-26 (next page) shows the design friction rate is less than 0.06 IWC/100, which is slightly outside the wedge. This problem is corrected by using equipment that has a more powerful blower.

7-5 Primary and Secondary Trunk System with an Operating Range Blower

Figure 7-18 shows a heat pump operating range blower delivering 1,500 Cfm to a duct system that has duct board trunks and metal runouts. The primary supply trunk has a reducing fitting to a secondary trunk, then this trunk tees to two more secondary trunks. The return path has four return branches and two return trunks.

Figure 7-19 summarizes the heating and cooling loads for the runout ducts and Figure 7-27 provides blower data for an operating range blower. When this equipment is shipped the blower cabinet will have a refrigerant coil, an electric resistance heater and an electronic filter.

Effective Length Calculation

Figure 7-21 provides effective length calculations for the duct runs. The Figure 7-22 summary shows that the effective length of the critical circulation path is 451 feet. This path flows through the #9 supply (192 feet) and the R4 return (259 feet).

Blower Cfm Set-Point

The ***Manual S*** design value for blower Cfm is 1,500 Cfm. From Figure 7-27, the two closest blower table set point options are 1,400 and 1,750 Cfm. Since the desired 1,500 Cfm set point is not available, 1,750 Cfm is used to size airways (see Section 3-3 this manual).

Blower Table Pressure

The performance of the ECM blower is summarized by Figure 7-27. In this case, footnotes advise that the blower data is not adjusted for air-side components installed at the factory, and that the pressure drop for such components shall be subtracted from the published external static pressure values.

Summary of Group 6A Equivalent Lengths				
Summary	**R1**	**R2**	**R3**	**R4**
Cfm1 / Cfm 2	1.00	0.75	0.56	1.00
Branch EL	75	68	33	75
Main EL	NA (0)	25	25	NA (0)

Figure 7-25

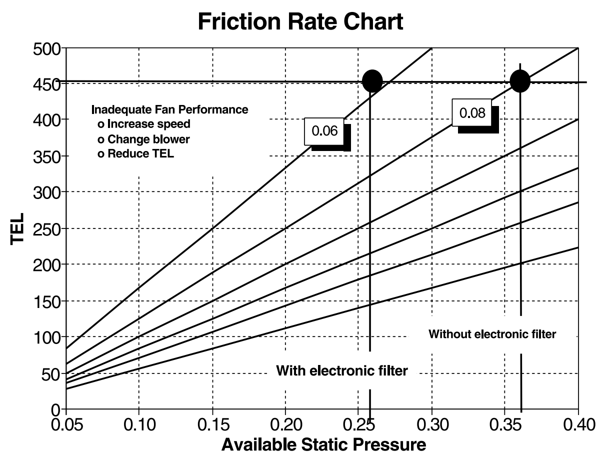


Figure 7-26

Component Pressure Drop

The heat pump will be shipped with a standard refrigerant coil (there are other coil options), an electric resistance coil and an electronic filter. Figure 2-28 (next page) shows the manufacturer's pressure drop data for these components. The total pressure drop for components installed in the cabinet is 0.65 IWC, and additional pressure is required for a supply air outlet, a return grille and a hand

ECM Blower Data for the Primary-Secondary Example												
Function	**Cfm Set Point**	**ESP (IWC) Range**	**External Static Pressure (IWC)**									
			0.10	**0.20**	**0.30**	**0.40**	**0.50**	**0.60**	**0.70**	**0.80**	**0.90**	**1.00**
Cooling Cfm Set Points	875	0.0 – 0.5	875									
	1,050	0.0 – 1.0	1,050						1,045			1,035
	1,225	0.0 – 1.0	1,215	1,220			1,225					1,220
	1,400	0.0 – 1.0	1,370	1,385		1,395				1,400	1,395	1,390
	1,750	0.0 – 1.0	1,750						1,745	1,740	1,735	1,725
Maximum	2,000	0.0 – 1.0	2,000				1,990	1,975	1,950	1,925	1,900	1,865

1) No OEM blower table notes pertaining to air-side components in place when the blower was tested.
2) Per other parts of the OEM's engineering data, deduct pressure drop for wet refrigerant coil, filter and electric heating coil.

Figure 7-27

damper, so the pressure drop for all system components is 0.74 IWC.

Cfm = 1,750
Refrigerant coil = 0.35 IWC
Electric coil = 0.18 IWC
Filter = 0.12 IWC
Total pressure drop for cabinet items = 0.65 IWC
Supply air outlet = 0.03 IWC
Return Grille = 0.03 IWC
Hand damper = 0.03 IWC
Total pressure drop for all components = 0.74 IWC

Default for Available Static Pressure

For an operating range blower, the default value for available static pressure is 70% (0.70 factor) of the maximum external static pressure value from the OEM's blower table (see Section 6-4). Since the maximum pressure for the OEM's blower table for the 1,725 Cfm setting is 1.0 IWC, the external static pressure for the Friction Rate Worksheet is 0.70 IWC.

Total System Resistance

The total airflow resistance produced by air-side components for a 1,725 Cfm flow rate is 0.74 IWC. Since the default external static pressure value for the Friction Rate Worksheet is 0.70 IWC, the available pressure for just the air-side components is slightly deficient.

However, and addition increment of pressure is required for the fittings and straight runs in the critical circulation path. The pressure required for the path depends on the friction rate design value. For the lowest value recommended by the Friction Rate Worksheet (0.06 IWC per 100 feet of duct), an additional 0.27 IWC is required for fittings and duct runs.

0.27 IWC = 0.06 x 451 / 100

Therefore, the total amount of external static pressure required for all pressure dissipating items in the critical circulation path is 1.01 IWC for 1,725 Cfm.

1.01 IWC = 0.74 for components + 0.27 ducts

This 1.00 IWC maximum available pressure vs. 1.01 IWC required pressure scenario is close enough to be a mathematically correct solution for airway sizing, but a design value for external static pressure that exceeds 70% of the maximum value is not recommended, per Section 6-4 guidance. Therefore, something must change.

System Design Adjustments

The first measure to reduce system airflow resistance is to shorten the length of the critical circulation path. The best way to do this is to use more efficient duct fittings. In this case, the Figure 7-21 duct fittings are relatively efficient, so other measures are investigated.

Coil Resistance (IWC)		
Cfm	Dry	Wet
1,225	0.11	0.18
1,400	0.15	0.26
1,750	0.22	0.35
2,000	0.28	0.46

Electronic Filter Resistance	
Cfm	IWC
1,225	0.06
1,400	0.08
1,750	0.12
2,000	0.15

Heater Resistance	
Cfm	IWC
1,225	0.09
1,400	0.13
1,750	0.18
2,000	0.23

Figure 7-28

The design blower Cfm for ***Manual S*** equipment selection was 1,500 Cfm. Figure 7-27 shows the closest values for the ECM blower are 1,400 Cfm and 1,750 Cfm. Per Section 3-3, the 1,750 value was used for duct design. If the system is designed for 1,400 Cfm, Figure 7-28 shows that the wet coil pressure drop is reduced by 0.09 IWC, the electronic filter pressure drop is reduced by 0.04 IWC and the electric heater pressure drop is reduced by 0.05 IWC, for a total reduction of 0.18 IWC. This reduces the pressure requirement from 1.01 to 0.83, but 0.83 exceeds the desired 0.70 value.

Total ESP required = 1.01 - 0.18 = 0.83 IWC

Find similar equipment that has a blower table that shows a maximum of 1.0 IWC of static pressure for something near 1,500 Cfm with footnotes that say a wet cooling coil, and standard filter were in place during the blower test (see Figure 7-29, next page). This will reduce the component pressure drop for Step-2 of the Friction Rate Worksheet from 0.74 IWC to something like 0.35 IWC, and the total system resistance for the components, duct

<table>
<tr><th colspan="13">ECM Blower Data for the Primary-Secondary Trunks</th></tr>
<tr><th rowspan="2">Function</th><th rowspan="2">Cfm Set Point</th><th rowspan="2">ESP (IWC) Range</th><th colspan="10">External Static Pressure (IWC)</th></tr>
<tr><th>0.10</th><th>0.20</th><th>0.30</th><th>0.40</th><th>0.50</th><th>0.60</th><th>0.70</th><th>0.80</th><th>0.90</th><th>1.00</th></tr>
<tr><td rowspan="5">Cooling Cfm Set Points</td><td>875</td><td>0.0 – 0.5</td><td colspan="10">875</td></tr>
<tr><td>1,050</td><td>0.0 – 1.0</td><td colspan="6">1,050</td><td colspan="3">1,045</td><td>1,035</td></tr>
<tr><td>1,225</td><td>0.0 – 1.0</td><td>1,215</td><td colspan="3">1,220</td><td colspan="5">1,225</td><td>1,220</td></tr>
<tr><td>1,400</td><td>0.0 – 1.0</td><td>1,370</td><td colspan="2">1,385</td><td colspan="4">1,395</td><td>1,400</td><td>1,395</td><td>1,390</td></tr>
<tr><td>1,750</td><td>0.0 – 1.0</td><td colspan="6">1,750</td><td>1,745</td><td>1,740</td><td>1,735</td><td>1,725</td></tr>
<tr><td>Maximum</td><td>2,000</td><td>0.0 – 1.0</td><td colspan="4">2,000</td><td>1,990</td><td>1,975</td><td>1,950</td><td>1,925</td><td>1,900</td><td>1,865</td></tr>
<tr><td colspan="13">1) Wet refrigerant coil and standard filter in place when the blower was tested.
2) Includes an allowance for supplemental electric heater (add 0.10 IWC to blower table values if there is no supplemental heater, or subtract the portion of the supplemental heater pressure drop that exceeds 0.10 IWC).</td></tr>
</table>

Figure 7-29

fittings and straight runs will drop form 1.01 to about 0.72, which is acceptable.

Total System Resistance for Alternative Equipment

If Figure 7-29 is the blower table for alternative equipment. The component pressure drop for the Friction Rate Worksheet is 0.21 IWC, based on Figure 7-28 pressure drops for an electric coil and an electronic filter (0.10 IWC is assumed to apply to a standard filter).

Cfm = 1,750
Refrigerant coil in place for blower test = 0 .0 IWC
Electric coil at 0.18 IWC = 0.18 - 0.10 = 0.08 IWC
Electronic vs. standard filter = 0.14 - 0.10 = 0.04 IWC
Total pressure drop for cabinet items = 0.12 IWC
Supply air outlet = 0.03 IWC
Return Grille = 0.03 IWC
Hand damper = 0.03 IWC
Total pressure drop for all components = 0.21 IWC

Design Friction Rate for Alternative Equipment

The default for external static pressure is 0.70 IWC and the total component pressure drop is 0.21 IWC, so the pressure for moving air through the critical circulation path is 0.49 IWC.

Per Figure 7-23, Step-4, the TEL of the critical circulation path is 446 Feet. Per the equation provided by Step-5 on the Friction Rate Worksheet, the friction rate value for airway sizing is 0.11 IWC.

FR = 0.49 x 100 / 446 = 0.11 IWC

Airway Size for Alternative Equipment

Figure 7-30 (next page) shows the airways sizes for the Figure 7-18 duct system and the Figure 29 blower. This solution is almost identical to the Figure 7-24 solution for the operating point blower because there is not much difference in the friction rate design values (0.08 for Figure 7-24 vs. 0.11 for Figure 7-30). The shaded cells on Figure 7-30 mark the difference between the two duct sizing worksheets.

7-6 Scrutinize Blower Data Footnotes

The Section 7-5 example for an operating range blower demonstrates the importance of blower table footnotes. If the Figure 7-27 blower is used for the Figure 7-18 duct system, the blower will have to operate very near to its external static pressure limit, with the airways sized for an 0.06 friction rate. If the Figure 7-29 blower is used for the Figure 7-18 duct system, the blower will operate at about 70% of its external static pressure limit, with the airways sized for an 0.11 friction rate. The Figure 7-29 blower is clearly the preferred solution for the Figure 7-18 duct system.

The blower table footnote issue is not peculiar to any particular type of blower motor technology. Each equipment manufacturer has their own format for presenting blower data. Some publish pressure values that have not been discounted for standard cabinet components and some publish discounted pressure values. Figure 7-29 provides an exhibit of discounted blower data. This blower is considerably more powerful than the Figure 7-27 blower.

- The Figure 7-29 blower produces about 1.5 IWC of blower pressure (0.5 IWC for cabinet components and 1.0 IWC of external static pressure for air distribution system).
- The Figure 7-27 blower produces 1.00 IWC of external static pressure for all cabinet components and the air distribution system, which means that

Duct Sizing Worksheet for the Primary-Secondary Example - Operating Range Blower									
HF = Blower Cfm / Manual J Heat Loss = 1,500 / 49,420 = 0.0304 **CF = Blower Cfm / Manual J Sensible Heat Gain =** 1,500 / 32,670 = 0.0459									**FR Value** 0.11
Supply-Side Runouts									
Supply - Trunk	**Heating Btuh**	**Cooling Btuh**	**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
S1 — ST-4	4,250	2,750	129	126	129	7	ok	7	121
S2 — ST-4	3,860	3,010	117	138	138	7	ok	7	121
S3 — ST-4	3,970	3,200	120	147	147	7	ok	7	127
S4 — ST-4	2,780	2,130	84	98	98	6	ok	6	86
S5 — ST-3	3,800	2,380	115	109	115	6	ok	6	106
S6 — ST-3	4,440	3,420	135	157	157	7	ok	7	138
S7 — ST-3	4,590	3,500	139	161	161	7	ok	7	142
S8 — ST-3	4,620	3,510	140	161	161	7	ok	7	143
S9 — ST-1	2,350	1,690	71	78	78	5	ok	5	71
S10 — ST-1	3,020	2,590	92	119	119	6	ok	6	100
S11 — ST-2	3,810	3,110	116	143	143	7	ok	7	123
S12 — ST-2	3,430	2,400	104	110	110	6	ok	6	101
S13 — ST-2	4,500	2,610	137	120	137	7	ok	7	122
Supply-Side Trunks									
Run numbers: S9, S10		**S-Trunk 1**	163	197	197	8	570	8	171
Run numbers: S12, S13		**S-Trunk 2**	357	373	373	10	690	10	346
Run numbers: S5 to S13		**S-Trunk 3**	1,049	1,157	1,157	16	825	16	1,046
Run numbers: Primary trunk		**S-Trunk 4**	1,500	1,500	1,500	18	850	18	1,500
Return-Side Runouts									
Return - Trunk	**Associated Supply Runs**		**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
R1 — RT-1	S9, S10		163	197	197	8	462	8	171
R2 — RT-1	S5, S6, S7, S8		529	588	588	12	775	14 std	529
R3 — RT-2	S1, S2, S3, S4		450	509	509	11	800	12	455
R4 — RT-2	S11, S12, S13		357	373	373	10	680	10	346
Return-Side Trunks									
Run numbers: R1 and R2		**R-Trunk 1**	692	785	785	14	750	16	700
Run numbers: R3 and R4		**R-Trunk 2**	807	882	882	14	840	16	801

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 7-30

there is about 0.50 IWC for the pressure drop through the critical circulation path.

Also note that Some manufactures test the blower to only 0.50 IWC, then use fan laws to extend the blower data beyond the 0.50 IWC. However, fan laws do not work very well if turbulence is produced by items in the air stream. In addition, some designs operate right next to the fan surge limit at 0.50 IWC, so a pressure above 0.50 IWC may cause fan surge. Therefore, have the OEM verify that the blower table is accurate for high pressure operation.

Sizing Flexible Constant Cfm Duct Systems

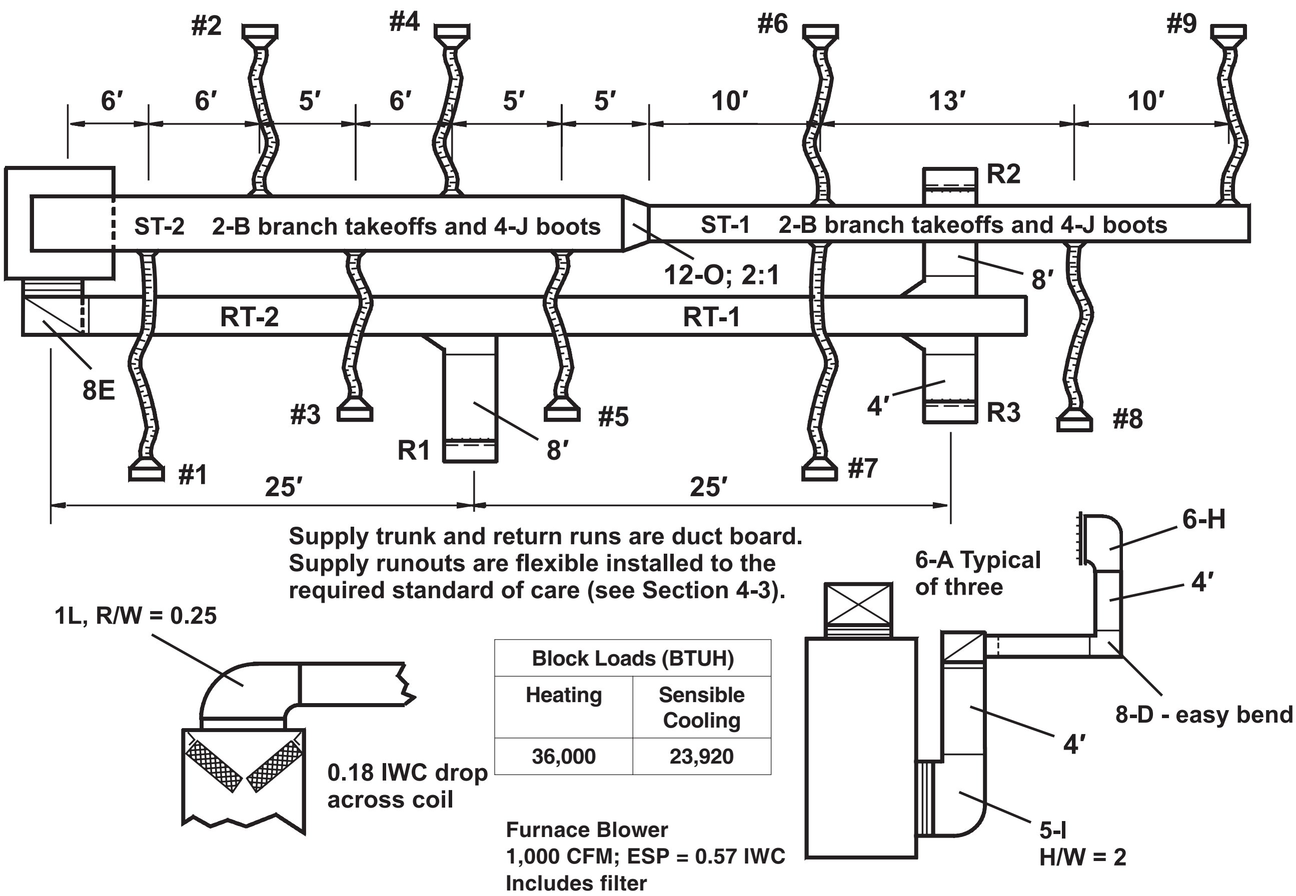
This section provides examples of airway sizing calculations for constant Cfm duct systems fabricated from flexible materials. These examples are for systems that comply with the required standard of care for installing flexible wire helix duct (see Section 4-3).

- Non compliant installations may have longer effective lengths than the measured point-to-point distance of the duct run, which causes larger pressure drops for duct runs.
- Appendix 16 provides tools for adjusting the effective length of installations that do not comply with the required standard of care for installing flexible wire helix duct.

The following examples are for an extended plenum system that has a rigid trunk and flexible runouts, and for a flexible wire helix system that has junction boxes. These examples are for single zone systems, so the standard ***Manual J*** procedure provides cooling load values for supply Cfm calculations (see Section 6-8, this manual).

8-1 Duct Board Trunk with Flex Runouts

Figure 8-1 shows a duct board, reducing-trunk plenum with flexible runouts. The flexible runouts have no significant excess length or significant sag. Heating and cooling is provided by a furnace equipped with an evaporator coil. The furnace has a standard throw away filter.



Block Loads (BTUH)	
Heating	Sensible Cooling
36,000	23,920

Figure 8-1

Duct Lengths and Manual J Loads for the Duct Board Trunk Flex Runout Example			
Run	**Length - Ft**	**Heat Btuh**	**Cool Btuh**
S1	22	3,810	3,110
S2	15	3,800	2,380
S3	12	3,970	3,200
S4	15	4,250	2,750
S5	12	3,860	3,010
S6	15	4,500	2,610
S7	22	4,590	3,500
S8	16	4,870	3,750
S9	15	2,350	1,690
R1	For supply runs 1, 2, 3, 4 and 5		
R2	For supply runs 6 and 9		
R3	For supply runs 7 and 8		

1) Block loads (Btuh): Heat = 36,000; Sensible cooling = 23,920
2) For cooling, fenestration excursion loads cause the sum of the room cooling loads to be more than the block cooling load (see Sections 6-8 through 6-12).

Figure 8-2

Figure 8-2 lists the heating and cooling loads for duct airway sizing, and Figure 8-3 provides the manufacturer's blower performance data.

Blower Data for the Duct Board Trunk Flex Runout Example			
Discharge CFM	**External Resistance (IWC) vs. Speed**		
	High	**Medium**	**Low**
800			0.53
900		0.65	0.45
1,000	0.63	0.57	0.37
1,100	0.48	0.49	0.29
1,200	0.33	0.41	
1,250	0.17		

1) Furnace blower tested with low efficiency filter in place.
2) If a cooling coil is required, subtract 0.18 IWC (wet coil) from the external resistance values that are listed in this table.

Figure 8-3

Effective Length Calculation

Considering system geometry and fitting types, runs 1, 7 and 9 are candidates for the longest supply run. Also note that the circulation path lengths are for supply run 1 and return R1, for supply run 9 and return R2, and for supply run 7 and return R3.

Figure 8-4 provides equivalent length calculations for these paths. Note that t here is no effective length adjustment for flexible runs that comply with the required

Effective Length Worksheet for the Duct Board Trunk Flex Runout Example								
Element	**Supply Run ID Number**			**Element**	**Return Run ID Number**			
	S1	S7	S9		R1	R2	R3	Notes
Trunk Length	6	43	66	**Trunk Length**	29	54	54	
Trunk Length				**Trunk Length**				
Trunk Length				**Trunk Length**				
Runout Length	22	22	15	**Runout Length**	12	12	8	
Group 1 (L)	40	40	40	**Group 5** (I)	30	30	30	(H/W = 2)
Group 2 (B)	45	40	20	**Group 6** (H)	15	15	15	
Group 3				**Group 7**				
Group 4 (J)	30	30	30	**Group 8** (D)	65	65	65	(ez bend)
Group 8				**Group 10**				
Group 9				**Group 11**				
Group 11				**Group 12**				Cfm1/Cfm2
Group 12 (O)		5	5	**Other** (6A br)	33	10	40	0.55 (R1) 0.39 (R2) 0.61 (R3)
Other				**Other** (6A m)	25	25	25	
Other				**Other** (8E)	10	10	10	
Total Length	143	180	176	**Total Length**	219	221	247	

Figure 8-4

standard of care (excess length 4% or less and negligible sag), so Figure 8-2 lengths are used for TEL calculations.

The Figure 8-5 summary shows the effective length of the critical circulation path is 427 feet. This path flows through the #7 supply (180 feet) and the R3 return (247 feet).

Design Friction Rate Calculation

Friction Rate Worksheet calculations determine the value for the design friction rate. For this example, airway sizes are based on 1,000 Cfm. Figure 8-3 shows that blower will deliver 1,000 Cfm when it operates against 0.57 IWC of external resistance and Figure 8-5 shows 247 feet for the critical circulation path.

Figure 8-6 (next page) shows that the total resistance for the evaporator coil (0.18 IWC), a supply outlet (0.03 IWC), a return (0.03 IWC), and a hand damper (0.03 IWC) is 0.27 IWC. Therefore, the available static pressure is 0.30 IWC, so the design friction rate is based on 0.30 IWC of pressure and 427 feet of effective length. Figure 8-6 shows the friction rate value for airway sizing is 0.07 IWC/100.

Duct Sizing Calculations

Figure 8-7 (ahead two pages) summarizes the duct sizing calculations for this example. Additional detail is provided here:

- Supply runs 1 through 5 are for supply trunk ST-2 and supply runs 6 through 9 are for supply trunk ST-1.
- Return runs R2 and R3 feed secondary return trunk RT-1.
- Return run R1 and return trunk RT-1 feed primary return trunk RT-2.
- The sum of the branch cooling Cfm values is greater than the blower Cfm value. (See Sections 6-8, 6-9 and 6-10.)
- Supply trunk ST-1 is a secondary trunk and supply trunk ST-2 is a primary trunk. Return trunk RT-1 is a secondary trunk and return trunk RT-2 is a primary trunk. (See Section 6-14.)
- The entering Cfm value for ST-1 equals the sum of the downstream branch Cfm values. The entering Cfm value for ST-2 equals the blower Cfm value. The entering Cfm value for RT-1 equals the sum of the upstream branch Cfm values. The entering Cfm value for RT-2 equals the blower Cfm values. (See Section 6-15.)
- All supply branch sizes were read from the Wire Helix Flexible Duct scale on the ACCA Duct Sizing Slide Rule.

Summary of Figure 8-4 Calculations			
Run	**S1**	**S7**	**S9**
Supply TEL	143	180	176
Run	**R1**	**R3**	**R2**
Return TEL	219	247	221

Figure 8-5

- All supply trunk sizes and the return duct sizes were read from the duct board scale on the ACCA Duct Sizing Slide Rule.
- Final airway sizes for supply runouts are based on the friction rate design value (0.07 IWC/100) because air velocities are less than 900 Fpm limit.
- Figure 8-3 shows that blower will deliver 1,000 Cfm when it operates against 0.57 IWC of external resistance. Final airway sizes for supply trunks ST-1 and ST-2 are based on the design friction rate (0.07 IWC/100) because air velocities are less than 900 Fpm limit.
- Final airway sizes for return branch ducts R1, R2 and R3 are based on the design friction rate value (0.07 IWC/100) because air velocities are less than the 700 Fpm limit.
- Final airway size for return trunk RT-1 is based on the friction rate design value (0.07 IWC/100) because air velocities is less than 700 Fpm.
- Final airway size for return trunk RT-2 is based on the maximum allowable velocity (700 Fpm) because the air velocity for the design friction rate slightly exceeds the limit.
- The R3 return run is longer than the R2 return run because the group 6A (branch) equivalent length value for the R3 run is 30 feet more than the equivalent length value for the R2 run.
- The ACCA Duct Slide Rule converts round sizes to equivalent rectangular sizes.

Comments and Observations

This example is characterized by the absence of accessory components, which makes a viable design possible. If a simple device, such as an electronic filter was included in the original design, medium speed blower operation has marginal ability to deliver design air flow.

Therefore, if accessories were installed, the duct sizing calculations would have been based on a high blower wheel speed blower. Also note that the effective length of the duct system can be reduced by 55 feet if the 8D elbow fittings for the return branches (EL = 65) are replaced with elbow fittings that have turning vanes (EL = 10).

Friction Rate Worksheet for the Duct Board Trunk Flex Runout Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.57 IWC Cfm = 1,000

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	0.18
Electric resistance heating coil	
Hot water coil	
Heat exchanger	
Low efficiency filter	
High or mid-efficiency filter	
Electronic filter	
Humidifier, UV lights, other	
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper, full open	

Total component losses (CPL) 0.27 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (0.57 - 0.27) = 0.30 IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (180 + 247) = 427 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.07 IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

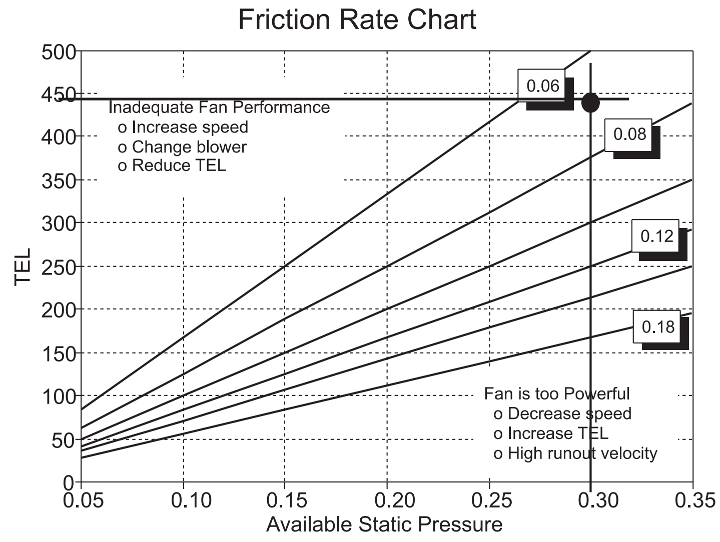


Figure 8-6

Duct Sizing Worksheet for the Duct Board Trunk Flex Runout Example

HF = Blower Cfm / Manual J Heat Loss = 1,000 / 36,00 = 0.0278 **CF = Blower Cfm / Manual J Sensible Heat Gain** = 1,000 / 23,920 = 0.0418	**FR Value** 0.07

Supply-Side Runouts									
Supply - Trunk	**Heating Btuh**	**Cooling Btuh**	**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
S1 — ST-2	3,810	3,110	106	130	130	8	ok	8	113
S2 — ST-2	3,800	2,380	106	99	106	7	ok	7	98
S3 — ST-2	3,970	3,200	110	134	134	8	ok	8	117
S4 — ST-2	4,250	2,750	118	115	118	7	ok	7	112
S5 — ST-2	3,860	3,010	107	126	126	8	ok	8	112
S6 — ST-1	4,500	2,610	125	109	125	8	ok	8	112
S7 — ST-1	4,590	3,500	128	146	146	8	ok	8	131
S8 — ST-1	4,870	3,750	135	157	157	8	ok	8	140
S9 — ST-1	2,350	1,690	65	71	71	6	ok	6	65
Supply-Side Trunks									
Run numbers: S6, S7, S8, S9		**S-Trunk 1**	453	483	483	11	720	11	448
Run numbers: Primary trunk		**S-Trunk 2**	1,000	1,000	1,000	15	830	15	1,000
Return-Side Runouts									
Return - Trunk	**Associated Supply Runs**		**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
R1 — RT-2	S1 through S5		547	604	604	12	760	12	552
R2 — RT-1	S6 and S9		190	180	190	8	570	8	205
R3 — RT-1	S7 and S8		263	303	303	10	580	10	243
Return-Side Trunks									
Run numbers: S6 through S9		**R-Trunk 1**	453	483	483	11	740	11	448
Run numbers: Primary trunk		**R-Trunk 2**	1,000	1,000	1,000	15	830	16	1,000

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 8-7

8-2 Flexible Wire Helix Duct System

On the next page, sketches and tables provide an example of a 900 Cfm flexible wire helix duct system (supply and return) with duct board junction boxes. Heating and cooling is provided by a heat pump equipped with a supplemental electric resistance heating coil and a standard throw away filter.

- Figure 8-8 (next page) shows the geometry of the duct system.
- Figure 8-9 (next page) lists heating and cooling loads for duct airway sizing.
- Figure 8-10 (next page) provides the manufacturer's blower performance data and a pressure drop for the electric heating coil.

Effective Length Calculation

Considering system geometry and fitting types, supply runs 7 and 8 are candidates for the longest supply run. Return R3 identifies the longest return path.

The there is no effective length adjustment for systems that comply with the required standard of care (excess length 4% or less, and negligible sag), so Figure 8-8 lengths are used for TEL calculations.

Note that junction box equivalent length depends on a reference velocity (see Appendix 3, Group 11). Before proceeding with effective length calculations, the practitioner selects a maximum air velocity for the duct system (600 Fpm for this example). Therefore, the Group 11 equivalent length for a junction box is 40 feet.

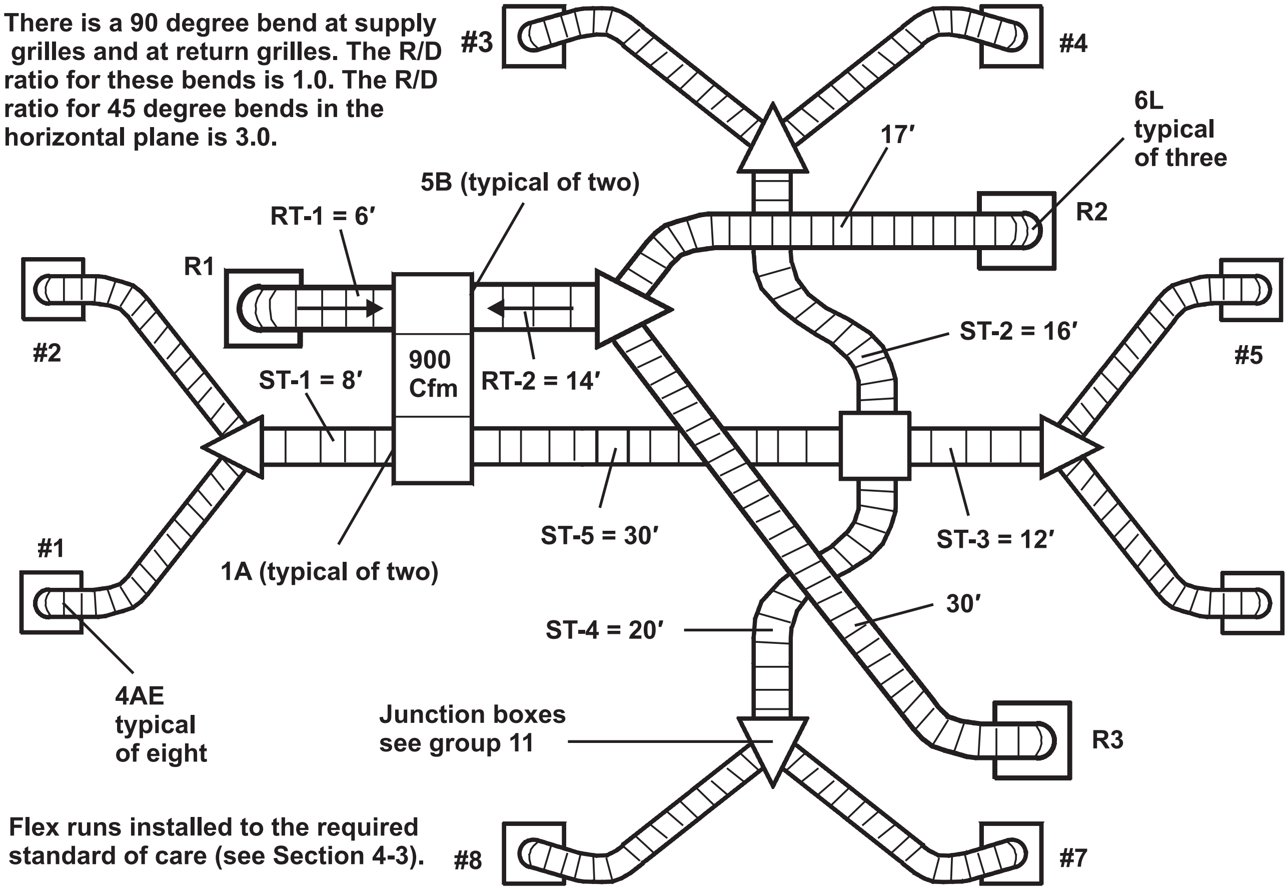


Figure 8-8

Duct Lengths and Manual J Loads for the Flexible Wire Helix System Example			
Run	**Length - Ft**	**Heat Btuh**	**Cool Btuh**
S1	16	3,800	2,380
S2	16	3,970	3,200
S3	14	4,250	2,750
S4	14	3,860	3,010
S5	14	4,500	2,610
S6	14	4,590	3,500
S7	14	4,870	3,750
S8	14	2,350	1,690
R1	For supply runs 1 and 2		
R2	For supply runs 3, 4 and 5		
R3	For supply runs 6, 7 and 8		

1) Block loads (Btuh): Heat = 32,190; Sensible cooling = 20,830
2) For cooling, fenestration excursion loads cause the sum of the room cooling loads to be more than the block cooling load (see Sections 6-8 through 6-12).

Figure 8-9

- The maximum recommended velocity for flexible duct is 900 Fpm (see Table A1-1), but this adds 55 feet to the equivalent length of a junction box (95 feet at 900 Fpm vs. 40 feet at 600 Fpm).

Blower Data for the Flexible Wire Helix System Example			
Discharge Cfm	**External Resistance (IWC) vs. Speed**		
	High	**Medium**	**Low**
750			0.49
800		0.58	0.41
850	0.65	0.50	0.33
900	0.60	0.42	0.25
950	0.45	0.34	
1,000	0.29		

1) Unit tested with low efficiency filter in place.
2) If resistance heating coils are required, subtract 0.12 IWC from the values that are listed in this table.

Figure 8-10

Effective Length Worksheet for the Flexible Wire Helix Example							
Element	**Supply Run:** S7 or S8			**Element**	**Return Run:** R3		
	Cut to Fit				Cut to Fit		
Trunk Length	30			**Trunk Length**	14		
Trunk Length	20			**Trunk Length**			
Trunk Length				**Trunk Length**			
Runout Length	14			**Runout Length**	30		
Group 1 (A)	35			**Group 5** (B)	40		
Group 2				**Group 6** (L)	20		
Group 3				**Group 7**			
Group 4 (AE)	55			**Group 8**			
Group 8				**Group 10**			
Group 9				**Group 11** Box 600	40		
Group 11 Box 600	2 x 40 = 80			**Group 12**			
Group 12				**Other**			
Other				**Other** Grp-11, 45	5		
Other Grp-11, 45	3 x 5 = 15			**Other** Grp-11, 90	10		
Other Grp-11, 90	10			**Other**			
Total Length	259			**Total Length**	159		

Figure 8-11

- Equivalent lengths for velocities that exceed 600 Fpm may used to size flexible duct airways, providing the blower can provide adequate external static pressure.

Figure 8-11 shows the equivalent length calculations for ducts that are cut to length (with an allowance for bends that function as fittings). The effective length of the longest supply run ranges is 259 feet and the effective length of the longest return run is 159 feet. Therefore, the total effective length of the critical circulation path is 418 feet.

Design Friction Rate Calculation

At medium speed, the blower delivers 900 Cfm when it operates against 0.42 IWC of external resistance. Since the resistance produced by the electric resistance coil (0.12 IWC), a supply outlet (0.03 IWC), a return (0.03 IWC) and a hand damper (0.03 IWC) is 0.21 IWC, the available static pressure is 0.21 IWC. Therefore, the design friction rate for the recommended standard of instillation care is based on 0.21 IWC of pressure and 418 feet of effective length. Figure 8-12 (next page) shows the design friction rate is less than 0.05 IWC/100, so medium blower wheel speed does not produce enough pressure.

There are two ways to deal with inadequate blower performance, increase wheel speed or reduce total effective length. For this example, the second option is investigated because junction boxes account for 120 feet of effective length.

If maximum air velocity is reduced to 500 Fpm, the equivalent length for a single junction box is reduced by 10 feet (from 40 feet to 30 feet). This reduces the total effective length of the critical circulation path by 30 feet (for three boxes).

Unfortunately, this option does not provide the desired result. As indicated by Figure 8-12, the design friction rate is still less than 0.06 IWC/100 when the total effective length is 388 feet. The next option is to use a 600 Fpm velocity limit and increase blower wheel speed.

At high speed, the blower delivers 900 Cfm when it operates against 0.60 IWC of external resistance. When external resistance (0.21 IWC) is subtracted from the new blower pressure, the design friction rate is based on 0.39 IWC of available pressure and 418 feet of effective length. These changes are made on Figure 8-12, which shows the design friction rate is 0.09 IWC/100 (rounded from 0.93) for the recommended standard of installation care.

Duct Sizing Calculations

Figure 8-13 (ahead two pages) shows the airway sizes for the recommended standard of installation care (duct runs cut to fit the span). Additional detail is provided here:

- Supply runs 3 and 4 are for secondary trunk ST-2; supply runs 5 and 6 are for secondary trunk ST-3; and supply runs 7 and 8 are for secondary trunk ST-4.

Friction Rate Worksheet for the Flexible Wire Helix Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = ~~0.42~~ **0.60** IWC   Cfm = 900

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	
Electric resistance heating coil	0.12
Hot water coil	
Heat exchanger	
Low efficiency filter	
High or mid-efficiency filter	
Electronic filter	
Humidifier, UV lights, other	
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	
Total component losses (CPL)	0.21 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (~~0.42~~ **0.60** - 0.21) = ~~0.21~~ **0.39** IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (259 + 159) = 418 Feet (600 Fpm velocity for boxes)

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = ~~0.05~~ **0.093 rounded to 0.09** IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

Friction Rate Chart

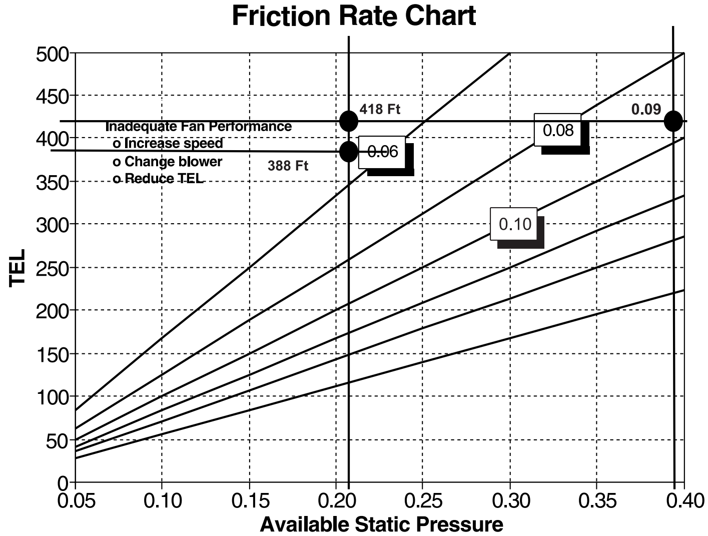


Figure 8-12

Duct Sizing Worksheet for the Flexible Wire Helix System Example

HF = Blower Cfm / Manual J Heat Loss = 900 / 32,190 = 0.0280
CF = Blower Cfm / Manual J Sensible Heat Gain = 900 / 20,830 = 0.0432

FR Value 0.09

Supply-Side Runouts

Supply - Trunk	Heating Btuh	Cooling Btuh	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
S1 — ST-1	3,800	2,380	106	103	106	7	About 600	7	100
S2 — ST-1	3,970	3,200	111	138	138	8		8	119
S3 — ST-2	4,250	2,750	119	119	119	7		7	113
S4 — ST-2	3,860	3,010	108	130	130	8		8	113
S5 — ST-3	4,500	2,610	126	113	126	7		7	114
S6 — ST-3	4,590	3,500	128	151	151	8		8	133
S7 — ST-4	4,870	3,750	136	162	162	8		8	142
S8 — ST-4	2,350	1,690	66	73	73	6		6	66

Supply-Side Trunks

Run numbers		Trunk	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
Run numbers:	S1, S2	**S-Trunk 1**	217	241	241	10	About 600	10	219
Run numbers:	S3, S4	**S-Trunk 2**	227	249	249	10		10	226
Run numbers:	S5, S6	**S-Trunk 3**	254	264	264	10		10	247
Run numbers:	S7, S8	**S-Trunk 4**	202	235	235	9		9	208
Run numbers:	S3 to S8	**S-Trunk 5**	683	748	748	15		16	681

Return-Side Runouts

Return - Trunk	Associated Supply Runs	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
R1 — RT-1	S1, S2	217	241	241	See RT1	See RT1	See RT1	219
R2 — RT-2	S3, S4, S5	353	362	362	11	Below 600	12 std	340
R3 — RT-3	S6, S7, S8	330	386	386	11		12	341

Return-Side Trunks

Run numbers		Trunk	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
Run numbers:	R1	**R-Trunk 1**	217	241	241	9	Below 600	9	219
Run numbers:	R2, R3	**R-Trunk 2**	683	748	748	15	About 600	16	681

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 8-13

- Supply runs 1 and 2 are for primary supply trunk ST-1; supply runs 3, 4, 5, 6, 7 and 8 are for primary supply trunk ST-5.
- Return branches R2 and R3 feed return trunk RT-2 and return R1 feeds return trunk RT-1. All duct sizes were read from the wire helix scale on the ACCA Duct Sizing Slide Rule.
- The sum of the branch cooling Cfm values is greater than the blower Cfm value. (See Sections 6-8, 6-9 and 6-10.)
- Supply trunks ST-1 through ST-5 are secondary trunks. Return trunks RT-1 and RT-2 are secondary trunks. (See Section 6-14.)
- The entering Cfm value for ST-1 through ST-5 equals the sum of the downstream branch Cfm values. The entering Cfm value for RT-1 and RT-2 equals the sum of the upstream branch Cfm values. (See Section 6-15.)
- A 600 Fpm velocity limit was imposed because a higher value produced a significant increase in the combined equivalent length of the junction box fittings.

- Final sizes of supply runouts are based on the friction rate design value (0.09 IWC/100) because air velocities are about 600 Fpm.
- The final sizes of the supply trunks (ST-1, ST-2, ST-3, ST-4 and ST-5) are based on the friction rate design value (0.09 IWC/100) because air velocities are about 600 Fpm.
- The final sizes of the return branch ducts (R2 and R3) are based on the friction rate design value (0.09 IWC/100) because air velocities are about 600 Fpm.
- The final sizes of the return trunks (RT-1 and RT-2) are based on the friction rate design value (0.09 IWC/100) because air velocities are about 600 Fpm or less.
- The final size of some duct runs were increased by 1-inch to conform with the standard sizes for the flex-duct product.

Comments and Observations

This duct system has a long effective length, but a relatively powerful blower (when operating at high speed), and the absence of accessory components make the design possible. If a simple device, such as an electronic filter (at 0.10 IWC), is added to the list of external components, the blower will be marginally compatible with the duct system (the design friction rate will be 0.07 IWC/100 Ft).

If the blower was less powerful and an accessory device was desired, the length of the critical path would have to be reduced. This could be accomplished by modifying routing geometry and by reducing maximum allowable velocity for the junction box fittings.

- The geometry could be revised so that the critical path does not pass through more than one junction box.
- The equivalent length for a junction box is reduced to 30 feet if the velocity limit is lowered to 500 Fpm.
- The pressure drop across all external components decreases with the square of the velocity ratio.

$$P_x = P_1 x \left[Cfm_x / Cfm_1 \right]^2$$

- To minimize duct run pressure drop, flexible wire helix duct shall not have more than 4% excess length.

Section 9

Air-Zoned Systems

Local room and space temperature excursions, compared the set-point of a central thermostat, are an attribute of single-zone, constant airflow rate (Cfm) systems. Acceptable performance is summarized by ACCA ***Manual Zr***, Figure 1-2.

Room and space temperature excursions may be eliminated or reduced by installing zone dampers in branch duct runs. The guidance provided by this section focuses on how zone dampers affect ***Manual D*** *procedures.* Refer to these ***Manual Zr*** locations for guidance pertaining to zoned comfort system design:

- Zoning benefits (Section 1).
- Zoning methods (Section 2).
- Making zoning decisions (Section 3).
- Causes of temperature and comfort problems (Appendix 1).

9-1 Procedure Overlap

The same equipment selection and sizing procedures apply to single-zone constant Cfm systems, and to air-zoned systems. These are noted here:

- A block load calculation for the entire space served by the central equipment and load calculations for each room and space is served by the equipment are required.
- A design value for blower Cfm is required for duct airway sizing. This is determined when ***Manual S*** procedures use ***Manual J*** loads and OEM expanded performance data to select and size central equipment.
- Blower Cfm delivery must be correlated with external static pressure values. This information is provided by the OEM's blower table. Such tables may, or may not, have footnotes that list the equipment cabinet components that were in place when the blower was tested. If blower table notes are missing, or if there is doubt about what cabinet components do, or do not, pertain to the blower table, the practitioner shall get accurate information from other sections of the OEM's engineering guidance, or from an OEM representative.

9-2 Air-Zoning Issues

The ***Manual D*** procedure for sizing duct airways for air-zoned systems is functionally identical to the constant Cfm procedure (see ***Manual Zr***, Appendix 8). There are some differences in detail for the topics listed below. Information and guidance for these topics is provided by Sections 9-3 through 9-8).

- Load calculation procedure.
- Zone damper performance and control.
- Bypass duct and dump zones.
- Equipment capacity control.
- Air distribution effectiveness.
- Air balancing for zone damper systems.

Minimize Excess Equipment Capacity

Manual J mandates that no factor of safety (as an explicit value, or by input data fudging) be applied to any step in the ***Manual J*** procedure, or to the total values for the heating load or cooling load. ***Manual S*** does allow a limited amount of over sizing when the equipment is selected for ***Manual J*** loads, based on an OEM's set of expanded performance data.

- This guidance exists because excess equipment capacity causes a variety of comfort and equipment performance problems. This is especially true for air-zoned systems because of various technical issues caused by interrupted, or throttled, supply air flow during part-load conditions.
- When zone dampers throttle supply air, excess equipment capacity acerbates the excess air problem (excess air equals the momentary difference between the blower Cfm and the sum of the Cfm values for the rooms and spaces).
- Excess air tends to reduce the number of zones, and tends to increase the complexity of the air-system control strategy.
- Primary equipment that has staged or variable capacity tends to improve the excess air situation, but this may not fully nullify an excess air problem.
- For any type of primary equipment, keep the ***Manual S*** allowance for excess capacity to an absolute minimum.
- See ***Manual Zr***, Section 8-12.

9-3 Load Calculations for Air-Zoned Systems

The load calculation procedure for air-zoned systems is somewhat different than for single zone systems. This is explained by the Eighth Edition of the unabridged version of ***Manual J***. Appendix 3 and Figure A11-1 of this document provides this guidance:

- The heating and cooling loads for the entire space served by the central equipment (i.e., the block load) is used to select and size primary equipment. The same block load procedure applies to single-zone systems, and to air-zoned systems.
- For single-zone systems, the room/space cooling load for fenestration is the sum of a daily average value plus an AED excursion value (see the excursion arrow on ***Manual J,*** Figure A3-2).
- For zoned systems, the room/space cooling load for fenestration is the true peak loads for a specific hour of day (see the maximum Btuh value on the hourly fenestration load curve for ***Manual J***, Figure 3-2).
- Per the two preceding bullets, the room/space cooling loads for a zoned system are larger than for a single-zone system. Therefore, the design values for room/space Cfm are larger than for a single-zone system. For heating, the room/space loads are the same for both types of system.
- As explained in Section 6 of this manual, the sum of the room/space sensible cooling loads will not be equal to the block sensible cooling load. This difference will be larger for an air-zoned system vs. a single-zone system. For heating, the sum of the room/space heating loads will equal the block heating load for both types of system.
- ***Manual Zr***, Section 4 explain how ***Manual J*** AED curves for rooms and spaces are used to group rooms and spaces into zones.

9-4 Zone Dampers

Zone dampers may be open-close, three position, or modulating (see **Manual Zr**, Section 5-5). The only thing that matters for duct airway sizing is the pressure drop through an open zone damper (this is a component pressure loss item on the Friction Rate Worksheet). The pressure drop value for a zone damper is obtained from the manufacturer's performance data.

Damper stops provide a simple mechanical method for managing excess air (***Manual Z*** calls this *distributed relief*). However, additional air management measures are normally required (see ***Manual Zr***, Figure 5-1 and Section 5-7). Note that zone temperature control is diminished when a damper stop allows air to flow through a zone damper that the zone thermostat wants to close. Therefore, the maximum supply air flow through a damper stop is limited. ***Manual Zr,*** Figure 8-1 provides a damper stop worksheet, see also Sections 8-5, 8-10 and Figure 8-2.

To manage excess air, zoning controls may use over-blow (digital damper stop control for a designated damper, which may have a fixed or conditional set-point), or selective throttling (digital damper-position override to the full-open position to create a conditionally selected dump zone, normally applied in concert with digital blower motor speed control). However, zone temperature control is diminished when over blow or selective throttling allows air to flow through a zone damper that the zone thermostat wants to close. Therefore, the maximum Cfm for over blow and selective throttling is limited. See ***Manual Zr***, Figure 5-1, Section 5-7, 8-6, 8-7, and Figure 8-2. See also, Section 6-12 for airway sizing guidance that pertains to circulation paths that have over-blow or selective throttling.

9-5 Bypass Duct and Dump Zones

A bypass duct and a dump zone provide similar methods for managing excess air in that the difference between the momentary blower Cfm and the momentary Cfm flowing to the rooms and spaces is routed though a dedicated circulation path. Comments about bypass air and dump zone air are provided below. ***Manual Zr***, Figure 5-1, Section 5-7, Section 8-10, and Figure 8-2 provide comprehensive guidance.

Bypass Duct

A bypass duct provides a commonly-used method for managing excess air, however, additional air management measures may be required. A bypass duct short-circuits blower discharge air to the return-side of the equipment cabinet. This causes the return air to get warmer and warmer for heating and colder and colder for cooling.

The bypass air Cfm value that will not cause a high-limit or low-limit problem for the primary heating-cooling equipment, or a blower stall problem is conditional. The design Cfm value for bypass duct sizing depends on the type of bypass damper control(counterweight damper vs. use of flow and temperature sensors for feed-back control of a modulating damper actuator).

The bypass air circulation path is in parallel with all the other supply-return paths of the duct system. The bypass path tends to have a relatively small total equivalent length. During system commissioning , a hand damper in the bypass path is adjusted so that bypass path resistance is similar to the resistance of the longest circulation path. This tends to stabilize operating conditions for the bypass damper, which enhances the operating range and authority of the bypass damper.

Manual Zr, Section 7 and Section 8 provide detailed guidance pertaining to bypass duct sizing and bypass air control. Per Section 7-10, the design value for bypass Cfm and a 900 Fpm air velocity value determine bypass airway size. Section 8-11 and Figure 8-6 show how the Bypass Airway Sizing Worksheet is used to determine the design value for bypass Cfm. The Excess Air Worksheet is a related calculation tool, see Section 8-10 and Figure 8-2. Sections 9, 10, and 11 provide examples that show calculations for managing excess air, and for bypass duct sizing. Appendix 3 goes into more detail about bypass air physics.

Dump Zone

A dump zone is similar to a bypass duct, but the ramping effect on return air temperature at the primary equipment is moderated because return air from the dump zone is mixed with other return air before it enters the equipment. A dump zone circulation path has the same issues as a bypass duct, as far as path pressure drop adjustment and path airflow control are concerned Temperature control in a dump zone will be poor because the flow of supply air to this space has nothing to do with the temperature in the space. Various issues arise if dump zone temperature excursions get too large, therefore the maximum flow through a dump zone is limited. See ***Manual Zr***, Section 8-4, Section 8-10 and Figure 8-2.

9-6 Equipment Capacity Control

Two-stage equipment and variable-capacity (modulating) equipment provide an effective air management tool. The ability to reduce blower Cfm as zone dampers close reduces the amount of excess air when the system operates at part-load. A coordinated reduction in the heating or cooling capacity of the primary equipment has the appropriate and necessary effect on the temperature ramp for bypass air. See ***Manual Zr***, Figure 5-1, Figure 5-3, Figure 5-9, and Sections 5-8 through 5-13.

9-7 Air Distribution Effectiveness

Zone damper action affects air distribution effectiveness. The open-close type flows the full room/space Cfm value, or essentially no Cfm (ignoring some leakage). Therefore, the open-close type is compatible with supply outlet hardware that is designed for constant Cfm. The modulating type throttles supply air Cfm incrementally. Therefore, the modulating type is not compatible with supply outlet hardware that is designed for constant Cfm. See ***Manual Zr***, Appendix 7 for guidance.

9-8 Air balancing for Zone Damper Systems

With the zone damper wide open and the bypass damper closed (if installed), using hand dampers to balance system airflow at the primary equipment, and the air flow to various rooms and spaces maximizes the authority and effectiveness of the zone dampers. That is, zone damper action (the cycling rate for the open-close type), or zone damper position (for the modulating type) is determined, to the best possible extent, by the need for space temperature control, and not by a need to throttle an excessive airflow rate that is not appropriate any operating condition. The same principle applies to use of a hand damper in a bypass duct. See ***Manual Zr***, Section 5-15, Section 6-13, Section A1-15, and all of Appendix 9.

9-9 Two-Zone Flip-Flop Damper

The basic zoning problem for a two-story home is that the upper level tends to be too warm and the lower level tends to be too cool (multiple fenestration exposures may create additional zoning issues). The consequences of the air buoyancy effect are minimized by installing two separate systems (one for each level), or by using one system with a diverting damper (flip-flop damper) in a trunk duct.

With a flip-flop system, heating and cooling capacity may be somewhat increased for one level and simultaneously reduced for the other level. The ***Manual D*** default for increasing and decreasing the total amount of airflow to a level is twenty percent of the design Cfm for the level. In other words, the supply Cfm delivered to the supply air outlets for a level may vary from the design Cfm by a 0.80 factor to a 1.20 factor). Section 12 provide an application example.

9-10 Duct Sizing Examples for Air-Zoned Systems

Manual Zr does not provide examples of ***Manual D*** procedures for air-zoned systems. See Section 10 through Section 13 of this manual for this type of guidance. The duct-sizing rules for zone-damper systems are summarized here:

- Equipment selection procedures determine the design value for blower Cfm.
- The ***Manual J*** block heating load is used to select and size central equipment, per ***Manual S*** procedures and limits. This is identical to the load for load for single zone systems. Because there are no time-of-day issues for the room and space heating loads, the sum of the local heating loads will equal the block heating load. Therefore, the sum of the local heating Cfm values will equal the blower Cfm value.

- The ***Manual J*** block value for the total cooling load (i.e., the sum of the sensible and latent block loads) is used to select and size central equipment, per ***Manual S*** procedures and limits. This block load value load is identical to the load for single zone systems. Because there are time-of-day issues for sensible room and space cooling loads, the sum of the local sensible cooling loads will not be equal to the block sensible cooling load (see Sections 6-8 through 6-12). Therefore, local airways are sized for worst case circumstances that may not simultaneously occur. Therefore, the sum of the local cooling Cfm values will be greater than the blower Cfm value.
- The OEM's blower table for the selected equipment and the design value for blower Cfm determine the external static pressure value for Step-1 on the Friction Rate Worksheet.
- OEM performance data provides pressure drop values for all airstream components that are external to the OEM's blower table. This list shall include the pressure drop for an open zone damper. The pressure drops for relevant items are summed, per Step-2 on the Friction Rate Worksheet.
- The available static pressure to move air through the straight duct runs and duct fittings equals the external static pressure from the blower table minus the sum of the pressure drop for airstream components that are external to the OEM's blower table. This calculation is made, per Step-3 on the Friction Rate Worksheet.
- Equivalent length values from the ***Manual D*** fitting tables and the measured lengths of the straight runs determine circulation path lengths. For wire flexible helix duct, point-to-point lengths are adjusted for installation deficiencies per Appendix 16 of this manual, as applicable.
- The values for the total effective length of the longest, circulation path is noted on the Friction Rate Worksheet, per Step-4.
- Step-5 on The Friction Rate Worksheet uses the available static pressure value for moving air through the straight runs and fittings in the critical circulation path, and the total effective length of the path to determine the friction rate value for duct airway sizing.
- For local (branch) ducts, there is a heating Cfm and a cooling Cfm. The Duct Sizing Worksheet calculates these values and uses the larger value (design Cfm) for airway sizing.
- The Duct Sizing Worksheet uses the friction rate value from the Friction Rate Worksheet and design Cfm values for branch ducts to determine local airway sizes.
- The Duct Sizing Worksheet uses the friction rate value from the Friction Rate Worksheet and design Cfm value for a trunk duct to determine trunk airway sizes.
- The design Cfm value for a primary trunk is the blower Cfm value. If blower Cfm values for heating and cooling are different, use the larger value.
- For secondary supply trunks, there is a set of heating Cfm values and a set of cooling Cfm values for the downstream supply air outlets. These values are summed separately, and the larger of the two sums is used to size a secondary trunk.
- For secondary return trunks, there is a set of heating Cfm values and a set of cooling Cfm values for the upstream return grilles. These values are summed separately, and the larger of the two sums is used to size a secondary trunk.
- Per ***Manual Zr***, a bypass duct airway size is based on the design bypass Cfm value from the Bypass Airway Sizing Worksheet and a 900 Fpm air velocity value (see ***Manual Zr***, Section 7-9 and Section 8-11).

9-11 Balancing Air-Zoned Systems

Because cooling loads have a time of day issue, the sum of the room and space (local) Cfm values will not be equal to the blower Cfm value. In addition, local heating and cooling Cfm values are seldom equal. Furthermore, the use of over blow for one or more zones increases the difference between the sum of the local Cfm values vs. the blower Cfm value. ***Manual Zr***, Appendix 9 provides guidance for balancing air-zoned systems, as summarized here:

- Proportional balancing resolves Cfm inequality issues.
- For gross system balancing, hand dampers are installed upstream from the zone dampers. For balancing zone-damper air that flows to two or more rooms/spaces (local air), additional hand dampers may be installed downstream from a zone damper.
- Blower Cfm is measured with all hand dampers open, and a closed bypass damper (if applicable). Blower Cfm is adjusted if it is deficient or excessive, compared to the design Cfm value.
- Zone dampers are adjusted with all downstream dampers open (if there are downstream dampers) and the bypass damper closed (if applicable).
- The design Cfm values (larger of the heating and cooling Cfm values) from the Duct Sizing Worksheet, are modified by an overblow

adjustment (as applicable) and summed, then this sum is divided by the design blower Cfm value to determine a diversity factor for proportional balancing. Then, zone Cfm values are multiplied by the diversity factor to obtain Cfm values for adjusting the hand dampers that are upstream from the zone dampers. This is demonstrated by ***Manual Zr***, Figure A9-1.

- Two or more branch balancing dampers may be installed downstream from a zone damper that serves two or more rooms/spaces. The total Cfm value for these rooms/spaces is the balancing Cfm value for the zone damper (per the preceding bullet). The diversity factor for the zone equals the sum of the averages for the heating and cooling Cfm values for the rooms and spaces in the zone divided by the balancing Cfm value for the zone damper. The balancing Cfm for a particular room/space equals the average of the heating and cooling Cfm values for the room/space, multiplied by the diversity factor for the zone. This is demonstrated by ***Manual Zr***, Figure A9-2.

Sizing Rigid Air-Zoned Duct Systems

This section demonstrates the ***Manual D*** procedure for two air-zoned systems. The first one has an operating point blower (PSC motor) and a bypass duct. The second one is for an OEM turn-key system that has a variable-speed compressor and an operating range blower (ECM motor). This system uses selective throttling for excess air control, and has no bypass duct.

10-1 Air-Zoned System with an Operating Point Blower and a Bypass Duct

Figure 10-1 provides a sketch of a 1,300 Cfm sheet metal zone damper system that has a heat pump that operates at a constant blower wheel speed. The supply-side of the air distribution system has four zones, a primary supply trunk, secondary supply trunks, and branch runouts.

- Figure 10-2 (next page) shows the geometry and details of the return system. There are four return grilles and two secondary return trunks.
- Figure 10-3 (next page) shows the local heating and cooling loads for duct airway sizing, and Figure 10-4 (next page) provides the heat pump manufacturer's blower data with conditional footnotes.

Effective Length Calculation

Considering system geometry and fitting types, the zone 2, 3, and 4 loops are candidates for the longest circulation path. Figure 10-5 (ahead two pages) shows the result of the effective length calculations for the supply paths and return paths. Figure 10-6 (ahead two pages) shows that the effective length of the critical circulation path is 355 feet.

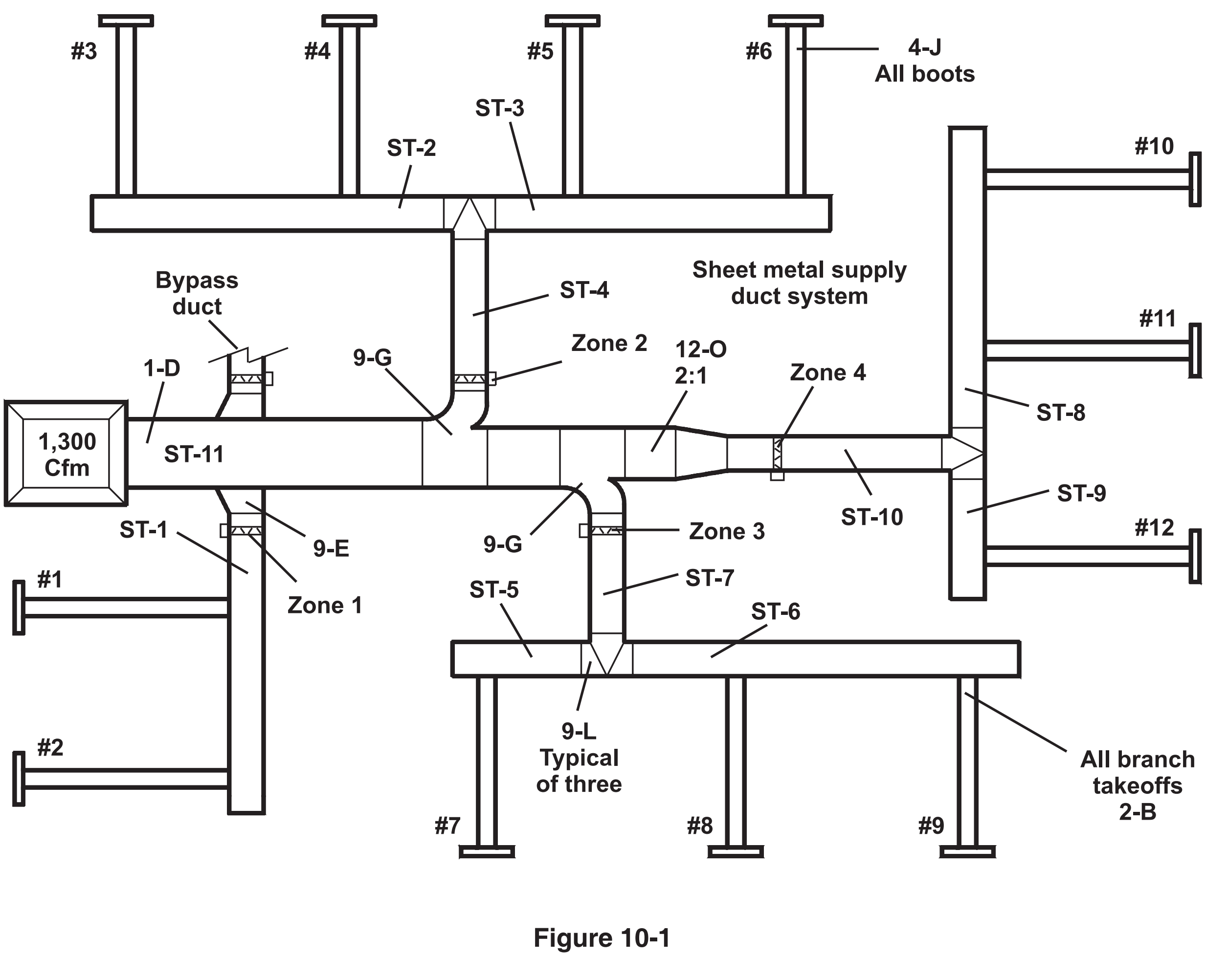


Figure 10-1

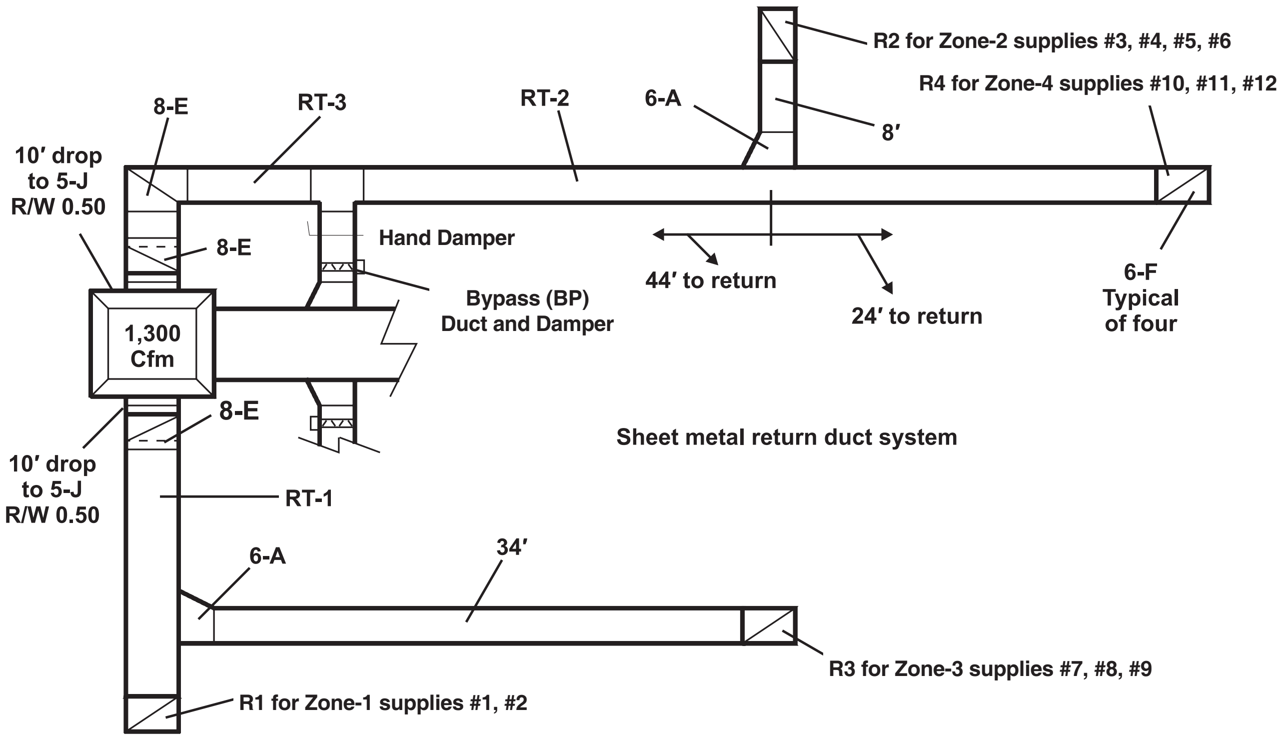


Figure 10-2

Design Friction Rate Calculation

Medium speed data shows the blower delivers 1,300 Cfm when it operates against 0.55 IWC of external resistance. Since the total resistance for the auxiliary heating coil

Duct Lengths and Manual J Loads for the Zoned, PSC Blower, Bypass Duct Example

Run	Trunk Run (Feet)	Branch Run (Feet)	Heating Btuh	Peak Cooling Btuh
1	14	12	4,250	3,300
2	24	12	3,860	3,612
3	54	10	3,970	4,000
4	40	10	2,780	2,663
5	40	10	3,800	2,975
6	54	10	4,440	4,275
7	46	10	4,590	3,500
8	48	10	4,620	3,510
9	62	10	2,350	1,690
10	66	12	3,020	2,590
11	54	12	3,810	3,110
12	56	12	3,430	2,400

1) Block loads (Btuh): Heat = 44,920; Sensible cooling = 33,690
2) Because of time-of-day issues, as they affect the block load and the room/space loads, the sum of the room cooling loads is more than the block cooling load (see Sections 9-3 and 9-10).

Figure 10-3

Blower Data and Component Pressure Drop for the Zoned, PSC Blower, Bypass Duct Example

Discharge Cfm	External Resistance (IWC) vs. Speed		
	High	Medium	Low
1,150			0.53
1,200			0.45
1,250		0.65	0.37
1,300		0.55	0.29
1,350	0.59	0.45	
1,400	0.48	0.34	
1,450	0.37		
1,500	0.26		

1) Heat pump blower tested with a wet refrigerant coil and a low efficiency filter in place.
2) If auxiliary heating coils are required, subtract 0.13 IWC from the values listed in this table.
3) If zone damper is installed in a supply duct, subtract 0.05 IWC (for an open damper) from the values listed in this table.

Figure 10-4

Effective Length Worksheet for the Zoned, PSC Blower, Bypass Duct Example									
Element	**Supply Run ID Number**				**Element**	**Return Run ID Number**			
	S3	S 9	S10	Notes		R2	R3	R4	Notes
Trunk Length	54	62	66		**Trunk Length**	54	22	78	
Trunk Length					**Trunk Length**				
Trunk Length					**Trunk Length**				
Runout Length	10	10	12		**Runout Length**	8	34		
Group 1 (D)	10	10	10		**Group 5** (J)	15	15	15	R/W=0.5
Group 2 (B)	20	20	20		**Group 6** (F)	25	25	25	
Group 3					**Group 7**				
Group 4 (J)	30	30	30		**Group 8** (E)	20	10	20	
Group 8					**Group 10**				
Group 9 (G)	35	35		(branch)	**Group 11**				
Group 11					**Group 12**				
Group 12 (O)			5		**Group 13**				
Group 13					**Other** (6A br)	40	33		Cfm1/Cfm2 R2 & R4 0.63; or 0.56 (R4)
Other (9G)		5	10	(main)	**Other (6A m)**			25	
Other (9L)	20	20	20		**Other**				
Other					**Other**				
Total Length	179	192	173		**Total Length**	162	139	163	

Figure 10-5

(0.13 IWC), supply outlet (0.03 IWC), return (0.03 IWC), open control damper (0.05 IWC), and open hand damper (0.03 IWC) is 0.27 IWC, the available static pressure is 0.28 IWC. Therefore, the design friction rate (0.08 IWC per 100 Ft) is based on 0.28 IWC of pressure and 355 feet of effective length. Figure 10-7 (next page) shows these calculations.

Summary of Figure 10-5 Calculations			
Run	**#3**	**#9**	**#10**
Supply TEL	179	192	173
Run	**R2**	**R3**	**R4**
Return TEL	162	139	163

Figure 10-6

Duct Sizing Calculations

Figure 10-8 (ahead two pages) summarizes the duct airway sizing calculations for this example. These comments apply to the calculations:

- Airway sizes were read from the Galvanized Metal Duct scale on the ACCA Duct Sizing Slide Rule.
- Supply runs 1 and 2 are for zone 1 and return R1; supply runs 3 through 6 are for zone 2 and return R2; supply runs 7 through 9 are for zone 3 and return R3; and supply runs 10 through 12 are for zone 4 and return R4.
- Supply runs 1 and 2 are for secondary trunk ST-1; supply runs 3 through 6 are for secondary trunks ST-2, ST-3 and ST-4; supply runs 7 through 9 are for secondary trunks ST-5, ST-6 and ST-7; and supply runs 10 through 12 are for secondary trunks ST-8, ST-9 and ST-10.
- ST-11 is a primary trunk.
- Return runs R1 and R3 feed secondary return trunk RT-1, and return runs R2 and R4 feed secondary return trunk RT-2.
- Secondary return trunk RT-2 and the bypass duct feed secondary return trunk RT-3.
- Supply runout sizes are for peak cooling Cfm values (based on the peak sensible cooling loads for the rooms).
- Supply runout sizes are based on the design friction rate (0.08 IWC/100) because air velocities are less than 900 Fpm.
- Air way sizes for secondary supply trunk ducts ST-1 through ST-10 are based on peak sensible cooling loads (sum of the downstream room Cfm values).

Friction Rate Worksheet for the Zoned, PSC Blower, Bypass Duct Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.55 IWC   Cfm = 1,300

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	______
Electric resistance heating coil	0.13
Hot water coil	______
Heat exchanger	______
Low efficiency filter	______
High or mid-efficiency filter	______
Electronic filter	______
Humidifier, UV lights, other	______
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	0.05
Total component losses (CPL)	0.27 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (0.55 - 0.27) = 0.28 IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (192 + 163) = 355 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.08 IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

Friction Rate Chart

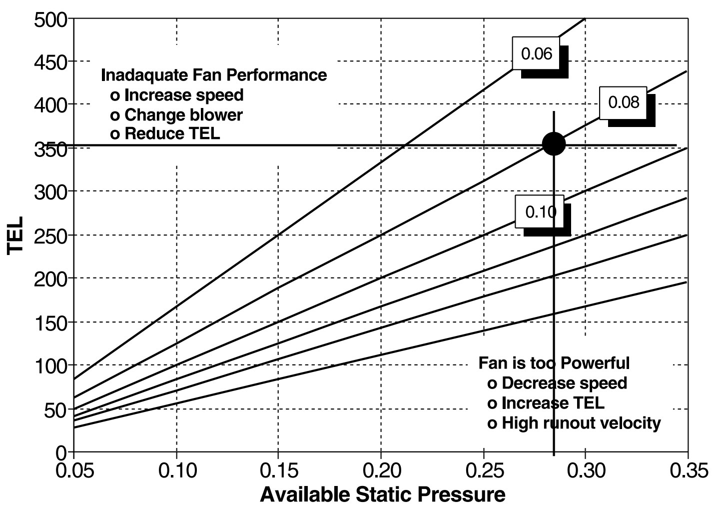


Figure 10-7

Duct Sizing Worksheet for the Zoned, PSC Blower, Bypass Duct Example

HF = Blower Cfm / Manual J Heat Loss = 1,300 / 44,920 = 0.0289
CF = Blower Cfm / Manual J Sensible Heat Gain = 1,300 / 33,690 = 0.0386

FR Value 0.08

Supply-Side Runouts

Supply - Trunk	Heating Btuh	Cooling Btuh	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
S1 — ST-1	4,250	3,300	123	127	127	7	ok	7	See Note 5
S2 — ST-1	3,860	3,612	112	139	139	7	ok	7	
S3 — ST-2	3,970	4,000	115	154	154	7	ok	7	
S4 — ST-2	2,780	2,663	80	103	103	6	ok	6	
S5 — ST-3	3,800	2,975	110	115	115	7	ok	7	
S6 — ST-3	4,440	4,275	128	165	165	8	ok	8	
S7 — ST-5	4,590	3,500	133	135	135	7	ok	7	
S8 — ST-6	4,620	3,510	134	135	135	7	ok	7	
S9 — ST-6	2,350	1,690	68	65	68	5	ok	5	
S10 — ST-8	3,020	2,590	87	100	100	6	ok	6	
S11 — ST-8	3,810	3,110	110	120	120	7	ok	7	
S12 — ST-9	3,430	2,400	99	93	99	8	ok	8	

Supply-Side Trunks

			Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
Run numbers:	S1, S2	**S-Trunk 1**	235	266	266	9	610	9	See Note 5
Run numbers:	S3, S4	**S-Trunk 2**	195	257	257	9	600	9	
Run numbers:	S5, S6	**S-Trunk 3**	238	280	280	9	645	9	
Run numbers:	S3 → S6	**S-Trunk 4**	433	537	537	12	705	12	
Run numbers:	S7	**S-Trunk 5**	133	135	135	7	520	7	
Run numbers:	S8, S9	**S-Trunk 6**	202	200	202	8	595	8	
Run numbers:	S7, S8, S9	**S-Trunk 7**	335	335	335	10	630	10	
Run numbers:	S10, S11	**S-Trunk 8**	197	220	220	8	645	8	
Run numbers:	S12	**S-Trunk 9**	99	93	99	6	505	6	
Run numbers:	S10 → S12	**S-Trunk 10**	296	313	313	10	610	10	
Run numbers:	Primary	**S-Trunk 11**	1,300	1,300	1,300	16	950	17	

Return-Side Runouts

Return - Trunk	Associated Supply Runs	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
R1 — RT-1	S1, S2	235	266	266	9	630	9	See Note 5
R2 — RT-2	S3 through S6	433	537	537	12	705	12	
R3 — RT-1	S7, S8, S9	335	335	335	10	640	10	
R4 — RT-2	S10, S11, S12	296	313	313	10	610	10	

Return-Side Trunks

			Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
Run numbers:	R1, R3	**R-Trunk 1**	570	601	601	12	790	13	See Note 5
Run numbers:	R2, R4	**R-Trunk 2**	729	850	850	14	800	16 std	
Run numbers:	R2, R4, BP	**R-Trunk 3**	Airway size based on 850 Cfm for RT2 or the bypass Cfm(see Note 6), use largest value.						
Run numbers:	Bypass duct - see Note 6.		Per ***Manual Zr***, Section 7-10, size the bypass airway for the bypass Cfm and 900 Fpm velocity.						

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 10-8

- Airway sizes for secondary supply trunks ST-1 through ST-10 are based on the design friction rate (0.08 IWC/100) because air velocities are less than 900 Fpm.
- The preliminary size of primary supply trunk ST-11 is based on 1,300 Cfm value for design blower Cfm.
- The final size of primary trunk ST-11 is based on the maximum allowable velocity (900 Fpm) because the velocity for the design friction rate (0.08 IWC/100) exceeds 900 Fpm.
- Airway sizes for zone return runs (R1, R2, R3 and R4) are based on the Cfm values for zone supply air Cfm.
- Final airway sizes for zone return runs (R1, R2, R3 and R4) are based on the design friction rate (0.08 IWC/100) because air velocities are acceptable (700 Fpm or less).
- The sizes of secondary return trunks RT-1 and RT-2 are based on the Cfm values for upstream return grilles.
- The final sizes of secondary return trunks RT-1 and RT-2 are based on the maximum allowable velocity because the velocity for the design friction rate (0.08 IWC/100) exceeds 700 Fpm.
- The design Cfm for the secondary return trunk RT-3 equals the design Cfm for RT-2 or the design Cfm for the bypass duct, use the larger value.
- The bypass duct airway is sized per ***Manual Zr*** procedures, as summarized by Section 9-5 of this manual.
- The ACCA duct slide rule converts round sizes to rectangular sizes for equivalent airflow resistance.

Comments and Observations

Since there are four exposures, zones do not peak simultaneously, diversity affects the block load, cooling equipment size, the design value for blower Cfm, and the size of primary supply trunk. Therefore, the sizing calculations for central components are identical to the calculations for a simple constant-volume system. But this is a zoned system, so the airway sizing procedure for any duct run that is not a primary run is based on peak room cooling loads.

- The duct layout features a system of secondary trunk ducts. This geometry is preferred because the number of control dampers is minimized. If an extended plenum design was used, each branch runout would have a control damper.
- A bypass duct and a bypass damper maintains adequate air flow thorough central equipment for any part-load condition. ***Manual Zr***, Section 7, Section 8, and Appendix 3 provide detailed information and guidance for determining bypass air Cfm, bypass duct size, bypass duct balancing, and bypass air control. See also, Sections 9-5, 9-6 and 9-10 of this manual.

10-2 Air-Zoned System with Equipment Speed Control and Selective Throttling

Figure 10-9 (next page) provides a sketch of a 1,050 Cfm air-zoned system. Heating and cooling is provided by a heat pump that has a variable-speed compressor and a variable-speed (ECM) blower motor. The primary equipment and the zoning controls are provided by the same OEM. This package uses selective throttling for excess air control, so per the OEM's system design guidance, there is no bypass duct. The two-zone, sheet metal, supply air system has a primary trunk duct, secondary trunks and branch runouts.

Figure 10-9 also shows the geometry and the fittings for the sheet metal return system. This system has two return branches and a primary return trunk.

Figure 10-10 (ahead two pages) shows the local heating and cooling loads for duct airway sizing, and Figure 10-11 (ahead two pages) provides the OEM's blower data with conditional footnotes.

Effective Length Calculation

Figure 10-12 (ahead two pages) summarizes the effective length calculations for supply runs and return runs. Considering system geometry and fitting types, there are no obvious longest path candidates.

Figure 10-13 (ahead two pages) shows the effective length of the critical circulation path is 188 + 167 = 355 feet. Note that return R1 is for zone 1 supply runs (outlets 1 through 4) and return R2 is for zone 2 supply runs (outlets 5 through 8).

Design Friction Rate Calculation

Blower data for speed tap 4 (Figure 10-11) shows the blower delivers 1,100 Cfm when it operates against an external resistance that ranges between 0.24 and 0.85 IWC. Therefore, a maximum of 0.85 IWC of external static pressure is available when the blower satisfies the system air flow requirement (1,050 Cfm). For an ECM blower motor, the default airway sizing value for external static pressure is 70% of the maximum value (per Section 6-5). Therefore, 0.60 IWC is the external static pressure value for Step-1 on the Friction Rate Worksheet.

Since the total resistance for an auxiliary heating coil (0.12 IWC), a supply outlet (0.03 IWC), a return grille (0.03 IWC), an open zone damper (0.05 IWC), and an open hand damper (0.03 IWC) is 0.26 IWC, the available static pressure is 0.34 IWC. Therefore, the friction rate design

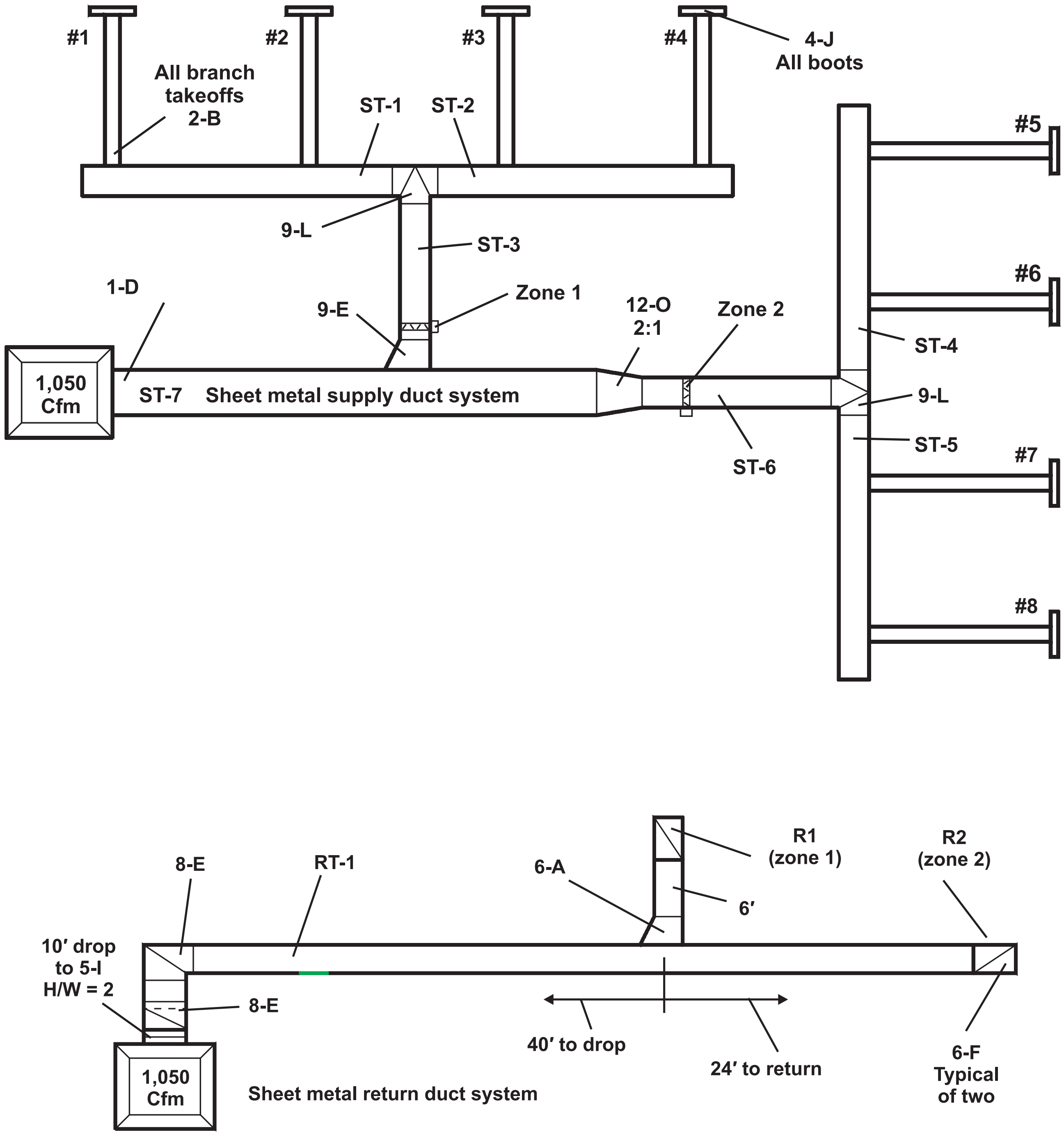


Figure 10-9

value is 0.10 IWC per 100 Ft, based on 0.34 IWC of pressure and 355 feet of effective length. These calculations are shown by Figure 10-14 (ahead two pages).

Duct Sizing Calculations

Figure 10-15 (ahead three pages) summarizes the duct sizing calculations for this example. These comments apply:

- Airway sizes were read from the Galvanized Metal Duct scale on the ACCA Duct Sizing Slide Rule.
- Supply runs 1 through 4 are for zone 1 and return R1, supply runs 5 through 8 are for zone 2 and return R2.

Duct Lengths and Manual J Loads for the Zoning with Selective Throttling Example				
Run	Trunks (Feet)	Branch (Feet)	Heating Btuh	Peak Cooling Btuh
1	48	10	4,250	3,630
2	36	10	3,860	3,973
3	36	10	3,970	4,224
4	48	10	2,780	2,812
5	62	12	3,800	3,273
6	48	12	4,440	4,703
7	48	12	4,590	4,813
8	62	12	4,620	4,827

1) Block loads (Btuh): Heat = 32,310; Sensible cooling = 29,331

2) Because of time-of-day issues, as they affect the block load and the room/space loads, the sum of the room cooling loads is more than the block cooling load (see Sections 9-3 and 9-10).

Figure 10-10

ECM Blower Data and Component Pressure Drop Notes for the Zoning with Selective Throttling Example			
Speed Setting	Cfm Setting	ESP (IWC)	Maximum RPM
TAP - 1	800	0.22 – 0.65	500
TAP - 2	900	0.23 – 0.70	600
TAP - 3	1,000	0.24 – 0.75	700
TAP - 4	1,100	0.24 – 0.85	850
TAP - 5	1,200	0.23 – 0.90	1,000
TAP - 6	1,300	0.19 – 0.95	1,200

1) The heat pump blower was tested with a wet refrigerant coil and a low efficiency filter in place.

2) If an electric supplemental heating coil is required, subtract 0.12 IWC from the ESP values listed in this table.

3) If a zone control damper is installed in the critical circulation, path subtract 0.05 IWC (for an open damper) from the ESP values listed in this table.

Figure 10-11

Effective Length Worksheet for the Zoning with Selective Throttling Example									
Element	Supply Run ID Number				Element	Return Run ID Number			
	S1 or S4	S2 or S3	S5 or S8	S6 or S7		R1	R2	Notes	
Trunk Length	48	36	62	48	Trunk Length	40	40		
Trunk Length					Trunk Length	10	10		
Runout Length	10	10	12	12	Runout Length	6	24		
Group 1 (D)	10	10	10	10	Group 5 (I)	30	30	(H/W = 2)	
Group 2 (B)	20	30	20	30	Group 6 (F)	25	25		
Group 3					Group 7				
Group 4 (J)	30	30	30	30	Group 8 (E)	10	10		
Group 8					Group 10				
Group 9 (L)	20	20	20	20	Group 11				
Group 11					Group 12				
Group 12 (O)			5	5	Group 13				
Group 13					Other (6A br)	18		(cfm1/cfm2 = 0.45)	
Other (9E)	50	50	5	5	Other (6A m)		18		
Other					Other (8E)	10	10		
Total Length	188	186	164	160	Total Length	149	167		

Figure 10-12

- Supply runs 1 through 4 are for secondary trunks ST-1, ST-2 and ST-3; supply runs 5 through 8 are for secondary trunks ST-4, ST-5 and ST-6.
- Return runs R1 and R2 feed primary return trunk RT-1.
- Supply runout sizes are for peak cooling Cfm values (based on the peak sensible cooling loads for the rooms).

Summary of Figure 10-12 Calculations				
Run	S1	S2	S5	S6
Supply TEL	188	186	164	160
Run	R1	R2		
Return TEL	149	167		

Figure 10-13

Friction Rate Worksheet for the Zoning with Selective Throttling Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.60 IWC   Cfm = 1,050

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	______
Electric resistance heating coil	0.12
Hot water coil	______
Heat exchanger	______
Low efficiency filter	______
High or mid-efficiency filter	______
Electronic filter	______
Humidifier, UV lights, other	______
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	0.05
Total component losses (CPL)	0.26 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (0.60 - 0.26) = 0.34 IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (188 + 167) = 355 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.10 IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

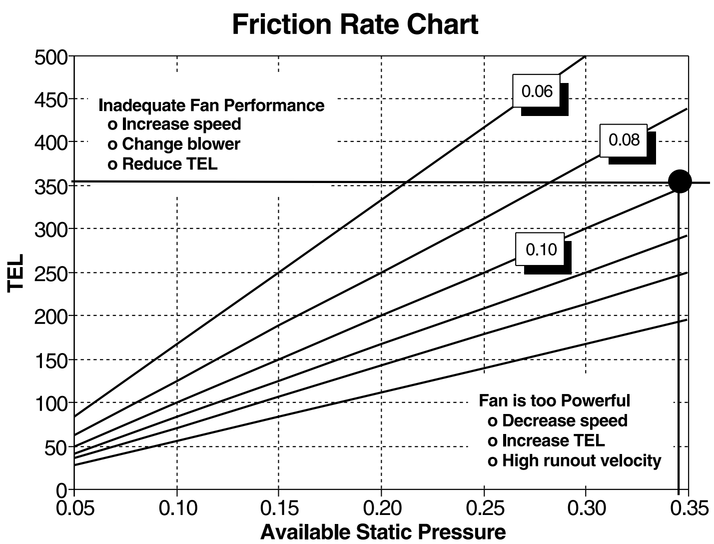


Figure 10-14

Duct Sizing Worksheet for the Zoning with Selective Throttling Example

HF = Blower Cfm / Manual J Heat Loss = 1,050 / 32,310 = 0.0325 **CF = Blower Cfm / Manual J Sensible Heat Gain =** 1,050 / 29,321 = 0.0358									**FR Value** 0.10
Supply-Side Runouts									
Supply - Trunk	**Heating Btuh**	**Cooling Btuh**	**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
S1 — ST-1	4,250	3,630	138	130	138	7	ok	7	See Note 5
S2 — ST-1	3,860	3,973	125	142	142	7	ok	7	
S3 — ST-2	3,970	4,224	129	151	151	7	ok	7	
S4 — ST-2	2,780	2,812	90	101	101	6	ok	6	
S5 — ST-4	3,800	3,273	123	117	123	6	ok	6	
S6 — ST-4	4,440	4,703	144	168	168	7	ok	7	
S7 — ST-5	4,590	4,813	149	172	172	7	ok	7	
S8 — ST-5	4,620	4,827	150	173	173	7	ok	7	
Supply-Side Trunks									
Run numbers:	S1, S2	**S-Trunk 1**	263	272	272	9	625	9	See Note 5
Run numbers:	S3, S4	**S-Trunk 2**	219	252	252	8	730	8	
Run numbers:	S1 → S4	**S-Trunk 3**	482	524	524	11	760	12 std	
Run numbers:	S5, S6	**S-Trunk 4**	267	286	286	9	695	9	
Run numbers:	S7, S8	**S-Trunk 5**	299	345	345	9	800	9	
Run numbers:	S5 → S8	**S-Trunk 6**	566	631	631	12	800	12	
Run numbers:	Primary	**S-Trunk 7**	1,050	1,050	1,050	15	870	15	
Return-Side Runouts									
Return - Trunk	**Associated Supply Runs**		**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
R1 — RT-1	S1 → S4		482	524	524	11	830	12	See Note 5
R2 — RT-2	S5 → S8		566	631	631	12	805	13	
Return-Side Trunks									
Run numbers:	Primary	**R-Trunk 1**	1,050	1,050	1,050	15	930	16	Note 5

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 10-15

- Supply runout sizes are based on the design friction rate (0.10 IWC/100) because the air velocities are less than 900 Fpm.
- Air way sizes for secondary supply trunk ducts ST-1 through ST-6 are based on peak sensible cooling loads (sum of the downstream room Cfm values).
- Airway sizes for secondary supply trunks ST-1 through ST-6 are based on the design friction rate (0.10 IWC/100) because air velocities are less than 900 Fpm.
- The size of primary supply trunk ST-7 is based on 1,050 Cfm value for design blower Cfm and the design friction rate (0.10 IWC/100) because the air velocity is less than 900 Fpm.
- Airway sizes for the zone return runs (R1 and R2) are for the zone supply air Cfm.
- Final airway sizes for the zone return runs (R1 and R2) are based on the maximum allowable velocity (700 Fpm) because the velocity for the design friction rate (0.10 IWC/100) exceeds 700 Fpm.
- The preliminary size of primary return trunk RT-1 is based on the design value for blower Cfm.

- The final size of primary return trunk RT-1 is based on the maximum allowable velocity (700 Fpm) because the velocity for the design friction rate (0.10 IWC/100) exceeds 700 Fpm.
- The ACCA Duct Sizing Slide Rule converts round sizes to equivalent rectangular sizes.

Comments and Observations

The conditioned space is has a South exposure, a West exposure and two partition walls, so the zone sensible cooling loads peak between 3 pm for the South zone and 6 pm for the West zone, so directional diversity, affects the block load, cooling equipment size, the design value for blower Cfm, and the size of primary trunk runs. Therefore, the equipment selection and sizing calculations for the heat pump are identical to the calculations for a single-zone, constant-volume system. But this is a zoned system, so the airway sizing procedure for any duct run that is not a primary run is based on peak room cooling loads.

- The duct layout features a system of secondary trunk ducts. This geometry is preferred because the number of control dampers is minimized. If an extended plenum design was used, each branch runout would have a control damper.
- The heating and cooling equipment has a variable-speed compressor, a blower driven by an ECM motor, and integrated OEM control logic that makes selective throttling decisions. Therefore, a bypass duct may not be required, per OEM guidance.
- The suggestion that a bypass duct is not required may only be conditionally true, depending on the minimum air flow rate produced by throttled zone dampers, the minimum air flow requirements for the refrigerant coil and the electric heating coil, and the noise and draft caused by forcing excess air through a the supply air outlet in a digital-selected dump zone.
- In addition, it may be that a digital-selected dump zone may be a critical space, as far as the occupants are concerned. When this is the case, this space may be disqualified for dump zone use (per control system settings), which means that some other, perhaps less-capable space, will be used as a dump zone.
- Third-party zoning devices and controls may not be able to control variable-speed equipment (blower and/or compressor) if the primary equipment manufactures's control schemes and control logic are proprietary (not available to third party vendors).

Section 11 — Illustrative Example

Sizing Flexible Air-Zoned Duct Systems

This section demonstrates the ***Manual D*** procedure for an air-zoning application that has a variable-speed compressor and an operating range blower (ECM motor). This system uses selective throttling for excess air control, and has no bypass duct. All duct runs are flexible wire helix duct. This example is for a system that complies with the required standard of care for installing flexible wire helix duct (see Section 4-3).

11-1 Flexible Air-Zoned System with Equipment Speed Control and Selective Throttling

Figure 11-1 (next page) provides a sketch of a 900 Cfm air-zoned, flexible wire-helix duct system. Heating and cooling is provided by a heat pump that has a variable-speed compressor and a variable-speed (ECM) blower motor. The primary equipment and the zoning controls are provided by the heat pump manufacturer. This package uses selective throttling for excess air control, so per the OEM's system design guidance, there is no bypass duct.

The four-zone, supply air system has four secondary trunks and branch runouts. There are three return grilles, one direct return branch, and one secondary return duct.

Figure 11-2 (next page) shows the local heating and cooling loads for duct airway sizing, and Figure 11-3 (next page) provides the OEM's blower data with conditional footnotes.

11-2 Effective Length Calculation

Effective length calculations are based on the standard of care for installing flexible wire helix duct (excess length 4% or less, with negligible sag). Note that junction box equivalent length depends on a reference velocity (see Appendix 3). A design velocity of 600 Fpm is used for this example. Therefore, the equivalent length of one junction box is 40 feet.

Figure 11-4 (ahead two pages) shows that the Zone-2 circulation path for supply runs 7 or 8 and the R3 return grille is the longest circulation path. For this circuit, the effective length of the supply path is 241 feet and the effective length of the return path is 150 feet. Therefore, the total effective length of the critical circulation path is 391feet.

11-3 Design Friction Rate Calculation

Blower data for speed tap 3 (Figure 11-3) shows the blower delivers 900 Cfm when it operates against a resistance that ranges between 0.24 and 0.85 IWC. Therefore, a maximum of 0.85 IWC of pressure is available when the blower satisfies the system air flow requirement (900 Cfm). For an ECM blower motor, the default airway sizing value for external static pressure is 70% of the maximum value (per Section 6-5). Therefore, 0.60 IWC is the external static pressure value for Step-1 on the Friction Rate Worksheet.

Since the resistance for the auxiliary heating coil (0.10 IWC), a supply outlet (0.03 IWC), a return (0.03 IWC), an open zone damper (0.05 IWC) and an open hand damper (0.03 IWC) is 0.24 IWC, the available static pressure is 0.36 IWC. Therefore, the friction rate design value is 0.10 IWC per 100 Ft, based on 0.36 IWC of pressure and 391 feet of effective length. These calculations are shown by Figure 11-5 (ahead three pages).

11-4 Duct Sizing Calculations

Figure 11-6 (ahead four pages) summarizes the duct sizing calculations for this example. These comments apply:

- Duct sizes were read from the Wire Helix Duct scale on the ACCA Duct Sizing Slide Rule.
- Supply runs 1 and 2 are for secondary supply trunk ST-1, supply runs 3 and 4 are for secondary trunk ST-2, supply runs 5 and 6 are for secondary trunk ST- 3 and supply runs 7 and 8 are for secondary trunk ST- 4.
- Supply runs 3, 4, 5, 6, 7 and 8 are for secondary supply trunk ST-5.
- Return branches R2 and R3 feed secondary return trunk RT-2 and return R1 flows to a direct return branch RT-1.
- Supply runout Cfm values and sizes are based on the peak sensible cooling loads for rooms.
- A 600 Fpm velocity limit for supply air was used because a larger value produces a significant increase in the equivalent length of junction box fittings.
- The final sizes of supply runouts are based on the design friction rate (0.09 IWC/100) because air velocities are less than 600 Fpm.

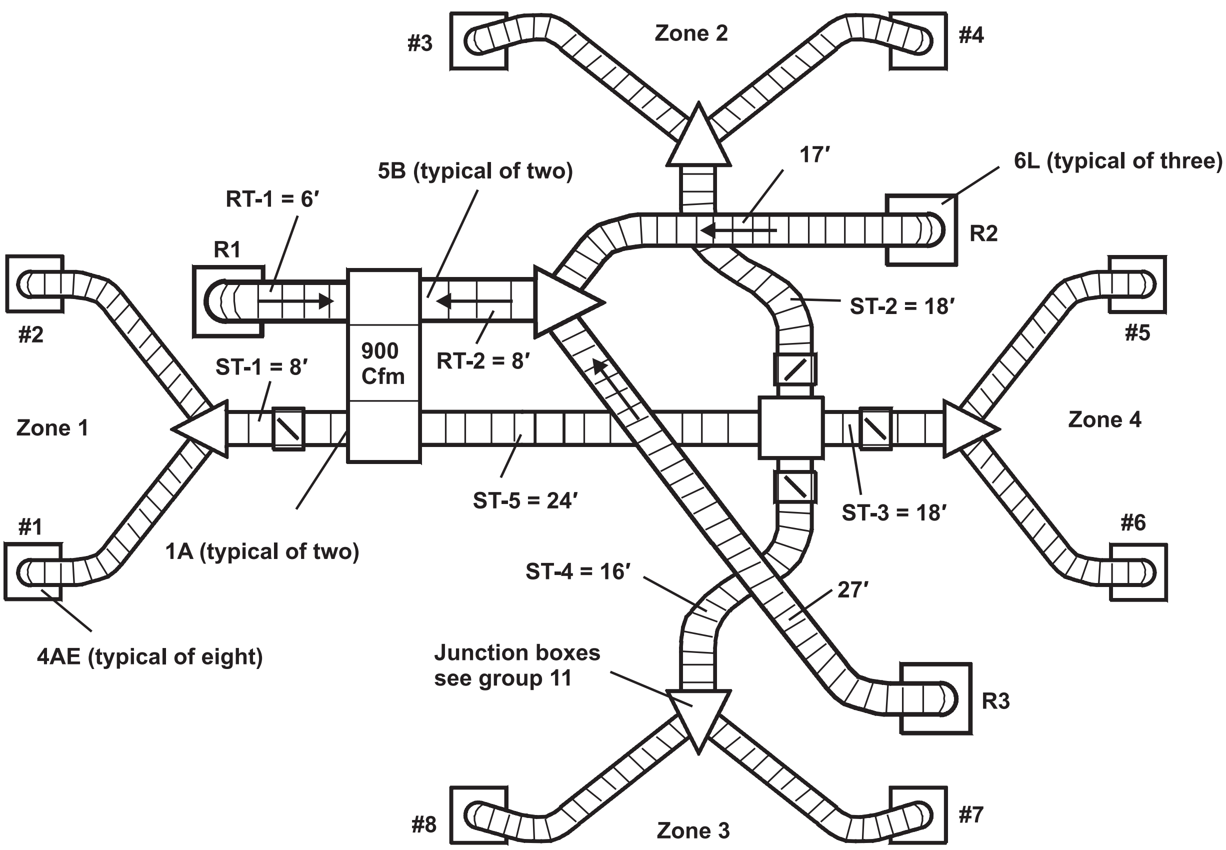


Figure 11-1

- The Cfm values and sizes of secondary trunks ST-1 through ST-4 are based on the design Cfm values for the rooms that they serve.

Manual J Loads for the Flexible Air-Zoned System			
Run	**Runout Size**	**Heating Btuh**	**Peak Cooling Btuh**
1	16'	3,800	2,380
2	16'	3,970	3,200
3	14'	4,250	2,750
4	14'	3,860	3,010
5	14'	4,500	2,610
6	14'	4,590	3,500
7	14'	4,870	3,750
8	14'	2,350	1,690
R1	For supply runs 1 and 2		
R2	For supply runs 3, 4 and 5		
R3	For supply runs 6, 7 and 8		

1) Block loads (Btuh): Heat = 32,190; Sensible cooling = 20,808
2) Because of time-of-day issues, as they affect the block load and the room/space loads, the sum of the room cooling loads is more than the block cooling load (see Sections 9-3 and 9-10).

Figure 11-2

- The size of secondary trunk ST-5 is based on the sum of the design Cfm values for supplies S3 through S8.
- The final sizes of supply trunks ST-1, ST-2, ST-3 and ST-4 are based on the friction rate design value (0.09 IWC/100) because air velocities are less than 600 Fpm.

ECM Blower Data and Component Pressure Drop Notes for the Flexible Air-Zoned System			
Speed	**Cfm**	**ESP (IWC)**	**Max RPM**
TAP - 1	700	0.23 – 0.80	600
TAP - 2	800	0.24 – 0.85	700
TAP - 3	900	0.24 – 0.85	850
TAP - 5	1,025	0.23 – 0.90	1,000
TAP - 6	1,150	0.19 – 0.95	1,200

1) The heat pump blower was tested with a wet refrigerant coil and a low efficiency filter in place.
2) If an electric supplemental heating coil is required, subtract 0.12 IWC from the ESP values listed in this table.
3) If a zone control damper is installed in the critical circulation, path subtract 0.05 IWC (for an open damper) from the ESP values listed in this table.

Figure 11-3

Effective Length Worksheet for the Flexible Air-Zoned System							
Element	Supply Run ID Number			Element	Return Run ID Number		
	S7 or S8	Notes			R3	Notes	
Trunk Length	32 x 1.0 = 32	Cut to fit		**Trunk Length**	8 x 1.0 = 8	Cut to fit	
Trunk Length				**Trunk Length**			
Trunk Length				**Trunk Length**			
Runout Length	14 x 1.0 = 14	Cut to fit		**Runout Length**	27 x 1.0 = 27	Cut to fit	
Group 1 (A)	35			**Group 5** (B)	40		
Group 2				**Group 6** **(L)**	20		
Group 3				**Group 7**			
Group 4 (AE)	55			**Group 8**			
Group 8				**Group 10**			
Group 9				**Group 11**	40	(1 @ 600 Fpm)	
Group 11	80	(2 @ 600 Fpm)		**Group 12**			
Group 12				**Group 13**			
Group 13				**Other** (11)	5	(one 45 degree bend)	
Other (11)	15	(three 45 degree bends)		**Other** (11)	10	(one 90 degree bend)	
Other (11)	10	(one 90 degree bend)		**Other**			
Total Length	241			**Total Length**	150		

Figure 11-4

- The final size for secondary trunk ST-5 was increased from 14 inches to 16 inches to keep the air velocity below 600 Fpm.
- The size of return branch R1 is based on the sum of the S1 and S2 supplies. The size of return branch R2 is based on the sum of the S3, S4 and S5 supplies. The size of return branch R3 is based on the sum of the S6, S7 and S8 supplies.
- The final size of the return ducts R1, R2 and R3 are based on the design friction rate (0.09 IWC/100) because the air velocity is less than 600 Fpm.
- The R1 branch is a direct return, so R1 and RT-1 are the same duct. The final size of secondary return trunk RT-1 is based on the friction rate design value (0.09 IWC/100) because the air velocity is less than 600 Fpm.
- The R2 and R3 branches flow to secondary return trunk RT-2. The preliminary size of the RT-2 trunk is based on the sum of the Cfm values for supplies S2 through S8.
- The final size for the RT-2 return trunk was increased from 14 inches to 16 inches to keep the air velocity below 600 Fpm.

11-5 Comments and Observations

Since there are four exposures, zones do not peak simultaneously, diversity affects the block load, cooling equipment size, the design value for blower Cfm, and the size of primary supply trunk. Therefore, the sizing calculations for central components, are identical to the calculations for a simple constant-volume system. But this is a zoned system, so the airway sizing procedure for any duct run that is not a primary run is based on peak room cooling loads.

- The duct layout features a system of secondary trunk ducts. This geometry is preferred because the number of control dampers is minimized. (If an extended plenum design was used, each branch runout would have a control damper.)
- The heating and cooling equipment has a variable-speed compressor, a blower driven by an ECM motor, and integrated OEM control logic that makes selective throttling decisions. Therefore, a bypass duct may not be required, per OEM guidance.
- The suggestion that a bypass duct is not required may be conditionally true, depending on the minimum air flow rate produced by throttled zone dampers, the minimum air flow requirements for the refrigerant coil and the electric heating coil,

Friction Rate Worksheet for the Flexible Air-Zoned System

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.60 IWC Cfm = 900

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	
Electric resistance heating coil	0.10
Hot water coil	
Heat exchanger	
Low efficiency filter	
High or mid-efficiency filter	
Electronic filter	
Humidifier, UV lights, other	
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	0.05
Total component losses (CPL)	0.24 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (0.60 - 0.24) = 0.36 IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (241 + 150) = 391 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.09 IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

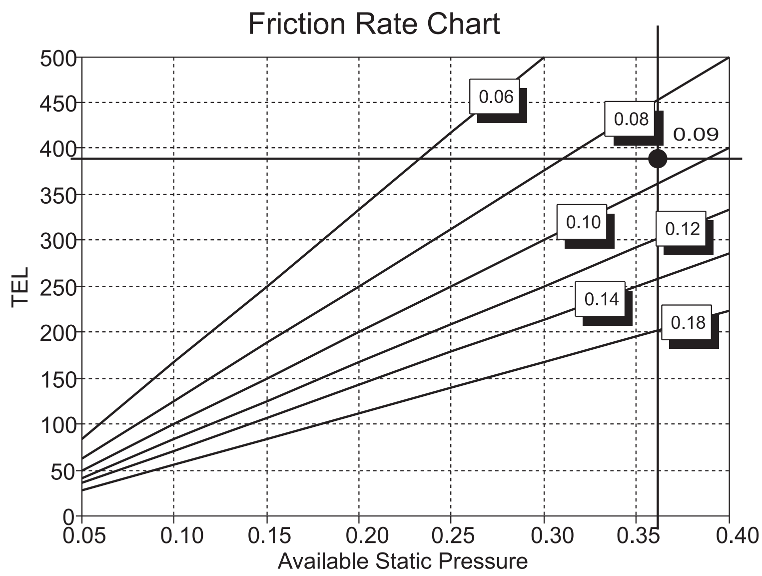


Figure 11-5

Duct Sizing Worksheet for the Flexible Air-Zoned System

	FR Value
HF = Blower Cfm / Manual J Heat Loss = 32,190 / 900 = 0.0280 **CF = Blower Cfm / Manual J Sensible Heat Gain =** 20,808 / 900 = 0.0433	0.09

Supply-Side Runouts

Supply - Trunk	Heating Btuh	Cooling Btuh	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
S1 — ST-1	3,800	2,380	106	103	106	7	ok	7	See Note 5
S2 — ST-1	3,970	3,200	111	139	139	8	ok	8	
S3 — ST-2	4,250	2,750	119	119	119	7	ok	7	
S4 — ST-2	3,860	3,010	108	130	130	8	ok	8	
S5 — ST-3	4,500	2,610	126	113	126	7	ok	7	
S6 — ST-3	4,590	3,500	129	152	152	8	ok	8	
S7 — ST-4	4,870	3,750	136	162	162	8	ok	8	
S8 — ST-4	2,350	1,690	66	73	73	6	ok	6	

Supply-Side Trunks

Run numbers:	S1, S2	**S-Trunk 1**	218	242	242	10	425	10	See Note 5
Run numbers:	S3, S4	**S-Trunk 2**	227	249	249	10	430	10	
Run numbers:	S5, S6	**S-Trunk 3**	255	265	265	10	480	10	
Run numbers:	S7, S8	**S-Trunk 4**	202	236	236	9	520	9	
Run numbers:	S3 → S8	**S-Trunk 5**	684	750	750	14	710	16	

Return-Side Runouts

Return - Trunk	Associated Supply Runs	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
R1 — RT-1	S1, S2	218	242	242	9	550	9	See Note 5
R2 — RT-2	S3, S4, S5	353	362	362	11	560	12 std	
R3 — RT-2	S6, S7, S8	331	387	387	11	600	12 std	

Return-Side Trunks

Run numbers:	R1 direct	**R-Trunk 1**	218	242	242	9	550	9	See Note 5
Run numbers:	R2, R3	**R-Trunk 2**	684	750	750	14	700	16	

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 11-6

and the noise and draft caused by forcing excess air through a the supply air outlet in a digital-selected dump zone.

- In addition, it may be that a digital-selected dump zone may be a critical space, as far as the occupants are concerned. When this is the case, this space may be disqualified for dump zone use (per control system settings), which means that some other, perhaps less-capable space, will be used as a dump zone.

Section 12 — Illustrative Example

Sizing Two-Zone Bi-Level Duct Systems

This section demonstrates the ***Manual D*** procedure for a two story dwelling that has central equipment, a lower level zone and an upper level zone. The goal is to minimize comfort problems caused by the buoyancy of warm air. This will reduce the floor-to-floor temperature difference, but will not reduce room-to-room temperature differences caused by seasonal and hourly changes in fenestration cooling loads.

12-1 Bi-Level Zoning with Central Equipment

Figures 12-1 and 12-2 (next page) show a two-zone system that serves the upper and lower levels of a two-story dwelling. This 1,300 Cfm system features two horizontal distribution systems (one for each floor), primary trunk ducts (supply and return), two zone control dampers, and central air handling equipment.

Design Concept

The most common zoning problem for a two-story home is that the upper level tends to be too warm, and the lower level tends to be too cool. This behavior is improved by installing two separate systems (one for each level), or by adding zone dampers and zoning controls to one central system. The air-zoning hardware and operating controls may be packaged with the central heating and cooling equipment, or it may be purchased by a third-party vendor.

For example, if additional cooling capacity is required for the second level, zone dampers route more supply Cfm to the second floor and reduce the flow to the first floor. Or, if more heating capacity is required for the lower level, the flow control dampers can route more supply Cfm to the first floor, and reduce the flow to the second floor. Therefore, ***Manual D*** calculations are adjusted so that the airway sizes for first-floor ducts and second-floor ducts can accommodate a 20 percent increase (a 1.2 overblow factor) in the Cfm that flows to each supply air outlets on each level.

Cooling Load Calculation

For this type of zoning, peak room loads are not relevant because there is no directional zoning for either level. Therefore the cooling load calculations for this two-zone system are identical to the calculations for a single-zone constant Cfm system. For equipment sizing, use the standard ***Manual J*** procedure to calculate the block load for the entire dwelling, and use the standard procedure for individual room loads (Section 6-8 discusses load calculation procedures).

Zone Dampers

Zone dampers may have an open-close action. or a modulating action. In either case, it is important to know something about the minimum flow rate through the damper. If the damper has a mechanical or electronic stop, there is reduced flow through a fully modulated damper. If there is no stop, flow can be completely throttled (with some leakage).

- Open-close zone dampers for a simple flip-flop system require a stop setting that routes 20% more air to the other level when the throttled zone damper is on its stop and the zone damper for the other level is full open. Determination of stop positions is an air balancing task (see Section 12-7).
- This way, the range of supply air Cfm control is about ± 20 percent of the single-zone Cfm for each level (when one damper is at the stop position, the other damper is wide open.
- Modulating dampers have a wide range of control. In this case, the range of supply air Cfm throttling can exceed ± 20 percent for each level. The benefit is that the flow of supply air may be adjusted to match distribution need for a momentary load condition.
- The guidance in this section does not apply to large changes in the Cfm supplied to a level (refer to ***Manual Zr*** procedures, and ***Manual D*** Section 10 examples).

System Controls

One thermostat is used for each zone (level). These thermostats adjust the position of the zone dampers and affect the operation of central equipment components in response to the simultaneous demand for zone air delivery. The details of the control strategy vary, depending on the product or vendor. In any case, it is important to know how the controls affect the maximum and minimum flow rates for each zone, and how they affect the minimum flow rate through central equipment.

Bypass Route

A bypass route (duct or dump zone) may, or may not, be required. This depends on the type of zone damper and the setting for the minimum position stop.

- A bypass route is not required if the zone dampers limit level airflow to about ± 20 percent of the single-zone Cfm for each level. For this scenario, total

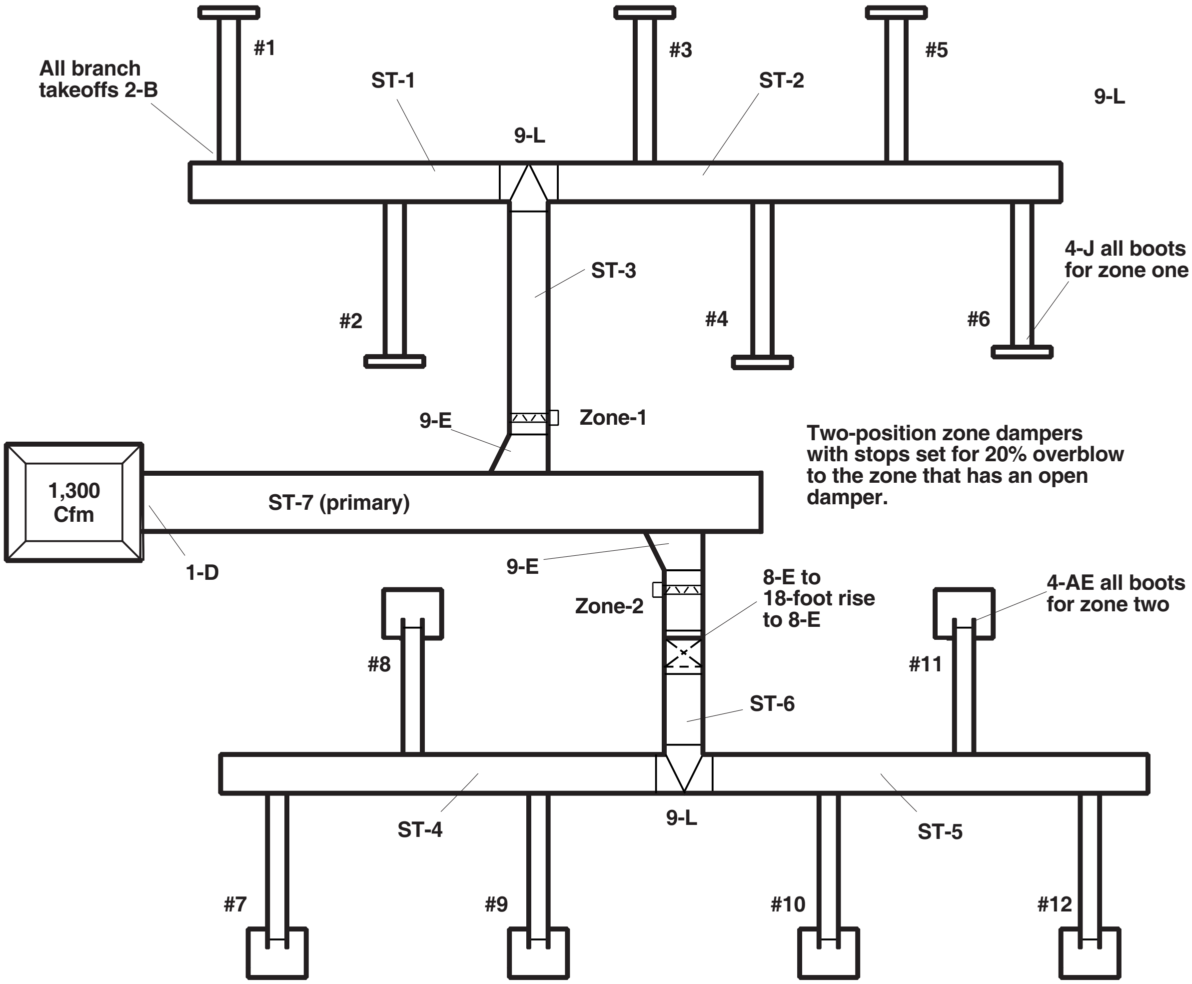


Figure 12-1

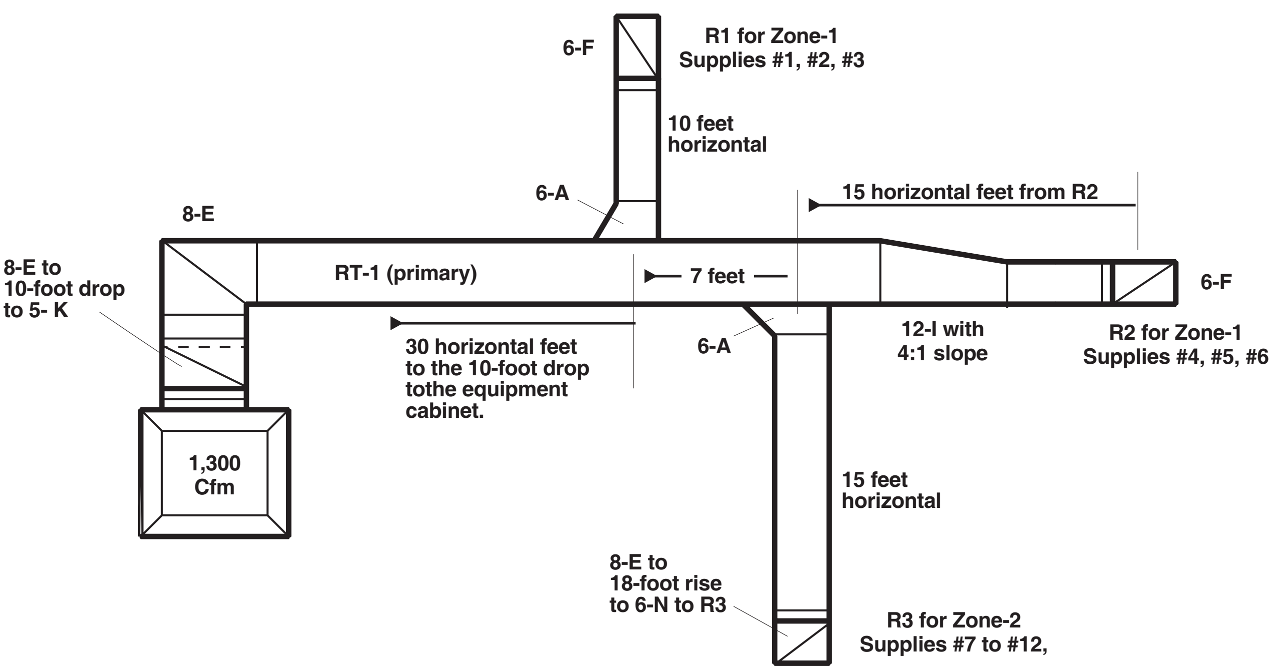


Figure 12-2

system air flow is roughly constant as one damper closes to its stop as the other damper goes to the full open position.

- A bypass route may be required if modulating zone dampers create an excess air issue. This depends on the maximum amount of excess air, and whether the primary system components (blower, compressor, electric coil or furnace heat exchanger) are equipped with some type of capacity control (motor speed control or staging, for example).
- See Section 9-5 and Section 9-6 of this manual for guidance pertaining to bypass ducts and dump zones.

Returns

A comprehensive return system shall be provided for each level. If there is one central return on each level, transfer grilles shall provide a low-resistance return path for each isolated room that receives supply air.

Sizing Calculations

This example shows metal construction, but the system may be fabricated from any combination of metal, duct board or flexible wire helix materials. In any case, the airway sizing calculations for this type of two-zone system is based on these concepts:

- Cooling load diversity applies to the central equipment and to each floor because both zones have four exposures.
- Cooling equipment size, blower Cfm and the flow rates for primary trunks are based on output from the standard ***Manual J*** procedure, ***Manual S*** equipment selection and sizing guidance, and expanded performance data published by the equipment manufacturer.
- Since air flow is not controlled on a room-by-room basis, room Cfm calculations are based on the standard ***Manual J*** procedure.
- Since the flow control dampers increase the percentage of blower Cfm that is routed to either level, Cfm values for each supply outlet, runout duct and secondary trunk duct are increased by 20 percent. This adjustment does not apply to the central components, or primary duct runs.
- The pressure drop for an open zone control damper is subtracted from the external static pressure that is provided by the blower when it delivers the design Cfm.

12-2 Loads and Performance Data

Manual J Eighth Edition, Version 2.10, or later (Sections 11-4, 11-5 and 11-7) provide equipment sizing loads and room loads. Figure 12-3 shows the heating loads, room cooling loads from the standard procedure, and the block load for sensible cooling. Figure 12-4 provides the manufacturer's blower data with footnotes pertaining to the test stand configuration, plus pressure drops for accessories and flow control devices.

Duct Lengths and Manual J load for the Two-Zone Bi-Level Example

Run	Trunks (Feet)	Branch (Feet)	Heating Btuh	Cooling Btuh
1	52	10	4,250	2,750
2	44	8	3,860	3,010
3	42	10	3,970	3,200
4	48	8	2,780	2,130
5	56	10	3,800	2,380
6	64	8	4,440	3,420
7	80	10	4,590	3,500
8	74	8	4,620	3,510
9	66	10	2,350	1,690
10	66	10	3,020	2,590
11	74	8	3,810	3,110
12	80	10	34,30	2,400

1) Block loads (Btuh): Heat = 44,920; Sensible cooling = 33,690

2) Because of time-of-day issues, as they affect the block load and the room/space loads, the sum of the room cooling loads is more than the block cooling load (see Sections 9-=3 and 9-10).

Figure 12-3

Blower Data and Component Pressure Drop for the Two-Zone Bi-Level Example

Discharge Cfm	External Resistance (WC) vs. Speed		
	High	Medium	Low
1,150			0.53
1,200			0.45
1,250		0.67	0.37
1,300		0.59	0.29
1,350	0.59	0.47	
1,400	0.48	0.36	
1,450	0.37		
1,500	0.26		

1) Heat pump unit tested with a wet refrigerant coil and a low efficiency filter in place.

2) If auxiliary heating coils are required, subtract 0.08 IWC from the values that are listed in this table.

3) If zone control damper is installed in the circulation path subtract 0.05 IWC (for an open damper) from the values that are listed in this table.

Figure 12-4

Effective Length Worksheet for the Two-Zone Bi-Level Example									
Element	**Supply Run ID Number**				**Element**	**Return Run ID Number**			
	S3	S6	S7 or S12	S9 or S10		R1	R2	R3	Notes
Trunk Length	42	64	80	66	**Trunk Length**	10	10	10	
Trunk Length					**Trunk Length**	30	30	30	
Trunk Length					**Trunk Length**		7	7	
Runout Length	10	8	10	10	**Runout Length**	10	15	33	
Group 1 (D)	10	10	10	10	**Group 5** (k)	10	10	10	
Group 2 (B)	40	20	20	35	**Group 6** (F)	25	25		
Group 3					**Group 7**				
Group 4 (J, AE)	30	30	55	55	**Group 8** (E all)	20	20	30	(10 each)
Group 8(2@E)			20	20	**Group 10**				
Group 9 (E br)	50	50	50	50	**Group 11**				
Group 11					**Group 12** (I)		10		(4:1)
Group 12					**Group 13**				
Group 13					**Other** (6A br)	10		60	cfm1/cfm2
Other (9L)	20	20	20	20	**Other** (6A m)		35	10	0.32 (R1) 0.67 (R3)
Other (9E main)			5	5	**Other**				35 = 25+10
Other					**Other** (6N)			10	
Total Length	202	202	270	271	**Total Length**	115	162	200	

Figure12-5

12-3 Effective Length Calculation

Considering system geometry and fitting types, supply runs 3, 6, 7 (or 12) and 9 (or 8) are candidates for the longest supply path. Note that return R1 is for zone-1 supply runs 1, 2 and 3, return R2 is for zone-1 supply runs 4, 5, and 6, and return R3 is for all the upper level supply runs.

Figure 12-5 shows the effective length calculations for selected duct runs. Figure 12-6 shows that the run to second floor supply 9 (or 10) is the longest supply path (271 Feet) and that run from second floor return R3 is the longest return path (200 feet). Therefore, the effective length of the critical circulation path is 471 feet.

12-4 Design Friction Rate Calculation

Medium speed blower data indicates the blower delivers 1,300 Cfm when it operates against 0.59 IWC of external resistance. Since the resistance for the auxiliary heating coil (0.08 IWC), a supply outlet (0.03 IWC), a return (0.03 IWC), an open balancing damper (0.05 IWC) and an open hand damper (0.03 IWC) is 0.22 IWC, the available static pressure is 0.37 IWC. Therefore, the design friction rate is based on 0.37 IWC of pressure and 471 feet of effective length. These calculations are summarized by Figure 12-7 (next page).

Summary of Figure 12-5 Calculations				
Level	**Lower**		**Upper**	
Run	**S3**	**S6**	**S7**	**S9**
Supply TEL	202	202	270	271
Run	**R1**	**R2**	**R3**	
Return TEL	115	162	200	

Figure 12-6

12-5 Duct Sizing Calculations

Figure 12-8 (ahead two pages) summarizes the duct sizing calculations for this example. These comments apply:

- All duct sizes were read from the metal duct scale on the ACCA Duct Sizing Slide Rule.
- Zone 1 includes all the lower level supply runs and Zone 2 includes all the upper level supply runs.
- Supply runs 1, 2 and 3 are for zone 1 and return R1; supply runs 4, 5 and 6 are for zone 1 and return R2; supply runs 7 through 12 are for zone 2 and return R3.
- Supply branches 1 through 6 are for secondary trunks ST-1, ST-2 and ST-3; and supply branches 7

Friction Rate Worksheet for the Two-Zone Bi-Level Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.59 IWC Cfm = 1,300

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	
Electric resistance heating coil	0.08
Hot water coil	
Heat exchanger	
Low efficiency filter	
High or mid-efficiency filter	
Electronic filter	
Humidifier, UV lights, other	
Supply outlet	0.03
Return grille	0.03
Balancing damper	0.03
Zone damper (full open)	0.05
Total component losses (CPL)	0.22 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (0.59 - 0.22) = 0.37 IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (271 + 200) = 471 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.078 IWC/100 (round to 0.08)

$$FR = \frac{ASP \times 100}{TEL}$$

Friction Rate Chart

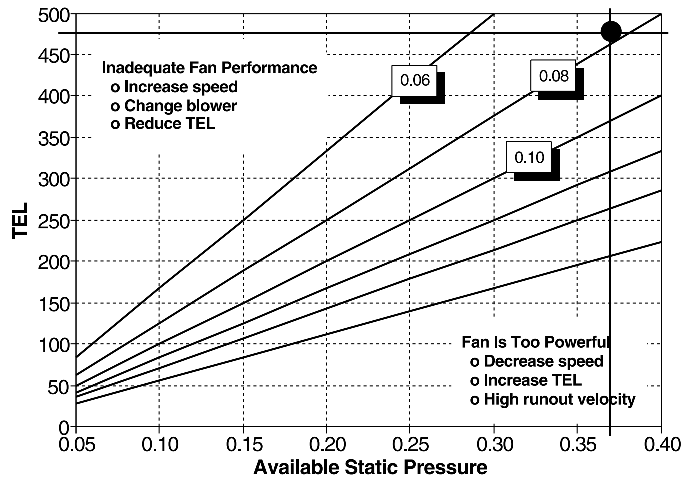


Figure 12-7

Duct Sizing Worksheet for the Two-Zone Bi-Level Example

HF = Blower Cfm / Manual J Heat Loss = 1,300/ 44,920 = 0.0289 **CF = Blower Cfm / Manual J Sensible Heat Gain** = 1,300 / 33,690 =0.0386	**FR Value** 0.08

Supply-Side Runouts									
Supply - Trunk	**Heating Btuh**	**Cooling Btuh**	**Heating Cfm**	**Cooling Cfm**	**Design Cfm + 20%**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
S1 — ST-1	4,250	2,970	123	115	147	7	ok	7	114
S2 — ST-1	3,860	3,251	112	125	151	7	ok	7	114
S3 — ST-2	3,970	3,456	115	133	160	7	ok	7	119
S4 — ST-2	2,780	2,300	80	89	107	6	ok	6	81
S5 — ST-2	3,800	2,570	110	99	132	7	ok	7	101
S6 — ST-2	4,440	3,694	128	143	171	8	ok	8	130
S7 — ST-4	4,590	3,780	133	146	175	8	ok	8	134
S8 — ST-4	4,620	3,791	134	146	176	8	ok	8	135
S9 — ST-4	2,350	1,825	68	70	85	6	ok	6	67
S10 — ST-4	3,020	2,797	87	108	130	7	ok	7	94
S11 — ST-5	3,810	3,359	110	130	156	7	ok	7	115
S12 — ST-5	3,430	2,592	99	100	120	7	ok	7	96
Supply-Side Trunks									
Run numbers:	S1, S2	**S-Trunk 1**	234	240	298	9	675	9	228
Run numbers:	S3 → S6	**S-Trunk 2**	433	464	569	12	725	12	431
Run numbers:	S1 → S6	**S-Trunk 3**	668	704	867	14	812	14	659
Run numbers:	S7 → S9	**S-Trunk 4**	334	363	435	11	660	11	336
Run numbers:	S10 → S12	**S-Trunk 5**	297	338	405	10	743	10	305
Run numbers:	S7 → S12	**S-Trunk 6**	631	700	840	14	786	14	641
Run numbers:	Primary	**S-Trunk 7**	1,300	1,300	1,300	16	932	16 std	1,300
Return-Side Runouts									
Return - Trunk	**Associated Supply Runs**		**Heating Cfm**	**Cooling Cfm**	**Design Cfm + 20%**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
R1 — RT-1	S1→ S3		349	373	458	11	694	11	347
R2 — RT-1	S4 → S6		318	330	409	11	621	11	312
R3 — RT-2	S7 → S12		631	700	840	14	786	16 std	641
Return-Side Trunks									
Run numbers:	Primary	**R-Trunk 1**	1,300	1,300	1,300	16	932	20 std	1,300

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 12-8

through 9 are for secondary trunks ST-4, ST-5 and ST-6.

- Secondary supply trunks ST-3 and ST-6 are for primary supply trunk ST-7.
- Returns R1, R2 and R3 feed primary return trunk RT-1.
- Blower Cfm is based on the peak black load for sensible cooling (the standard ***Manual J*** procedure).
- Room Cfm values are from standard ***Manual J*** procedure. (Each zone has four exposures, so each zone has cooling load diversity.)

- Because of the protocol for bi-level design, traditional (single-zone) Cfm values for branch runs and secondary trunks are increased by a factor of 1.2 (two-position dampers with 80 percent stops shift 20 percent of the heating or cooling capacity from the lower level to the upper level, or vice versa.)
- The final sizes for supply runouts and the secondary supply trunks (ST-1 trough ST-6) are based on the design friction rate (0.08 IWC/100) because air velocities are less than 900 Fpm.
- The size of primary supply trunk ST-7 is based on 1,300 Cfm. (The blower Cfm does not change when a portion of the blower Cfm is diverted from one floor to the other floor.)
- The final size of primary supply trunk ST-7 is based on the maximum allowable velocity (900 Fpm) because the air velocity for the design friction rate (0.08 IWC/100) exceeds 900 Fpm.
- The sizes of the zone return runs (R1, R2 and R3) are based on the adjusted Cfm values that were used to size the supply ducts.
- The final sizes of the zone return ducts (R1 and R2) are based on the design friction (0.08 IWC/100) because the air velocities are acceptable (700 Fpm or less).
- The final size of zone return duct R3 is based on the maximum allowable velocity (700 Fpm) because the air velocity for the design friction rate (0.08 IWC/100) exceeds 700 Fpm.
- The size of primary return trunk RT-1 is based on 1,300 Cfm. (Blower Cfm does not change when a some air flow is diverted from one floor to the other floor.)
- The final size of primary return trunk RT-1 is based on the maximum allowable velocity (700 Fpm) because the air velocity for the design friction rate (0.08 IWC/100) exceeds 700 Fpm.
- The ACCA Duct Sizing Slide Rule converts round sizes to equivalent rectangular sizes.

Balancing Cfm for the Two-Zone Bi-Level Example

Balancing Cfm Values for Level-One (Lower Level)					
Supply	**Normd Cfm**	**Share**	**OB Factor**	**Level Cfm**	**OB Cfm**
S1	114	17.3%			137
S2	114	17.3%			137
S3	119	18.1%	1.20	791	143
S4	81	12.3%			97
S5	101	15.3%			121
S6	130	19.7%			156
Total	659	100%		**Total**	791
Balancing Cfm Values for Level-Two (Upper Level)					
Supply	**Normd Cfm**	**Share**	**OB Factor**	**Level Cfm**	**OB Cfm**
S7	134	20.9%			161
S8	135	21.1%			162
S9	67	10.5%	1.20	769	80
S10	94	14.7%			113
S11	115	17.9%			138
S12	96	15.0%			115
Total	641	100%		**Total**	769

1) Normd = Normalized, OB = Overblow
2) Share = Percentage of level Cfm to each supply = (Normed Cfm for supply) / (Sum of normed supply Cfms for the level).
3) Level Cfm = Overblow factor x Sum of normalized supply Cfm values for the level.
4) Overblow Cfm = Share percentage x Level Cfm

Figure 12-9

12-6 Comments and Observations

This zoning arrangement has a limited objective, which is to reduce temperature control problems caused by the buoyancy of warm air. It can divert a little more blower Cfm to the level that is too warm (upper level) or the level that is too cool (lower level), but it does not have the ability to control space temperatures on a room-by-room basis. In this regard, system capabilities and owner satisfaction are diminished if the supply outlets are improperly sized, or return paths are inadequate.

For this type of zoned system, the concept of peak room load does not apply because the entire zone is a single floor that has diversity. The capacity of the central heating and cooling equipment and the design value for blower Cfm are based on the block cooling load for the entire dwelling. Room cooling load is determined by the standard ***Manual J*** procedure.

The concept of diversity applies to the airway sizing calculations for primary supply trunk (ST-1) and primary return trunk (RT-1). Therefore, as far as central components are concerned, the sizing calculations are identical to the calculations for a single-zone constant Cfm system. But as far as the rest of the air distribution system is concerned, design Cfm values are increased by a factor of 1.2 so duct airways can accommodate level-to-level capacity shifts.

This example is for constant capacity equipment that operates at a constant blower wheel speed, and for open-close zone dampers that have minimum position stops. Therefore, a bypass duct is not required because the blower Cfm and airflow rate through the central equipment is approximately constant has 20 percent of the supply air Cfm is diverted from one level to the other level.

12-7 Balancing a Simple Damper-Stop System

Figure 12-9 (previous page) shows how to calculate balancing Cfm values for 20 percent overblow to each level. Note that this does not occur simultaneously, one level has overblow when the other level is throttled, so the blower Cfm is relatively constant. These concepts are demonstrated by Figure 2-9:

- The normalized Cfm values from the Duct Sizing Worksheet are the default balancing Cfm values for a single-zone, constant Cfm system (see Section 6-11). These values add up to the blower Cfm.
- Some of the normalized Cfm values are for the lower level, and some are for the upper level.
- The total normalized Cfm for a level is the sum of the normalized supply outlet Cfms for the level.
- There is a total normalized Cfm value for each level. These two values add up to the blower Cfm.
- Each supply outlet gets a share of the total Cfm for a level (expressed as a percentage value in the example, or use the equivalent factor).
- The air flow to a level is increased by 20 percent (a 1.20 overblow factor) when the zone damper for the level is wide open while the other zone damper is on its stop. With branch runout dampers are open, air balancing measurements determine stop positions for 20 percent overblow to each level.
- With one zone damper open while the other zone damper is on its stops, the hand dampers for the supply air outlets on overblow level are adjusted for the overblow Cfm to each supply.
- The positions of the zone dampers are switched and the supply air outlets on new overblow level are adjusted for the overblow Cfm to each supply.
- If local hand damper adjustments cause more than a 10% change in level overblow (the target is 18% to 22% overblow to a level), adjust the damper stops as required.
- After a period of seasonal use, it may be that occupant experience causes further adjustment on one or more local hand dampers (to compensate for the lack of directional zoning for a level).

Zone Damper Retrofit

Single zone constant Cfm systems are converted air-zoned systems by adding zone thermostats, zone dampers, and supervisory controls. This requires a significant amount of field work and design work.

- Performance data for the existing blower is required. If this is not available, air-balancing instruments and procedures determine blower performance.
- Equipment performance data is required for excess air calculations, and to select measures for dealing with excess air.
- Information pertaining to the heating-cooling capacity of primary equipment and the capacity of ancillary devices, such as an electric resistance heating coil is required. If the equipment capacity is staged or modulated, performance data for full capacity and the associated blower Cfm, and for minimum capacity and the associated blower Cfm is required
- The existing duct system geometry must be sketched with airway sizes, duct run lengths, ***Manual D*** fitting numbers, and duct materials noted. This sketch must be related to the rooms and spaces on the floor plan.
- For duct in an unconditioned space, duct leakage and duct insulation must be determined to be adequate, or upgraded to a standard of care.
- A ***Manual J*** load calculation for zoned systems is required. This information and ***Manual Zr*** procedures determine the preliminary zoning plan. The final zoning plan depends on the result of the excess air calculations (if necessary, merging one or more zones reduces the amount of excess air).
- Airway sizes for a zoned system are calculated, per ***Manual D*** procedures. Existing airway size measurements are compared to calculated airway sizes for the retrofit system. If existing airway sizes are deficient for the available static pressure, some fittings and/or duct runs shall be altered or changed.
- Some form of blower capacity control may be required, or the existing blower may be replaced with a blower that is suitable for zoning duty.

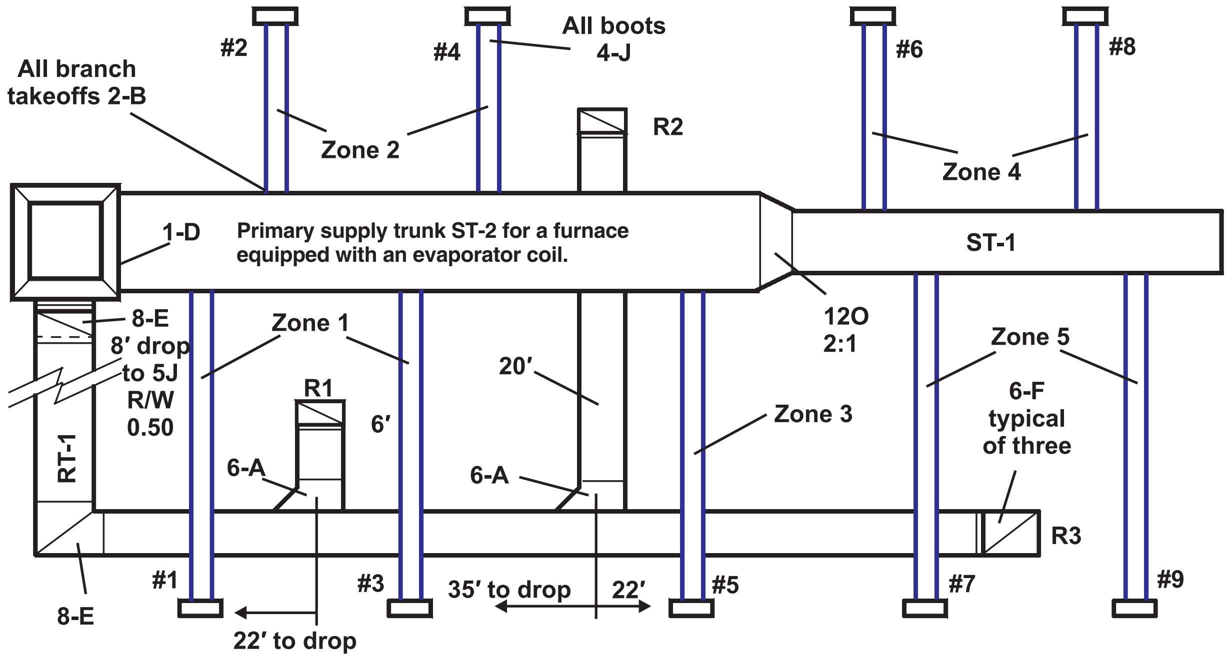


Figure 13-1

13-1 Candidate System

Figure 13-1 (previous page) shows a single-zone, constant Cfm, constant equipment speed system. The plan is to convert this system to a five-zone system. At this point, the practitioner speculates that the conversion will be generally compatible with the existing equipment and duct system.

The supply-side of the system has reducing plenum (secondary trunk ST-1 and primary trunk ST-2) and branch runouts. The return-side has three return runs that feed a primary return trunk (RT-1) that has one airway size along its length. The entire duct system is fabricated from metal and is reasonably tight.

13-2 Zoning Geometry Vs. Zoning Hardware

This system was not designed for air-zoning service, so zones are not fed by a system of secondary supply trunks. Therefore, zones are created by installing zone dampers in the branch runout ducts.

- Secondary trunks are desirable because two or more compatible rooms may be converted to a zone that has one thermostat for one zone damper installed in a secondary supply trunk. The exisitng duct system does not offer this oppertunity.
- For extended plenum, branch run-out geometry, two or more compatible rooms may form a zone that has one thermostat for two or more zone dampers installed in the branch runout ducts to each room.
- This example has five zones and five thermostats. One zone has one control damper and one thermostat. The other zones have two control dampers operated by one thermostat.

13-3 Bypass Duct

A bypass duct and bypass air control damper are required because the blower, compressor and furnace do not have some type of capacity control (staging or motor speed control, for example).The bypass duct airway is sized per ***Manual Zr*** procedures, as summarized by Section 9-5 of this manual and the last bullet item for Section 9-10 of this manual. Follow third party vendor instructions if zoning components and controls are used with heating-cooling equipment provided by others.

13-4 Returns

A low-resistance return path is required for each zone. If any room is isolated from one of the returns, transfer grilles shall complete the circulation path.

Manual J loads for the Zone Damper Retrofit Example

Run	Glass faces	Heating Btuh	Peak Cooling Btuh
1	S	3,810	3,888
2	W	3,800	2,975
3	E	3,970	3,200
4	W	4,250	3,438
5	E	3,860	3,010
6	W	4,500	3,263
7	E	4,590	3,500
8	N	4,870	3,750
9	N	2,350	1,690

1) Block loads (Btuh): Heat = 36,000; Sensible cooling = 26,000
2) Because of time-of-day issues, as they affect the block load and the room/space loads, the sum of the room cooling loads is more than the block cooling load (see Sections 9-3 and 9-10).

Figure 13-2

13-5 Data Collection

The practitioner shall verify that the existing blower and duct system is compatible with the proposed modification. This work begins with a ***Manual J*** load calculation (assuming that the practitioner does not have access to the original design calculations and equipment performance data.)

Manual J calculations shall be made because heating loads and peak sensible cooling loads determine supply Cfm for rooms and zones. Figure 13-2 shows the heating loads, peak room cooling loads and block cooling load produced by ***Manual J*** Eighth Edition, Version 2.10, or later (Sections 11-4, 11-5 and 11-7).

A field survey shall be done to evaluate the air moving capability of the blower and the carrying capacity of existing duct runs. Collect this information:

- Airway sizes for existing trunks and runouts (supply and return).
- The measured length of each runout (supply and return).
- The measured length of each trunk section (from the branch runout fitting to the air handler (supply and return).
- A ***Manual D*** group number for each fitting that is installed in the existing system.
- The blower Cfm and the available static pressures (ESP) for each PSC blower speed setting, or ECM blower Cfm setting.

- The available static pressure is the pressure difference across the air handler cabinet (positive pressure at the discharge collar minus the negative pressure at the return collar).

ESP (IWC) = Supply Pressure - (- Return Pressure)

Flow and Pressure Measurements

If manufacturer's blower performance data is not available, air balancing instruments are used to measure Cfm and static pressure. This is done with open hand dampers, a clean filter and a clean refrigerant coil (preferably a wet coil if the climate produces a latent load).

- For each PSC blower speed setting or relevant ECM blower Cfm settings, measure blower Cfm and positive static pressure in the supply trunk downstream from blower (before the first branch run); and return Cfm and negative static pressure in the return trunk upstream from the blower (after the last branch return).
- Do not measure unstable flow. Measure where air has a fully developed velocity profile. If possible, stay three to five feet downstream from a discharge opening, fitting or damper, and one to two feet upstream from a return opening.
- The testing environment is far from ideal (i.e, laboratory conditions), so the two Cfm values may be different.
- More Cfm data is obtained by measuring and summing flows through supply outlets and returns. These two sums may not agree, and either or both values may not match the Cfm value from the trunk measurement.
- Duct leakage affects supply and return flow. Data should be more consistent if the duct system is properly sealed. If supply and return ducts leak, trunk measurements are trusted more than discharge or return measurements.
- Look at the values produced by the Cfm tests, think about the factors that affected each measurement, then select a blower Cfm value.

Example

System geometry is summarized by Figure 13-1 and Figure 13-2 provides heating and cooling load values. Figure 13-3 provides dimensional data and summarizes the results of the blower tests for each blower wheel speed.

Figure 13-4 compares the blower test points with the system resistance curve. This diagram is useful because it checks the accuracy of the blower test. In this case, there is a high level of confidence in the accuracy of the blower test because there is good correlation between three test points and the system curve. (Section 2-5 provides instructions for drawing the system curve.)

Survey Data for the Zone Damper Retrofit Example

Supply Runs			
#	Size	Runout	Trunk
1	6"	24'	6'
2	6"	14'	12'
3	6"	24'	18'
4	6"	14'	24'
5	6"	24'	34'
6	6"	18'	48'
7	6"	24'	58'
8	6"	14'	63'
9	6"	24'	66'
ST-1	11"	(12" x 8")	
ST-2	14"	(20" x 8")	

Return Runs			
#	Size	Runout	Trunk
R1	10"	6'	30'
R2	12"	24'	43'
R3	8"	22'	43'
RT-1	16"	(20" x 10")	

PSC Blower Test Data		
Speed	Cfm	IWC
Low	840	0.20
Medium	920	0.31
High	1,060	0.36
Includes filter, coil and heater.		

Figure 13-3

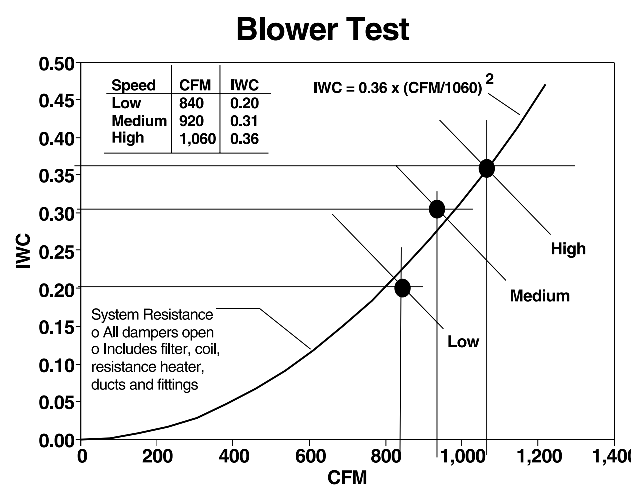


Figure 13-4

13-6 Airway Sizing Calculations

This example is for sheet metal construction, but this system could be fabricated from any combination of metal, duct board or flexible wire helix materials. In any case, the airway sizing calculations for this type of system are based on these concepts:

- Since zone dampers will be added to the system, the pressure drop for an open damper is

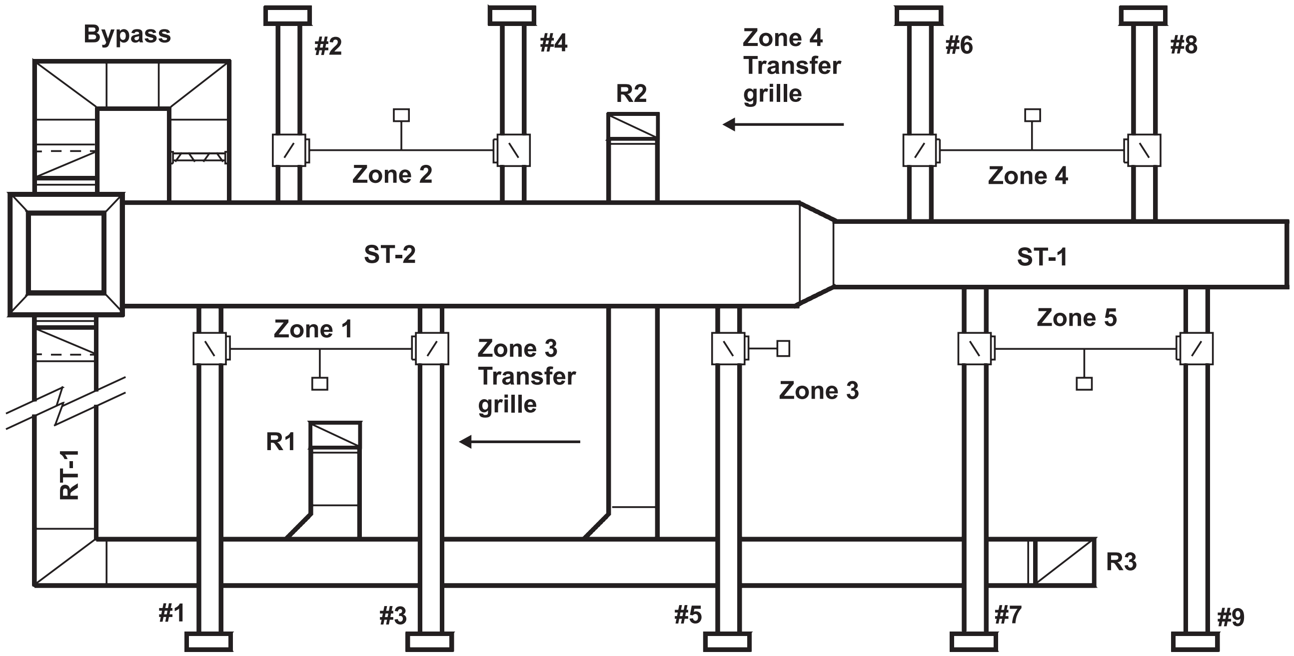


Figure 13-5

Effective Length Worksheet for the Zone Damper Retrofit Example									
Element	**Supply Run ID Number**				**Element**	**Return Run ID Number**			
	S1	S6	S9			R1	R2	R3	Notes
Trunk Length	6	48	66		**Trunk Length**	22	35	35	
Trunk Length					**Trunk Length**	8	8	8	
Runout Length	24	18	24		**Runout Length**	6	24	22	
Group 1 (D)	10	10	10		**Group 5** (J)	15	15	15	(R/W = .5)
Group 2 (B)	45	40	20		**Group 6** (F)	25	25	25	
Group 3					**Group 7**				
Group 4 (J)	30	30	30		**Group 8** (E)	20	20	20	(2 @ 10')
Group 8					**Group 10**				
Group 9					**Group 11**				
Group 11					**Group 12**				
Group 12 (O)		5	5		**Group 13**				
Group 13					**Other** (6A br)	10	60		Cfm1/Cfm2 0.39 (R1)
Other					**Other** (6A m)		10	35	0.72 (R2)
Other					**Other**				35 = 25+10
Total Length	115	151	155		**Total Length**	106	197	160	

Figure 13-6

subtracted from the available static pressure from the blower test.

- Zones combine rooms that have similar seasonal and hourly load characteristics. For this example, glass exposure direction and glass area are the primary factors that affect zoning decisions.
- The heating and cooling capacity of the central equipment and the flow rates for the primary

trunk ducts (supply and return) are based on the block load for the entire dwelling.

- Room Cfm values are based on room heating loads and peak sensible cooling loads. These load are summarized by Figure 13-2.

13-7 Effective Length Calculation

Figure 13-5 (previous page) shows the proposed modifications. Note that supply runs 1 and 3 are for zone 1, supply runs 2 and 4 are for zone 2, supply run 5 is for zone 3, supply runs 6 and 8 are for zone 4, and supply runs 7 and 9 are for zone 5. When transfer grilles are installed, return R1 is for zones 1 and 3, return R2 is for zones 2 and 4 and return R3 is for zone 5. Considering system geometry and fitting types, supply runs 1, 6, and 9 and returns R1, R2 and R3 may be in the longest circulation path.

Figure 13-6 (previous page) shows the effective length calculations for selected duct runs, and Figure 13-7 summarizes results. The summary shows that the effective length of the longest circulation path is 348 feet. This path is for supply run 6, which has a 151 foot length and return run R2, which has a 197 foot length.

Design Friction Rate Calculation

Blower test data shows that when the blower operates at high speed, 1,060 Cfm flows through a duct system that produces 0.36 IWC of external resistance. Since there were no zone dampers in the system when the blower was tested, the resistance for an open zone damper (0.05 IWC) is subtracted from the 0.36 IWC value. Therefore, the design friction rate is based on 0.31 IWC of pressure and 348 feet of effective length. These calculations are summarized by Figure 13-8 (next page).

Duct Sizing Calculations

Figure 13-9 (ahead two pages) summarizes the duct sizing calculations for this example. These comments apply :

- All duct sizes were read from the metal duct scale on the ACCA Duct Sizing Slide Rule.
- Supply runs 1 and 3 are for zone 1 and return R1; supply runs 2, and 4 are for zone 2 and return R2; supply run 5 is for zone 3 and return R1, supply runs 6, and 8 are for zone 4 and return R2; supply runs 7, and 9 are for zone 5 and return R3.
- Supply branches 6 through 9 are for secondary supply trunk ST-1. Supply branches 1 through 9 are feed by primary supply trunk ST-2.
- Return runs R1, R2 and R3 feed primary trunk RT-1.

Supply runout sizes are based on the Cfm values for the heating loads and the peak sensible cooling loads for the rooms.

Summary of Figure 13-6 Calculations			
Run	**#1**	**#6**	**#9**
Supply TEL	115	142	155
Run	**R1**	**R2**	**R3**
Return TEL	106	193	160
Path Total	221	335	315

Figure 13-7

- The final sizes of the supply runouts are based on the design friction rate (0.09 IWC/100) because air velocities are less than 900 Fpm.
- The size of supply trunk ST-1 is based on the Cfm values for zones 4 and 5.
- The size of supply trunk ST-1 is based on the design friction rate (0.09 IWC/100) because air velocity is less than 900 Fpm.
- The size of supply trunk ST-2 is based on blower air delivery (1,000 Cfm).
- The final size of supply trunk ST-2 is based on the maximum allowable velocity (900 Fpm) because the air velocity for the design friction rate (0.09 IWC/100) exceeds 900 Fpm.
- The sizes of the zone return runs (R1, R2 and R3) are based on Cfm values for the heating loads and the peak cooling loads.
- The final sizes of zone return ducts R1 and R2 are based on the maximum allowable velocity (700 Fpm) because the air velocities for the design friction rate (0.09 IWC/100) exceed 700 Fpm.
- The final size of the zone return duct R3 is based on the design friction rate (0.09 IWC/100) because the air is acceptable (700 Fpm or less).
- The size of return trunk RT-1 is based on blower air delivery (1,000 Cfm).
- The final size of return trunk RT-1 is based on the maximum allowable velocity (700 Fpm) because the air velocity for the design friction rate (0.09 IWC/100) exceeds 700 Fpm.
- A bypass duct with a modulating bypass air damper controlled by a temperature sensor and a pressure sensor, and other air management measures maintain adequate air flow thorough central equipment for any part-load condition.
- The bypass duct airway is sized per ***Manual Zr*** procedures, as summarized by Section 9-5 of this manual.
- The ACCA Duct Sizing Slide Rule provides equivalent rectangular sizes.

Friction Rate Worksheet for the Zone Damper Retrofit Example

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = 0.36 IWC   Cfm = 1,060

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	Included in test data
Electric resistance heating coil	Included in test data
Hot water coil	
Heat exchanger	
Low efficiency filter	Included in test data
High or mid-efficiency filter	
Electronic filter	
Humidifier, UV lights, other	
Supply outlet	Included in test data
Return grille	Included in test data
Balancing damper	Included in test data
Zone damper (full open)	0.05
Total component losses (CPL)	0.05 IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (0.36 - 0.05) = 0.31 IWC

Step 4) Total Effective Length (TEL)

Supply-side TEL + Return-side TEL = (151 + 197) = 348 Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = 0.09 IWC/100

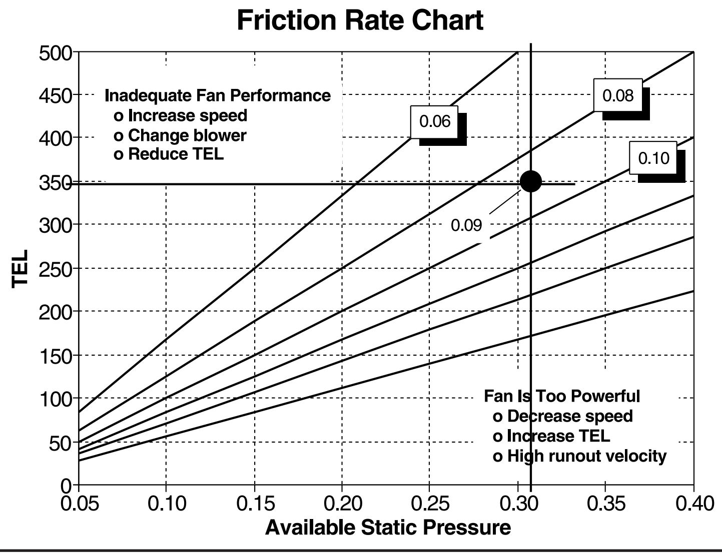


Figure 13-8

Duct Sizing Worksheet for the Zone Damper Retrofit Example									
HF = Blower Cfm / Manual J Heat Loss = 1,060 / 36,000 = 0.0294 **CF = Blower Cfm / Manual J Sensible Heat Gain =** 1,060 / 26,000 = 0.0408									**FR Value** 0.09
Supply-Side Runouts									
Supply - Trunk	**Heating Btuh**	**Cooling Btuh**	**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
S1 — ST-2	3,810	3,888	112	159	159	7	ok	7	See Note 5
S2 — ST-2	3,800	2,975	112	121	121	7	ok	7	
S3 — ST-2	3,970	3,200	117	131	131	7	ok	7	
S4 — ST-2	4,250	3,438	125	140	140	7	ok	7	
S5 — ST-2	3,860	3,010	113	123	123	7	ok	7	
S6 — ST-1	4,500	3,263	132	133	133	7	ok	7	
S7 — ST-1	4,590	3,500	135	143	143	7	ok	7	
S8 — ST-1	4,870	3,750	143	153	153	7	ok	7	
Supply-Side Trunks									
Run numbers: S6 → S9		**S-Trunk 1**	480	498	498	11	755	11	See Note 5
Run numbers: Primary		**S-Trunk 2**	1,060	1,060	1,060	14	992	16 std	
Return-Side Runouts									
Return - Trunk	**Associated Supply Runs**		**Heating Cfm**	**Cooling Cfm**	**Design Cfm**	**Round Size**	**Velocity Fpm**	**Final Size**	**Normed Cfm**
R1 — RT-1	S1, S3, S5		342	412	412	10	756	10	See Note 5
R2 — RT-1	S2, S4, S6, S8		512	548	548	11	830	12	
R3 — RT-1	S7, S9		204	212	212	8	607	8	
Return-Side Trunks									
Run numbers: Primary		**R-Trunk 1**	1,060	1,060	1,060	14	992	16 std	See Note 5

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

Figure 13-9

Comments and Observations

This example deals with two air distribution systems; the as-built system (which may not be properly designed or documented), and the desired system, which has been designed to function as a five-zonesystem. Geometrically, these systems are identical (except for the bypass duct), and they both use the same blower, but there are differences in some duct airway sizes.

Figure 13-10 (nest page) compares round-duct sizes for the two systems. This shows that some of the existing supply-side runouts are 1-inch smaller than the size required for the zoned system. As far as air delivery is concerned, this 1-inch difference is not critical unless the runout duct is in the longest circulation path or a path that is nearly as long as the critical path. (The design friction rate is based on the length of the critical circulation path.) Since runs 6, 7 and 8 are part of a long circulation path, the existing 6-inch ducts are replaced with 7-inch ducts.

Figure 13-10 shows that most of the existing supply-side airways and return-side airways are large enough for the zoned airflow rates and the design friction rate; but three of the runs (ST-2, R1 and RT-1) are 1-inch too small if airway size is based on the recommended limit for air velocity. Fortunately, modifications are not be required because a 1-inch discrepancy will not result in excessive velocities.

Figure 13-5 shows a zoning plan that is intended to be compatible with ***Manual Zr*** procedures for selecting zones. This five-zone plan was used by demonstrate airway sizing procedures for a retrofit system, and it is adequate for this purpose because the airway sizing procedures do not depend on the number of zones.

However, it may be that single-zone, constant Cfm equipment with no capacity control, a bypass duct, and other air relief measures will not manage the excess air for a five-zone plan. If ***Manual Zr*** procedures show this to be the case, the number of zones must be reduced.

The maximum amount of excess air is the difference between the blower Cfm and the smallest zone Cfm value (i.e., the zone damper for the critical zone is open, all other zone dampers are closed). Merging one or more zones reduces the amount of excess air, but this also reduces the amount of zone control. In this regard, the need to manage excess air supercedes the desire for more zones.

The scope of a retrofit project is more than just adding zone dampers to an existing single-zone, constant Cfm system. Air management considerations and duct system modifications vary on a case-by-case basis. ***Manual Zr*** procedures determine what air management measures are required for the attributes of the primary heating-cooling equipment vs. the number of zones. Manual D procedures determine if airway sizes are adequate, or if some of the attributes of the duct system must be changed.

Duct System Changes for the Zone Damper Retrofit Example

Run	Existing Size	Desired Size		Change Required
		For FR	For Fpm	
#1	6"	7"	7"	ok
#2	6"	6"	6"	ok
#3	6"	7"	7"	ok
#4	6"	7"	7"	ok
#5	6"	6"	6"	ok
#6	6"	7"	7"	New run
#7	6"	7"	7"	New run
#8	6"	7"	7"	New run
#9	6"	5"	5"	ok
ST-1	11"	11"	11"	ok
ST-2	14"	14"	15"	ok
R1	10"	10"	11"	ok
R2	12"	11"	12"	ok
R3	8"	8"	8"	ok
RT-1	16"	14"	17"	ok

The existing blower is acceptable, if operated at high speed.

Figure 13-10

Applicable Appendices

Applicable appendices are part of the standard.

Appendix 1 — Table and Equations

Appendix 2 — Friction Charts, Duct Slide Rules and Equivalency Tables

Appendix 3 — Fitting Equivalent Lengths

Appendix 4 — Fitting Equivalent Length Adjustments

Appendix 5 — Terminology

Appendix 1

Tables and Equations

The following tables and equations are used to design air distribution systems. In some cases, a brief description or discussion is provided with an equation or table. Refer to the main body of this manual for more information about a particular item.

A1-1 Air Velocity

Table A1-1 provides guidelines for air velocity through duct airways, supply outlets and return grilles. If recommended velocities are used, system resistance and generated noise are less than if maximum velocities are used.

- As far as resistance is concerned, the maximum velocity limit may be exceeded if the available external static pressure produced by the blower equals or exceeds system resistance.
- The design friction rate is affected if air velocity limits are exceeded (fitting equivalent lengths are for 900 Fpm or less. See Appendix 4 for the adjustment procedure).
- System resistance considerations supercede velocity considerations (minimum acceptable airway size shall be based on the local Cfm value and the design friction rate). Air way size shall be increased if the local air velocity exceeds the maximum limit.
- As far as noise is concerned, the maximum velocity limit may be exceeded if duct runs, supply outlets and return grilles do not produce objectionable noise (see Appendix 13).
- Low velocity is not an issue (see Appendix 15).

A1-2 Door Cut for Return Air

A door cut establishes a return path from a room that is isolated from a central return. This practice is not recommended by ***Manual D***.

- Table A1-2 (next page) shows that adequate flow area requires substantial undercut.
- Adequate door cuts create appearances issues.

Air Velocity for Noise Control Subject to Notes 1, 2 and 8

Component	Supply-Side (Fpm)				Return-Side (Fpm)			
	Conservative		Maximum		Conservative		Maximum	
	Rigid	Flex	Rigid	Flex	Rigid	Flex	Rigid	Flex
Trunk Ducts	700	700	900	900	600	600	700	700
Branch Ducts	600	700	900	900	500	600	700	700
Supply Outlet Face Velocity	Size for Throw		700 Note 7		—		—	
Return Grille Face Velocity	—		—		—		500	
Filter Grille Face Velocity	—		—		—		300	

1) The design friction rate is affected if air velocity exceeds 900 Fpm (fitting equivalent lengths are for 900 Fpm or less).
2) System resistance considerations supercede velocity considerations (minimum acceptable airway size shall be based on the local Cfm value and the design friction rate). Air way size shall be increased if the local air velocity exceeds the maximum limit.
3) This table applies to metal duct with transverse seams and metal fittings (duct runs and fittings not lined or wrapped with insulating material).
4) This table applies to flexible wire helix duct with duct board junction box fittings.
5) Maximum velocities may be exceeded when construction has less surface irregularities (no transverse seams or less irregularity at transverse seams, and very efficient fittings); and has a sound absorbing attribute (duct board or duct liner).
6) Authoritative guidance concerning velocity limits for aerodynamically efficient and/or sound absorbing designs is not available at this time.
7) The velocity limit for a supply outlet may be ignored if the noise criteria (NC) value for a grille, register or diffuser is 30 or less over the range of Cfm values that will flow through the device (or combination of devices, if a damper is involved), during any mode of system operation.
8) Air velocity limits are superceded by measured noise criteria (NC) values for low rise dwellings (Notes 1 and 2 still apply).
   · NC values measured by sound meter in middle of the room when normal human ear perceives maximum HVAC system noise.
   · Measured NC equals or exceeds 30 with comfort system off; measured NC shall not increase by more than 3 with comfort system on.
   · Measured NC less than 30 with comfort system off; measured NC shall not exceed 33 with comfort system on.

Table A1-1

Door Cut Height for 300 Fpm Air Velocity							
Cfm Under Door	Door Width (Inches)						
	24	30	36	42	48	54	60
	Clearance (Inches) to Floor or Top of Carpet						
100	2.0	1.6	1.3	1.1	1.0	0.9	0.8
200	4.0	3.2	2.7	2.3	2.0	1.8	1.6
300	6.0	4.8	4.0	3.4	3.0	2.7	2.4
400	8.0	6.4	5.3	4.6	4.0	3.6	3.2
500	10.0	8.0	6.7	5.7	5.0	4.4	4.0
600	12.0	9.6	8.0	6.9	6.0	5.3	4.8
700	14.0	11.2	9.3	8.0	7.0	6.2	5.6
800	16.0	12.8	10.7	9.1	8.0	7.1	6.4
900	18.0	14.4	12.0	10.3	9.0	8.0	7.2
1,000	20.0	16.0	13.3	11.4	10.0	8.9	8.0
1,200	24.0	19.2	16.0	13.7	12.0	10.7	9.6
1,400	28.0	22.4	18.7	16.0	14.0	12.4	11.2
1,600	32.0	25.6	21.3	18.3	16.0	14.2	12.8

Table A1-2

- Adequate door cuts create privacy issues (a significant amount of noise is transmitted though a small crack, a door cut provides negligible attenuation).

A1-3 Sensible Heat Equation

For generalized thermodynamic calculations, the sensible heat equation defines the relationship between supply air Cfm, the load for the conditioned space, supply air temperature (dry-bulb) and room air temperature (dry-bulb). Separate calculations are made for cooling and heating because the load (heating or cooling), thermostat set point and supply air temperature depend on the mode of operation.

$$\text{Heating Cfm} = \frac{\text{Heating Load Btuh}}{1.1 \times (\text{Supply DB} - \text{Room DB})}$$

$$\text{Cooling Cfm} = \frac{\text{Sensible Cooling Load Btuh}}{1.1 \times (\text{Room DB} - \text{Supply DB})}$$

A1-4 Room Air Flow Equations

For *Manual D* calculations, room air flow (supply Cfm) depends on the ratio of the room load to the design load (block heating load or block sensible cooling load from *Manual J* calculations). This ratio is applied to the design blower Cfm (determined during the *Manual S* equipment selection procedure). This calculation is made in two steps: first, a heating factor or cooling factor is evaluated, then the room Cfm is evaluated.

$$\text{Heating Factor (HF)} = \frac{\text{Design Blower Cfm}}{\text{Design Heating Load}}$$

$$\text{Cooling Factor (CF)} = \frac{\text{Design Blower Cfm}}{\text{Design Sensible Cooling Load}}$$

$$\text{Room Cfm (Heat)} = \text{HF} \times \text{Room Heating Load}$$

$$\text{Room Cfm (Cool)} = \text{CF} \times \text{Room Sensible Cooling Load}$$

A1-5 Friction Rate Equation

The friction rate (FR) equation relates available pressure (IWC), total effective length (TEL) with a friction rate (IWC/100 Ft). This equation is important because all duct sizing slide rules and friction charts are based on the friction rate concept.

$$\text{FR (IWC / 100 Ft)} = \frac{\text{Available Pressure (IWC)} \times 100}{\text{TEL (Ft)}}$$

If the friction rate equation is rearranged, the pressure drop across (PD) a duct run equals the product of the friction rate (FR) and the total effective length (TEL).

$$\text{PD (IWC)} = \text{FR (IWC/100 Ft)} \times \text{TEL (Ft)}$$

A1-6 System Resistance Equation

The duct system (or duct run) resistance curve (duct curve) is a graph of duct pressure drop (PD) as a function of air flow rate (Cfm). The effect of blower wheel speed changes are estimated by drawing the duct curve and the blower performance curve on the same graph. For this equation, PD_X represents the pressure drop (IWC) at the Cfm of interest (Cfm_X) and PD_1 represents the pressure drop (IWC) at a known Cfm (Cfm_1).

$$PD_x = PD_1 \times (Cfm_x \;/\; Cfm_1)^2$$

A1-7 Duct Flow Equation

The Cfm that flows through a duct airway equals the cross-sectional area (A in SqFt) multiplied by the average velocity of the flow (V in Fpm). For design work, flow rate and design velocity are known, so cross-sectional area is calculated. For balancing work, cross-sectional area (see Section A1-10) and average velocity are measured, so Cfm is calculated.

A (SqFt) = Cfm / V (Fpm)
Cfm = V (Fpm) x A (SqFt)
V (Fpm) = Cfm / A (SqFt)

A1-8 Equivalent Duct Size Based on Friction Rate

For a given airway material, these equations produce duct sizes that have the same friction rate (IWC/100 Ft). For round airways diameter D is in inches. For rectangular airways sides W (width) and H (height) are in inches. For oval airways flow area A is in square-inches and perimeter P is in inches.

Round equivalent of a rectangular airway

$$D = 1.30 \times \frac{(W \times H)^{0.625}}{(W + H)^{0.250}}$$

Round equivalent for an oval airway

$$D = 1.55 \times \frac{A^{0.625}}{P^{0.250}}$$

A1-9 Hydraulic Diameter

For a given material, duct shapes that have the same hydraulic diameter (HD) have the same friction rate (IWC/100). This equation determines the hydraulic diameter for any airway shape. All dimensions are in inches.

$$HD = \frac{4 \times \text{Cross-sectional Area}}{\text{Perimeter}}$$

A1-10 Areas and Perimeters

These equations return cross-sectional area (A) and perimeter (P) for round, rectangular and oval shapes. All dimensions are in inches, so the corresponding area is in square inches. For square feet area, divide the square inch area by 144.

Round shape (radius R and diameter D)
$A = \pi R^2$
$P = \pi D$

Rectangle Shape (width W and height H)
$A = W \times H$
$P = 2W + 2H$

Oval Shape (minor diameter B and major diameter A)
$A = \pi B^2 / 4 + B \times (A-B)$
$P = \pi B + 2 \times (A-B)$

Triangular Shape (base B, height H and side S)
$A = 0.50\ B \times H$
$P = B + 2S$

Duct Roughness Factors (RF)	
Material	***Manual D* Default**
Metal	1.00
Duct Board	1.39
Duct Liner	1.65
Wire Helix (default standard of care)	1.76
See Section 4-3 for default standard of care.	

Table A1-3

A1-11 Duct Slide Rule Equations

For round duct, airway diameter (D in Inches) depends on the air flow rate (Cfm), a friction rate value (IWC per 100 Ft) for metal duct (MFR) and a duct material roughness factor (RF). The following equation summarizes the relationships:

- Table A1-3 provides duct roughness factors.
- Airway diameters produced by the following equation are approximate (typically within one-half inch, or less, of diameter read from a friction chart or duct sizing slide rule).
- See the side bar for more information about this equation.
- See Sections A1-8 and A1-9 for equations that convert a round airway to an equivalent-resistance shape.

$$D\ (In) = 0.55 \times \frac{Cfm^{0.40}}{(MFR\ /\ RF)^{0.20}}$$

For example; find the airway diameter for a round metal duct for 600 Cfm and a 0.12 IWC/100 Ft friction rate. Then find the airway diameter for a flexible wire helix duct.

$$\text{Metal } D\ (In) = 0.55 \times \frac{600^{0.40}}{(0.12\ /\ 1.00)^{0.20}} = 10.9$$

$$\text{Flex } D\ (In) = 0.55 \times \frac{600^{0.40}}{(0.12\ /\ 1.76)^{0.20}} = 12.2$$

For round duct, air velocity (V in Fpm) depends on the air flow rate (Cfm) and duct diameter (D in Inches). This equation summarizes the relationships:

$$V\ (Fpm) = 183.4 \times (Cfm / D^2)$$

For round duct, velocity pressure (VP in IWC) depends on the air flow rate (Cfm) and duct diameter (D in Inches). This equation summarizes the relationships.

$$VP\ (IWC) = 0.002092 \times (Cfm^2 / D^4)$$

A1-12 Duct System Efficiency

Refer to Appendix 10 and Appendix 11 for information about duct system efficiency. Also see Section 23 and Table 7 of the unabridged version of ***Manual J***, Eighth Edition.

A1-13 Metal Duct Performance for Standard Air

Mathematical models for duct performance are provided by various editions of the ASHRAE Handbook of Fundamentals. The relevant equations for round duct are provided here:

1) $PD = 12 \times FF \times (L / D) \times VP$

Where:
PD = Pressure drop (IWC)
FF = Friction factor (Figure A1-1, next page)
L = Duct length (Ft)
D = Round duct diameter (In)
VP = Velocity Pressure (IWC)

2) $ff = 0.11 \times (ar / D + 68 / RE)^{0.25}$

2A) If ff ³ 0.018; then $FF = ff$

2B) If $ff < 0.018$; then $FF = 0.85 \times ff + 0.0028$

Where:
ff = A conditional friction factor (Figure A1-2, next page)
ar = absolute roughness of material (Ft)
For galvanized steel, transverse seams: ar = 0.0003
D = Round duct diameter (In)
RE = Reynolds Number for duct flow

3) $RE = 8.56 \times D \times V$

Where:
RE = Reynolds Number for standard air
D = Round duct diameter (In)
V = Average air velocity through airway (Fpm)

Looking at these equations, it is immediately apparent that they relate duct diameter (D), duct chart friction rate (FR in IWC/100 FT) and Cfm. In other words, they provide this relationship:

$D = \Phi(Cfm \text{ and } FR)$

In other words, the diameter is a function of Cfm and friction rate (as demonstrate by friction charts and duct slide rules).

However, the equations do not have much utility in this form. A little manipulation produces a user-friendly model, as demonstrate here.

A) Rearrange Equation 1 (solve for D):

$D = 12 \times FF \times (L / PD) \times VP$

B) Note that L / PD is the reciprocal of pressure drop per foot of length, and that this is equivalent to the reciprocal of the friction rate, so the equation becomes:

$D = 12 \times FF \times (100 / FR) \times VP$

C) Note that velocity pressure (VP) is a function of Cfm and duct diameter (D), as demonstrated here (where, A is the cross-sectional area of the airway in SqIn).

$A = (3.142 / 4) \times D2$
$A = 0.7855 \times D2$

$V = Cfm / A$
$V = Cfm / (0.7855 \times (D2 / 144))$
$V = Cfm / (0.00545 \times D2)$
$V = 183.35 \times (Cfm / D2)$

$VP = (V / 4{,}007)2$
$VP = (183.352 \times Cfm2) / (D4 / 4{,}0072)$
$VP = 0.002094 \times (Cfm2 / D4)$

D) The velocity pressure equation (step-3) is inserted into the duct diameter equation (step-2). Then constants are combined, variables are combined and the equation is rearranged.

$D = 12 \times FF \times (100 / FR) \times VP$
$D = 12 \times FF \times (100 / FR) \times VP \times 0.002094 \times (Cfm2 / D4)$
$D5 = 2.512 \times FF \times FR \times Cfm2$

E) FF is a conditional function of D (see equation 2), so the complexity of the D5 equation is significantly increased. To keep things simple, an approximate numerical value for FF is used. Equations 3, 2, 2A and 2B provided the basis for this calculation:

$RE = 8.56 \times D \times V$
$V = 183.35 \times (Cfm / D2)$
$RE = 8.56 \times D \times 183.35 \times (Cfm / D2)$
$RE = 1{,}569.45 \times (Cfm / D)$

$ff = 0.11 \times (ar / D + 68 / RE)0.25$
Substitute $1{,}569.45 \times (Cfm / D)$ for RE
$ff = 0.11 \times (ar / D + (0.043327 \times D / Cfm))0.25$

ff is a function of D and RE (see equation 2). Spreadsheet calculations produce ff values for duct diameters that range from 6 inches to 34 inches and Cfm values that range from 100 Cfm to 4,800 Cfm. (Table A1-3 previous page).

FF is a conditional function of ff (see equations 2A and 2B). Spreadsheet calculations are used to produce FF values for duct diameters that range from 6 inches to 34 inches and Cfm values that range from 100 Cfm to 4,800 Cfm. (Figure A1-1). For Figure A1-1, the average FF value is 0.021.

The average FF values is substituted into the D5 equation with this result (RF = 1.0 for metal).

$$D\ (In) = 0.553\ x\ \frac{Cfm^{0.40}}{(MFR\ /\ RF)^{0.20}}$$

F) For duct materials other than metal, the metal friction rate value (MFR) in the denominator of the diameter equation is divided by the duct roughness factor (RF). RF values are provided by Table A1-3.

Galvanized Metal Duct FF Value (ar = 0.0003)

D (In)	Cfm					
	100	300	600	1,200	2,400	4,800
6	0.025	0.019	0.017	0.015	0.013	0.012
7	0.026	0.020	0.017	0.015	0.013	0.012
8	0.027	0.020	0.018	0.015	0.014	0.012
9	0.028	0.021	0.018	0.016	0.014	0.012
10	0.028	0.022	0.018	0.016	0.014	0.013
12	0.030	0.023	0.019	0.016	0.014	0.013
14	0.031	0.023	0.020	0.017	0.015	0.013
16	0.032	0.024	0.020	0.017	0.015	0.013
18	0.033	0.025	0.021	0.018	0.016	0.014
20	0.034	0.026	0.021	0.018	0.016	0.014
24	0.036	0.027	0.022	0.019	0.016	0.014
26	0.036	0.027	0.023	0.019	0.017	0.015
28	0.037	0.028	0.023	0.020	0.017	0.015
30	0.037	0.028	0.024	0.020	0.017	0.015
32	0.038	0.029	0.024	0.020	0.017	0.015
34	0.038	0.029	0.025	0.021	0.018	0.015

Figure A1-1

Galvanized Metal Duct ff Value (ar = 0.0003)

D (in)	Cfm					
	100	300	600	1,200	2,400	4,800
6	0.025	0.019	0.016	0.014	0.012	0.011
7	0.026	0.020	0.017	0.014	0.013	0.011
8	0.027	0.020	0.017	0.015	0.013	0.011
9	0.028	0.021	0.018	0.015	0.013	0.011
10	0.028	0.022	0.018	0.015	0.013	0.012
12	0.030	0.023	0.019	0.016	0.014	0.012
14	0.031	0.023	0.020	0.017	0.014	0.012
16	0.032	0.024	0.020	0.017	0.015	0.012
18	0.033	0.025	0.021	0.018	0.015	0.013
20	0.034	0.026	0.021	0.018	0.015	0.013
24	0.036	0.027	0.022	0.019	0.016	0.014
26	0.036	0.027	0.023	0.019	0.016	0.014
28	0.037	0.028	0.023	0.020	0.017	0.014
30	0.037	0.028	0.024	0.020	0.017	0.014
32	0.038	0.029	0.024	0.020	0.017	0.014
34	0.038	0.029	0.025	0.021	0.017	0.015

Figure A1-2

Appendix 2

Friction Charts, Duct Slide Rules and Equivalency Tables

Friction charts and duct slide rules correlate performance parameters for duct air flow. The variables are air flow rate (Cfm), friction loss per 100 feet of straight duct (FR value), round duct diameter (inches) and the average velocity of the air moving through the duct (Fpm). Duct slide rules and equivalency tables provide ancillary information, such as converting round airways to equivalent rectangular airways, or vice versa.

A2-1 Friction Charts

Friction charts (and duct slide rules) provide a graphic model of the relationships that govern ducted air flow. Figure A2-1 (next page) provides an example of a friction chart. For this chart, air flow rate is represented by the Cfm values on the Y-axis, friction rate is represented by the FR values on the X-axis, round duct size is represented by the lines that slope from the lower left to the upper right, and air velocity is represented by the lines that slope from the upper left to the lower right.

If any two of these four variables are known, the lines that represent these variables intersect at some point on the chart and values for the other two variables are read from the chart, based on the intersection point. For example, if 100 Cfm is flowing through a 7-inch galvanized metal duct, the friction rate is 0.04 IWC/100 and air velocity is 360 Fpm (approximately). Figure A2-1 shows this mapping.

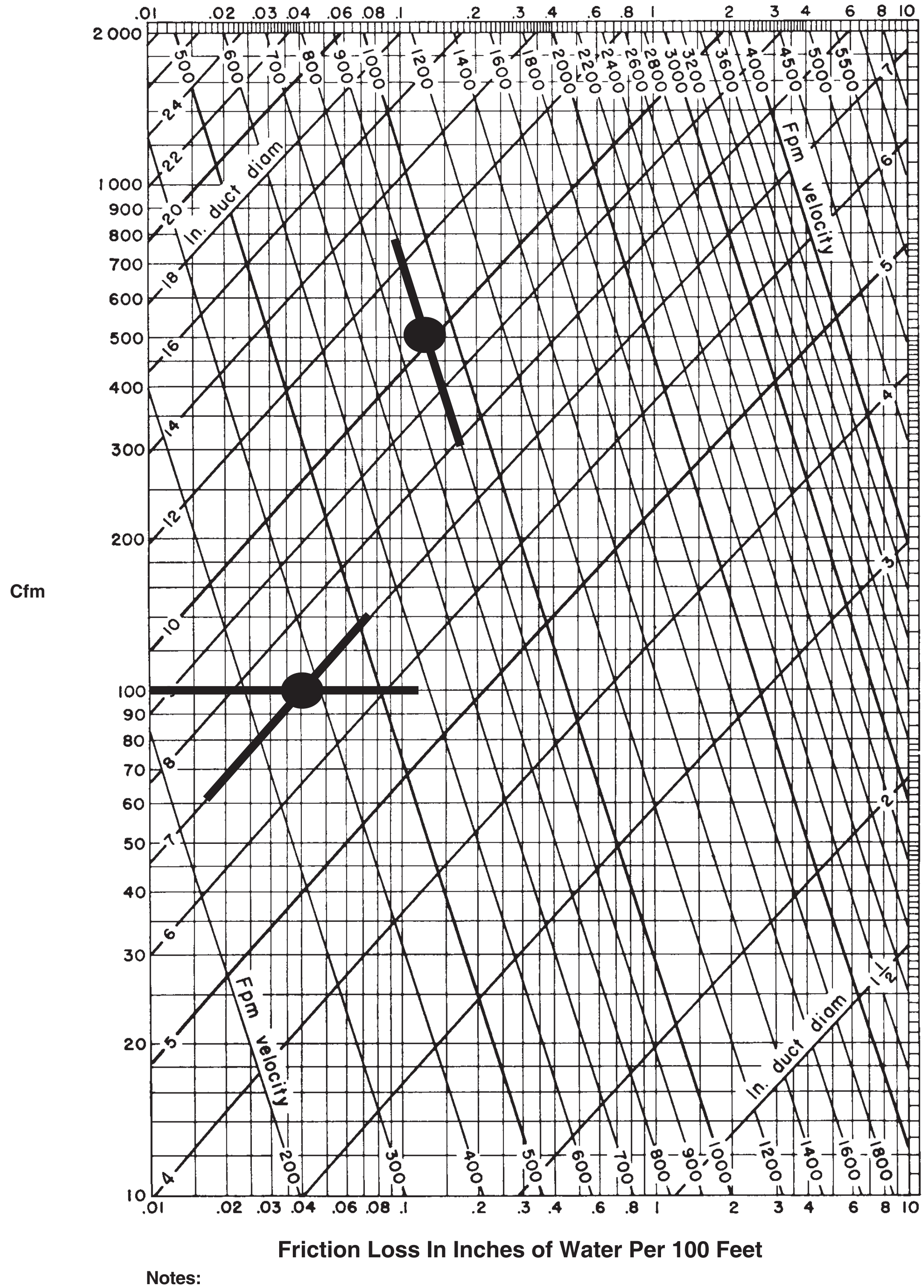
Other inputs are processed in a similar manner. For example, Figure A2-1 also shows that if the flow rate is 500 Cfm and the maximum allowable velocity is 900 Fpm, the smallest possible duct diameter is 10 inches (approximately) and the corresponding friction rate is 0.13 IWC/100 (approximately).

Friction Rate versus Pressure Drop

Because of friction, there is a pressure drop (ΔP IWC) across a duct run. This pressure drop is related to, but is not normally equal to, the friction rate. The exception occurs when the duct run is exactly 100 feet long. In this case the pressure drop is read directly from the duct slide rule because there is no difference between the friction rate (IWC/100 Ft) and the pressure drop value (IWC ΔP for a 100 Ft duct). For example, if the relevant friction rate is 0.08 IWC/100 and the duct is 100 feet long, the pressure drop from entrance to exit is 0.08 IWC.

When a duct run is not exactly 100 feet long, the friction rate and the pressure drop are not equal. This equation returns the pressure drop (IWC) for a given friction rate (IWC/100) and length (Ft).

Figure A2-1
Round Galvanized Metal Duct
10 Cfm to 2,000 Cfm

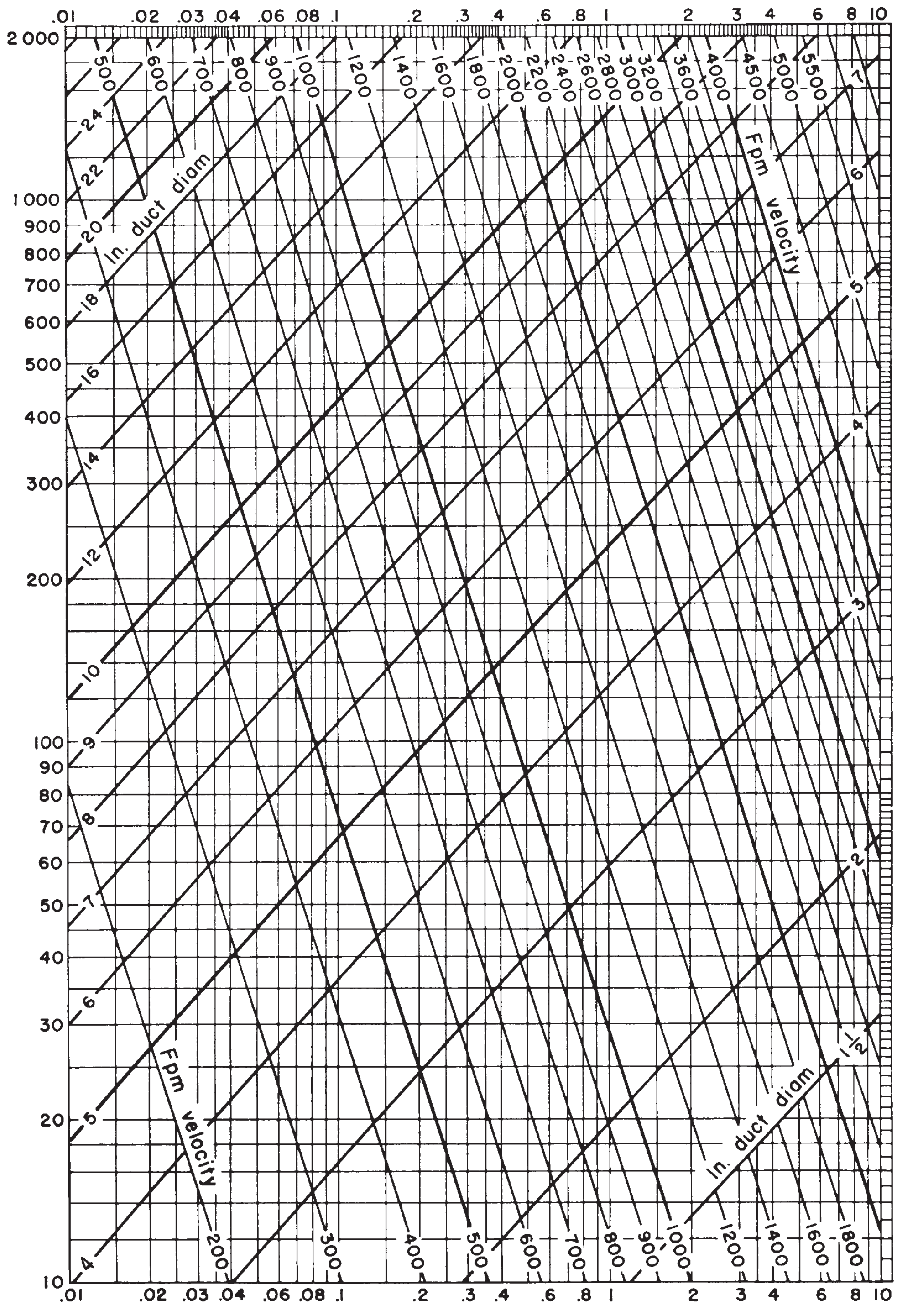


Notes:
1) Correction required for non-standard air.
2) 40 joints per 100 feet.
3) Roughness = 0.0005 feet.

Chart 1

Round Galvanized Metal Duct
10 Cfm to 2,000 Cfm

Cfm



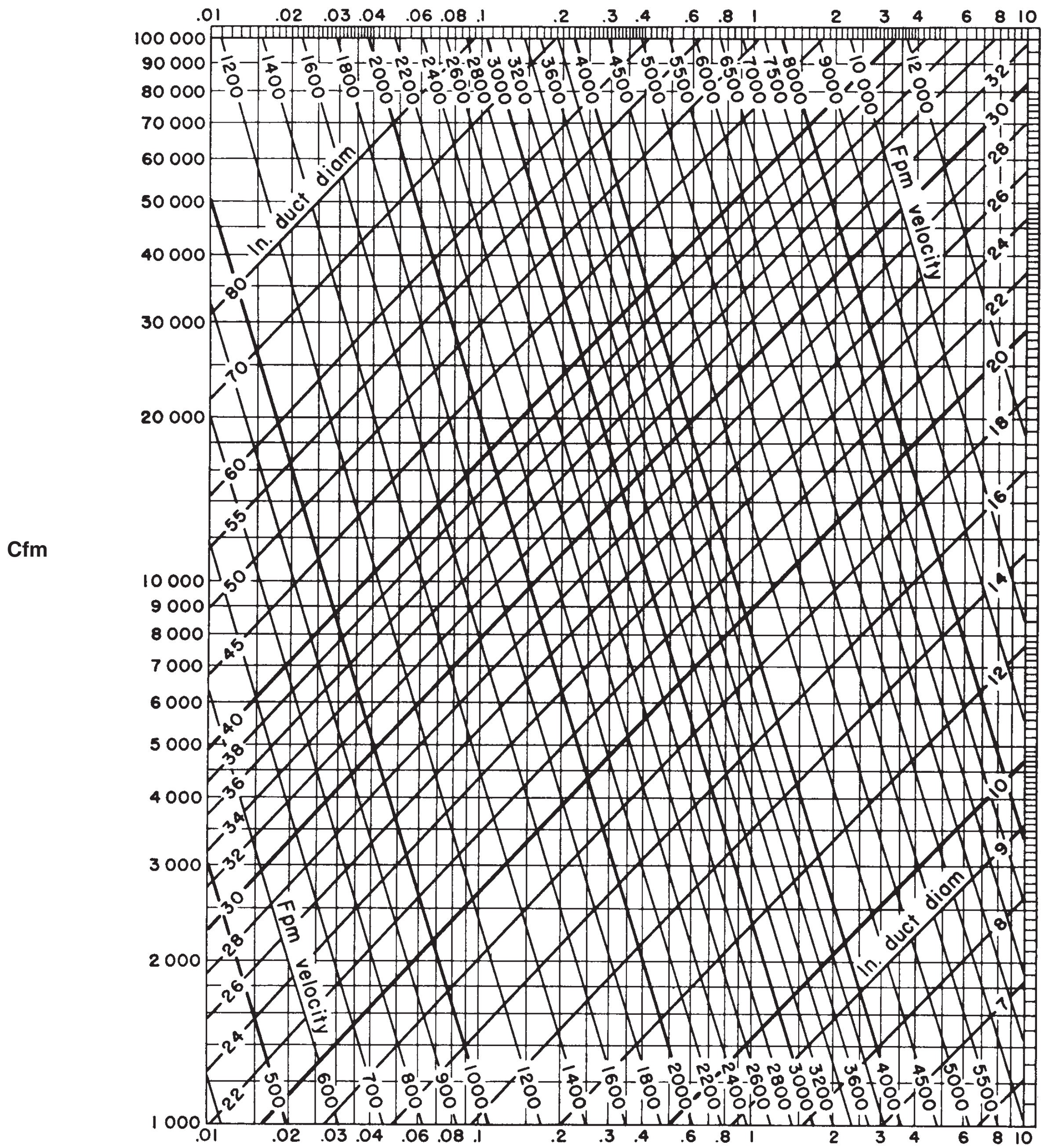
Friction Loss In Inches of Water per 100 Feet

Notes:

1) Correction required for non-standard air.
2) 40 joints per100 Feet.
3) Roughness = 0.0005 Feet

Chart 2

Round Galvanized Metal Duct
1,000 Cfm to 100,000 Cfm

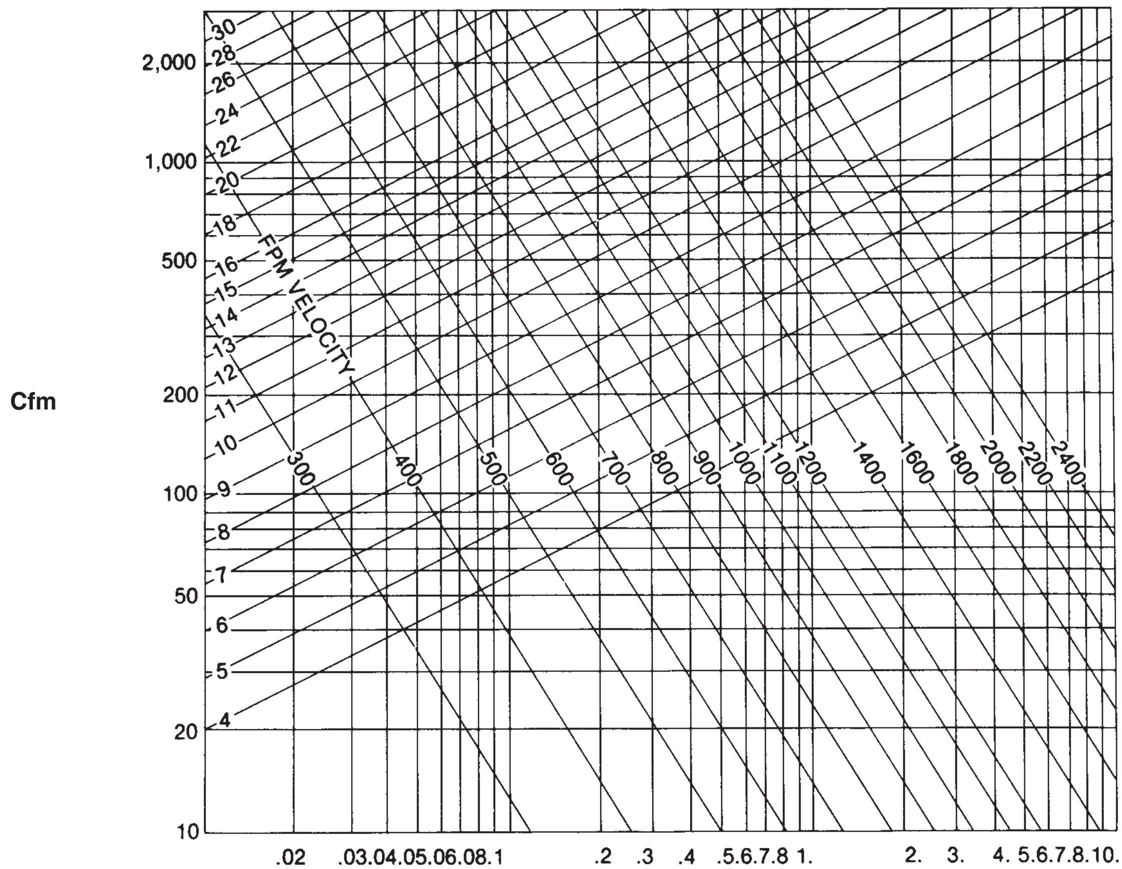


Notes:

1) Correction required for non-standard air.
2) 40 joints per100 Feet.
3) Roughness = 0.0005 Feet

Chart 3

Fiberglass Duct Board
10 Cfm to 2,000 Cfm



Friction Loss In Inches of Water per 100 Feet

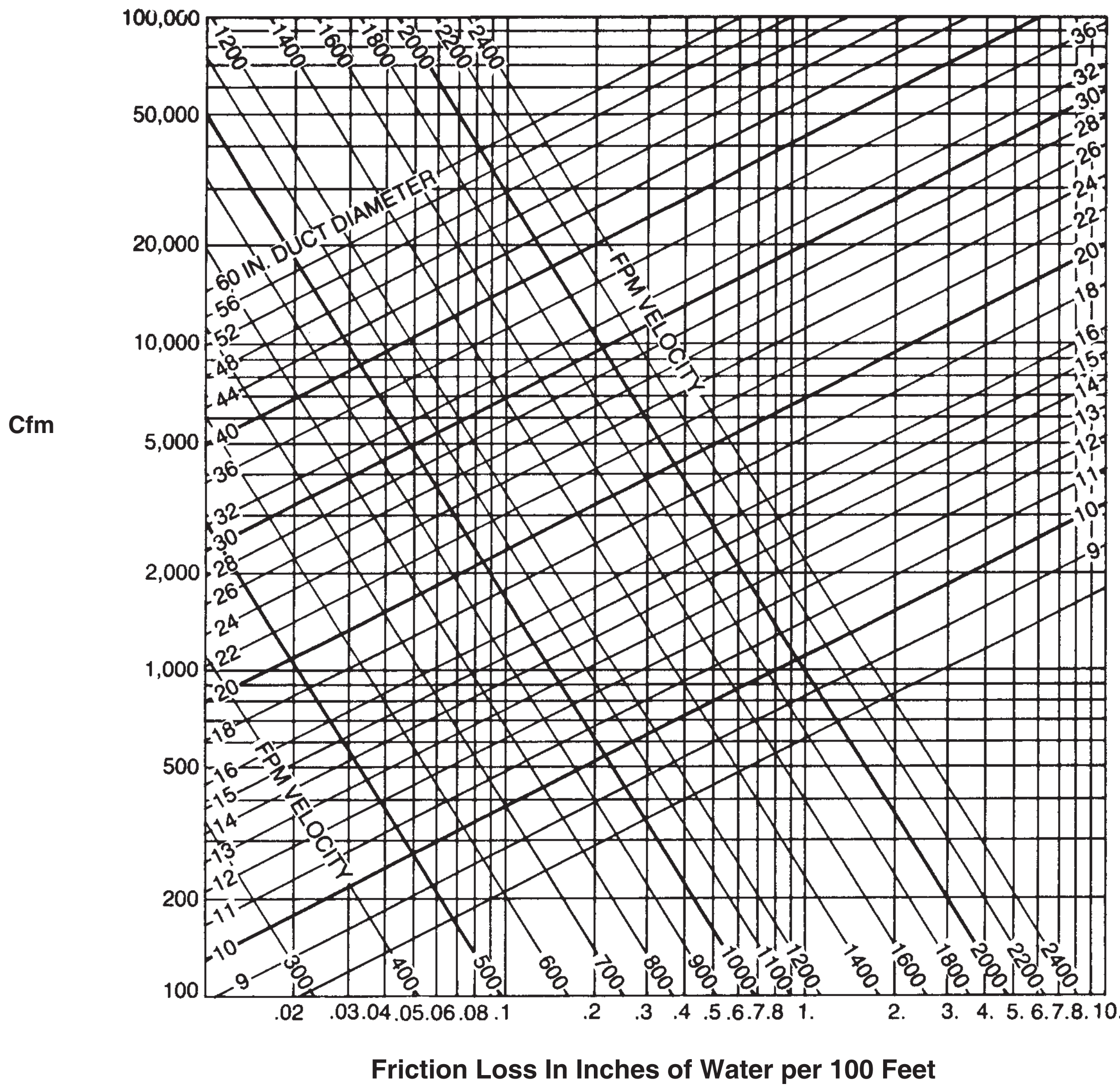
Notes:

Duct friction charts vary from product to product (depending on construction details). The friction chart or duct slide rule provided by the manufacturer of a particular board product supercedes the ***Manual D*** friction chart and the ACCA slide rule.

1) Correction required for non-standard air
2) Maximum allowable velocity 2,400 Fpm
3) Maximum allowable temperature 250°F
4) Maximum allowable pressure 2.0 IWC

Chart 4

Fiberglass Duct Board
100 Cfm to 100,000 Cfm



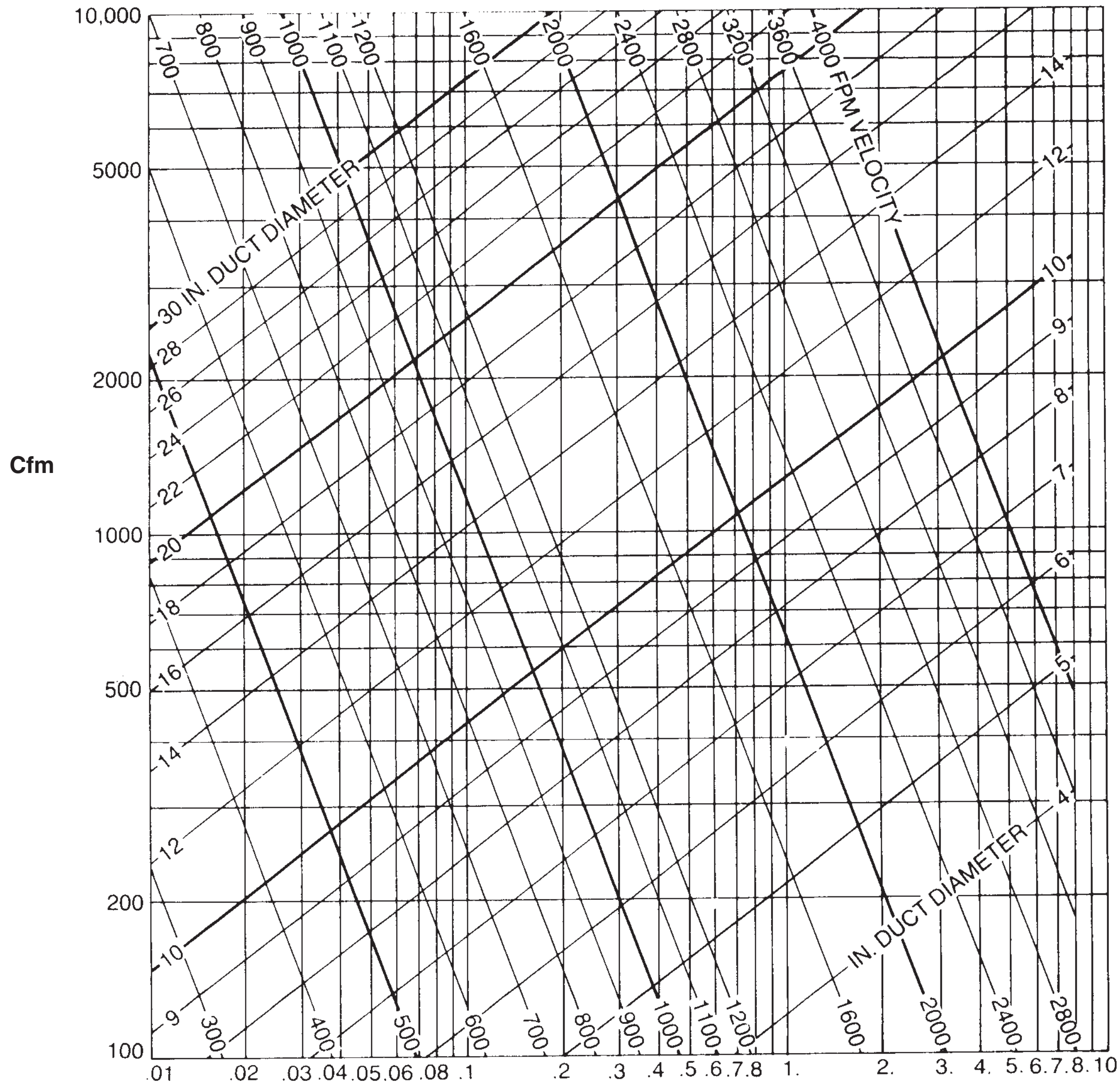
Notes:

Duct friction charts vary from product to product (depending on construction details). The friction chart or duct slide rule provided by the manufacturer of a particular board product supercedes the ***Manual D*** friction chart and the ACCA slide rule.

1) Correction required for non-standard air.
2) Maximum allowable velocity 2,400 Fpm.
3) Maximum allowable temperature 250°F.
4) Maximum allowable pressure 2.0 IWC.

Chart 5

Rigid Round Fiberglass Duct

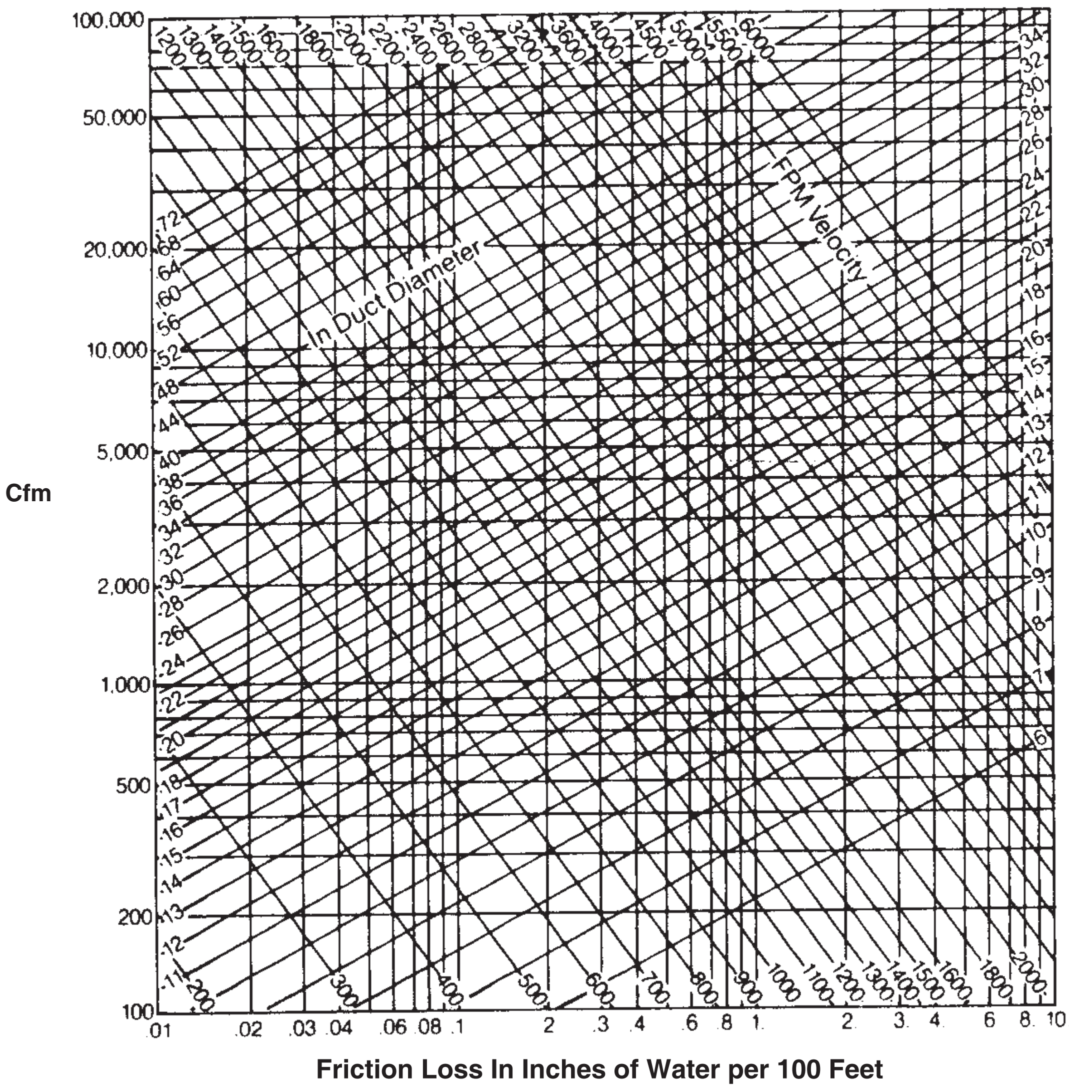


Friction Loss In Inches of Water per 100 Feet

Note:
Duct friction charts vary from product to product (depending on construction details). The friction chart or duct slide rule provided by the manufacturer of a particular board product supercedes the ***Manual D*** friction chart and the ACCA slide rule.

Chart 6

Fiberglass Duct Liner with Facing



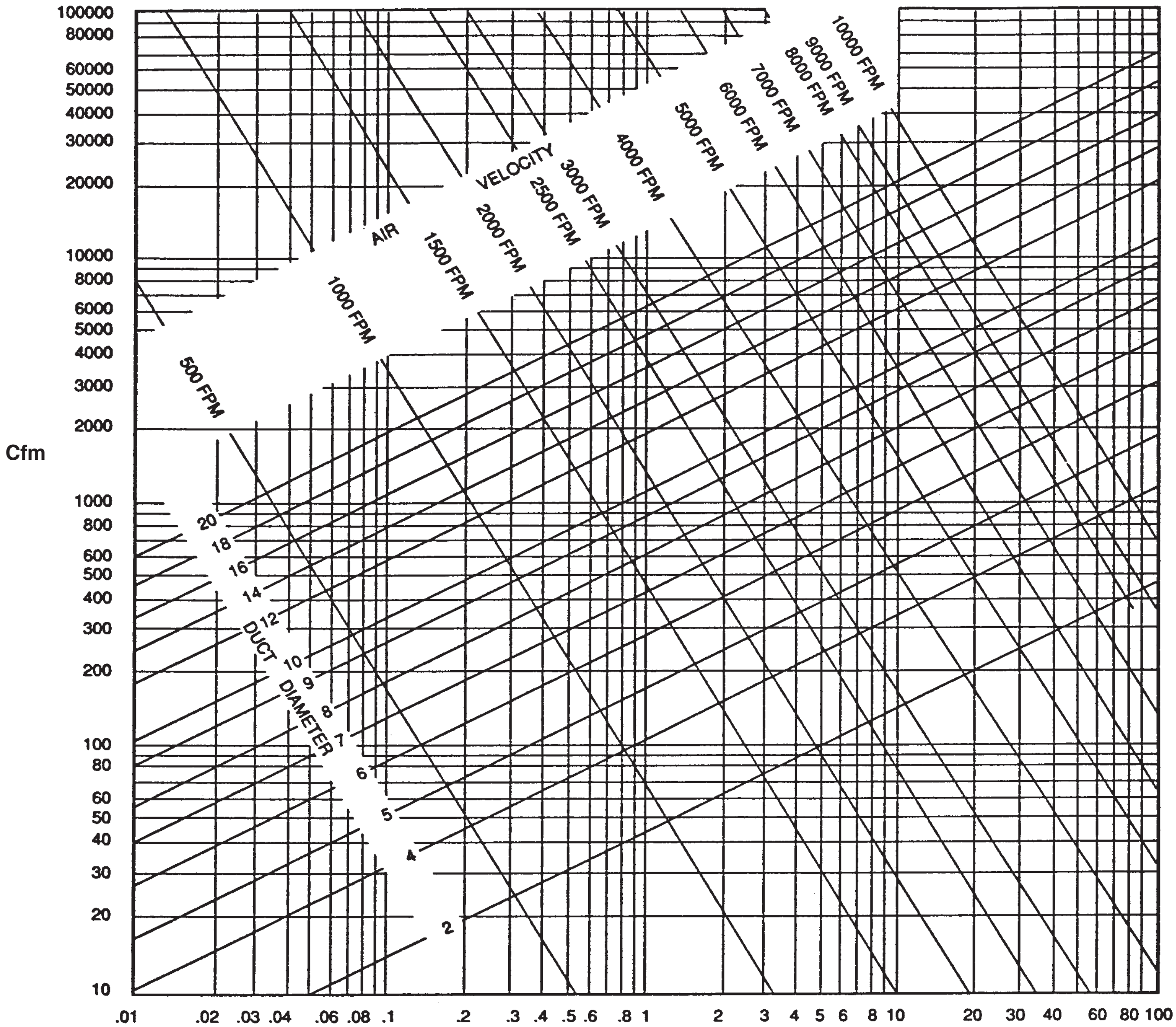
Notes:

Duct friction charts vary from product to product (depending on construction details). The friction chart or duct slide rule provided by the manufacturer of a particular duct liner product supercedes the ***Manual D*** friction chart and the ACCA slide rule.

1) Correction required for non-standard air.
2) Maximum recommended velocity 5,000 Fpm.
3) Maximum allowable temperature 250°F.

Chart 7

Flexible Wire Helix Core Ducts



Friction Loss In Inches of Water per 100 Feet

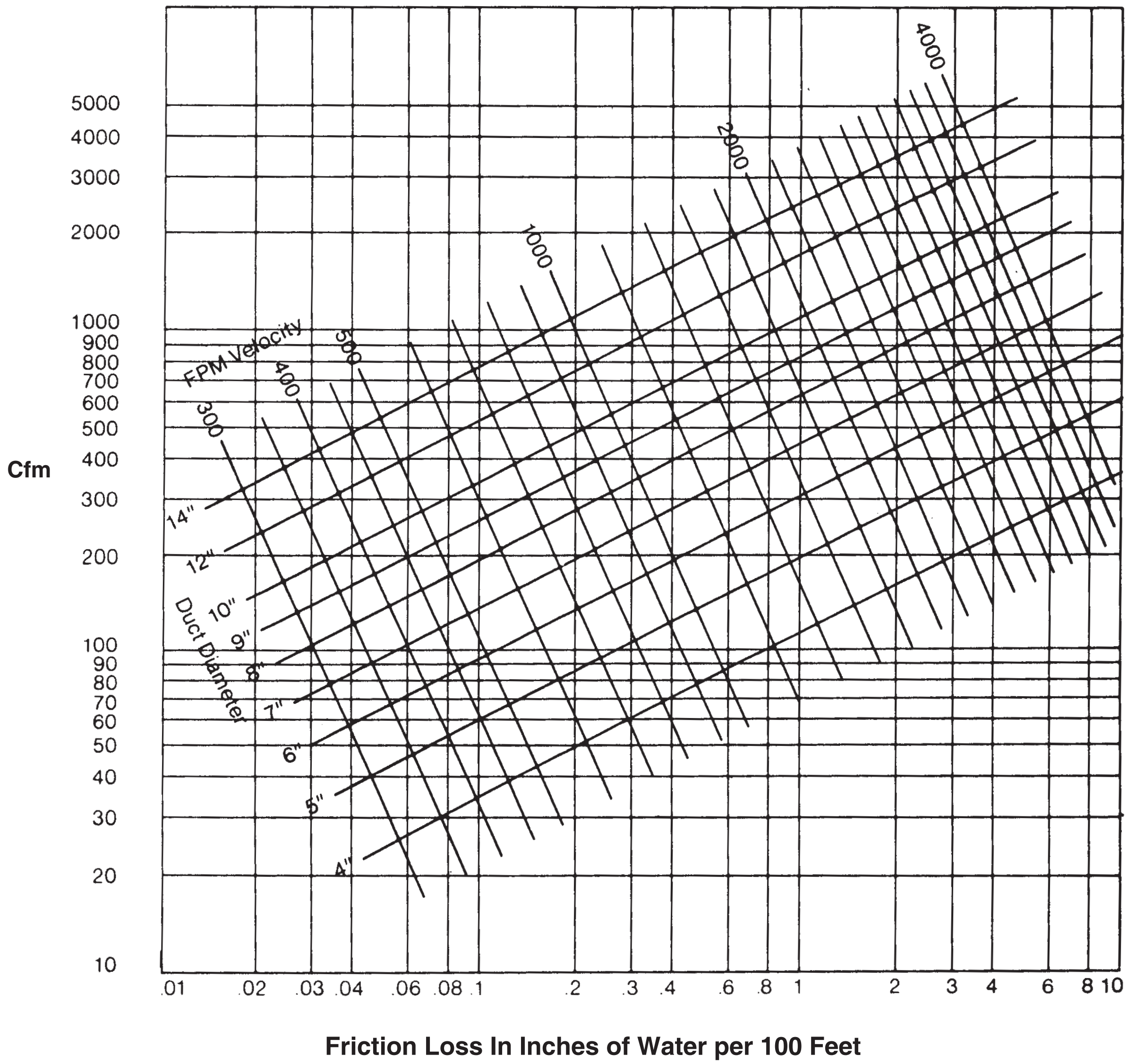
Notes:

Duct friction charts vary from product to product (depending on construction details). The friction chart or duct slide rule provided by the manufacturer of a helix product supercedes the ***Manual D*** friction chart and the ACCA slide rule.

1) Correction required for non-standard air.
2) Maximum velocity 2,400 Fpm.
3) Maximum temperature 250°F.
4) Maximum positive pressure
   up to 12" I.D. - 2.0 IWC
   over 12" I.D. - 1.0 IWC
5) Maximum negative pressure 1.0 IWC.

Chart 8

Flexible Metal Ducts



Notes:

Duct friction charts vary from product to product (depending on construction details). The friction chart or duct slide rule provided by the manufacturer of a particular board flexible metal supercedes the ***Manual D*** friction chart and the ACCA slide rule.

1) Correction required for non-standard air.
2) Maximum velocity 8,000 Fpm.
3) Maximum temperature 250°F (insulated), 400°F (not insulated).
4) Maximum positive or negative pressure (IWC) for standard non-perforated, non acoustical duct = 8.0 IWC

Chart 9

Circular Equivalents of Rectangular Ducts
Based on Equal Friction Rate

Side (In)	Side Dimension (Inches)																
	4	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
3	3.8	4.2	4.6	4.9	5.2	5.5	5.7	6.0	6.2	6.4	6.6	6.8	7.0	7.2	7.3	7.5	7.7
4	4.4	4.9	5.3	5.7	6.1	6.4	6.7	7.0	7.3	7.6	7.8	8.0	8.3	8.5	8.7	8.9	9.1
5	4.9	5.5	6.0	6.4	6.9	7.3	7.6	8.0	8.3	8.6	8.9	9.1	9.4	9.6	9.9	10.1	10.3
6	5.3	6.0	6.6	7.1	7.6	8.0	8.4	8.8	9.1	9.5	9.8	10.1	10.4	10.7	11.0	11.2	11.5
7	5.7	6.4	7.1	7.7	8.2	8.7	9.1	9.5	9.9	10.3	10.7	11.0	11.3	11.6	11.9	12.2	12.5
8	6.1	6.9	7.6	8.2	8.7	9.3	9.8	10.2	10.7	11.1	11.5	11.8	12.2	12.5	12.9	13.2	13.5
9	6.4	7.3	8.0	8.7	9.3	9.8	10.4	10.9	11.3	11.8	12.2	12.6	13.0	13.4	13.7	14.1	14.4
10	6.7	7.6	8.4	9.1	9.8	10.4	10.9	11.5	12.0	12.4	12.9	13.3	13.7	14.1	14.5	14.9	15.2
11	7.0	8.0	8.8	9.5	10.2	10.9	11.5	12.0	12.6	13.1	13.5	14.0	14.4	14.9	15.3	15.7	16.0
12	7.3	8.3	9.1	9.9	10.7	11.3	12.0	12.6	13.1	13.7	14.2	14.6	15.1	15.6	16.0	16.4	16.8
13	7.6	8.6	9.5	10.3	11.1	11.8	12.4	13.1	13.7	14.2	14.7	15.3	15.7	16.2	16.7	17.1	17.5
14	7.8	8.9	9.8	10.7	11.5	12.2	12.9	13.5	14.2	14.7	15.3	15.8	16.4	16.8	17.3	17.8	18.2
15	8.0	9.1	10.1	11.0	11.8	12.6	13.3	14.0	14.6	15.3	15.8	16.4	16.9	17.4	17.9	18.4	18.9
16	8.3	9.4	10.4	11.3	12.2	13.0	13.7	14.4	15.1	15.7	16.4	16.9	17.5	18.0	18.5	19.0	19.5
17	8.5	9.6	10.7	11.6	12.5	13.4	14.1	14.9	15.6	16.2	16.8	17.4	18.0	18.6	19.1	19.6	20.1
18	8.7	9.9	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	20.2	20.7
19	8.9	10.1	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	21.3
20	9.1	10.3	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
22	9.5	10.8	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	9.8	11.2	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
26	10.1	11.5	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
28	10.4	11.9	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
30	10.7	12.2	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
32	11.0	12.6	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
34	11.3	12.9	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
36	11.5	13.2	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
38	11.8	13.5	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
40	12.0	13.8	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
42	12.3	14.0	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
44	12.5	14.3	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
46	12.7	14.6	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
48	12.9	14.8	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
50	13.2	15.1	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

Round Diameter (In) = $1.3 \times ((a \times b)^{0.625}) / (a + b)^{0.25})$

a = Length of side one (In); b = Length of side two (In)

Chart 9 Continued

Circular Equivalents of Rectangular Ducts Based on Equal Friction Rate

Side (In)	Side Dimension (Inches)																
	22	24	26	28	30	32	34	36	38	40	42	44	46	48	50	52	54
3	8.0	8.3	8.5	8.8	9.0	9.3	9.5	9.7	9.9	10.1	10.3	10.5	10.7	10.9	11.0	11.2	11.4
4	9.5	9.8	10.1	10.4	10.7	11.0	11.3	11.5	11.8	12.0	12.3	12.5	12.7	12.9	13.2	13.4	13.6
5	10.8	11.2	11.5	11.9	12.2	12.6	12.9	13.2	13.5	13.8	14.0	14.3	14.6	14.8	15.1	15.3	15.5
6	12.0	12.4	12.8	13.2	13.6	14.0	14.4	14.7	15.0	15.3	15.6	15.9	16.2	16.5	16.8	17.1	17.3
7	13.0	13.5	14.0	14.5	14.9	15.3	15.7	16.1	16.5	16.8	17.1	17.5	17.8	18.1	18.4	18.7	19.0
8	14.1	14.6	15.1	15.6	16.1	16.5	17.0	17.4	17.8	18.2	18.5	18.9	19.3	19.6	19.9	20.2	20.6
9	15.0	15.6	16.2	16.7	17.2	17.7	18.2	18.6	19.0	19.5	19.9	20.3	20.6	21.0	21.4	21.7	22.0
10	15.9	16.5	17.1	17.7	18.3	18.8	19.3	19.8	20.2	20.7	21.1	21.5	21.9	22.3	22.7	23.1	23.5
11	16.8	17.4	18.1	18.7	19.3	19.8	20.4	20.9	21.4	21.8	22.3	22.7	23.2	23.6	24.0	24.4	24.8
12	17.6	18.3	19.0	19.6	20.2	20.8	21.4	21.9	22.4	22.9	23.4	23.9	24.4	24.8	25.2	25.7	26.1
13	18.3	19.1	19.8	20.5	21.1	21.8	22.4	22.9	23.5	24.0	24.5	25.0	25.5	26.0	26.4	26.9	27.3
14	19.1	19.9	20.6	21.3	22.0	22.7	23.3	23.9	24.5	25.0	25.6	26.1	26.6	27.1	27.6	28.0	28.5
15	19.8	20.6	21.4	22.1	22.9	23.5	24.2	24.8	25.4	26.0	26.6	27.1	27.7	28.2	28.7	29.2	29.7
16	20.4	21.3	22.1	22.9	23.7	24.4	25.1	25.7	26.4	27.0	27.6	28.1	28.7	29.2	29.8	30.3	30.8
17	21.1	22.0	22.9	23.7	24.4	25.2	25.9	26.6	27.2	27.9	28.5	29.1	29.7	30.2	30.8	31.3	31.8
18	21.7	22.7	23.5	24.4	25.2	26.0	26.7	27.4	28.1	28.8	29.4	30.0	30.6	31.2	31.8	32.3	32.9
19	22.3	23.3	24.2	25.1	25.9	26.7	27.5	28.2	28.9	29.6	30.3	30.9	31.6	32.2	32.8	33.3	33.9
20	22.9	23.9	24.9	25.8	26.6	27.5	28.3	29.0	29.8	30.5	31.2	31.8	32.5	33.1	33.7	34.3	34.9
22	24.0	25.1	26.1	27.1	28.0	28.9	29.7	30.5	31.3	32.1	32.8	33.5	34.2	34.9	35.5	36.2	36.8
24	25.1	26.2	27.3	28.3	29.3	30.2	31.1	32.0	32.8	33.6	34.4	35.1	35.9	36.6	37.2	37.9	38.6
26	26.1	27.3	28.4	29.5	30.5	31.5	32.4	33.3	34.2	35.1	35.9	36.7	37.4	38.2	38.9	39.6	40.3
28	27.1	28.3	29.5	30.6	31.7	32.7	33.7	34.6	35.6	36.4	37.3	38.1	38.9	39.7	40.5	41.2	41.9
30	28.0	29.3	30.5	31.7	32.8	33.9	34.9	35.9	36.8	37.8	38.7	39.5	40.4	41.2	42.0	42.8	43.5
32	28.9	30.2	31.5	32.7	33.9	35.0	36.1	37.1	38.1	39.0	40.0	40.9	41.8	42.6	43.5	44.3	45.1
34	29.7	31.1	32.4	33.7	34.9	36.1	37.2	38.2	39.3	40.3	41.3	42.2	43.1	44.0	44.9	45.7	46.5
36	30.5	32.0	33.3	34.6	35.9	37.1	38.2	39.4	40.4	41.5	42.5	43.5	44.4	45.3	46.2	47.1	48.0
38	31.3	32.8	34.2	35.6	36.8	38.1	39.3	40.4	41.5	42.6	43.7	44.7	45.7	46.6	47.5	48.4	49.3
40	32.1	33.6	35.1	36.4	37.8	39.0	40.3	41.5	42.6	43.7	44.8	45.8	46.9	47.9	48.8	49.7	50.7
42	32.8	34.4	35.9	37.3	38.7	40.0	41.3	42.5	43.7	44.8	45.9	47.0	48.0	49.1	50.0	51.0	52.0
44	33.5	35.1	36.7	38.1	39.5	40.9	42.2	43.5	44.7	45.8	47.0	48.1	49.2	50.2	51.2	52.2	53.2
46	34.2	35.9	37.4	38.9	40.4	41.8	43.1	44.4	45.7	46.9	48.0	49.2	50.3	51.4	52.4	53.4	54.4
48	34.9	36.6	38.2	39.7	41.2	42.6	44.0	45.3	46.6	47.9	49.1	50.2	51.4	52.5	53.6	54.6	55.6
50	35.5	37.2	38.9	40.5	42.0	43.5	44.9	46.2	47.5	48.8	50.0	51.2	52.4	53.6	54.7	55.7	56.8

Round Diameter (In) = 1.3 x ((a x b)$^{0.625}$) / (a + b)$^{0.25}$)
a = Length of side one (In); b = Length of side two(In)

Chart 10

Circular Equivalents of Oval Ducts Based on Equal Friction Rate

Minor Dia. (In)	Major Diameter (Inches)																
	4	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**
3	3.6	4.0	4.4	4.8	5.1	5.4	5.7	5.9	6.1	6.4	6.6	6.8	7.0	7.1	7.3	7.5	7.6
4	~	4.6	5.1	5.5	5.9	6.3	6.6	6.9	7.2	7.5	7.7	8.0	8.2	8.4	8.6	8.8	9.0
5	~	~	5.6	6.1	6.6	7.0	7.4	7.8	8.1	8.4	8.7	9.0	9.3	9.5	9.8	10.0	10.2
6	~	~	~	6.6	7.1	7.6	8.1	8.5	8.9	9.2	9.6	9.9	10.2	10.5	10.8	11.1	11.3
7	~	~	~	~	7.6	8.2	8.7	9.1	9.6	10.0	10.4	10.7	11.1	11.4	11.7	12.0	12.3
8	~	~	~	~	~	8.6	9.2	9.7	10.2	10.6	11.0	11.4	11.8	12.2	12.5	12.9	13.2
9	~	~	~	~	~	~	9.6	10.2	10.7	11.2	11.7	12.1	12.5	12.9	13.3	13.7	14.0
10	~	~	~	~	~	~	~	10.6	11.2	11.7	12.2	12.7	13.2	13.6	14.0	14.4	14.8
11	~	~	~	~	~	~	~	~	11.6	12.2	12.7	13.3	13.7	14.2	14.7	15.1	15.5
12	~	~	~	~	~	~	~	~	~	12.6	13.2	13.8	14.3	14.8	15.3	15.7	16.1
13	~	~	~	~	~	~	~	~	~	~	13.6	14.2	14.8	15.3	15.8	16.3	16.8
14	~	~	~	~	~	~	~	~	~	~	~	14.6	15.2	15.8	16.3	16.8	17.3
15	~	~	~	~	~	~	~	~	~	~	~	~	15.6	16.2	16.8	17.3	17.9
16	~	~	~	~	~	~	~	~	~	~	~	~	~	16.6	17.2	17.8	18.3
17	~	~	~	~	~	~	~	~	~	~	~	~	~	~	17.6	18.2	18.8
18	~	~	~	~	~	~	~	~	~	~	~	~	~	~	~	18.6	19.2

Minor Dia. (In)	Major Diameter (Inches)																
	22	**24**	**26**	**28**	**30**	**32**	**34**	**36**	**38**	**40**	**42**	**44**	**46**	**48**	**50**	**52**	**54**
20	21.2	22.4	23.5	24.5	25.4	26.3	27.2	28.0	28.8	29.6	30.3	31.0	31.7	32.4	33.0	33.6	34.2
22	~	23.3	24.4	25.5	26.5	27.5	28.4	29.3	30.2	31.0	31.8	32.5	33.3	34.0	34.7	35.3	36.0
24	~	~	25.3	26.4	27.5	28.6	29.6	30.5	31.4	32.3	33.1	33.9	34.7	35.5	36.2	36.9	37.6
26	~	~	~	27.3	28.4	29.5	30.6	31.6	32.6	33.5	34.4	35.3	36.1	36.9	37.7	38.4	39.2
28	~	~	~	~	29.3	30.4	31.6	32.6	33.7	34.6	35.6	36.5	37.4	38.2	39.1	39.9	40.6
30	~	~	~	~	~	31.3	32.5	33.6	34.7	35.7	36.7	37.7	38.6	39.5	40.4	41.2	42.0
32	~	~	~	~	~	~	33.3	34.5	35.6	36.7	37.7	38.8	39.7	40.7	41.6	42.5	43.3
34	~	~	~	~	~	~	~	35.3	36.5	37.6	38.7	39.8	40.8	41.8	42.7	43.7	44.6
36	~	~	~	~	~	~	~	~	37.3	38.5	39.6	40.7	41.8	42.8	43.8	44.8	45.8
38	~	~	~	~	~	~	~	~	~	39.3	40.5	41.6	42.8	43.8	44.9	45.9	46.9
40	~	~	~	~	~	~	~	~	~	~	41.3	42.5	43.7	44.8	45.9	46.9	47.9
42	~	~	~	~	~	~	~	~	~	~	~	43.3	44.5	45.7	46.8	47.9	48.9
48	~	~	~	~	~	~	~	~	~	~	~	~	~	~	49.3	50.5	51.7

Round Diameter (In) = 1.55 x (A $^{0.625}$/ P $^{0.25}$)
A (SqIn) = 3.14 x b2 / 4 + b x (a - b) and P (In) = 3.14 x b + 2 x (a - b)
a = Major diameter (In); b = Minor diameter (In)

Appendix 3

Fitting Equivalent Lengths

This appendix provides information about the air flow resistance produced by various types of supply and return fittings. For residential duct systems, this resistance is quantified by assigning an equivalent length value to each type of fitting.

A3-1 Equivalent Length

Many practitioners assume that the equivalent length of a fitting is an invariable and unconditional description of the aerodynamic efficiency of the fitting. This is not true! The equivalent length of a fitting is in fact a conditional pressure loss value that has been converted to a "length" by using an equation that contains an arbitrary friction rate value.

Some of the conditions that affect the aerodynamic performance of the fitting may include the geometry, the entering and leaving flow rate(s), or the entering and leaving velocities. For example, Figure A3-1 shows that the fitting loss coefficient (C) for a simple elbow depends on the geometry of the fitting and the Cfm that flows through the fitting (Reynolds number). For 2,000 or more Cfm, the loss coefficient varies from 0.14 to 1.50 when the Reynolds number correction is 1.0.

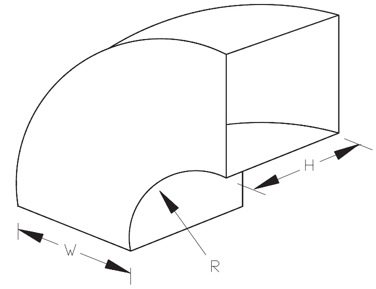
The effective loss coefficient is converted to a pressure loss by multiplying it by the velocity pressure (VP) produced by the flow. For a residential system, velocity pressure may vary from less than 0.01 IWC (at 400 Fpm) to more than 0.05 IWC (at 900 Fpm). This means that the pressure drop (PD) across the fitting could vary from 0.0014 IWC to 0.076 IWC, as demonstrated here:

Air velocity (Fpm) = V
VP (IWC) = (V / 4,007)2
PD (IWC) = C x N x VP

Cfm = 1,200 and R/W = 0.5; so N = 1.0
PD (0.14; 400) = 0.14 x 1.0 x (400 / 4,007)2 = 0.0014 IWC
PD (1.5; 900) = 1.5 x 1.0 x (900 / 4,007)2 = 0.076 IWC

Note that fitting loss is in pressure units (IWC), which is exactly what is required. (The pressure produced by the blower must be equal to, or greater than, the pressure loss for the straight runs and fittings in a duct run).

Fitting equivalent length values are a convenience. This way, the airflow resistance of a duct run is represented by its effective length (straight run length plus the equivalent length of the fittings). Once effective length is calculated, a friction rate (FR) value converts effective length to pressure loss, or vice versa, per this equation:



Fitting Loss Coefficient (C)											
R/W	H/W										
	0.25	0.50	0.75	1.00	1.50	2.00	3.00	4.00	5.00	6.00	8.00
0.5	1.50	1.40	1.30	1.20	1.10	1.00	1.00	1.10	1.10	1.20	1.20
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

Reynolds Number Correction (N)		
Cfm	**R/W <0.75**	**R/W > 0.75**
50 to 200	1.15	1.60
200 to 400	1.10	1.50
400 to 800	1.05	1.35
800 to 1,000	1.03	1.30
1,000 to 1,500	1.00	1.15
1,500 to 2,000	1.00	1.05
Above 2,000	1.00	1.00
Adjusted loss coefficient = C x N		

Figure A3-1

Length (Ft) = 100 x Pressure drop / FR

If just one fitting is involved, this equation is used to calculate its equivalent length. For example, the equivalent length of the fitting that is described above could vary from less than 1 foot to more than 94 feet, depending on the loss coefficient value, the reference velocity, and the friction rate value. Some equivalent length values (rounded feet) for this fitting are provided here for FR= 0.08 (IWC/100Ft), and for FR = 0.15 (IWC/100 Ft).:

100 x 0.0014 / 0.08 = 2
100 x 0.0014 / 0.15 = 1
100 x 0.076 / 0.08 = 95
100 x 0.076 / 0.15 = 51

Since equivalent length (EL) values are conditional, they should be accompanied by a note that lists the reference velocity (V_r) and the reference friction rate value (FR_r) that were used to generate the equivalent length information.

A3-2 Default Equivalent Length Values

In this appendix, the equivalent length values for supply-side fittings are for 900 Fpm air velocity and for a 0.08 IWC/100 Ft friction rate. The equivalent length values for return-side fittings are for 700 Fpm air velocity and for a 0.08 IWC/100 Ft friction rate. The equivalent length values for fittings that are used on either side of the system are for 900 Fpm air velocity and for a 0.08 IWC/100 Ft friction rate. This information appears at the top of each page of the fitting tables.

A3-3 Equivalent Length Values for Other Scenarios

The equivalent length (EL_x) for another velocity (V_x) or another friction rate (FR_x) is calculated by this equation:

ELx = EL x (Vx / Vr)2 x (FRr / FRx)

For example, some default equivalent length values are based on 900 Fpm velocity and a 0.08 friction rate. If the default EL value is 65 feet, the equivalent length for 700 Fpm and a 0.12 friction rate is 26 Feet.

$El_x = 65 \times (700 / 900)^2 \times (0.08 / 0.12) = 26$ *Feet*

Figures A3-2 and A3-3 provide equivalent length values for various combinations of velocity and friction rate. These figures show that equivalent length values are sensitive to the friction rate and very sensitive to velocity.

- If alternative EL values are used, output from the total effective length (TEL) calculation is reduced and the design friction rate value (FR) increases.
- If alternative EL values are used, the reference air velocity for supply and return fittings must not exceed the Table A1-1 limit.

Equivalent Length Versus Velocity for 0.08 IWC/100Ft Friction Rate

Table 3 EL (Ft)	Velocity (Fpm)					
	400	500	600	700	800	900
5	1	2	2	3	4	5
10	2	3	4	6	8	10
15	3	5	7	9	12	15
20	4	6	9	12	16	20
25	5	8	11	15	20	25
30	6	9	13	18	24	30
35	7	11	16	21	28	35
40	8	12	18	24	32	40
45	9	14	20	27	36	45
50	10	15	22	30	40	50
55	11	17	24	33	43	55
60	12	19	27	36	47	60
65	13	20	29	39	51	65
70	14	22	31	42	55	70
75	15	23	33	45	59	75
80	16	25	36	48	63	80
85	17	26	38	51	67	85
90	18	28	40	54	71	90
95	19	29	42	57	75	95
100	20	31	44	60	79	100

Figure A3-2

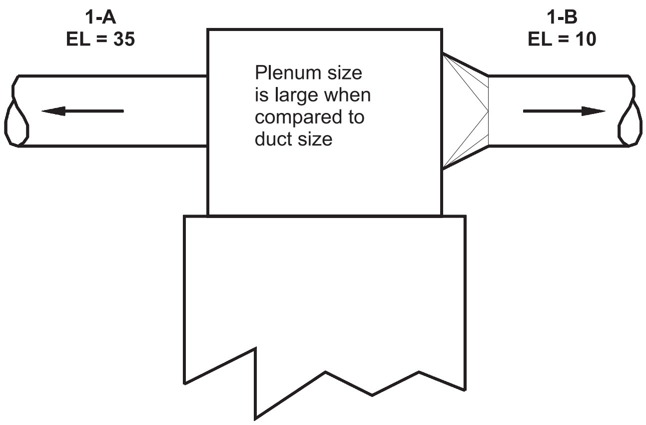
Equivalent Length Versus Velocity for 0.14 IWC/100Ft Friction Rate

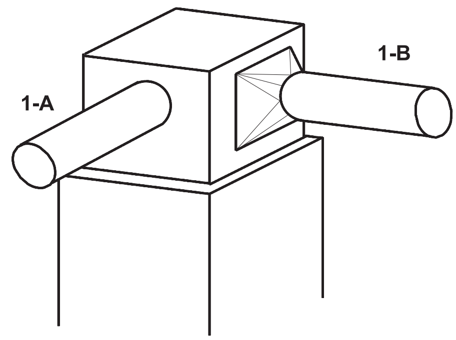
Table 3 EL (Ft)	Velocity (Fpm)					
	400	500	600	700	800	900
5	1	1	1	2	2	3
10	1	2	3	3	5	6
15	2	3	4	5	7	9
20	2	4	5	7	9	11
25	3	4	6	9	11	14
30	3	5	8	10	14	17
35	4	6	9	12	16	20
40	5	7	10	14	18	23
45	5	8	11	16	20	26
50	6	9	13	17	23	29
55	6	10	14	19	25	31
60	7	11	15	21	27	34
65	7	11	17	22	29	37
70	8	12	18	24	32	40
75	8	13	19	26	34	43
80	9	14	20	28	36	46
85	10	15	22	29	38	49
90	10	16	23	31	41	51
95	11	17	24	33	43	54
100	11	18	25	35	45	57

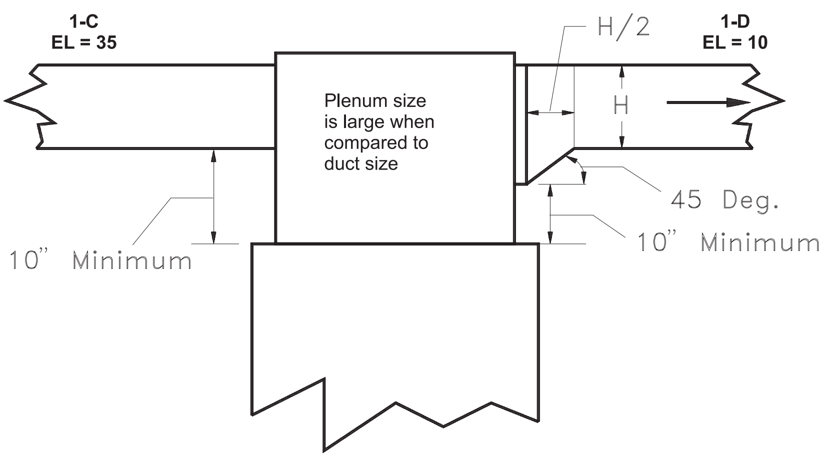
Figure A3-3

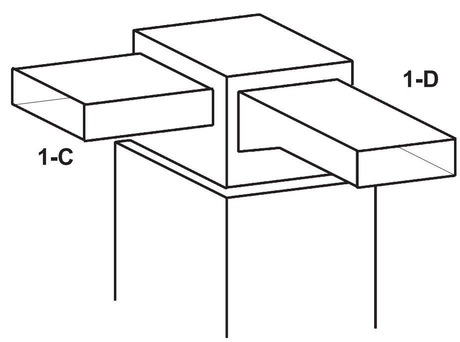
Group 1
Supply Air Fittings at the Air Handling Equipment

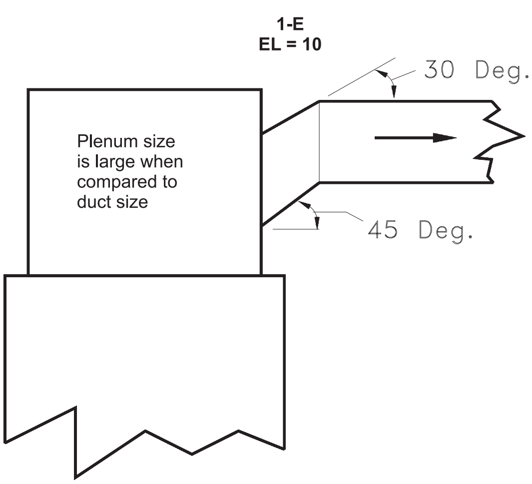
Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

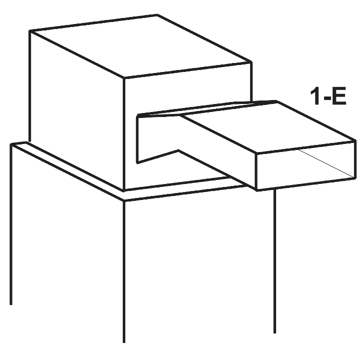






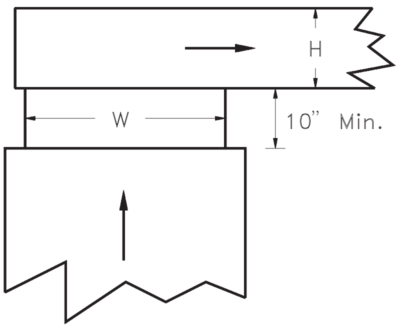


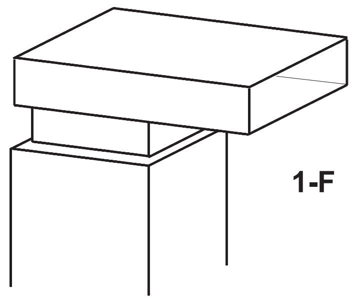




Group 1 — Continued
Supply Air Fittings at the Air Handling Equipment

Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

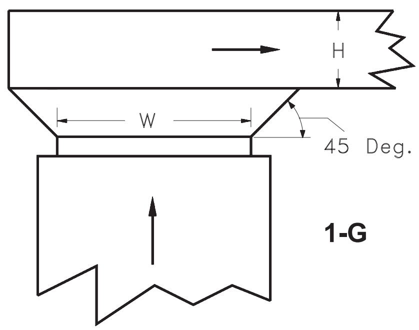




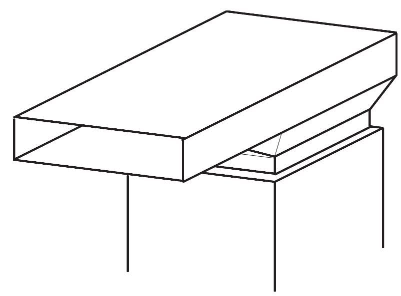
1-F

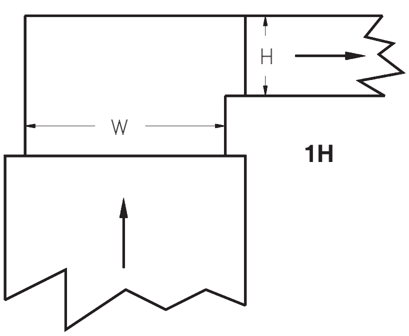
Bull Head 1F	H / W	EL
	0.50	120
	1.0	85

Tapered Head 1G	H / W	EL
	0.50	35
	1.0	25



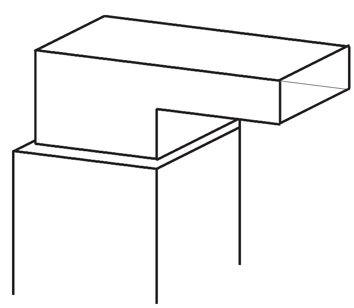
1-G

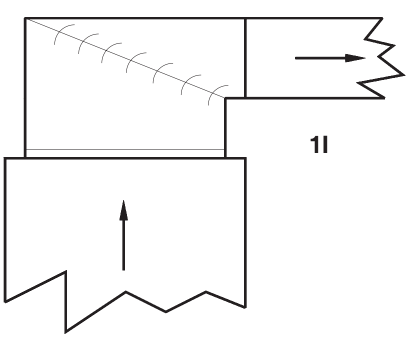




1H

No Vanes 1H	H / W	EL
	0.5	120
	1.0	85



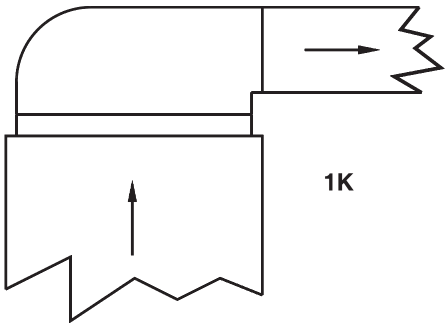


1I

With Vanes 1I	EL = 20

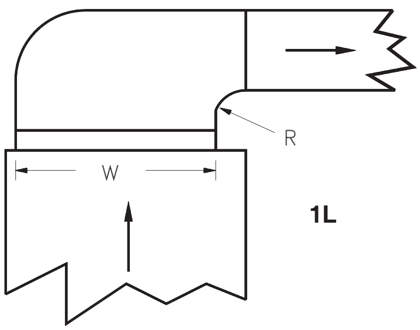
Group 1 — Continued
Supply Air Fittings at the Air Handling Equipment

Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

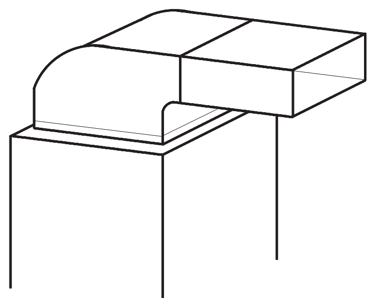


1K

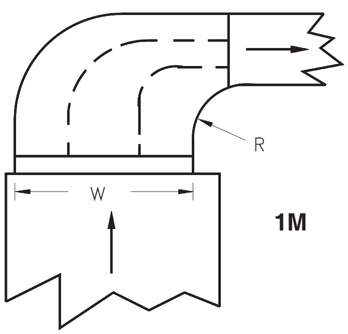
Miterd Inside Corner 1K	**EL = 85**



1L



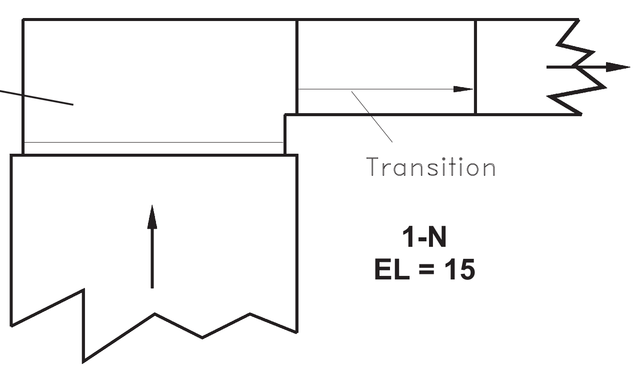
Radius Ell No Vanes 1L	**R / W**	**EL**
	0.25	40
	0.50	20
	1.0	10



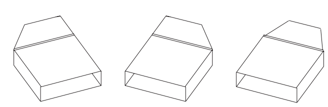
1M

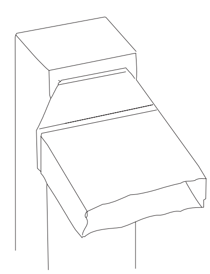
		1-Vane	**2-Vane**
Radius Ell with Vanes 1M	**R / W**	**EL**	**EL**
	0.05	30	20
	0.25	20	10
	0.50	10	10

Add the transition EL to the upstream elbow EL (or tee EL)



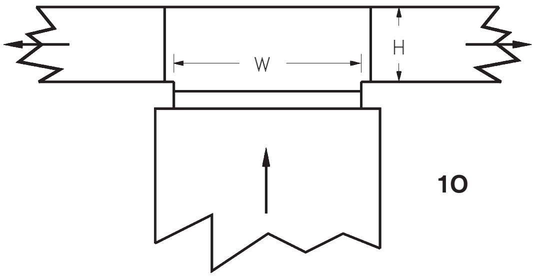
1-N
EL = 15





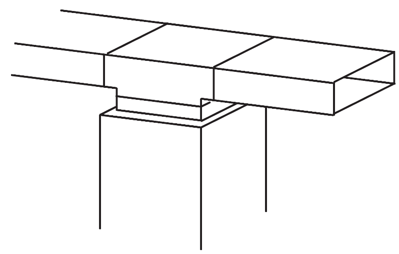
Group 1 — Continued
Supply Air Fittings at the Air Handling Equipment

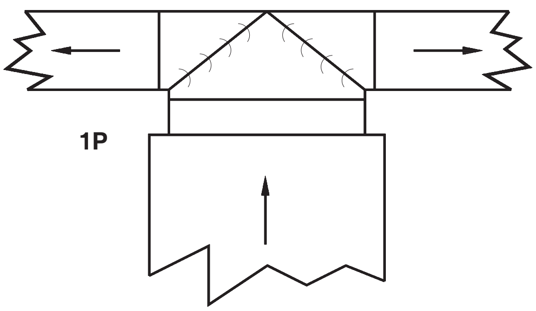
Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet



1O

Bull Head No Vanes 1O	H / W	EL
	0.50	120
	1.0	85



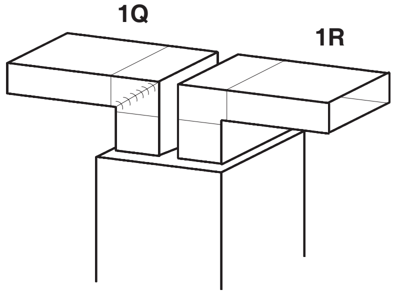


1P

Vaned Tee 1P	EL = 20

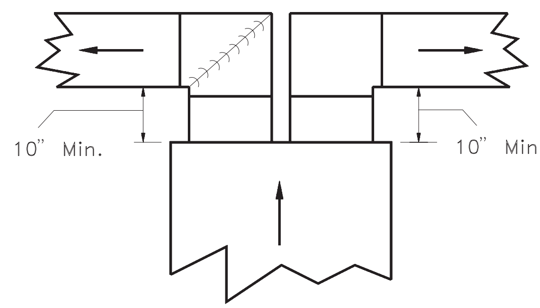
1Q

1R

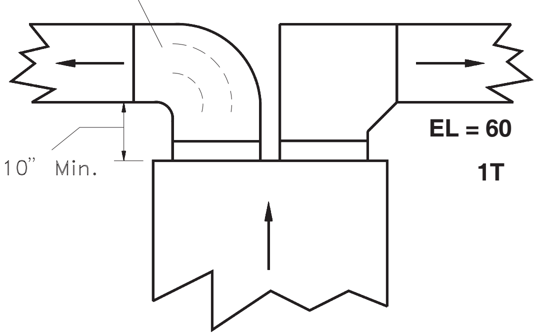


1Q
EL = 50

1R
EL = 120



Radius Elbow 1S	Vanes	EL
	0	60
	1	40
	2	30

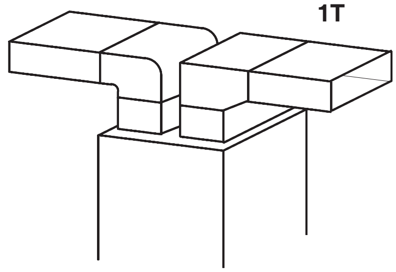


EL = 60

1T

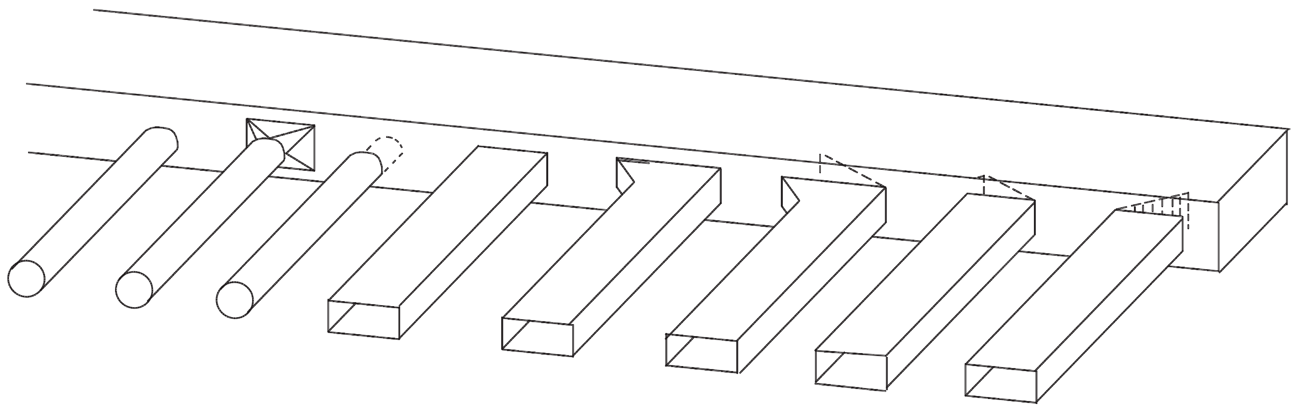
1S

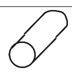
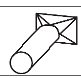
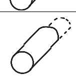
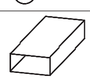
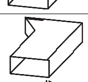
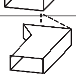
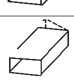
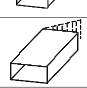
1T



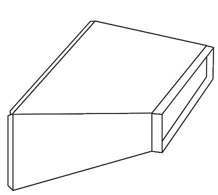
Group 2
Branch Takeoff Fittings at the Supply Trunk

Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

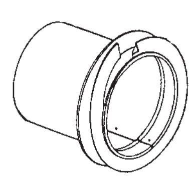


EL Values		Number of Downstream Branches to End of Trunk Duct or Number of Downstream Branches to a Trunk Reducer					
Fitting		**0**	**1**	**2**	**3**	**4**	**5 or More**
	**2A**	**35**	**45**	**55**	**65**	**70**	**80**
	**2B**	**20**	**30**	**35**	**40**	**45**	**50**
	**2C**	**65**	**65**	**65**	**65**	**70**	**80**
	**2D**	**40**	**50**	**60**	**65**	**75**	**85**
	**2E**	**25**	**30**	**35**	**40**	**45**	**50**
	**2F**	**20**	**20**	**20**	**20**	**25**	**25**
	**2G**	**65**	**65**	**65**	**70**	**80**	**90**
	**2H**	**70**	**70**	**70**	**75**	**85**	**95**

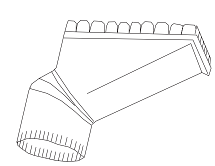
Note: If the trunk has a reducer, count down to the reducer; then begin a new count after the reducer.



Refer to Fitting 2D



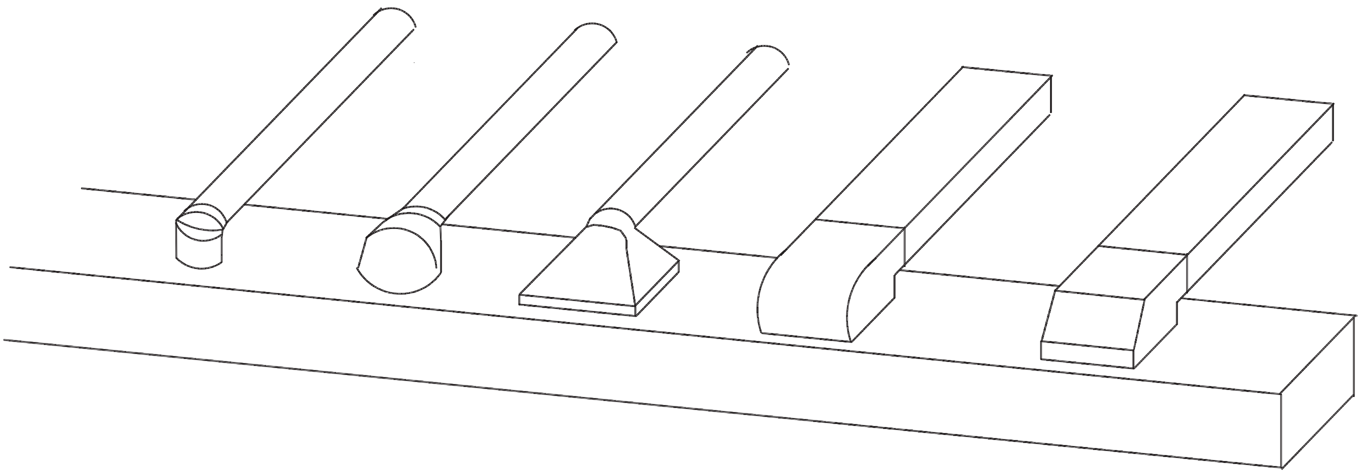
Refer to Fitting 2A or 2C

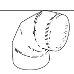
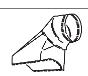
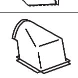
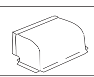
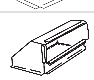


Refer to Fitting 2E

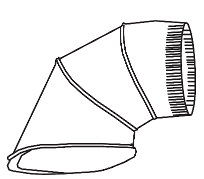
Group 2 — Continued
Branch Takeoff Fittings at the Supply Trunk

Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

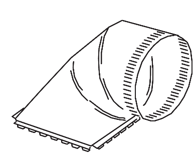


EL Values		Number of Downstream Branches to End of Trunk Duct or Number of Downstream Branches to a Trunk Reducer					
Fitting		0	1	2	3	4	5 or More
	2I	65	75	85	95	100	110
	2J	50	60	65	70	75	80
	2K	50	60	65	70	75	80
	2L	70	80	90	95	105	115
	2M	70	80	90	95	105	115

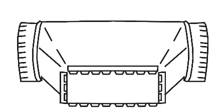
Note: If the trunk has a reducer, count down to the reducer; then begin a new count after the reducer.



Refer to Fitting 2J



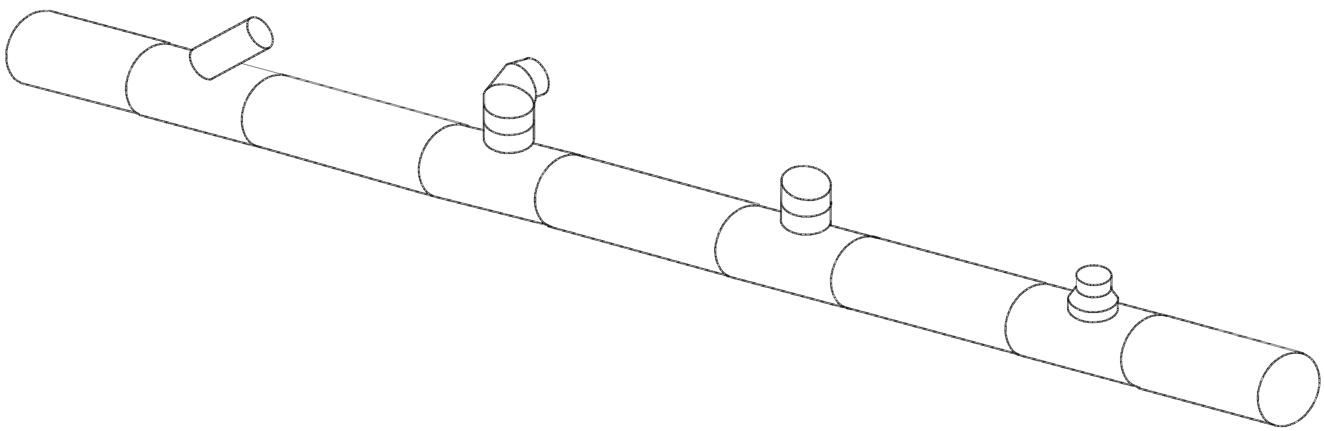
Refer to Fitting 2J

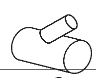
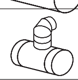
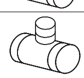
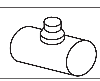


Refer to Fitting 2I

Group 2 — Continued
Branch Takeoff Fittings at the Supply Trunk

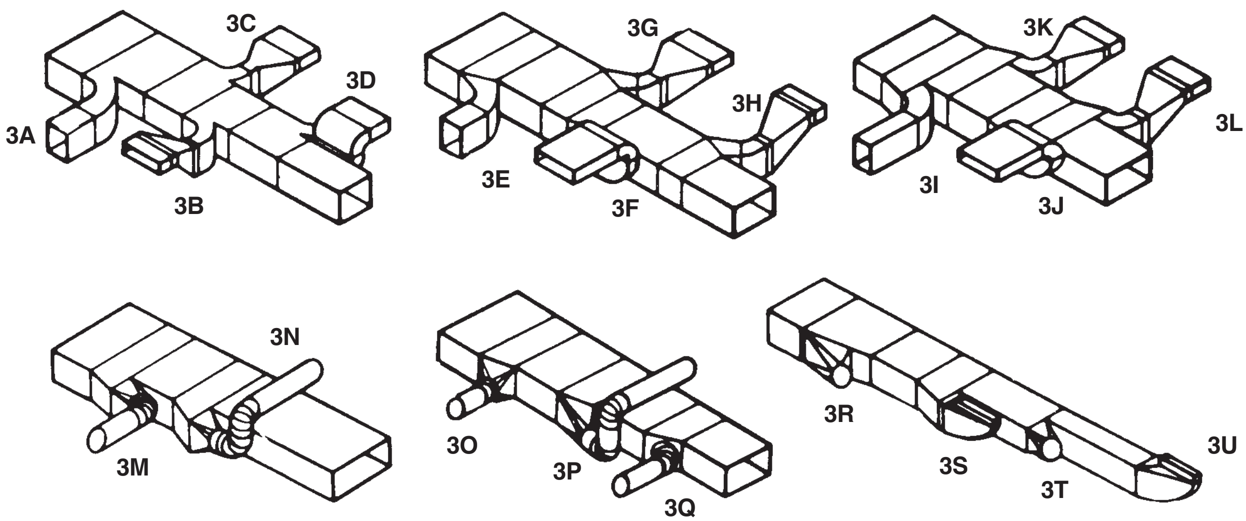
Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet



EL Values		Number of Downstream Branches to End of Trunk Duct or Number of Downstream Branches to a Trunk Reducer					
Fitting		0	1	2	3	4	5 or More
	2N	35	35	40	40	40	40
	2O	55	65	75	85	90	100
	2P	50	55	60	65	70	75
	2Q	10	10	15	20	20	25
Note: If the trunk has a reducer, count down to the reducer; then begin a new count after the reducer.							

Group 3
Reducing Trunk Takeoff Fittings

Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet



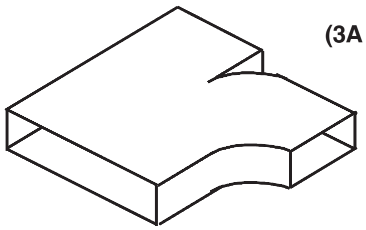
Fitting ID	EL	Description of Assembly
3A and 3I	15	Full radius takeoff
3B and 3L	30	Full radius takeoff plus offset transition
3C and 3K	20	Full radius takeoff plus straight transition
3D and 3J	35	Radius takeoff elbow (see 3S) plus easy-bend elbow
	55	Tight radius takeoff elbow (see 3S) plus easy-bend elbow
	110	Miterd inside corner takeoff elbow (see 3S) plus easy-bend elbow
3E	30	Transition wall takeoff
3F	3D + 15	Transition wall takeoff elbow (radius, tight radius or Miterd corner) plus easy-bend elbow
3G	35	Transition wall takeoff plus straight-aspect transition
3H	35	Transition wall takeoff plus offset-aspect transition
3M	25	In line eased takeoff fitting (see 3T) plus one elbow
3N	40	In line eased takeoff fitting (see 3T) plus two elbows
3O and 3R	20	Transition wall eased takeoff fitting (see note)
3P	50	Transition wall eased takeoff fitting plus two elbows (see note)
3Q	35	Transition wall eased takeoff fitting plus one elbow (see note)
3S and 3U	15	Full radius takeoff elbow
	35	Tight inside radius takeoff elbow
	90	Miterd inside corner takeoff elbow
3T	10	In line eased takeoff fitting

Note: Add 15 feet to the equivalent length if a round sleeve is simply butted to the transition wall.

Group 3 — Continued
Reducing Trunk Takeoff Fittings

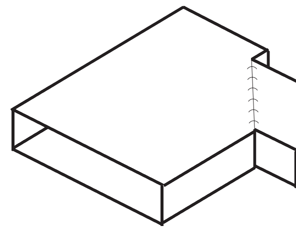
Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC. per 100 Feet

(3A or 3I)

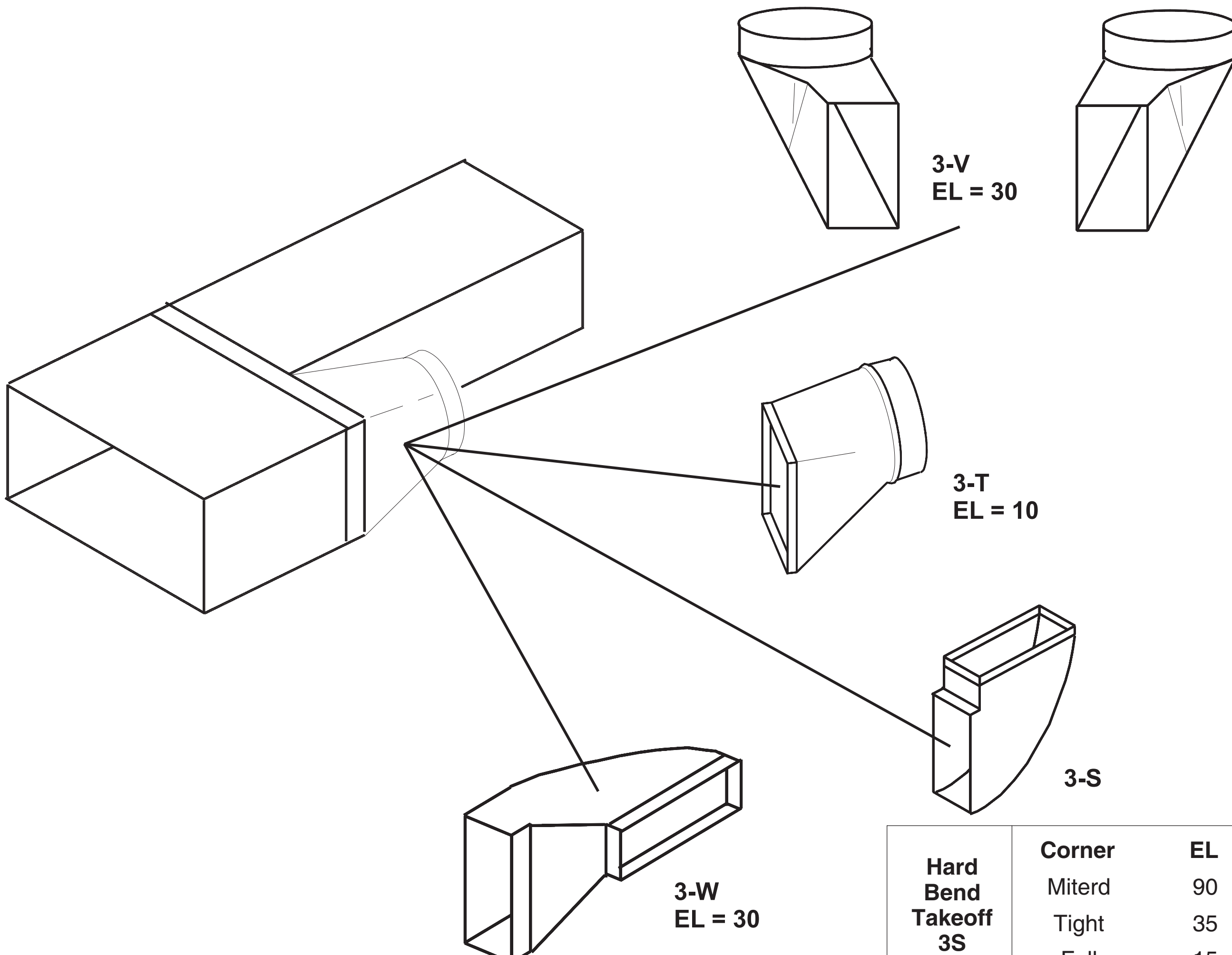


Full Radius Takeoff 3A or 3I	EL = 15

3U



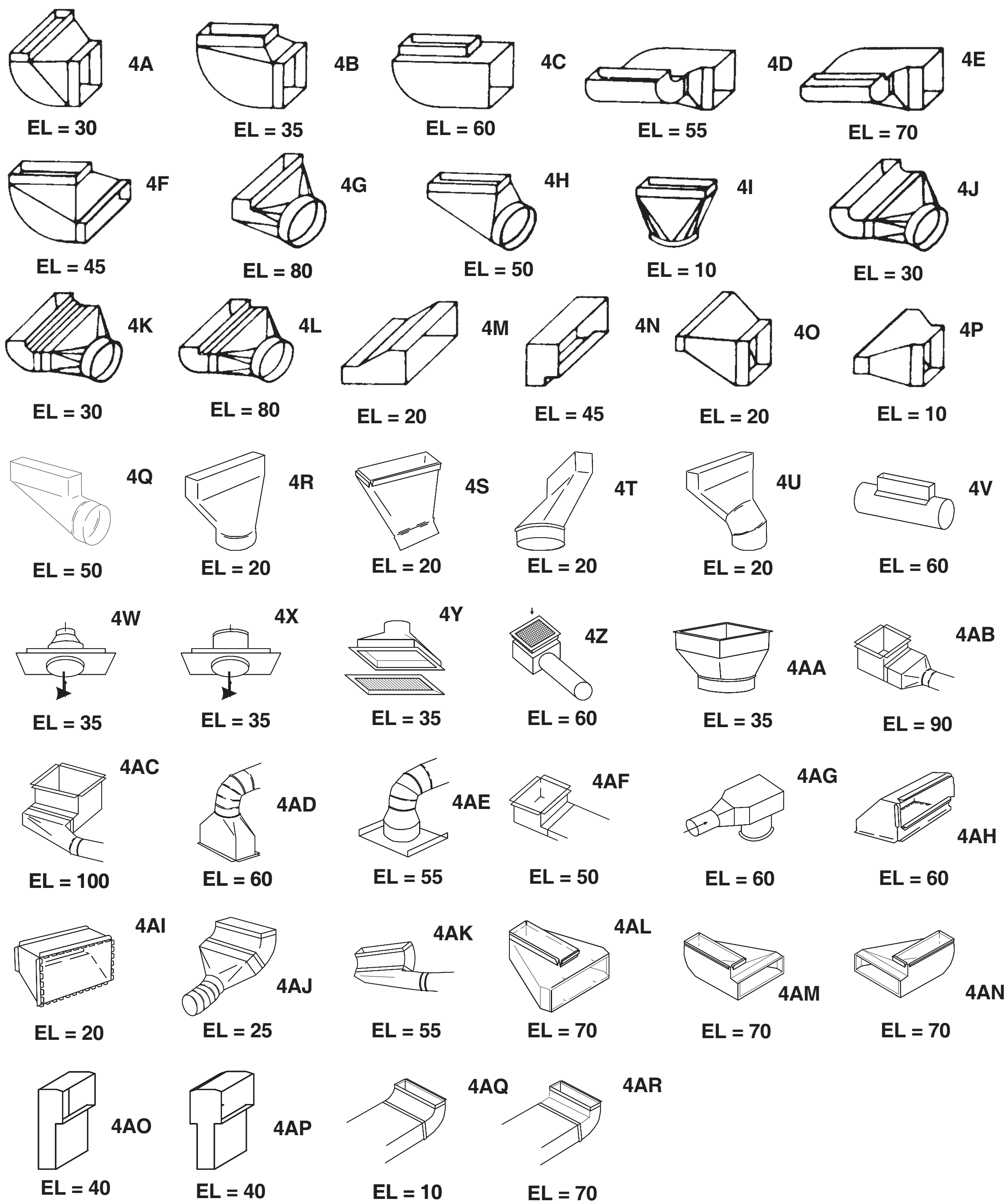
	Vanes	**EL**
Miterd Takeoff 3U	Yes	10
	No	80



	Corner	**EL**
Hard Bend Takeoff 3S	Miterd	90
	Tight	35
	Full	15

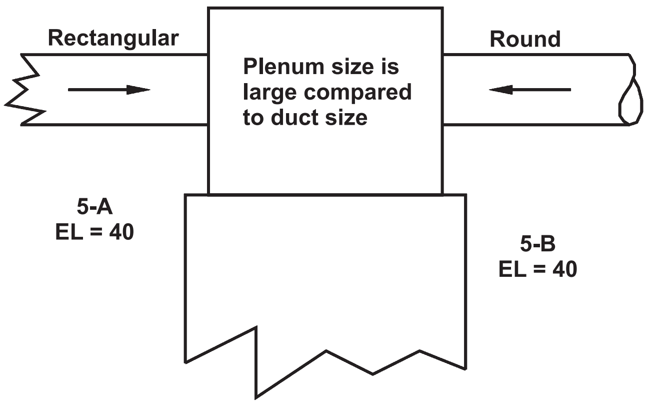
Group 4
Supply Air Boot and Stack Head Fittings

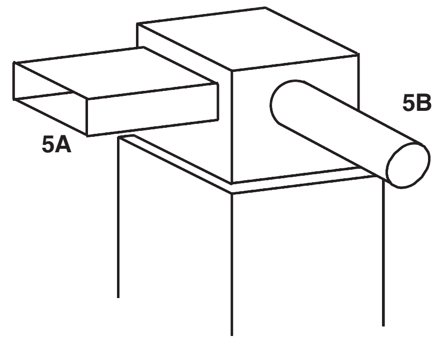
Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

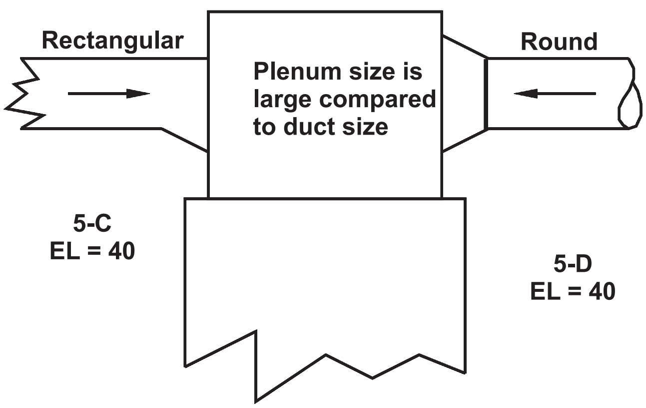


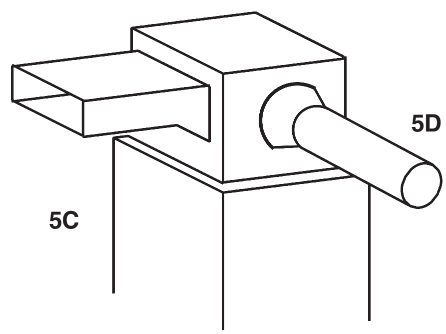
Group 5
Return Air Fittings at the Air Handling Equipment

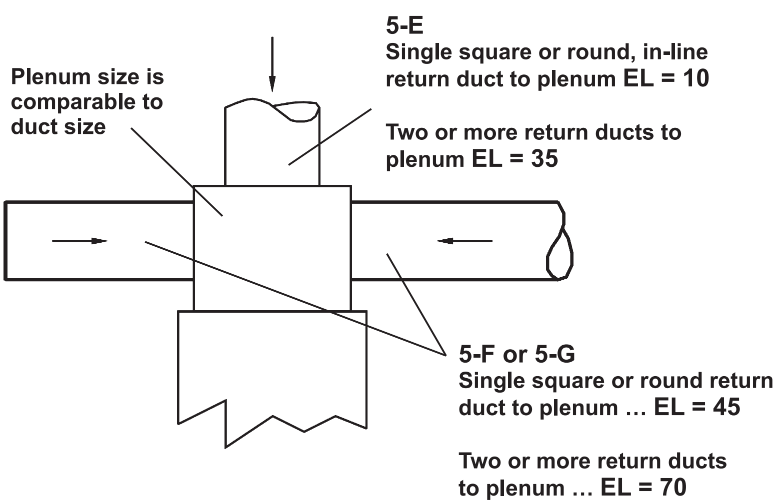
Reference Velocity = 700 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

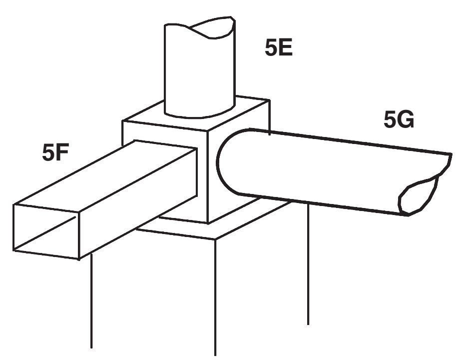






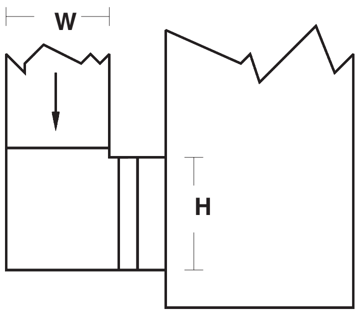


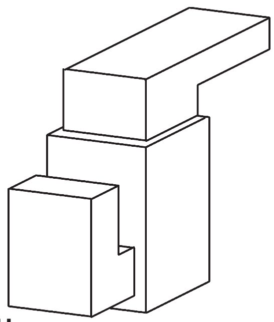




Group 5 — Continued
Return Air Fittings at the Air Handling Equipment

Reference Velocity = 700 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

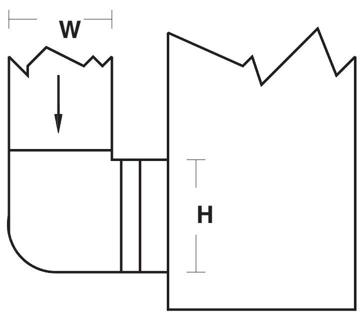


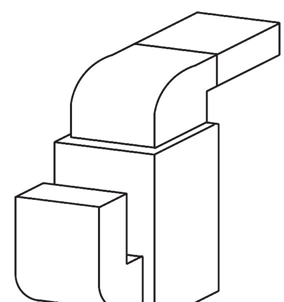


5H

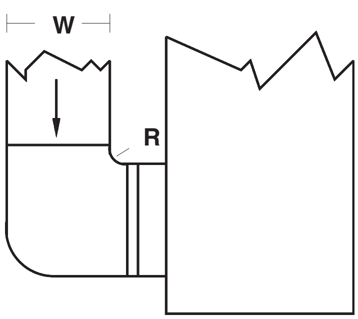
Square Elbow 5H	H / W	EL
	1	45
	2	30

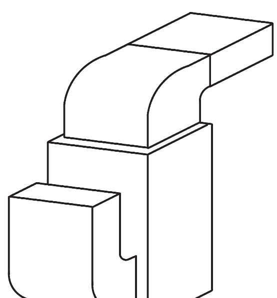
Miterd Inside Corner 5I	H / W	EL
	1	45
	2	30





5I

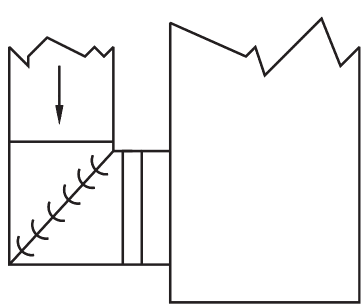


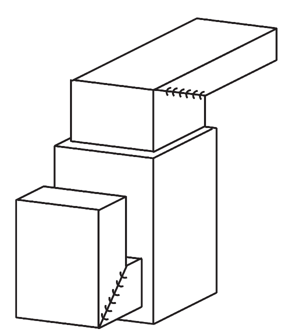


5J

Radius Elbow 5J	R / W	EL
	0.25	20
	0.50	15
	1.00	10

Square Elbow with Vanes 5K	EL
	10



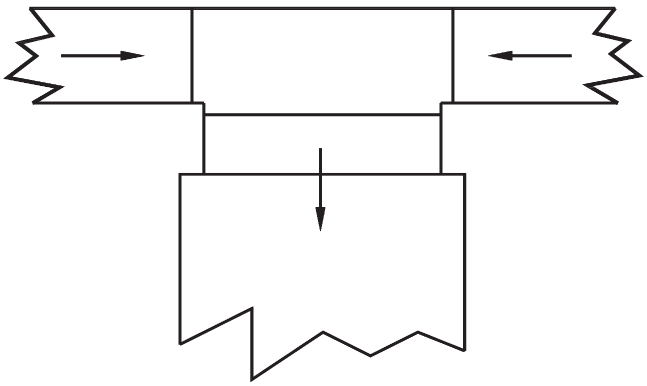


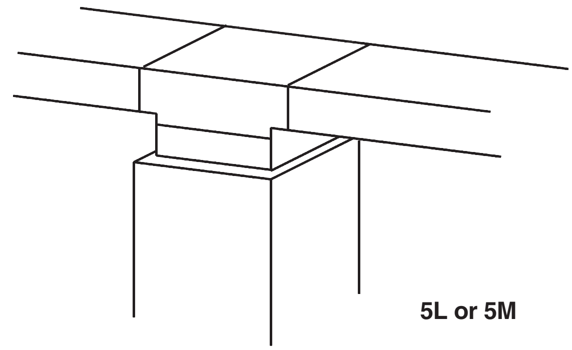
5K

Group 5 — Continued
Return Air Fittings at the Air Handling Equipment

Reference Velocity = 700 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

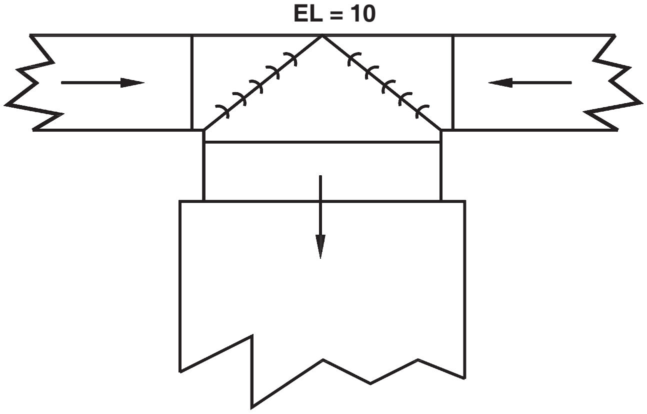
5L
EL = 75



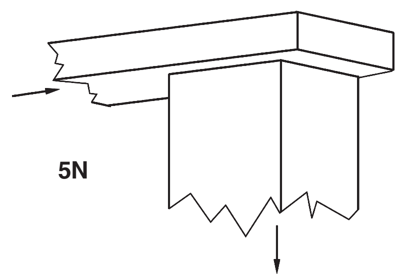


5L or 5M

5M
EL = 10

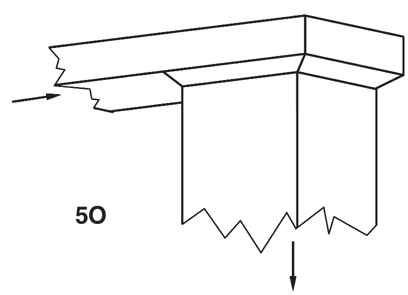


EL = 55



5N

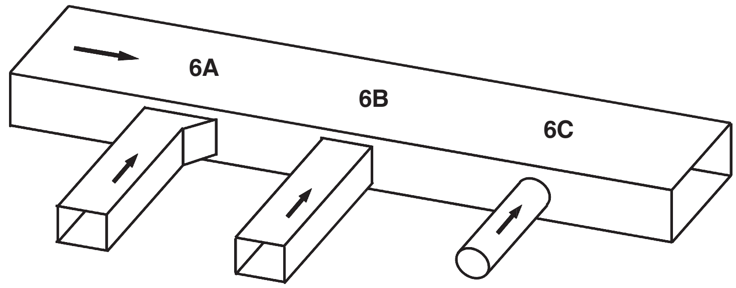
EL = 35

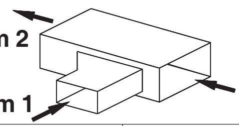
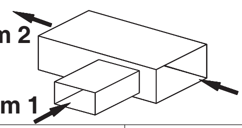
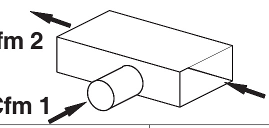


5O

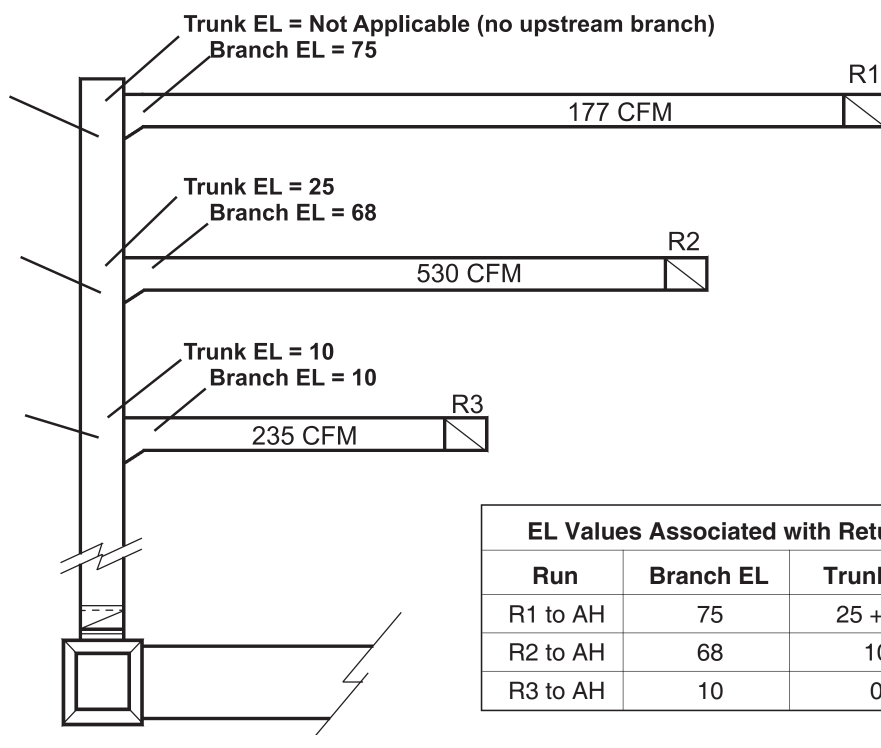
Group 6
Branch Return Air Fittings at the Return Trunk

Reference Velocity = 700 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet



	 Cfm 2 Cfm 1 6A		 Cfm 2 Cfm 1 6B		 Cfm 2 Cfm 1 6C	
Cfm1/Cfm2	**Branch EL**	**Trunk EL**	**Branch EL**	**Trunk EL**	**Branch EL**	**Trunk EL**
0.40 or less	10	10	10	10	10	10
0.50	25	25	40	25	30	25
0.60	40	25	40	25	50	25
0.70	60	25	75	25	75	25
0.80	75	25	110	25	115	25
1.00	75	NA	110	NA	115	NA

The branch EL value applies to the turn and the trunk EL value applies to the upstream fittings (see example below).

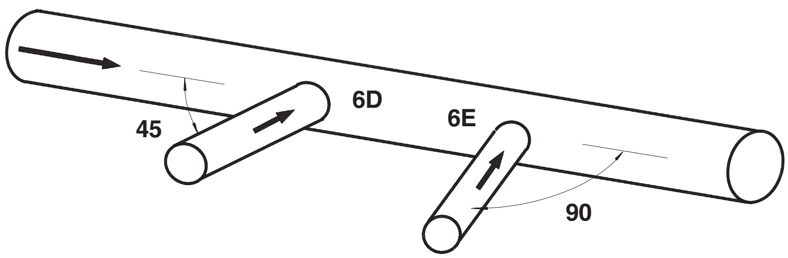


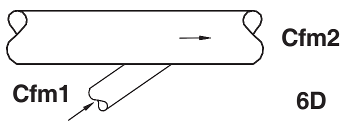
EL Values Associated with Return Trunk Fittings			
Run	**Branch EL**	**Trunk EL**	**Total EL**
R1 to AH	75	25 + 10	110
R2 to AH	68	10	78
R3 to AH	10	0	10

Figure 3-5

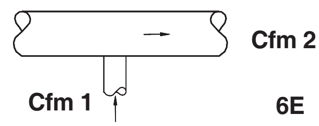
Group 6 — Continued
Branch Return Air Fittings at the Return Trunk

Reference Velocity = 700 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

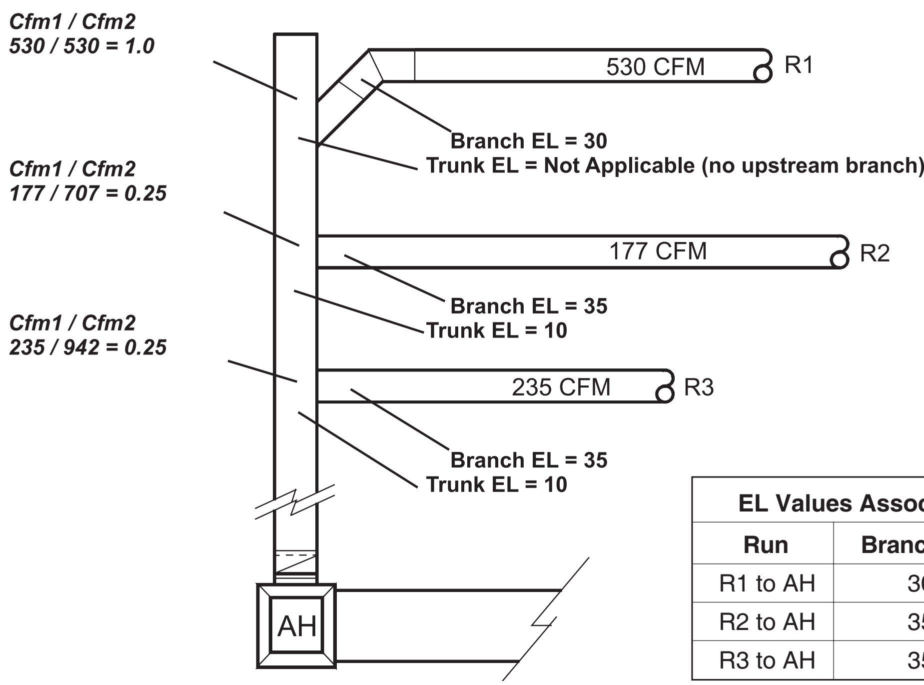




Cfm1/Cfm2	Branch EL	Trunk EL
0.10	10	5
0.20	15	5
0.30	20	5
0.40	30	5
0.60	30	5
0.80	30	5
1.00	30	NA
Refer to example below		



Cfm1/Cfm2	Branch EL	Trunk EL
0.10	15	10
0.20	30	10
0.30	40	10
0.40	40	10
0.50	40	25
0.80	40	25
1.00	40	NA
Refer to example below		

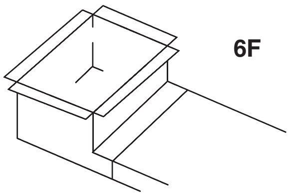


EL Values Associated with Return Trunk Fittings			
Run	Branch EL	Trunk EL	Total EL
R1 to AH	30	10 + 10	50
R2 to AH	35	10	45
R3 to AH	35	0	35

Figure 3-5

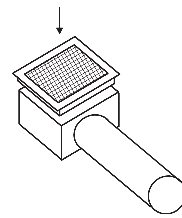
Group 6 — Continued
Return Air Boot Fittings

Reference Velocity = 700 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet



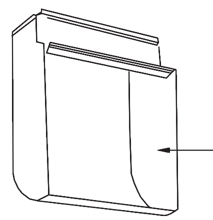
6F

EL = 25



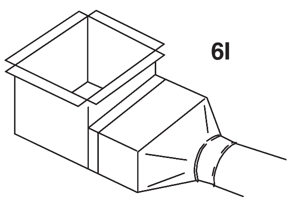
6G

EL = 30



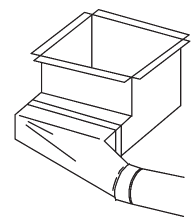
6H

EL = 15



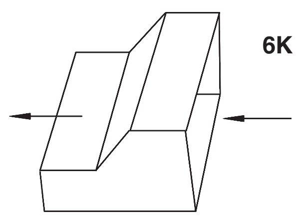
6I

EL = 30



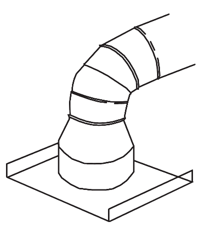
6J

EL = 55



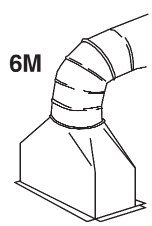
6K

EL = 10



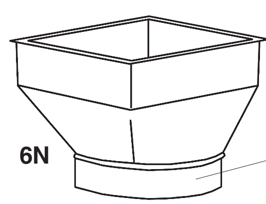
6L

EL = 20



6M

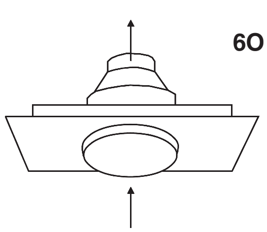
EL = 20



6N

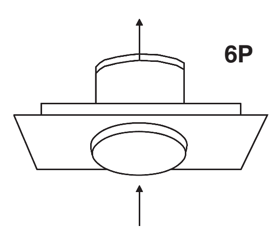
Round or Square

EL = 10



6O

EL = 10



6P

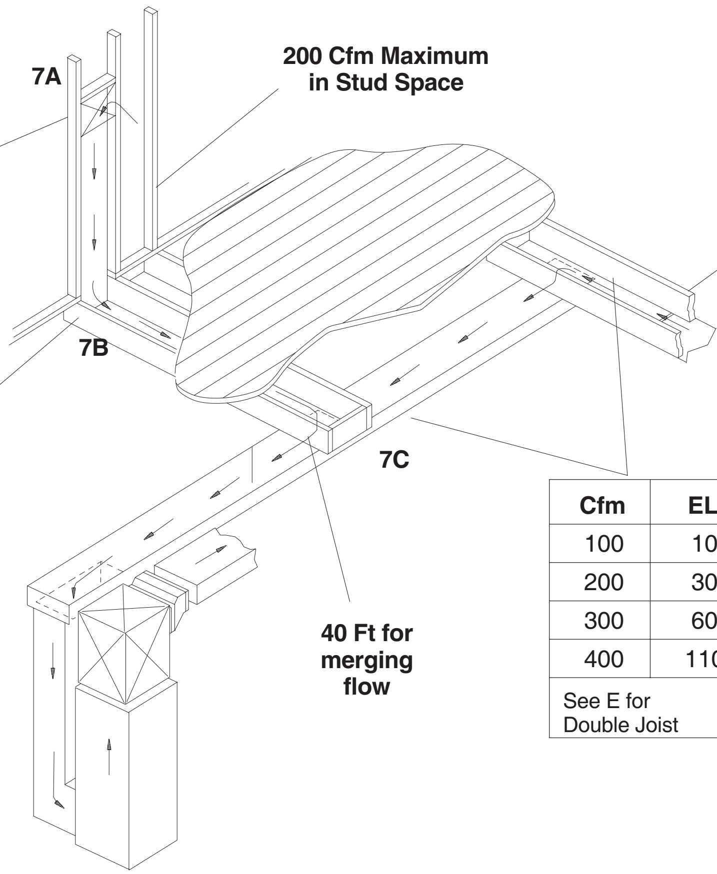
EL = 5

Group 7
Panned Joists and Panned Stud Return Air Fittings

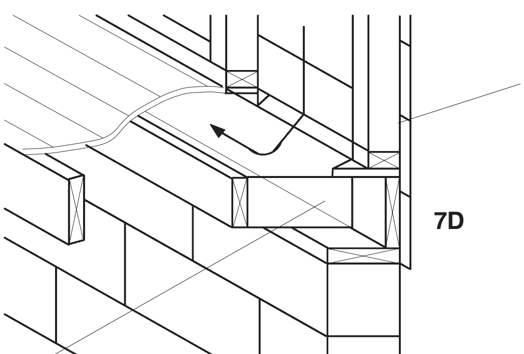
Reference Velocity = 700 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

Cfm	EL
100	25
150	
200	

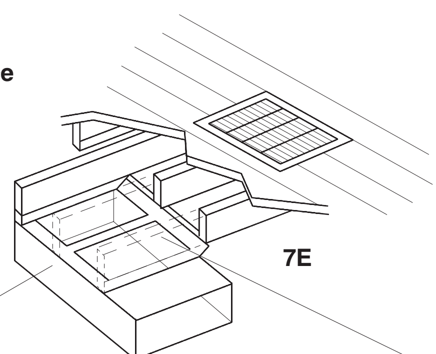
Cfm	EL
100	10
150	15
200	25
See D for Double Turn	



Cfm	EL
100	10
200	30
300	60
400	110
See E for Double Joist	



Cfm	EL
100	20
150	50
200	90



Cfm	EL
200	10
400	30
600	60
800	110

Panning is not recommended by *Manual D*. Panned joist and stud space airway leakage increases duct loads, may cause space pressure problems, and may cause indoor air quality problems. Equivalent length and airway sizing procedures are approximate, and there is no sizing procedure if there are obstructions in the airway. Group 7 equivalent length values are used to evaluate existing systems. See Section 4-5 for more discussion.

Group 8
Elbows and Offsets

Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

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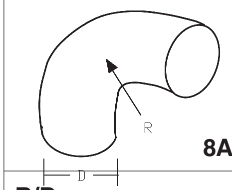
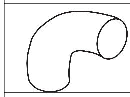
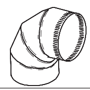
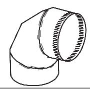
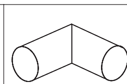
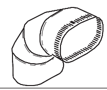
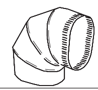
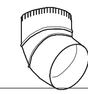
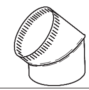
<tr><td rowspan="3">
8A
R/D</td><td colspan="8">Round and Oval Elbow EL Values</td></tr>
<tr><td></td><td></td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>Smooth</td><td>4 or 5 Piece</td><td>3 Piece</td><td>Smooth Miterd</td><td>Easy Bend</td><td>Hard Bend</td><td>3-Piece 45°</td><td>2-Piece 45°</td></tr>
<tr><td>Miterd (R = 0)</td><td>—</td><td>—</td><td>—</td><td>75</td><td rowspan="2">4-Piece
25</td><td rowspan="2">4-Piece
30</td><td rowspan="4">10</td><td rowspan="4">15</td></tr>
<tr><td>0.75</td><td>20</td><td>30</td><td>35</td><td>—</td></tr>
<tr><td>1.0</td><td>15</td><td>20</td><td>25</td><td>—</td><td rowspan="2">3-Piece
30</td><td rowspan="2">3-Piece
35</td></tr>
<tr><td>1.5 or Larger</td><td>10</td><td>15</td><td>20</td><td>—</td></tr>
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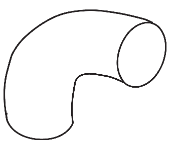
<tr><td rowspan="3">
8A — Continued</td><td colspan="8">For Smooth Radius Round Elbows
Angles (θ) Less Than 90°
Multiply EL by One of the Following Factors</td></tr>
<tr><td>20°</td><td>30°</td><td>45°</td><td>60°</td><td>75°</td><td>110°</td><td>130°</td><td>150°</td></tr>
<tr><td>0.31</td><td>0.45</td><td>0.60</td><td>0.78</td><td>0.90</td><td>1.13</td><td>1.20</td><td>1.28</td></tr>
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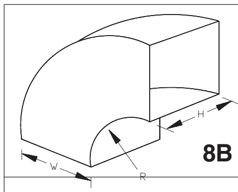
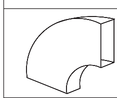
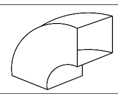
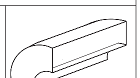
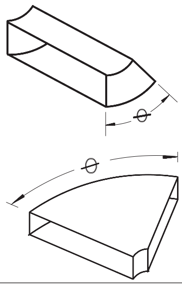
<tr><td rowspan="3">
8B
R/W</td><td colspan="3">Radius Elbow EL Values</td></tr>
<tr><td></td><td></td><td></td></tr>
<tr><td>Hard Bend</td><td>H / W = 1</td><td>Easy Bend</td></tr>
<tr><td>Miterd (R = 0)</td><td>90</td><td>75</td><td>65</td></tr>
<tr><td>0.25</td><td>35</td><td>30</td><td>25</td></tr>
<tr><td>0.5 or Larger</td><td>20</td><td>15</td><td>10</td></tr>
<tr><td rowspan="3"></td><td colspan="3">For Angles (θ) Less Than 90° Multiply EL by One of the Following Factors</td></tr>
<tr><td>30°</td><td>45°</td><td>60°</td></tr>
<tr><td>0.45</td><td>0.60</td><td>0.78</td></tr>
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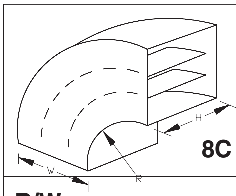
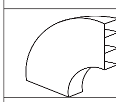
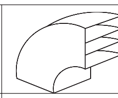
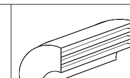
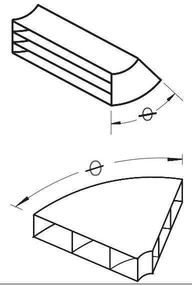
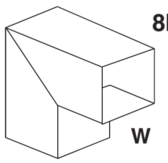
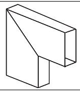
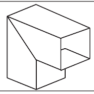
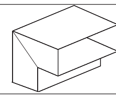
<tr><td rowspan="3">
8C
R/W</td><td colspan="3">Radius Elbow EL Values</td></tr>
<tr><td></td><td></td><td></td></tr>
<tr><td>Hard Bend</td><td>H / W = 1</td><td>Easy Bend</td></tr>
<tr><td>Miterd (R = 0)</td><td>30</td><td>25</td><td>40</td></tr>
<tr><td>0.25</td><td>10</td><td>10</td><td>10</td></tr>
<tr><td>0.5 or Larger</td><td>5</td><td>5</td><td>5</td></tr>
<tr><td rowspan="3"></td><td colspan="3">For Angles (θ) Less Than 90° Multiply EL by One of the Following Factors</td></tr>
<tr><td>30°</td><td>45°</td><td>60°</td></tr>
<tr><td>0.45</td><td>0.60</td><td>0.78</td></tr>
</table>

Group 8 — Continued
Elbows and Offsets

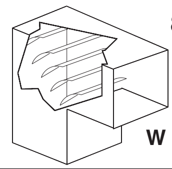
Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

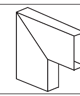
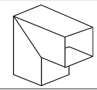
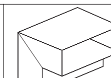
8D



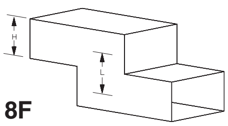
No Vanes	Square Elbow EL Values		
			
	Hard Bend	H / W = 1	Easy Bend
	80	80	65

8E



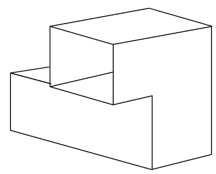
Single Thickness Turning Vanes	Square Elbow EL Values		
			
	Hard Bend	H / W = 1	Easy Bend
	10	10	10

L/H	EL
1	160
2	260
4	190



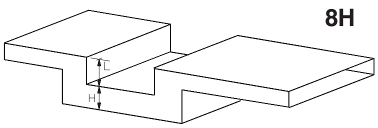
8F

No Vanes



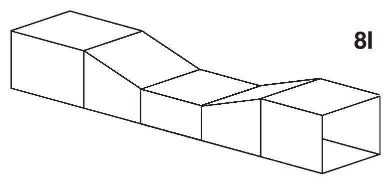
8G

EL = 200
No Vanes



8H

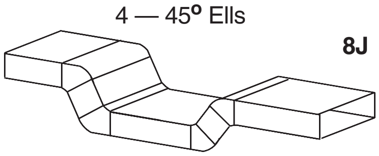
EL's H/L	No Vanes	With Vanes
0.5	55	—
1.0	330	55
1.5	430	55
2.0	470	55



8I

EL = 20

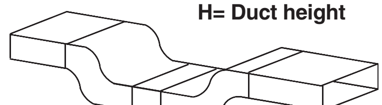
4 — 45º Ells



8J

EL = 20

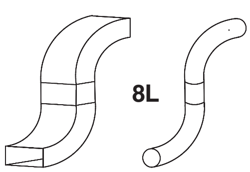
R = Inside radius
H= Duct height



4 — 90º Ells

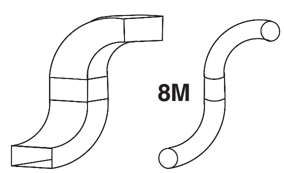
8K

R/H	EL
Miterd (R=0)	250
0.25	100
0.50	20
1.00	20



8L

Double Ell — 1 Plane
1.7 x EL value for single elbow

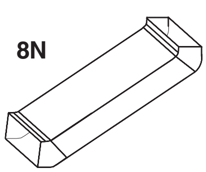


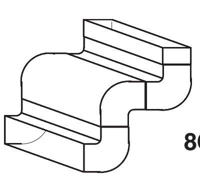
8M

Double Ell — 2 Plane
2.0 x EL value for single elbow

EL = 10

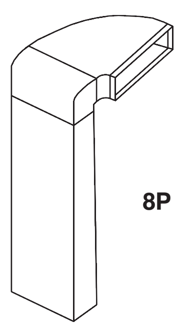
8N





8O

Inside Corner	EL
Miter (R=0)	235
R = 0.25	90
R > 0.50	45



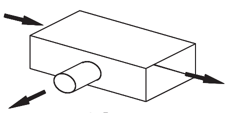
8P

EL Values	Inside Corners	
Riser	Miter	Radius
3-1/4 x 10	75	60
3-1/4 x 12	90	75
3-1/4 x 14	90	75

Group 9
Supply Trunk Junction Fittings

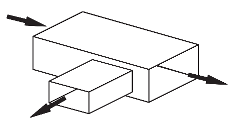
Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

The equivalent lengths for this group apply when the branch duct is a secondary trunk duct. See Group 2, Branch Takeoff Fittings for branch runout equivalent length values.



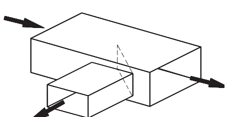
9A

	EL
Branch	80
Main	5



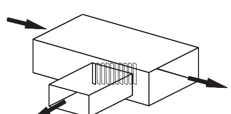
9B

	EL
Branch	80
Main	5



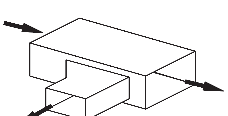
9C

	EL
Branch	80
Main	5



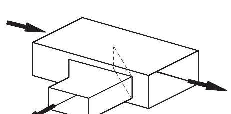
9D

	EL
Branch	75
Main	5



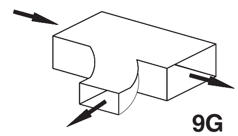
9E

	EL
Branch	50
Main	5



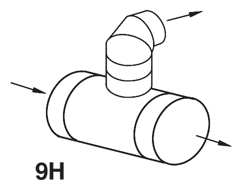
9F

	EL
Branch	45
Main	5



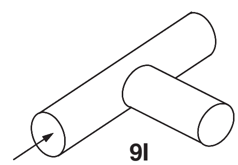
9G

	EL
Branch	35
Main	5



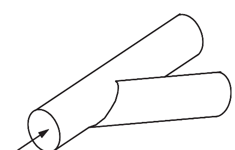
9H

	EL
Branch	100
Main	5



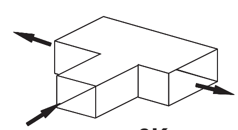
9I

	EL
Branch	85
Main	5



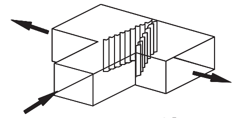
9J

	EL
Branch	25
Main	5



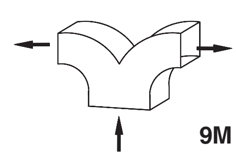
9K

EL = 65



9L

EL = 20



9M

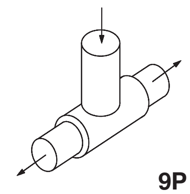
EL = 20

Group 9 — Continued
Supply Trunk Junction Fittings

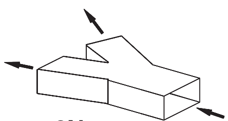
Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

The equivalent lengths for this group apply when the branch duct is a secondary trunk duct. See Group 2, Branch Takeoff Fittings for branch runout equivalent length values.

EL = 70

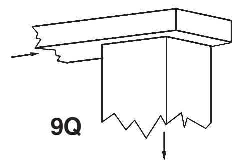


9P



9N

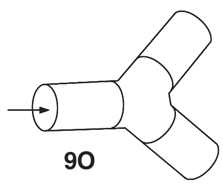
EL = 15



9Q

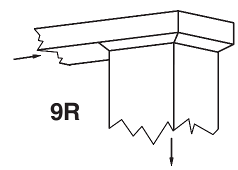
EL = 55

EL = 15



9O

EL = 35



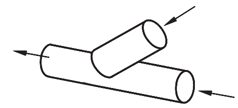
9R

Group 10
Return Trunk Junction Fittings

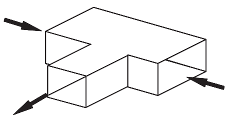
Reference Velocity = 700 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet

The equivalent lengths in this group apply when the flow in two return trunks merge. See Group 6, Branch Return Fittings, for branch return equivalent length values.

EL = 25

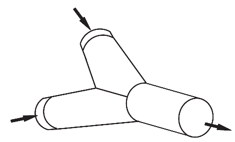


10E



10A

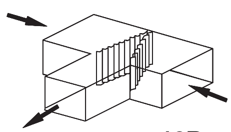
EL = 75



10F

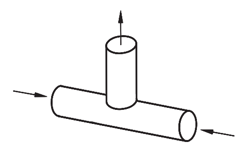
EL = 35

EL = 10

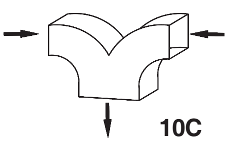


10B

EL = 75



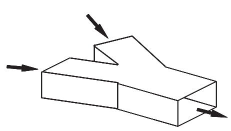
10G



10C

EL = 10

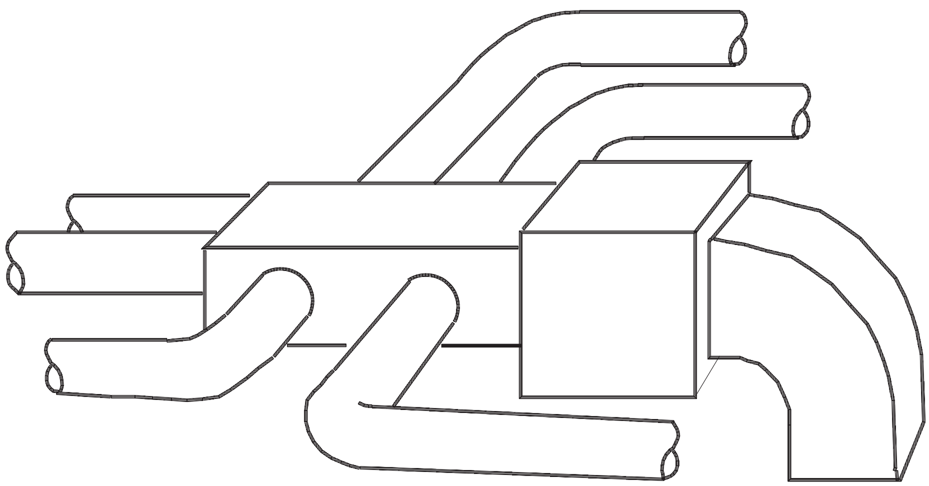
EL = 25

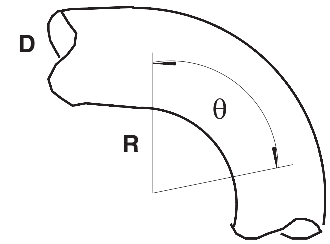


10D

Group 11
Flexible Duct Junction Boxes and Radius Bends

Reference Velocity = As Indicated
Reference Friction Rate = 0.08 IWC per 100 Feet

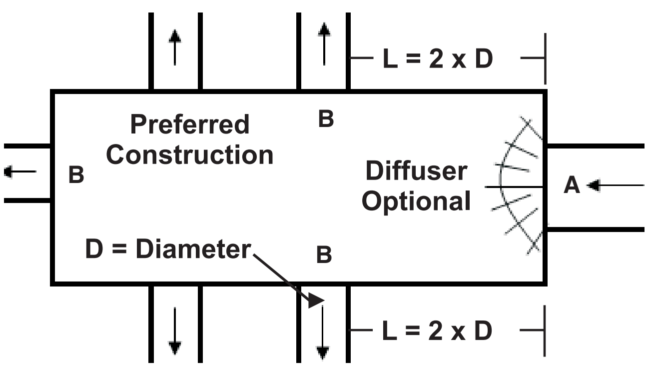




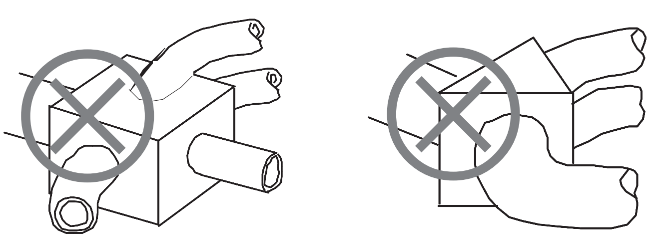
For bends that are not equal to 90°, multiply the 90° equivalent length by the ratio of the desired angle to the 90° angle.

Example: IF R/D = 1.0, find the EL for a 45° bend if the velocity equals 700 Fpm.

15 x 45 / 90 = 7.5 Ft



Recommended (compatible with Group 11 EL values)
- Entrance (A) has a diffuser fitting that recovers velocity pressures and prevents swirl (optional).
- Straight approach(A) and straight exit (B).
- Exit opening on side (no top or bottom exits).
- Exit opening at least two diameters from entrance (L).
- Make box as small as possible, but comply with L = 2 x D.



Not Recommended (Group 11 EL values may be too small)
- Turn or bend near entrance or exit.
- Top or bottom exits.
- Exit opening less than two diameters from entrance.

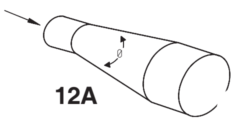
Equivalent Length Values

Velocity in Flex Duct (Fpm)	Junction Box (Ft) Notes 1, 2 and 3	90° Bend (Ft) R / D Ratio (In / In) [4]			
		1.0	1.5	2 to 3	4 to 5
400	20	5	5	5	5
500	30	5	5	5	5
600	40	10	5	5	5
700	60	15	10	5	5
800	75	15	10	10	8
900	95	20	15	10	8

1) No anti-swirl regain diffuser at entrance.
   - Swirl tends to feed one side of the box and starve the other side.
   - Swirl may be induced by spiral wire geometry.
   - Swirl attributes (such as direction) may change when the blower shuts down and restarts.
2) Straight-run approach and a straight-run departures (no turns in duct runs near the junction box).
3) Entrance and exits on side of box (no top or bottom openings).
4) Radius of turn divided by diameter of duct.

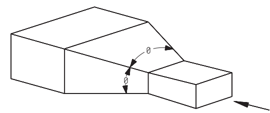
Group 12
Transitions (Diverging)

Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet



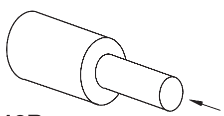
12A

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	20	40
2:1	20	40
4:1	20	30



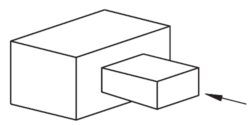
12F

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	20	40
2:1	20	40
4:1	15	30



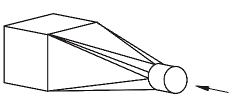
12B

EL Values	A1/A2	A1/A2
Slope	2	4
Abrupt	20	40



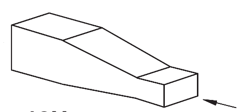
12G

EL Values	A1/A2	A1/A2
Slope	2	4
Abrupt	20	40



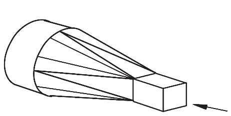
12C

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	20	40
2:1	20	40
4:1	20	30



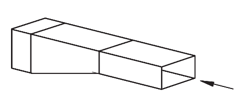
12H

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	20	40
2:1	20	40
4:1	15	25



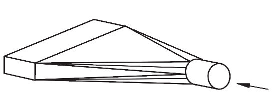
12D

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	20	40
2:1	20	40
4:1	20	30



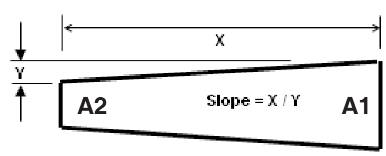
12I

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	20	35
2:1	15	25
4:1	10	10



12E

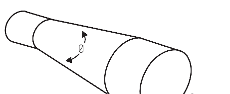
EL = 25



A1 / A2 = Larger Area / Smaller Area

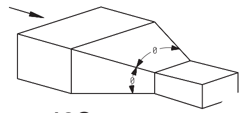
Group 12
Transitions (Converging)

Reference Velocity = 900 Fpm
Reference Friction Rate = 0.08 IWC per 100 Feet



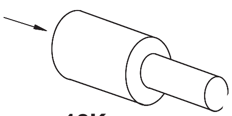
12J

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	10	10
2:1	5	5
4:1	5	5



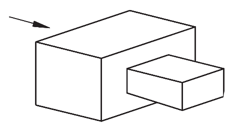
12O

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	10	10
2:1	5	5
4:1	5	5



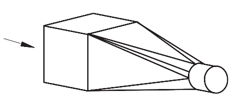
12K

EL Values	A1/A2	A1/A2
Slope	2	4
Abrupt	25	25



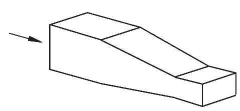
12P

EL Values	A1/A2	A1/A2
Slope	2	4
Abrupt	30	30



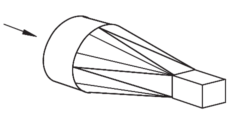
12L

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	10	10
2:1	5	5
4:1	5	5



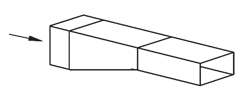
12Q

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	10	10
2:1	5	5
4:1	5	5



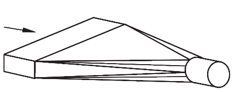
12M

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	10	10
2:1	5	5
4:1	5	5



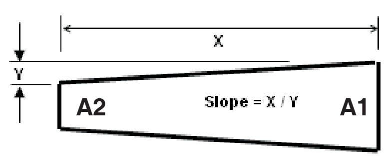
12R

EL Values	A1/A2	A1/A2
Slope	2	4
1:1	5	5
2:1	5	5
4:1	5	5



12N

EL = 10



A1 / A2 = Larger Area / Smaller Area

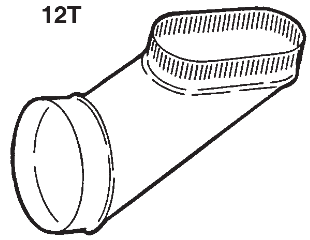
Group 12
Oval Transition Plenums and Abrupt Squeezes

Reference Velocity = 900 Fpm or as Indicated
Reference Friction Rate = 0.08 IWC per 100 Feet

12S

EL = 30

12T

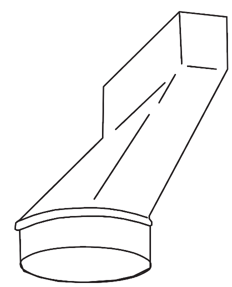


EL = 30

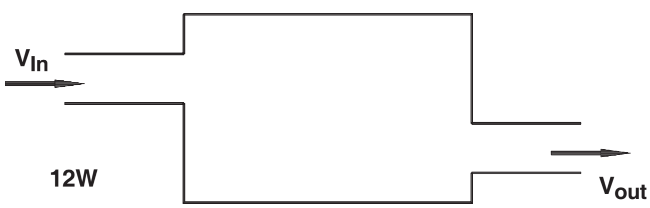
EL = 25

12U

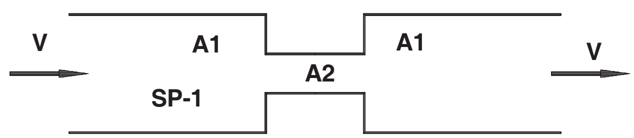
EL = 30



12V



12W



12X

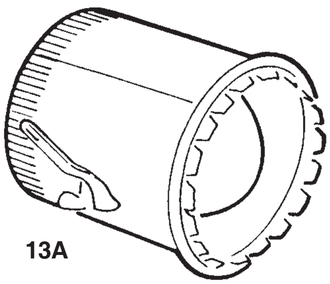
EL Values for Flow Through Large Plenum				
O_{ut} (Fpm)	V_{in} (Fpm)			
	600	700	800	900
600	40	40	35	35
700	55	55	50	50
800	70	70	70	65
900	90	90	90	85

Abrupt Squeeze				
Velocity (Fpm)	A1 / A2 = 2		A1 / A2 = 4	
	EL	Min SP-1	EL	Min SP-1
600	65	0.12	245	0.52
700	90	0.16	330	0.71
800	115	0.20	430	0.92
900	145	0.26	545	1.17

Min SP-1 = Minimum upstream static pressure (IWC) for positive static pressure at A2 (air velocity doubles or quadruples at A2).

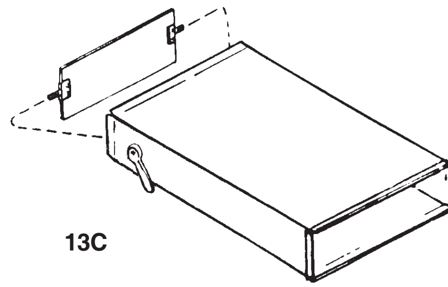
Group 13
Manual Balancing Dampers

Component Pressure Loss (CPL) for Damper Blade in Wide-Open Position
Reference Velocity = 900 Fpm



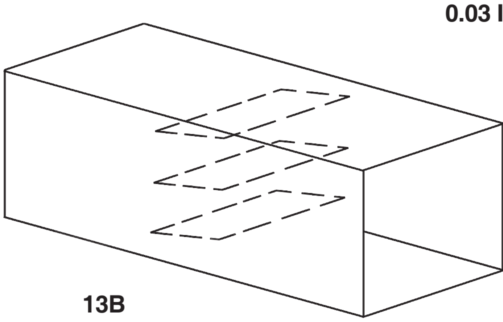
13A

Component Pressure Loss 0.03 IWC



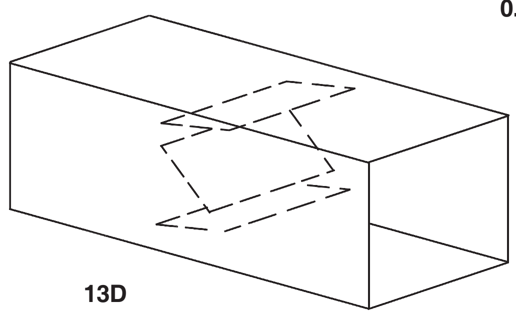
13C

Component Pressure Loss 0.03 IWC



13B

Component Pressure Loss 0.03 IWC

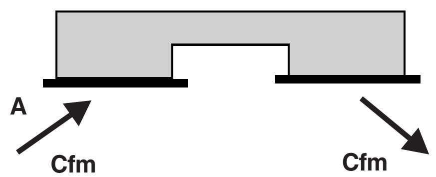


13D

Component Pressure Loss 0.03 IWC

Group 14
Transfer Duct or Transfer Grille
Component Pressure Loss (CPL) for Recommended Practice

Transfer Duct



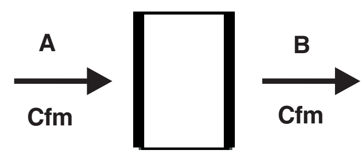
Maximum A-B Pressure Drop for Assembly = 0.05 IWC

The A-B pressure drop includes these items:

- Two return grilles at 0.02 IWC per grille, or less (use OEM performance data and Cfm for size)
- Grille frame size determines duct airway size; or flex duct equivalent
- Two 90 degree fittings; or flex duct equivalent (15 feet equivalent length per fitting, or less)

Example Grille-Airway Size	
Cfm	**Size**
100	12 x 8
200	14 x 10
300	18 x 12
400	20 x 12

Transfer Grille



Maximum A-B Pressure Drop for Assembly = 0.05 IWC

The A-B pressure drop includes two return grilles at 0.025 IWC per grille, or less.

Example Grille Size	
Cfm	**Size**
100	12 x 8
200	14 x 10
300	18 x 12
400	20 x 12

Appendix 4

Fitting Equivalent Length Adjustments

The Appendix 3 fitting equivalent length values are default values because they are based on a specific air velocity and a specific friction rate value. This provides a simple solution for fitting resistance, but it offers no credit for more efficient designs. This Appendix provides a procedure that trades air flow velocity for fitting efficiency.

A4-1 Pressure Drop In A Duct Run

Duct fittings produce a resistance to air flow. This resistance creates a pressure drop that is measured in inches water column (IWC). This pressure drop is physically equivalent to the pressure drop produced by a straight section of duct. Therefore, the total pressure drop for any duct run (supply or return) equals the sum of the fitting pressure drops and the straight section pressure drops.

A4-2 System Resistance

For airway sizing, the maximum pressure drop for the entire duct system (system resistance) equals the sum of the largest supply-side pressure drop and the largest return-side pressure drop. This includes the pressure drop for relevant straight sections, fittings and air-side components (damper, coil, supply grille, return grille, filter-grille, etc.) installed in the flow path. The route through the relevant supply and return runs is called the critical circulation path.

A4-3 Air Delivery Vs. System Resistance

A blower's ability to move air through a duct system depends on the amount of power delivered to the motor, transferred through the drive to the blower wheel and then to the air stream. Available motor power overcomes the aerodynamic and frictional resistance of the duct system. If system resistance is relatively large, the system flow rate is reduced, or vice versa. In other words, the blower has a limited amount of power, which can be used to move a lot of air against a small resistance, or to move a small amount of air against a large resistance. Therefore, the practitioner has to make sure that system resistance is compatible with the desired air flow rate and the available blower power.

A4-4 Controlling System Pressure Drop

The practitioner has a number of ways to control the system resistance. These are listed here:

- Use air-side components that have the smallest pressure possible drop. For example, a manufacturer might offer two refrigerant coils for a gas furnace. These coils have may the same cooling capacity, but one may produce less air flow resistance. (This concept applies to filters, grilles, electric resistance coils, open dampers and so forth.)
- Use duct materials that have a smooth surface. For example, metal generates less resistance than rigid fiberglass and fiberglass generates less resistance than flexible wire-helix material.
- Use aerodynamically efficient fittings. Or, as far as ***Manual D***, is concerned, use fittings that have small equivalent length values.
- Use larger air ways to slow air flow. Straight run resistance and fitting resistance decreases dramatically as the air velocity decreases (size of the airway increases).

A4-5 Equivalent Length Values Are Conditional

The Appendix 3 equivalent length values are default values that depend on the velocity and friction rate values listed in the Group Number boxes at the top of the appendix pages. These values are provided here:

- 900 Fpm and 0.08 IWC per 100 Feet for supply ducts.
- 700 Fpm and 0.08 IWC per 100 Feet for return ducts.

A4-6 Trading Fitting Length For Velocity

Fittings have a significant effect on total system resistance. In fact, the collective fitting resistance for the critical circulation path is normally much larger than the straight run resistance. This means that if a practitioner uses fittings that have relatively large equivalent length values (for cost or availability reasons, or because of personal preference), air velocity must be limited to a value that makes fitting resistance compatible with blower capability. In this regard, fitting equivalent length values get smaller than the Appendix 3 values if air velocity is less than the reference velocity (see Section A4-5).

A4-7 The Wingard Equation

ACCA member Joe Wingard (Cooper Wingard Design) has developed an equation that can be used to design duct systems that are short on blower power. It allows the practitioner to determine the maximum air flow velocity that is compatible with the blower and the fittings. In the following equation V_{max} is the maximum air flow velocity, ASP is the available static pressure from Step-3 on the Friction Rate Worksheet (external static pressure from the blower table, minus the collective pressure drop for the air-side components), F_X is an arbitrary friction rate value (use any value between 0.06 and 0.18), SL is the total length of all the straight runs in the critical circulation path, EL_S is the total of the supply run equivalent length values from Appendix 3, and EL_r is the total of the return run equivalent length values from Appendix 3.

$$V_{max} = 900 \times \left[\frac{100 \times ASP - F_X \times SL}{0.08 \times (EL_S + 1.653 \times EL_r)} \right]^{0.50}$$

A4-8 Application of the Wingard Equation

A furnace is equipped with a cooling coil. The furnace blower table shows that the blower delivers 1,400 Cfm when working against 0.52 IWC of resistance, and the footnotes indicate that the blower was tested with the standard throwaway filter in place. Manufacturer's data also shows the pressure drop across the refrigerant coil is 0.23 IWC at 1,400 Cfm. The collective pressure drop for the supply grille, return grille and branch damper is 0.09 IWC. The collective equivalent length for the supply-side fittings is 300 ft, and the associated straight run length is 50 ft. The collective equivalent length for the return-side fittings is 60 ft, and the associated straight run length is 15 ft.

ASP = 0.52 - 0.23 - 0.09 = 0.20 IWC
SL = 50 + 15 = 65 ft
EL_S = 300 ft
EL_r = 60 ft

$$V_{max} = 900 \times \left[\frac{100 \times 0.20 - F_X \times 65}{0.08 \times (300 + 1.653 \times 60)} \right]^{0.50}$$

$$V_{max} = 900 \times \left[0.626 - 2.035 \times F_x \right]^{0.50}$$

Now the practitioner has the option to specify a value for the friction rate F_x. Any value between 0.06 and 0.18 is acceptable because this range of values produces a reasonable compromise between airway size and available blower power. The results for the entire range of F_x values is provided by Figure A4-1.

Velocities for Various F_X Values

F_X	0.06	0.08	0.10	0.12	0.14	0.16	0.18
V_{Max}	638	612	585	556	525	493	458

Figure A4-1

A4-9 Modified Duct Sizing Procedure

A V_{max} value and the related F_X value are used to size the duct runs. Simply use the ACCA Duct Sizing Slide Rule and the following procedure to find the airway size for each section of duct:

- Determine the Cfm for each section of trunk duct and for each runout duct.
- Use the sectional Cfm values and the desired V_{max} value to size the duct runs.
- Use the sectional Cfm value and F_x value to size the duct runs.
- Select the larger of the two sizes and install this size.

Note that any combination of F_X and V_{max} values will work, but smaller airway sizes are generated by using smaller F_X values, which are 0.06, 0.07, 0.08, 0.09 or 0.10.

Example of Sizing Procedure

Figure A4-3 (next page) provides a simple example that demonstrates the alternative sizing procedure. Sectional flow rates are noted on the sketch. V_{max}, F_X sizes and airway sizes are listed by Figure A4-2. All airway sizes were read from the ACCA Duct Sizing Slide Rule. An 0.06 F_X value was used for this design. Figure A4-1 shows that V_{max} is 638 Fpm for this design.

Trunk Section	Cfm	Size @ 638 fpm	Size @ F_X 0.06	Min Size	Install Size
1	1,400	20.1	17.4	20.1	20
2	1,300	19.5	17.1	19.5	20
3	1,100	17.9	15.9	17.9	20
4	950	16.7	15.1	16.7	20
5	900	16.2	14.8	16.2	16
6	800	15.3	14.2	15.3	16
7	600	13.3	12.7	13.3	16
8	400	10.7	11.0	11.0	12
9	250	8.5	9.2	9.2	12
10	150	6.5	7.5	7.5	12
Branch Runs	**Cfm**	**Size @ 638 fpm**	**Size @ F_X 0.06**	**Design Size**	**Install Size**
50's	50	3.7	5.0	5.0	5
100's	100	5.4	6.5	6.5	7
150's	150	6.5	7.5	7.5	8
200's	200	7.6	8.4	8.4	9

Figure A4-2

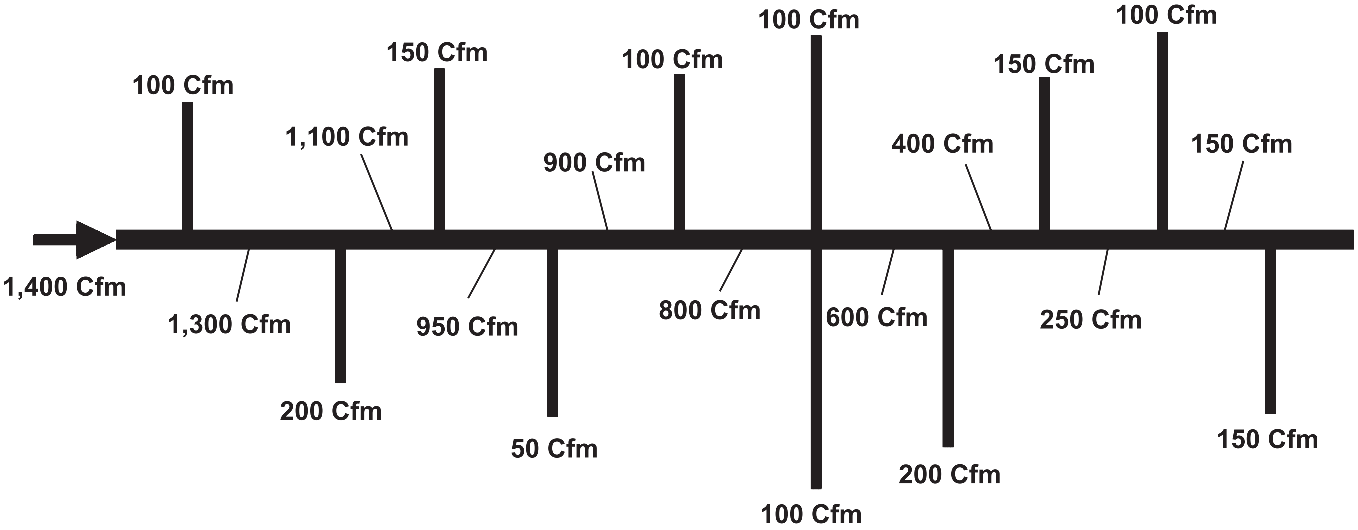

100 Cfm
150 Cfm
100 Cfm
100 Cfm
150 Cfm
100 Cfm
1,100 Cfm
900 Cfm
400 Cfm
150 Cfm
1,400 Cfm
1,300 Cfm
950 Cfm
800 Cfm
600 Cfm
250 Cfm
200 Cfm
50 Cfm
100 Cfm
200 Cfm
150 Cfm

Figure A4-3

Appendix 5

Terminology

Adequate Exposure Diversity (AED): A ***Manual J*** term for dwellings that do not have a significant spike in the daily cooling load on the central cooling equipment.

AED Excursion: The peak value for the daily cooling load exceeds the average daily load by more than a factor of 1.3 (a ***Manual J*** issue).

Air Distribution Effectiveness: The ability of grilles, registers or diffusers to adequately mix supply air with room air.

Air Distribution System: Supply duct runs, return duct runs, blower, supply outlets, return grilles, balancing dampers and optional automatic flow control (modulating or fixed-position zone dampers).

Air Handling Equipment or Air Handler: The cabinet containing the blower (i.e., a furnace; a single-package outdoor heating and cooling unit; the indoor section of a split air conditioning or heat pump system).

Air-Side Component: Equipment or component installed in a duct run or equipment cabinet (filter, refrigerant coil, water coil, electric heating coil, humidifier, automatic flow control damper, supply grille, return grille, hand damper, for example).

Air Way: The flow area for a length of duct that has constant perimeter dimensions.

Air Way Dimensions: Round duct diameter, or the width and height of rectangular duct or oval duct.

Air Zoning: An air distribution system that is equipped with a set of supply air dampers and associated controls, that provides zone temperature control.

Approach: The flow pattern of the flow entering a grille, register or diffuser.

Block Load: The total heating or cooling load for the entire space served by a piece of heating or cooling equipment (which may be one room or space, a group of rooms and spaces, or the entire dwelling).

Boot: A duct fitting that transforms branch runout geometry to the collar or neck geometry of a grille, register or diffuser (entrance and exit shapes are topologically equivalent, but may, or may not, have equal flow areas).

Branch Duct: A duct that routes air to one supply air outlet.

Branch Return: A duct that routes air collected at one return grille.

Bypass Air: Air that leaves the primary equipment and short circuits back to the equipment though a dedicated duct run (bypass duct). The condition of the bypass air is essentially the same as the supply air. Bypass air is mixed with normal return air, and possibly outdoor air, before it enters the equipment. This process causes a temperature ramp at the equipment; which means that the discharge air temperature increases over time for heating, or decreases over time for cooling, but settles at a final value. Should the final values exceed the equipment's limit temperature for discharge air, the equipment's limit control will stop the equipment (assuming appropriate controls).

Bypass or Bypass Duct: An engineered flow path that routes excess supply air back to the heating-cooling equipment or dump zone.

Compression: When the length of the centerline of an installed flexible wire helix duct is less than its fully extended (pulled taught) length.

Conditioned: Room or space that is heated, or heated and cooled.

Constant Cfm: Blower operates at one Cfm for any heating or cooling load condition (heating Cfm may be optionally different than cooling Cfm**).**

Critical Circulation Path: The supply run that has the longest effective length plus the return run that has the longest effective length.

Damper: An adjustable (manually set or actuator controlled) blade or louver inserted in an airway (to adjust the flow rate through the airway).

Design Air Flow: The design value for blower Cfm shall be determined by ***Manual J*** loads, use of equipment manufacturer's expanded performance data and ***Manual S*** procedures. The design values for sizing duct runs shall be determined by ***Manual D*** procedures (compare the heating Cfm with the cooling Cfm and use the larger value).

Desired Air Flow: The target values for system Cfm and the Cfm supplied to the rooms and space when the system is adjusted and balanced. Ducts are sized for the maximum flow requirement, but target values for system balancing depend on system type (single-zone, constant-cfm or multi-zone, variable-cfm) and the strategy for dealing with different Cfm requirements for heating and cooling.

Diffuser: A supply air terminal that discharges air in many directions and multiple planes (may, or may not, be equipped with a flow control damper).

Discounted Excursion Load: For ***Manual J*** Eighth Edition calculations, 30 percent of the average daily load is subtracted from the peak fenestration load (see ***Manual J*** Appendix 3 and Appendix 11).

Distributed Relief: Engineered air leakage through zone damper stops (the system designer must decide which zones will have damper stops, and provide design values for stop leakage).

Diversity: When glass faces different directions, fenestration loads peak at different times of year and hours of day. Therefore, the instantaneous aggregate fenestration load is always less than the sum of the peak fenestration loads for all items in the set.

Draft: Air motion at some point in the in the conditioned space exceeds 50 Fpm.

Drop: A plume of cold air discharged from a ceiling or high side wall outlet falls into the occupied zone before being mixed with room air.

Duct Run: An airflow connection that has a starting point at a piece of equipment, air-side device or component, or fitting; and a termination point at a piece of equipment, air-side device or component, or fitting. For flexible wire helix duct, an engineered bend is an elbow fitting (for the purpose of determining equivalent length for a duct run). See also Engineered Bend and Span Length.

Dump Zone: An unoccupied room or space used to route excess air flow back to the blower as thermostatically controlled zone dampers throttle supply air flow to conditioned spaces.

Dwelling: Low-rise residential housing; home/house, duplex units, triplex units, townhouse, condominium three stories or less.

Effective Length: Measured length (centerline) of all duct sections in a duct run, plus the sum of all fitting equivalent lengths in the run (flexible wire helix duct has an equivalent length).

Engineered Bend: For flexible wire helix duct, a bend that serves as a Group 11 elbow fitting. See also, Duct Run. and Span Length.

Equipment Sizing: Using equipment manufacturer's expended performance data (see ***Manual S***) to match equipment heating and cooling capacity with ***Manual J*** heating and cooling loads.

Equivalent Length: For duct fittings, the air flow resistance produced by a fitting is equivalent to feet of straight duct that produces the same air flow resistance (this fabricated relationship is subject to explicit caveats, see Appendix 4).

Excess Air: For air-zoned systems, the difference between the momentary need for supply air Cfm, and the momentary value for blower Cfm.

Excess Length: For flexible wire helix duct, the fully extended cut length exceeds the straight line span length.

Excessive sag: See Sag.

Extended Plenum: A main trunk that feeds branch ducts (perimeter dimensions are constant along the length).

Grille: A cover for the opening at the end of a supply air duct or entrance to a return air duct, consisting of a frame and a grid of vanes or louvers.

- The grid may be a simple matrix of loosely spaced vanes (see through), or angled vanes or louvers that may, or may not, overlap (see through, limited sight line, or no sight line).
- Supply grille discharge may be in one to four directions, directional discharge is in one plane (a grille with a damper is a register).
- A return air grille or return air filter grille.
- For ***Manual D***, grille is a generic term for air mixing and air collection hardware.

Face Velocity: Grille or diffuser flow rate (Cfm) divided by free area (SqFt).

Fenestration: Windows, glass doors and skylights.

Free Area: The flow area inside the boundary of a grille frame (discharge Cfm divided by face velocity).

High Velocity Air: A draft (more than 50 Fpm; 25 to 35 Fpm is ideal).

Impermeable Membrane: For below grade walls, an effective barrier to gases, odors and moisture in the surrounding soil. (The ASTM E283 Standard provides instructions for testing the performance of membrane material).

Inlet: Return grille.

Isolated Room: A room with no local return and an interior door.

Load: Sensible heating load or sensible cooling load (output from ***Manual J***).

- Envelope loads (block; zone; space; room).
- System loads (engineered ventilation, duct, hot water pipe, etc.).

- Equipment sizing loads (envelope loads plus system loads).

Load Condition: Instantaneous heating or cooling load.

Loop: For flexible wire helix duct, four smooth, aerodynamic, close-coupled bends that cause no change in direction (entering air velocity vector equals exiting air velocity vector).

Long Arc Sag: See Sag.

Low-Resistance Return Path: See Section 4-9 and Section A1-2.

Main Trunk: A length of supply duct or return duct that routes the entire flow (Cfm) discharged from the blower.

- Perimeter dimensions are constant along the length.
- The flow rate (Cfm) at the entrance of a supply duct equals the blower Cfm.
- Downstream flow rates diminish if air is routed to secondary trunks and/or branch ducts.
- The flow rate at the exit of a return duct equals the blower Cfm. Upstream flow rates diminish if air is collected from secondary trunks and/or branch returns.

Manual J: The current unabridged version of the ACCA/ANSI Standard.

Multi-Speed Operating Point Blower: A blower that has two or more distinct speed settings; manually set or conditionally set by automatic control (PSC Blower). For a given speed setting, each blower Cfm value in the blower table has a different external static pressure value.

Multi-zone: Heating and/or cooling for two or more conditioned spaces is controlled by two or more local thermostats (one thermostat per zone; one central source of heat and cooling; or distributed sources of heat and cooling).

Negligible Compression: ADC FD72-R1, the test code from which flexible duct friction charts are developed, require the duct be stretched to its fullest length by applying a force of 25 pounds pull, holding for one minute, and then allowing the duct to retract to its normal length.

Negligible Sag: See Sag

Noise Criteria (NC): A single value rating for human perception of noise produced by a supply air outlet. NC values may, or may not, appear in product performance data (more common for commercial products, less common for residential products).

Occupied Zone: The volume of conditioned space that is two feet from all walls and 6.5 feet high.

Operating Control: Controls that monitor space conditions and adjust system operation to maintain space temperature and humidity, providing the equipment is not subjected to an abusive operating condition. Operating controls must pass a 1,000,000 or more cycle test. A safety control only needs to pass a 50,000 cycle test, therefore a safety control shall not be used as an operating control. See also, Safety Control.

Optional *Manual J* Procedure: Room, space or zone sensible cooling load for fenestration equals the peak cooling load for the day. Computer software performs these calculations (they cannot be performed by hand).

- See the unabridged edition of ***Manual J***, Version 2.10 or later, Appendix 11, Figure A11-1.
- The sensible cooling load has no diversity.
- The peak zone load is used to size local equipment that serves a portion of the dwelling, which consequently determines the design value for blower Cfm for the local equipment.
- The associated room loads are peak cooling loads for the whole day, which consequently determine the design values for room or space cooling Cfm for any type of variable Cfm system, or a local constant Cfm system.
- For constant Cfm systems and central variable Cfm systems, see the definition of the Standard ***Manual J*** Procedure.

Outlet: Grille, register or diffuser.

Over Blow: For air zoning, the Cfm delivered to a zone exceeds the design value for zone Cfm. This provides a method of air relief as other zone dampers close.

Peak Load: The ***Manual J*** value for the peak sensible cooling load for a zone. The zone may be one room, or a set of rooms and spaces. The optional ***Manual J*** procedure provides peak load values for zone system design (not applicable to single zone systems that serve a set of rooms and spaces).

Primary Trunk: For the supply-side; the section of duct immediately downstream from the blower that feeds the entire distribution system. For the return-side; the section of duct immediately upstream of the blower, that collects air from the entire return system.

Register: Supply air grille with integral damper.

Return: Return air grille or return air filter grille.

Room Load: See load.

Run: See Duct Run.

Sag: When the centerline of a flexible duct that runs from point A to point B is not coincident with a straight line from point A to point B. The duct centerline may have simple (arc) or complex (snake) curvature between the entrance and exit openings.

Negligible Sag: 2.5 inches sag per 5 feet of span (0.5 inch vertical displacement per foot of span).

Short arc sag: 5.0 inches sag per 5 feet of span or 5.0 inches sag per 10 feet of span (1.0 to 2.0 inches per foot).

Long arc sag: 10.0 inches sag per 5 feet of span or 10.0 inches sag per 10 feet of span (0.5 to 1.0 inches per foot).

Excessive sag: Sag that exceeds two inches per foot.

Safety Control: Controls that monitor minimum air flow, excessive air temperature, low air temperature, amperage, voltage, refrigerant pressure, etc.. Safety controls must pass a 50,000 cycle test. A operating control only needs to pass a 1,000,000 cycle test, therefore a safety control shall not be used as an operating control. See also, Operating Control.

Secondary Trunk: For the supply-side; the section of duct that feeds part of the distribution system (for more than on room or space). For the return-side; the section of duct that collects air from the part of the return system (for more than one return).

Selective Throttling: As capacity control reduces blower Cfm and/or Btuh capacity at part-load, software/firmware routes excess air to one or more zones that are conditionally used as dump zone for a limited period of time.

Short Arc Sag: See Sag.

Single-Zone: Heating and/or cooling for all conditioned spaces is controlled by one central thermostat (one central source of heat and cooling).

Space Load: See load.

Span Length: For flexible wire helix duct, the straight line distance (SLD) between the entrance and exit of the duct run; or the SLD between the entrance and a engineered bend; or the SLD between two engineered bends; or the SLD between an engineered bend and the exit. See also Engineered Bend and Duct Run.

Standard Air: Standard air has a specific weight of 0.075 pounds per cubic foot, which is the specific volume of 70°F air at sea level.

Standard *Manual J* Procedure: Space sensible cooling load for fenestration equals the average daily fenestration load plus the AED excursion adjustment load. The excursion adjustment load is zero if a dwelling, room or space has AED, or it may range from zero to thousands of Btuh if the dwelling, room or space does not have AED. Computer software performs these calculations (they cannot be performed by hand).

- See the unabridged edition of ***Manual J***, Version 2.10 or later, Appendix 11, Figure A11-1.
- The block cooling load for the entire conditioned space is used to size central cooling equipment, which consequently determines the design value for blower Cfm.
- The associated room loads are average cooling loads for the whole day, which consequently determine the design values for room or space cooling Cfm for constant Cfm systems.
- For variable Cfm systems, see the definition of the Optional ***Manual J*** Procedure

Stratification: Pockets or layers of cold air or warm air (due to lack of mixing supply air with room air).

Supply Air Outlet: Diffuser, register or grille.

System Curve: A graph of the air flow resistance (IWC) produced by the critical circulation path vs. system flow rate (Cfm). Where air flow resistance accounts for all pressure dissipating items not considered by the equipment manufacturer's blower table (duct runs, duct fittings, air-side components and accessories).

Thermal Mass: The heat storage capacity of building materials, structural panels, structural assemblies and furnishings.

Throw: The distance that it takes for the primary flow from a supply air outlet to slow from the face velocity at the outlet (typically 600 Fpm to 800 fpm, if the outlet is sized correctly) to a relatively slow value (typically 50 Fpm to 100 Fpm, as designated by the OEM's performance data for the product).

Tight or Semi-Tight: Tight or semi-tight construction, as defined by ***Manual J***, Table 5A and Table 5B.

Transfer Grilles: Method of establishing a low-resistance a return air path between rooms with interior doors (other than bath rooms) and a central return (i.e., a door grille or a transfer duct with grilles; see ACCA ***Manual T***, Section 11-4).

Variable Air Volume: Heating or cooling capacity delivered to a conditioned space is adjusted by modulating supply air Cfm.

Variable-Speed: Incremental speed settings; conditionally selected by automatic control.

Vapor Retarder or Barrier: A membrane that significantly reduces the flow of water vapor from a more humid ambient to a less humid ambient. Refer to standards and codes for guidance.

For example (Sept 2008), IRC 2006 language in M1601.3.4 states "A vapor retarder having a maximum permeance of 0.05 *perm [(2.87 ng/(s·m2·Pa)] in accordance with ASTM E96, or aluminum foil with a minimum thickness of 2 mils (0.05 mm), shall be installed on the exterior of insulation on cooling supply ducts that pass through non-conditioned spaces conducive to condensation."*

VAV Damper: A modulating component controlled by a local (room or zone) thermostat.

Variable-Speed Operating Range Blower: A blower driven by a variable-speed motor (ECM motor). For constant Cfm systems, the blower Cfm value is practitioner-selected and automatic controls determine the motor speed that will produce the desired blower Cfm (a range of external static pressure is associated with the Cfm set point). For variable Cfm systems with zone dampers, automatic controls adjust motor speed to satisfy system air flow requirements.

Water Coil: Hot water coil; chilled water coil; change-over coil (one coil used for hot water heating and chilled water cooling).

Wheel Speed: The RPM of a squirrel cage blower or a propeller fan (i.e, fan speed blower speed).

Zone: A room or space, or a set of rooms and spaces, that has its own point of temperature sensing and control. (The associated comfort system/equipment/devices must be capable of satisfying the requests issued by local control.)

Zone Damper: An automated open-close damper, multi-position damper, or modulating damper in a supply air duct, typically controlled by a zone thermostat.

Zone Damper System: An engineered package of heating-cooling equipment and its operating and safety controls, duct work, supply air outlets and returns, zone thermostats and dampers, local zoning controls, and a central zone-control panel. There may be a bypass air duct and a bypass damper. The zoning controls and bypass controls (if applicable) must be compatible with the actions of the heating-cooling equipment controls.

Zone Load: See load.

Zoned System: The temperature in two or more rooms or spaces is controlled by thermostats located in selected rooms or spaces. Each thermostat controls a throttling damper located in a supply air duct. The blower may have one speed, multiple speeds, or speed may vary. Heating and cooling equipment may have one capacity, staged capacity or variable capacity. A bypass duct may, or may not, be required. System operating controls monitor airflow and adjust the position of the bypass damper (if installed). System operating controls monitor equipment operation and adjust equipment capacity or cycle equipment capacity accordingly. Equipment safety controls protect against low air flow, high discharge temperature (heating) and low return air temperature or discharge temperature (cooling).

Informative Appendices

Informative Appendices are not part of the standard.

Appendix 6 — Duct Construction Standards

Appendix 7 — Standard of Care and Continuity

Appendix 8 — Residential Air Distribution Systems

Appendix 9 — Equipment and Air-Side components

Appendix 10 — Duct System Efficiency

Appendix 11 — Duct Leakage and System Interactions

Appendix 12 — Air Quality Issues

Appendix 13 — Noise

Appendix 14 — Testing and Balancing

Appendix 15 — Air Velocity for Ducts and Grilles

Appendix 16 — Excess Length and Sag in Flexible Duct

Appendix 17 — Symbols and Abbreviations

Appendix 18 — *Manual D* Worksheets

Appendix 6 (Informative; not Part of the Standard)

Duct Construction Standards

This appendix provides information about duct system fabrication, installation and sealing requirements. These requirements pertain to material performance requirements, fabrication procedures, installation techniques and duct sealing methods. Comprehensive and authoritative guidance is provided by the cited codes, manuals, standards, and good practices documents.

A 6-1 Pertinent Standards and Codes

Duct construction codes, manuals, standards, and good practices deal with:

- Material performance requirements.
- Assembly and fabrication techniques.
- Installation procedures.
- Closure and sealing.
- Insulation requirements.
- Repair issues.

Some of this guidance applies to fabricator-installers and the remainder of the information concerns companies that manufacturer basic materials.

Fibrous Board Duct Standards

The North American Insulation Manufacturers Association (NAIMA) manuals apply to fibrous glass duct systems. Refer to the **Fibrous Glass Residential Duct Construction Standards Fifth Edition 2003** publication for information about systems that will be subjected to pressures (positive or negative) that are less than 0.50 IWC. Note that this manual does not apply to every residential system because some residential equipment has a blower that produces pressures that exceed 0.50 IWC. In these cases refer to the **NAIMA manual; Fibrous Glass Duct Construction Standards Sixth Edition 200**8 (addresses low velocity systems, 2.0 IWC maximum static pressure). Both of the cited NAIMA manuals are subject to these limitations:

- Duct board 1 inch thick to 2 inch thick
- Round rigid fibrous glass duct
- Flexible duct
- Operating temperatures below 250°F
- Vertical risers no more than two adjacent stories high

Also refer to Underwriters Laboratory (UL) publication **UL 181 Edition 10 2005** *(Standard for Factory-Made Air Ducts and Air Connectors)*. This publication covers fibrous board and flexible duct materials requirements, and fabrication techniques. Some of the subjects discussed in this standard are:

- Strength test (tension and torsion).
- Load versus deflection tests.
- Structural integrity tests (containment and collapse).
- Duct leakage tests.
- Flame spread and smoke development.
- Burning and flame penetration.
- Corrosion and erosion tests.
- Resistance against mold and mildew.
- Puncture resistance and impact tests.

For information on duct sealing, refer to **UL 181A Edition 3 2005** *(Standard for Closure Systems for use with Rigid Air Ducts)* and to **UL 181B Edition 2 2005** *(Standard for Closure Systems for use with Flexible Air Ducts and Connectors)*. These standards include information that was extracted from UL 181 and information submitted by duct tape manufacturers. Some of the requirements for this standard are:

- Continuous seal on all joints and seams.
- Pressure sensitive tape must conform to UL 181A and 181B.
- Heat-activated tape must conform to UL 181A and 181B.
- Proper application of staples and tape.
- Apply glass fabric and mastic as per manufacturer's instructions (mastic and tape are usually associated with fire- rated assemblies).

The Sheet Metal and Air Conditioning Contractors National Association (SMACNA) also publishes standards that apply to fibrous board duct systems. Refer to the **Fibrous Glass Duct Construction Standard, 7th Edition, 2003.**

The integrity of a duct system, as far as closure is concerned, also depends on:

- Supports and hangers (to prevent stress on seams and joints).
- Reinforcement (to prevent panel sag due to gravity and budge due to pressure differences).
- External loads (to avoid loads that might damage ducts and fittings).
- Interface connections (screws and washers attach fiber board duct to sheet metal flanges and tabs).

Flexible Duct Standards

For information about flexible duct systems, refer to the **ADC Flexible Duct Performance and Installation Standard, 5th edition, 2010**, and the **ADC Flexible Air Duct Test Code FD72-R1, 3rd Edition, 2006, or ASHRAE Standard 210**; as published by the Air Diffusion Council (ADC). These standards cover materials, performance requirements, fabrication techniques, installation procedures and performance testing. Also refer to the **UL 181 Edition 11 2013** (Standard for Factory-Made Air Ducts and Air Connectors), **UL 181A Edition 4 2013** (Standard for Closure Systems for use with Rigid Air Ducts), and **181B Edition 3 2013** (Standard f*or Closure Systems for use with Flexible Air Ducts and Connectors)* and to the **ANSI/SMACNA 006-2006** *(HVAC Duct Construction Standards)*, as published by SMACNA. Some of the guidance that is provided by these standards relates to the following:

- Connections and splices (tape, mastic, fasteners, fittings, and closure methods).
- Excessive length (cut to length, no coiling, bagging, detours or compression).
- Short lengths (do not excessively stretch duct).
- Supports and hangers.
- Change of directions (avoid tight bends that restrict airway).
- Exposure (avoid sunlight, heat or physical damage).

Metal Duct Standards

Metal duct construction standards are published by the American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) **2012 HVAC Systems and Equipment Handbook** (Chapter 19 Duct Construction). See also; **National Fire Protection Association NFPA Standard 90B 2012** (Standard for the Installation of Air Conditioning/Ventilating Systems); and the **IAPMO / ANSI Uniform Mechanical Code 2012.**

These documents provide guidance regarding installation and fabrication of metal ducts, and metal ducts that have duct liner.

Information on external duct insulation (duct wrap) is provided by the ***Commercial and Industrial Insulation Standards, 6th Edition, 2006 publication***, as published by Midwest Insulation Contractors Association (MICA).

The content of these standards pertain to:

- Fabrication (seams, joints, fittings and connections).
- Metal gauge requirements.
- Reinforcement, bracing.
- Closure.
- Insulation attachment.
- Fasteners.
- Adhesives.
- Hangers.
- Supports.

Thermoplastic (PVC) Duct Standards

Information about PVC duct systems is provided by the SMACNA **Thermoplastic Duct (PVC) Construction Manual, 2nd Edition, 1995**.

Duct Insulation Standards

Information on duct insulation materials requirements, thermal performance, installation techniques, and sealing requirements is provided by the SMACNA **ANSI/SMACNA 006-2006** *(HVAC Duct Construction Standards)* and the MICA **Commercial and Industrial Insulation Standards, Sixth Edition** (a manual), Also refer to the NAIMA publication titled **A Guide to Insulated Air Duct Systems, 2003**. The guidance provided by these documents is summarized here:

- Duct Liner
  - ◇ Apply adhesives liner to metal (90% coverage).
  - ◇ Adhesive material performance (ASTM C916).
  - ◇ Mechanical fasteners required.
  - ◇ End caps and facing prevent erosion.
- Duct Wrap
  - ◇ Install with facing exposed.
  - ◇ Cut to recommended “stretched-out” dimension.
  - ◇ Compressed thickness not less than 75 percent.
  - ◇ Do not compress at corner bends.
  - ◇ Apply staples and tape as per MICA standard.
- Rigid Board Insulation
  - ◇ Use pins and clips as per applicable standards.
  - ◇ Seal joints with tape.

◇ Use fabric and mastic to build weatherproof jacket.

Other Documents

The preceding documents are not the only standards that apply to duct systems. Other documents that pertain to duct system materials, construction, installation, testing, safety, and efficiency are listed here:

- Air Conditioning Contractors of America (ACCA) **ANSI/ACCA 5 QI-2010 HVAC Quality Installation Specification.**
- **Uniform Mechanical Code (UMC).**
- International Code Council (ICC) **International Energy Conservation Code (IECC)**, IECC-2006.
- International Code Council (ICC) **International Mechanical Code (IMC)**, IMC-2006.
- International Code Council (ICC) **International Residential Code (IRC)**, IRC-2006.
- **ANSI/ASHRAE/IESNA Standard 90.1-2007** (Energy *Standard for Buildings Except Low-Rise Residential Buildings*) and **ANSI/ASHRAE/IESNA Standard 90.2-2007** (*E*nergy Efficient Design of Low-Rise Residential Buildings).
- SMACNA **HVAC Air Duct Leakage Test Manual, 2nd Edition, 2012.**

National Fire Protection Association **NFPA Standard 90A 2006** *(Standard for the Installation of Air Conditioning/Ventilating Systems)*, and **NFPA Standard 90B 2012** *(Standard for the Installation of Warm Heating/Air Conditioning/Ventilating Systems)*; fire and smoke standards.

- Related items from American Society of Testing and Materials (ASTM):
  - ◇ **ASTM C 612 2004** (Standard Specification for Mineral Fiber Block and Board Thermal Insulation).
  - ◇ **ASTM C 916 1985** *(Standard Specification for Adhesives for Duct Liner Insulation).*
  - ◇ **ASTM C 1071 2005** *(Standard Specification for Fibrous Glass Duct Lining Insulation – Thermal and Sound Absorbing Material).*
  - ◇ **ASTM C 1290 2006** (*Flexible Fibrous Glass Blanket Insulation Used to Externally Insulate HVAC Ducts).*
- Other ASTM and UL standards that deal with materials performance requirements and performance testing; also refer to local codes and regulations.

A6-2 Performance Checklists

The NAIMA document entitled **A Guide to Insulated Air Duct Systems, 2003** includes field inspection checklists that are used to evaluate fabrication materials and workmanship. Abbreviated checklists are summarized here. More detail about any item in these checklists is found in the documents and standards that are published by ADC, MICA, NAIMA, and SMACNA.

Fibrous Glass Duct Construction

(Refer to NAIMA **Fibrous Glass Duct Construction Standard, 7th Edition 2007** for details).

- Is duct system static pressure within specified limits?
- Is the EI rating (475, 800, or 1400) printed on the board?
- Are all sheet metal accessories of galvanized steel?
- Is foil closure tape marked UL 181A-P or UL 181A-H?
- Is all duct stock labeled UL 181?
- Do glass fabric and mastic closures meet UL 181 A-M requirements?
- Are all seams and joints properly stapled or, where stapling flaps are not available, are tape tabs used, minimum of one per side, 12 inches (300mm) on center?
- Does fitting fabrication meet NAIMA standard requirements?
- Does equipment installation meet NAIMA standard requirements?
- Do reinforcement elements meet NAIMA requirements?
- Do hangers and supports meet NAIMA requirements?
- Are ducts free from unrepaired tears or punctures?

Flexible Duct Systems

Refer to ADC Flexible Duct Installation Standard, 4th Edition, 2003 for details.

- Is duct system static pressure within product limits?
- Does insulation R-value meet code requirements?
- Is the UL 181 listing label attached to the flexible duct material?
- Are closure system materials listed and labeled to the UL181 B standard and marked UL181 B-FX (for tape), UL181 B-M (for mastic), or UL181 B-C (for non-metallic mechanical fasteners – straps)?
- Are connections to trunk ducts airtight and insulated?
- Do closures otherwise meet ADC requirements?
- Does flexible duct support meet ADC manual requirements?

- Is the duct system free from sharp bends or kinks?
- Are vertical flexible duct runs correctly supported?

Metal Ducts with Duct Liner

Refer to **NAIMA, A Guide to Insulated Air Duct Systems, 2003** for details.

Has duct liner been installed with air-stream surface printing visible?

- Does duct liner completely cover all inside surfaces of the system, including fittings?
- Is the duct liner free of visible damage (tears, punctures, abrasions)?
- Is duct certified to comply with ASTM C 1071?
- Has duct liner been adhered to sheet metal with adhesive meeting ASTM C 916?
- Are fasteners of the proper type and properly installed perpendicular to sheet metal?
- Are fastener washers cupped or beveled, installed so as not to cut into duct liner?
- Are fasteners spaced correctly for system air velocity?
- Do fastener heads or washers compress duct liner no more than 1/8 inch (3mm)?
- Are leading edges and transverse joints factory-coated, or are they field-coated with adhesive meeting requirements of ASTM C 916?
- Are transverse joints firmly butted, with no gaps or open seams?
- Are all corner joints compressed and overlapped or folded?
- Are longitudinal joints at corners unless duct size or product dimensions prohibit?
- Are top panels of duct liner board supported by side panels?
- Are all leading edges finished with sheet metal nosing if air velocity requires it?
- If installation is two layer, is second layer securely bonded to the first layer?
- Are all sheet metal joints tightly sealed to prevent air leakage?
- Has construction debris been blown or removed from ducts?
- Are sources of potential moisture controlled in order to keep duct liner dry?

Metal Ducts with External Wrap

Refer to **MICA National Commercial & Industrial Insulation Standards Sixth Edition** (a manual), for details.

- Is duct system operating within humidity and temperature range for which duct wrap insulation is rated (40ºF – 250ºF, or 4ºC – 121ºC)?
- Were all circumferential and longitudinal seams or joints in sheet metal duct work tightly sealed before applying duct wrap insulation?
- Is duct wrap certified to comply with ASTM C 1290?
- Is the duct wrap insulation's installed R-value clearly printed on the facing?
- Are all the insulation seams properly stapled with outward-clinching staples every 6 inches (150mm) or tacked using tape across seam?
- When a vapor retarder is required, are seams tightly taped with pressure-sensitive tape or sealed with glass fabric and mastic?
- Was correct stretch-out dimension used so duct wrap is not excessively compressed?
- If rectangular ducts are 24-inches (600 mm) wide or greater, is duct wrap insulation secured to bottom of duct with mechanical fasteners to prevent sagging of insulation?

Metal Ducts with Rigid External Insulation

Refer to **NAIMA A Guide to Insulated Air Duct Systems, 2003** for details.

- Are insulation boards certified to comply with ASTM C 612?
- Were all circumferential and longitudinal seams or joints in sheet metal duct work tightly sealed before installing insulation?
- Are mechanical fasteners spaced on 16" to 18" (400mm to 450mm) centers starting no more than 3" (76mm) from joints?
- Are mechanical fasteners spaced at the correct intervals?
- Where a vapor barrier is required, are seams of insulation boards tightly taped or sealed with glass fabric and mastic?
- Is pressure-sensitive tape at least 3-inches (76mm) wide over all seams and joints?
- Is field jacketing material evenly and uniformly applied, with no gaps or seams?
- Where a vapor barrier is required, are all fasteners tightly sealed with pressure-sensitive tape matching the insulation facing?

A6-3 Recommended Materials Based on Location

Duct fabrication materials and duct insulating materials must be selected for the intended use. This decision is

Recommended Applications for Duct materials							
Duct Location	**Duct Material and Type of Insulation**						
	Duct Board	**Rigid Round Fiber Glass**	**Sheet Metal with Liner (1)**	**Sheet Metal with Wrap**	**Sheet Metal Bare**	**Sheet Metal Rigid Exterior**	**Flexible Insulated**
Attic	X	X	X	X			X
Basement — Unconditioned	X	X	X	X			X (2)
Basement — Conditioned	X	X			X		X (2)
Enclosed Crawlspace	X	X	X	X			X
Open Crawlspace			X			X	
Exterior Wall Cavity or Chase	X	X	X				X
Interior Wall Cavity or Chase	X	X			X		X
Soffit or Ceiling Plenum	X	X	X	X			X
In Conditioned Space	X	X			X		X
Roofor Outdoor Location						X	

Notes:
1) The R-value must be suitable for the application.
2) There must be no possibility of physical damage.

Figure A6-1

based on the location of the duct system because of the potential for damage caused by physical abuse by humans or animals, wetting or condensation. Figure A6-1 (next page) provides guidance on this matter.

Appendix 7 (Informative; not Part of the Standard)

Standard of Care and Continuity

The standard of care recommended by ***Manual D*** is summarized here. These directives affect the accuracy of the airway sizing calculations. They also affect duct system efficiency, ***Manual J*** duct loads, equipment size, installed cost, operating cost and occupant comfort.

- ***Manual D*** calculations may not be relevant if the standard of care is deficient for any design-installation task, or if there is general deficiency in the overall standard of care for the project.
- For continuity, ***Manual D*** airway sizing calculations are based on ***Manual J*** load calculations and on blower data from ***Manual S*** equipment selection procedures.
- Airway sizing calculations are affected by the standard of care used for load calculations and the standard of care used for equipment selection procedures.
- The accuracy of airway sizing calculations depends on proper use of blower data, component pressure drop data, material friction rate data, measured length values, equivalent length values for flexible duct, equivalent length values for fittings and duct sizing tools (charts or slide rules).

A7-1 Statutory Requirements

The duct system must conform to local codes and standards. Codes may cite standards, or local utilities may set standards. The Federal Housing Authority sets standards. Banks and lending agencies may have standards. The corporation or entity that builds tract housing, or builds a custom home may have standards.

A7-2 General *Manual D* Requirements

There are general requirements that apply to all duct materials. These are listed here:

- Duct airway sizes are determined by ***Manual D*** procedures.
- Duct fittings should be relatively efficient (fitting equivalent length typically produces a much larger path pressure drop than the associated straight run length).
- Appendix 3 is used to compare fitting equivalent length values. Select and use the fittings that have smaller equivalent length values.
- The airway sizing calculations are invalid if the fittings that are actually installed in the duct system are not identical to the fittings used for ***Manual D*** calculations (design, then verify).
- Fabrication methods and materials and retrofit methods and materials conform to industry standards and good practice guidelines (see Appendix 6).
- The friction chart or duct slide rule used to size duct airways summarize the performance of the duct material used to fabricate the duct run. This information should be obtained from the manufacturer of the duct material (if available).

  Note: Friction charts or duct slide rules for flexible wire helix duct model the performance of straight duct that has been stretched and allowed to relax to its natural length (the friction chart or duct sizing slide rule is based on the ADC FD72-R1 test code).
- The airway sizing calculations may be invalid if the methods and materials used to fabricate the actual duct system are not the same as the methods and materials used for ***Manual D*** calculations (design, then verify).
- Fabrication methods and materials conform to Underwriters Laboratory documents and National Fire Protection documents (see Appendix 6).

Duct Sealing and Duct Insulation Issues

Airway sizes depend on the output from load calculations that may include heating loads and sensible cooling loads for duct runs. For continuity, the installed duct wall R-value and the actual sealing effort conform to the values used for ***Manual J*** calculations.

- Exposed duct runs should be comprehensively sealed. See Appendix 6 for relevant standards.
  a) There are significant differences in sealing efforts. The ***Manual J*** default (based on ASHRAE Standard 152 research) for an average sealing effort is 0.12 Cfm/SqFt of duct surface area on the supply-side and 0.24 Cfm/SqFt of duct surface area on the return-side. ***Manual J*** also offers two sealing options that are significantly tighter than average and one that is leakier than average.
  b) There are significant differences in the performance of unsealed duct. The ***Manual J*** default (based on ASHRAE Standard 152 research) for an average unsealed duct is 0.35 Cfm/SqFt of duct surface area on the supply-side and 0.70

Cfm/SqFt of duct surface area on the return-side. Duct systems can be leakier than this, but such systems demonstrate no standard of care.

- Currently (2009), generic code templates and some local codes specify R8 insulation for exposed duct runs (R6 for ducts between floor joists). See Appendix 6 for listing of code bodies.
- ***Manual J*** duct load calculations are invalid if the job site duct insulation R-value and sealing effort are not the same as the values used for ***Manual J*** calculations (design, then verify).
- The accuracy of ***Manual J*** calculations affects subsequent ***Manual S*** calculations for installed equipment size.

Balancing

Duct systems should have balancing dampers (for total control of system Cfm, the Cfm delivered to each room or space and the Cfm returned from each return grille). Duct systems should be balanced after they are installed.

A7-3 Requirements for Installing Flexible Duct

Flexible wire helix duct is a special case, because it is flexible. If the length of a section of metal duct or duct board is obviously too long for the span it will not fit. This is no problem for flexible duct because it can be compressed, curved, snaked, looped and coiled. However, this attribute has a price, which is the increase in path pressure drop. Appendix 17 discusses this issue in detail.

Figure A7-1 compares airway sizes for installation practices that can be evaluated by ***Manual D*** procedures. This figure shows that duct diameter must be increased to compensate for installation practices that increase duct run pressure drop. For flexible wire helix duct, this is the standard of care recommended by ***Manual D***:

- Duct sections cut to length, 0% to 4% greater than the span length.
- Duct centerline relatively straight, no significant sag or snaking (2.5 inches sag per 5 feet of span, or less).
- The radius of a bend or turn not be less than the diameter of the airway (R/D = 1.0 or greater).
- No crimping or crushing at any point in a duct run.
- Friction charts and duct slide rules model length and sag conditions specified by the latest version of the ADC FD72-R1 test code (see Section A16-1). Use information provided by the product manufacture, if available.

A7-4 Generic Issues

The installation variables for that are common to all types of duct systems include friction chart information, seam and joint construction and fittings. Differences in the standard of care may affect system design calculations.

Comparative Example: Flexible Duct Diameter (Inches) Vs. Installation Practice [1]

Cfm	Negligible Compression [2]					15% Compression		35% Comp	45% Comp
	No Sag	Long Sag 5 In/10 Ft	Long Sag 10 In/10Ft	Short Sag 5 In/5 Ft	Short Sag 10 In/10 Ft	No Sag	With Sag	With Sag	
100	6.6	6.7	7.0	6.7	7.0	7.6	7.7	8.6	9.4
200	8.5	8.6	9.0	8.6	9.0	9.8	9.9	11.0	12.0
400	11.1	11.3	11.7	11.2	11.8	12.8	13.0	14.4	15.7
800	14.5	14.6	15.3	14.6	15.3	16.6	16.9	18.8	20.5
1,200	16.8	17.0	17.8	17.4	18.0	19.3	19.7	21.8	23.8
1,600	18.6	18.9	19.7	19.3	20.5	21.4	21.8	24.2	26.4
2,000	20.2	20.5	21.4	21.0	22.2	23.2	23.7	26.3	28.6
2,500	22.0	22.3	23.3	22.8	24.8	25.3	25.8	28.6	31.2
3,000	23.7	24.0	25.1	24.6	26.7	27.3	27.8	30.8	33.6

1) Duct diameter must be increased to compensate for pressure drop caused by unacceptable installation practices.
2) Based on ***Manual D*** friction chart for flexible wire helix duct and an a design friction rate of 0.10 IWC / 100 Ft.
3) Compression and Sag: See Appendix 17 for more information.

Figure A7-1

- A generic friction chart from Appendix 2 or the ACCA Duct Sizing Slide Rule may not be exactly correct for a particular product. Use a proprietary friction chart, if available.
- Regarding the use of generic friction charts or equivalent slide rules; airway size depends on many variables besides surface roughness. In addition, there are the issues of rounding to a standard size and air velocity limits. In other words, a minor difference in surface roughness may have a relatively small effect on airway sizes.

  For example, the friction chart friction rate for flexible wire helix duct tends to be twice as large as the metal duct value. When airways are sized with both tools, some flexible duct sizes are the same, some flexible runouts are one inch larger, and some flexible trunks are two inches larger, but this is for a 100 percent difference in surface roughness. Duct size differences are significantly reduced if the variation in friction chart friction rate is 20 percent or less.
- ***Manual D*** procedures use equivalent length (duct length multipliers) for flexible wire helix runs that have excess length with negligible compression, and for runs that have significant compression and sag. This way, existing flexible wire helix friction charts and slide rules (for perfect installation) apply to all Table A16-1 and Table A16-2 scenarios (see Appendix 16).
- Stainless steel, aluminum, clean carbon steel, spiral galvanized steel, smooth plastic and PVC materials have roughness indexes that are less than the value for galvanized metal duct (at 40 joints per 100 feet). If the galvanized metal scale is used for these materials, the error will be small and on the safe side.
- Flexible metal and concrete ducts have roughness indexes that are similar to the roughness index of flexible wire helix duct that has 4% or less excess length and negligible sag. Therefore, a flexible wire helix friction chart, or slide rule (for compression and sag that complies with ***ADC Flexible Duct Installation Standard*** guidance), may be used for these materials.
- The spacing and shape of transverse joints affects the friction rate of the duct run. This is a non-issue for acceptable workmanship, but it could be an issue if joint surfaces significantly protrude into the airway.
- Fitting selection is an extremely important issue. Appendix 3 fittings have documented equivalent lengths. Any listed fitting may be used for design calculation, but efficient fittings are preferred.
- The use of undocumented fittings for ***Manual D*** calculations may produce an invalid solution for one or more airways sizes.
- Installation of fittings that are different than used for the ***Manual D*** calculations may produce an invalid solution for one or more airways sizes.
- There is no ***Manual D*** procedure for estimating the consequences of significant crimping and crushing.
- Leakage at seams, transverse joints, longitudinal joints and fitting connections is not an ***Manual D*** issue, but this is an important ***Manual J*** issue.

A7-5 Procedure Capability vs. Standard of Care

Manual D procedures are based on mathematical models for specific construction scenarios and operational circumstances. This means that calculated airway sizes may be invalid if the actual as-installed condition deviates from the design criteria.

Appendix 8 (Informative; not Part of the Standard)

Residential Air Distribution Systems

No single type of air distribution system is ideal for every dwelling. Sometimes two or more types of air distribution systems are required for a single structure. The performance of an air distribution system should be compatible with the dwelling's structural features and envelope performance characteristics, the local climate and the calculated heating and cooling loads for the various rooms and spaces, of the house and the capability of the blower.

A8-1 Single-Zone Vs. Multi-Zone System

There is a uniform sensation of comfort for the entire living space, when the temperature difference between any two rooms is 2°F or less. However, for most architecture it is difficult, if not impossible, to satisfy this requirement with a single-zone air distribution system. The primary issues are summarized here:

- Glass heat gain varies with exposure direction, month and hour of day. This is especially problematic when rooms are isolated from each other (most rooms and spaces have interior doors).
- Warmer air rises to the upper level, colder air drops to the lower level. This is problematic for split level homes, multistory dwellings and town houses.
- One or more rooms may require a given supply air Cfm for cooling and a significantly different supply air Cfm for heating.
- One or more rooms may be subjected to a load condition that is substantially different than the load condition applied to the primary living space (typically basement rooms, attic rooms and rooms over a garage).
- Construction attributes (insulation, infiltration and thermal mass) for one or more rooms may be significantly different than the attributes of the envelope surrounding the primary living space (typically rooms added to an older, inefficient structure).
- Some floor plans spread out in many directions. The greater the sprawl, the greater the chance that climatological variables will cause temperature control problems in one or more rooms.
- Occupants may desire different thermostat settings for their personal space.

However, the comfort provided by small differences in room to room temperatures is obtained at a cost that may be unacceptable to a builder or owner (zoned comfort systems are more expensive to install than single zone systems). These considerations apply:

- For a single-zone system, the temperature at the thermostat is maintained at the outdoor design conditions for winter and summer (code conditions, or ***Manual J*** Table 1A or Table 1B conditions).
- Room temperatures may be warmer and/or colder than the temperature at the thermostat.
- A room-to-room temperature difference that exceeds 2°F by a few degrees is not a health and safety issue for most people.
- Room to room temperature difference is a business issue. Comfort system performance satisfies negotiated performance requirements at the quoted price.
- Performance-price issues settled prior to designing the comfort system. This written agreement should be part of the contract document.

Performance Standard for Room Temperatures

Manual D concerns itself with duct system design, so it does not provide a performance standard for room-to room temperature difference. ***Manual D*** defers to guidance provided by ACCA ***Manual Zr***.

- ***Manual Zr*** is an ANSI stanadrd that provides "good practice" guidance for residential zoning.
- ***Manual Zr***, Figure 1-2 summarizes temperature and humidity goals for residentail single-zone and milti-zone systems.

Guidance for Designing Zoned Systems

Comprehensive application of principles, concepts and information provided by ACCA manuals is a prerequisite for zoned comfort system design. Use the current versions of these design tools.

- ***Manual RS*** provides general information and commentary pertaining to zoning issues.
- The full version of ***Manual J*** evaluates and quantifies fenestration cooling load vs. time of day behavior for the block load calculation, for room/space load calculations, and for zone load calculations. This information is used to make zoning decisions.

- ***Manual S*** equipment sizing procedures apply to single zone systems and multizone systems (the ***Manual J*** block load for equipment sizing is the same for both types of systems).
- ***Manual Zr*** provides comprehensive design guidance for air-zoned systems (and for zoning with duct-less equipment).
- ***Manual T*** and ***Manual D*** provide comprehensive guidance for air distribution system design.
- ACCA revises and updates its manuals on a continuous basis. The content of existing manuals is coordinated with the content of new manuals. Refer to the ACCA website (www.ACCA.org) for information about current products.

A8-2 Air Distribution System Performance Depends on Air Flow and Air Mixing

A deficiency in the air distribution system (blower, duct runs, supply outlets and returns) may produce local comfort problems in any dwelling, or stratification problems in multilevel dwellings. This means that the air distribution system must be designed to deliver the correct air flow to each supply air outlet; that each outlet must be carefully sized and located; and that the return air system must be designed to assure there is a low-resistance return air path between every room and space and the return-side of the air handler. (At least one return air opening is required for each level and one return air opening or transfer grille is required for every room that can be isolated by an interior door.)

Most practitioners appreciate the significance of the supply and return air paths and flows, but some underestimate the importance of the supply air outlets. For example, if the supply outlets are too large, supply air will not be projected out into the room and will not mix with room air.

During the cooling season, supply air tends to drop to the floor. At the upper level of a dwelling, supply air may 'fall' out of the ceiling outlets, accumulate on the floor and drain down the stairwell or fall off the balcony. When this cascade of cool air reaches the floor of the lower level, it adds to the puddle of cold air created by the over-sized outlets that serve the lower level.

When this happens, room air, which is relatively warm, floats to the upper level and stratifies near the ceiling. This layer of warm air can penetrate the living space if the mixing action of the supply outlets is deficient. The net effect is that the upper level is too warm and the lower level is too cold. Furthermore, the same problem is experienced during the heating season. In this case, over-sized supply air outlets and inadequate mixing causes warm supply air to stratify at the ceiling of the upper level.

A8-3 Continuous Blower Operation

In theory, uniform room temperatures could be provided by careful adjustment of the branch duct balancing dampers and continuous blower operation. This way, each room would receive the correct amount of heat (or cooling) when the central heating (or cooling) unit is operating; and when the unit is off, the air in the various rooms will be continually mixed by the action of the blower. But, this strategy may not provide the desired result.

- A fixed branch damper setting will not balance the heating (or cooling) capacity of the supply air with the room load because the room load continually changes as the outdoor temperature, solar gains and internal loads vary.
- The off-cycle blending action provided by continuous blower operation will not be effective unless the performance of the air distribution system is flawless. This means that an ample amount of air must be continuously extracted from every room and replaced with an equivalent amount of recirculated air. Furthermore, the recirculated air has to be thoroughly mixed with room air (by the action of the supply outlets).

A8-4 Central and Distributed Air Handling Systems

Traditionally, dwellings are equipped with air distribution systems that feature a single, centrally located air handler. In most cases there is only one system, but some dwellings have two or more systems. Multiple single-zone systems are used to solve:

- Zoning problems produced by two or more levels.
- Zoning and installation problems associated with heating and cooling very large homes.

Regardless of how many central systems are used, each air handler may have a duct system that delivers supply air to a suite of rooms. Figure A8-2 (next page) summarizes the performance characteristics of a central single-zone system.

A central air handling system can provide zone control if it is equipped with zone dampers, special controls and certain mechanical components. Most dwellings require only one of these systems, but two or more systems are appropriate if the dwelling is exceptionally large or architecturally complicated. The performance characteristics of central multi-zone systems are summarized by Figure A8-3 (next page).

One of the disadvantages of central air handling systems is that the trunk ducts take up a lot of space that might otherwise be used as living space. In order to solve this problem, some builders and equipment manufacturers are experimenting with integrated structural-system air-system designs that feature thermally efficient envelopes, open truss construction and distributed air-handling equipment. Figure A8-4 (next page) provides an example of a distributed air handling system. This system has these characteristics:

- The floor plan is divided into zones and one air handler is provided for each zone. These air handlers are very small, only a few hundred Cfm flows through each unit. Because they are small, the air handlers can be located near the rooms and spaces they serve.
- Each air handler has its own duct system; there are no large trunk sections and the duct runs are short.
- The heating and cooling loads are small (due to efficient construction techniques), so duct sizes are small (6 inches or less.)
- When there is more than one level, ducts are routed through the floor joists (open trusses are used above the first floor ceiling).
- A system can be a constant-volume system or an air-zoned system.
- Each air handler is equipped with a heating coil (electric or hot water) and a cooling coil (refrigerant or chilled water).
- Remote fuel conversion equipment, which may be an ordinary water heater and a simple water cooler, provides hot and cold water for the water coils.

A8-5 Classification of Supply Duct Systems

Supply duct systems are characterized by the duct geometry, the location of the supply outlets and the duct material. Information about each of these characteristics is required to completely describe a supply duct system.

Supply Duct Geometry

Supply air duct runs have three basic configurations; a trunk and branch configuration, a radial configuration or a perimeter loop configuration. Each of these configurations have different spatial requirements, performance characteristics, and installation cost. Figure A8-5 (next page) shows the geometry of the basic configurations. More information about the advantages and disadvantages of various configurations is provided in Sections A8-8, A8-9 and A8-10.

Supply Air Outlet Location

Depending on the location of the supply outlets, an air distribution system could be described as a perimeter system, a ceiling supply system, or an inside wall supply system. Some general comments about the strengths and weakness of these systems are made below. Refer to ACCA *Manual T* for detailed information about the performance characteristics of supply air outlets.

Central Single-Zone System

- No control over airflow as room loads vary.
- Central equipment is controlled by a single thermostat.
- Blower Cfm could be constant.
- Blower Cfm could vary (mult-speed or variable-speed blower)

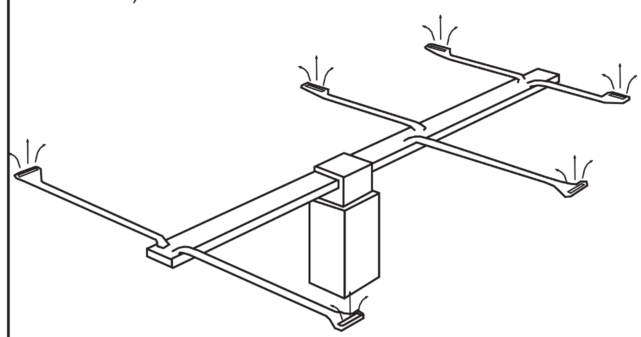


Figure A8-2

Central Air-Zoned System

- Airflow is controlled on a zone-by-zone basis (control dampers are installed in the secondary trunk ducts and/or branch ducts and controlled by room or zone thermostats.)
- The central equipment is controlled by a microprocessor that monitors the zone thermostats.
- The blower Cfm could be constant, but a bypass damper may be required to maintain the airflow through primary equipment.
- The blower Cfm could vary if blower and compressor RPM are continually adjusted.

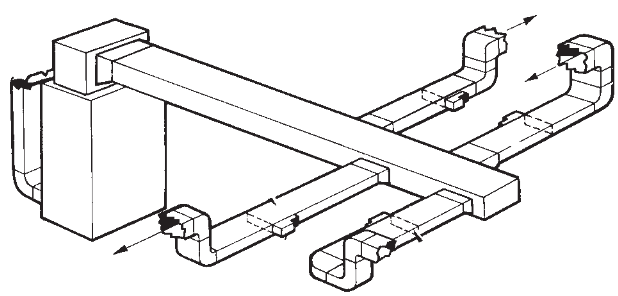


Figure A8-3

Perimeter Systems

Perimeter systems blanket portions of the exterior walls with supply air. This is accomplished by using floor, baseboard, or low sidewall, outlets that discharge supply air straight up the wall. If the supply outlets are sized

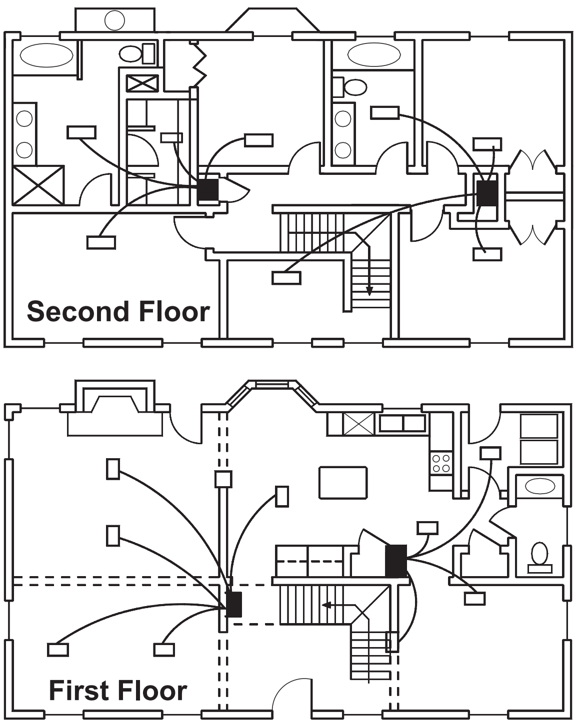


Figure A8-4

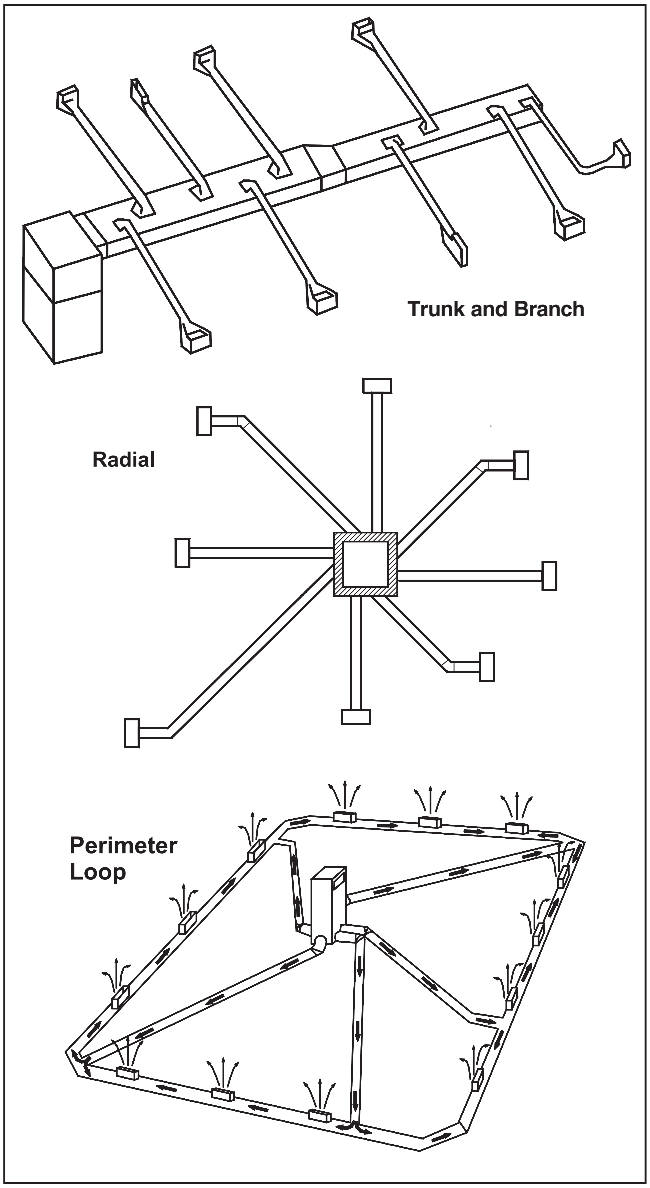


Figure A8-5

correctly, the discharge pattern will extend up to the ceiling. (Never use floor or low side-wall outlets that blow air into the interior of the room. Discharge should be vertical, parallel to the wall.)

It also is possible to use ceiling outlets that discharge air straight down the wall, but this arrangement is more suited for heating rooms that cannot be served by a below-the-floor duct system. (Discharging cold air straight down a wall will cause the air to stratify along the floor. A horizontal, parallel to the ceiling, discharge is preferred for cooling.)

Traditionally, perimeter systems have been used for cold climate dwellings because they provide more comfort at the floor level than the two other types of systems. However, ceiling or inside wall systems can be used in a cold climate, if the dwelling has a thermally efficient envelope and a heated basement. But, when slab construction or exposed floors are involved, perimeter systems are preferred, even if the envelope is well insulated.

Ceiling Supply Systems

Ceiling supply outlets should discharge air parallel to the ceiling. If ceiling outlets are sized correctly the discharge pattern will extend to the walls. (Never use outlets that blow air down into the interior of the room.)

Ceiling systems provide optimum performance for cooling, so they are commonly used in warm climate dwellings. (Cold floor problems may be experienced during the heating season when ceiling outlets are installed in a room that has a slab floor or an exposed floor.)

High Inside Wall Supply Systems

High sidewall supply outlets should discharge air parallel to the ceiling toward the outside wall. If the outlets are sized correctly, the discharge pattern will extend to the opposite wall and high velocity air will not drop into the occupied zone. (Excessive drop during the cooling season is a common problem for high sidewall outlets.)

High sidewall outlets perform best during the cooling mode, so they are more suitable for warm climate dwellings. (Cold floor problems could be experienced during the heating season when high sidewall outlets are installed in a room that has a slab floor or an exposed floor.)

Duct Material

Duct runs are fabricated from different types of materials. Each type has advantages and disadvantages. Selection of the duct material will normally depend upon the location of duct work. In some instances it may be determined by the local building code.

Systems placed in attics, basements, or crawlspaces offer the most options regarding materials. On the other hand, when ducts are installed underground, or encased in concrete, the practitioner is obligated to use materials that are certified for these applications. A discussion of the common types of duct materials as it pertains to their range of application is provided here:

Stainless Steel

Stainless steel is used for ducts that will be subjected to moisture. Applications include exhaust ducts for spaces that have hot tub or pool, for shower room exhaust, and for ducts that are exposed to the weather. This material is not combustible or subject to corrosion. Stainless steel ducts are slightly smoother than galvanized steel ducts. The pressure drop across a section of stainless steel duct will be marginally less than the pressure drop across a similar section of galvanized steel duct.

Galvanized Steel

Galvanized steel is widely used for supply, return and exhaust ducts. This material is not combustible. Galvanized steel is subject to corrosion by concrete, so if it is embedded in concrete, it must be protected by a suitable coating such as (but not limited to) plastic, asphalt or bituminous mastic. Galvanized steel duct sections will float when the concrete is being poured. Painting is recommended when galvanized steel ducts are installed outdoors.

Plastic-coated Steel

Plastic-coated steel ducts may be used for below-grade duct systems and for applications that involve moisture. This material is not combustible or subject to corrosion by concrete, but the duct sections will float when covered by concrete. Plastic-coated ducts are smoother than galvanized steel ducts. The pressure drop across a section of plastic-coated duct will be a little less than the pressure drop across a similar section of galvanized steel duct. Plastic-coated steel ducts are not be used when temperatures exceed 200°F.

Aluminum

Aluminum may be used for ducts that are subject to moisture or exposed to the weather. This material is not combustible. Bare aluminum ducts are not suited for exposure to chlorine or lime. Since aluminum is subject to corrosion by concrete, it should not be embedded in concrete unless it is protected by a suitable coating such as (but not limited to) plastic, asphalt or bituminous mastic. Aluminum duct sections will float when the concrete is being poured. Aluminum ducts are smoother than galvanized steel ducts. The pressure drop across a section of aluminum duct will be slightly less than the pressure drop across a similar section of galvanized duct.

Plastic Polyvinyl Chloride (PVC)

Plastic polyvinyl chloride (PVC) may be used for ducts that are subject to moisture. Applications include hot tub, pool and shower exhaust ducts, and ducts that are exposed to the weather. PVC ducts are not subject to rust and corrosion, so they can be embedded in a concrete slab or used for below-grade duct systems. Plastic duct sections will float when the concrete is being poured. Plastic ducts are smoother than galvanized steel ducts. The pressure drop across a section of plastic duct will be slightly less than the pressure drop across a similar section of galvanized steel duct.

Fiberglass Reinforced Plastic

Fiberglass reinforced plastic duct may be used for air conveying ducts that are subject to moisture. Applications include hot tub, pool and shower exhaust ducts, or ducts that are exposed to the weather. This material is also used for below-grade duct systems. Fiberglass reinforced plastic duct sections will float when the concrete is being poured. They are rougher than galvanized steel ducts. The pressure drop across a section of fiberglass reinforced plastic duct will be larger than the pressure drop across a similar section of galvanized steel duct.

Rigid Fibrous Glass

Rigid fibrous glass (duct board) is used for HVAC duct systems that will not be subjected to physical abuse or moisture. Rigid fibrous glass should not be embedded in concrete or buried in the ground. Such ducts offer the advantages of sound attenuation and insulation and are rougher than galvanized steel ducts. The pressure drop across a section of fibrous glass duct will be greater than the pressure drop across a similar section of galvanized steel duct. Rectangular shapes are normally fabricated from duct board. Prefabricated rigid round shapes also are available. Fibrous glass ducts are not recommended for use with kitchen hoods and are not be used when temperatures exceed 250°F.

Fibrous Glass Duct Liner

Fibrous glass duct liner offers the advantages of sound attenuation and insulation. Lined ducts are rougher than galvanized steel ducts. The pressure drop across a section of lined duct will be larger than the pressure drop across a similar section of galvanized steel duct. Lined ducts are not be used when temperatures exceed 250°F.

Flexible Wire Helix Ducts

Flexible wire helix material may be used for ducts that will not be subjected to physical abuse, weather or moisture. Flexible wire helix ducts should not be embedded in concrete or buried in the ground. Such duct systems offer the advantages of sound attenuation and ease of installation (as far as airway fabrication is concerned). These ducts also offer the advantage of built-in insulation when they are wrapped with an insulating jacket. Flexible wire helix ducts behave as if the interior is rougher than galvanized steel ducts and fibrous glass ducts (see Section 4-2). The pressure drop across a section of flexible duct is considerably larger than the pressure drop across a similar section of galvanized steel duct (this depends on the amount of excess length produced by installation practices). Flexible helix ducts are not be used when temperatures exceed 250°F.

Flexible Metal Ducts

Flexible metal ducts may be used for residential systems, but wire helix ducts are more common. Flexible metal ducts are commonly fabricated from aluminum or stainless steel and they can be purchased with an insulating jacket. This material is not combustible. Flexible metal ducts are rougher than galvanized steel ducts. The pressure drop across a section of flexible metal duct will be larger than the pressure drop across a similar section of galvanized steel duct.

Concrete

Concrete ducts are used for underground duct systems. This material is not combustible and does not float. Concrete ducts are rougher than galvanized steel ducts. The pressure drop across a section of concrete duct will be greater than the pressure drop across a similar section of galvanized steel duct.

Asbestos Cement

Asbestos cement material has been used for below-grade duct systems. Use of this material was discontinued because asbestos is considered to be a health hazard.

Materials and Installation Standards

Material and installation requirements may be defined by local codes and regulations; also refer to the standards that are published by the Air Conditioning Contractors of America (ACCA), National Fire Protection Association (NFPA); the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE); the Sheet Metal and Air Conditioning Contractors National Association (SMACNA); the North American Insulation Manufacturers Association (NAIMA); the International Code Council (ICC); the U.S. Department of Housing and Urban Development (HUD); and the Federal Housing Administration (FHA).

A8-6 Classification of Return Duct Systems

Return duct systems are primary characterized by the number of the return openings. Return inlet location, duct run geometry and duct material, are secondary attributes that describe a return duct system.

Number of Return Openings

Return duct systems are classified as a single, central return system; a multiple return system; or as a system that has a return in every room. Regardless of which type of return duct system is used, a low-resistance return air path must connect every room to the air handler. A system that features a return in every room automatically satisfies this requirement.

If a single return system or a multiple return system is used, provide a low-resistance path between every isolated room and the closest return air opening. This path may be established by using transfer grilles or door grilles. Refer to Appendix 3, fitting Group 14 the recommended design procedure.

Return Inlet Location

Return air duct systems also are characterized by the location of the return air openings. If all of the return openings are installed in the ceilings or located high on the walls, the system is called a "high return system." If all of the return openings are installed in the floor or in a low sidewall position, the system is called a "low return system."

Since the return inlet location (high or low) has a negligible effect on the air motion within the room (refer to ACCA ***Manual T***, Section 7), return openings may be placed at positions that are compatible with the location of the equipment and the duct runs. High return systems are typically used for rooms that have air handling equipment above the ceiling and low returns are the logical choice for rooms that have air handling equipment below the floor.

*The air motion within the occupied zone depends on the performance of the supply air grille or register. If this component is sized correctly, the jet of discharge air entrains a large amount of room air as it transitions to a secondary air pattern. The amount of air in the secondary air pattern is about 10 to 20 times more than the supply air Cfm value (see **Manual T**, Section 2-4.)*

Even more mixing occurs as the secondary air exchanges its momentum with the room air. (This mixing action occurs outside of the occupied zone, which means it has to occur near a wall or ceiling.) Ultimately, all of the air in the occupied zone is put in motion and there will be no drafts or stratified air in the occupied *zone. (Stratification does not cause discomfort if it occurs outside of the occupied zone, near the ceiling, for example.)*

Return Duct Geometry

A return duct system could have a trunk and branch configuration, a radial configuration, or a perimeter loop configuration. Return-side geometry is described by the number and location of the return inlets and the spatial requirements of the supply air duct system.

Trunk and branch return systems are very common because they provide an effective way to connect the air handler with multiple returns located near the core of the floor plan. Radial duct systems also could be used if there is no interference with the supply duct runs. Returns are not usually located around the perimeter of the floor plan, but if they were, a radial system or a loop system could be used to connect the returns to the air handler.

Sometimes the return duct system consists of a simple stub duct or elbow, and in some cases there is no return duct. For these designs, the air handler (and its return air opening) is either installed close to a central return grille or is located within the living space, perhaps in a utility closet.

- If a central return is used, provide a low-resistance return path between every room and the central return.
- Equipment noise can be a problem when the equipment is located within, or close to, the living space.

Return Duct Material

Return ducts can be made out of any of the materials that are used for supply ducts. Information about these materials is provided above in Section A8-5.

A8-7 System Selection

A particular set of duct system attributes will not arbitrarily apply to all dwellings, because spatial relationships and performance requirements vary on a case by case basis. At the beginning of a project, the practitioner must identify and prioritize relevant issues and select a distribution system that is compatible with local code, the attributes of the structure, and the comfort level specified by the contract document.

The four primary factors that affect the air distribution system selection process are zoning requirements, envelope construction details, climate, and installation cost. Other factors include duct losses, building codes, energy codes, and noise levels.

Zoning Requirements

Some dwellings must have zone control (considering ***Manual Zr*** guidance) and many dwellings would benefit from zone control. As explained in Section A8-1 (and ***Manual Zr***), zoning decisions depend on the floor plan, the construction details, and on contractual preferences specified by the owner or builder. If zoning is authorized by the owner or builder, it may be provided by installing one or more of the following types of ducted air distribution systems:

- Multiple, single zone or constant Cfm systems.
- A central zone damper system.
- Distributed air-handling units equipped with one change-over water coil, chilled and hot water coils, refrigerant coil and electric heating coil, or chilled water coil and electric heating coil.

Envelope Construction Features

Envelope construction features affect zoning requirements, and they influence decisions related to supply air outlet positions, return air inlet locations, equipment placement, and duct run locations. As far as air distribution hardware and duct placement are concerned, the most important construction features are the type of foundation, the type of roof (attic or no attic) and the number of floors or levels.

Note: Because of energy use, operating cost and undesirable effects on comfort system performance, it is suggested that duct runs should be installed within the conditioned space. In this regard, home builders and HVAC contractors are encouraged to cooperate on designs that optimize the overall performance of the architecture and the comfort system.

Slab Construction

For a structure on a ground slab, air handling equipment may be located within the dwelling, in the attic, or outdoors. If the blower is located in the dwelling, or in an outdoor cabinet, it may feed air to ducts embedded in the slab, or to ducts located in the attic or above a drop ceiling. In any case:

- A perimeter supply air system is compatible with ducts below the slab.
- Ceiling outlets are compatible with attic ducts.
- High inside wall outlets may be used for ducts in a drop ceiling cavity. (If attic ducts drop to high sidewall outlets, the drops have to penetrate the plate at the top of the partition walls.)

Depending on where the air handling equipment is located, air may be returned through:

- A central return located at or near the air handler (transfer grilles can provide a low-resistance return path from every isolated room).
- Air may be returned through multiple returns located in the ceiling (providing multiple return duct runs can be routed back to the unit).
- Rooms that have an interior door and no local return grille require transfer grilles.

Crawlspace Construction

For a structure over a crawlspace, air handling equipment may be located within the dwelling, in the crawlspace, in the attic, or outdoors. If the blower is located in the dwelling or in an outdoor cabinet, it may feed air to crawlspace ducts, or to ducts located in the attic or above a drop ceiling. In any case:

- A perimeter supply air system is compatible with ducts below the slab.
- Ceiling outlets are compatible with attic ducts.
- High inside wall outlets may be used for ducts in a drop ceiling cavity. (If attic ducts feed drop to high sidewall outlets, the drops have to penetrate the plate at the top of the partition walls.)

Depending on where the air handling equipment is located, air may be returned through:

- A central return located at or near the air handler (if transfer grilles provide a low-resistance return path from every isolated room).
- Air may be returned through multiple returns located in the ceiling (providing multiple return duct runs can be routed back to the unit).
- Rooms that have an interior door and no local return grilles, require transfer grilles.

Basement Construction

When the structure has a basement, blower equipment and duct runs are usually located in the basement; and a perimeter supply air system is normally used to deliver air to first floor rooms. One or more sidewall outlets (installed in the supply trunk) may crudely project air into the basement, providing the basement is not a living space. If the basement is a living space, air may be supplied through ceiling outlets, high sidewall outlets, or through low outlets positioned around the perimeter of the basement.

- Supply air outlets must have appropriate throw and drop for the geometry of the space.
- If low perimeter outlets are used, in the basement they may be fed by duct runs that drop from the basement ceiling or by a duct system embedded in the basement floor. (Low supply outlets are preferred for basement living spaces because they provide more comfort.)

Low returns may be used to route first floor air back to the air handler, and high returns may be used to route the basement air back to the air handler.

Multi-Story and Split-Level Dwellings

Multiple systems solve zoning problems and duct routing problems (when it is too difficult to route risers and drops from one level to another). If multiple systems are used, a multi-story or split-level dwelling is treated as a combination of the types of construction that are listed above. For example, if the dwelling has two levels on framing, a level on a slab, and a basement; one system could be installed in the basement, and a second system could be installed in the attic. The basement system could serve the first level, the slab level and the basement. The attic system could serve the second floor.

Climate

The local climate is considered when locating supply air outlets. Perimeter supply air systems furnish superior heating performance and ceiling supply air systems provide optimum cooling performance. It follows that a perimeter system is preferred for a cold climate and a ceiling system is preferred for a hot climate. For some climates, heating and cooling are equally important. In this case, the perimeter system is preferred for most dwellings because the cooling performance of a perimeter system is better than the heating performance of the ceiling system.

Even though the perimeter system offers the best heating performance, it is not absolutely necessary to install a perimeter system in a cold climate dwelling. Ceiling or high sidewall systems may provide acceptable comfort if the space is tight and well insulated, if it does not have unusually large glass areas, and if it is above a heated basement. However, a perimeter system is recommended for rooms that have a slab floor, and for rooms that are located above an unheated crawlspace.

The local climate also dictates the envelope insulation requirements, which are important because insulation details affect the occupant's perception of comfort system performance and the air distribution system.

- When the dwelling is properly insulated, the air distribution system performance and the comfort of the occupant is enhanced.
- When insulation is inadequate, performance and comfort are degraded.
- Local codes and standards determine minimum duct insulation and duct sealing requirements.
- Current versions of international, national and regional codes and industry standards provide

current consensus guidance pertaining to duct insulation and duct sealing requirements (see Appendix 6).

Installation Costs

Installation costs are minimized by keeping duct runs as short and simple as possible, by using as few fittings as possible, by using as few returns as feasible, and by using materials that are inexpensive and easy to install. In this regard, air outlet locations and return air inlet positions significantly effect the installed cost of a duct system. For example, the installation costs are relatively low for radial duct runs that connect ceiling outlets to an air handler located above a central ceiling return.

Man-hours and workmanship may be the primary factors affecting installation costs. When comparing cost for different duct materials, compare the cost for full compliance with methods and material standards, installation codes and manufacturer recommendations.

Duct Losses

Duct conduction and leakage losses affect comfort, energy use, and operating costs. Ideally, ducts should be installed inside the conditioned space. When ducts are installed in an unconditioned space, they should be sealed and insulated, and in some cases, a vapor retarder may be required. Comments about duct losses, as they pertain to the location of a duct run, are provided below. More information about insulating and sealing exposed ducts is provided in Appendix 10.

Slab Construction

Radial and perimeter-loop duct systems embedded in a concrete slab lose heat through the edge of the slab. However, a radial system has less heat loss than a perimeter loop system. Moisture is another problem. When ducts are installed below-grade, moisture should not accumulate inside of the duct system because this affects indoor humidity and the quality of the indoor air (possible mold, mildew and related health problems). Moisture should not be allowed to saturate the duct bedding material because this increases heat transfer through the duct walls.

Open Crawlspace

Ducts installed in an open crawlspace are subject to outdoor temperature and humidity conditions so they should be sealed and insulated. Duct insulation requires a vapor retarding jacket if the temperature of the outer surface of the duct material can be lower than the dew point temperature of the outdoor air. (A vapor retarder is required for climates that routinely have outdoor dew point temperatures in excess of 60°F during the cooling season.)

Enclosed Crawlspace

Ducts installed in an enclosed crawlspace are subject to a temperature and humidity condition that falls somewhere between the outdoor condition and the indoor condition. If the crawlspace walls are tight and well insulated, crawlspace temperature may be close to the room temperature, probably within 20°F or less. Therefore, the amount of duct insulation is less than the requirement for an open crawlspace. In any case, duct runs should be sealed. (A vapor retarder is required for climates that routinely have outdoor dew point temperatures in excess of 60°F during the cooling season.)

Basements

Basements should be tight and well insulated, especially if used as a living space and maintained at the indoor design condition during the heating and cooling seasons. Duct insulation is not required in this case, but duct runs should be sealed. If the basement is not used as a living space it may be heated, but not cooled. In this case, duct insulation is recommended because it will reduce the duct losses during the cooling season.

Attics

Attic ducts are subject to an ambient temperature and humidity condition that may be equal to (during winter), or more severe (during summer) than, the outdoor air condition. Therefore, attic duct runs must be sealed and insulated. Duct insulation requires a vapor retarding jacket if the temperature of the outer surface of the duct material can be lower than the dew point temperature of the outdoor air. (A vapor retarder is required for climates that routinely have outdoor dew point temperatures that exceed 60°F during the cooling season.)

Drop Ceilings

Ducts installed in a drop ceiling are subject to a temperature and humidity condition that falls somewhere between the outdoor condition and indoor condition. If the drop ceiling cavity is tight, well insulated and protected by a vapor retarder, the ceiling cavity condition will be similar to the indoor design condition. In this case, duct insulation is not required, but the duct runs must be sealed. If the ceiling cavity condition is expected to be substantially different than the indoor design condition, the duct runs must be insulated, sealed, and wrapped with a vapor retardant jacket if the ceiling cavity has no vapor retarder.

Ducts in Outside Wall Stud Spaces

Risers and drops installed in an outside wall stud space are subject to a temperature and humidity condition that falls somewhere between the outdoor condition and the indoor condition. These ducts must be sealed and insulated. Duct insulation requires a vapor retarding jacket if the temperature of the outer surface of the duct material can be lower than the dew point temperature of the outdoor air. (A vapor retarder is required for climates that routinely have outdoor dew point temperatures in excess of 60°F during the cooling season.)

Note: Avoid routing ducts through outside wall stud spaces; this minimizes issues that may be difficult and costly to resolve. For example, the size of the stud space has to be compatible with the size of the duct, which is not known until the architect or builder finalizes the wall design. Additional stud space is required for adequate insulation, and there is the question of whether the insulation should be placed on the outside wall or on the duct. The appropriate location for risers and drops is in dedicated chases or wall cavities that are within the conditioned space.

Noise

Primary equipment is preferably located outside the conditioned space. Excessive blower noise, noise generated in the duct system, and noise generated by dampers, supply air outlets, or return air grilles, is not acceptable. Good practice methods and procedures maintain an acceptable noise level in the conditioned space (see Appendix 13).

Codes

Building codes, energy codes, fire codes, insurance regulations and utility regulations, may require, or disallow, certain envelope construction features, and they might mandate or prohibit certain air system installation procedures and practices. Duct placement, duct materials, duct insulation, duct sealing and vapor barriers are some of the things that might be codified. Make sure that the installation is in compliance with all of the codes and regulations that apply at the site.

A8-8 Trunk and Branch Systems

The various types of trunk and branch systems are identified by more common names — an "extended plenum" system, or a "reducing plenum" system, for example. The flexible duct system also is classified as a trunk and branch system. Information about the various types of trunk and branch systems is provided here:

Extended Plenum System

The extended plenum system is the most common residential duct system. This arrangement consists of a relatively large trunk duct that is essentially an elongated plenum, and a number of small branch ducts that deliver supply air to each room (see Figures A8-6 and A8-7). This system is easy to fabricate and inexpensive to install. The trunk run is just a straight section of rectangular or round duct; and simple fittings and boots connect branch ducts and supply air outlets.

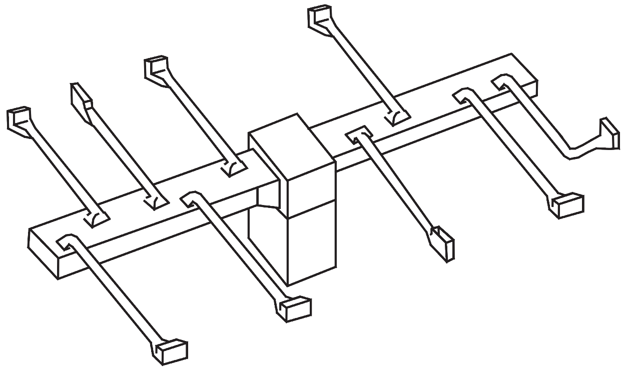


Figure A8-6

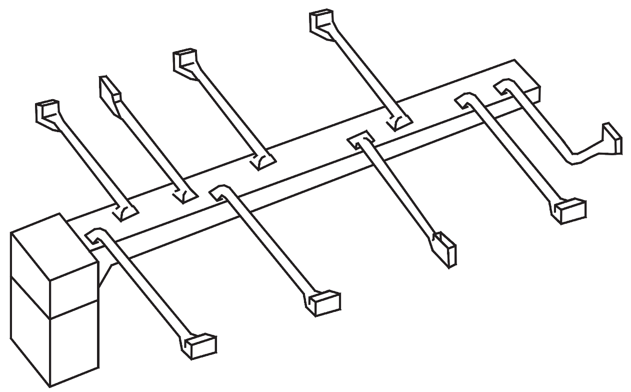


Figure A8-7

The air handler may be centrally located with respect to the floor plan of the dwelling (if possible and practical). In this case, two plenums would extend across the length of the dwelling, as indicated by Figure A8-6. This configuration tends to reduce the equivalent length of the branch take-off fitting that are close to the air handler, but this may be a minor issue as far as the total effective length of the duct system is concerned. (Minimize effective length and airway sizes by using plenum, branch take-off and boot fittings that have a small equivalent length.)

Quite often, for one reason or another, the air handler cannot be centrally located with respect to the floor plan. In this case, one long plenum may extend cross the length of the dwelling, as illustrated by Figure A8-7. This trunk geometry produces large differences in the lengths of the various supply paths; and these differences make it difficult to turn the air into branch ducts near the air handler.

Reducing Plenum System

A reducing plenum design, as illustrated by Figure A8-8, (next page) improves the performance of a long supply plenum because the reduction in trunk airway size tends to

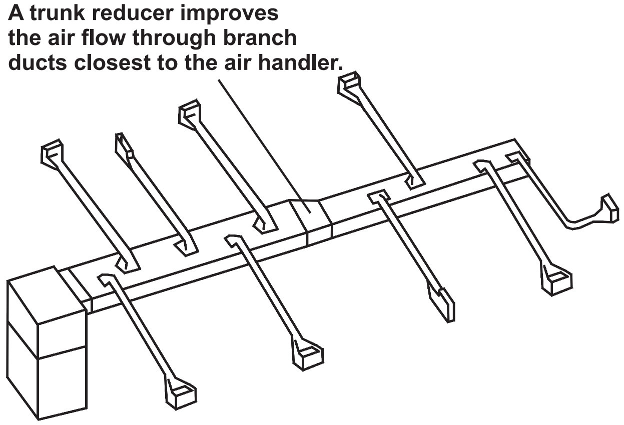


Figure A8-8

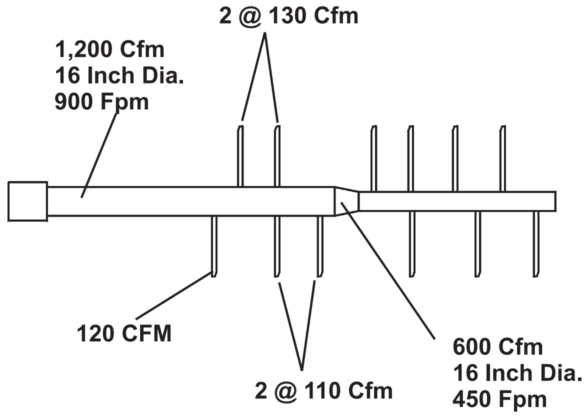


Figure A8-9

equalize branch duct takeoff fitting losses, which helps turn the air into branches near the air handler.

Historically, duct design manuals (including the first edition of ***Manual D***) recommended that a plenum reduction be located at a point that is about 24 feet downstream from the air handler. However, this recommendation is arbitrary because it is based on the assumption that a substantial number of branch ducts are installed in the first 24 feet of the plenum, and that a similar number of branch ducts are downstream from the reduction. Since this is not generally true, it is better to use air velocity to determine the location of the trunk reducer.

The size of a plenum airway can be reduced if the air velocity just upstream from a branch duct slows to about 50 percent of the initial velocity.

Figure A8-9 shows how to apply this rule. In this case, the reduction after the fifth branch duct takeoff fitting is appropriate because the air velocity has dropped from 900 Fpm to 450 Fpm.

- Velocity at the entrance to the plenum is at, or close to, the maximum recommended velocity.
- For a constant sectional area, Cfm and velocity along the plenum decrease at each branch take-off.
- The sectional area of the downstream plenum is sized for the entrance Cfm and the design friction rate value.

Air velocity at any point along a duct run depends on the Cfm flowing through the duct at that point and on the cross sectional area of the duct at that point. This velocity (Fpm) may be determined by using the ACCA duct sizing slide rule or by dividing the Cfm value by the sectional area (SqFt).

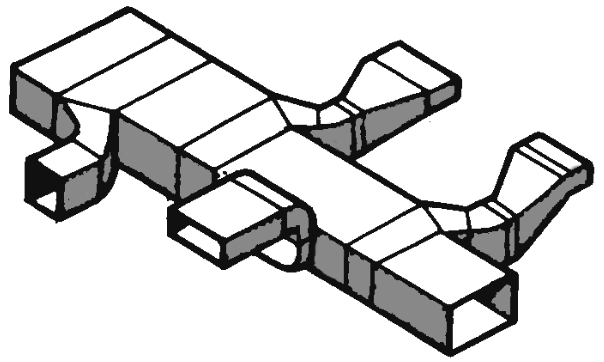


Figure A8-10

Reducing Trunk Systems

Figure A8-10 provides an example of a reducing trunk system. This design is similar to an extended plenum design, except the sectional area of the trunk is reduced after every branch take-off. This geometry is produced by the equal friction method (which is the most common procedure for designing commercial duct systems). This design uses less material than an extended plenum system, but fabrication time increases because each trunk section is a different size. For residential duct systems, it is usually less expensive to use an extended plenum design or a reducing plenum design.

Primary and Secondary Trunk Systems

Some trunk and branch systems have a primary trunk and two or more secondary trunks, as illustrated by Figure A8-11 (next page). This design may be used in a home that spreads out in two or more directions. For this geometry, the tee at the end of the main trunk is equivalent to the reducer fitting in a reducing plenum system.

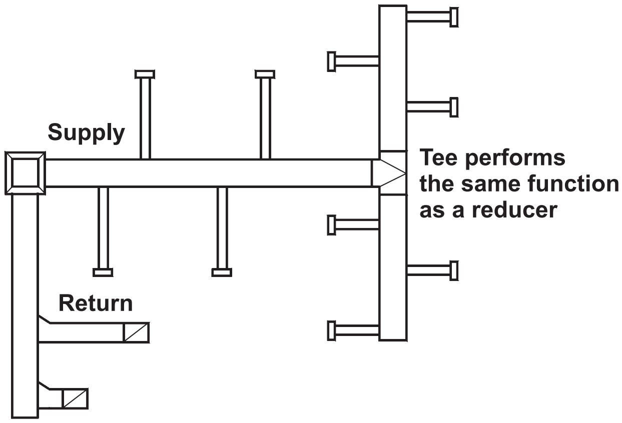


Figure A8-11

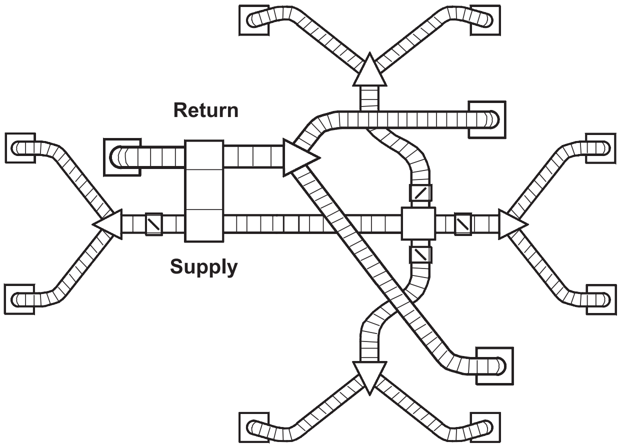


Figure A8-12

- Velocity at the entrance to the main trunk is at, or close to, the maximum recommended velocity.
- For a constant sectional area, Cfm and velocity along the main trunk decrease at each branch take-off.
- The sectional area of a secondary trunk is sized for the entrance Cfm and the design friction rate value (see the Section 7-4 example).

Flexible Duct Systems

Flexible, wire helix duct systems consist of large diameter flexible trunk ducts, triangular or rectangular junction boxes (which are normally fabricated from duct board) and smaller diameter flexible branch ducts. This type of system is normally used for attic installations, but basement and crawlspace installations are possible. Regardless of location, flexible duct runs should be cut to length and installed as straight as possible. (Kinked turns, coiling, looping and compressed coils create unnecessary pressure losses and reduced air flow.) Figure A8-12 provides an example of a flexible duct system.

A8-9 Radial Duct Systems

Radial duct systems have a supply plenum that feeds branch ducts arranged in a radial pattern. These systems may be easy and inexpensive to install because there is no trunk duct. Radial geometry is often used when a duct system is installed under a concrete slab because low perimeter supply outlets provide superior comfort during the heating season. These systems also are installed in attics, and they may be installed in a crawl space or a basement (if there is adequate head room under the joists).

Radial geometry is associated with designs that have a centrally located air handler (with respect to perimeter supply air outlets), but symmetry is not mandatory. If the duct runs are sized by ***Manual D*** procedures, any amount of off-set is acceptable.

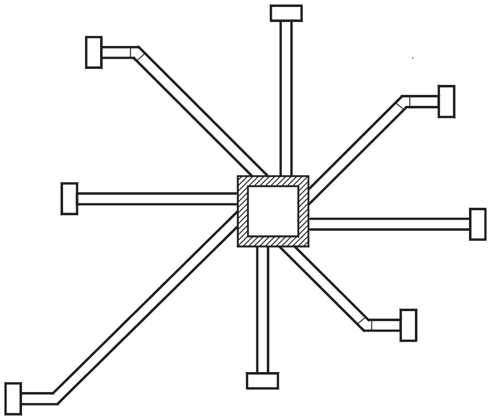


Figure A8-13

Figure A8-13 provides an example of a radial duct system that has rigid (metal or plastic) ducts. Rigid materials are used for duct runs below a slab floor, but they also may be used for crawlspace and attic locations. Flexible, wire helix duct material is commonly used when a radial system is installed in an attic, and may be used in a crawlspace.

A radial duct system may be installed above a drop ceiling. Figure A8-14 (next page) shows how radial duct runs supply air to high sidewall outlets located on the inside walls of a thermally efficient structure.

A8-10 Perimeter Loop Systems

Figure A8-15 (next page) provides an example of a perimeter loop duct system. This design improves floor-level comfort for cold- climate dwellings built on a ground slab.

- A loop system provides a little more ankle comfort than a radial system, but the loop system is more

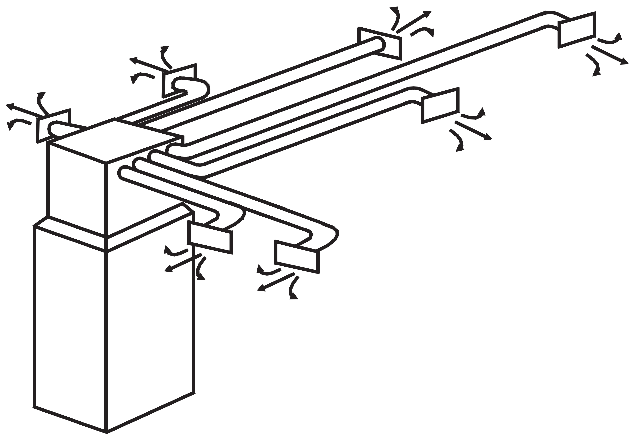


Figure A8-14

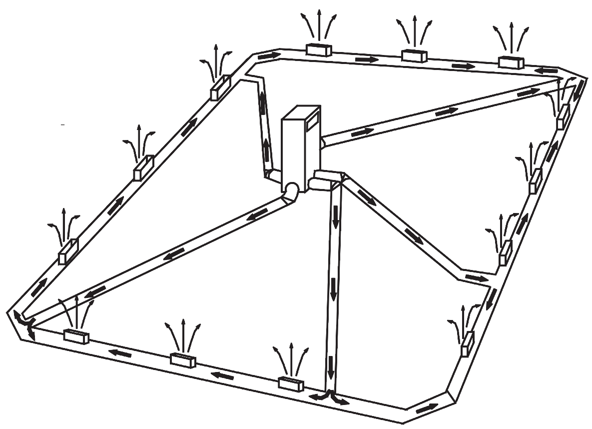


Figure A8-15

difficult to design, more expensive to install and has a larger slab-edge heat loss.

- Perimeter supply outlets, blowing straight up the wall are effective for heating and cooling (if correctly sized).

A8-11 Return Air Paths

The return air system must provide a low-resistance return air path between every conditioned space and the return-side of the air handler. If these paths are missing or deficient, air flow through some or all supply air outlets is affected.

For example, if a thick carpet and a closed door isolate one or more rooms from a central return, the rooms will be pressurized and the flow of supply air to these rooms will be reduced (and may be inadequate). For these designs, interior doors function as balancing dampers, throttling supply air as they close (the last few inches of gap controls the entire throttling range).

In addition, reduced air flow to closed rooms causes increased air flow in the remaining rooms. The net result is that the air distribution system is thrown out of balance, some rooms get too little air and other rooms get too much air. (Under these circumstances, envelope infiltration may increase because some rooms will be pressurized and some rooms will be subject to a negative pressure.)

Return in Every Room

The ultimate return air system has a ducted return for every room or space that is, or can be, isolated from the rest of the dwelling. This assures adequate return flow when interior doors are closed. This strategy also provides more privacy because transfer grilles are not required. Another advantage is that the return air system is quiet, because the return air openings are small and because they are acoustically isolated (by distance and multiple turns) from the blower. There also is an aesthetic advantage, because individual return grilles are much smaller than a central return grille. However, this design is geometrically complex, it requires more space, it may increase the effective length of the critical return path, it increases the number of potential leakage points and it is relatively expensive.

Central Return

A single central return is the least expensive system to install. (In multi-level dwellings, a central return is required for each level.) Usually, the return duct is short and the return-side pressure drop is small. This design is compact, easy to install and inexpensive. The disadvantages are that each isolated room must be equipped with a transfer grille, equipment noise may not be effectively isolated from the living space, and a large return grille may be unattractive.

Multiple Returns

A system that features multiple return air openings has the performance benefits provided by individual returns with the space-cost benefits of a single central return. With this design, a return air opening is provided for every major room or space and transfer grilles are used for secondary rooms. (At least one return is required for each level of a multilevel or multistory dwelling.)

A8-12 Plenum Systems

It is possible to use a crawlspace or a hall ceiling cavity as a supply air plenum or as a return air plenum. In either case, the desire to reduce installation cost is the primary motive for converting a structural cavity to a plenum. ACCA does not recommend this practice, but plenum systems may provide acceptable performance and comfort if certain installation requirements are satisfied. These requirements are summarized here:

Crawlspace Supply Plenum

As shown by Figure A8-16, a crawlspace supply plenum is used with a perimeter air distribution system. However, this system is significantly different than other types of perimeter systems because there are no duct runs between the air handler and the supply air openings.

For this design, short, dampered duct runs and aerodynamic boots are used to regulate and control air flow. (The purpose of a short feeder duct and boot is to control the acceleration of the air so it obtains a uniform velocity profile before it enters the supply grille. The purpose of the hand damper is to balance the air flow.) Plenum pressure can be maintained at about 0.10 to 0.15 inches water gauge. This is the approximate pressure drop through an abbreviated, dampered duct run, a boot fitting and a supply air outlet.

- Entrance to stub duct about 0.02 IWC.
- Pressure for wide open damper = 0.03 IWC.
- Fitting loss (30 to 80 Ft) = 0.03 to 0.08 IWC.
- Pressure for supply grille = 0.03 IWC.

The problems with this design are plenum leakage, plenum heat loss, moisture control, acceptable air outlet performance, and odor control. This design should not be used if one or more of these problems cannot be satisfactorily resolved.

- The entire crawlspace must be tightly sealed so that it can be pressurized (tightness should be verified by a pressure test). Air leakage to the outdoors, to unconditioned spaces and to conditioned spaces is not acceptable.
- The plenum walls and floor should be well insulated (R-19 walls and R4 floor). During winter, the crawlspace temperature will be warmer than the space temperature.
- The crawlspace floor and walls should be covered with a vapor retarder. Supply air should not absorb moisture from the soil, or moisture that leaks or migrates through walls.
- Each supply outlet should be equipped with a short "feeder duct," consisting of a converging transition, a stub duct equipped with a hand damper and an aerodynamic boot fitting.
- Odors, mildew and other air quality problems are caused by soil moisture and soil gas. Supply air must be isolated from the soil by airtight walls and an impermeable membrane system that is sealed at the foundation perimeter and all penetrations, with a trapped low-point drain.
- If a furnace is installed in the crawlspace, it must be a direct vent design (combustion air from outdoors by sealed duct, sealed combustion chamber and sealed vent).

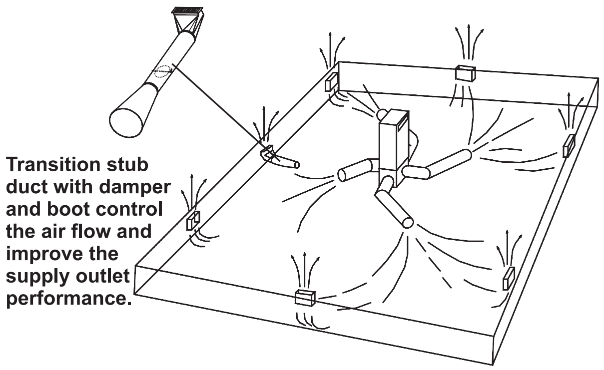


Figure A8-16

Ceiling Cavity Supply Plenum

A ceiling cavity supply plenum system may be used in conjunction with a high inside wall air distribution system. (It also is possible to run one or more ducts from the plenum to one or more ceiling outlets.)

As with the crawlspace design, plenum pressure is maintained at about 0.10 to 0.15 inches water gauge. This is the static pressure required to flow air to and through an abbreviated, dampered duct run, a boot fitting and a supply air outlet.

The problems with this design are ceiling plenum leakage, plenum heat loss and heat gain and acceptable air outlet performance. This design should not be used if one or more of these problems cannot be satisfactorily resolved.

- The entire ceiling cavity should be tightly sealed so that it can be pressurized (tightness should be verified by a pressure test). Air leakage to the attic, to unconditioned spaces and to conditioned spaces is not acceptable.
- Exposed plenum surfaces should be well insulated (R-19). During winter, cavity temperature will be warmer than the space temperature; and colder than space temperature during the summer.
- Exposed plenum surfaces should have an external vapor retarding membrane. Attic moisture (or moisture in an unconditioned space) must not migrate to the plenum.
- Each supply outlet should be equipped with a short "feeder duct," consisting of a converging transition, a stub duct equipped with a hand damper and an aerodynamic boot fitting.

Duct System Attributes for Various Locations						
Location	**Distribution**	**Supply Runs**	**Return Runs**	**Duct Shape**	**Duct Materials**	**Insulation**
Equipment in basement — air delivery to first floor and basement.	First Floor: Low perimeter supplies and low return(s). Basement: Ceiling, high or low perimeter supplies.	Trunk ducts below first floor framing. Runout ducts below or between floor joists. No panned airways.	Duct runs below first floor framing and/or between floor joists. No panned airways.	Rectangular trunks provide more headroom; round or rectangular branch duct runouts.	Metal or duct board trunks (consider potential for physical abuse); metal, rigid, fiberglass or flex duct runouts.	Duct insulation required if the basement is not fully conditioned.
Equipment in crawlspace; air delivery to the first floor.	Low perimeter supplies and low return(s).	Trunk ducts below floor framing; runouts below or between floor joists. No panned airways.	Duct runs below floor framing and/or between floor joists. No panned airways.	Rectangular or round trunks; round or rectangular branch duct runouts.	Metal for open crawlspace (animals or rodents); metal, rigid fiberglass or flex duct for closed crawspace.	Duct insulation required if crawlspace is open or if enclosed with poorly insulated, leaky walls.
Equipment in basement or crawlspace; air delivery to the second floor.	Low perimeter supplies and low return(s); or ceiling supplies and high return(s).	Interior risers in partition walls, chases or closets; perimeter runouts between floor joists; or horizontal runs in the attic. No panned airways.	Drops in interior partition walls, chases or closets; horizontal runs between joists; or horizontal runs in the attic. No panned airways.	Rectangular or oval risers and drops; round or rectangular perimeter run-outs; round or rectangular trunks and run-outs in attic.	Metal or duct board risers; metal, rigid fiberglass or flex duct for perimeter runouts; metal, rigid fiberglass or flex duct for the attic.	Insulation not required for duct in interior wall (duct must be inside of the wall insulation if it is in an exterior wall); insulate attic duct runs.
Equipment installed on the first floor; air delivery to the first floor (slab or crawlspace construction).	Low perimeter supplies, or ceiling supplies, or high side-wall supplies; single central return or multiple returns (located low or high, depending on the installation cost).	Perimeter outlets supplied by ducts buried in slab or installed in crawl space; ceiling outlets supplied by duct system in attic; sidewall outlets supplied by duct system in drop ceiling.	Central return at unit (with return path from every room);or return duct system below crawlspace floor; or return duct system in attic with drop to unit.	Round duct embedded in slab; rectangular or round duct in crawlspace; rectangular or round duct in attic; rectangular duct in ceiling cavity.	Plastic or plastic covered metal in slab; metal in open crawlspace (animals or rodents); metal, rigid fiberglass or flex for enclosed crawlspace or attic; metal or rigid fiberglass in ceiling cavity.	Slab edge insulation for ducts embedded in slab; duct insulation is required in an attic, in an open crawlspace, or if an enclosed crawlspace is poorly insulated.
Equipment installed on the first floor; air delivery to the first floor and second floor.	Perimeter ceiling outlets for first floor; perimeter floor outlets for second floor; return(s) for each level.	Supply trunk located in drop ceiling; runouts between ceiling joists to first and second floor outlets.	Returns on both levels; drops located in interior partition walls, chases or closets.	Rectangular trunks provide more headroom; round or rectangular branch duct runouts.	Metal or rigid fiberglass in ceiling cavity; metal, rigid fiberglass or flex duct for perimeter runouts.	Insulation not required for duct in interior ceiling cavity.
Split level home, no basement.	Central equipment in crawlspace or on slab level. Low perimeter supplies for all levels. Ducts in crawlspace, embedded in slab and in ceiling cavity below upper level. Refer to situations described above for details.					
Split level home with basement.	Central equipment in basement. Low perimeter supplies for all levels. Ducts in basement, embedded in slab or in crawlspace, risers in partitions to second floor outlets. Refer to situations described above for details.					
1) Duct runs located outdoors, in unconditioned spaces, or within the framing spaces should be sealed (***Manual J*** maximum for sealed duct is 0.12 Cfm/SqFt for supply duct and 0.24 Cfm/SqFt; tighter construction is preferred, See Unabridged ***Manual J***, Eighth Edition, Figure 23-7). 2) Exposed duct runs (to outdoors or unconditioned space) should be insulated (R6 minimum, R8 preferred).						

Figure A8-17

Return Plenums

A tight crawlspace or ceiling cavity may serve as a return plenum. In this case, return air flows through plenum openings, then to the return-side of the blower cabinet. The problems with this design are plenum leakage, plenum heat losses, moisture control and indoor air quality. A return air plenum shall not be used if each one of these problems cannot be satisfactorily resolved by engineering specifications and construction practices.

- The entire plenum should be tightly sealed so that it can be depressurized (tightness should be verified by a pressure test). Air leakage from the outdoors, from unconditioned spaces and from conditioned spaces is unacceptable.
- Exposed plenum surfaces should be well insulated. Cavity temperature will be close to space temperature during the winter and summer.
- Exposed plenum surfaces should have a vapor retarding membrane. Return air must not absorb moisture from the soil, or moisture that leaks or migrates through exposed wall and ceiling surfaces.
- If a crawlspace is used as a plenum, return air should be isolated from the soil by an impermeable membrane because odors, mildew and other air quality problems are caused by soil moisture and soil gas.
- If a furnace is installed in a return plenum, it must be a direct vent design (combustion air from outdoors by sealed duct, sealed combustion chamber and sealed vent).

A8-13 Architectural Compatibility

Ducts are routed through available, unused space, or seldom-used space. Figure A8-17 (previous page) summarizes the possibilities for installing ducts in various locations.

Appendix 9 (Informative; not Part of the Standard)

Equipment and Air-Side Components

This section introduces conventional types of residential HVAC equipment and components. This presentation emphasizes attributes and characteristics that affect air-side performance.

A9-1 Air Distribution System Components

A complete air distribution system, whether intended for winter, summer, or year-round use, consists of primary equipment, secondary equipment, air-side components and duct runs. Primary equipment has components that are installed in the air stream and components that have nothing to do with the air-side of the system. Secondary equipment and air-side devices are always installed in the air stream.

Primary Equipment

Primary equipment provides the basic functions for comfort conditioning (heating, sensible cooling, latent cooling and air filtration) and an air moving component (the blower). This equipment could be a furnace, a cooling-only unit, or a heat pump. Unitized equipment contains air-side components that are an integral part of the assembly (a blower, cooling coil or heat exchanger) and accessory devices that are easily be added to, or removed from, the basic assembly (a filter or a supplemental electric resistance heater). Some packages place all of the components in one cabinet (single package unit) and other designs place the air-side components (blower, coil and filter) in a separate cabinet (split system).

Secondary Equipment

Secondary equipment includes components normally supplied with the basic equipment package, and components that supplement or modify the performance of the basic package. Examples include electric, duct-mounted heating coils, DX cooling coils, heat-reclaim water coils, media filters, electronic filters and humidifiers.

Air-Side components

Air-side components are used to control air flow. Supply outlets mix supply air with room air, grilles capture the return air, dampers and junction boxes control the air flow in the duct system.

A9-2 Primary Heating and Cooling Equipment

Forced air heating, cooling, or year-round conditioning equipment, includes an energy conversion component, which may be fuel-fired or electrically powered. Fuel-fired components usually have a burner (gas or oil fired) and a combustion chamber (heat exchanger). Electrically powered components range from a simple electric resistance heating coil, to an assembly of refrigeration cycle machinery and hardware (air conditioner or heat pump). In any case, the primary equipment will include a blower. Therefore, as far as the duct system is concerned, the performance of the heating and cooling equipment is defined by the performance and the arrangement of the blower section.

Furnaces

A fuel-fired or electric furnace is the primary component of most forced air heating systems. For cooling, furnaces are fitted with a refrigerant coil. Furnace manufacturers provide a number of configuration options that are designed to fit in the available space, which may be a basement, crawlspace, attic, ceiling cavity, or closet. The cabinet configuration also depends on he location of the supply trunk and return trunk (above, below, or in-line).

Low-Boy Arrangement

A low-boy arrangement locates the blower section near floor level and to the side or rear of the heat exchanger. This design minimizes the head room used by the equipment and duct system. These units are usually installed in basements because they have a low profile and a vertical discharge. In this location, the vertical discharge is compatible with duct runs located above the equipment. Figure A9-1 provides examples of gas-fired and oil-fired low-boy furnaces.

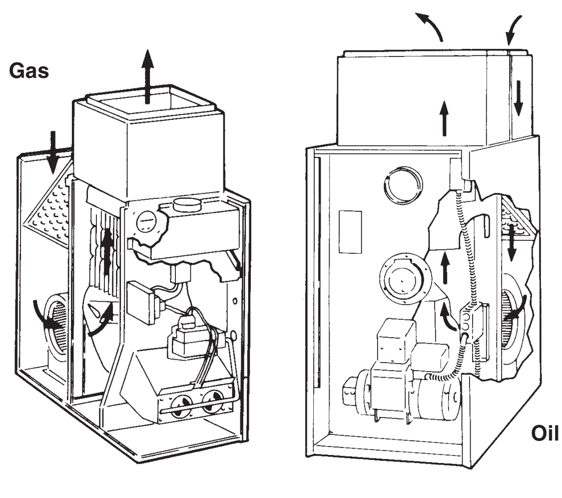


Figure A9-1

Up-flow Arrangement

An up-flow (hi-boy) arrangement locates the blower section directly below the heat exchanger. This configuration is generally installed in closets, small utility rooms and basements (when ceiling height permits). Because this cabinet has a vertical discharge, it is compatible with overhead duct runs. Figure A9-2 provides examples of gas-fired and oil-fired up-flow furnaces.

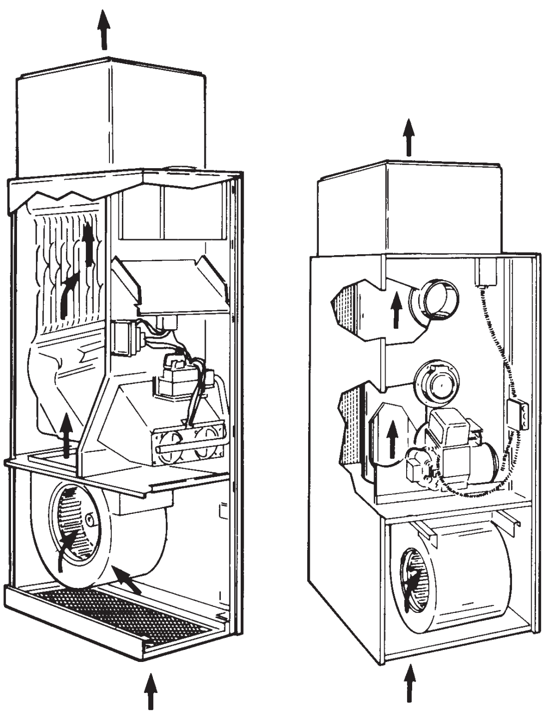


Figure A9-2

Down-flow Arrangement

The down-flow arrangement places the blower compartment above the heat exchanger. This configuration is used when supply ducts are below the equipment. Down-flow units are usually installed in dwellings that have slab floor or crawlspace construction, and are normally in a closet or small utility room. Figure A9-3 provides examples of gas-fired and oil-fired down-flow furnaces.

Horizontal Arrangement

Horizontal furnaces require a minimum amount of vertical clearance because the blower section and the heat exchanger are on the same level. This type of furnace is commonly installed in an attic or crawlspace, but it could be hung below a basement ceiling or a utility room ceiling. Since the unit features a horizontal discharge, it may be connected to a duct system located above, below or at the same level as the furnace. Figure A9-4 provides examples of horizontal gas-fired and oil-fired furnaces.

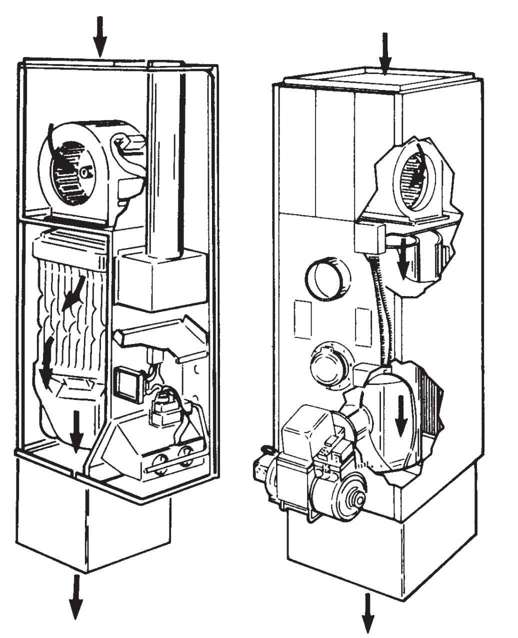


Figure A9-3

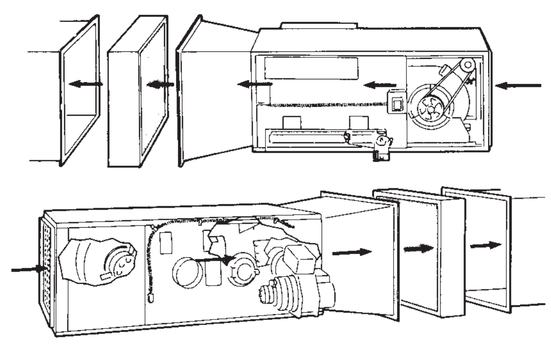


Figure A9-4

Multi-position Furnaces

Most manufacturers offer a multi-position furnace. This equipment may provide up-flow, down-flow and horizontal arrangements.

Blower Performance

Some furnaces are designed for heating-only applications and others are compatible with heating-cooling applications. The difference is that the heating-cooling equipment has a more powerful blower, which compensates for the air flow resistance produced by the cooling coil.

In any case, blower performance data is usually part of the technical information published by the furnace manufacturer. These tables or graphs are used to design the duct system.

Manufacturer's blower performance data is produced by testing a furnace that has a specific set of air-side components. This test *may not account for all of the air-side devices that might be added to the basic package (a cooling coil or filter upgrade, for example. (Sections 1-10, 3-3 (page 21), 3-5 (page 23), 3-7, 4-7, 4-8, 7-4, 7-5 provide more information on this subject.*

Air Conditioners and Heat Pumps

Air conditioners and heat pumps (air-source and water-source) are sold as split systems and as a single package. Split systems have an outdoor section (condensing unit) and an indoor section (air handler). Single package units have both sides of the refrigerant system in one outdoor cabinet, typically on a roof curb or ground pad.

Split Systems — Cooling-Only and Heat Pump

Split system air handlers normally have a refrigerant coil, blower and filter (an electric resistance coil is commonly included with, or added to, heat pump equipment.) This equipment is available in up-flow, down-flow and horizontal configurations. This provides necessary options for locating the air handler (attic, basement, crawlspace or utility closet) and the duct runs (above, below or at the same level as the equipment). Figure A9-5 provides an example of split system equipment.

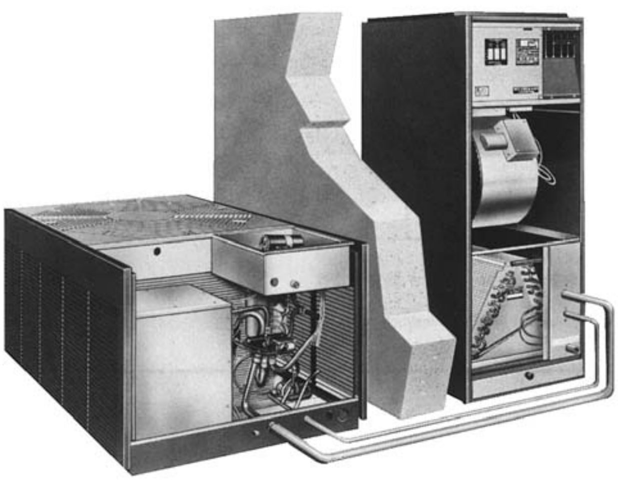


Figure A9-5

Cooling Added to a Furnace

Split system air conditioning equipment is used with forced air furnaces (fossil fuel or electric). For this design, a refrigerant coil is added to the discharge side of the furnace (the blower and the filter are part of the furnace package.) Cooling coils, coil cabinets and coil casings are available for up-flow, down-flow and horizontal applications. Figure A9-6 shows an up-flow furnace equipped with a cooling coil.

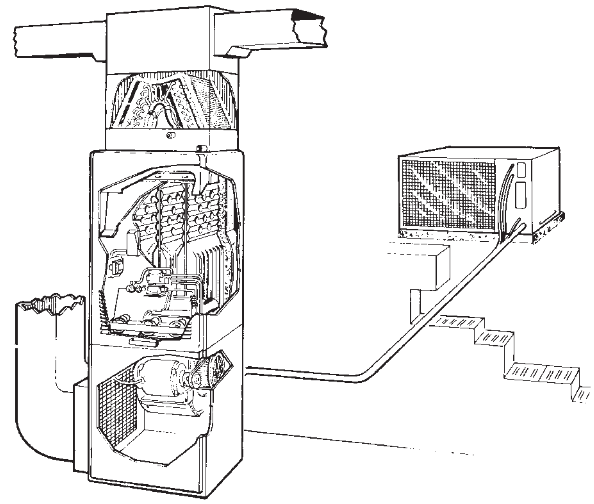


Figure A9-6

Heat Pump Added to a Furnace

Split system heat pump equipment may be used with a fossil fuel furnace. In this case, a dual purpose (heating-cooling) refrigerant coil is added to the discharge side of the furnace. (The blower and the filter are part of the furnace package. Heat pump controls switch the function of the indoor and outdoor coils, and lock out heat pump operation when the furnace is firing.)

Single Package Systems — Cooling-Only and Heat Pump

Self contained cooling units and heat pumps have the refrigerant-side components and air-side components (blower, coil and filter) in one cabinet. (Heat pump units typically have an electric resistance heating coil that supplements heat pump output during cold weather.) Because this cabinet is outdoors, supply and return ducts must penetrate a wall or roof.

Blower Performance

A blower performance table or graph is usually provided with technical information published by equipment manufacturers. Such tables or graphs are used to design the duct system.

Manufacturer's blower performance data is produced by testing an air-handler that has a specific set of air-side components. This test may not account for all the air-side components that might be in the equipment cabinet (electric resistance heat or filter upgrade, for example). Sections 1-10, 3-3 (page 21), 3-5 (page 23), 3-7, 4-7, 4-8, 7-4, 7-5 provide more information on this subject.

A9-3 Secondary Equipment

Secondary equipment includes filter upgrades, supplemental electric or water heating coils and humidifiers. A refrigerant coil added to a furnace is secondary equipment.

Secondary equipment usually increases resistance to air flow. This resistance equals the pressure drop across the component produced by the flow through the component. For a given component, pressure drop vs. Cfm values are provided by manufacturer's performance data.

The pressure drop for any secondary or ancillary component not mentioned by the blower table, or its footnotes, is subtracted from the external static pressure value read from the blower table. (Refer to Sections 1-10, 3-3 (page 21), 3-5 (page 23), 3-7, 4-7, 4-8, 7-4, 7-5 for more information about blower performance data.)

Figure A9-7

Figure A9-8

Filters

Filters remove particles (dust and pollen, for example) and mists entrained in the flow of return air. However, most filters do not remove the small particles, gases and odors that cause air quality and health problems, or pollutants produced by tobacco smoke. (High efficiency filters may be used to trap small particles; absorption and air-washing equipment may be used to remove odors and gases.)

Media filters and electronic filters are normally used for residential applications. Media filters provide two types of cleaning action.

- They strain particles from the air, but this action only removes the particles that are larger than the openings in the media.
- They use a viscous coating or an electrostatic charge to snare small particles that come in contact with the fibers of the filter.

Various types of filters are described here. In some situations it may be necessary to pass air through two filters in order to obtain the desired result.

Viscose Media Filters

Viscose media filters are normally furnished with residential equipment. These products have a media (mat of coarse material such as glass fiber, expanded metal, animal hair, nylon thread, or some combination of these materials) that is coated with a sticky substance. The media strains the air, but it also acts like low-velocity centrifuge. When air passes through the mat, it changes direction suddenly and often. This churning action causes the particles — which are unable to change direction as quickly as the air — to strike and stick to the media.

The standard panel filter supplied with furnaces and air handlers protect the HVAC equipment from lint, fibers and large particles, but do not remove most of the smaller particles that affect indoor air quality. The pressure drop across a viscous media panel (when it is clean) is about 0.10 inches water column (IWC), but a more precise value is obtained from the manufacturer's performance data. Figure A9-7 provides examples of a viscose media filter.

Electronic Air Cleaners

Electronic air cleaners ionize air stream particles. Then the particles are attracted to a charged plate. Once in contact with the plate, the particles lose their charge and are held on the plate by natural adhesion, or a viscous film. The effectiveness of residential products may range from unimpressive, to good (depending on the rating produced by authorized testing and rating procedures. The more effective products eliminate particles that cause allergic reactions, smudges and stains, and may be partially effective on tobacco smoke. Figure A9-8 (previous page) provides an example of an electronic air cleaner.

A pre-filter is commonly installed upstream from the electronic filter (its purpose is to remove small particles). The pressure drop across an electronic air cleaner may range from about 0.10 IWC to about 0.20 IWC. Refer to manufacturer's performance data for the actual value.

- If a factory-installed panel filter is used as a pre-filter for an electronic air cleaner, the ancillary pressure drop is the pressure drop for the electronic assembly.
- If a factory-installed panel filter is replaced by an electronic air cleaner that has its own pre-filter, the ancillary pressure drop is the difference between the electronic assembly pressure drop and the factory filter pressure drop.

Charged-Media Filters

Charged-media air cleaners also use an electrostatic field, but the voltage is applied to the filter media. When particles pass through this filter, they are attracted to the media by electrostatic action. However, this design is not as effective as an electronic air cleaner because the particles have a limited ability to be polarized by the field. The effectiveness of this type of filter is only slightly better than the effectiveness of a conventional viscous media filter. Refer to the manufacturer's performance data for the pressure drop across this type of filter.

Plastic Static-Charge Filters

When air passes through a plastic media at a relatively high velocity, the media becomes charged with static electricity. However, the ionizing effect on undesirable particles is less than what is produced by a charged-media filter, and the charge diminishes as the relative humidity of the return air increases. The effectiveness of this type of filter is similar to that of a conventional viscous media filter. Refer to the manufacturer's performance data for information about the pressure drop across this type of filter.

Plastic Foam Filters

Open-pore plastic foam filters depend upon a straining action to remove large particles, and use a clinging action to capture small particles. The effectiveness of this type of filter is similar to a conventional viscous media filter. Refer to the manufacturer's performance data for information about the pressure drop across this type of filter.

Dry Extended Surface Filters

Dry, extended surface filters are made of materials that have very fine pores. Normally these filters use a pleated geometry to reduce the air flow resistance created by the media and to increase the holding capacity of the media. The effectiveness of this of this design depends on the product. The most effective products eliminate particles that cause allergic reactions, smudges and stains; and are partially effective at removing very small particles and capturing contaminates in tobacco smoke. Note that the pressure drop across an extended surface filter is normally much larger than the 0.10 IWC (approximate) value for the standard filter supplied with the equipment. Refer to the manufacturer's performance data for information about the pressure drop across this type of filter.

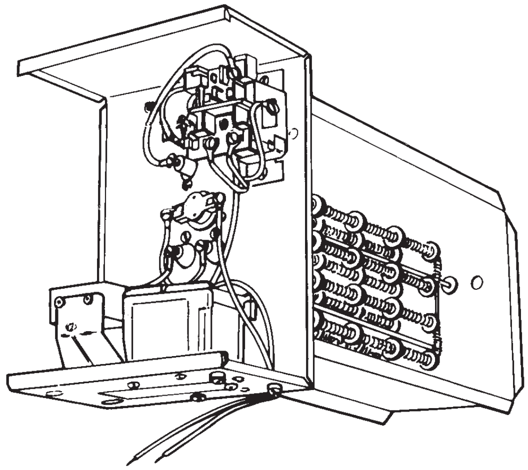


Figure A9-9

Supplemental Heaters

An electric heating coil added to a heat pump air handler is the most common type of supplemental heat. This component is controlled by a central thermostat and it helps heat every room or space served by the air handler.

Small electric coils, installed in runout ducts or special boot fittings, provide supplemental heat for particular rooms and spaces. These heaters may be controlled by a low-limit thermostat that operates independently of the central thermostat.

The pressure drop across a supplemental electric heater can vary from less than 0.10 IWC to more than 0.20 IWC, depending on the product and the air flow rate. Refer to the manufacturer's performance data for this important pressure drop information. Figure A9-9 shows an example of a duct heater.

Humidifiers

Add-on humidifiers increase indoor humidity during the heating season. These components are classified as adiabatic, or isothermal.

- Adiabatic components do not have their own source of heat. The heat of evaporation comes from the supply air. This produces a drop in the dry-bulb temperature of the supply air. A wet media humidifier is an example of adiabatic humidification equipment.

- Isothermal humidification equipment does not produce a temperature change. The heat of evaporation is provided by an integral heater or external heat source. A heated pan humidifier is an example of isothermal humidification equipment.

Indoor air quality is an important winter humidification issue. Humidifiers must not produce biological contaminants, such as bacteria, algae, mold or fungi.

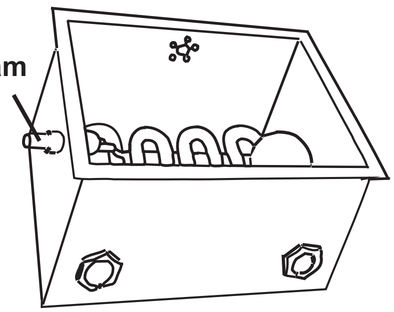
- Most of these biological agents do not cause serious health problems, but they can cause allergic reactions and odor problems.
- There is one important exception; the bacteria associated with Legionnaires disease is deadly.

Reservoirs, drain pans, dripping or spitting nozzles, and condensation on duct materials or building components, are all potential sources of biological contamination. The likelihood of the occurrence of an air quality problem depends on the type of humidification component. In general, sterile feed water, a high operating temperature, proper sizing and installation, proper water treatment, scheduled inspection and proper maintenance will keep biological growth under control.

Most residential humidification components are installed in a duct or plenum, but there are components that are installed in a room. All types of components must be mounted so that clearances are maintained and condensation is prevented (i.e, according to applicable codes and manufacturer's instructions). A brief review of common types of residential humidification equipment is provided here:

Pan Humidifiers

Pan humidifiers are simple components that evaporate water from the surface of a heated reservoir. (If the reservoir is not heated, the component is not very effective.) Heat is supplied by an electric coil. These components do not cause a significant change in air temperature because the heat of evaporation is supplied by the component. Pan humidifiers are suitable for small humidification loads, and they can be installed in a supply air duct or in a room. Regular blow down and weekly or monthly maintenance is required. This keeps the pan and the heating coils free of biological, chemical or mineral deposits. When properly installed, there should be no water droplet, wetting, or duct corrosion problems. These humidifiers are on-off components; because of the dynamics of the evaporation process, they have a sluggish response to a call for moisture and they are slow to shut down. If the parts of the humidifier project into the air stream, they may produce a resistance to the air flow, but this pressure drop information may not be documented in the manufacturer's performance data. (The pressure drop penalty will probably be less than 0.10 IWC.) Figure A9-10 shows a pan humidifier.



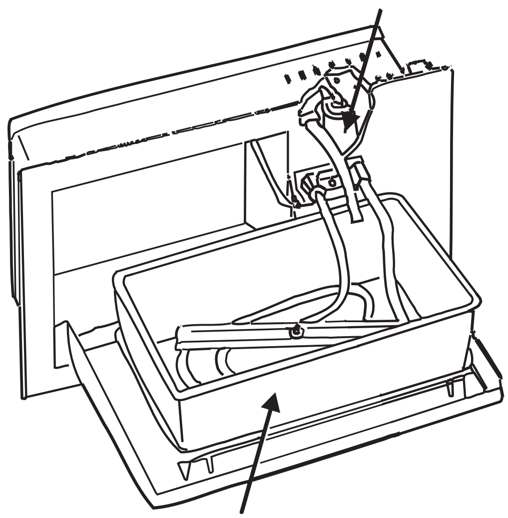


Figure A9-10

Wetted Media Humidifiers

Wetted media humidifiers evaporate water from the surface of a wet pad. The media is wetted by a nozzle or by immersion in a sump. (Nozzle units use considerably more water, but they provide continual cleansing (blow down.) Usually, the heat of evaporation is extracted from the air flowing through the media. This causes a drop in air temperature. (In some cases, performance is enhanced by a supplemental heater that adds heat to the water or the air.) Wetted media humidifiers are suitable for small humidification loads, and there are many different types of duct-mounted and self-contained room units to choose from. These components require regular maintenance to clear the sump or reservoir of any biological, chemical or mineral deposits. When properly installed, there should be no water droplet, wetting or duct corrosion problems. Mineral fallout (dusting) is not a problem with this type of equipment. Wetted media humidifiers are on-off components. Depending on the design, they have a slow to good response to a call for moisture, and a reasonably quick response to a shut-down command.

Figure A9-11 shows a wetted media unit that protrudes into the duct system. This component will produce a resistance to the air flow, but this pressure drop information may not be documented in the manufacturer's performance data. (The pressure drop penalty will probably be less than 0.10 IWC.)

Figure A9-11

Figure A9-12 shows a duct-mounted bypass unit. In this case the air flow through the media is induced by the pressure difference across the blower equipment. This design does not produce much resistance to the air flow in the supply duct, but air is continuously diverted through the bypass duct, which causes a corresponding reduction in the supply air Cfm.

Figure A9-13 shows a fan powered bypass unit. In this case an integral fan causes the air to flow from the duct, through the media and back to the duct. This equipment does not produce much resistance to the air flow in the supply duct and it does not cause a reduction in the supply Cfm.

Atomizing Humidifiers

Atomizing humidifiers spray a fine mist of water droplets into the air. Therefore, the air supplies the heat of evaporation, which produces a drop in air temperature. Spinning disk and diffusing screen units are designed for very small loads. Small nozzle-spray units are available. These components are normally installed in a duct. There is a possibility of wetting nearby surfaces if the nozzle is not properly sized, controlled and installed. Regular maintenance is required to keep biological and mineral deposits under control. Small atomizing humidifiers are usually on-off components, and they have a good response to a call for moisture and to a shut-down command. Units that do not protrude into the air stream have no effect on the air flow in the supply duct. Figure A9-14 (next page) provides an example of an atomizing humidifier.

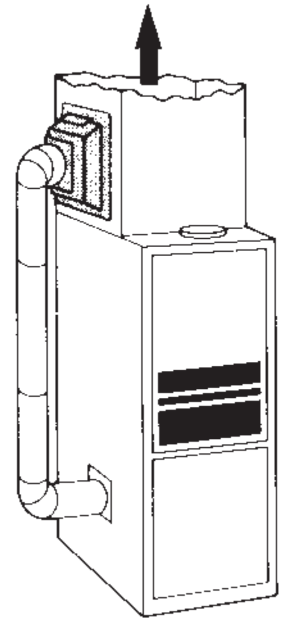


Figure A9-12

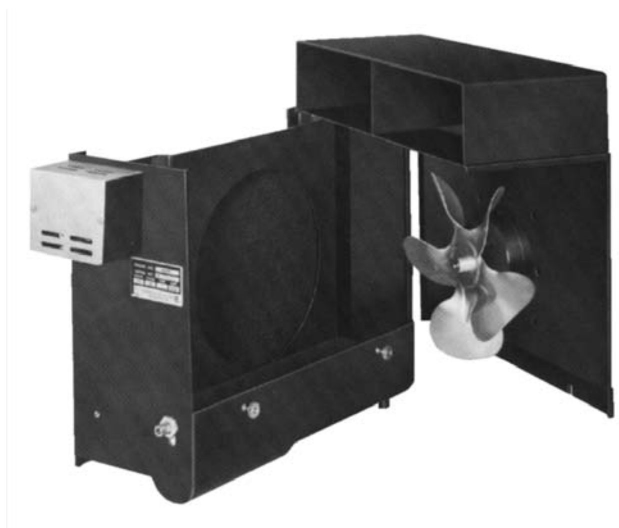


Figure A9-13

Refrigerant Coils and Water Coils

Refrigerant coils added to furnaces may be part of a cooling-only system or part of an add-on heat pump system. Water coils added to an air distribution system may be part of a hot water heating system or a thermal storage system. In either case, the coil adds considerable resistance (pressure drop) to the supply-side of the air distribution system. Therefore, it is very important to verify that the furnace blower has the power to overcome this additional resistance. Pressure drop values for refrigerant coils and water coils is provided by manufacturer's performance data. Figure A9-6 shows a furnace that is equipped with a refrigerant coil.

- The pressure drop across a wet refrigerant coil is greater than the pressure drop across a similar dry coil.
- Normally the wet-coil value is required, but the dry-coil value is used when a dry climate causes the ***Manual J*** latent load to be zero or negative.

A9-4 Air-Side components

Air-side components include supply air grilles, registers, diffusers, return grilles, filter grilles, balancing dampers, zone dampers and flex-duct junction boxes. These components increase resistance to the air flow when added to a duct system. Relevant issues are listed here:

- All supply and return runs are in parallel.
- For supply-side calculations, there is only one supply outlet (and one hand damper or zone damper, if applicable).
- For return-side calculations, there is only one return (and one hand damper, if applicable).
- There may be one or more junction boxes in a supply path or return path.
- Applicable component pressure drops are subtracted from the external static pressure produced by the blower.
- Flexible duct junction boxes have an equivalent length (see below).
- Refer to Sections 1-10, 3-3 (page 21), 3-5 (page 23), 3-7, 4-7, 4-8, 7-4, 7-5 for more information about adjusting blower performance data.

Supply and Return Hardware

Supply diffusers, registers, and grilles are designed to mix the supply air with room air. Returns are usually grilles, but they could be registers. The pressure drop across one of these components is usually less than 0.03 IWC.

- The ***Manual D*** default (0.03 IWC) may be used for all duct sizing calculations.

Figure A9-14

- Manufacturer's pressure drop data may be substituted for the ***Manual D*** default.
- Refer to ACCA ***Manual T*** for information about selecting, sizing and positioning this hardware.

Dampers

Balancing dampers (hand dampers) and zone dampers are normally located in branch supply ducts. Balancing dampers are manually set to provide the desired flow to a room or space. Zone dampers, typically controlled by a room thermostat, maintain room temperature at the desired set-point.

The pressure drop across a hand damper in the full open position is about 0.03 IWC. The pressure drop produced by an open zone damper may be considerably more than 0.03 IWC (flow control authority depends on the ratio of the component pressure drop to the system pressure drop).

- The ***Manual D*** default (0.03 IWC) may be used for any hand damper.
- The pressure drop value for a fully open zone damper is obtained from manufacturer's performance data.

Flex-Duct Junction Boxes

A pressure loss occurs when air enters a flex-duct junction box, and a second pressure loss occurs when air leaves the box. These losses depend on the geometry of the junction box, on the upstream air velocity, and on the downstream air velocity. ***Manual D*** uses equivalent length values to account for these effects. (see Appendix 3, Group 11).

- Group 11 equivalent length values are for the worst-case geometry (i.e., the box is large compared to the size of the upstream duct, so all upstream velocity pressure is lost as the air enters the box).

- Group 11 equivalent length values are for a straight-run approach and a straight run departure. Turns or bends near entrance or exit openings significantly affect box performance, and invalidate the Group 11 equivalent length values.
- Note that equivalent length values are very sensitive to air velocity (moderate velocities are preferred).

Appendix 10 (Informative; not Part of the Standard)

Duct System Efficiency

Duct system efficiency depends on the air flow resistance produced by duct runs, heat transfer through duct walls and leakage at seams and joints. Because air velocity is relatively low, air flow resistance is not an important efficiency issue, but it is an important airway sizing issue.

- When duct runs are installed in unconditioned spaces or outdoors, conduction and leakage losses and gains have a significant effect on duct system efficiency, heating-cooling equipment size, energy use and operating cost.
- Conduction and leakage affect comfort, and leakage can affect indoor air quality and may create health and safety problems.

A10-1 Aerodynamic Efficiency

If flow rate and cross-sectional area are constant, aerodynamic efficiency depends on airway shape. In this regard, a round shape is the most efficient because the friction rate (pressure drop per 100 feet of length) of a round airway is always less than the friction rate for other shapes.

Square shapes and rectangular shapes are fairly efficient, but the aerodynamic efficiency decreases as airway aspect ratio increases. (Oval shapes are slightly more efficient than rectangular shapes that have the same aspect ratio.) Figure A10-1 provides a comparison of the aerodynamic efficiency of round, rectangular and oval shapes.

Aerodynamic Efficiency of Duct Shapes

Cfm = 1,000 Area = 1.25 SqFt Velocity = 800 Fpm

Shape	Aspect Ratio	Friction Rate FR	Airway Size (Inches)
Round	NA	0.066	15.14
Square	1:1	0.070	13.42 x 13.42
Rectangular	2:1	0.075	9.49 x 18.97
Oval	2:1	0.070	10.04 x 20.08
Rectangular	4:1	0.095	6.71 x 26.83
Oval	4:1	0.090	6.90 x 27.58
Rectangular	8:1	0.130	4.74 x 37.95
Oval	8:1	0.125	4.81 x 38.47

Friction rates from the ACCA Duct Sizing Slide Rule.
Sheet metal duct material.
FR = IWC per 100 feet of duct.
1.25 SqFt = 180 SqIn.

Figure A10-1

Aerodynamic Efficiency of Duct Material

Cfm = 1,000 Area = 1.25 SqFt Velocity = 800 Fpm

Shape	Aspect Ratio	Friction Rate (FR)		
		SM	DB	Flex
Round	NA	0.066	0.080	0.120
Square	1:1	0.070	0.085	NA
Rectangular	2:1	0.075	0.092	NA
Rectangular	4:1	0.095	0.130	NA
Rectangular	8:1	0.130	0.180	NA

Friction rates from the ACCA Duct Sizing Slide Rule.
SM = Sheet Metal; DB = Duct Board; Flex = Wire Helix Duct.
FR = IWC per 100 feet of duct.

Figure A10-2

Equal Resistance Designs

Cfm = 1,000 Friction Rate = 0.10 IWC per 100 Feet

Material	Airway Size (Inches)	Airway Area (SqFt)	Air Velocity (Fpm)
Sheet Metal	14.0 Dia.	1.07	935
Duct Board	14.6 Dia	1.16	862
Sheet Metal	6.7 x 26.8	1.25	800
Duct Board	6.9 x 27.6	1.32	758
Flex Duct (4%)	15.7 Dia.	1.34	746

Sizes read from the ACCA Duct Sizing Slide Rule.
Sizes = Inches Area = SqFt Velocity = Fpm.

Figure A10-3

The aerodynamic efficiency of a straight duct run also depends on the surface roughness of the airway material. Plastic and sheet metal surfaces are more efficient than fibrous glass surfaces (duct board and duct liner) and fibrous glass surfaces are more efficient than flexible wire helix surfaces (flex duct). Figure A10-2 compares the aerodynamic efficiency of three popular duct materials.

Since available static pressure is limited, airway sizes are increased to compensate for inefficient airway shapes and/or rough airway material. Figure A10-3 demonstrates this concept. Note that as the aerodynamic inefficiency increases, cross-sectional areas and the amount of fabrication material increase, and air velocity decreases.

A10-2 Conduction Losses

Conductive heat gain and heat loss occurs when duct surfaces are exposed to the outdoor air, or when ducts are installed in an unconditioned space. This affects the size and efficiency of the comfort system, but the indoor design condition (desired temperature and humidity) is maintained when duct loads are added to the equipment sizing load. But, comfort is not the only consideration. Conduction loads should be minimized because they degrade system efficiency, waste energy, increase equipment loads, and increase annual operating cost.

Minimum R-Value

Manual J duct load tables show that ducts installed outdoors or, in unconditioned spaces should have R-6 insulation. They also show that R-8 is preferred if duct runs are exposed to temperatures that approach or exceed winter and summer design temperatures.

- ***Manual J*** and ***Manual D*** procedures and guidance related to duct insulation is superseded by local codes and regulations.
- For 2009, national and international energy codes specify R-8 insulation for ducts located outside the conditioned space, and R-6 for ducts in floor joist space.
- Refer to the current version of a code, standard or regulation published by any body or authority.

Effective R-Values

As demonstrated here, there are many ways to obtain a duct insulation R-value. These suggestions recognize that it is difficult to install glass fiber blankets without causing some compression. Since some insulation manufacturers suggest that 50 percent compression is typical, recommended blanket thicknesses are adjusted for 50 percent compression. (A 50 percent compression translates to a 39 percent reduction in insulating efficiency.)

Effective R-2 (blanket adjusted for 50% compression)

- 1 inch of 0.6 to 0.75 Lb/CuFt. glass fiber blanket.
- ½ inch of 2 to 3 Lb/CuFt glass fiber duct liner.
- ½ inch of 3 to 10 Lb/CuFt glass fiber board.
- Flexible duct with 1 inch glass fiber jacket.

Effective R-4 (blanket adjusted for 50% compression)

- 2 inches of 0.6 to 0.75 Lb/CuFt glass fiber blanket.
- 1 inch of 1.5 to 3 Lb/CuFt glass fiber duct liner.
- 1 inch of 3 to 10 Lb/CuFt glass fiber board.
- Flexible duct with glass fiber jacket rated at R4.

Effective R-6 (blanket adjusted for 50% compression)

- 3 inches of 0.6 to 0.75 Lb/CuFt glass fiber blanket.
- 2½ inches of 1.0 Lb/CuFt glass fiber blanket.
- 1½ inches of 1.5 to 3.0 Lb/CuFt glass fiber duct liner.
- 1½ inches of 3 to 10 Lb/CuFt glass fiber board.
- Flexible duct with glass fiber jacket rated at R6.

Effective R-8 (blanket adjusted for 50% compression)

- 1 inch duct liner plus 2 inches glass fiber blanket.
- 1 inch duct board plus 2 inches glass fiber blanket.
- 1½ inches duct liner plus 1 inch glass fiber blanket.
- 1½ inches duct board plus 1 inch glass fiber blanket.
- Flexible duct with glass fiber jacket rated at R8.

Effective R-value Determined by Code

The IECC code (2009) is specific in that "duct insulation thermal performance shall be determined by the installed wall thickness divided by the thermal conductivity of the insulation at that installed thickness."

A10-3 Vapor Retarders

In humid climates, duct board or external wrap may not provide enough insulation to prevent condensation on duct walls. Facings and wraps that have a perm rating of 0.50 or less shall be installed when the average cooling season (July/August) outdoor dew point exceeds 60°F.

Figure A10-4 (next page) roughly identifies locations that may have excessive dew points. Average monthly dew point temperature maps for the USA, and average monthly dew point temperature values for specific cities are provided by Version 2 of the ***Climate Atlas of the United States*** produced by NOAA's National Climatic Data Center (to purchase, do a web search for Climate Atlas of the United States).

A10-4 Leakage Losses

If duct runs are installed outdoors, or in an unconditioned space, leakage increases equipment heating and cooling loads, increases operating costs, and can have an adverse effect on comfort and indoor air quality. Leakage also can produce room-to-room pressure differences, increase envelope infiltration loads, and disrupt the balance of the air distribution system. This translates to a need for larger heating and cooling equipment, increased air flow rates (Cfm), larger ducts, larger blower, increased installation cost, a larger demand on the gas and electric grids, increased energy use, and increased operating cost. (When leaky ducts characterize local installation practices, practitioners install more heating and cooling capacity than would otherwise be required.)

Duct leakage to or from the conditioned space is less of an issues as far as energy use and operating cost are concerned, but it can have an adverse affect on comfort system performance. Duct sealing is required if leakage produces room-to-room pressure differences that increase envelope infiltration loads and/or disrupt the balance of the air distribution system.

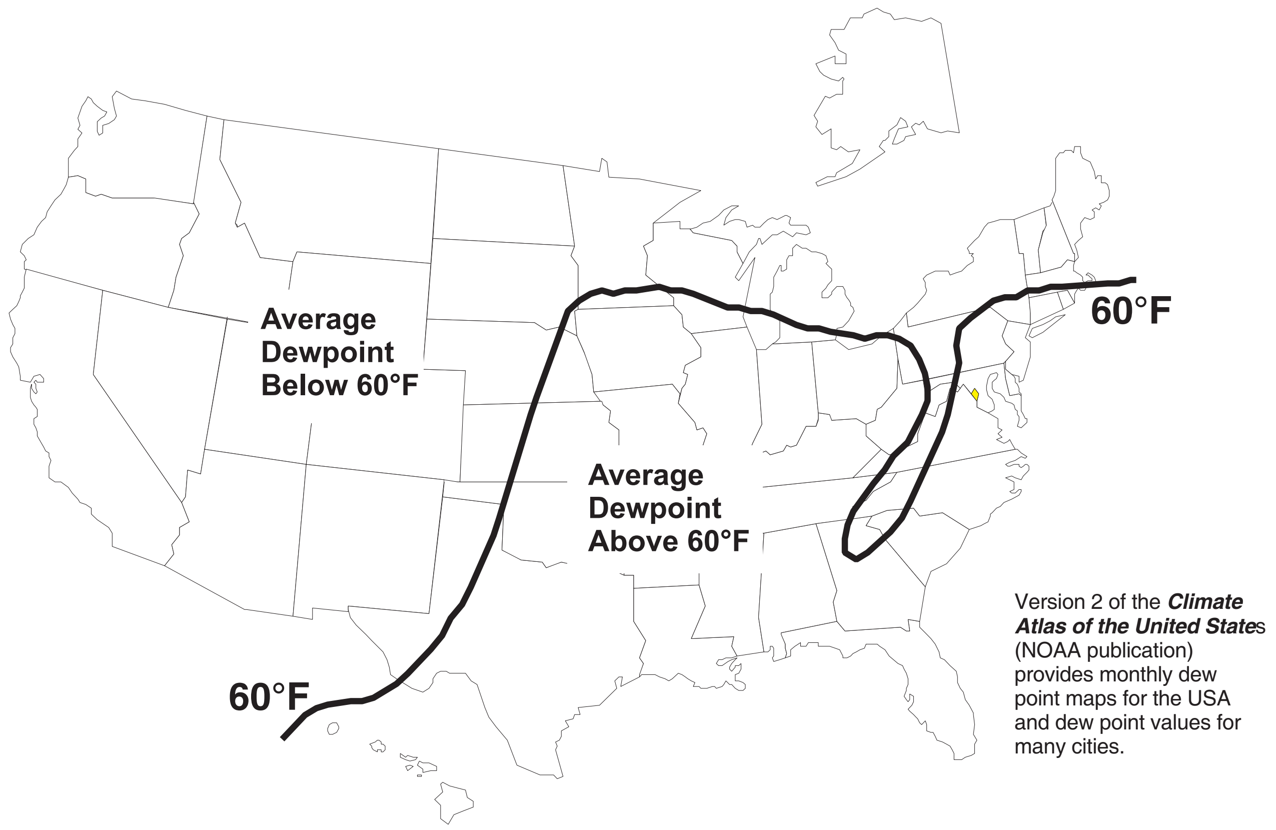


Figure A10-4

A10-5 Duct Sealing Requirements

Leakage is a function of the pressure in the duct, airway shape and size, duct length, construction details, sealing materials and workmanship. General comments pertaining to duct leakage are provided below. Refer to Appendix 6 for a list of codes and standards that specify minimum leakage rates, sealing requirements, sealing methods and sealing materials.

Location

There are rigorous standards for sealing ducts installed outdoors or in an unconditioned space. Relaxed standards apply to ducts located in the conditioned space.

Duct Pressure

Approved construction methods and materials depend on duct pressure. Residential low velocity duct systems operate at low pressure (± 2.0 IWC or less)

Duct Runs and Extended Plenums

Unsealed metal ducts, extended plenums and blower plenums leak at transverse and longitudinal seams, and at fitting connections. The amount of leakage depends on workmanship. Codes and industry standards specify metal gauges, fabrication techniques and sealing requirements.

Fibrous board ducts and plenums have potential leakage points at transverse and longitudinal seams and at fitting connections. Longitudinal seams should be reasonably tight if duct sections are assembled with approved methods and materials. Transverse seams should be sealed with the same care as longitudinal seams, but this is not always the case. Fitting connections may not be sealed at all. Codes and industry standards specify fabrication techniques and sealing requirements.

Sections of round spiral duct, round fibrous board duct, round flexible wire helix duct and plastic pipe are inherently tight. Transverse seams, plenums, fitting connections and flex duct junction boxes are the primary source of leakage. Codes and industry standards specify fabrication techniques and sealing requirements. Fire codes may not allow plastic in above grade locations.

- Plastic ducts are well suited for below-grade installations because they are watertight and because they do not rust or corrode.
- Local codes and utility regulations may conditionally restrict the use of certain types of duct materials.

Stud Spaces and Panned Joist Spaces

Stud spaces and panned joist spaces are commonly used as airways. Unfortunately, this method of solving duct routing problems produces leaky airways.

- There may be leakage paths to or from an attic, a basement, a crawlspace, or the outdoors.
- Stud spaces, panned joist spaces, structural chases and structural cavities should not be used as an airway unless the airway is completely sealed. This means sealing all related structural cracks, joints and penetrations, and sealing the joints and seams of the panning material.
- The National Fire Code and the International Building Code requires *no combustible material in the supply air or return air streams*. Even if a local inspector does not enforce this requirement, the national code is the minimum fire protection standard and by law. Therefore, practitioners should conform to "National Code" because they are held responsible if a fire happens.

Fittings and component Interface Connections

Leakage occurs at the joints and seams of branch takeoff fittings, elbows, tees, wyes, and component interface connections (regardless of the type of material that is used to fabricate the duct system).Leakage occurs where a duct run interfaces with an air handler. Leakage occurs at flexible duct junction box connections. Codes and industry standards specify sealing requirements for these connections.

Leakage at Register and Grille Flanges

Substantial leakage can occur at supply outlets and return grilles. (There are potential leakage paths to an attic, basement or other unconditioned spaces.) This leakage occurs when a duct, boot or transition box is not sealed to the frame of the register or grille. Codes and industry standards specify sealing requirements for the matting surface of a duct or duct fitting, and the flange or frame of a diffuser, register or grille.

Return and Discharge Plenums

Return plenums and discharge plenums at central air handling equipment tend to leak at the seams and joints. In some cases, an air handler is on top of a plenum that is fabricated out of wood framing and plasterboard.

- For a given crack size, leakage near the blower is maximized because the magnitude of duct pressure is larger than the pressure at some remote point in the duct system.
- Plaster board plenums tend to have serious leakage problems at seams and pipe penetrations.
- Plaster board plenums can be outrageously leaky if one or two of the plenum walls are part of a partition wall or exterior wall.
- If the partition or wall plaster board stops at the top of the plenum, the plenum is open to a wall cavity. Since this cavity extends from ceiling to floor, there may be leakage to the outdoors, an attic, a garage, a basement or a crawlspace.
- Plaster board plenums should be paneled and thoroughly sealed with suitable mastic.

Cabinet Leakage

Air handler cabinets leak at bent or missing panels, poor or damaged panel seals, cracks at the duct connection points, cabinet penetrations and knockout openings. Single package air handlers are completely exposed to the weather, which increases the potential for rust and corrosion that can creates leakage paths. Any type of cabinet leak is very important because the maximum pressure differences occur at the air handler. All air handler cabinet leaks should be sealed.

Equipment in Closet with Louvered Door

Leakage from an attic, basement, crawlspace or other unconditioned space may occur when the air handler is in a closet and uses a louvered door as a central return grille. If the free area of the louver opening is too small, the closet will be at negative pressure, causing leakage paths to adjacent spaces.

Methods, Materials and Workmanship

All seams and joints should be fabricated in accordance with industry standards. Tapes and mastics should be applied in accordance with industry standards. Tapes and mastics should be suitable for the intended application (regarding deterioration caused by aging, moisture and sunlight). There should be no outrageous oversights (missing duct runs, missing fittings, disconnections and large penetrations and openings that are not sealed). Relevant codes and standards are listed in Appendix 6.

Damage

Even if the duct system is properly installed, serious leakage problems can be caused by system abuse. This damage could be caused by other trades, occupants, pets and rodents. If damage potential is high, the system should be fabricated from suitable materials and protected by effective guards and shields at valuable points.

A10-6 Duct Leakage Estimates

Figure A10-5 and Figure A10-6 (next page) are used to estimate duct leakage. Figure A10-5 provides leakage class categories for various types of materials and

Figure A10-6 correlates leakage rate (Cfm per 100 square feet of duct surface area) with duct pressure (IWC).

The accuracy of a leakage estimate depends on selecting a leakage class value (CL) for the duct system. This is highly speculative for an unsealed system because the range of CL values is so large. Reasonable accuracy is expected if the duct system is fabricated and sealed to industry standards.

The accuracy of the leakage estimate also depends on the static pressure inside the duct. This is problematic because static pressure gradually changes in the direction of the flow and abruptly changes when the flow passes through a component. Therefore, when the duct pressure gradient is reasonably constant a along duct run, use the average pressure for the duct run.

- Separate leakage calculations are made for each side of the system.
- Pressure is positive on the supply-side of the system and negative on the return-side of the system, and the magnitude of these two pressures could be quite different.
- Some duct systems are fabricated from more than one material (a metal and duct board system, for example). In this case use a weighted average leakage class based on the surface areas of each type of material.

Leakage Class (CL)		
Duct System	**Sealed**[1]	**Unsealed**[6]
Fabricated Sheet Metal[2]	48	48 to 192
Fibrous Glass[3]	24	24 to 192
Wire Helix (Flexible)[4]	12	12 to 192
Metal—Round Spiral[5]	6	6 to 96
Plastic (Glued Joints)	3	NA

1) Traverse joints, fitting joints and boot-grille flanges.
2) Rectangular trunk and round runouts.
3) Rigid glass trunks and flexible or sheet metal runouts.
4) Flexible trunks, flexible runouts with junction boxes.
5) Round spiral trunks and round spiral runouts.
6) Based on measurements made during field tests. In extreme. cases (excessive use of building cavities and panning, deterioration, physical damage or disconnections) the leakage Cfm can be equal to 30 or 40 percent of the blower Cfm.

Figure A10-5

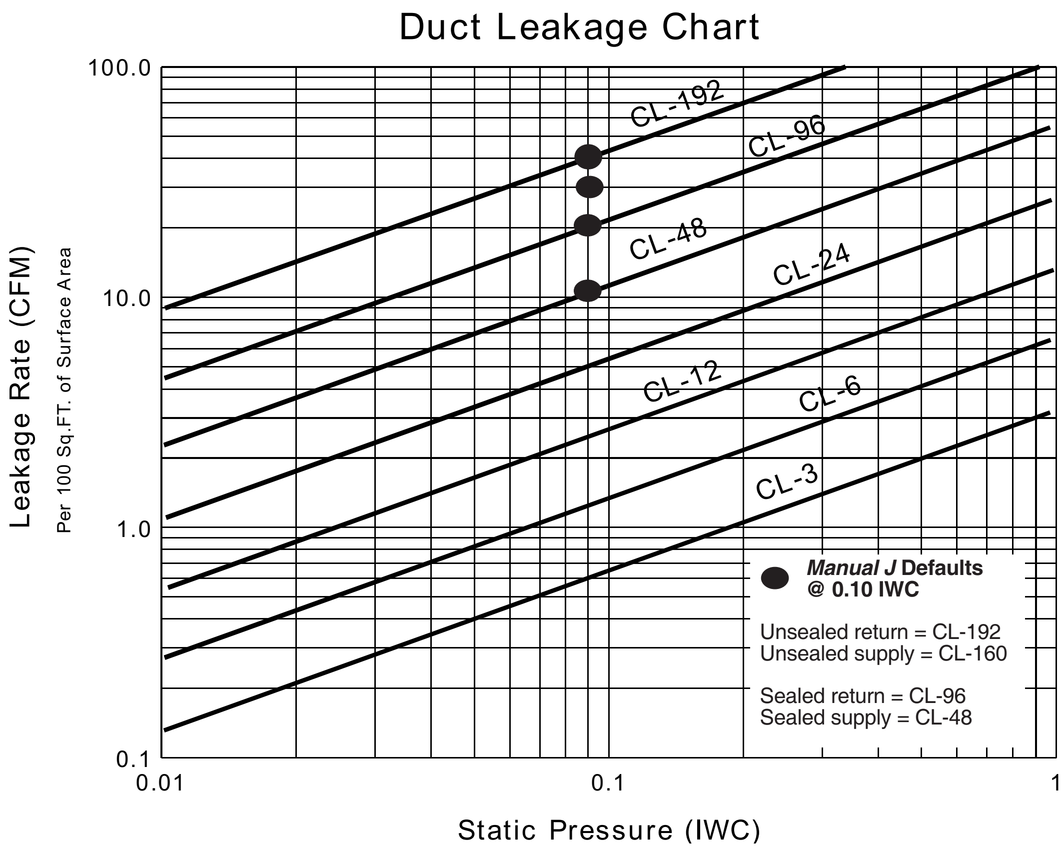


Figure A10-6

Example

Estimate leakage Cfm for a duct system in an unconditioned space. At 1,000 Cfm, blower pressure is 0.55 IWC, but 0.12 IWC is dissipated by ancillary components installed in the air handler cabinet. The total equivalent length of the longest supply run is 325 feet, and the total equivalent length of the longest return run is 120 feet. The surface area of the supply system is 430 SqFt (250 SqFt for rectangular trunks and 180 SqFt for round runouts), and there is 120 SqFt of return duct area. All of duct runs are sheet metal and all transverse seams are sealed.

Leakage class = 48
External pressure = 0.55 - 0.12 = 0.43 IWC

$$\text{Starting supply pressure} = \frac{0.43 \times 325}{325 + 120} = 0.31 \text{ IWC}$$

Average supply-side pressure = 0.31 / 2 = 0.155 IWC
Starting pressure on return-side = 0.43 - 0.31 = 0.12
Average return-side pressure = 0.12 / 2 = 0.06 IWC

Supply-side leakage rate = 15 Cfm per 100 SqFt
Return-side leakage rate = 8 Cfm per 100 SqFt

$$\text{Supply-side leakage} = \frac{15 \times 430}{100} = 64.5 \text{ CFM}$$

$$\text{Return-side leakage} = \frac{8 \times 200}{100} = 9.6 \text{ CFM}$$

Total leakage = 64.5 + 9.6 = 74.1 Cfm
Percent of blower Cfm = 74.1 / 1,000 = 0.074 = 7.4%

A10-7 Leakage Loads

Field studies for all parts of the USA, show that duct leakage has an adverse effect on the efficiency of residential comfort systems and building envelope leakage. This translates to more purchased energy, more demand on the electric, or gas grid and increased cost for heating and cooling a dwelling.

Infiltration Loads Depend on Blower Operation

Duct leakage increases equipment run time and blower operating hours. This increases infiltration, because envelope leakage increases when the blower operates. Figure A10-7 shows how blower operation affects the infiltration rate when leaky ducts are located in an unconditioned space.

Loads Induced by Duct Leakage

More energy is required to heat, cool and dehumidify when duct leakage increases the infiltration load and the duct load. Heating load, sensible cooling load and latent cooling load are affected.

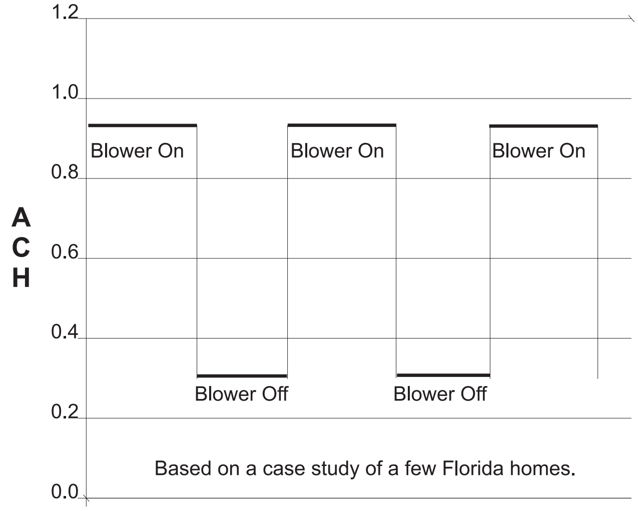


Figure A10-7

- If duct leakage causes negative space pressure, outdoor air enters through cracks in the building envelope.
- If duct leakage causes positive space pressure, ambient air enters through cracks in the return duct.
- Ambient conditions range from hostile (roof, attic, outdoors or open crawlspace) to benign (enclosed crawlspace or basement).
- The interaction between duct leakage, the pressure in the occupied space and infiltration is discussed in Appendix 11.

The size of the leakage-induced infiltration load (and the amount of wasted energy) depends on leakage Cfm and the condition (temperature and moisture) of the leaving air and entering air.

- Conditioned air is expelled from the envelope or supply duct.
- Replacement air is drawn from outdoors or an ancillary space.

Examples

The following examples show that large heating and cooling loads are produced by 100 Cfm of duct leakage. (Field tests have found many homes that have more than 100 Cfm of duct leakage. Some homes had leakage rates of 300 to 400 Cfm, or more!)

Example 1

Calculate the cooling and heating loads for leaky supply ducts in an open crawlspace. Base the calculations on supply-side leakage that produces negative space pressure and increases envelope infiltration by 100 Cfm. Make this calculation for a hot humid day when the condition of the supply air is 55°F and 60 Gr/Lb, and the condition of the outdoor air is 90°F and 90 Gr/Lb; and on a cold winter day when the temperature of the heated air is 105°F and the temperature of the outdoor air is 20°F.

Sensible load = 1.1 x 100 x (90-55) = 3,850 Btuh
Latent load = 0.68 x 100 x (90-60) = 2,040 Btuh
Total cooling load = 5,890 Btuh
Total heating load = 1.1 x 100 x (105-20) = 9,350 Btuh

Example 2

Calculate the cooling and heating loads for leaky return ducts in an open crawlspace. Base the calculations on return-side leakage that produces positive space pressure and produces 100 Cfm of envelope exfiltration. Make this calculation for a hot humid day when the condition of the return air is 75°F and 65 Gr/Lb, and the condition of the outdoor air is 90°F and 90 Gr/Lb; and on a cold winter day when the temperature of the return air is 70°F and the temperature of the outdoor air is 20°F.

Sensible load = 1.1 x 100 x (90-75) = 1,650 Btuh
Latent load = 0.68 x 100 x (90-65) = 1,700 Btuh
Total cooling load = 3,350 Btuh
Heating load = 1.1 x 100 x (70-20) = 5,500 Btuh

Example 3

Calculate the cooling and heating loads for leaky return ducts in an attic. Base the calculations on return-side leakage that produces positive space pressure and produces 100 Cfm of envelope exfiltration. Make this calculation for a hot humid day when the condition of the return air is 75°F and 65 Gr/Lb, and the condition of the attic air is 130°F and 90 Gr/Lb; and on a cold winter day when the temperature of the return air is 70°F and the temperature of the attic air is 20°F.

Sensible load = 1.1 x 100 x (130-75) = 6,050 Btuh
Latent load = 0.68 x 100 x (90-65) = 1,700 Btuh
Total cooling load = 7,750 Btuh
Heating load = 1.1 x 100 x (70-20) = 5,500 Btuh

Example 4

Calculate the cooling and heating loads for a leaky duct system in an attic. Base the calculations on 100 Cfm of supply-side leakage and 100 Cfm of return-side leakage (which has no effect on space pressure). Make this calculation for a hot humid day when the condition of the supply air is 55°F and 60 Gr/Lb, and the condition of the attic air is 130°F and 90 Gr/Lb; and on a cold winter day when the temperature of the supply air is equal to 105°F and the temperature of the attic air is equal to 20°F.

Sensible load = 1.1 x 100 x (130-55) = 8,250 Btuh
Latent load = 0.68 x 100 x (90-60) = 2,040 Btuh
Total cooling load = 10,290 Btuh
Total heating load = 1.1 x 100 x (105-20) = 9,350 Btuh

A10-8 Duct Loads

The full version of the Eighth Edition of ***Manual J*** (version 2.10 or later) provides duct load tables for various locations and types of duct systems. These tables provide duct load factors for sensible heating and sensible cooling and latent load values for latent cooling, which depend on duct wall R-value, duct leakage and duct surface area. When processed by ***Manual J*** procedures they appear as a heating load, a sensible cooling load and a latent cooling load on the ***Manual J*** load summary form (Form J1). Relevant output and sensitivities are listed here:

- Duct load for heating (Btuh).
- Duct load for sensible cooling (Btuh).
- Duct load for latent cooling (Btuh)
- Duct insulation R-value = 2.0, 4.0, 6.0 and 8.0.
- Supply leakage (Cfm per SqFt duct surface) = 0.06, 0.09, 0.12, 0.24, 0.35.
- Return leakage (Cfm per SqFt duct surface) = 0.06, 0.15, 0.24, 0.47, 0.70.
- The defaults for a sealed duct system are 0.12 supply and 0.24 return with R-6 insulation.
- The default for unsealed duct system are 0.35 supply and 0.70 return with R-6 insulation.
- The defaults for supply-side surface area and return-side surface area are listed on the duct table.
- Sensible load factors and the latent load value can be adjusted for installed surface area.
- ***Manual J*** procedures process other combinations of R-value, leakage and surface area.
- See ***Manual J***, Section 23 and Table 7.

A10-9 Supply Duct Load Reduces Delivered Capacity

For ***Manual J*** calculations, the supply-side (sensible) duct load is part of the block load for equipment sizing. In addition to increasing the load on the equipment, supply duct losses reduce the delivered capacity of supply air discharged from heating-cooling equipment.

- Duct leakage reduces the Cfm delivered to the space, and duct wall conduction reduces supply air temperature for heating and increases supply air temperature for cooling.
- The net capacity for the conditioned space equals the capacity of the supply air discharged from the heating or cooling equipment, minus the capacity

loss for the combined effect of duct leakage and supply air temperature change.

- When ***Manual J***, ***Manual D*** and ***Manual S*** procedures are used, the capacity delivered to the conditioned space will maintain comfort conditions in the room or space. (Supply air discharged from the equipment has enough capacity for the duct load and the space load.)

A10-10 Return Duct Load Affects Refrigeration Equipment Capacity

For ***Manual J*** calculations, the return-side (sensible and latent) duct loads are part of the block load for equipment sizing. In addition, the return-side duct loads may significantly affect the performance of refrigeration cycle equipment.

- The capacity of refrigeration cycle equipment depends on the condition of the air entering the equipment. For heating, the temperature of the air entering the equipment is colder than space return air. For cooling, the temperature of the air entering the equipment is warmer than space air, and it may have more or less moisture than space air.
- A return duct load has a small affect on heat pump heating capacity, as documented in manufacturer's performance data. This affect may be ignored, because it marginally affects balance point calculations (second stage heat makes up for the small loss of heat pump capacity).
- The sensible and latent capacity of cooling equipment depends on the dry-bulb temperature and wet-bulb temperature of the air entering the equipment.
- For cooling, a return duct heat gain increases entering dry-bulb temperature.
- For humid climate cooling, a return duct moisture gain increases the wet-bulb temperature of the air entering the cooling equipment.
- For dry climate cooling, a return duct moisture loss reduces the wet-bulb temperature of the air entering the cooling equipment, but this is usually not an issue of concern.
- A return duct load has no significant affect on furnace heating capacity.

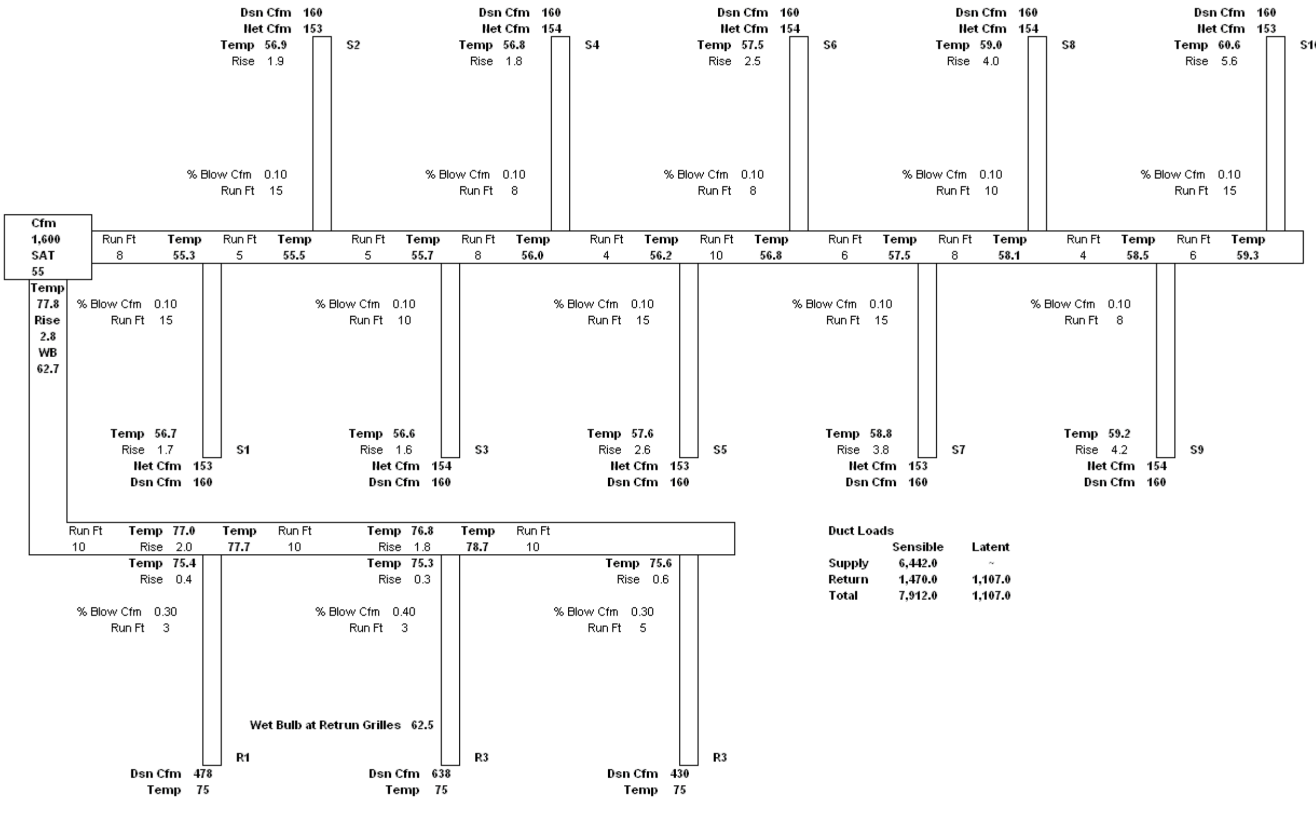


Figure A10-8

- For heating with any type of equipment, a return duct load reduces discharge air temperature (the temperature rise produced by heating equipment is added to a temperature that is colder than the temperature or the air return from the space).
- When ***Manual J***, ***Manual D*** and ***Manual S*** procedures are used, the installed equipment capacity will maintain comfort conditions in the room or space. (The equipment has adequate capacity for the duct loads and the space loads.)

A10-11 Illustrative Example of Duct System Loads

Figure A10-8 (previous page) shows a simple attic duct system that has 520 SqFt of supply duct surface area and 226 SqFt of return duct surface area. The heat pump blower delivers 1,600 Cfm to the duct system. The ducts are sealed, but the workmanship is average (***Manual J*** leakage is 0.12 Cfm/SqFt for the supply ducts and 0.24 Cfm/SqFt for return ducts). All duct walls have R6 insulation. All duct runs have a 2:1 aspect ratio. The ambient attic temperature is 120°F. The altitude of the location is close to sea level (this affects air temperature calculations). For simplicity, all supply branches deliver ten percent of the system air flow.

- The sensible supply duct load is 6,442 Btuh, the sensible return duct load is 4,842 Btuh, the total sensible duct load is 11,284 Btuh, and the latent return duct load is 1,107 Btuh.
- Figure A10-8 shows the cooling supply air temperatures at each supply outlet and the temperature of the air entering the heat pump. Notice that the air leaves the heat pump at 55°F and leaves the S10 supply outlet at 60.6°F. Notice that the return air enters the return grilles at 75°F and enters the heat pump at 77.8°F.
- Figure A10-8 shows the latent load for return ducts produces a 62.7°F entering wet-bulb temperature at the heat pump duct (the air entered the return grilles at 62.5°F wet-bulb.)
- Figure A10-8 shows the airway sizing Cfm values (Dsn Cfm) and the net Cfm values for the supply outlets (for this example, the system is balanced so that each supply outlet is penalized for 10% of the trunk duct leakage).

Other combinations of duct insulation R-value and duct sealing were investigated for the Figure A10-8 duct system. The results of this effort are summarized by Figure A10-9. This information confirms what the industry already knows… adequate duct insulation and careful duct sealing have a dramatic affect on duct system performance.

Duct Performance Summary for Illustrative Example

R	Seal	SAT °F	EAT °F	EWB °F	Sens Btuh	Lat Btuh
0	Tight	79.9	81.2	62.6	36,337	415
	Avg		82.2	62.7	38,117	1,107
	Loose		85.0	63.1	43,892	3,228
2	Tight	67.0	78.1	62.6	17,685	415
	Avg		79.1	62.7	19,682	1,107
	Loose		82.1	63.1	26,752	3,228
4	Tight	62.5	77.1	62.6	11,709	415
	Avg		78.1	62.7	13,774	1,107
	Loose		81.2	63.1	21,260	3,228
6	Tight	60.6	76.7	62.6	9,189	415
	Avg		77.8	62.7	11,284	1,107
	Loose		80.8	63.1	18,945	3,228
8	Tight	59.1	76.4	62.6	7,368	415
	Avg		77.5	62.7	9,483	1,107
	Loose		80.5	63.1	17,172	3,228

1) R = R-value of duct wall insulation.
Tight = 0.06 Cfm/SqFt for supply and 0.06 Cfm/SqFt for return.
Avg = 0.12 Cfm/SqFt for supply and 0.24 Cfm/SqFt for return.
Loose = 0.35 Cfm/SqFt for supply and 0.7 Cfm/SqFt for return.
2) Avg = ***Manual J*** default for sealed, and loose = ***Manual J*** default for not sealed.
3) SAT = Supply air temperature at outlet 10; EAT = dry-bulb temperature of air entering heat pump; EWB = wet-bulb temperature of air entering heat pump; Sens = total sensible load for supply and return ducts; Lat = latent load for return ducts.

Figure A10-9

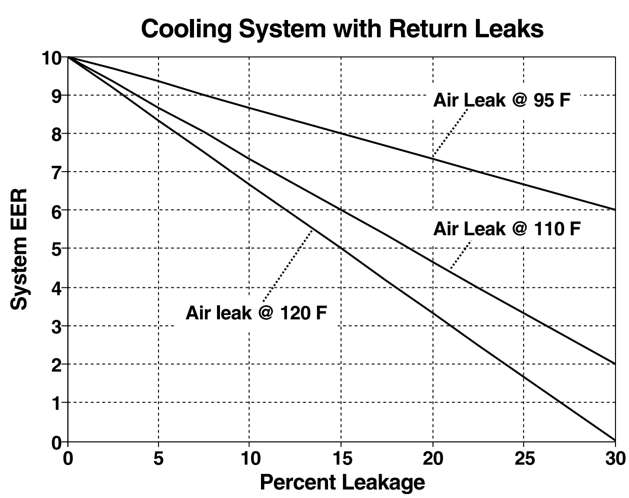


Figure A10-10

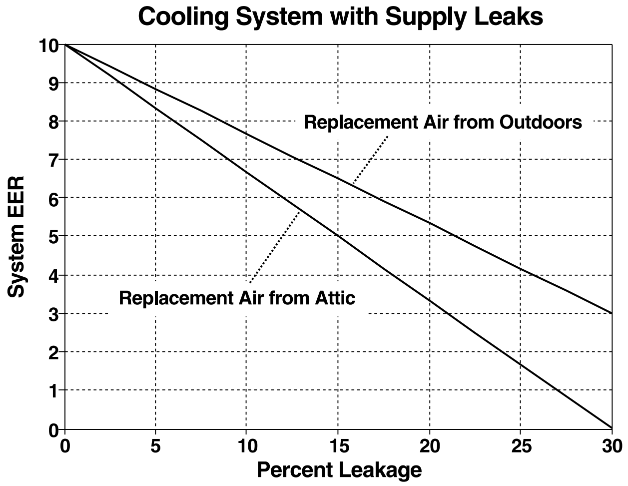


Figure A10-11

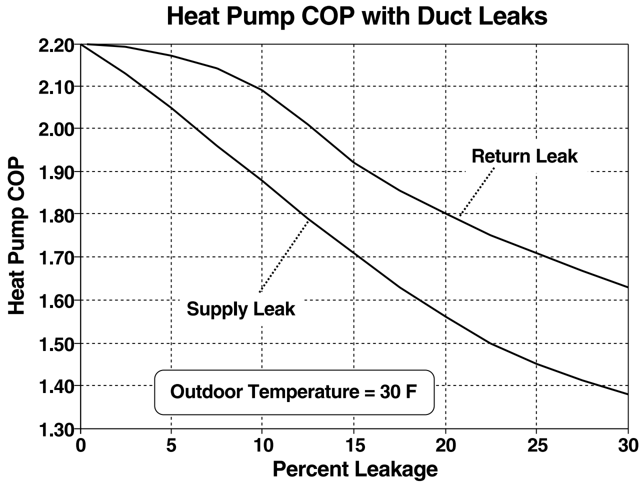


Figure A10-12

A10-12 Efficiency, Operating Cost and Demand Load

Duct leakage can affect the efficiency of the heating and cooling equipment, the building envelope and the overall efficiency of the integrated system. Duct leakage also affects operating cost and demand load.

Cooling System Efficiency

Figure A10-10 (previous page) relates cooling equipment efficiency to return duct leakage. This shows how the seasonal efficiency is affected by return-side leaks. Figure A10-11 relates cooling equipment efficiency to supply duct leakage. This shows how the seasonal efficiency is affected by supply-side leaks. In this case, seasonal EER values are for the overall efficiency of the integrated system (envelope, ducts and equipment). These efficiencies are based on required equipment output energy with no duct leaks and required equipment input energy with duct leaks.

Effect on Refrigeration Cycle Efficiency

Return-side leaks have a significant effect on the condition of the air that enters a refrigerant coil, which affects the condensing unit efficiency (Figure A11-14, next page). And, mechanical efficiency is reduced when return-side leaks draw air that contains dirt, dust and pollen. This clogs filters, fouls coils and covers blower wheel surfaces. (A clogged return air filter-grille exacerbates return-side leakage problems because more air is drawn from the unconditioned space as less air flows through the filter.)

Effect on Heat Pump Efficiency (Heating)

For an increased infiltration load, wasted energy depends on equipment operating mode and the efficiency of the component that provides the increment of capacity used to condition replacement air. For example, the efficiency of a heat pump depends on the use of second-stage heat, which depends, in part, on the size of the infiltration load caused by duct leakage.

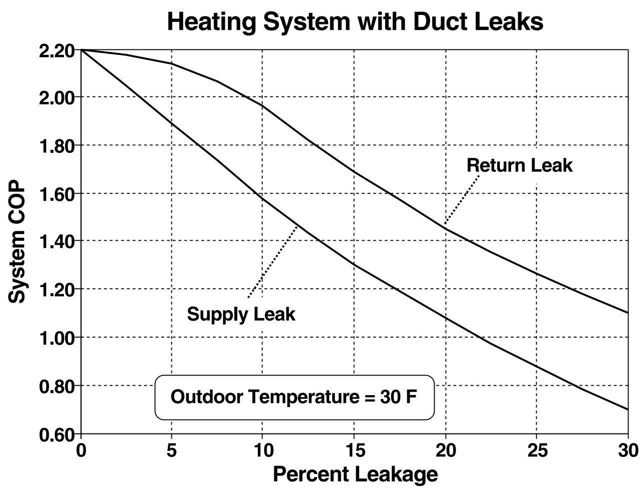


Figure A10-13

Figure A10-12 shows how duct leakage affects the coefficient of performance (COP) of a heat pump unit (refrigeration machinery and electric resistance heating coil). Note that the efficiency of the heat pump deceases as duct leakage increases. This occurs because duct leakage produces a larger heating load, which causes an upward shift in the balance point of the heat pump equipment. This shift means that more second-stage heat (COP = 1.0) is used during then heating season.

Figure A10-13 shows another way to think about duct leakage and heating efficiency. For this diagram, COP values are not for the mechanical efficiency of a heat pump. These COP values quantify the overall efficiency of the integrated system (envelope, ducts and machinery). This index is based on output energy for no duct leakage and the input energy with duct leakage. (For a given amount of leakage, Figure A10-12 efficiencies are lower than Figure A10-13 efficiencies.)

Operating Cost

Operating cost is directly related to energy use. When duct leakage causes an increase in the equipment load, or a reduction in the equipment efficiency, there is a corresponding increase in operating cost. The size of the increase depends on the size of the added load, the efficiency reduction for the equipment that satisfies the added load and the marginal cost of the additional increment of power or fuel.

Demand Loads

Utility demand load increases in proportion to the collective instantaneous inefficiency of the comfort systems connected to the grid. This efficiency is affected by duct leakage.

- Heating and cooling loads peak during extreme weather (***Manual J*** design conditions or worse).
- When equipment loads peak, blowers operate continuously (or most of the time), infiltration loads increase, duct leakage increases, equipment loads increase, and equipment efficiency may be affected.
- When blowers operate continuously, the collective inefficiency of the load-side of the utility grid increases.
- Duct leakage loads produce a significant and avoidable demand on the utility service.

Figure A10-14 shows how duct leaks affect the demand load produced by a air-air heat pump during a cold winter day in Northern Florida. This diagram shows that KW demand is reduced by 30 percent if duct leakage is reduced by 70 percent. (A larger reduction in demand is expected for a similar dwelling in a colder climate.)

Figure A10-15 shows how duct leaks affect the demand for an air-air heat pump system during a hot summer day in Northern Florida. This diagram shows that KW demand is reduced by 25 percent if duct leakage is reduced by 70 percent.

A10-13 Figure of Merit

A figure of merit (FOM) is used to rate the efficiency of an air distribution system. This index is calculated by dividing the energy that would be used if there was no duct system by the energy that would be used if there is a duct system.

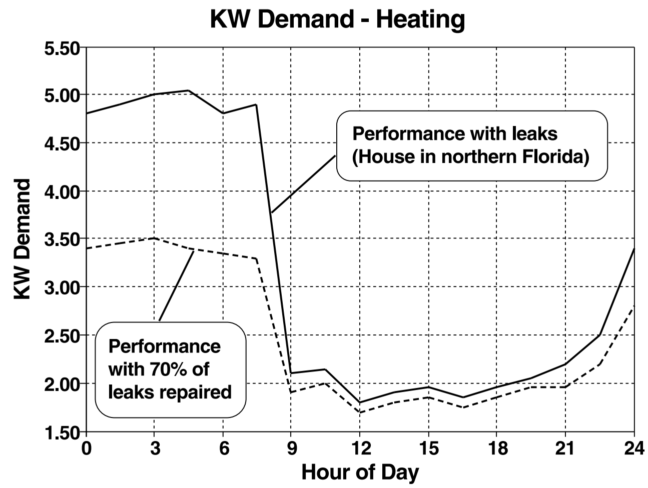


Figure A10-14

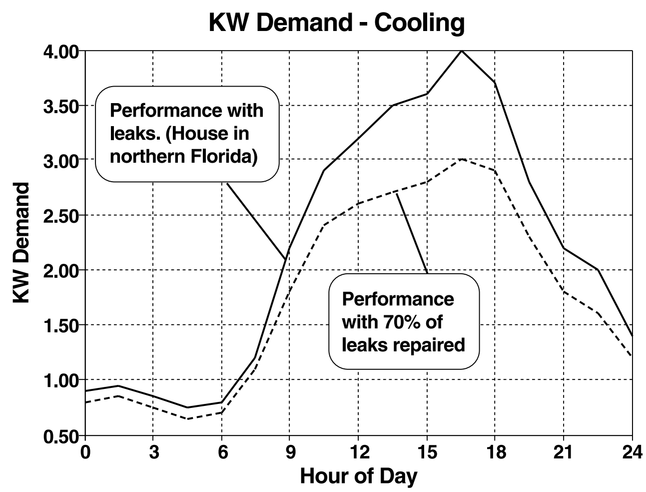


Figure A10-15

$$FOM = \frac{\text{Conditioning energy without ducts}}{\text{Conditioning energy with ducts}}$$

As explained by sections A10-1 through A10-11, many factors affect the figure of merit. Primary items are blower power, supply-side conduction losses, return-side conduction losses, supply-side leakage losses, return-side leakage losses, reductions in equipment efficiency and changes (increase or decrease) in the infiltration load. The effect of these factors is more apparent if the figure of merit equation takes this form:

$$FOM = \frac{(MEF)_{with}}{(MEF)_{no}} \times \frac{(Envelope)_{no}}{(Envelope)_{with} + DLL + DWCL}$$

- $(MEF)_{with}$ is the mechanical efficiency of the fuel conversion equipment when used with a distribution system. This factor includes blower energy and the effect that leakage and conduction losses have on the annual efficiency of the equipment. (Return-side leakage and conduction losses affect equipment efficiency because they alter the condition of the air entering the equipment.)
- $(MEF)_{no}$ is the mechanical efficiency of the fuel conversion equipment with no air distribution system.
- $(Envelope)_{no}$ is the annual energy required for the structure with no air distribution system.
- $(Envelope)_{with}$ is the annual energy required for the structure that has an air distribution system. This factor includes infiltration load adjustments for supply-side and return-side leakage.
- DLL is the energy for loads produced by supply-side and return-side leakage.
- DWCL is the energy for duct wall loads produced by supply-side and return-side conduction.

For example, calculate the FOM value if the annual COP (heating and cooling combined) of the equipment is 2.1 without a duct system and 1.9 with a duct system; and the annual envelope energy requirement is 300 million Btu/Yr without a duct system and 315 million Btu/Yr with a duct system; and duct leakage losses add 20 percent to the annual energy load; and the duct conduction losses add 10 percent to the annual energy load.

$$FOM = \frac{1.9}{2.1} \times \frac{300}{(315 + 60 + 30)} = 0.67$$

Appendix 11 (Informative; not Part of the Standard)

Duct Leakage and System Interactions

Duct leakage and return path restrictions affect the efficiency of the duct system, the performance of the building envelope, the efficiency and effectiveness of the HVAC equipment, the capacity of the exhaust equipment, and the power of the vents for fuel burning components. In most cases these effects are interactive. This appendix discusses these relationships.

A11-1 Complex Systems

An air distribution system can be very simple, as far as cause-and-effect relationships are concerned, providing that the duct system is continuous, tight, well insulated and only has one inlet and one outlet. Figure A11-1 shows this type of system.

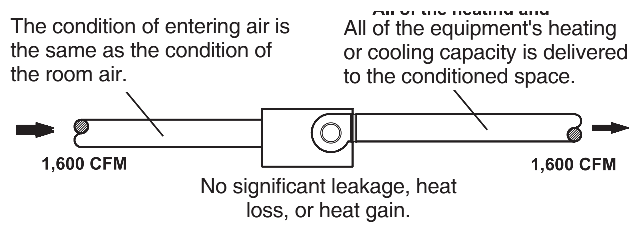


Figure A11-1

As far as the air flow is concerned, the operating point of this simple duct system is completely defined by blower performance (flow versus resistance curve) and duct performance (resistance versus flow curve), as indicated by Figure A11-2. This operating point is obvious because there is only one duct run and because there is no interaction between the duct system, the building envelope or any other system that serves the dwelling.

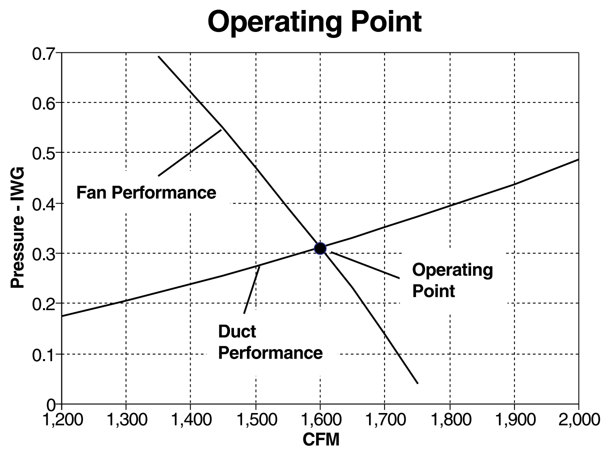


Figure A11-2

Figure A11-3 (next page) shows a complex system that consists of the building envelope, the HVAC equipment, a manifold duct system, a discontinuous return path, exhaust equipment, appliances and a fireplace. This system is characterized by the interaction, coupling and connections of the various subsystems. In this case, the performance of one subsystem depends on the performance of the other subsystems.

The common thread that ties all the systems together is pressure. For example, pressure differences affect infiltration rate, duct leakage, supply Cfm values, return Cfm values, flues and vent performance, combustion appliances, fireplace performance and exhaust equipment performance.

A11-2 Pressure Differences

Relevant pressure differences include the indoor-to-outdoor difference, room-to-room differences; the difference between a conditioned space and an unconditioned or, ancillary space; and the difference between space pressure and the pressure in a duct run. These pressure differences are affected by envelope leakage areas, wind velocity, the height of the structure, flues and vents, exhaust equipment, some types of appliances, the operating mode of the blower (on-off), duct leakage areas, leakage areas for interior partitions and doors, and the continuity of the return air system. These pressure differences are quite small, usually a few Pascals (25 Pascals is approximately 0.10 IWC), and they are not constant. Therefore, as various pressures fluctuate, infiltration rates and air flow rates may increase, or decrease, or they may have a binary nature, appearing when certain conditions exist and disappearing when these conditions change. More information about the various factors that affect the pressure in a conditioned space is provided here:

Envelope Leakage Areas

The envelope leakage areas (seams, holes and cracks) determine the tightness of the dwelling. The size and location of these openings affect infiltration, exfiltration and the ability of the envelope to hold pressure.

Wind

The wind causes infiltration. Wind blowing on an exterior surface produces a high pressure at that surface. At

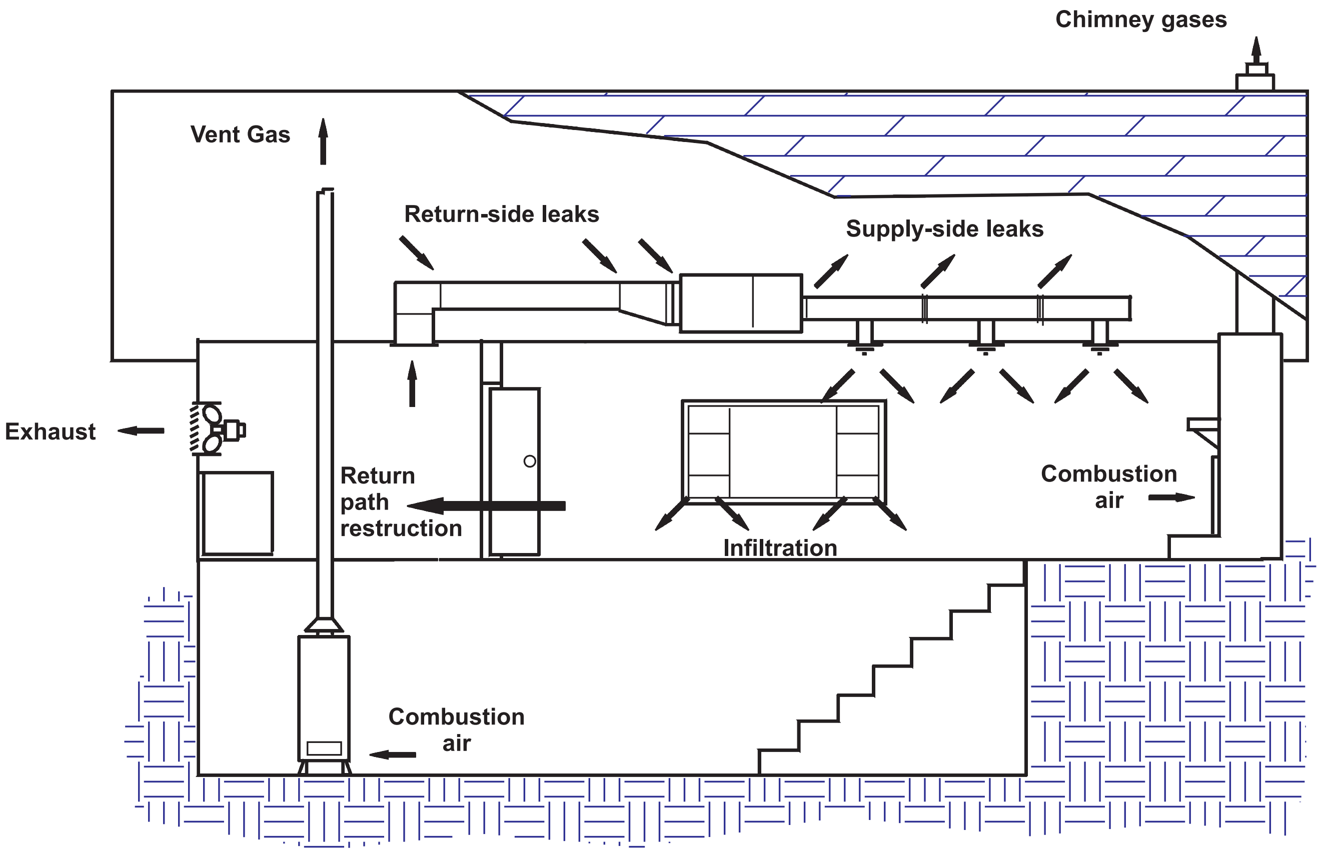


Figure A11-3

the same time, a low pressure occurs on the leeward side of the structure. This condition can cause a positive or negative pressure in the dwelling, depending on wind direction and the relative location of the leakage areas.

- If the wind blows toward a large amount of the crack area and if there is negligible leeward crackage, the space is pressurized and indoor air leaks out through neutral cracks.
- If there is negligible windward crack area and a large amount of leeward crack area, indoor air is pulled through leeward cracks, the space is depressurized and outdoor enters through neutral cracks.
- If the crack area for a single-family detached home is evenly distributed around the perimeter of the structure, the pressure in the dwelling is usually slightly negative.
- Tests on single-family Florida homes indicate that indoor pressures typically range from 0 to -4 Pascals, depending on wind velocity.
- Figure A11-4 shows the pressure conditions that are likely to occur when the wind acts alone.

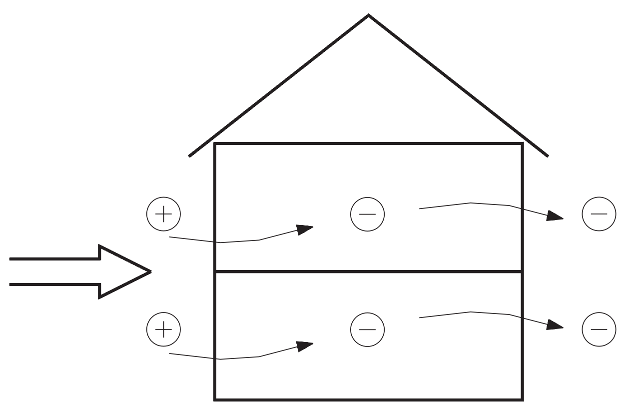


Figure A11-4

Buoyancy Effect

The height of the structure determines the power of the buoyancy effect; therefore the effect is more important for a multistory dwelling. When the buoyancy effect acts alone, buoyancy forces produce a pressure difference between the lower level and the upper level. Depending on the height of the structure and outdoor air temperature, the pressure difference can range from 0 Pascals to more than 8 Pascals.

During winter, the pressure is positive at the upper level (0 to +4 Pascals or more, with respect to outdoors) and negative at the lower level (0 to -4 Pascals or less, with respect to outdoors). This pressure difference causes air to flow through cracks distributed around the lower level and out of the cracks distributed around the upper level. This situation is reversed in the summer if the dwelling is air conditioned, but pressure difference are smaller because indoor-outdoor temperature difference is smaller. Figure A11-5 shows pressure conditions when the buoyancy effect acts alone during winter.

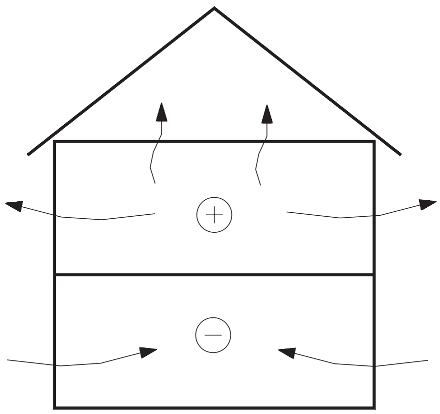


Figure A11-5

Flues and Vents

When a vent acts alone, buoyancy forces produce a pressure difference between the top of the vent and the inside of the dwelling. This causes a negative pressure in the conditioned space. Draft pressure depends on the height of the vent and the temperature difference between the flue gas and the outdoor air. When heating equipment operates on cold days, draft pressure can be quite small, perhaps as small as -5 Pascals; and when the heating equipment is off, the draft pressure is even smaller. This negative pressure causes air to flow through all the cracks in the building envelope. Figure A11-6 shows the pressure conditions that are created when the vent effect acts alone.

Exhaust Equipment and Appliances

Exhaust fans and some appliances, such as clothes dryers, produce a negative pressure in the dwelling. Small exhaust fans and clothes dryers produce negative pressures that range from -1 to -6 Pascals. Large exhaust fans can produce negative pressures that can exceed -30 Pascals. An all cases, negative pressure causes air to flow through all the cracks in building envelope and negative space pressure can cause venting problems if it overpowers the natural draft produced by a vent, stack or chimney. Figure A11-7 shows the pressure conditions that are created when the exhaust effect acts alone.

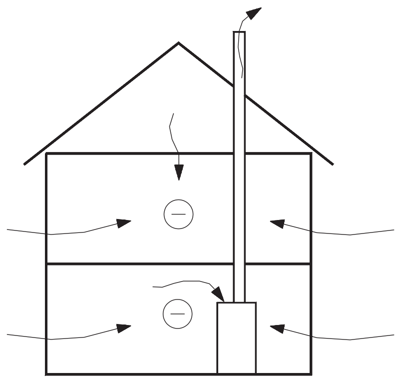


Figure A11-6

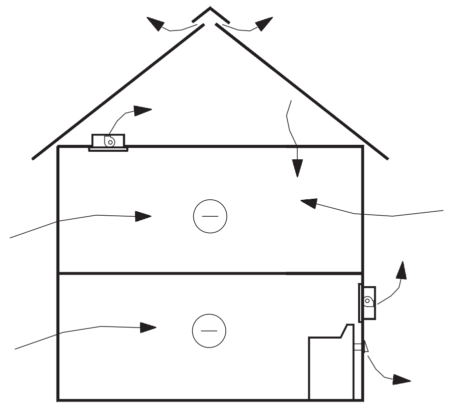


Figure A11-7

Blower Operation

Blower operation causes substantial pressure fluctuations at various points in a complex system. When the blower is on, it pressurizes supply-side ducts and depressurizes return-side ducts. Blower operation may affect the pressure in the conditioned space.

- Discounting duct leakage; a conditioned space is pressurized if it is decoupled from a return air path (no return or transfer grille); and some other part of the conditioned space will be at negative pressure.
- If leaky ducts are installed in an ancillary space, the pressure in the conditioned space can be positive or negative.

The positive pressure inside the supply-side of a duct system typically ranges from 0.10 to 0.30 IWC. (25 to 75 Pascals). The negative pressure inside the return-side of a duct system typically ranges from -0.10 to -0.30 IWC. (-25 to -75 Pascals). The pressures in a conditioned space or an ancillary space usually range from 0 to 25 Pascals, depending on the effectiveness of the return system, the tightness of the duct runs and the tightness of the space. The various pressures and pressure differences produced by blower operation affect duct leakage, envelope leakage, and the performance of supply outlets, flues, vents and chimneys.

Duct Leakage

When duct runs are exposed to outdoor air (attic, open crawlspace or vented cavity), leakage can produce a positive pressure, or a negative pressure, in the conditioned space. Return-side leaks cause positive space pressure, and supply-side leaks cause negative space pressure.

- Air that flows through return-side leaks enters the dwelling with the supply air and exfiltrates through the envelope crackage. As far as the conditioned space is concerned, supply Cfm exceeds return Cfm, so the space is pressurized.
- Air that flows through supply-side leaks reduces the amount of air that is delivered to the conditioned space. As far as the conditioned space is concerned, return Cfm exceeds supply Cfm, so the space is depressurized.
- If there are leaks on both sides of the system, space pressure is positive if return-side leaks dominate; space pressure is negative if supply-side leaks dominate, and space pressure is neutral if supply-side leakage equals return-side leakage.
- Figure A11-8 shows the pressure conditions for dominate return-side leakage. Figure A11-9 shows the pressure conditions for dominate supply-side leakage.

When there is dominate leakage, the amount of positive pressure or negative pressure in the conditioned space depends on the unbalanced leakage Cfm and on the tightness of the structure. If supply-side leaks dominate, space pressure can range from -1 to -6 Pascals (or lower). If return-side leaks dominate, space pressure can range from +1 to +6 Pascals (or higher). Figure A11-10 summarizes this behavior.

When the entire duct system is in the conditioned space, duct leakage can produce a positive or negative pressure in the area where the duct is located. Return-side leaks produce a negative space pressure and supply-side leaks produce a positive space pressure. The amount of pressure in a particular area depends on local leakage Cfm and the tightness of the room. Local pressures can range from +8 to -8 Pascals or more.

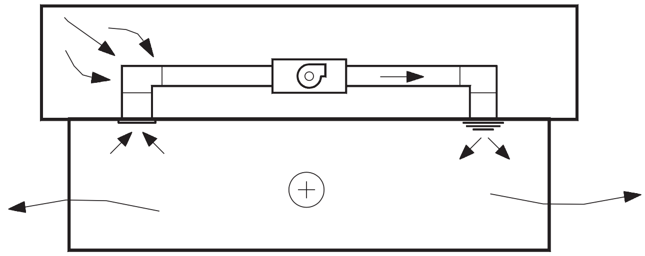


Figure A11-8

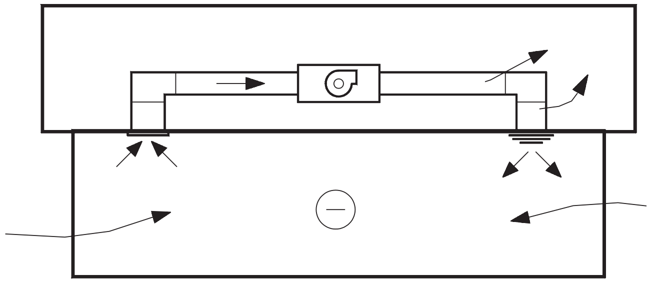


Figure A11-9

Duct Leakage versus Pressure in Space				
Envelope Tightness	**Return Leakage**	**Supply Leakage**	**Air Balance**	**Space Pressure**
Tight	250	0	+ 250	+ 6
Loose	250	0	+ 250	+ 2
Tight	250	100	+ 150	+ 3
Loose	250	100	+ 150	+ 1
Tight	250	250	0	0
Loose	250	250	0	0
Tight	100	250	- 150	3
Loose	100	250	-150	-1
Tight	0	250	-250	-6
Loose	0	250	-250	-2

Figure A11-10

Discontinuous Return Path

If there are no other factors involved, the pressure inside any room and the pressure in the core of the dwelling depend on return air path restrictions. If there is a low-resistance return air path for every room, supply Cfm and return Cfm are equal for every room, so space pressures are neutral. An adequate return path is established by providing a ducted return for every room or space, or by using one or more central returns (in the core areas) supplemented by adequate transfer grilles (which provide paths to central returns).

If one or more rooms are isolated from a central return (by tight-fitting interior doors which are closed), the pressure in the isolated room will be positive and the pressure in the core of the dwelling will be negative. The positive pressure in an isolated room depends on the supply Cfm, the tightness of the duct run and the tightness of the room. When there are no other factors involved, a positive pressure of a few Pascals to more than 10 Pascals is possible. (If a room is perfectly sealed, space pressure will equalize with the pressure in the branch supply air duct and there will be no air flow to the room.)

The negative pressure produced in the core area depends on the difference between the return Cfm and the supply Cfm; the tightness of the surrounding walls, interior partitions, windows and doors; and on the tightness of the return ducts. When there are no other factors involved, a negative pressure of a few Pascals to more than -10 Pascals is possible.

Sometimes a primary return air path is restricted by a filter-grille that has a clogged filter. In this case, the return-side duct leakage is dramatically increased and this increases the pressure in most rooms or the entire conditioned space. In general, the pressure changes produced by an inadequate return path affect duct leakage, envelope leakage and the performance of supply outlets, flues, vents and chimneys.

A11-3 Synergistic Effects

The leakage and air flow rates for the building envelope (which includes exhaust equipment and fireplaces), the duct system, the HVAC equipment (and vents) and some household appliances (and vents) are affected by pressure differences. These pressure differences are caused by various "drivers" such as wind, indoor buoyancy forces caused by air temperature differences, buoyancy forces that affect vents and chimneys, exhaust equipment, the comfort system blower and duct leaks.

Figure A11-11 summarizes the effect that the various drivers have on the pressure in the conditioned space. Note that it would be unusual to find any one of these drivers acting alone. Normally, a set of drivers act in concert, so a wide range of space pressure conditions are possible, depending on which drivers are active. For example, if there is no wind, and no vent or exhaust equipment in operation, space pressure is close to zero if no heating or cooling is required (duct leakage and return path effects are negligible). On the other hand, space pressure could be decidedly negative (-10 to -20 Pascals) if one or more exhaust systems operate when blower

Pressure Drivers — Residential Structures

Subsystem	Pressure Produced (Pascals) (1)		Conditions
	Lower Level	Upper Level	
Envelope Crackage	+ 4 to - 4	+ 4 to - 4	Depends on the wind velocity and crack locations
Buoyancy Effect	0 to - 4	0 to + 4	Depends on height and indoor/outdoor temperatures
Toilet Exhaust	- 1 to - 6	- 1 to - 6	When fan is operating
Dryer Vents	- 1 to - 6	- 1 to - 6	When dryer is operating
Kitchen Exhaust	- 10 to - 30	- 10 to - 30	When fan is operating
Fossil Fuel Vents (2)	- 1 to - 5	- 1 to - 5	When burner is firing, combustion air from indoors
Fire Place Flue (2)	- 1 to - 5	- 1 to - 5	When in use, combustion air from indoors
Supply Duct Leaks (U) (3)	- 1 to - 8	- 1 to - 8	When blower is operating
Return Duct Leaks (U) (3)	+ 1 to + 6	+ 1 to + 6	When blower is operating
Supply Duct Leaks (C) (3)	+ 1 to + 8	+ 1 to + 8	When blower is operating
Return Duct Leaks (C) (3)	- 1 to - 10	- 1 to - 10	When blower is operating
No Return in Room (4)	+ 2 to + 10	+ 2 to + 10	Blower operating, interior door closed
Return in Core Area (4)	- 2 to - 15	- 2 to - 15	Blower operating, one or more interior doors closed

Note 1) The pressure ranges in this table are common, but higher or lower pressures are possible.
Note 2) This is the pressure in the room that contains the combustion component, not the draft pressure.
Note 3) U designates ducts that are in an unconditioned space. C designates ducts that are in a conditioned space.
Note 4) Inadequate return path — interior doors isolate perimeter rooms from a central return in the core area.

Figure A11-11

operation activates a supply leak driver or a return path driver. Or, space pressure could be positive if relevant duct leakage and return path drivers act when other drivers are dormant. Comments on how the space pressure affects the performance of the various subsystems are provided here:

Supply Cfm

Room supply Cfm is reduced when the room is pressurized and increased when the room is at negative pressure. Blower Cfm also is affected by room pressures. For example, when a dwelling has a single central return, the supply Cfm to rooms that are isolated from the return (by closing interior doors) decreases, the Cfm delivered to core rooms (that benefit from the central return) increases, and blower Cfm decreases. In this case, blower Cfm decreases because the interior doors act like balancing dampers, which effectively increase system resistance as they close. Figures A11-12 and A11-13 illustrate this behavior.

Figure A11-12 shows system flow rates when all interior doors are open. In this case the blower delivers 1,000 Cfm against 0.20 IWG of resistance and the flows to rooms A, B, C and D are 100 Cfm, 150 Cfm, 350 Cfm and 400 Cfm.

Figure A11-13 shows what happens when the interior doors for rooms A and B are closed. In this case, the flow to rooms A and B drops to 52 Cfm and 105 Cfm; the flow to rooms C and D increases to 370 Cfm and 420 Cfm; system resistance increases to 0.22 IWC; and the flow through the blower decreases to 947 Cfm.

- Closed doors throttle the flow to some rooms and cause a larger flow to the other rooms.
- Larger flows to some rooms cause more resistance in the ducts that serve these rooms.
- The resistance for every supply duct run increases, so blower Cfm is reduced.

Envelope Leakage Rate

Room infiltration Cfm is reduced when the room is pressurized and increased when the room is at negative pressure. But, Figure A11-11 shows that room pressure is affected by many factors. Therefore, space pressure and infiltration rate are determined by a complex interaction of wind velocity, envelope leakage, duct leakage, blower operation (on vs. off), the stack effect, exhaust fans, flues for combustion appliances, return air paths and the location of the filter.

Note that changes in the infiltration rate can reduce or increase the pollutants and the humidity in the conditioned space, depending on the situation. If pollutants or humidity are generated within the space, adverse effects are diluted by clean, dry infiltration. If pollutants or humidity are in the outdoor air, adverse effects are created by infiltration.

Also note that the performance of flues and vents for combustion equipment and fireplaces is very sensitive to the pressure in the room that has the combustion component. A negative pressure as small as -3 Pascals can cause vent back drafting. Since space pressure is constantly changing, the danger of back drafting depends on the set of drivers that are active at a given time. Figure A11-11 demonstrates that there are many scenarios that can cause a back drafting problem. A simple act like energizing an exhaust fan, operating a clothes dryer, closing an interior door or opening an interior door can cause an unsafe

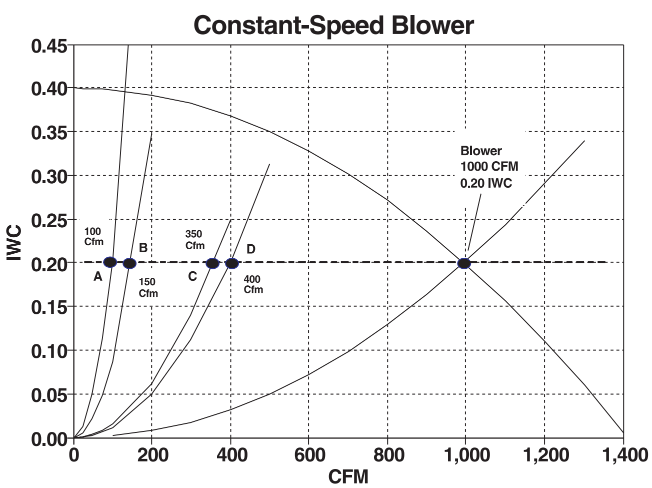


Figure A11-12

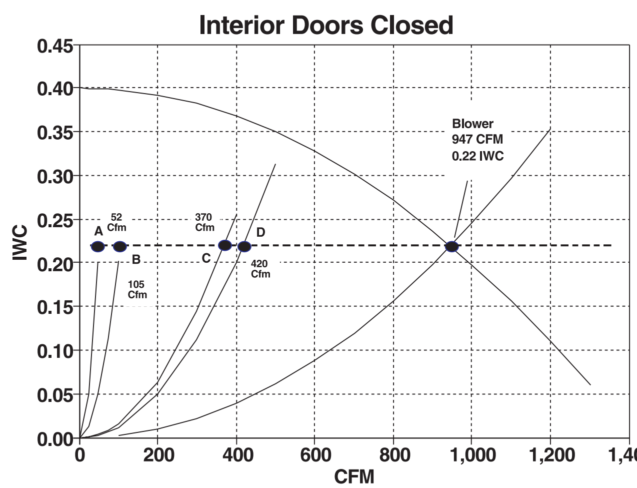


Figure A11-13

condition. In some cases, sealing duct leaks and/or structural leaks can produce a negative pressure condition that causes a venting problem.

Exhaust Equipment

The performance of exhaust equipment and vented household appliances is affected by the pressure in the room that has the equipment. Exhaust Cfm decreases as space pressure gets more negative and increases as space pressure gets more positive. (If a set of pressure drivers creates a large negative pressure in the room, the effectiveness of the exhaust equipment can be completely neutralized.)

Flow rates for various types of residential exhaust equipment are listed below (assuming the equipment has access to an unrestricted supply of air). Obviously, if two or more components operate simultaneously, they compete for available supply of air and the flow through each component is reduced. This situation is exacerbated if other drivers, such as duct leaks or return path problems contribute to the negative pressure situation.

- Kitchen = 100 to 200 Cfm.
- Bath = 50 Cfm.
- Clothes dryer = 100 to 150 Cfm.
- Central vacuum = 100 Cfm.
- Kitchen range = 250 to 500 Cfm.

Supply Duct Leaks (Unconditioned Space)

Leaks from a supply duct to an unconditioned space do not affect the performance (capacity and efficiency) of the mechanical equipment, but they do degrade the overall performance of the HVAC system because they increase the load on the equipment. Supply-side leaks also can create comfort and air quality problems.

- This type of leak depressurizes the conditioned space and increase infiltration to the conditioned space.
- Additional infiltration from the outdoors or from an ancillary space, affects operating cost, comfort and indoor air quality.
- This type of leak wastes conditioned air (heated, or cooled and dehumidified), which is replaced by unconditioned infiltration air. This exchange wastes energy and increases operating cost.
- Loss of conditioned supply air is more serious than loss of return air because the temperature difference and the moisture difference between the wasted air and the replacement air is larger. (During the cooling season, replacement air may be hotter than the outdoor air; air that infiltrates from an attic, for example.)
- This type of leak does not alter the temperature rise across the heating equipment, or the temperature-humidity drop across a cooling coil, because it does not alter the condition of the air entering the equipment. (The condition of the air leaving the equipment depends on the condition of air entering the equipment, the blower Cfm and the capacity of the equipment.)
- This type of leak does not affect the mechanical efficiency of the cooling equipment because they do not alter the condition of the air entering the indoor refrigerant coil.
- This type of leak produces an unnecessary load on the auxiliary heating coil for a heat pump. This raises the thermal balance point and reduces the seasonal efficiency of the heat pump system.

Supply Duct Leaks (Conditioned Space)

When air movement within a dwelling is unrestricted, supply leaks to the conditioned space do not substantially affect equipment performance (capacity and efficiency) and they do not increase equipment load. However, this type of leak affects the pressure in isolated rooms, and this can have an indirect effect on equipment load.

- This type of leak does not increase the infiltration load or the load on the equipment, providing that the air is free to move around the various rooms in the dwelling.
- This type of leak does not alter the temperature rise across heating equipment or the temperature-humidity drop across a cooling coil because it does not alter the condition of the air entering the equipment.
- This type of leak does not affect the equipment efficiency because it does not alter the condition of the air entering the equipment.
- This type of leak tends to increase the pressure in a room that does not have an adequate return air path.
- If this type of leak pressurizes a space, there is less infiltration to the space.
- If rooms isolated by closed interior door are pressurized, exfiltration from these rooms causes other rooms to be at negative pressure, so these rooms have more infiltration. This increases infiltration load, equipment load and operating cost.
- This type of leak pressurizes a space and reduces supply Cfm to the space.
- The air balance for all rooms is affected by room-to-room pressure differences.
- If this type of leak pressurizes a space, contaminants could be transferred to other rooms in the dwelling by local infiltration.

Return Duct Leaks (Unconditioned Space)

Leaks from an unconditioned space to a return duct increase the load on the mechanical equipment, affect the performance (capacity and efficiency) of the equipment and degrade the overall performance of the HVAC system. Return-side leaks also can create comfort and air quality problems.

- This type of leak pressurizes the conditioned space and decreases infiltration to the space.
- This type of leak draws air from the outdoors or from an ancillary space, so it increases operating cost.
- This type of leak draws air from the outdoors or from an ancillary space, so it can have an adverse affect on indoor humidity.
- This type of leak draws air from the outdoors or from an ancillary space, so it can affect air quality in the conditioned space. (Which may improve or degrade air quality, depending on the quality of the air that surrounds the return duct. If replacement air is of good quality, it dilutes concentrations of space contaminants. If the replacement air is of poor quality, it degrades indoor air quality.)
- This type of leak wastes neutral air (air that exfiltrates from the rooms) and replaces it with air that has not been conditioned at all. This exchange wastes energy and increases the operating cost.
- The loss of neutral air is not as serious as the loss of fully conditioned supply air because the temperature difference and the moisture difference between the wasted air and the replacement air is smaller. (During the cooling season, the replacement air may be hotter than the outdoor air - air that infiltrates from an attic, for example).
- This type of leak alters the temperature rise for heating equipment, the temperature-humidity drop for cooling equipment and the sensible heat ratio of the cooling coil. In some case, leaving air (supply air) may not be capable of neutralizing the space load, which may be a heating load, a sensible cooling load or a latent load. (Leaving air condition depends on the condition of the entering air, blower Cfm and equipment capacity.)
- This type of leak affects the mechanical efficiency of the cooling equipment because it alters the condition of the air entering the indoor refrigerant coil.
- This type of leak produces an unnecessary load on the auxiliary heating coil for a heat pump. This raises the thermal balance point and reduces the seasonal efficiency of the heat pump system.

Figure A11-14 shows how the return-side leaks in an attic duct affects the cooling equipment performance. For this example, the return duct has 20 percent leakage and the attic is hot and humid. (The attic condition is typical for the Southeast and Midwest on a hot summer day.)

Note that when compared to a no-leakage scenario, the sensible load increases by 13,200 Btuh and latent load increases by 4,900 Btuh. Also note that the sensible capacity increases by about 4,000 Btuh, the latent capacity decreases by about 1,800 Btuh and the leaving dry-bulb and wet-bulb temperatures increase dramatically. It is obvious that this system will not be able to maintain an acceptable level of comfort on a hot summer day.

Return Leakage Effect — Design Day Cooling

	No Leakage	20% Leakage
Coil Cfm	1,200	1,200
Leakage Cfm	0	240
Sensible load	28,000	41,200
Sensible capacity	28,320	32,250
Latent load	6,500	10,900
Latent capacity	7,080	5,250
Entering DB	75	85
Leaving DB	54	61
Entering WB	63	67
Leaving WB	52	57

1,200 Cfm cooling unit, leaky return duct in attic.
Outdoor air condition = 95°F Dry-bulb and 95 Gr / Lb.
Attic Dry-bulb = 125°F, Attic moisture = 95 Gr / Lb.
Equipment size based on “tight” ducts.
Cooling capacities extracted from manufacturers’ data.

Figure A11-14

Figure A11-15 (next page) shows how return-side leaks for an attic duct affect the heating performance of an air-source heat pump. For this example, the return duct has 20 percent leakage rate and attic temperature equals outdoor temperature.

Note that 240 Cfm of return-side leakage increases the design heating load by 18,480 Btuh. Also note that the balance point increases by about 7 degrees and the design load on the electric heating coil increases by 5.4 KW. These effects translate to a substantial increase in energy use and operating cost. This is demonstrated by a bin calculation that shows the heating energy requirement increases by 9,600 Kwh. At 0.10 cents per Kwh, wasted energy adds $960 to the annual heating bill.

Approximate Energy Wasted by Leakage				
Bin	**COP**	**Bin Hrs**	**Leakage Load**	**Btu x 10^6**
0-10	1.0	75	17,107	1.28
10-20	1.0	341	14,360	4.90
20-30	1.0	1,076	11,613	12.50
30-40	1.5	1,525	8,867	9.01
40-50	2.5	1,285	6,120	3.15
50-60	2.9	1,420	3,373	1.65
60-70	3.4	1,563	627	0.29
Akron, Ohio 240 Cfm			**Total Btu/Yr Kwh/Yr**	**32,780,000 9,604**

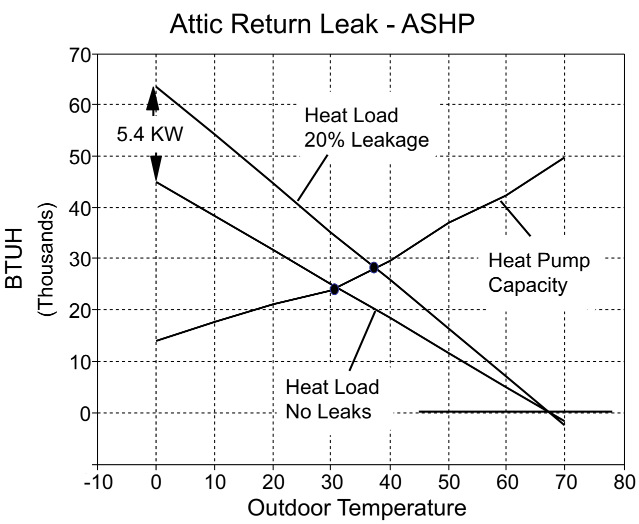


Figure A11-15

Return Duct Leaks (Conditioned Space)

When air movement within the dwelling is unrestricted, leaks from the conditioned space to a return duct do not substantially affect equipment performance (capacity and efficiency) and they do not increase equipment load. However, this type of leak can affect the pressure in isolated rooms and this could have an indirect effect on equipment load.

- This type of leak does not increase the infiltration load or the load on the equipment, providing that air is free to move around the various rooms in a dwelling.
- This type of leak does not alter the temperature rise for heating equipment, or the temperature-humidity drop across a cooling coil because it does not alter the condition of the air entering the equipment.
- This type of leak does not affect equipment efficiency because it does not alter the condition of the air entering the equipment.
- This type of leak tends to reduce the pressure in a room that is isolated from other rooms.
- If his type of leak depressurizes a room, it increases local infiltration.
- If isolated rooms are depressurized, increased infiltration to these rooms causes other rooms to be pressurized, so air exfiltrates from these rooms. This increases infiltration load, equipment load and operating cost.
- If this type of leak depressurizes a space, it increases supply Cfm to the space.
- The air balance for all rooms is affected by room-to-room pressure differences.

A11-4 Excessive Space Humidity at Part-Load

Figure A11-14 shows how latent load increases when leaky ducts are installed in a humid, unconditioned space. Note that this example is for the full sensible load at the ***Manual J*** design condition.

Duct leaks can create unmanageable problems at part-load, especially for locations that have a high average dew point temperature during the summer months (see Figure A10-4). For these locations the net affect of the latent duct leakage load eliminates any chance of controlling space humidity when the sensible load is less than fifty percent of the ***Manual J*** load.

A11-5 Building Damage

Since duct leakage can affect space pressure, it can affect infiltration or exfiltration rates. Consequently, it can affect indoor humidity, the direction of moisture migration and the rate of moisture migration. Therefore, building damage such as mold, mildew and rot may occur when duct leakage produces a condition that causes high indoor humidity or uncontrolled moisture migration.

- Damage can be caused by moisture that condenses on exposed surfaces (windows and window frames are usually the first surfaces to show condensation) or within the layers of a structural panel (an outside wall, for example).
- In the winter, condensation occurs when dew point of the indoor air or exfiltrating air is higher than the temperature of an exposed surface or a concealed structural surface.

- The potential for serious damage is increased if condensation freezes.
- In the summer, condensation can occur if the outdoor air is above 75°F, providing that the temperature of a structural surface is below 75°F. Although this is not a problem for most of the country, it can occur in locations that have very high humidity during the summer.

Appendix 12 (Informative; not Part of the Standard)

Air Quality Issues

Duct leakage and inadequate return air paths can have a significant effect on indoor air quality. In some cases the problems created by the air distribution system can be serious.

A12-1 Problems Caused by the Duct System

The air quality problems produced by a defective air distribution system fall in three categories, which are comfort, health and safety. One of these problems, or a combination of these problems, is likely to occur if the air distribution system has excessive leakage and/or return path restrictions.

Comfort

Comfort is compromised when infiltration creates drafts. Infiltration also can cause the indoor humidity to be too high or too low.

- For a given envelope leakage area, the indoor-outdoor pressure difference controls the infiltration rate.
- The indoor-outdoor pressure difference can be affected by duct leakage.

Comfort is degraded when supply air outlets and returns are not able to deliver and extract the required air flow. If room or space air flow is not correct, the heating or cooling capacity of the supply air will not be balanced with the heating or cooling load. If supply air Cfm is not compatible with the size of the supply air outlet, drafts or pockets of stagnant air can cause problems in the occupied zone.

- Supply air Cfm and return air Cfm are affected by the room-to-duct pressure differences.
- Room pressure is affected by return path resistance, envelope leakage and leakage to adjacent spaces.

Leakage on the return-side of the duct system can have a significant effect on equipment performance and its ability to control the temperature and humidity of discharge air. This, in turn, affects comfort.

- The condition of the air leaving the indoor refrigerant coil depends on the temperature and humidity of the air that enters the coil.
- The temperature of the air leaving the heating equipment depends on the temperature of the air entering the equipment.
- The condition of the entering air is affected by return-side duct leakage.

Health

Health can be adversely affected when duct leakage produces a condition (infiltration or negative pressure) that introduces pollutants (dust, dirt, spores, fumes, odors, vapors, sewer gas, soil gas, radon gas, etc.) to the occupied space. This may occur directly by envelope leakage or indirectly by return air duct leakage.

- Return duct leakage increases when the leaks are downstream from a dirty filter grille.
- Outdoor air may be of poor quality.
- Polluted air may come from an attic, garage, or crawlspace.

Duct leakage also can produce conditions that lead to mold and mildew, with potential health and odor problems.

- Drip pans and cooling coils are vulnerable to biological growth if dirt and dust are pulled in with return-side duct leaks.
- In humid climates, duct leakage can cause high indoor humidity during the cooling season, which can lead to biological growth in the conditioned space.
- Condensation produces an environment for biological growth when cold air leaks from a supply duct, impinges on a nearby surface and causes the temperature of the surface to fall below the of the ambient air.

Every dwelling needs a minimum amount of fresh air because health and comfort are adversely affected when the infiltration rate is too low. Therefore, it is possible for duct leakage to have a positive effect on health and comfort. If duct leakage increases the infiltration rate, a tight structure benefits from the leakage, provided that the infiltrating air is of good quality. In this case a duct sealing project might degrade indoor air quality (unless mechanical ventilation provides an adequate supply of fresh air).

Safety

Duct leakage can effect the pressure in rooms, spaces and buffer zones. In some cases, leakage causes negative pressures, which can adversely affect the operation of combustion appliances, fireplaces, flues and vents.

Dangerous and even deadly situations occur when duct leaks produce pressures that cause back drafting, spilling and flame rollout.

Duct leakage also can cause furnace heat exchanger damage. This occurs when the furnace and refrigerant coil are located in a small enclosed equipment room or closet that is not well ventilated. During summer, cool air can leak from the supply plenum and/or furnace cabinet. This cools the furnace heat exchanger. If the temperature of the heat exchanger metal falls below the temperature of the outdoor air, condensation occurs on the combustion side of the heat exchanger. This condensation can cause rust and leaks.

A12-2 Collective Effect of Pressure Drivers

As noted, duct leakage and return path restrictions can cause pressure conditions that adversely affect indoor air quality. These pressure drivers act in concert with other divers that are not related to the air distribution system. Therefore, air distribution system effects must be analyzed from a systems point of view. This approach focuses on cause-and-effect relationships as they apply to occupants, duct leakage, return air path geometry, envelope leakage, exhaust fans, equipment performance characteristics, the effectiveness of the flues and vents and the combustion air requirements of fuel burning appliances. This is discussed in Appendix 11.

A12-3 Duct Board and Duct Liner

According to the North American Insulation Manufacturers Association (NAIMA), there is no research that indicates that duct board or duct liner creates a hazard that will have a chronic effect on the health of the occupants or the installer-fabricator. However, airborne glass fibers can cause temporary skin and respiratory irritations. These irritations are a medical reaction to glass fibers that rub against or become embedded in tissue surfaces. But, if the duct system is properly designed, installed and "blown down," the concentration of airborne fibers in the conditioned space is too small to create a health or comfort problem for occupants.

- When properly installed, duct board and duct liner does not erode over time.
- As far as fabrication and testing is concerned, workers can shield themselves from loose fibers by wearing protective clothing and filter components, and by observing the appropriate work practices.
- In 2001, the International Agency for Research on Cancer (IARC), reclassified glass wool as a Group 3, not classifiable as a carcinogenic to humans.

A12-4 Duct Cleaning

If dust, dirt, mildew and mold accumulate in the air distribution system, they can cause allergic reactions and create health hazards. If an inspection uncovers an accumulation of foreign material or biological growth on flow path surfaces, the problem may be corrected by duct cleaning. However, there is a possibility that the remediation work can create a more serious problem, if procedures are not appropriate for the condition, or if cleaning work is not performed in accordance with industry standards. (In some cases encapsulation is the preferred remediation strategy.)

A12-5 Dirty Socks Syndrome

People that have studied the problem believe that the dirty socks syndrome (a foul odor produced by a heat pump) is caused by an accumulation of biological contaminants on the indoor heat pump coil. These researchers speculate that microorganisms commonly found in the soil, water and air, are deposited on the indoor coil during the summer, when the coil is cold and wet. These colonies thrive and grow during the heating season, when the indoor coil is dry and warm.

However, this coating, by itself, does not produce the odor problem, even if spores are entrained in the supply air. Evidently, the odor is produced when the airborne contaminants are burned. This condition occurs during the heat pump defrost cycle when the electric resistance heater is energized; but even then, it might not occur unless the indoor coil is wet. In other words, there may be preconditions for the odor problem:

- Biological contaminants must be deposited on the coil.
- The indoor coil might have to be wet.
- Electric resistance heat must be on.

Note that two of these conditions make the occurrence of the odor problem somewhat arbitrary.

- The coil may not be contaminated.
- Even if the coil is coated, indoor humidity (during the heating season) may be too low to wet the indoor coil during the defrost cycle.

In any case, if the first condition is eliminated (no contamination), the problem cannot occur. Therefore, duct leakage could be a contributing factor if the leakage introduces contaminants to return air or causes contaminated air to infiltrate to the conditioned space.

The dynamics of the dirty socks problem are still not fully understood. Contact the Air Conditioning and Refrigeration Institute (AHRI) for information that postdates the publication of this manual.

Appendix 13 (Informative; not Part of the Standard)

Noise

Noise is produced by the blower and can be generated by air-side components, dampers, fittings, supply outlets and return grilles. Noise is generated when air flow velocities are too high. Noise is produced by equipment vibrations.

Noise can propagate through the duct system, it can be transmitted through, or radiated from, duct walls and it can be transmitted through the structural components of a building.

Comfort system noise should not create an unacceptable condition in the occupied space. Noise is controlled by observing air velocity limits, by using aerodynamic fittings, by using duct liner or duct board, by avoiding line-of-sight connections between a noise source and a supply outlet or return opening, and by appropriate balancing damper location. Ducts should be fabricated with adequate reinforcement and breaks. Equipment and ducts should be mounted or supported with components that isolate or absorb vibration.

A13-1 Blower Noise

Blower noise propagates downstream through the supply duct and upstream, against the flow or the return duct. If not attenuated, blower noise enters a room at supply air outlets and return openings. Blower noise also can be transmitted to the surrounding space through the equipment cabinet and/or duct walls.

A13-2 Noise Generated by Duct Runs

Noise is generated by turbulence at some point in a duct run. This is typically caused by inefficient fittings (elbows, tees, transitions and takeoffs) and sternly throttled dampers. Turbulence can be created by accessories and heat transfer components installed in airways. Regardless of source, generated noise propagates downstream though the supply duct and upstream in a return duct. If not attenuated, noise enters a room at supply air outlets and return air openings. Generated noise also can be transmitted to the surrounding space through the duct walls.

A13-3 Noise Generated by Air Distribution components

Noise is generated when air flows through a grille, register or diffuser. Intensity depends on air velocity through the face or neck of the component. If either of these velocities is too high, or if the air distribution component has a damper that is sternly throttled, unacceptable noise will propagate to the room.

A13-4 Transmitted Noise

Equipment room noise (or any type of noise) can be transmitted through walls, partitions, ceilings and floors. The amount of noise that enters a room depends on the tightness of the structural assembly and the type of construction material. Tight, massive construction provides the most attenuation. Light construction provides a small amount of attenuation. Small cracks and openings completely destroy the ability of a structural component to attenuate noise.

A13-5 Crosstalk

Crosstalk refers to a situation where noise created in one room is transmitted to another room via duct runs or a transfer grille. Crosstalk is likely to create a problem if the duct system or return path produces a line-of-sight connection for two rooms.

A13-6 Vibration

Mechanical vibrations are produced by all types of rotating and reciprocating equipment. These vibrations can create noise problems and in some cases they can cause structural damage.

Aerodynamic vibrations are produced when an unstable flow pattern is created by a blower or duct component. Unstable flows can cause the air flow to pulsate and surge, and in some cases the ducts may vibrate. These types of vibrations are not normally a problem if the air-side of the system is designed properly.

A13-7 Attenuation

Attenuation refers to the reduction in the intensity of a noise that is propagating through a duct system. For example, the power level of the noise that is generated by a blower will continuously decrease as it moves through the various components of the duct system. This type of attenuation is always desirable because it dissipates the blower noise before it enters the room.

A13-8 Room Absorption Effect

The construction and size of the room has a large effect on the perceived intensity of the sound that enters the room.

Noise generated by the comfort system is louder in a small hard room than for a large soft room. (Room attenuation depends on construction material, furnishings, carpets and window treatments.)

A13-9 Noise Criteria

The ear is sensitive to air pressure waves and to the frequency of these waves. The frequency spectrum for ear sensitivity is divided into eight octave bands. The perception of noise in any one of these bands depends on the intensity of the pressure wave for the band. So, perceived noise is defined by eight different pressure levels, one for each octave band. Perceived noise also depends on the acoustical attributes of the room and the neighborhood.

This complexity is resolved by using a single value descriptor (NC value) to rate the perception of noise for various environments. This index is a performance attribute of supply air and return air hardware.

NC values are provided for grilles, registers and diffusers. When this information is available, it is provided by product manufacturer's performance data. This information is commonly available for commercial components, but may not be published for some residential components. The industry recommended NC values for private dwellings are provided here:

- Rural/suburban single family dwellings = 20–25.
- Urban single family dwellings = 25–30.
- Multi-family dwellings = 35 to 45.

A13-10 Designing for Noise Control

Noise control is a design requirement. Most generated noise problems are avoided by using ***Manual D*** procedures to size airways; and to select fittings, blowers and air-side components.

In some cases, the practitioner must provide some type of attenuation or isolation to avoid noise problems caused by system components. Relevant recommendations and procedures are discussed here.

Supply and Return Hardware

The performance of supply air outlets and return grilles is summarized in manufacturers' engineering data, but for residential equipment, this may not include noise criteria values. If NC values are not available, face velocity or neck velocity is used as a guideline.

A properly installed supply outlet will not generate objectionable noise if the face velocity or neck velocity is 700 Fpm or less. This means that supply outlets are selected for three criteria, which are: required supply Cfm, required throw, and face or neck velocity. As far as

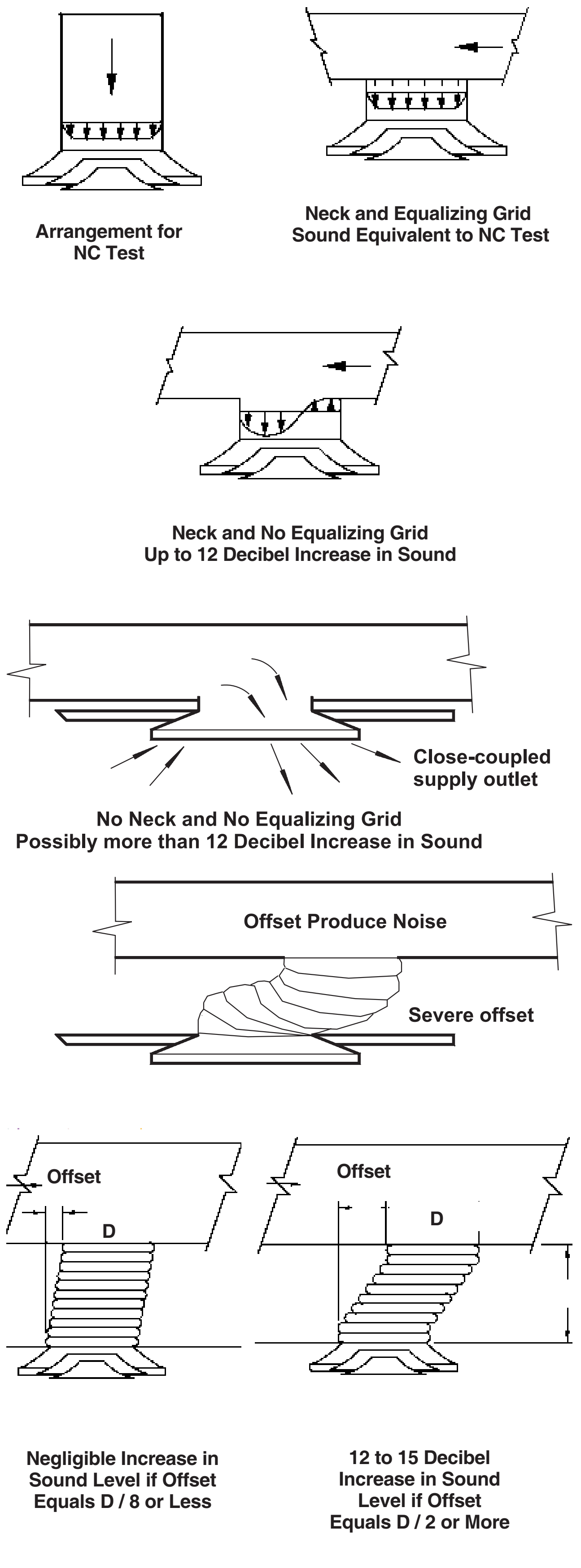


Figure A13-1

returns are concerned, objectionable noise should not be a problem if the face velocity is 500 Fpm or less.

Note that even if face or neck velocity is in an acceptable range, noise can be produced by improper installation practices. For example, Figure A13-1 (previous page) shows rigid duct installations that have non-uniform air flow approaching the blades or vanes of the supply outlet. This produces turbulence, generates noise, and distorts the supply air pattern. Figure A13-1 also shows that noise is produced by an abrupt flexible duct offset between the supply air outlet and the duct connecting point.

- Air should approach supply grilles, registers and diffusers as a uniform, perpendicular flow (no turbulence or eddies).
- Necks, collars, turning components and equalizing grids should be used to control the flow to the outlet.

Noise Generated by Flow Control Components

A register damper should not be used as a primary balancing damper because it will generate noise when adjusted to a hard-throttle position. (Registers are only suitable for making minor adjustments to the supply air flow rate.) When considerable flow reduction is required, the adjustment should be made by setting a hand damper that is installed as far from the supply air outlet as possible.

Velocity

Air velocity is very important, as far as generated noise is concerned. Excessive noise will not be produced by fittings and components if air velocities through duct runs are within the limits that are recommended by Table A1-1 of this manual (page125).

Blower Wheel Speed

Blower wheel speed reductions translate to slower flow rates and less noise, but a speed change may not be acceptable, as far as comfort is concerned. For example, a system that is providing satisfactory comfort (heating, cooling and dehumidification) before a speed change, may not perform satisfactorily after a speed change. It is important to remember that wheel speed is directly related to equipment capacity and air delivery rates.

Efficient Fittings

When velocity limits are observed, flow through aerodynamically efficient fittings will not be a source of noise. However, the potential for generated noise increases as the sophistication of fitting geometry decreases (see Figure A13-2). As far as noise and pressure drop are concerned, fittings that have radius turns, turning vanes and gradual transitions are preferable to square, unvaned fittings and abrupt transitions.

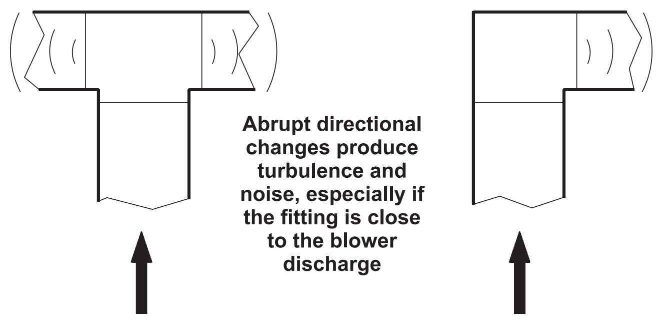


Figure A13-2

Duct Geometry

Strive to acoustically decouple the blower, or any other source of noise (a balancing damper, for example), from the occupied space. Design the duct system so that there are no short, straight connections between a blower (outlet and inlet) or air-side component and a room. Even long straight connections should be avoided.

When duct flow velocities are low, elbows, tees, take-off fittings and transitions attenuate noise. Ideally, the flow should turn at least twice as it moves from the source of the noise to the room. Figure A13-3 (next page) shows supply and return openings that are too close to the blower, as far as noise control is concerned, and alternative designs that reduce blower noise.

Upstream Turbulence

Fittings and dampers should only be installed where air flow has a smooth, well ordered flow pattern. Do not install any type of fitting or damper in the turbulent wake of an upstream disturbance. In addition, incorrect branch take-off design at the end of a supply trunk will cause local inefficiency and turbulence that may result in inadequate airflow through the branch and unacceptable noise (see Section 5-6).

Provide Attenuation

Propagating noise is attenuated by turns and sound absorbing materials. In this regard, lined elbows are desirable because they are effective attenuating components, especially if liner is placed directly in the elbow, a short distance upstream from the elbow and a short distance downstream from the elbow.

A properly designed, acoustically lined plenum attenuates noise. As far as straight runs are concerned, duct liner and duct board provide significant attenuation. For some designs, duct system components may not provide

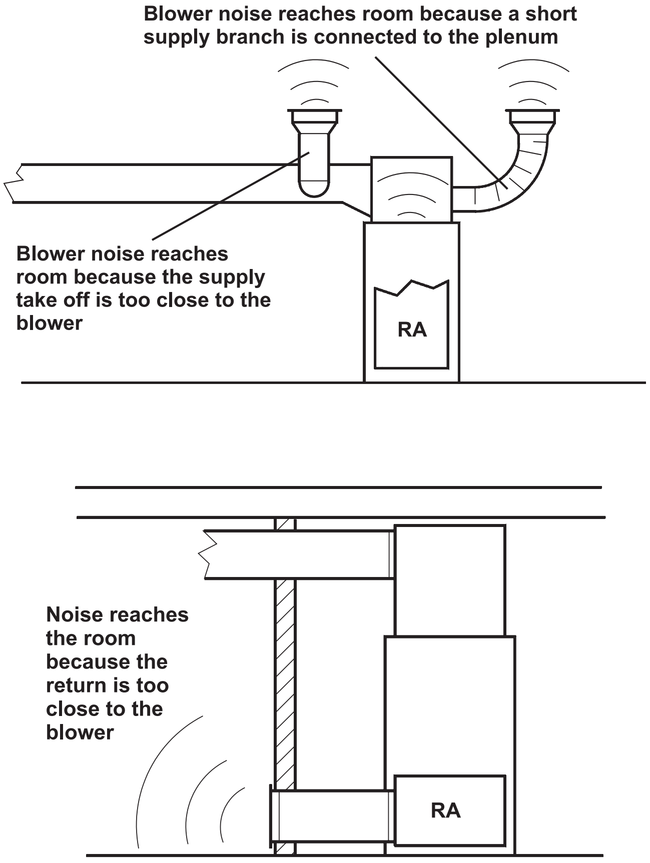


Figure A13-3

enough attenuation, even if they are lined. When this is the case, sound traps provide required attenuation.

- Line the first 5 feet of a supply trunk.
- Line the first 10 feet of a return run.
- Design the run so that it has two 90-degree turns.

Duct Shape

Turbulence may cause a duct wall to vibrate. Round shapes resonate less than rectangular shapes because curvature tends to stiffen duct walls. Crossbeams, beading and reinforcement reduce noise radiated from rectangular shapes.

Prevent Crosstalk

Crosstalk refers to occupant generated noise that is transmitted from room to room. Crosstalk can be reduced or prevented by using duct geometry that has no line-of-sight connection between any supply or return openings (see Figure A13-4). Also avoid using door and wall grilles to transfer return air from a room to a central return.

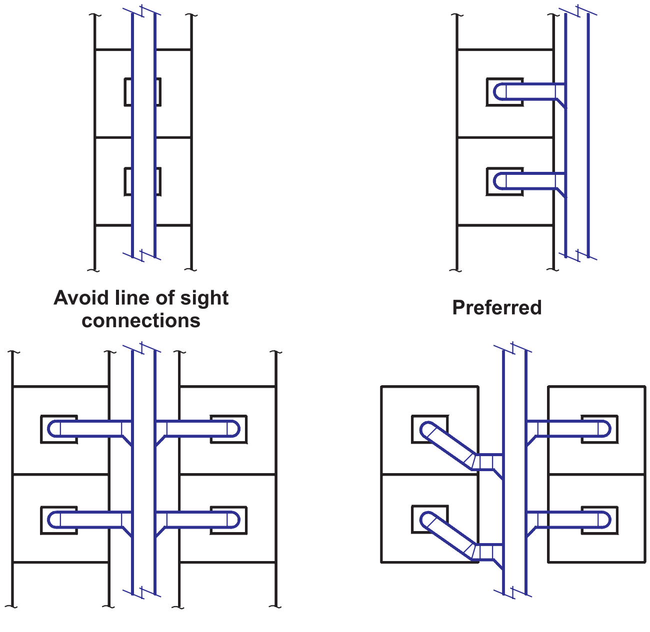


Figure A13-4

If return air transfer is required, install two grilles in the ceiling (on each side of the partition) and connect them with a lined duct or a fibrous-board duct that has two 90-degree elbows. (see Appendix 3, Group 14.)

Airway materials are important. A system that features duct liner, duct board and flex duct materials is superior to a sheet metal system, as far as attenuation is concerned.

Duct Leakage

Seal the cracks, seams and joints in duct runs and equipment panels located in the occupied space or if such surfaces are acoustically coupled to the occupied space. When airways are tightly sealed, noise propagated along duct runs (blower noise, for example) is isolated from the room because room air is not affected by the pressure pulses in the duct.

Low Frequency Noise

Duct board and duct liner provide some attenuation of high frequency sound, but most duct materials and duct liners are transparent to low-frequency sound. Rooms under lay-in ceilings are especially susceptible to low frequency noise. Structural mass provides the best defense against a breakout of low frequency noise. However, the cost advantages of using lightweight materials usually determine material choice.

Block Transmission

Sound can enter the occupied space by transmission through walls, ceilings and floors. Make sure that the structural panels between the equipment room and the occupied space are completely sealed (no cracks or penetrations). Sound absorbing material, or in extreme cases, additional wall mass, may be required if transmission is a problem after structural panels are sealed.

Provide Vibration Isolation

Blower wheels must be balanced. Blowers and blower cabinets must be installed in accordance with the manufacturer's recommendations. Vibration pads should be installed under equipment. Flexible connections should be installed between the blower cabinet and the duct work. Also use flexible connectors for rigid conduit and piping connections. Ducts should be supported and isolated in accordance with SMACNA standards. Whenever possible locate the HVAC equipment away from the rooms that should be quiet.

Appendix 14 (Informative; not Part of the Standard)

Testing and Balancing

For comfort and efficiency, air balance is as important as design calculations and installation protocol. This section provides an overview of air-side testing, balancing and inspection work.

A14-1 Scope of Work

A dwelling has many systems. Some have nothing to do with the comfort conditioning system, but others are part of the comfort system, or they interact with the comfort system. The list of systems that may have to be tested includes the structural envelope, the air distribution system, the ventilation system, refrigerant-based heat transfer systems, fossil fuel heating systems, venting systems, water-based heat transfer systems, electrical systems and control systems. As far as the air-side performance is concerned, there are tests for the structural envelope, air distribution system, ventilation system and venting systems.

A14-2 Blower Door Testing

The practitioner evaluates the performance of the structural envelope because it determines the load on the heating and cooling equipment. Envelope attributes also affect occupant comfort, indoor air quality and utility cost.

- If an existing structure is involved, a construction-features survey and a blower door test provide information for troubleshooting and retrofit work.
- For new construction, the performance promised by drawings, calculations and specifications is validated by inspections and tests.

As far as the air distribution system is concerned, a blower door test can be used to evaluate duct system leakage. This may be accomplished by using the subtraction method or the flow hood method.

Subtraction Method

The entire duct system is placed under a negative pressure (50 Pascals or less), depending on the leakage rate at the various leakage points, and on the air flow resistance for various duct runs. Two leakage measurements are required for the subtraction test, one with the supply outlets and return grilles open, and one with the supply outlets and return grilles sealed. Then, a duct leakage rate estimate is obtained by subtracting the second leakage Cfm value from the first leakage Cfm value. The deficiencies of this test are listed here:

- Only measures leakage for ducts located outside the conditioned space.
- Does not duplicate the pressure gradients, pressure differences and the flow conditions that exist when the air distribution system operates.
- Poor accuracy if the envelope is very leaky.
- Accuracy is sensitive to small errors in test data.
- Accuracy is affected by the wind conditions.
- Not accurate when the duct leakage rate is low.
- Overemphasizes leakage near grilles and registers.
- Des not measure leaks to the conditioned space.

Leakage to Outdoors Method

The flow hood test involves sealing all supply resisters and returns, except for one return grill. When the blower door depressurize the dwelling, duct leakage is evaluated by using a flow hood (or calibrated fan system) to measure air flow through the open return. This test produces an accurate leakage measurement because the measurement is a single test is applied directly to the duct system, and because flow hoods are fairly accurate instruments. The deficiencies of this test are listed here:

- Only measures leakage for ducts located outside the conditioned space.
- Does not duplicate the pressure gradients, pressure differences and the flow conditions that exist when the air distribution system operates.
- The negative pressure in the duct is approximately 25 Pascals near the open return, but pressures at other points in the duct system are less negative, depending on the distance from the open grille and the resistance of the low path.
- Duct pressure varies with the position of the leakage point.
- Overemphasizes leakage near grilles and registers.
- Does not measure leaks to the conditioned space.

Blower Door Test Vs. Infiltration Rate

When blower door equipment measures the envelope leakage rate, the conditioned space is depressurized to -25 Pascals. The resulting flow rate through the blower door is not equivalent to the infiltration Cfm, it is only an indication of building tightness. ***Manual J***, Eighth Edition Version 2.10 or later (unabridged) provides a procedure for converting blower door data to an infiltration Cfm value.

A14-3 Calibrated Airflow Matching System

A calibrated airflow matching system is similar to a small blower door. This component measures duct leakage directly. When this test is performed, all supply outlets and return grilles are sealed, then the duct system is pressurized (or depressurized) by the apparatus. This test produces a credible leakage Cfm estimate because the measurement is based on a single test that is applied directly to the duct system, and because the duct blaster is a fairly accurate instrument. The deficiencies of this test are listed here:

- Does not discriminate between leakage to a conditioned space and leakage to an unconditioned space.
- Does not measure leaks to unconditioned spaces.
- Must use a blower door test to measure leaks to outdoors.
- Does not duplicate the pressure gradients, pressure differences and the flow conditions that exist when the air distribution system operates. (Both sides of the duct system are pressurized during the test.)
- Duct pressures vary with position of the leakage point.
- Overemphasizes leakage near grilles and registers.

A14-4 Pressure Measurements

As explained in Appendix 11, the pressures in the various rooms and spaces fluctuate, depending on which pressure drivers are active. These pressures and pressure differences should be measured because the information can be used to evaluate duct system tightness, the continuity of return air paths, the effectiveness of the exhaust and venting systems, and envelope infiltration.

A14-5 Vents, Chimneys and Exhaust Systems

The capacity of vents, chimneys and exhaust systems are certified by measuring flow rates for the most adverse operating conditions. Relevant conditions are created when the applicable pressure drivers combine to produce a peak negative pressure in a room or space.

A14-6 Safety

It is important to understand that leakage tests and duct sealing efforts can create an unsafe condition. This can occur when leakage testing, or duct sealing, produces a negative pressure in a room or space that contains atmospheric combustion equipment.

- The relationship between duct leakage and room or space pressure is discussed in Appendix 11.
- Flue gas may back draft through combustion equipment (properly vented combustion equipment can back draft when the space pressure is as small as -1 to -5 Pascals).
- Depressurization may cause flame rollout at a furnace or water heater.
- The combustion efficiency of a burner may be affected by the pressure in the equipment room.

Because of the potential danger, leakage testing and duct sealing work must adhere to documented procedures. If there is any question regarding safety, a series of carbon monoxide (CO) tests should be made before and after a duct sealing project. Do not proceed with a testing and sealing project if a CO test indicates unacceptably high levels in the conditioned space, or the flue gas. Also issue warnings to the occupants and alert the proper authority, so that the problem is immediately corrected.

A14-7 Air-Side Balancing

The primary objective of balancing work is to ensure that each room receives the desired flow of supply air (design Cfm values are provided by the Duct Sizing Worksheet.) This involves taking measurements and balancing damper adjustments.

Flow measurements are made at supply air outlets and returns, but other types of flow, pressure and temperature measurements are used to fully evaluate duct system performance, blower performance, or to estimate duct leakage. A brief summary of air balancing tests is provided here:

Supply Cfm and Return Cfm

Flow through supply air outlets and returns can be measured by a flow hood, a velometer (with probe), a vane anemometer, or a thermal anemometer. A flow hood is the easiest to use (provided that the size and shape of the hood is compatible with the size and shape of the grille, register or diffuser) because it returns a Cfm value. Other methods are less convenient because multiple measurements may have to be averaged, proprietary factors (conditional A_k values published in manufacturer's data tables) may be required to convert instrument readout data to a Cfm value, or an area measurement may be used to convert velocity data to a Cfm value.

Duct Flow

Duct flow can be directly measured by using an anemometer, velometer, or a manometer and a pitot tube. All of these instruments use the duct traverse technique. This produces a series of data points. If a velometer is used, velocity data is averaged, and multiplied by cross-sectional area to obtain a Cfm value. If a pitot tube is used, velocity pressure readings are converted to velocity values, then the velocity values are averaged, then average velocity is multiplied by internal cross-sectional area to obtain a Cfm value.

Duct flow also can be measured indirectly, providing the flow passes through an air-side component (a fin-tube coil, for example). When this method is used, component pressure drop is measured with a pitot tube or static pressure gauge, then the pressure drop value is converted to a flow rate. This requires manufacturer's performance data (a graph or table that correlates Cfm with pressure drop).

Blower Cfm

The blower Cfm can be evaluated by measuring the flow through a trunk duct located immediately upstream or downstream from the blower. Or, if a blower table is available, blower Cfm is estimated by correlating blower pressure change and blower wheel speed with blower Cfm. (Do not assume that blower Cfm is represented by the sum of the supply outlet Cfm values, or the sum of the return air Cfm values. This assumption is invalidated by duct leakage.)

Duct Leakage

Duct system leakage can be estimated by comparing upstream flow with downstream flow. On the supply-side of the system, upstream flow equals the flow in the primary supply trunk (near the blower, before the first branch take-off), and the downstream flow equals the total flow through the supply outlets. On the return-side of the system, upstream flow equals the total flow through the return grilles, and downstream flow equals the flow in the return trunk (near the blower, downstream from the last return branch).

Appendix 15 (Informative; not Part of the Standard)

Air Velocity for Ducts and Grilles

Table A1-1 provides guidance pertaining to velocity through duct airways and face velocity at supply air grilles, registers and diffusers. This guidance lists recommended velocity and maximum velocity, but says nothing about minimum velocity.

A15-1 Continuity Equation

Continuity is the most basic of fluid flow principles. The concept is that steady flow into a pipe, duct or plenum, equals steady flow out of a pipe, duct or plenum (i.e., matter is neither created or destroyed). This equation applies the principle to a duct run:

$$ECfm_1 + ECfm_2 + \ldots ECfm_n = LCfm_1 + LCfm_2 + \ldots LCfm_n$$

- $ECfm_{(1 \rightarrow n)}$ is the flow entering the duct from various engineered openings and the Cfm leaking into the duct at joints, seams, penetrations and cracks.
- $LCfm_{(1 \rightarrow n)}$ is the flow leaving the duct from various engineered openings and the Cfm leaking from the duct at joints, seams, penetrations and cracks.

Things get remarkably clear when there is one engineered opening, one engineered exit and no leaks.

$$Cfm_{in} = Cfm_{out}$$

We gain more insight by adding detail to this equation. If area is in Square Feet (SqFt), and velocity is in Feet per Minute (Fpm) we get:

$$Area_{in} \times Velocity_{in} = Area_{out} \times Velocity_{out}$$

Suppose we want to route 1,000 Cfm through a rigid trunk duct. Per Table A1-1, the recommended velocity is 700 Fpm and the maximum velocity is 900 Fpm. So…

Minimum area (SqFt) = 1,000 / 900 = 1.11
Recommended area (SqFt) = 1,000 / 700 = 1.43

Suppose duct area exceeds 1.43 SqFt, lets say its 3.0 SqFt or 6.0 SqFt. Then…

Velocity (Fpm) = 1,000 / 3.0 = 333
Velocity (FPM) = 1,000 / 6.0 = 167

If we summarize (see Figure A15-1), we see that velocity has no affect on the delivery of air routed through a section of duct.

Cfm	Velocity Fpm	Area SqFt	Diameter Inches	Friction Rate
1,000	900	1.11	14.3	0.086
1,000	700	1.43	16.2	0.046
1,000	333	3.0	23.5	< 0.01
1,000	167	6.0	33.2	<< 0.01

1) The friction rate is for metal duct.
2) < = less than << = much less than.

Figure A15-1

A15-2 Why Worry About Air Velocity

The primary problem caused by air velocity is turbulence that produces noise. So as far as Table A1-1 is concerned, objectionable noise is an issue.

Another problem caused by low velocity is that sectional duct area increases as velocity decreases. This translates to a need for more installation space, more duct material, and if airways size gets very large, heaver duct-wall material and bracing. So for practical economic and competitive reasons, duct airway sizes tend to be as small as possible, and corresponding velocities tend to be as large as permitted (or larger when practitioners ignore codes and standards).

Benefits of Low Velocity

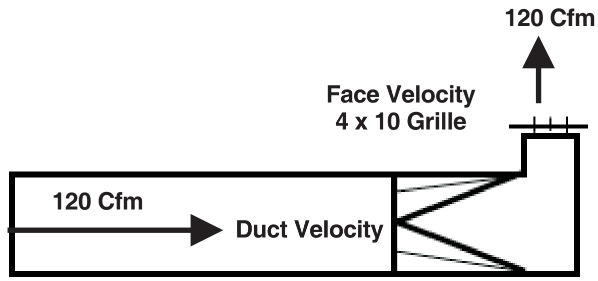
For a given Cfm, air flow resistance decreases as airway size increases and air velocity decreases, as demonstrated by Figure A15-1. This means that less blower power is required to move air through the duct.

Velocity Affects Takeoff Fitting Performance

If a supply trunk has a constant airway size, Cfm and air velocity decrease after each branch takeoff. This reduces the downstream friction rate, but increases the total equivalent length of the upstream fittings.

- See Appendix 3, Fitting Group 2 (pages 151-153).
- Branch takeoff equivalent length significantly increases with the number of downstream branches.
- The number of downstream branches are counted to the end of the trunk.

- If there is a trunk reducer, downstream branches are counted to the reducer, then a second count starts after the reducer (1, 2, 3, etc.).
- If a reducer is installed where trunk air velocity drops to the 400 to 500 Fpm range, there are less downstream branches, fitting resistance is reduced, supply run resistance is reduced, and less blower power is required.
- Velocity affects junction box performance.
- Equivalent length of a flexible duct junction box increases with the velocity of the entering air and the velocity of the leaving air.
- See Appendix 3, Fitting Group 11.
- Junction box equivalent length significantly increases with each 100 Fpm of velocity increase.
- Low velocity translates to less fitting resistance, so the resistance of the various duct routes is reduced, and less blower power is required.
- Plenums are just large junction boxes, so these comments also apply to plenums.



Duct Diameter Inches	Duct Area SqFt	Duct Velocity Fpm	Free Area SqFt	Face Velocity Fpm
6	0.20	611	0.17	700
8	0.35	344	0.17	700
10	0.55	220	0.17	700
12	0.79	153	0.17	700

Figure A15-2

Velocity Affects Transition Fittings

See Appendix 3, Fitting Group 12 (pages 170 - 172). Transition fittings increase or decrease air velocity with no change in Cfm. If there is a large change in velocity, say from a very low velocity to the maximum allowable velocity, there is a large change in airway size. So if the ratio of two sectional areas is large, the equivalent length of the fitting is significantly increased.

Myth

Some practitioners say… "We need high velocity in the duct so the supply air will mix with room air."The continuity equation shows that this is an absurd interpretation of reality, as demonstrated by Figure A15-2.

$$Cfm_{in} = Cfm_{out}$$
$$Area_{in} \times Velocity_{in} = Area_{out} \times Velocity_{out}$$

Face area (SqFt) = 4 x 10 = 40 SqIn = 0.35
Face velocity (Fpm) = 120 / 0.17 = 700
Duct diameter (SqFt) = 6 Inches; Duct area = 0.20
Duct velocity = (Fpm)120 / 0.20 = 611
Duct diameter = etc.
Duct velocity = etc.

In case someone is wondering, other common myths are listed here:

1) 500 SqFt per Ton
2) 400 Cfm per Ton
3) Use 0.10 IWC/100 Ft to size duct airways.
4) Use high returns for cooling and low returns for heating, or vice versa, depending on who your are talking to.

Velocity Affects Boot Performance

Boot fittings are transitions that produce working relationships between runout ducts and supply outlets. Refer to Figure A15-2.

- Air enters the boot at runout duct velocity.
- The boot changes the direction of the air.
- The boot changes the velocity of the air.
- Air approaches the grille going the desired direction with approximately the desired velocity.
- Air leaves the grille in desired direction at increased velocity (face velocity).
- Direction changes and velocity changes come at a cost. The price is the pressure drop produced by the boot.

For ***Manual D*** calculations, boot pressure drops are expressed as equivalent lengths. And, fitting equivalent length values are based on a set of assumptions (see the first page of Appendix 3 and see the Group 4 fittings).

- Equivalent length depends on entering and leaving geometry and airway velocities.
- ***Manual D*** equivalent lengths are for 900 Fpm velocity, which is a worst-case assumption (assuming Table A1-1 velocity limits are observed).
- Lower velocities generally translate to smaller, or slightly smaller, equivalent lengths.
- Marginally overestimating equivalent length provides a factor of safety for airway sizes.

- It is possible to connect a radically over-sized runout to a normal size supply grille, but the boot fitting will be a handmade work of art.
- An appropriately sized runout duct is a matter of practicality and economics.

A15-3 Balancing Dampers

Low air velocity implies over-sized airways, which leads to questions about deficient supply air Cfm and system air balance. Figure A15-2 shows that large airways do not affect supply Cfm. As explained here, and there is no reason to worry about losing system air balance, because it never existed in the first place.

- ***Manual D*** airway sizes are based on a "worst case" friction rate (i.e., for the circulation path that has the longest effective length). This means that airway sizes for the longest runs are correct and that the sizes of the other runs are larger than necessary. Therefore, the shorter runs will deliver excessive air flow.
- ***Manual D*** sizes duct runs for the worst-case condition, which may be heating or cooling. In either case, airway sizes are compatible with maximum air flow requirement, and larger than necessary for the lesser requirement.
- If a larger airway size is dictated by a velocity limit, the duct run produces less air flow resistance than the smaller size that satisfies the friction rate procedure. This reduction in resistance increases air flow.
- Flow rate is affected when a duct slide rule size is rounded to a standard size. Air flow increases when the size is rounded to the next larger standard size.

To correct this situation, install balancing dampers in the branch runout ducts. When the balancing dampers are adjusted, the total effective length of all circulation paths are approximately equal, and each path will flow the desired Cfm.

A15-4 Air Mixing and Room Air Motion

Supply outlets are selected and sized to perform specific tasks, as listed below. The first two tasks require an exchange of momentum (Mass x Velocity), the third item limits available momentum.

- Mix supply air with room air.
- Produce comfortable air motion in the occupied zone.
- Quiet operation.

When the blower is switched on, room air is just sitting there; it has relatively large mass, no velocity and no momentum. The momentum of the supply air depends on the flow rate (Cfm) and the face velocity of the air leaving the supply outlet (which has nothing to do with the velocity through any duct anywhere in the system). Then momentum is exchanged:

- A jet of supply air is projected into the room, but not into the occupied zone (because that would cause a draft).
- Because of viscosity, the jet of supply air drags some room air with it.
- Momentum is exchanged, the leading edge of the jet begins to slow down and room air begins to move and circulate.
- Air is continuously discharged from the supply outlet, so momentum is continuously imparted to room air.
- Supply air is mixed with room air as room air rolls and tumbles.

Self Balancing Design

A duct system could be self-balancing, if duct sections are sized to ensure that the pressure drop for each possible circulation path is exactly equal to the available static pressure. In this case the following rules apply:

- Trunk ducts common to multiple circulation paths, must be sized for the path that has the longest effective length.
- Since trunk sizes will be too large for shorter circulation paths, runout ducts must be sized to compensate for trunk sections that do not provide enough air flow resistance.
- Room supply air Cfm is typically different for heating and cooling, so room Cfm design values are the average of the heating Cfm and cooling Cfm.

However, a self-balancing design is not practical and the system could be noisy. These comments apply:

- The calculations become more complex and time consuming.
- Non-standard airway sizes are required to obtain the desired pressure drops.
- Air velocity at points in the shorter circulation paths could be too high.
- Room supply air Cfm is typically different for heating and cooling, so seasonal performance is compromised by a self-balancing design.

The problems produced by a self-balancing design are difficult to reconcile. The preferred solution is to install a balancing damper in each runout duct.

- Friction at ceilings, walls, windows and floors limits room air velocity (dissipates room air momentum, creating heat).
- After a few minutes, things settle down and air velocities at various points in the room stabilize (steady-state operation).
- If the outlet is the correct type and size for the application, there will be desirable air motion in the occupied zone, no drafts, no objectionable stagnate areas and good mixing.
- If the outlet is not the correct type and/or size for the application, there may be a draft or stagnate air in the occupied zone, and/or inadequate mixing.

Supply outlet performance is significantly affected by approach. Air must flow directly at the inlet of the grille, diffuser or register in a uniform manner.

- The flow vector should be perpendicular to the plane of the opening, the velocity profile should be uniform, there should be no turbulent eddies.
- Use geometrically compatible, aerodynamically efficient boots, fittings and necks to route air to supply outlets.
- A balancing damper behind a supply outlet may disrupt approaching flow, depending on the design.

A15-5 Conclusion

There are scores of things to worry about when designing and installing a comfort system. Low velocity through a duct airway is not one of them.

Appendix 16 (Informative; not Part of the Standard)

Excess Length and Sag in Flexible Duct

Duct run pressure drops may be significantly increased (and airflow rates reduced) if flexible duct is improperly installed (see Section 4-3). This Appendix provides tools for evaluating the performance of non-compliant installations. This information will make people aware of the ramifications and penalties of poor installation practices.

- Practitioners can reverse engineer existing duct systems, identify problem areas, and quantify the benefits provided by corrective adjustments.
- Conscientious designers will be more vocal and diligent in how their flex duct designs are installed in the field.

A16-1 Natural Length of Flexible Wire Helix Duct

When removed from its packaging, the natural or free length of a straight piece of flexible wire helix duct can be shorter than its fully stretched length, especially if effort is not taken to fully extend the duct before cutting it to the desired span length. This issue is not relevant to ***Manual D*** calculations (see A16-3), but it is relevant to published friction charts and duct slide rules.

ADC FD72-R1 is the test code for producing flexible duct friction charts. This code requires that the duct be stretched to its fullest length by pulling with 25 pound force, holding for one minute, and then allowing the duct to retract to its normal length.

A16-2 Default Pressure Drop for Flexible Wire Helix Duct

Appendix 2, Friction Chart 7, and the flexible duct scale on the ACCA duct sizing slide rule, are for installations that have little or no compression (0% to 4%) and negligible sag (½ inch per foot maximum). Table A16-1 (next page) and Table A16-2 (next page) provide equivalent length adjustment factors for installations that are improperly installed.

- Use Table A16-1 (next page) for duct runs where compression does not exceed 4% (1 foot per 25 feet).
- Use Table A16-2 (next page) for duct runs that have more than 4% compression.

A16-3 Default for Excess Length

Excess length equals the difference between the fully stretched cut length and the measured, straight-line, entrance-to-exit span length. Excess length always increases duct run pressure drop.

- A straight, fully stretched cut length has no compression (0% compression) and minimum pressure drop (performance similar to metal duct that has the same diameter).
- Pressure drop increases (about twice as much as a metal duct), if the duct is not fully stretched (close to 4% compression).
  - a) Friction charts that conform to the ADC FD72-R1 test code model performance for ducts that have 4% or less compression.
  - b) Chart 7, in Appendix 2 models ducts that have 4% or less compression.
- Relatively straight duct runs with not more than 4% compression, is the default scenario for ***Manual D*** design procedures.

A16-4 Affect of Excess Length

Excess flexible duct length effects total effective length (TEL) calculations. Excess length has no effect on any other ***Manual D*** procedure, but a larger TEL value produces a smaller design friction rate value.

$$FR = \frac{ASP \times 100}{TEL} \quad (IWC\,/\,100\,Ft)$$

A16-5 Excess Length Geometry

The centerline geometry for excess length may be a simple straight line, a gradual curve, a series of curves, a bend, or a series of bends. ***Manual D*** treats gradual curves as additional length with negligible compression, and classifies a bend as an elbow fitting.

- Excess length causes compression of the inner core if the centerline of the duct is relatively straight.
- Excess length may create a curve or series of linked curves that have little or no compression.
- If excess length produces an uncompressed curve or bend, the radius of curvature determines if the curve or bend is just extra equivalent length, or an elbow.
- If excess length acts like an elbow, the bend has a pressure loss coefficient that depends on the ratio of the centerline radius and the duct diameter (just like the R/D ratio for a manufactured fitting).

<table>
<tr><th colspan="7">Equivalent Length Multipliers[1] for Wire Helix Duct that has Excess Length and Negligible Compression</th></tr>
<tr><th rowspan="2">Duct Diameter (Inches)</th><th rowspan="2">ADC Standard 2.5" sag in 5 Ft</th><th colspan="2">Short Arc Sag</th><th colspan="2">Long Arc Sag</th></tr>
<tr><th>5" sag in 5 Ft</th><th>10" sag in 5 Ft</th><th>5" sag in 10 Ft</th><th>10" sag in 10 Ft</th></tr>
<tr><td>6</td><td rowspan="10">1.0</td><td rowspan="6">1.05</td><td rowspan="5">1.33</td><td rowspan="10">1.07</td><td rowspan="10">1.32</td></tr>
<tr><td>8</td></tr>
<tr><td>10</td></tr>
<tr><td>12</td></tr>
<tr><td>14</td></tr>
<tr><td>16</td><td>1.40</td></tr>
<tr><td>18</td><td rowspan="3">1.20</td><td rowspan="2">1.60</td></tr>
<tr><td>20</td></tr>
<tr><td>22</td><td rowspan="2">1.80</td></tr>
<tr><td>24</td><td>1.40</td></tr>
<tr><td>Bagged</td><td colspan="5">Excess length should not be stored in a bag.</td></tr>
<tr><td>Coiled</td><td colspan="5">Excess length should not be coiled.</td></tr>
</table>

1) The recommended (ADC) standard of care is 4%, or less excess length and negligible sag (2.5 inches sag per 5 feet of span).
2) These multipliers apply to airway sizing tools (friction chart or slide rule) that model the performance of duct that has less than 4% excess length and negligible sag (test stand condition).
3) The measured span length of flexible duct is the straight line length from entrance to exit (no Group 11 turns), from an entreamce to a Group 11 turn, the distance between Group 11 turns, or from a Group 11 turn to the exit.
4) The equivalent length of uncompressed, sagging duct equals the product of the measured span length and a sag multiplier.
6) These equivalent length values do not apply to any other type of duct material.
7) Short arc sag: One to two inches sag per foot of span. Long arc sag: One-half to one inches sag per foot of span.
8) Airway sizes for wire helix duct are read from the ***Manual D*** wire helix friction chart, or equivalent, or use the wire helix scale on the ACCA Duct Sizing Slide Rule, or equivalent.
9) Duct friction charts depend on construction details). The friction chart, or duct slide rule, provided by the manufacturer of a particular flexible duct product, supercedes the ***Manual D*** friction chart and the ACCA slide rule.

Table A16-1

- A sag in the vertical plane is the same as a curve or bend in the horizontal plane (sag implies curve or bend).
- The radius of curvature for excess length is related to inches of sag per foot of span.
- The maximum sag allowed by the Air Diffusion Council's (ADC) installation standard (***Flexible Duct and Installation Standards, Fifth Edition***) is ½ inch per foot of length (2-1/2 inches for a 5 foot span).

<table>
<tr><th colspan="4">Equivalent Length Multipliers[1] for Wire Helix Duct with Compression and Sag</th></tr>
<tr><th rowspan="2">Compres- sion</th><th colspan="3">Superimposed Sag</th></tr>
<tr><th>Negligible</th><th>1-Inch / Ft</th><th>2-Inch / Ft</th></tr>
<tr><td>0% to 4%</td><td>1.0</td><td colspan="2">1.1</td></tr>
<tr><td>15%</td><td>2.0</td><td colspan="2">2.2</td></tr>
<tr><td>30%</td><td>3.4</td><td colspan="2">3.7</td></tr>
<tr><td>45%</td><td>5.2</td><td colspan="2">5.7</td></tr>
<tr><td>Bagged</td><td colspan="3">Excess length should not be stored in a bag.</td></tr>
<tr><td>Coiled</td><td colspan="3">Excess length should not be coiled.</td></tr>
</table>

1) The recommended standard of care is 4%, or less coil compression and negligible sag (2.5 inches sag per 5 feet of span).
2) These multipliers apply to airway sizing tools (friction chart or slide rule) that model the performance of duct that has less than 4% excess length and negligible sag (test stand condition).
3) Compression occurs when excess length is squeezed into a shorter straight line span (see Table A16-3, next page). It is possible to have excess length with negligible compression.
4) The measured span length of flexible duct is the straight line length from entrance to exit (no Group 11 turns), from an entrance to a Group 11 turn, the distance between Group 11 turns, or from a Group 11 turn to the exit.
5) The equivalent length of compressed duct equals the product of the measured span length and a compression-sag multiplier.
6) These equivalent length values do not apply any other type of duct material.
7) Airway sizes for wire helix duct are read from the ***Manual D*** wire helix friction chart, or equivalent, or use the wire helix scale on the ACCA Duct Sizing Slide Rule, or equivalent.
8) Duct friction charts depend on construction details). The friction chart, or duct slide rule, provided by the manufacturer of a particular flexible duct product, supercedes the ***Manual D*** friction chart and the ACCA slide rule.

Table A16-2

- If sag exceeds ½ inches per foot, the installation does not comply with the ADC standard.
- ***Manual D*** defines non-compliant sag as any configuration that exceeds the maximum ADC allowance of 2½ inches sag per 5 foot duct span.
- For estimating the affect of non-compliant sag, ***Manual D*** used models for short arc sag, long arc sag, and excessive sag.
  a) Short arc sag is defined as 1.0 to 2.0 inches sag per foot of span (length adjustment factors are provided for a 5 foot span).
  b) Long arc sag is defined as 0.5 to 1.0 inches sag per foot of span (length adjustment factors are provided for a 10 foot span).
  c) Excessive sag is defined as sag that exceeds two inches per foot (more than 10 inches in a 5 foot span or 20 inches in a 10 foot span).

A16-6 Equivalent Length for Cut Lengths that have Negligible Compression

If a duct run has 4% or less compression, and if sag does not exceed two inches per foot (short arc sag or long arc sag), use Table A16-1 (previous page) to produce an equivalent length value for the run. Perform separate equivalent length calculations, if the excess length run has engineered bends that simulate elbows.

- Multiply the straight span length by the equivalent length multiplier for sag (Table A16-1), and enter the equivalent length value on the Effective Length Worksheet (as a trunk length or runout length).
- If there are engineered bends (elbows) in the entrance-exit path, use a Group 11 equivalent length value for each bend in path. Sum the equivalent length values, and enter the sum on the Effective Length Worksheet (as a Group 11 length).

Excessive sag (two or more inches sag per foot of span) is evaluated as a fitting loss. Excessive sag may be a continuous series of linked bends, or the bends may be separated by sections of relatively straight duct.

- Use a Group 11 equivalent length value for each bend (180 degrees or less) in the path that transverses the span. Sum the equivalent length values and enter the sum on the Effective Length Worksheet (as a Group 11 length).
- If the run has one or more relatively straight sections (less than 2 inches sag per foot), use Table A16-1 to produce an equivalent length value for these sections, and enter this equivalent length on the Effective Length Worksheet (as a trunk length or runout length).

A16-7 Flexible Wire Helix Duct Compression

For a duct that has a straight centerline, the amount of compression (C) equals the difference between its fully stretched cut length (SCL), and the measured,

Flexible Wire Helix Duct Compression																				
Span Feet (SL)	Fully Stretched, Straight Line Cut Length Feet (SCL)																			
	5	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100
5	0%																			
10		0%																		
15			0%	33%																
20				0%	25%															
25					0%	20%	40%													
30						0%	17%	33%												
35							0%	14%	29%	43%										
40								0%	13%	25%	38%									
45									0%	11%	22%	33%	44%							
50										0%	10%	20%	30%	40%						
55											0%	9%	18%	27%	36%					
60												0%	8%	17%	25%	33%	42%			
65													0%	8%	15%	23%	31%	38%		
70														0%	7%	14%	21%	29%	36%	43%
75															0%	7%	13%	20%	27%	33%
80																0%	6%	13%	19%	25%
85																	0%	6%	12%	18%
90																		0%	6%	11%
95																			0%	5%

Table A16-3

straight-line, entrance to exit span length (SL), divided by the span length, expressed as a percentage.

The compression equation is provided here, and Table A16-3 (previous page) summarizes output from this equation:

$$C\,(\%) = \frac{(SCL - SL)}{SL} \times 100$$

A16-8 Equivalent Length for Compression

Compressed duct, with or without sag, significantly increases air flow resistance. This is demonstrated by Table A16-2, which shows that the equivalent length multiplier for compressed duct ranges from 1.0 to 5.2 with no sag, and from 1.1 to 5.7 with sag. Therefore, standard wire helix duct friction charts (see Section A16-2), or duct slide rule equivalents (4% compression or less with negligible sag), do not apply to duct that has more than 4% compression. However, equivalent length adjustment factors allow the use of standard friction charts, or duct slide rules for determining the effect on airway sizing and duct performance.

- Multiply the straight span length by the equivalent length multiplier for compression and sag, and enter the value on the Effective Length Worksheet (as a trunk length or runout length).
- If there are engineered bends in the entrance-exit path, use a Group 11 equivalent length value for each bend in path. Sum the equivalent length values and enter the sum on the Effective Length Worksheet (as a Group 11 length).

A16-9 Comparative Examples

Section 8 has two flexible duct examples that show total effective length calculations for flexible duct runs that do not have excess length. Comparative examples for duct runs that have excessive length are provided here:

Example 1

At the beginning of Section 8, Figure 8-1 provides a sketch of a duct system that has a duct board plenum and flexible runouts that are cut to length. Figure 8-4 shows the effective length calculations for this duct system. For this example, runouts with short arc sag (5 inches of sag per 5 feet of span and negligible compression) are compared to cut to length runouts.

- Equivalent length multipliers for short arc sag are provided by Table A16-1.
- Figure A16-1 shows the consequences of non-complaint workmanship, as it pertains to total effective length.

Example 2

In the middle of Section 8, Figure 8-8 provides a sketch of a complete flexible duct system that has all duct runs cut to length, and Figure 8-11 shows the effective length calculations for this duct system. For this example, runs that have excess length as short arc sag (10 inches per 5 foot span), and runs that have excess length with 15% compression with sag are compared to cut to length runs.

- Equivalent length multipliers for short arc sag are provided by Table A16-1.
- Equivalent length multipliers for compression with sag are provided by Table A16-2.

Effective Length Worksheet (Flex Runout Example)

Element	Supply Run: S7			Element	Return Run: R3			
	Cut to Fit	Arc Sag			Cut to fit	Arc Sag		Notes
Trunk Length	43	43		**Trunk Length**	54	54		
Runout Length	22	22 x 1.05 = 23		**Runout Length**	8	8 x 1.05 = 8		Short arc sag
Group 1 (L)	40	40		**Group 5** (I)	30	30		(H/W = 2)
Group 2 (B)	40	40		**Group 6** (H)	15	15		
Group 4 (J)	30	30		**Group 8** (D)	65	65		(ez bend)
Group 12 (O)	5	5		**Other** **(6A br)**	40	10		
Other				**Other** (6A m)	25	25		
Other				**Other** (8E)	10	10		
Total Length	180	181		**Total Length**	247	247		

Figure A16-1

Effective Length Worksheet (Wire Helix Example)							
Element	**Supply Run:** S7 or S8			**Element**	**Return Run:** R3		
	Cut to Fit	Arc Sag	Compression		Cut to Fit	Arc Sag	Compression
Trunk Length	30	1.40 x 30 = 42	2.2 x 30 = 66	**Trunk Length**	14	14 x 1.40 = 20	14 x 2.2 = 31
Trunk Length	20	1.33 x 20 = 27	2.2 x 20 = 44	**Trunk Length**			
Runout Length	14	1.33 x 14 = 19	2.2 x 14 = 31	**Runout Length**	30	30 x 1.33 = 40	30 x 2.2 = 66
Group 1 (A)	35	35	35	**Group 5** (B)	40	40	40
Group 2				**Group 6** (L)	20	20	20
Group 4 (AE)	55	55	55	**Group 8**			
Group 9				**Group 11** Box 600	40	40	40
Group 11 Box 600	2 x 40 = 80	2 x 40 = 80	2 x 40 = 80	**Group 12**			
Other				**Other** Grp-11, 45	5	5	5
Other Grp-11, 45	3 x 5 = 15	3 x 5 = 15	3 x 5 = 15	**Other** Grp-11, 90	10	10	10
Other Grp-11, 90	10	10	10	**Other**			
Total Length	259	283	336	**Total Length**	159	175	212

Figure A16-2

Effect on Total Length and Friction Rate				
Duct System	**Example**	**TEL (Ft)**	**ESP (IWC)**	**FR (IWC/100)**
Fig 8-1	Cut to fit	427	0.30	0.070
	5" by 5' sag	428		0.070
Fig 8-8	Cut to Fit	418	0.39	0.093
	10" by 5' sag	458		0.085
	15% with sag	548		0.071
Cut to Fit: Installed with 0% to 4% excess length and 0.50 Inches sag per foot.				

Figure A16-3

- Figure A16-2 (next page) shows the consequences of non-complaint workmanship, as it pertains to total effective length.

Affect on Friction Rate

Figure A16-3 summarizes the affect on total effective length and design friction rate for the two example problems.

- No significant effect for Example 1 (flex runouts).
- Significant change for Example 2 (all runs are flexible duct). Note that the compression affect is much larger than short arc sag effect.

Affect on Airway Size

A lower friction rate translates to larger airway sizes, but this effect tends to be masked by a need to round to a

Workmanship Vs. Round Airway Diameter				
Duct run	**Cfm**	**Fitted**	**4%-Sag**	**15%-Sag**
TEL (Ft)		418	458	548
FR (IWC/100)		0.090	0.085	0.070
1 — ST1	106	7	7	7
2 — ST1	138	8	8	8
3 — ST2	119	7	7	8
4 — ST2	130	8	8	8
5 — ST3	126	7	7	8
6 — ST3	151	8	8	8
7 — ST4	162	8	8	8
8 — ST4	73	6	6	6
S-Trunk ST1	241	10	10	10
S-Trunk ST2	249	10	10	10
S-Trunk ST3	264	10	10	10
S-Trunk ST4	235	9	9	10
S-Trunk ST5	748	16	16	16
R2	362	11	11	12
R3	386	11	11	12
RT1 and R1	241	9	9	10
RT2	748	16	16	16
Sizes read from ACCA Duct Sizing Slide Rule. Standard size plus 0.49 Inches, or less, rounded down to a standard size. Standard size plus 0.50 Inches, or more, up to a standard size.				

Figure A16-4

standard size. Figure A16-4 compares standard airway sizes for the Figure 8-8 flexible duct system for three workmanship scenarios (cut to length, short arc sag and compression).

- No change in airway sizes for short arc sag.
- About one third of the airway sizes increased by compression with sag.

Appendix 17 (Some items do not appear in the standard)

Symbols and Abbreviations

Delta

ΔGrains	Grains difference for a psychometric process
ΔT	Temperature difference (see TD)

A

A	Area in square feet
ACF	Altitude correction factor (see Table 10A, ***Manual J*** Eighth Edition)
Aprox	Approximate
ASHP	Air-source heat pump
ASP	Available static pressure (pressure available to move air through straight runs and fittings, which equals external static pressure minus the pressure drop for air-side components that were not in place during the blower test)
Asoc	Associated

B

BPF	By-pass fraction (regarding bypass Cfm for constant Cfm duct)
Btuh	British Thermal units per Hour

C

C	Compression (for flexible duct)
C-Btuh	Sensible cooling load (Btuh)
C-Cfm	Cooling Cfm
CF	Cooling factor in Cfm per Btuh of sensible load
Cfm	Cubic feet of air per minute
CL	Leakage class (Cfm leakage per 100 SqFt of duct surface area)
C-Load	Sensible cooling load (Btuh)
COP	Coefficient of performance
CPL	Component pressure loss in IWC (total pressure drop for air-side components that were not in place during the blower test)

D

DAT	Discharge air temperature (°F)
D-Cfm	Design Cfm
Dsn	Design

E

EAT	Entering air temperature (°F)
ECM Blower	A blower driven by a motor equipped with sensors and software. The motor obeys software commands to maintain a Cfm set point over a range of external static pressures
EER	Energy Efficiency Ratio
EL	Fitting equivalent length in feet
ESP	External static pressure (pressure value from manufacturer's blower table, which must be adjusted for the pressure drop for air-side components that were not in place during the blower test)
EWB	Entering wet bulb temperature (°F)

F

Flex	Flexible wire helix duct
Fpm	Air velocity in feet per minute
FR	Friction rate as IWC/100 Feet of Straight Duct
Ft	Feet

G

Grains	Grains of moisture in a pound of air

H

H-Btuh	Heating load (Btuh)
H-Cfm	Heating Cfm
HF	Heating factor in Cfm per Btuh of heating load
H-Load	Heating load (Btuh)
HSPF	Heating season performance factor
HVAC	Heating ventilation and air conditioning

I

IWC	Inches water column

K

Kw	Kilowatts

L

LAT	Leaving air temperature (°F)
Lat Btuh	Latent Btuh

M

MAT	Mixed air temperature (°F)

N

NC	Noise criteria (see Appendix 13)

P

PA	Pascals (1.0 IWC = 249 PA)
PD	Pressure drop in IWC

R

R	Radius in inches or feet
RA	Return air
Rigid	Metal duct with or without liner, or duct board
RPM	Revolutions per minute

S

SAT	Supply air temperature (°F)
SC	Straight cut length for flexible duct (Feet)
Seer	Seasonal energy efficiency rating
Sens Btuh	Sensible Btuh
SL	Span length for flexible duct (Feet)
SLD	Straight line distance in feet
SP	Thermostat set point

T

TD	Dry-bulb temperature difference in °F (across a furnace heat exchanger or electric heating coil; across a cooling coil; between room air and supply air; between return air and air entering HVAC equipment; across a duct wall; etc.)
TEL	Total equivalent length in feet
Ton	12,000 Btuh of cooling capacity

V

V	Air Velocity in feet per minute
VAV	Variable air volume
VP	Velocity pressure in IWC

W

WSHP	Water-source heat pump

Appendix 18 (Use not Mandated by the Standard)

Manual D Worksheets

Training exercises that use templates familiarize the student or practitioner with ***Manual D*** tables, equations and procedures. This training is valuable for proper use of third-party, ***Manual D*** software.

- The duct sizing worksheets can be used for hand calculations, and/or to summarize solutions to example problems.
- Blank copies of the duct sizing worksheets are provided by the last three pages of this appendix.

A18-1 Effective Length Worksheet

The Effective Length Worksheet is not dependant on output from any other worksheet. Do these calculations first:

- Make a sketch of the duct system geometry.
- Show location of the blower equipment.
- Show all supply outlets with supply grille identification numbers (in some cases more than one supply outlet may be used for a room).
- Show all return grilles with return grille identification numbers.
- Show all straight run sections with measured or calculated lengths (reasonably accurate estimate is acceptable, say ± 10%).
- Show the location of all fittings with related group numbers and equivalent lengths.
- Select candidates for the longest supply run and the longest return run (or do them all, if not sure).
- Use the worksheet to calculate effective lengths.

A18-2 Friction Rate Worksheet

The Friction Rate Worksheet processes equipment manufacturer's performance data, and output from the Effective Length Worksheet. This is the most important part of the ***Manual D*** procedure because it demonstrates (or not) that blower performance is compatible with the proposed duct system.

- Use ***Manual J*** to calculate the block heating load, the block sensible cooling load, and the block latent cooling load.
- Use ***Manual J*** loads, expanded manufacturer's performance data, and ***Manual S*** procedures to select equipment.
- The design value for blower Cfm is determined when equipment is selected. (Equipment performance depends on blower Cfm at the selected blower speed, and performance is matched to the ***Manual J*** loads.)
- Obtain the equipment manufacturer's blower table for the selected equipment.
- For the design Cfm, read the corresponding external static pressure value from the blower table.
- Read the blower table notes to find what equipment was in place when the blower-table test was conducted.
- If an air-side component(s), other than those listed in the blower table notes, is to be installed in the blower cabinet, or at some point in the duct system, obtain manufacturer's performance data for the component(s). Then, for the design Cfm, read the corresponding pressure drop(s) for the component(s).
- Enter the blower Cfm, blower table pressure, and ancillary component(s) pressure drop(s) on the Friction Rate worksheet.
- Also enter the pressure drop for one supply air grille, one return air grill, and one hand damper (the default values are 0.03, 0.03 and 0.03 IWC, or obtain exact values from manufacturer's performance data).
- Use the Friction Rate Worksheet to calculate available static pressure, total effective length, and the design friction rate value.

A18-3 Duct Sizing Worksheet

The Duct Sizing Worksheet and a duct slide rule, or friction cart, convert the design friction rate and a set of Cfm values to duct airway sizes. Then air velocities are checked, and if air velocity is excessive, airway size is increased to lower air velocity.

- Use ***Manual J*** to calculate the block heating load and the block sensible cooling load (the same information that was used to select equipment).
- Enter the block heating load, block cooling load, and the design value for blower Cfm in the spaces at the top of the worksheet, then calculate the heating factor (HF) and the cooling factor (CF).
- Refer to the Friction Rate Worksheet and copy the design friction rate value to the top of the Duct Sizing Worksheet.

- Use ***Manual J*** to calculate the room heating and cooling loads.
- If two or more supply outlets are used for a room, split the room heating load and room cooling load into parts.
- Enter the heating and cooling loads for all supply outlets on the worksheet (correlate with outlet identification numbers).
- Multiply the heating loads by the heating factor to find heating Cfms and enter these values on the worksheet.
- Multiply the cooling loads by the cooling factor to find cooling Cfms and enter these values on the worksheet.
- For each supply outlet, select the larger of the heating Cfm value or cooling Cfm value, and enter the design Cfm values on the worksheet.
- Use a duct slide rule, or friction chart, to find the round duct runout size (the sizing tool must be for the actual airway material), and enter the preliminary sizes on the worksheet.
- Use a duct slide rule, or friction chart, to check airway velocity and enter the velocity values on the worksheet.
- If one or more velocities are too high, re-size the duct for an acceptable velocity, and enter the final sizes on the worksheet.
- Correlate trunk sections with downstream branch sections, and calculate heating and cooling Cfm for each unique section of trunk duct, then enter these values on the worksheet (see Sections 6-12 through 6-16).
- Use the heating and cooling factors to determine heating and cooling Cfm and the design Cfm (larger of the two values).
- Use a duct slide rule, or friction chart, to find the round duct runout size (the sizing tool must be for the actual airway material), and enter the preliminary sizes on the worksheet.
- Use a duct slide rule, or friction chart, to check airway velocity and enter the velocity values on the worksheet.
- If one or more velocities are too high, re-size the duct for an acceptable velocity, and enter the final sizes on the worksheet.
- Assign Cfm values to each return grille and repeat the process for the return-side of the system.

Effective Length Worksheet

Element	Supply Run ID Number				Element	Return Run ID Number			
Trunk Length					Trunk Length				
Trunk Length					Trunk Length				
Trunk Length					Trunk Length				
Runout Length					Runout Length				
Group 1					Group 5				
Group 2					Group 6				
Group 3					Group 7				
Group 4					Group 8				
Group 8					Group 10				
Group 9					Group 11				
Group 11					Group 12				
Group 12					Group 13				
Group 13					Other				
Other					Other				
Total Length					Total Length				

Friction Rate Worksheet

Step 1) Manufacturer's Blower Data

External static pressure (ESP) = ______ IWC Cfm = __________

Step 2) Component Pressure Losses (CPL)

Direct expansion refrigerant coil	______
Electric resistance heating coil	______
Hot water coil	______
Heat exchanger	______
Low efficiency filter	______
High or mid-efficiency filter	______
Electronic filter	______
Humidifier, UV lights, other	______
Supply outlet	______
Return grille	______
Balancing damper	______
Zone damper (full open)	______
Total component losses (CPL)	______ IWC

Step 3) Available Static Pressure (ASP)

ASP = (ESP - CPL) = (______ - _______) = _______ IWC

Step 4) Total Effective Length (TEL)

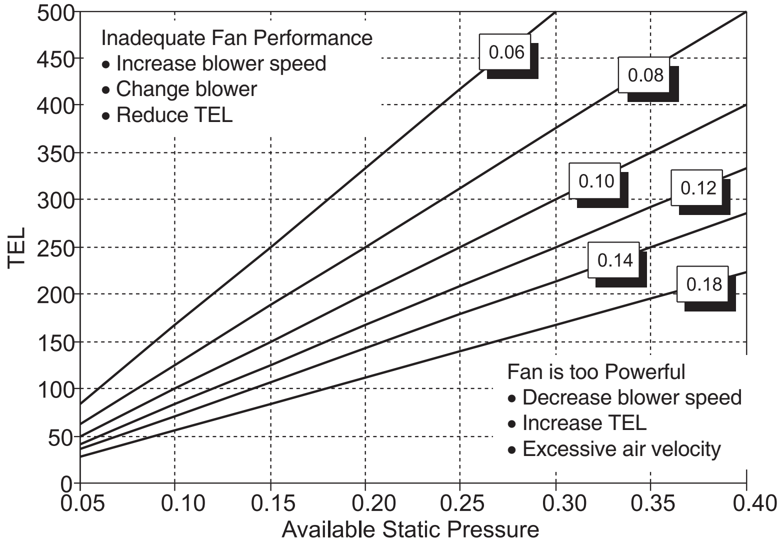
Supply-side TEL + Return-side TEL = (_______ + _______) = _______ Feet

Step 5) Friction Rate Design Value (FR)

FR value from friction rate chart = ______ IWC/100

$$FR = \frac{ASP \times 100}{TEL}$$

Friction Rate Chart



Duct Sizing Worksheet

HF = Blower Cfm / Manual J Heat Loss = ( ) / ( ) =
CF = Blower Cfm / Manual J Sensible Heat Gain = ( ) / ( ) =

FR Value

Supply-Side Runouts

Supply - Trunk	Heating Btuh	Cooling Btuh	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
S1 —									
S2 —									
S3 —									
S4 —									
S5 —									
S6 —									
S7 —									
S8 —									
S9 —									
S10 —									
S11 —									
S12 —									

Supply-Side Trunks

Run numbers:	S-Trunk 1							
Run numbers:	S-Trunk 2							
Run numbers:	S-Trunk 3							
Run numbers:	S-Trunk 4							

Return-Side Runouts

Return - Trunk	Associated Supply Runs	Heating Cfm	Cooling Cfm	Design Cfm	Round Size	Velocity Fpm	Final Size	Normed Cfm
R1 —								
R2 —								
R3 —								
R4 —								
R5 —								
R6 —								
R7 —								
R8 —								

Return-Side Trunks

Run numbers:	R-Trunk 1							
Run numbers:	R-Trunk 2							
Run numbers:	R-Trunk 3							
Run numbers:	R-Trunk 4							

1) Room heating and cooling Btuh obtained per Section 6 guidance.
2) Heating Cfm for runouts = HF x Heating Btuh; Cooling Cfm for runouts = CF x Sensible Cooling Btuh.
3) For trunks, sum heating Cfm values for branches served by the trunk, and sum the cooling Cfm values for branches served by the trunk. The design Cfm for branches and trunks is equal to the larger of the heating Cfm or cooling Cfm values for the run.
4) Round size is based on FR value. Final size is based on FR value if air velocity is acceptable, or the maximum allowable velocity value. Final size may be a standard round size, or a standard equivalent rectangular size.
5) Normed Cfm = Normalized Cfm for air balancing single-zone systems (see Section 6-23). For zoned systems, see Section 9-11.
6) Per ***Manual Zr***, Sections 7-9 and 8-11, a bypass airway is sized for 900 Fpm and the bypass Cfm from the Bypass Cfm Worksheet.

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