

5. Air Systems, Heating and Cooling, Part I

In Chapters 3 and 4, we learned how to design and draw hydronic and electrical heating systems. Both these systems are excellent for their purpose. Not so many years ago, particularly in cold climates, heating was all that was required for a building HVAC system. Adequate ventilation was usually provided by natural infiltration in construction that was deliberately not airtight. In warmer climates, summer heat was relieved, somewhat, by the use of fans. With the advent of economically and mechanically practical refrigeration machines, the demand for interior space cooling, in addition to the usual heating, grew quickly. This led rapidly to the development of ducted air systems, which could provide cool dry air for summer comfort and humidified warm air for winter comfort. In this chapter, we will learn the principles of ducted air systems and their application to the design of warm air heating. In Chapter 6, the use of refrigeration machines to provide cooling (and heating) will be studied, using the information on ducted air distribution that we will learn in this chapter. Study of this chapter will enable you to:

1. Recognize and understand all the components of low pressure, low velocity, all-air heating systems.
2. Calculate the heat-carrying capacity of ducted airflow.
3. Be familiar with the characteristics of air pressure in ducted air flow, including measurement techniques.
4. Calculate duct air pressures including static pressure, velocity pressure and total pressure.
5. Be familiar with the major types of warm air furnaces, including components, accessories, duct arrangements and operating characteristics.
6. Understand the components and construction of duct systems, including ducts and fittings.

7. Be completely familiar with air distribution outlets, including registers, diffusers and grilles. This includes understanding the outlets' operating characteristics and how to select, locate and properly apply them.
8. Understand the various types of all-air systems in use today. This includes understanding the duct arrangements of single-zone systems.
9. Calculate air friction in duct systems using charts, tables and duct slide rule-type calculators. This includes round, rectangular and oval ducts of all materials.
10. Understand all stages of the design procedure for warm air duct systems. This includes duct size calculation by four different methods, as applicable to the duct system.

All-Air Systems

Our discussion in this chapter will be restricted to arrangements that are known as all-air systems: specifically, low pressure, low velocity, all-air systems. These systems use ducts to carry warm or cool air from the central point where it is "made" to the various spaces in a building. At these terminations, the air is distributed within the space by specially designed air outlets. This type of system is completely different from water/air systems, such as those shown schematically in Figure 3.1(b) and (c). There, the heat-carrying medium is water, which transfers its heat (or coolness) to air at the terminal point. Such a system is really hydronic. It uses a heat exchange device such as a fan coil unit to deliver the heating or cooling in the form of warm or cool air at the terminals. Water/air systems are commonly used in large buildings where the distances involved make piping more practical and economical than ductwork. There are many other considerations involved in the selection of a system type for a building. They are, however, not the responsibility of the technologist and will, therefore, not be discussed here.

5.1 Air System Characteristics

The great advantage of all-air systems is their ability to provide year-round comfort air conditioning with a single system. Originally, the term *air conditioning* referred only to cooling, and even today it is used in this sense. However, the HVAC profession tends to use the term in its broadest sense, that

is, conditioning of room air to provide year-long comfort. This means control of air temperature and humidity. It also means controlled ventilation and air purification.

It might be helpful at this point to review the human comfort material in Chapter 1. Briefly, most people are comfortable with the following indoor conditions:

Winter: 68–74°F DB, 35–50% RH, maximum air velocity 40 fpm

Summer: 73–79°F DB, 25–60% RH, maximum air velocity 50 fpm

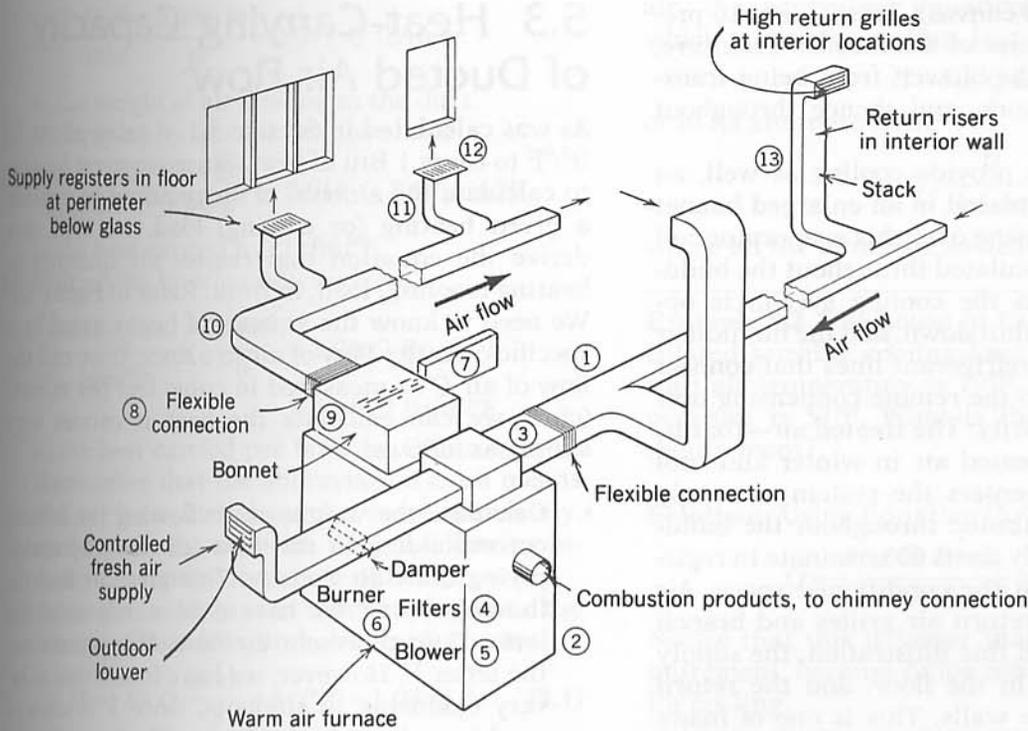
All year: mean radiant temperature (MRT) approximately the same as the air temperature

These are ideal conditions, which, as we will learn are almost impossible to maintain uniformly throughout a space. Even in well-designed spaces the temperature may vary vertically as much as 5°F, and from the center of the room to the walls, it varies even more. The extent of these variations depends on the design and placement of the room air outlets. This subject will be discussed in detail in the section dealing with air registers and diffusers.

The principal disadvantage of air as a heat-carrying medium is its required volume. Air weighs approximately 0.075 lb/ft³ (slight variation with temperature). This means that 1 lb of air occupies 13.3 ft³. Since the specific heat of air is only 0.24 Btu/ft³/°F, it takes 13.3/0.24 or 55 ft³/°F to carry 1 Btu of heat! This accounts for the large cross-sectional area of duct required to supply even a relatively small heating or cooling load, such as in a residential installation. (By way of comparison, 55 ft³ of water carries 3458 Btu/°F). On the other hand, air is easily humidified (for winter requirements) and almost as easily dehumidified. It is also easily filtered, cleaned and exchanged with fresh air when required. In the United States, the overwhelming majority of residential heating and cooling installations are all-air systems, as are a large portion of commercial installations. It is, therefore, obvious that their advantages outweigh their disadvantages, from both engineering and economic points of view.

5.2 Components of All-Air Systems

Although the details of air systems vary from one installation to another, the essential components



LEGEND

- | | |
|---------------------------------|---------------------------------------|
| ① Return air duct | ⑦ Humidifier |
| ② Alternate return air location | ⑧ Flexible duct connection |
| ③ Return air duct connection | ⑨ Evaporator coil location (optional) |
| ④ Air filter | ⑩ Main supply duct |
| ⑤ Blower | ⑪ Branch supply duct |
| ⑥ Burner | ⑫ Supply air register |
| | ⑬ Branch return duct |

Figure 5.1 Typical warm air furnace installation showing components and accessories. When cooling is required, a larger bonnet, capable of containing an (A-frame) evaporator coil, is constructed. The arrangement shown has the furnace in the basement of a one-story building. Warm air ducts are arranged in a perimeter system with supply registers in the floor under windows and return grilles high on inside walls. This arrangement is suitable for year-round heating and cooling.

of all forced-air systems are shown in Figure 5.1. Follow the description of the parts with the labels on the illustration. Air from the building spaces is returned to the furnace via a system of return air ducts ①. This return air may enter at the top of the furnace as shown or at the bottom ②. The return air passes through a filter located either at the duct entrance ③ or within the furnace enclosure ④. The filter can be either mechanical or electrostatic. The

air then passes through the blower ⑤, which adds static pressure and velocity to the air stream. It proceeds to the burner heat exchange mechanism ⑥, where it is heated and its temperature raised between 45 and 80°F. The air is then humidified by a humidifier ⑦ located at the bonnet or immediately thereafter, in the first supply duct section ⑧. Notice that the supply and return ducts are connected to the furnace with flexible connec-

tions (usually treated canvas). This is done to prevent the vibration noise of the furnace enclosure, which is caused by the blower, from being transmitted to the ductwork, and thence throughout the building.

If the system is to provide cooling as well, an evaporator coil ⑨ is placed in an enlarged bonnet (plenum). The air passing over this evaporator coil is cooled and then circulated throughout the building. Obviously, when the cooling system is operating, the heater is shut down, and the humidifier is disconnected. The refrigerant lines that connect the evaporator coil to the remote condensing unit are not shown, for clarity. The treated air—that is, humidified filtered heated air in winter and cool dry air in summer—enters the system of supply ducts ⑩ and is distributed throughout the building. The branch supply ducts ⑪ terminate in registers ⑫ or diffusers in the conditioned space. Air is returned through return air grilles and branch return air ducts ⑬. In this illustration, the supply registers are placed in the floor, and the return grilles high on inside walls. This is one of many possible arrangements that will be discussed in detail in the section on air outlets later in this chapter.

Notice also in Figure 5.1 that there is a controlled fresh air supply that connects into the return air system. This provides make-up air to compensate for air that is exhausted from kitchens and bathrooms. It is customary not to return air from these spaces because of odors and high humidity. The amount of make-up air is easily controlled by a damper in the intake air duct. Combustion air can be taken from the basement or from a separate combustion air intake (not shown). The latter is the preferred method, particularly in cold climates, because it avoids infiltration of cold outside air into the basement, which can appreciably increase the building heating load.

Additional components of air systems that are not shown in Figure 5.1 include air flow (volume) dampers in branch ducts and special duct fittings. In commercial systems, there are many sophisticated air temperature and air volume controls devices such as mixing boxes, variable air volume (VAV) boxes and terminals and the like. These, however, are not normally the responsibility of HVAC technologists except for showing them on the working drawings. All the components of air systems will be discussed in detail later in this chapter. First, however, an understanding of the properties of moving air is required. This, therefore, will be the subject we turn to next.

5.3 Heat-Carrying Capacity of Ducted Air Flow

As was calculated in Section 5.1, it takes about 55 ft^3/F to carry 1 Btu of heat. Since we must be able to calculate the amount of air required to provide a given heating (or cooling) load, we will now derive the equation that relates air quantity to heating (cooling) load, in Btuh. Refer to Figure 5.2. We need to know the amount of heat carried by a specific quantity flow of air in a duct. If we call the flow of air Q as measured in cubic feet per minute (cfm), we can calculate the heat it carries very simply as follows:

- Calculate the weight of air flowing (in lb/min) corresponding to this flow (in cfm), by multiplying Q by air density. This gives us flow (in lb/min). (*Note:* We have deliberately used the letter Q to represent air flow. Many texts use the letter V . However, we have found this to be very confusing to students, since V is always used for volume and velocity. The letter Q normally represents volume flow such as cfm.)
- Multiply the flow (lb/min) by the specific heat to obtain the Btu/min/ $^{\circ}\text{F}$ heat flow.
- Convert the heat flow to Btuh/ $^{\circ}\text{F}$.
- Using the design temperature change in the air, calculate the heat flow in Btuh, corresponding to a flow of Q cfm.

The calculation is, therefore, as follows:

- (a) We begin with an air flow in a duct of Q cfm.
- (b) Multiplying by density, we have

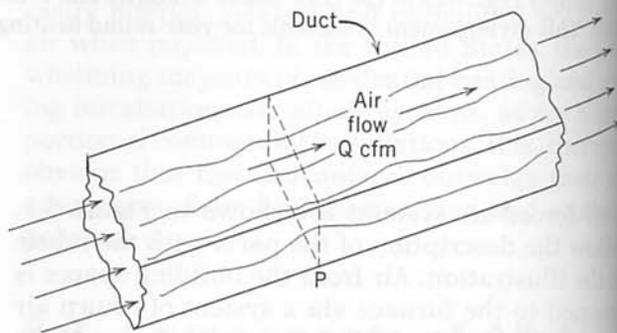


Figure 5.2 The quantity of air flowing in a duct is measured in cubic feet per minute (cfm), as it flows past a cross-sectional plane P .

$$\frac{Q \cancel{\text{ft}^3}}{\text{min}} \times \frac{0.075 \text{ lb}}{\cancel{\text{ft}^3}} = 0.075 Q \text{ lb/min}$$

as the weight of air flowing in the duct.

(c) Multiplying by specific heat, we have

$$\frac{0.075 Q \cancel{\text{lb}}}{\text{min}} \times \frac{0.24 \text{ Btu}}{\cancel{\text{lb}}/^\circ\text{F}} = 0.018 Q \text{ Btu/min}/^\circ\text{F}$$

as the heat carried by Q cfm/ $^\circ\text{F}$.

(d) Multiplying by 60 min/h, we have

$$\frac{0.018 Q \text{ Btu}}{\cancel{\text{min}}/^\circ\text{F}} \times \frac{60 \cancel{\text{min}}}{\text{h}} = 1.08 Q \text{ Btu/h}/^\circ\text{F}$$

$$= 1.08 Q \text{ Btuh}/^\circ\text{F}$$

as the heat carried per hour by Q cfm, per $^\circ\text{F}$. (Remember that the abbreviation Btuh means Btu per hour, and is written more accurately, mathematically, as Btu/h. We are following industry convention in the use of Btuh.)

(e) Assuming a drop in temperature Δt , the amount of heat H in Btuh, lost by Q cfm is

$$H = 1.08 Q \frac{\text{Btuh}}{^\circ\text{F}} \times \Delta t (^\circ\text{F}) = 1.08 Q \Delta t \quad (5.1)$$

where

H is heat delivered in Btuh

Q is airflow in cubic feet per minute and

Δt is the change (drop) in temperature of the air from supply to return.

(f) Conversely, if we know the amount of heat flow required and want to know how much air is needed to supply the heat, we can use the same equation. Since $H = 1.08 Q \Delta t$

$$Q = \frac{H}{1.08 \Delta t} \quad (5.2)$$

where the terms are the same.

An example should make the use of this important equation clear.

Example 5.1 A heat loss calculation for a new residence indicates a heat loss of 84,000 Btuh for the entire house. What is the required air output of the furnace, assuming a return air temperature of 68°F and a furnace air supply temperature of 140°F .

Solution: The temperature difference between incoming and outgoing air is $140^\circ\text{F} - 68^\circ\text{F} = 72^\circ\text{F}$. Using Equation (5.2), we have

$$Q = \frac{84,000 \text{ Btuh}}{1.08 (72^\circ\text{F})} = 1050 \text{ cfm}$$

The same equation, with one small change, can be used for cooling. Due to the higher density of cold

air, the equivalent equation to Equation (5.1), which is used for cooling, is

$$H = 1.1 Q \Delta t \quad (5.3)$$

or in its alternate form:

$$Q = \frac{H}{1.1 \Delta t} \quad (5.4)$$

where all the terms are as defined previously.

Example 5.2 The house in Example 5.1 has a calculated sensible cooling load of 46,000 Btuh. Return air temperature is 79°F , and supply air temperature is 57°F . What is the blower air output requirement?

Solution: Using Equation (5.4), we have

$$Q = \frac{46,000 \text{ Btuh}}{1.1 (79 - 57)^\circ\text{F}} = 1900 \text{ cfm}$$

Notice that this is larger than the heating air requirement, because Δt for heating is larger than Δt for cooling.

5.4 Air Pressure in Ducts

Refer to Figure 5.1. The furnace blower (also called the system fan) delivers energy to the air in the system. This energy takes the form of air pressure, which causes the air to circulate through the supply and return ducts. This pressure can be expressed as the sum of two quantities: static pressure and velocity pressure (energy). Expressed in an equation, this is written

$$\text{Total pressure} = \text{Static pressure} + \text{Velocity pressure}$$

or

$$P_T = P_S + P_V \quad (5.5)$$

Static pressure, also called static head, is the pressure that the air has at rest. It is sometimes called spring pressure because it can be thought of as the pressure that pushes on the sides of the duct. See Figures 5.3 and 5.4. It can also be thought of as the potential energy of the system that is gradually converted to kinetic energy in order to keep the air moving against the system friction. Indeed, it is sometimes defined as the pressure, or static head, required to overcome the system friction. Since that is so, static pressure drops gradually and continuously as we move away from the blower along the supply duct, provided that the duct dimension does not change.

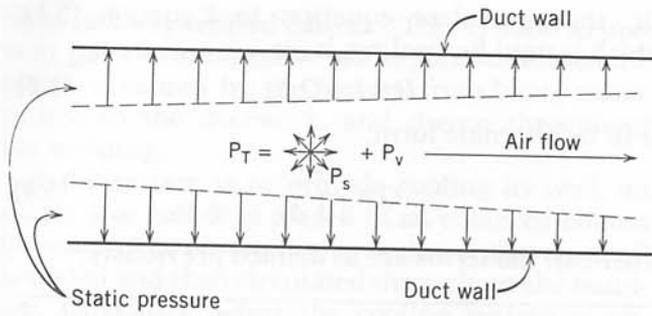
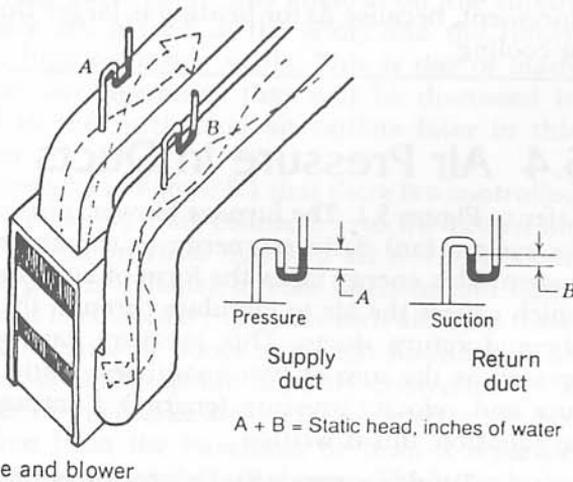


Figure 5.3 The total pressure (P_{TOT}) of air flowing in a duct is the sum of the static pressure P_s and the velocity pressure P_v . The static pressure acts like a spring and pushes against the duct walls. The energy of velocity pressure is felt only in the direction of flow and remains constant as long as the duct size does not change. However, the static pressure drops as we proceed in the direction of flow. It is "used up" by friction.



Furnace and blower

Figure 5.4 The static pressure in the supply duct is positive and can be measured by a manometer as A inches of water column. The pressure in the return duct is negative (suction) and is measured as B inches of suction.

Velocity pressure is not pressure at all; it is the kinetic energy of the moving air stream. It is converted to pressure by a process called *static regain* when the velocity of the moving air is changed. When you extend your hand through the window of a moving automobile, you feel this air pressure. The same is true when a stream of water from a hose strikes the hand. The fluid velocity in both instances drops sharply, and its kinetic energy is converted to pressure. To give you an impression

of the magnitude of this pressure, a simplification will help. A maximum air velocity of 11.4 mph is common in residential main supply ducts. This velocity corresponds to a speed of 11.4 mph. The static pressure is, therefore, quite small. It is not so important in calculating losses in ductwork as we will learn later on. It is also very important in commercial installations where an air velocity of 2000 fpm (22.7 mph) and even higher is common. In residential duct work, velocity pressure is frequently ignored because it is so small. The static pressure (or head) can be accurately calculated from Equation (5.6)

$$P_v = (V/4005)^2$$

where

P_v is the velocity pressure in inches water column

V is the air velocity in feet per minute (fpm)

A few accurate calculations will help you get a feel for the pressures involved.

Example 5.3 A residential duct system is shown in Figure 5.4 with air velocities of 900 fpm in the main duct and 600 fpm in the branches. What are the static pressures in both?

Solution: Using Equation (5.6), we have

(a) in the main duct

$$P_v = (900/4005)^2 = 0.05 \text{ in. w.g.}$$

(b) in the branch duct

$$P_v = (600/4005)^2 = 0.022 \text{ in. w.g.}$$

Since the entire pressure available for the air to do work in a residential system rarely exceeds 0.1 in. w.g., the velocity pressure in the mains is not so important in marginal designs, whereas generally it can be ignored in the branch ducts. Exact calculations, as we will learn, should consider velocity pressure. However, residential duct systems are almost always oversized to reduce noise, and the air flow is regulated and balanced by dampers. Because the loss in fittings is usually overestimated for these reasons, most designers do not calculate velocity pressures in small or medium-size residential design.

The diagrams in Figure 5.5 should help you understand the pressure relationships of air in a duct. Because the pressures involved are so small, an inclined tube manometer is used in actual field work. By inclining the tube, a very small pressure can cause a large movement of the

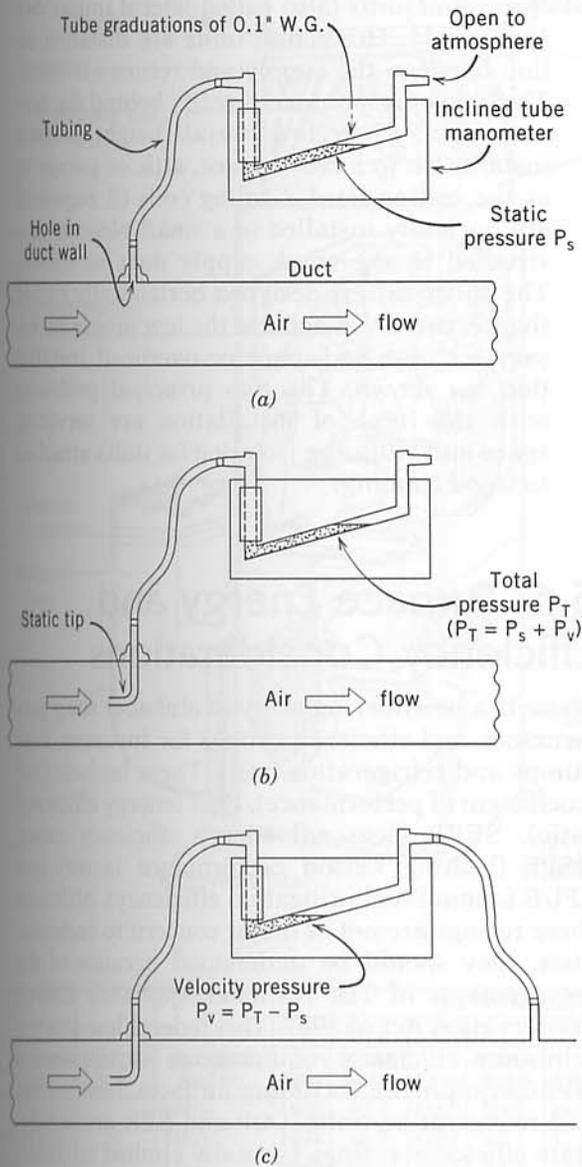


Figure 5.5 (a) Static pressure in a duct can be readily measured using an inclined tube manometer, which is also known as a draft gauge. (b) Placing a static tip into the center of the duct will measure total pressure. (c) Velocity pressure is the difference between total pressure and static pressure. It can be measured by developing a back pressure P_S acting against a total pressure P_T .

(oil) in the tube. The tube is marked (graduated) in tenths of an inch of water. Commercial manometers can be read to an accuracy of 0.03 in. w.g. (water gauge). Some units are marked as inches water column (WC), which is identical to water gauge (w.g.). The two terms are used interchangeably, with w.g. being more common.

In Figure 5.5(a) a hole in the duct is connected by a piece of flexible hose to the inclined tube manometer, which is also known as a *draft gauge*. Since the other end of the inclined tube is open to the atmosphere, the gauge will read the static (spring) pressure that is pushing on the duct walls. In Figure 5.5(b), a simple tube called a static tip is inserted in the hole, facing the air stream. It now measures the static pressure plus the velocity pressure, that is, the total pressure P_T . Imagine the face of the tube as another duct wall. As such, it measures static pressure. To this pressure P_S is added the velocity pressure P_V caused by the drop in air velocity inside the tube to zero, giving a reading of total pressure P_T .

If we now connect the other end of the gauge to another hole slightly downstream, we will develop a back-pressure of P_S pushing against a forward pressure of P_T in the gauge. The result is velocity pressure because

$$P_T = P_V + P_S \quad (5.5)$$

and, therefore,

$$P_T - P_S = P_V$$

We will have a great deal more to say about duct pressures when we study friction losses in ducts and fittings. At this point, you should remember two facts:

1. Static pressure is required to overcome duct friction.
2. Velocity pressure is usually small and remains constant as long as the cfm and duct area do not change.

Warm Air Furnaces

The heart of a warm air heating system is either a warm air furnace or a heat pump. In this chapter, we will study furnaces; in Chapter 6 heat pumps, which supply both heating and cooling, will be studied. Warm air furnaces are used primarily in residences and small commercial and institutional buildings.

5.5 Furnace Components and Arrangements

The components of a warm air furnace, as we can see in Figure 5.1 are standard. They consist of the heating unit itself (which can be either gas, oil or

electric fueled), the blower, humidifier, air filter, supply duct plenum and controls. The supply duct plenum may contain an evaporator coil if cooling is being supplied in addition to heating. In such installations, the refrigeration compressor and condenser are remotely located—usually outside the building. The arrangement of the components and the overall size and shape of the furnace depend on where the furnace is to be installed and, more particularly, where the supply and return ducts are to go. The four basic physical designs of furnaces follow.

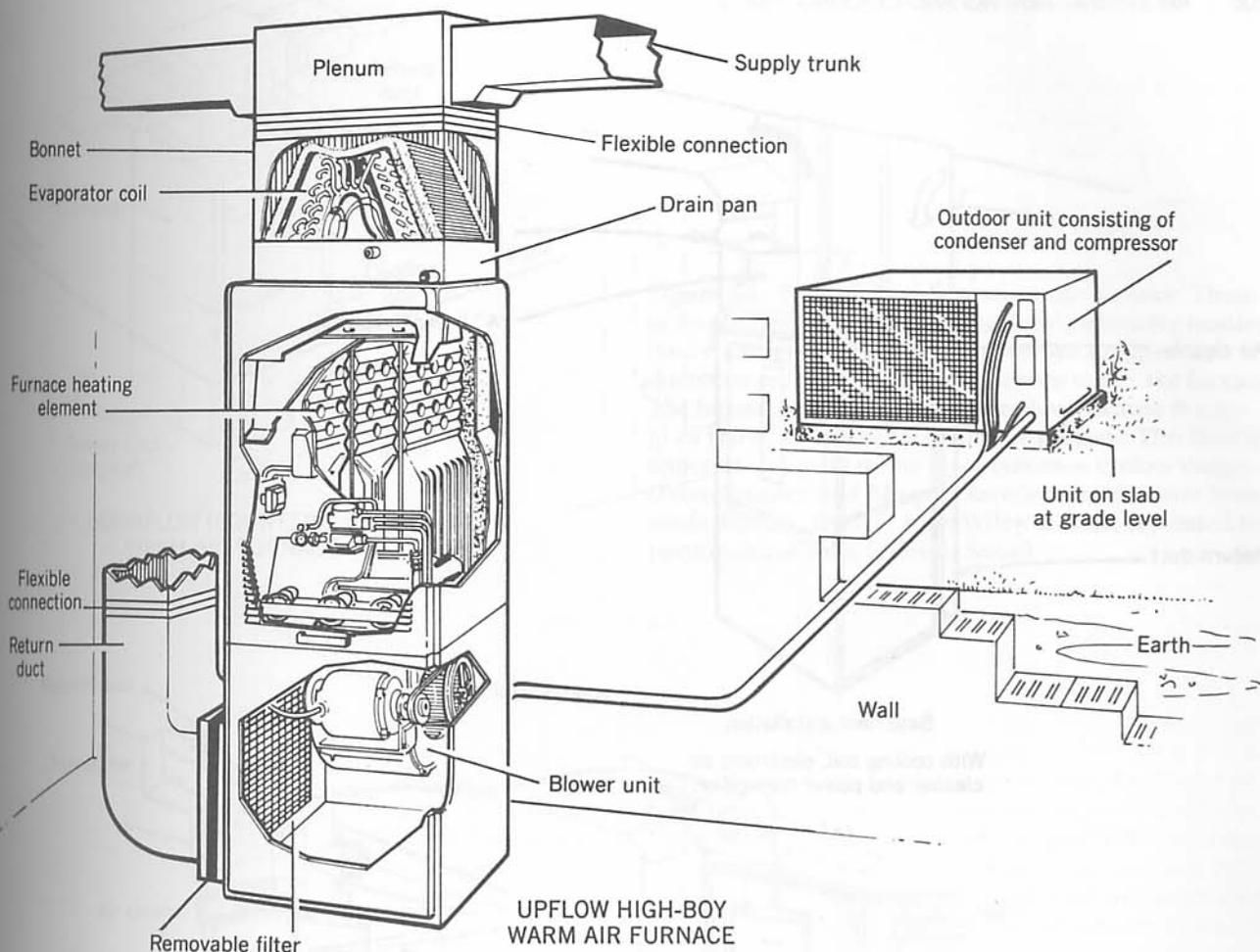
- (a) *Upflow (high-boy) units.* See Figures 5.6 and 5.7. These units are full-height. The supply ducts exit the top plenum, which has space for an evaporator cooling coil. The return duct runs overhead but drops to enter the furnace housing at the bottom, through a side panel. The blower is normally installed at the bottom of the enclosure, adjacent to the entry point of the return duct. Units of this design can be installed in a full-height basement or in a closet/utility room that can accommodate overhead supply ducts and overhead and bottom return ducts.
- (b) *Downflow (counterflow) units.* See Figures 5.8 and 5.9. These units were originally designed specifically for use with perimeter heating-only systems in slab-on-grade or low crawl space houses. A typical unit is installed in a closet or utility room, directly over a small sheet metal plenum. (An enclosed crawl space should not be used as a plenum because of problems with pesticides and insects.) Ducts emerge from the sheet metal plenum to feed perimeter floor outlets or a perimeter loop. (See Figure 5.46, page 271.) The almost universal requirement for summer cooling in new construction has led to the addition of cooling coils in these downflow units. The blower, which is located above the heat exchanger, supplies conditioned air directly into the underfloor plenum. Return air enters the furnace enclosure at the top.
- (c) *Low-boy units.* See Figure 5.10. These units are similar to the upflow (high-boy) units. The return duct enters at the top of the furnace enclosure. The blower is located at floor level as in the high-boy unit. The heat exchanger, however, is also at this level, to the side or in front of the blower. This requires a wider casing as shown. A typical application for this design is a low ceiling basement where a full-height unit with plenum cooling coil would exceed the available ceiling height.

- (d) *Horizontal units* (also called lateral units). See Figure 5.11. Horizontal units are installed in-line between the supply and return air ducts. The blower is mounted directly behind the heat exchanger. Their low overall height permits installation in a crawl space, attic or garage or at the ceiling level. Cooling coils (if required) are normally installed in a small plenum constructed in the trunk supply duct as shown. The entire unit is designed horizontally rather than vertically to achieve the low profile necessary for cramped-space or overhead installation, as shown. The two principal problems with this type of installation are servicing space and vibration isolation for units attached to wood framing.

5.6 Furnace Energy and Efficiency Considerations

There is a bewildering array of alphabet soup performance and efficiency ratings for furnaces, heat pumps and refrigeration units. These include COP (coefficient of performance), EER (energy efficiency ratio), SEER (seasonal energy efficiency ratio), HSPF (heating season performance factor) and AFUE (annual fuel utilization efficiency). Although these ratings are not of major concern to technologists, they should be understood because of the requirements of The National Appliance Energy Conservation Act of 1987. This federal law sets out minimum efficiency requirements for all sorts of HVAC equipment, including furnaces, heat pumps and refrigeration units. COP and EER are steady-state efficiency ratings normally applied to water-source heat pumps. HSPF and SEER are seasonal efficiency ratings that are usually used for air source heat pumps.

Of all the ratings, the only one applicable to warm air furnaces is AFUE. It is applied to gas- and oil-fired heating equipment and is listed in the *Directory of Certified Furnace and Boiler Efficiency Ratings* published by GAMA (Gas Appliance Manufacturer's Association). The AFUE furnace rating was developed to take into account actual operating conditions rather than only laboratory-style tests. These conditions include flue losses and other variable factors such as on-off cycling. Cycling is related to weather patterns, design conditions selected and deliberate oversizing. The federal law, effective January 1992, mandates a minimum AFUE of 78% for warm air furnaces larger than 45 MBH and smaller than 225 MBH.



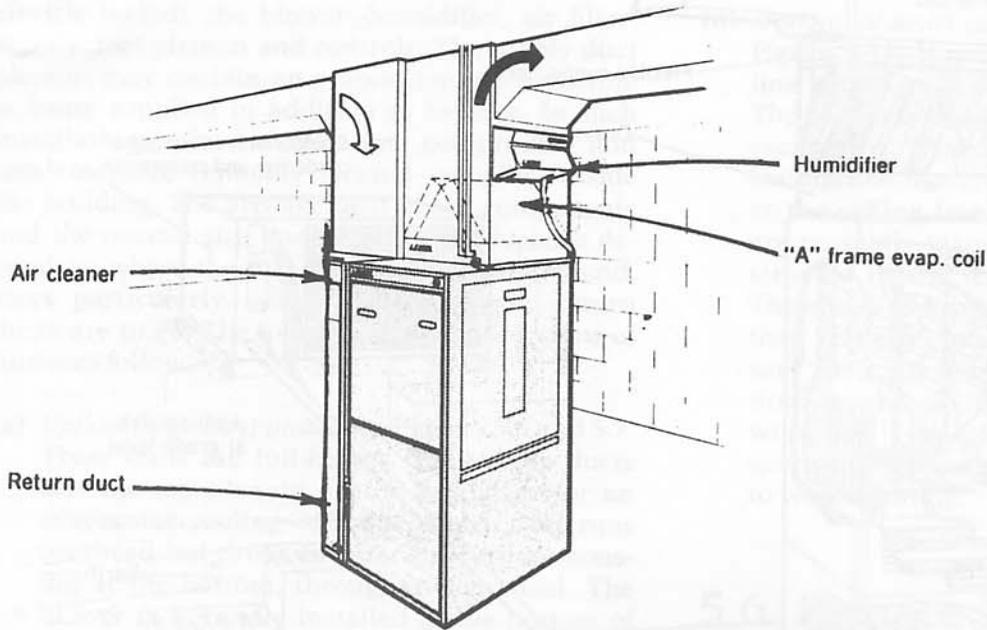
UPFLOW HIGH-BOY
WARM AIR FURNACE

(a)

BLOWER SIZE	MOTOR H.P.	BLOWER SPEED	EXTERNAL STATIC PRESSURE—INCHES WATER COLUMN													
			0.1		0.2		0.3		0.4		0.5		0.6		0.7	
			CFM	TEMP. RISE	CFM	TEMP. RISE	CFM	TEMP. RISE	CFM	TEMP. RISE	CFM	TEMP. RISE	CFM	TEMP. RISE	CFM	TEMP. RISE
11.8x8	1/2	HIGH	1609	32.4	1550	33.6	1491	35.0	1431	36.4	1365	38.2	1280	40.7	1202	43.4
		MED.	1294	40.3	1268	41.1	1231	42.4	1188	43.9	1131	46.1	1070	48.9	992	52.6
		LOW	1045	49.9	1021	51.1	995	52.4	960	54.3	924	56.5	875	59.6	820	63.6
11.8x8	1/2	HIGH	1609	40.1	1550	41.6	1491	43.3	1431	45.1	1365	47.3	1280	50.4	1202	53.7
		MED.	1294	49.9	1268	50.9	1231	52.4	1188	54.3	1131	57.1	1070	60.3	992	65.1
		LOW	1045	61.8	1021	63.2	995	64.9	960	67.2	924	69.8	875	73.8	820	78.7
11.8x10.6	3/4	HIGH	1642	52.4	1568	54.9	1505	57.2	1441	59.7	1359	63.3	1270	67.8	1170	78.5
		MED. HI	1621	53.1	1557	55.3	1487	57.9	1415	60.8	1357	63.4	1252	68.7	1152	74.7
		MED. LOW	1465	58.7	1399	61.5	1343	64.1	1285	67.0	1197	71.9	1122	76.7	1030	83.6
		LOW	1335	64.5	1295	66.4	1249	68.9	1207	71.3	1152	74.7	1084	79.4	1010	85.2

(b)

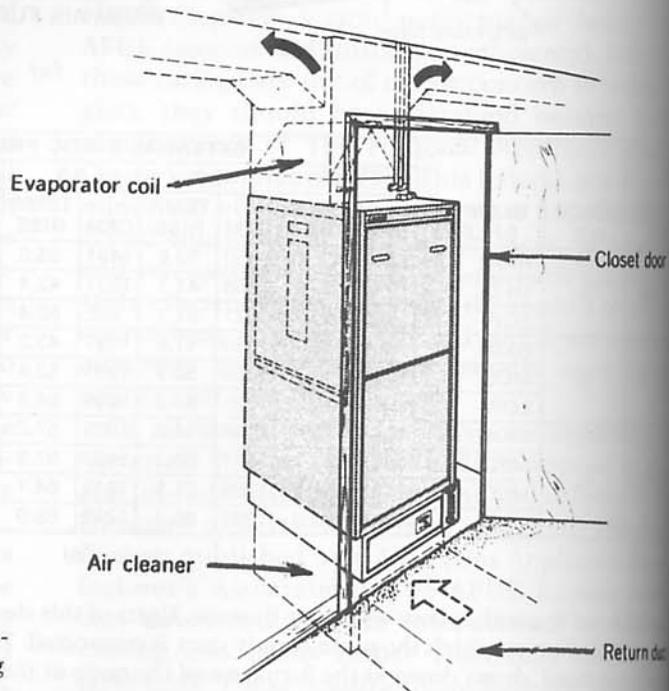
Figure 5.6 (a) Typical upflow warm air furnace. Units of this design force heated air into a top plenum to which the main supply duct is connected. The return duct, also usually overhead, drops down at the furnace and connects at the bottom of one side of the furnace enclosure. Flexible canvas connectors at the supply duct exit and the return duct entry reduce transmitted noise and vibration from the furnace. Upflow units can be installed in full-height basements, closets and utility rooms. Refrigerant piping, which connects the cooling coil (evaporator) to a remote refrigeration compressor and condenser, is shown for information. (Drawing reproduced with permission from ACCA Manual C, p. 17.) (b) Typical external static pressure data for a residential upflow furnace.



Basement installation

With cooling coil, electronic air cleaner and power humidifier.

(a)



Closet installation

With cooling coil and electronic air cleaner.

(b)

Figure 5.7 Typical upflow furnace installations. (a) Standard high-ceiling basement installation, with electronic air cleaner in lieu of a mechanical filter. Cooling coil piping is not shown for clarity. Humidifier is usually mounted on the supply duct. (b) Closet installation, with return air entering from below. The air cleaner is mounted at the junction of the return air duct and the base of the furnace. This type of installation is appropriate for a slab-on-grade or a low crawl space house.

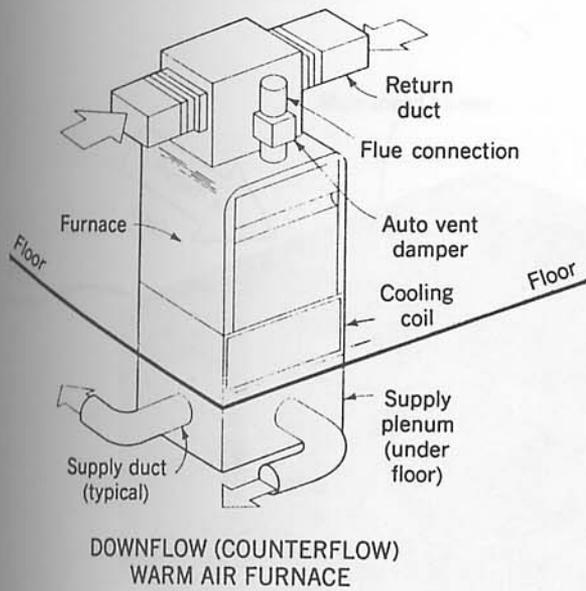


Figure 5.8 Typical downflow warm air furnace. These units are designed to serve floor-level perimeter heating (and cooling) outlets. These outlets are supplied by ducts connected to a subfloor plenum under the furnace. The furnace is also called counterflow because it supplies warm air downward into the plenum. This flow is opposite (counter) to the more common upflow design. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

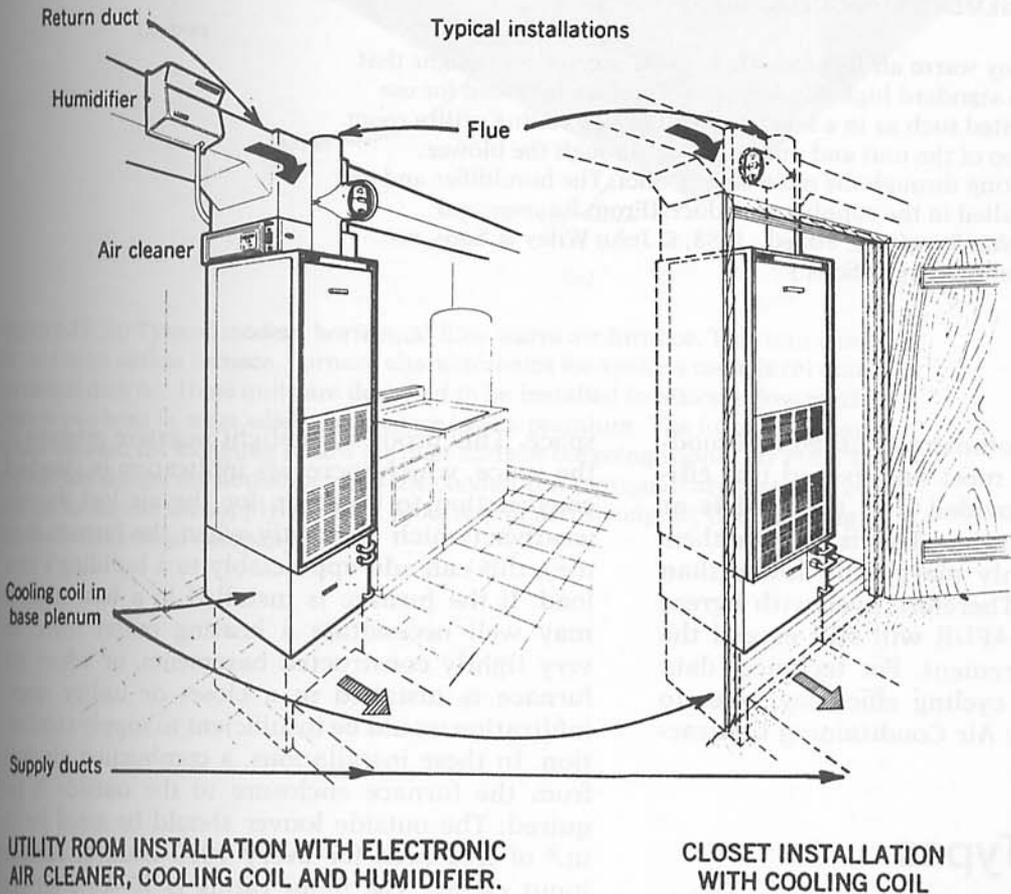


Figure 5.9 Typical installations of downflow units. These furnaces are specifically designed for slab-on-grade or low crawl space houses using perimeter heating (and cooling). The blower forces air down into an underfloor plenum from where it is distributed to perimeter outlets. An optional cooling coil can be installed in a base plenum. Return air enters overhead. Filters, air cleaners and humidifiers are mounted overhead as shown. (Built-in humidifiers are mounted inside the furnace enclosure, in the supply air path.)

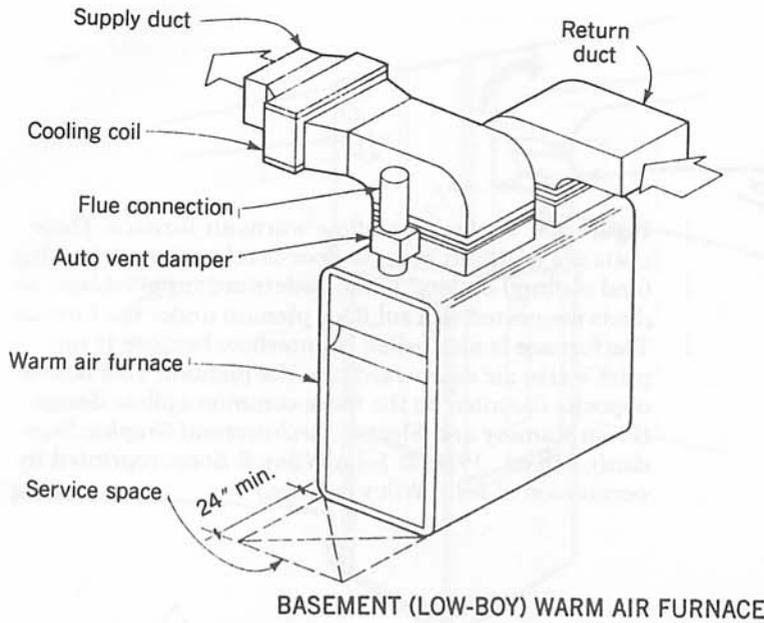


Figure 5.10 Typical low-boy warm air furnace. These units are upflow designs that are wider and shorter than standard high-boy furnaces. They are intended for use where ceiling height is limited such as in a low basement or low ceiling utility room. The return air enters the top of the unit and makes a loop through the blower, heater and filter before exiting through the main supply duct. The humidifier and optional cooling coil are installed in the supply trunk duct. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

(residential and small commercial usage). All modern warm air furnaces meet and exceed this efficiency requirement provided that the load is at least 25% of the unit's rating. This is because their cycling efficiency is only marginally lower than steady-state efficiency. Therefore, even with a great deal of on-off cycling, AFUE will still exceed the minimum AFUE requirement. For technical data on residential furnace cycling efficiency, refer to Manual S published by Air Conditioning Contractors of America (ACCA).

5.7 Furnace Types

There are three principal types of warm air furnaces being produced today: conventional, condensing and pulse.

a. Conventional Gas or Oil Furnaces

Older furnaces use an atmospheric heat exchanger that takes combustion air from the surrounding

space. This produces a slight negative pressure in the space, which increases infiltration in standard construction to make up for the air lost. In cold weather (which is exactly when the furnace is in use), this can add appreciably to a building's heat load. If the furnace is installed in a basement, it may well necessitate a heating outlet there. In very tightly constructed basements, or where the furnace is installed in a closet or utility space, infiltration would be insufficient to supply combustion. In these installations, a combustion air duct from the furnace enclosure to the outside is required. The outside louver should be sized for 1 in.² of free area for every 1000 Btuh of furnace input rating. The input rating is 10–25% greater than the output rating.

Older furnaces do not meet the 78% AFUE requirement; they run somewhere between 50 and 65% efficiency. Modern conventional gas and oil furnaces have a sealed heat exchanger, which takes combustion air from outside, usually using a forced draft or induced draft fan to do so. Many units also

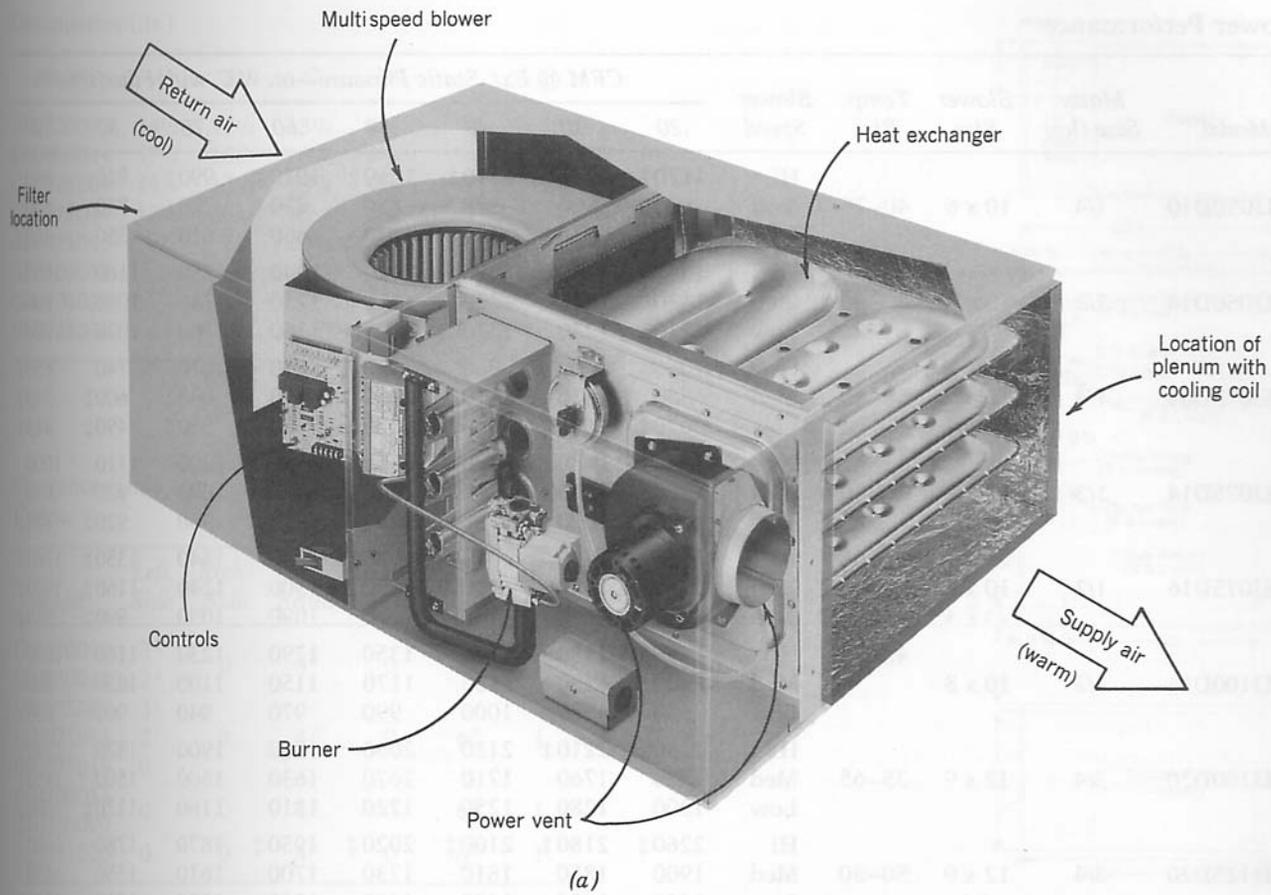


Figure 5.11 (a) Typical modern horizontal flow warm air furnace. This unit can also be used as an upflow furnace. Furnace characteristics for various models (b) and dimensional data (c). These units are designed to be installed in attics or low pitched roofs or overhead in areas where floor space is at a premium. The furnace is placed in-line between the incoming return air duct and the outgoing supply air duct. Like the low-boy design, the humidifier and the cooling coil (optional) are installed outside the furnace in the supply air duct. (Courtesy of Armstrong Air Conditioning, a Lennox International company.)

Blower Performance

Model	Motor Size (hp)	Blower Size	Temp. Rise	Blower Speed	CFM @ Ext. Static Pressure—in. W.C. with Filter(s)†							
					.20	.30	.40	.50	.60	.70	.80	.90
GHJ050D10	1/4	10 x 6	40–70	Hi	1170‡	1130‡	1110‡	1060‡	1020‡	990‡	910	840
				Med	900	900	870	850	830	790	740	650
				Low	720	710	690	670	650	610	550	460‡
GHJ050D14	1/3	10 x 8	20–50	Hi	1580	1540	1490	1420	1340	1250	1160	1050
				Med	1430	1420	1350	1310	1250	1780	1090	990
				Low	1210	1220	1220	1190	1160	1100	1020	920
GHJ075D09	1/4	10 x 6	50–80	Hi	1050	1000	960	930	870	810	740	650‡
				Med	850	830	810	760	720	680‡	600‡	540‡
				Low	670‡	660‡	650‡	620‡	580‡	550‡	490‡	410‡
GHJ075D14	1/3	10 x 8	40–70	Hi	1480‡	1430‡	1390	1350	1280	1200	1110	1000
				Med	1270	1270	1250	1210	1160	1090	920	830
				Low	1070	1090	1080	1060	1030	980	920‡	820‡
GHJ075D16	1/2	10 x 8	36–65	Hi	1870	1790	1710	1630	1540	1440	1350‡	1260‡
				Med	1500	1450	1410	1370	1300	1240	1160‡	1070‡
				Low	1170	1170	1160	1150	1090	1050	990‡	910‡
GHJ100D14	1/3	10 x 8	45–75	Hi	1450	1430	1370	1350	1290	1230	1160	1070
				Med	1200	1180	1180	1170	1150	1100	1030	960
				Low	980	980	1000	990	970	940	900	830
GHJ100D20	3/4	12 x 9	35–65	Hi	2290‡	2210‡	2130	2060	1980	1900	1820	1740
				Med	1820	1760	1710	1670	1630	1600	1500	1450
				Low	1300	1280	1250	1220	1210	1160	1120‡	1000‡
GHJ125D20	3/4	12 x 9	50–80	Hi	2260‡	2180‡	2100‡	2020‡	1950‡	1870	1780	1680
				Med	1900	1850	1810	1730	1700	1610	1550	1450
				Low	1440	1420	1390	1360	1320	1270	1210	1130‡

Notes: † .50 in. w.c. max. approved ext. static pressure. Airflow rated with AFILT524-1 filter kit.

‡ Not recommended for heating; Temperature rise may be outside acceptable range.

Physical and Electrical

Model	Input (Btu/h)	Output (Btu/h)	AFUE (ICS)	Nom. Cooling Cap.	Gas Inlet (in.)	Flue Size (in.)	Volts/Ph/hz	Min. Time Delay Breaker or Fuse	Nominal F.L.A.	Trans. (V.A.)	Appr. Weight (lbs.)
GHJ050D10	50,000	40,000	80.7	1.5–2.5	1/2	4	115/1/60	15	5.7	40	120
GHJ050D14	50,000	40,000	81.7	2.5–3.5	1/2	4	115/1/60	15	8.2	40	130
GHJ075D09	75,000	60,000	80.4	1.5–2.5	1/2	4	115/1/60	15	5.7	40	125
GHJ075D14	75,000	60,000	80.4	2.5–3.5	1/2	4	115/1/60	15	8.3	40	135
GHJ075D16	75,000	60,000	80.5	3.0–4.0	1/2	4	115/1/60	15	8.3	40	145
GHJ100D14	100,000	80,000	80.2	2.5–3.5	1/2	4	115/1/60	15	9.1	40	155
GHJ100D20	100,000	80,000	80.6	3.5–5.0	1/2	4	115/1/60	15	12.2	40	165
GHJ125D20	125,000	100,000	80.6	3.5–5.0	1/2	5*	115/1/60	15	12.2	40	170

* Connection to the combustion blower is 4 inch. Vent must be 5 in. for Cat. 1 installation

(b)

Figure 5.11 (Continued)

Dimensions (in.)

Model	A	B	C	D
GHJ050D10	14 ^{1/2}	13 ^{1/2}	13 ^{1/4}	47/8
GHJ050D14	17 ^{1/2}	16 ^{1/2}	16 ^{1/4}	6 ^{3/8}
GHJ075D09	14 ^{1/2}	13 ^{1/2}	13 ^{1/4}	47/8
GHJ075D14	17 ^{1/2}	16 ^{1/2}	16 ^{1/4}	6 ^{3/8}
GHJ075D16	22	21	20 ^{3/4}	8 ^{5/8}
GHJ100D14	22	21	20 ^{3/4}	8 ^{5/8}
GHJ100D20	22	21	20 ^{3/4}	8 ^{5/8}

Clearances (in.)

Upflow

Model	Clearance		Clearance			
	Left Side	Right Side	Front	Back	Vent	Top
GHJ050D10	3 ¹	0	4 ²	0	6 ³	1
GHJ050D14	2 ¹	0	4 ²	0	6 ³	1
GHJ075D09	3 ¹	0	4 ²	0	6 ³	1
GHJ075D14	2 ¹	0	4 ²	0	6 ³	1
GHJ075D16	0	0	4 ²	0	6 ³	1
GHJ100D14	0	0	4 ²	0	6 ³	1
GHJ100D20	0	0	4 ²	0	6 ³	1
GHJ125D20	0	0	4 ²	0	6 ³	1

¹0" if B1 vent is used

²2" if B1 vent is used

³1" if B1 vent is used

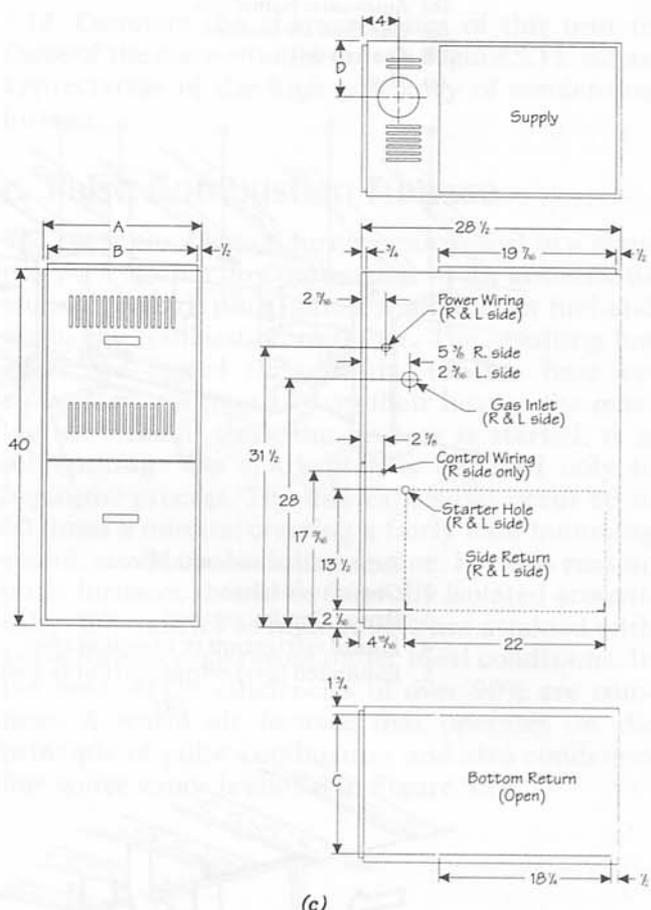
Horizontal

Model	Clearance		Air Flow							
	Left Side	Right Side	Front	Back	Vent	L to R		R to L		
						Top	Bottom	Top	Bottom	
GHJ050D10	1	1	18	0	6 ²	1	3 ¹	3 ³	0	
GHJ050D14							2 ¹	2 ³		
GHJ075D09							3 ¹	3 ³		
GHJ075D14	1	1	18	0	6 ²	1	2 ¹	2 ³	0	
GHJ075D16							0	1		
GHJ100D14	1	1	18	0	6 ²	1	0	1	0	
GHJ100D20										
GHJ125D20	1	1	18	0	6 ²	1	0	1	0	

¹0" if B1 vent is used

²1" if B1 vent is used

³1" if B1 vent is used

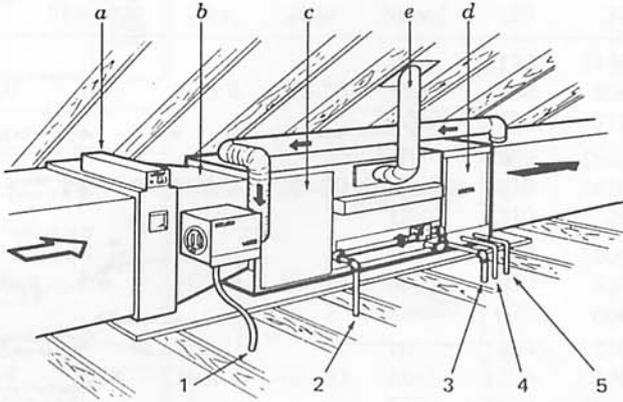


(c)

Figure 5.11 (Continued)

TYPICAL INSTALLATIONS

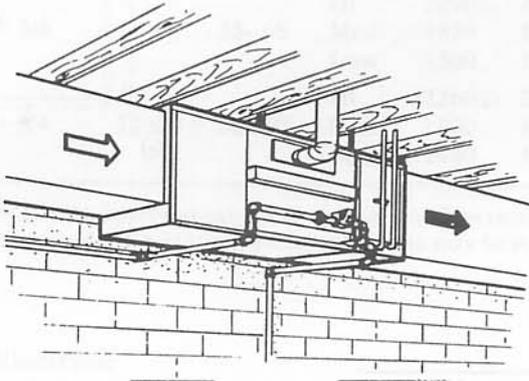
- (a) Electronic air cleaner
- (b) Automatic humidifier
- (c) Gas-fired air furnace
- (d) Cooling coil
- (e) Flue



Attic installation
with cooling coil,
electronic air
cleaner and automatic humidifier

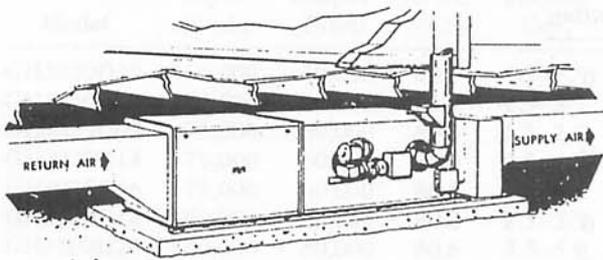
1. Tap water *in* to humidifier.
2. Gas *in* to furnace.
3. Drain. Condensate *out*.
4. Liquid refrigerant *in* to evaporator.
5. Expanded (gas) refrigerant *out* to condenser.

(d)



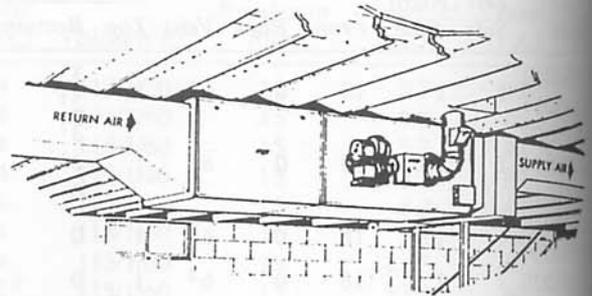
Basement installation
with cooling coil

(e)



Crawl Space Horizontal Installation

(f)



Suspended Horizontal Installation

(g)

Figure 5.11 (d–g) Typical installations of horizontal-type warm air furnaces in an attic (d), suspended in a basement (e, g), and in a shallow crawl space (f). Units (d) and (e) contain cooling coils mounted in the supply trunk duct. Units attached to, or resting on, wood fram-

ing as in (d), (e) and (g), must be mounted on vibration isolators. Crawl space units (f) are installed on concrete pads below the house and, therefore, require less vibration isolation.

have automatic vent dampers or power venting to reduce losses. These modern units have AFUE efficiencies of 80–84%. They are referred to as mid-efficiency units. One such unit and its characteristics is shown in Figure 5.11(a–c).

b. Condensing Furnaces

Condensing furnaces achieve AFUE efficiencies above 90% by recovering heat that normally goes up the chimney as hot flue gas at 400–500°F. Most of this heat is carried in superheated water vapor (steam). By channeling the flue gas through a secondary heat exchanger inside the furnace, it is possible to reduce its temperature to about 150°F. In so doing, the superheated water vapor (steam) condenses into water and gives up 1000 Btu/lb of water. This warm water condensate is then drained away into the sewer system, while the heat recovered is added to the furnace output. Another advantage of this design is that it eliminates the need for an expensive masonry chimney. All that is required is a small diameter (2–3 in.) plastic pipe to exhaust the few remaining flue gases. A typical unit of this design and its characteristics are shown in Figure

5.12. Compare the characteristics of this unit to those of the conventional unit in Figure 5.11, for an appreciation of the high efficiency of condensing furnace.

c. Pulse Combustion Furnace

The pulse combustion furnace burns fuel in a manner very similar to combustion in an automobile engine. A spark plug ignites a mixture of fuel and air in the combustion chamber. The resulting hot gases are forced through the furnace heat exchanger where they give up their heat to the moving air stream. Once the process is started, it is self-igniting. The spark plug is required only to begin the process. The miniexplosions occur 60 to 80 times a minute, creating a fairly loud humming sound, similar to an idling engine. For this reason, pulse furnaces should be carefully isolated acoustically. Efficiencies as high as 96% are attained with pulse furnaces operating under ideal conditions. In the field, AFUE efficiencies of over 90% are common. A warm air furnace that operates on the principle of pulse combustion and also condenses flue water vapor is shown in Figure 5.13.

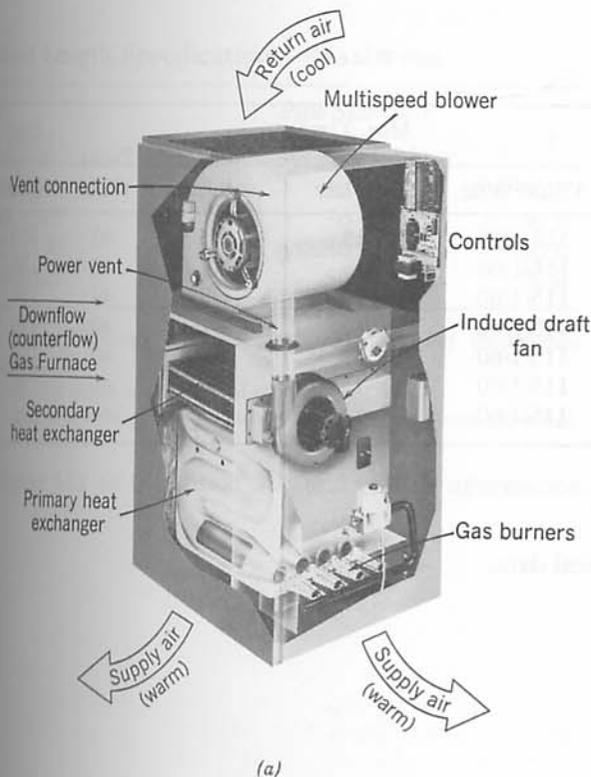


Figure 5.12 (a) Modern condensing-type downflow (counterflow) warm air furnace. (See Figures 5.8 and 5.9.) Note the use of a secondary heat exchanger that functions to remove most of the heat from the furnace flue gas. The vent connection is a plastic pipe that vents to outside air. (Courtesy of Armstrong Air Conditioning, a Lennox International company.)

Blower Specifications

Model	Motor Size (hp)	Blower Size	Temp. Rise	Blower Speed	CFM @ Ext. Static Pressure—in. W.C. with Filter(s) ¹							
					.20	.30	.40	.50	.60	.70	.80	.90
GCK050D10	1/4	10 x 6	40–70	Hi	1080 ²	1030	1010	960	900	840	770	680
				Med	870	840	820	790	740	700	640	550
				Low	680	670	650	620	590 ²	560 ²	490 ²	430 ²
GCK050D12	1/3	10 x 8	30–60	Hi	1350	1310	1230	1180	1100	1030	940	850
				Med	1260	1230	1170	1120	1050	980	910	830
				Low	1150	1110	1080	1040	990	920	850	830
GCK075D14	1/2	10 x 9	40–70	Hi	1700 ²	1640 ²	1560	1490	1410	1330	1240	1150
				Med	1550	1470	1410	1340	1280	1200	1130	1030
				Low	1330	1280	1240	1190	1120	1070	1010	910
GCK075D20	3/4	12 x 9	40–70	Hi ²	2240 ²	2160 ²	2070 ²	1980 ²	1890 ²	1800 ²	1700 ²	1610 ²
				Med	1910 ²	1830 ²	1780 ²	1700 ²	1630 ²	1550	1470	1390
				Low	1420	1380	1360	1310	1270	1220	1160	1100
GCK100D14	1/2	10 x 9	55–85	Hi	1670 ²	1580 ²	1500	1420	1340	1230	1110	1000
				Med	1540	1450	1380	1310	1220	1130	1020	890
				Low	1360	1300	1240	1180	1100	1010	910 ²	780 ²
GCK100D20	3/4	12 x 9	45–75	Hi	2260 ²	2160 ²	2090 ²	1990 ²	1910 ²	1810	1700	1590
				Med	1740	1670	1640	1580	1550	1460	1390	1310
				Low	1400	1380	1350	1310	1270	1230	1180	1110
GCK125D20	3/4	12 x 9	45–75	Hi	2190	2130	2030	1960	1850	1790	1690	1600
				Med	1910	1850	1770	1730	1640	1590	1520	1420
				Low	1500	1460	1420	1400	1360 ²	1310 ²	1250 ²	1180 ²

¹.50 in. w.c. max. approved ext. static pressure. Airflow rated with AFILT525-1 filter kit.

²Not recommended for heating; Temperature rise may be outside acceptable range.

Physical and Electrical

Model	Input (Btuh)	Output (Btuh)	AFUE (ICS)	Nom. Cooling Cap.	Gas Inlet (in.)	Volts/Ph/hz	Min. Time Delay Breaker or Fuse	Nominal F.L.A.	Trans. (V.A.)	Appr. Weight (lbs.)
GCK050D10	50,000	45,000	90.0	1.5–2.5	1/2	115/1/60	15	5.5	40	150
GCK050D12	50,000	45,000	90.0	2.5–3.0	1/2	115/1/60	15	7.5	40	150
GCK075D14	75,000	67,500	90.0	2.5–3.5	1/2	115/1/60	15	9.4	40	180
GCK075D20	75,000	67,500	90.0	3.5–5.0	1/2	115/1/60	15	12.1	40	190
GCK100D14	75,000	90,000	90.0	2.5–3.5	1/2	115/1/60	15	9.2	40	190
GCK100D20	100,000	90,000	90.0	3.5–5.0	1/2	115/1/60	15	12.0	40	195
GCK125D20	125,000	112,500	90.0	3.5–5.0	1/2	115/1/60	15	12.0	40	215

(b)

Figure 5.12 (b) Blower specifications and physical and electrical data.

Dimensions (in.)

Model	A	B	C	D
GCK050D10				
GCK050D12	17 1/2	16 1/2	16	3 5/8
GCK075D14	22	21	21 1/2	3 3/8
GCK075D20	26 1/2	25 1/2	25	5 5/8
GCK100D14	22	21	20 1/2	3 3/8
GCK100D20	26 1/2	25 1/2	25	5 5/8
GCK150D20	26 1/2	25 1/2	25	3 1/8

Clearances (in.)

Model	Top	Side	Front	Back	Vent
GCK050D10					
GCK050D12	1	0	2	0	0
GCK075D14					
GCK075D20	1	0	2	0	0
GCK100D14					
GCK100D20	1	0	2	0	0
GCK150D20	1	0	2	0	0

Vent Length Specifications—Maximum

Model	Pipe Size (in.)		
	2	2 1/2	3
GCK050	50 ft.	50 ft.	50 ft.
GCK075	50 ft.	50 ft.	50 ft.
GCK100	50 ft.	50 ft.	50 ft.
GCK125	N/A	50 ft.	50 ft.

Notes: Allowance for vent terminal included in lengths shown.

One 90° elbow equals 5 ft. of pipe.

Min. length 5 ft. and 1 elbow not including the vent terminal.

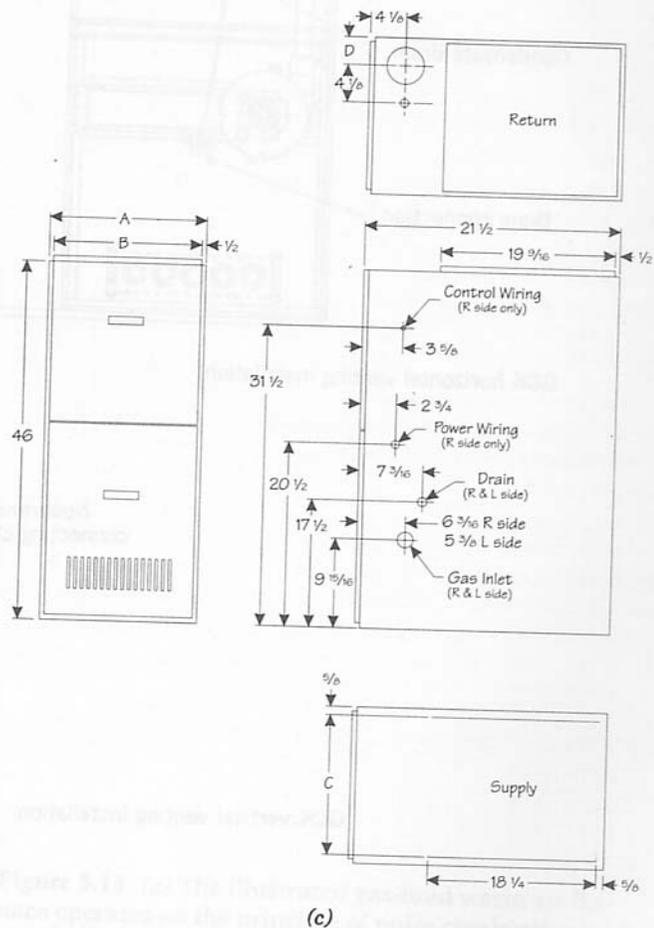
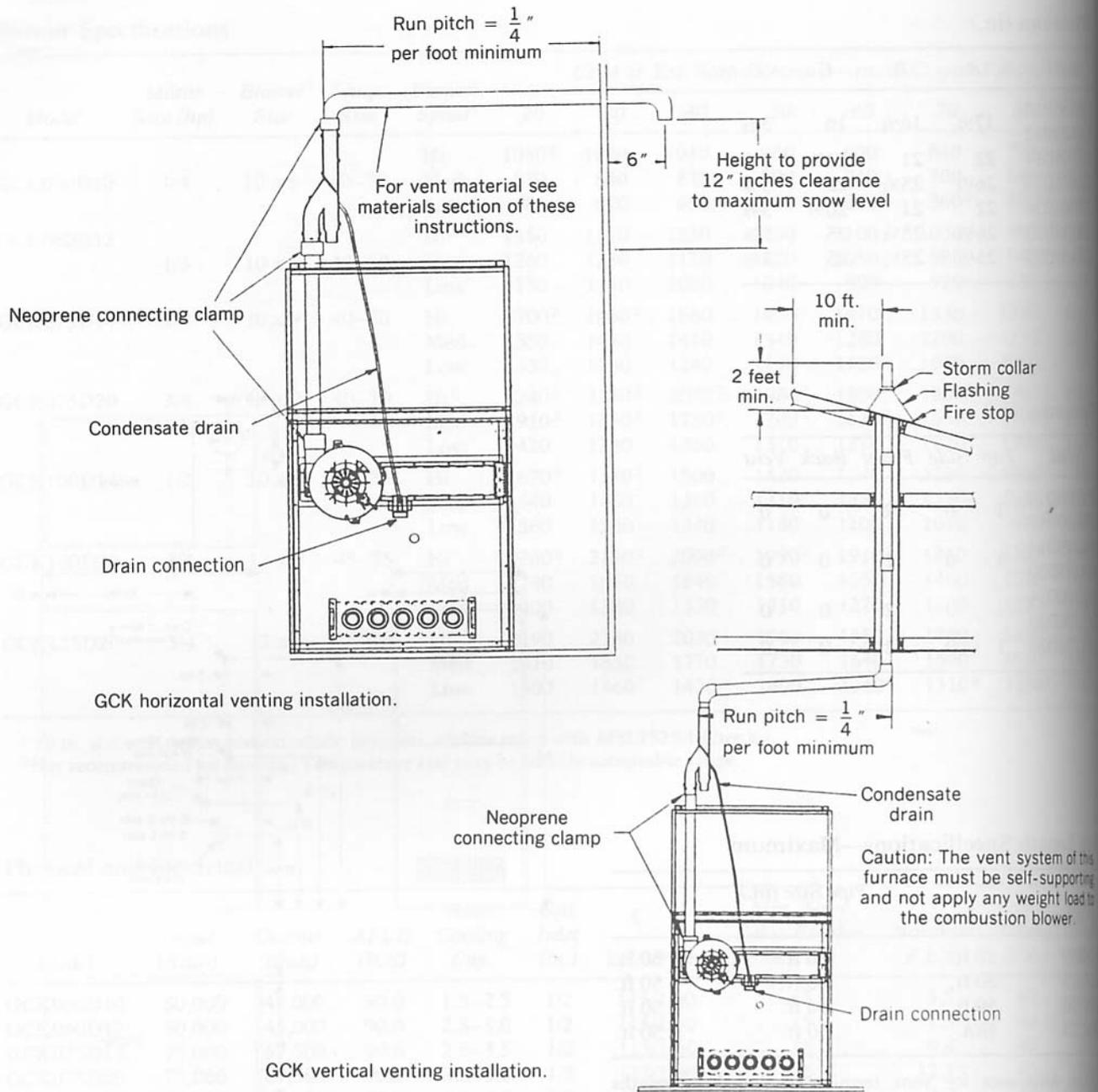
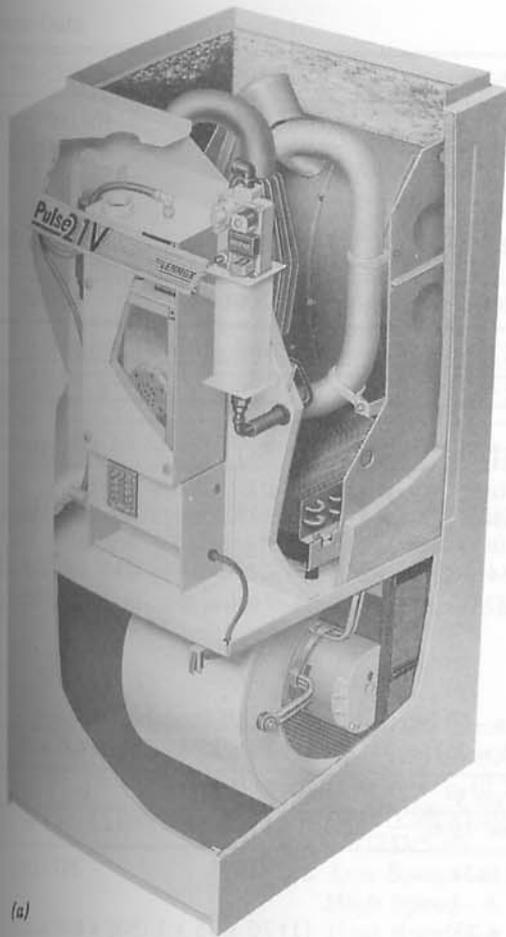


Figure 5.12 (c) Dimension data and venting information.



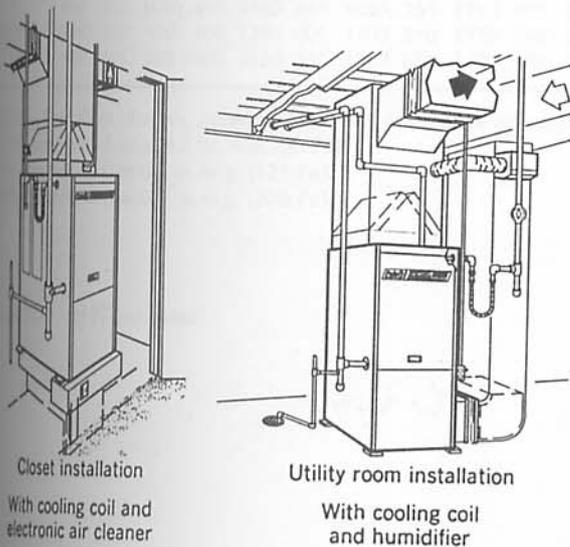
(d)

Figure 5.12 (d) Venting and condensate drain arrangements.



(a)

Typical applications



Closet installation
With cooling coil and
electronic air cleaner

Utility room installation
With cooling coil
and humidifier

(b)

Figure 5.13 (a) The illustrated gas-fired warm air furnace operates on the principle of pulse combustion and also condenses the water in the flue gas. As a result, no flue or chimney is required. Combustion air to the sealed combustion unit is drawn in through the same PVC pipe that exhausts the remaining cool flue gases. Burner control is automatic, producing heat output in proportion to the air delivered by the blower. The blower is electronically speed controlled to maintain a specified air flow (cfm) throughout the entire static pressure range. These units are rated 60,000–100,000 Btuh input, with a maximum external static pressure of 0.80 in. w.g., including filter resistance. (b) Typical installations. (Photo and data courtesy of Lennox Industries.)

Specifications

Model No.		G21V3-60	G21V3-80	G21V5-80	G21V5-100
Input—Btuh (kW)		60,000 (17.6)	80,000 (23.4)	80,000 (23.4)	100,000 (29.3)
Output—Btuh (kW)		57,000 (16.7)	76,000 (22.3)	75,000 (22.0)	95,000 (27.8)
* A.F.U.E.		94.3%	94.5%	93.4%	94.5%
California Seasonal Efficiency		92.5%	92.4%	90.9%	91.5%
Temperature rise range—°F(°C)		40–70 (22–39)	45–75 (25–41)	35–65 (19–36)	40–70 (22–39)
High static certified by A.G.A./C.G.A.—in wg. (Pa)		.80 (200)	.80 (200)	.80 (200)	.80 (200)
Gas Piping Size I.P.S.—in. (mm)	Natural	1/2 (13)	1/2 (13)	1/2 (13)	1/2 (13)
	** LPG/Propane	1/2 (13)	1/2 (13)	1/2 (13)	1/2 (13)
Vent/Intake air pipe size connection —in. (mm)		2 (51)	2 (51)	2 (51)	2 (51)
Condensate drain connection—in. (mm) SDR11		1/2 (13)	1/2 (13)	1/2 (13)	1/2 (13)
Blower wheel nom. diameter x width —in. (mm)		10 x 8 (254 x 203)	10 x 8 (254 x 203)	11 1/2 x 9 (279 x 229)	11 1/2 x 9 (279 x 229)
Blower motor hp (W)		1/2 (373)	1/2 (373)	1 (746)	1 (746)
Blower motor minimum circuit ampacity			12.0		17.4
Maximum fuse or circuit breaker size (amps)			15.0		25
Electrical characteristics		120 volts—60 hertz—1 phase (All models)			
Number and size of filters—in. (mm)		(1) 16 x 25 x 1 (406 x 635 x 25)		(1) 20 x 25 x 1 (508 x 635 x 25)	
Nominal cooling that can be added	Tons	1 1/2, 2, 2 1/2 or 3	2, 2 1/2 or 3	3 1/2, 4 or 5	3 1/2, 4 or 5
	kW	5.3, 7.0, 8.8 or 10.6	7.0, 8.8 or 10.6	12.3, 14.1 or 17.6	12.3, 14.1 or 17.6
Shipping weight—lbs. (kg) 1 package		250 (113)	250 (113)	297 (135)	297 (135)
External Filter Cabinet (furnished) *Filter size—in. (mm)		(1) 16 x 25 x 1 (406 x 635 x 25)		(1) 20 x 25 x 1 (508 x 635 x 25)	
** LPG/Propane kit		LB-65810B (46J46)	LB-65810B (46J46)	LB-65810B (46J46)	LB-65810C (46J46)

* Filter is not furnished with cabinet. Filter cabinet utilizes existing filter supplied with G21V unit.

* Annual Fuel Utilization Efficiency. Isolated combustion system rating for non-weatherized furnaces.

** LPG/Propane kit must be ordered extra for field changeover.

(c)

Figure 5.13 (c) Specification and blower data.

Blower Data

G21V3-60-80 BLOWER PERFORMANCE
 0 through 0.80 in. w.g. (0 Through 200 Pa) External Static Pressure Range

VSP2-1 Blower Control—Factory Settings

G21V3-60	G21V3-80
Low Speed—1	Low Speed—1
High Speed—4	High Speed—4
Heat Speed—1	Heat Speed—2

BDC2 Jumper Speed Positions

"ADJUST" Jumper Setting	"LOW" Speed (Cool Or Continuous Fan)								"HIGH" Speed (Cool)								"HEAT" Speed							
	1		2		3		4		1		2		3		4		1		2		3		4	
	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s
+	540	225	700	330	830	390	1000	470	1150	545	1260	595	1400	660	1410	665	1150	545	1250	590	1350	635	1420	670
NORM	490	230	630	295	740	350	880	415	1040	490	1140	540	1240	585	1265	595	1030	485	1140	540	1220	575	1300	615
—	440	210	560	265	670	315	800	380	940	445	1030	485	1140	540	1160	545	920	435	1020	480	1100	520	1190	560

NOTE—The effect of static pressure and filter resistance is included in the air volumes listed.

G21V5-80-100 BLOWER PERFORMANCE
 0 through 0.80 in. w.g. (0 Through 200 Pa) External Static Pressure Range

VSP2-1 Blower Control—Factory Settings

G21V5-80	G21V5-100
Low Speed—1	Low Speed—1
High Speed—4	High Speed—4
Heat Speed—1	Heat Speed—2

BDC2 Jumper Speed Positions

"ADJUST" Jumper Setting	"LOW" Speed (Cool Or Continuous Fan)								"HIGH" Speed (Cool)								"HEAT" Speed							
	1		2		3		4		1		2		3		4		1		2		3		4	
	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s	cfm	L/s
+	800	380	1050	495	1410	665	1620	765	1710	805	2030	960	*2270	*1070	*2270	*1070	1900	895	2140	1010	*2270	*1070	*2270	*1070
NORM	720	340	950	450	1280	605	1500	710	1570	740	1850	875	2100	990	2220	1050	1700	800	1940	915	2080	980	2200	990
—	620	295	850	400	1120	530	1310	620	1420	670	1650	780	1860	880	1990	940	1520	715	1730	815	1860	880	1940	990

NOTE—The effect of static pressure and filter resistance is included in the air volumes listed.

*2300 cfm (1085 L/s) at 0.2 in. w.g. (50 Pa).

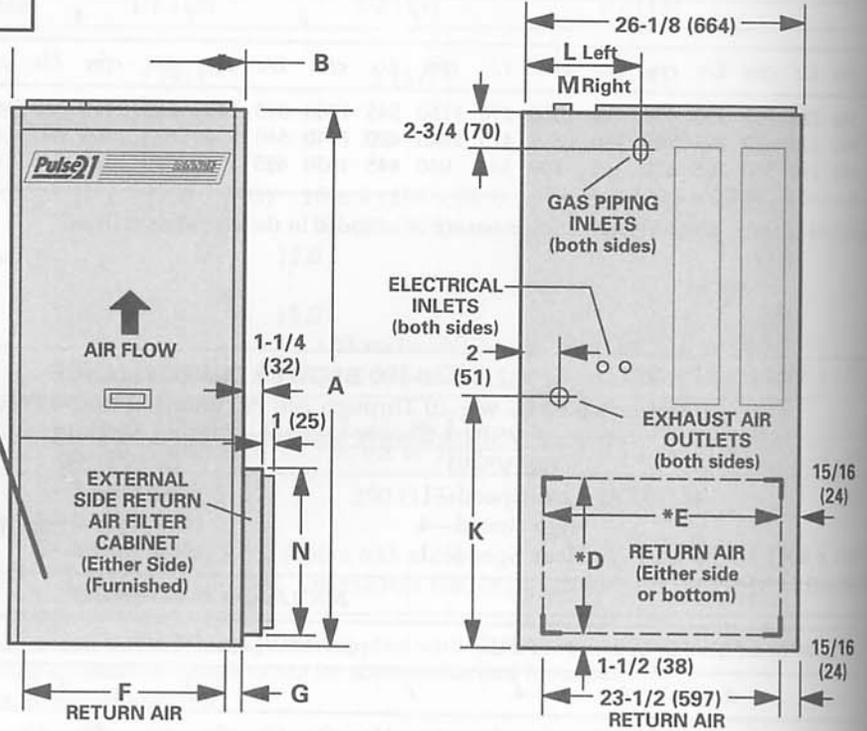
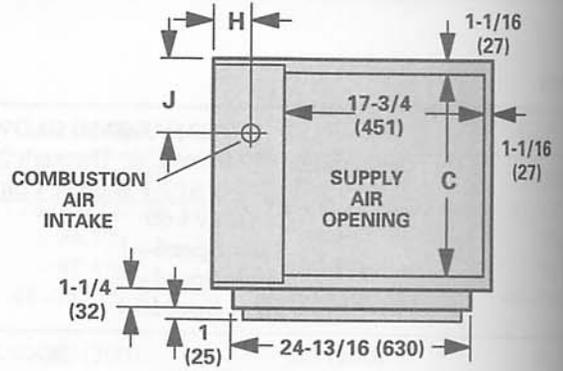
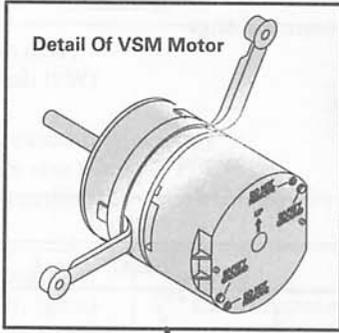
2100 cfm (990 L/s) at 0.5 in. w.g. (125 Pa).

2000 cfm (990 L/s) at 0.8 in. w.g. (200 Pa).

(c)

Figure 5.13 (c) (Continued)

ELECTRONICALLY VARIABLE SPEED (VSM) MOTOR



*Unit or External Side Return Air Filter Cabinet

Model No.		A	B	C	D	E	F	G	H	J	K	L	M	N
G21V3-60 G21V3-80	in.	49	21-1/4	19-1/8	14-1/2	18-1/2	14-1/2	3-3/8	4-1/2	8-1/2	20-1/4	7-1/4	5-1/4	16
	mm	1245	540	486	368	470	368	86	114	216	514	184	133	406
G21V5-80 G21V5-100	in.	53	26-1/4	24-1/8	18-1/2	23-1/2	18-1/2	3-7/8	2-1/2	11	24-1/4	4-5/8	4-5/8	20
	mm	1346	667	613	470	597	470	98	64	279	616	117	117	508

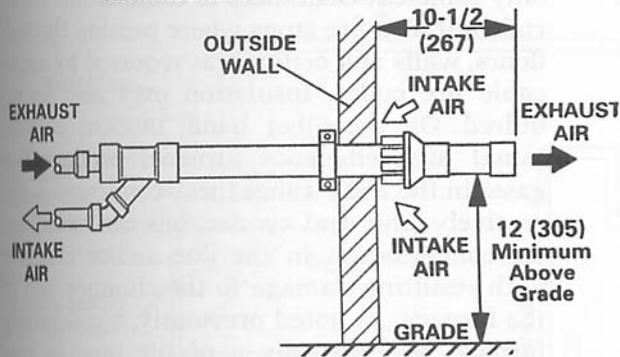
INSTALLATION CLEARANCES — ALL MODELS

Sides	0 inch (0 mm)
Rear	0 inch (0 mm)
Top	1 inch (25 mm)
Front	0 inch (0 mm)
Front (service)	36 inches (914 mm)
Floor	Combustible
Exhaust Pipe	0 inches (0 mm)
Exhaust Pipe Side	6 inches (152mm) (service only)

(d)

Figure 5.13 (d) Physical dimensions and clearances.

CONCENTRIC WALL TERMINATION APPLICATIONS



CONCENTRIC ROOF TERMINATION APPLICATIONS

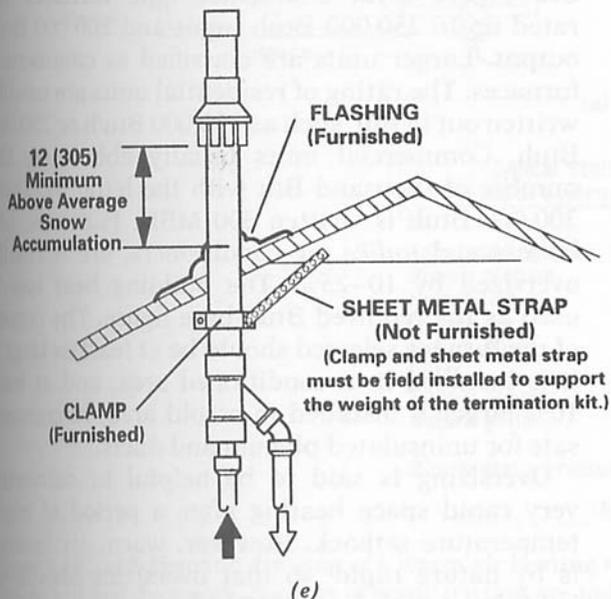


Figure 5.13 (e) Flue termination details.

5.8 Warm Air Furnace Characteristics

The data required by a system designer are available in manufacturers' published material. Typical dimensional and technical data are given in Figures 5.11–5.13. Of particular importance to the engineering technologist are the furnace output rating in Btuh and air delivery information. The latter gives the cfm of air delivered and the corresponding air temperature rise as a function of static pressure, for the various blower speeds.

a. Dimensional Data

The data required here are the furnace length and width ("footprint"), overall height including air plenum if any, service accessibility clearance, required clearances to combustible materials, clearance for filter replacement and venting data. Each of these items must be considered individually after a preliminary duct layout has been made, as we will explain.

- (1) Overall dimensions of the furnace and the required clearances control its placement in the building. Basements in residences usually have low ceilings that may not permit the use of a full-size high-boy unit with a cooling coil in the plenum. Remember also to include the required overhead clearance to combustible materials. Removable filters are often placed in the return air duct connection. This not only requires a full filter length of clearance, for filter removal, but also sets off the return duct, making the assembly of return duct, filter assembly and furnace very wide.
- (2) The minimum dimensions of the stripped furnace unit must be obtained from the manufacturer to ensure accessibility to the proposed location. This is particularly important with replacement units and with furnaces intended for installation in attics, closets, utility rooms and other tight, confined spaces with limited access.
- (3) A vent connection to the furnace flue collar is required for every furnace, to exhaust flue gases. The vent pipe size, maximum length and required clearances depend on the type of furnace used. Venting tables are published by GAMA. Specific venting information for each furnace is provided by the manufacturer. Remember that flue gases can be as hot as 600°F

with conventional furnaces. This requires not only sufficient clearances to combustible materials but also fire stops where passing through floors, walls and ceilings, as required by applicable fire codes. Insulation may also be required. On the other hand, modern conventional high efficiency furnaces produce flue gases in the 250°F range (near-condensing). Excessively long vent connections can cause water condensation in the flue and/or chimney with resulting damage to the chimney and to the furnace. As noted previously, a condensing furnace requires only a plastic pipe to vent cool flue gas and a drain connection to remove condensate. See Figure 5.12(d).

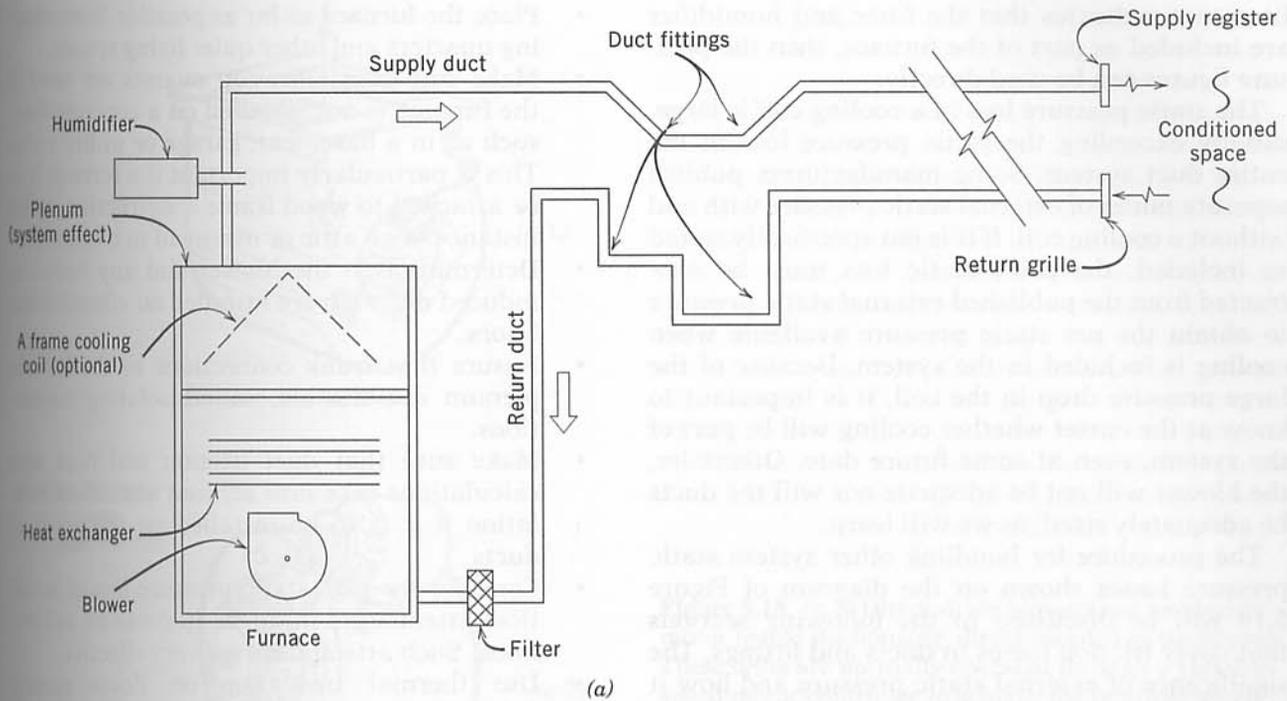
b. Output Rating

See Figure 5.12. Residential-type furnaces are rated up to 250,000 Btuh input and 200,000 Btuh output. Larger units are classified as commercial furnaces. The rating of residential units are usually written out in full, such as 250,000 Btu/h or 250,000 Btuh. Commercial units usually abbreviate the number of thousand Btu with the letter *M*. Thus 300,000 Btuh is written 300 MBH. Furnaces, like boilers and *unlike* air conditioners, are normally oversized by 10–25%. The building heat loss is used as the required Btuh base figure. The output of the furnace selected should be at least as large as it is installed in a conditioned area, and at least 10% larger if installed in a cold area, to compensate for uninsulated plenum and ducts.

Oversizing is said to be helpful in delivering very rapid space heating after a period of night temperature setback. However, warm air heating is by nature rapid, so that oversizing should be held to a minimum. Furthermore, oversizing causes frequent heating system cycling. This results in unpleasant temperature swings because air has almost negligible heat retention properties (specific heat). Therefore, as soon as the blower stops, the air temperature in rooms and ducts begins to drop rapidly. This effect can be overcome to an extent by continuous blower operation, combined with burner cycling.

c. Air Delivery Data

Refer to the table of air delivery information in Figure 5.6. Refer also to Figure 5.14, which shows the location of static pressure losses in a typical air system. Figure 5.14 is a schematic drawing showing essentially the same elements as Figure 5.1.



Typical Static Pressure
Loss of System Components

Item	Loss, in. W.G.
Evap. cooling coil	0.2 - 0.3
Supply plenum	0.0 - 0.05
Humidifier	0.0 - 0.02
Supply duct	0.1 - 0.2
Return duct	0.05 - 0.15
Filter	0.1 - 0.2
Supply register	0.01 - 0.07
Return grille	0.01 - 0.07
Blower static pressure	0.1 - 0.8

(b)

Figure 5.14 (a) Schematic drawing of a warm air heating system with (optional) cooling coil and ductwork. Sources of static friction are indicated. (b) List of typical static pressure losses of system components.

you consult the air delivery table in Figure 5.6, you will notice that the air delivery columns are listed under various values of external static pressure. These figures refer to the static pressure available external to the basic furnace. Since different manufacturers supply different items as part of the furnace, it is essential for the designing technologist to know, in advance, which items are included and which are not.

Specifically, the most common items that may or may not be included are the filter and the humidifier. Refer now to Figure 5.14. Mechanical fil-

ters installed in the return duct line are not included as part of the furnace, whereas electrostatic filters usually are. Similarly, plate-type humidifiers installed in the supply duct are not included in the basic furnace, whereas atomizing types frequently are. This means that in an installation such as shown in Figure 5.14, the designer must subtract the static pressure loss of the filter (which is considerable) and the static pressure loss in the humidifier, from the external static pressure given in the table, to obtain the pressure available at the plenum. On the other hand, if the manufacturer's

literature indicates that the filter and humidifier are included as part of the furnace, then the pressure figures can be used directly.

The static pressure loss in a cooling coil is large, usually exceeding the static pressure loss in the entire duct system. Some manufacturers publish separate tables of external static pressure with and without a cooling coil. If it is not specifically stated as included, the coil's static loss must be subtracted from the published external static pressure to obtain the net static pressure available when cooling is included in the system. Because of the large pressure drop in the coil, it is important to know at the outset whether cooling will be part of the system, even at some future date. Otherwise, the blower will not be adequate nor will the ducts be adequately sized, as we will learn.

The procedure for handling other system static pressure losses shown on the diagram of Figure 5.14 will be discussed in the following sections that cover friction losses in ducts and fittings. The significance of external static pressure and how it is used will become clear later on. For the moment, however, the technologist must keep in mind the clarifications necessary with regard to the humidifier and filter.

Note further from the table in Figure 5.6 that, as the cfm rises, the temperature rise drops. This is entirely logical. The air is heated as it passes through the heat exchanger. Since the combustion rate is fixed, the more air that passes per minute, the less it is heated, and vice versa. The use of these figures will also become clear later on, in our system design discussion.

5.9 Noise Considerations

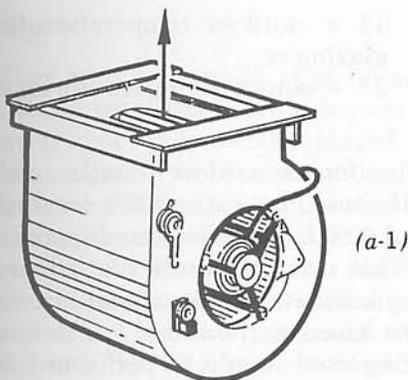
All mechanical equipment generates noise. Ducts are excellent carriers of sound. Since they interconnect all the rooms in a building, they comprise a very effective but generally undesirable noise-conducting system. Not only will combustion and blower noise be heard in every room, but sound will also carry from room to room. Treatment of these acoustical problems is not usually the responsibility of the engineering technologist. We will, therefore, mention only a few of the considerations that are of interest and importance to technologists.

The best way to avoid noise problems is not to generate the noise in the first place. As applied to warm air furnaces this means that designers should

- Place the furnace as far as possible from sleeping quarters and other quiet living spaces.
- Make sure that vibration mounts are used, if the furnace is not installed on a concrete floor such as in a basement, garage or utility room. This is particularly important if a furnace is to be attached to wood frame construction, as for instance in an attic or overhead in a basement.
- Determine that the blower and any forced or induced draft fan are installed on vibration isolators.
- Ensure that trunk connections to the furnace plenum are flexible, sound-isolating connections.
- Make sure that duct friction and duct area calculations take into account acoustical insulation if it is to be installed on the inside of ducts.
- Consider the high static pressure loss of acoustical attenuators in ducts, in pressure calculations. Such attenuators are very effective.
- Use thermal insulation on ducts passing through nonconditioned spaces because it can be very effective in reducing vibration of metal ducts, particularly those with large dimensions. This reduction is most effective when the thermal insulation is glued to the metal duct surface.

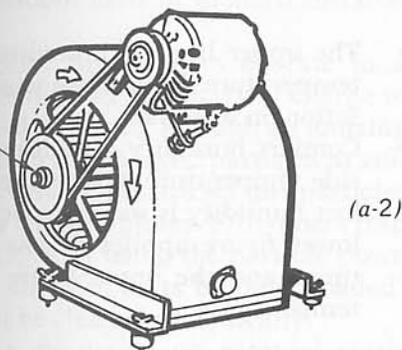
5.10 Warm Air Furnace Blowers

The terms *fan* and *blower* are used interchangeably when referring to the furnace fan. In most furnaces the unit is actually a centrifugal blower of the type shown in Figure 5.15a. The function of the blower is to circulate the air in the duct system. Most blowers are belt driven from a multisheave pulley. This permits changing the belt position on the drives, and thereby the blower speed. In some furnaces, the blower is directly coupled to the motor. There, a speed-controlled motor is used to permit changing the blower speed. As demonstrated previously, speed control of the blower is necessary when changing over from heating to cooling because of the different air quantities required for the two services. A speed change motor also be required in the initial system balancing after installation. Typical furnace blower curves are shown in Figure 5.15b. The system friction curve is simply a graph of the ductwork static friction at different values of air flow in the system.



Direct-drive blower

Multiple diameter sheaves
(for speed control)



Belt-driven blower

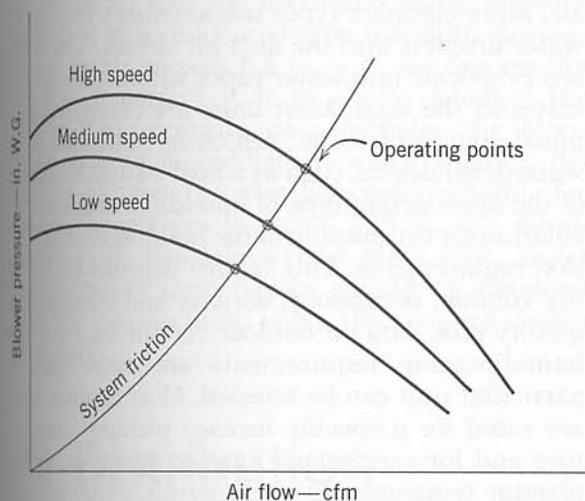


Figure 5.15 (a-1) Direct-drive blowers use an electric motor inside the housing, direct coupled to the blower. These units are normally operated through a variable-speed motor controller to achieve the required speed control. **(a-2)** Speed control of belt-driven blowers is usually accomplished by use of multiple sheaves on the motor and blower. A speed change requires physically moving the drive belt from one sheave to another. (Reproduced with permission from *ACCA Manual C*, p. 24.)

Figure 5.15 (b) Typical static pressure/airflow characteristics of a multispeed centrifugal blower. This is the type normally used in residential-type warm air furnaces. The operating points of the duct system are noted at the intersection of the blower curves and the system friction curve.

The intersection of the system curve with the blower curves shows the operating points possible at different blower speeds.

Residential-type furnaces are usually supplied as package units with a blower design that the furnace manufacturer has found to be satisfactory for most applications. The only leeway the designer has with respect to the blower is speed control. In contrast, commercial furnaces can be equipped with blowers that meet the specifications of the system designer. For our purposes—residential and small commercial installations—the package-type furnace and blower will be used. The manufacturer usually presents blower data in tabular form as shown in Figure 5.13. Some manufacturers prefer to present the data in graphic form, similar to that shown in Figure 5.15(b) (without the system curve, of course).

5.11 Furnace Accessories

The two essential warm air furnace accessories are a humidifier and an air filter/cleaner.

a. Humidification

Humidification is almost always required in winter. This is particularly true in residences because the humid air created in the house in areas such as bathrooms, kitchens and laundries is exhausted to the outside. Make-up air from the outside is dry, because cold air carries very little moisture. In small commercial buildings, ventilation achieves the same result by bringing in dry air to replace stale but more humid exhausted air. In both cases, humidification is, therefore, required.

As an example of the dryness of heated make-up air, consider a typical condition. If exterior air at 30°F and 50% RH (typical winter air conditions) is heated in a warm air furnace to 140°F, it will have a relative humidity approaching zero. (See the psychrometric chart in Figure 2.18.) Even after it mixes with room air and its temperature drops to 74°F, this air will have only a 10% RH. This dry air will act to reduce the humidity of all the air in the room. The exact effect will depend on the ventilation rate, occupancy and so on. However, it is clear that humidification is required to compensate for the introduction of very dry air into the building.

Excess humidity is also undesirable. From a purely comfort point of view, an RH of up to 50% is acceptable. High humidity can cause condensation on windows and metal sash. As a rule of thumb, indoor relative humidity should not exceed

- 15 + outdoor temperature (in °F) for single-glazing or
- 25 + outdoor temperature for double-glazing

Thus for an outdoor temperature of 15°F, indoor RH should not exceed 30% for single-glazed spaces and 40% for double-glazed spaces.

This rule of thumb, like all such rules, is only approximate. For an actual installation, a calculation based on the actual *R/U* factors for the window being used should be performed. In the absence of manufacturers' data, the data given in Table 2.1 can be used. Two important facts with regard to winter inside humidity should be remembered.

- The upper limit of humidity drops as outside temperature drops, if we wish to avoid condensation on windows.
- Comfort humidity also drops with colder outside temperature. For this reason, winter comfort humidity is usually given as 35–50%. The lower figure applies at low outside temperatures, and the upper figure applies at higher temperatures.

Refer to the results of Problem 2.11 for more data on this important subject.

There are many types of humidifiers available. The simplest types use pans or multiple porous plates from which water is evaporated. Evaporation can be natural or aided by heat and/or forced air. More complex types use atomizers that spray water droplets into the duct air stream. The droplets evaporate into water vapor within a few feet of travel in the duct. Most units are connected to a piped source of water, which is controlled by a water-level device, such as a float switch. Selection of the appropriate type of humidifier requires calculation (or estimation) of the building humidification requirements. This in turn depends on building volume, occupancy, activity and construction quality plus data on outdoor conditions. Once the humidification requirements are established, a particular unit can be selected. Most furnace units are rated for a specific furnace plenum temperature and for continuous furnace operation. Other plenum temperatures and furnace on-off cycling must be considered in sizing a humidifier. Since actual humidification calculations can only approximate actual field conditions, a humidistat and/or other control device is recommended to control humidification.

b. Air Cleaning

A mechanical filter is the simplest type of air-cleaning device. It can be of the renewable (washable) or throw-away type. It is usually placed in the return air duct, adjacent to the connection to the furnace. See Figure 5.6(a). These units consist of a frame containing treated coarse fibrous material such as glass wool, plastic thread or a combination of materials. Throw-away units called viscous impingement filters remove dust and dirt from the air, not only by filtering but also by capturing impinging particles on their treated sticky surfaces. Renewable, washable units are less effective, because they operate only as mechanical filters. They are seldom used in modern installations.

So-called electronic air cleaners operate in a number of ways. Some add an electrical charge to dust and dirt particles passing through an ionizing section and then trap the charged particles at collection electrodes. Other types attract dirt particles to electrostatically charged plates. Still others trap dust and dirt particles by using the particle's own electrical charge. All these units become "loaded" with dirt and must be cleaned periodically.

As noted in our discussion on external static pressure in Section 5.8.c, the static pressure loss in filters and humidifiers must be considered. Most furnaces are tested and rated with a clean filter installed. This is true for most, but not all. The static pressure loss in a clean filter varies from 0.1 to 0.35 in. w.g. and is 50% higher in a dirty filter. If we remember that the maximum static pressure developed by a residential-type warm air furnace blower rarely exceeds 0.8 in. w.g., we can readily see how important it is to clarify whether the blower static pressure data includes the filter. Static drop in humidifiers is much smaller, depending on type. It varies between negligible for the atomizing types to about .025 in. w.g. maximum for the multiple wetted plate type. Exact static pressure drop figures should be obtained from the manufacturer, using the system's design air velocity.

Air Distribution System

Having discussed the warm air furnace and its accessories, which is the heart of any warm air

heating system, we will now proceed to the air distribution system. This system carries the warm air from the furnace to the various rooms in the building, via a network of supply ducts and fittings. At the duct terminations in the various rooms, the supply air is dispersed by registers specifically designed to distribute the air in optimal fashion. The "used" air is then collected at return air grilles and brought back to the furnace in return ductwork for filtering, reheating, humidification and recirculation. If the building HVAC system is to supply summer cooling as well as winter heating, the duct system may have to be larger than for heating alone. Also, the room air outlets may be different, both in design and placement. These items will become clear as our study progresses.

If we refer again to Figure 5.1, we can see all the essential elements of an air distribution system: ducts, duct fittings, air outlets and air control devices. An understanding of the construction and functioning of each of these system components is essential to an understanding of the system functioning, as a whole.

5.12 Ducts

a. Rectangular Ducts

The most commonly used materials for construction of rectangular ducts in low pressure, low velocity forced-air systems are galvanized steel, aluminum and rigid fibrous glass.

- (1) Galvanized steel is probably the most widely used material for supply and return ducts. When used outdoors, painting is recommended even though the zinc galvanizing acts as an effective weatherproofing for at least 5 years. Galvanized sheet steel has high strength and is rust-resistant; nonporous; highly durable; readily cut, drilled and welded; and easily painted. It is also widely available in the United States in a variety of qualities. The most commonly used type is called lock-form quality, which describes the usual method of joint closure.

Minimum metal gauges for steel and aluminum duct are given in SMACNA standards. The principal disadvantages of galvanized steel duct are its weight and its acoustical characteristics. In addition to being an excellent channel for noise transmission, it is also a source of noise from vibration, particularly in large ducts. Addition of thermal insulation on the

outside, especially if glued, will dampen the metal vibration but will not attenuate the duct's noise transmission ability. For that purpose, acoustical damping inside the duct is required.

- (2) Aluminum is often substituted for galvanized steel because it is lighter and much more corrosion- and weather-resistant. It also has a more attractive appearance, which is a consideration in installations using exposed ductwork. Among the disadvantages of aluminum duct are high cost, low physical strength in the thicknesses used for ducts and a thermal expansion coefficient more than double that of steel. Aluminum is also difficult to weld. The large thermal expansion is not normally a problem in residential work but can be a duct length limiting factor in commercial work. Aluminum is smoother than galvanized steel and, therefore, has a lower static friction drop. Comparative figures for this characteristic are found in Figure 5.54.
- (3) Rigid fibrous glass board, normally 1-in. thick, is used frequently to fabricate rectangular duct. The material is a composite of fibrous glass board with a factory-applied facing of plain or reinforced aluminum. This facing acts as a finish and as a vapor barrier. Rigid fibrous glass board has distinct advantages over metal duct including (light) weight, good thermal insulation and acoustical qualities and simplicity of fabrication and installation. The principal disadvantages of this material are relatively high cost, low physical strength, sensitivity to moisture and pressure limitations (2 in. w.g.). In addition, in some areas of the country, it is not acceptable according to local codes. Another disadvantage is its higher static friction loss as compared to metal duct. Although this last item is rarely a deciding factor, it may cause the designer to use larger and, hence, more expensive ducts. Use of fibrous glass duct is limited to locations where the duct is not subject to physical damage. Also, because of its lack of physical strength, risers are limited to about 20 ft.
- (4) Other materials including black carbon steel, stainless steel, fiberglass-reinforced plastic (FRP), gypsum board and polyvinyl chloride (PVC) are all used for special-purpose rectangular duct. They are not considered to be general-purpose duct materials and will, therefore, not be discussed here. You can consult publications of the Sheet Metal and Air Conditioning Con-

tractors National Association, Inc. (SMACNA) for further information on these duct materials as well as construction and installation standards for all ductwork.

b. Round Ducts

Round metal ducts are very common in HVAC systems because they are strong, rigid, efficient and economical. Round ducts have the lowest static friction loss of any shape, and the highest ratio of cross-sectional area to perimeter of any shape. That means that for a given cross-sectional area, a round duct will use less material than any other shape. It is, therefore, cheaper than any other shape. The considerations involved in duct shape are discussed at length in Section 5.22 where duct friction is analyzed.

Round ducts also have the advantage of great rigidity as a result of shape. This makes them ideal for installation in locations where the ducts are subject to physical abuse. This rigidity also minimizes noise and vibration transmission. Another advantage of round ducts is economy of insulation due to the minimal perimeter of the round shape. The only major disadvantage of round ducts is that they often will not fit into tight locations such as hung ceilings or between joists, because of their shape. Such locations require rectangular ducts. Round ducts are generally metallic, although prefabricated round rigid fibrous glass duct is available.

c. Oval Ducts

Oval ducts were developed to solve the bulky shape problem of round ducts without losing their advantage. They are almost as efficient and rigid as round ducts, and their flatter profile allows their use in tight locations such as between wall studs and between joists in framed houses. Their only disadvantage is, at this writing, their price premium. Table 5.5 (page 287) lists oval duct sizes and their round duct equivalents.

d. Flexible Ducts

Flexible ducts are available in two basic designs: metallic, both insulated and bare, and mesh covered, insulated, metallic helix (see Figure 5.16). All flexible ducts have considerably higher friction than their nonflexible equivalent. This, however, is not usually an important factor since flexible ducts are most often used in short runs. Despite their

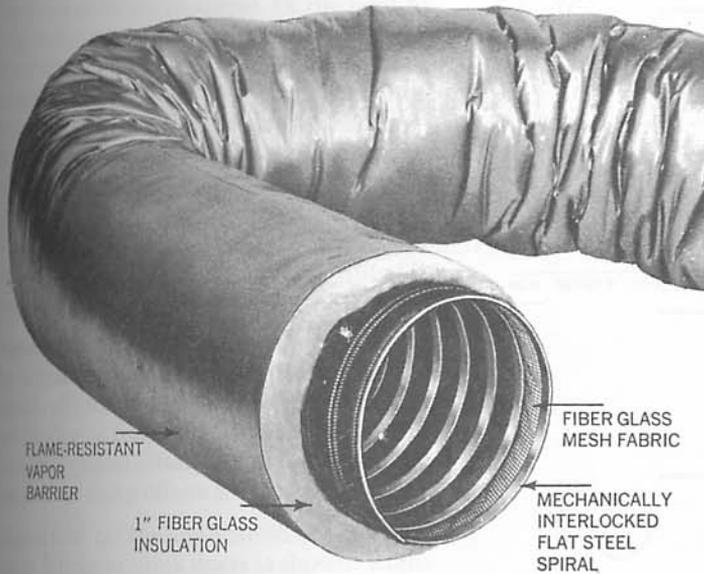


Figure 5.16 Insulated flexible duct, usable in low velocity systems. It consists of a steel helix (spiral) covered with a fiberglass fabric and thermal insulation. These ducts can be spliced, clamped and formed into oval ends. Their principal use is for short, angled runs and sharp turns.

high cost, they are considerably cheaper than the custom-made joints and fittings used with rigid duct.

5.13 Duct Fittings and Air Control Devices

A well-designed duct system has low static friction loss. Any change in duct size or direction can cause turbulence in the air stream. This is turn increases the static head loss in the duct system. (This subject will be treated more full in Section 5.21.) As a result, a good duct system uses duct fittings that are specifically designed to minimize air turbulence. This means that transitions are smooth [Figure 5.17(a)], branch takeoffs are gradual [Figure 5.17(b)], elbows are long and rounded, [Figure 5.17(c)], and so on. Figure 5.18 shows a few of the many types of duct fittings and a typical duct hanging detail.

Figure 5.19 shows some of the more commonly used duct air control devices including turning vanes, splitters and volume dampers. Volume dampers are constructed in single-leaf (e) or multileaf (opposing blade) design (c). The purpose of volume dampers is simply to control the quan-

tity of air in a duct. Every branch duct is equipped with a volume damper, which is capable of being locked in position. Turning vanes (f) are used where there is insufficient space for a long sweep elbow and a sharp turn right angle elbow must be used. Figure 5.20 shows the drawing symbols and abbreviations commonly used on HVAC working drawings.

5.14 Duct Insulation

Duct insulation is generally provided on ducts in accordance with energy codes and ASHRAE standards. In their absence, use the following guidelines:

- (a) Do not insulate ducts passing through a conditioned space, or through furred interior spaces.
- (b) Do not insulate ducts of "heat-only" systems that pass through unheated basements.
- (c) Do insulate ducts of heating/cooling systems passing through unheated basements. This will probably result in a requirement for a heating register in the duct to compensate for the lack of basement heating from duct heat loss, in winter.

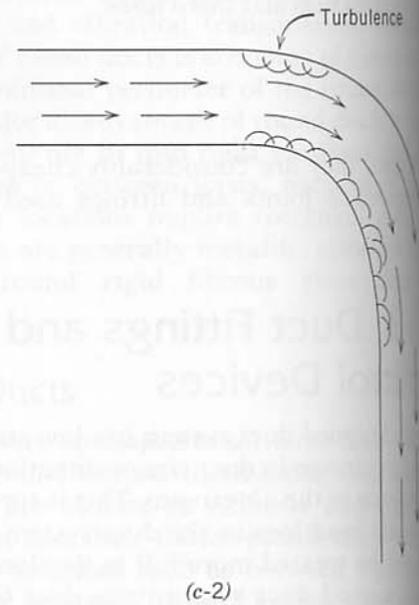
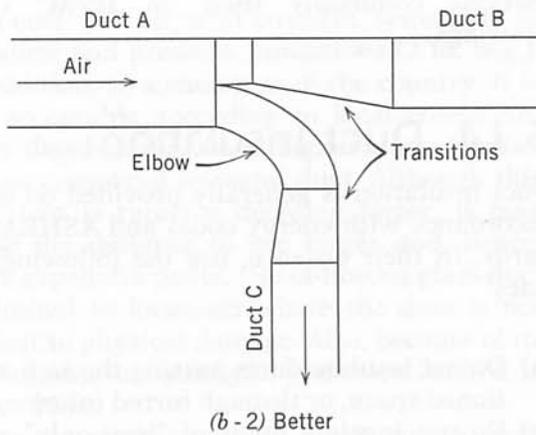
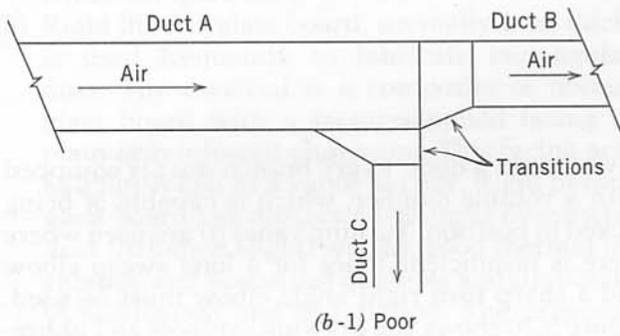
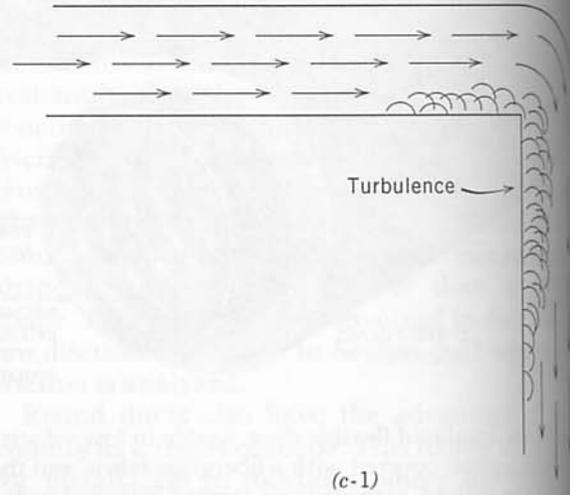
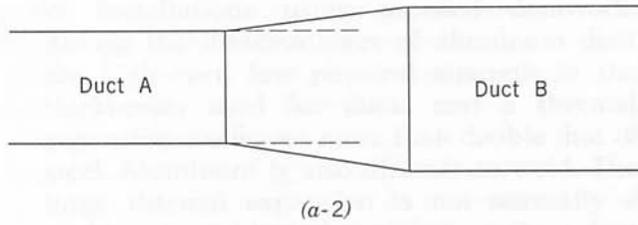
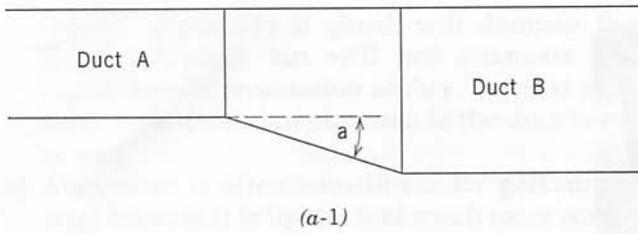


Figure 5.17 Transitions can be either single (a-1) or double (a-2). Angle a should be as small as possible, to reduce turbulence, and should not exceed 20° . If space is tight, requiring a larger angle, a double transition (a-2) can be used. Takeoff connections should not be at right angles (b-1) but should use a long elbow (b-2) connected in the direction of airflow. Sharp turns (c-1) cause severe turbulence and result in high static friction losses. Long gradual elbows (c-2) have minimal turbulence and low losses.

Figure 5.18 (a) Air boot fittings are terminations of branch ducts, onto which registers, diffusers and grills are mounted. Of the boot fittings shown, types H, I and J connect to round branch ducts; types A, C and O connect to square branches, and types M and N connect to rectangular branches. Note that transitions are gradual and that right angles are avoided wherever possible. For an explanation of "equivalent length" see Section 5.23. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

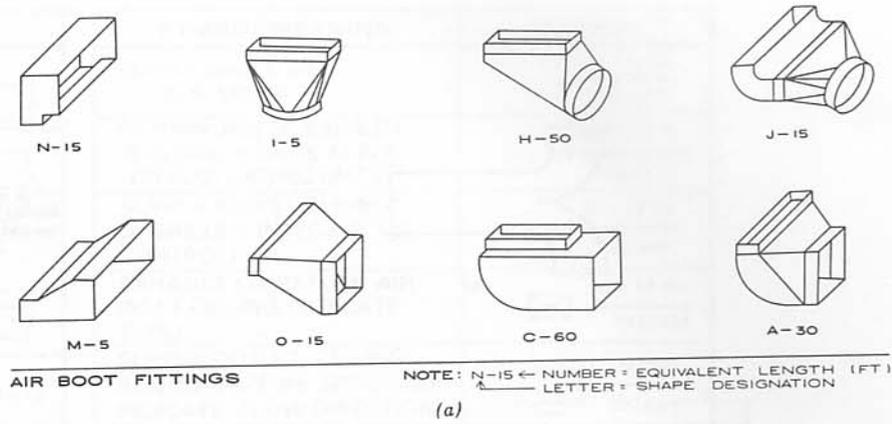


Figure 5.18 (b) Angles, elbows and offsets can be of standard design (B, G, I, E) or fabricated specifically for job conditions (K, L, M). Offsets are fabricated from two elbows and a straight section. Gradual offsets (L, M) have much lower losses than sharp offsets like type K.

Figure 5.18 (c) The application of elbows and angle fittings to trunk ducts is shown. Right angle elbow D uses internal turning vanes to reduce turbulence. See text.

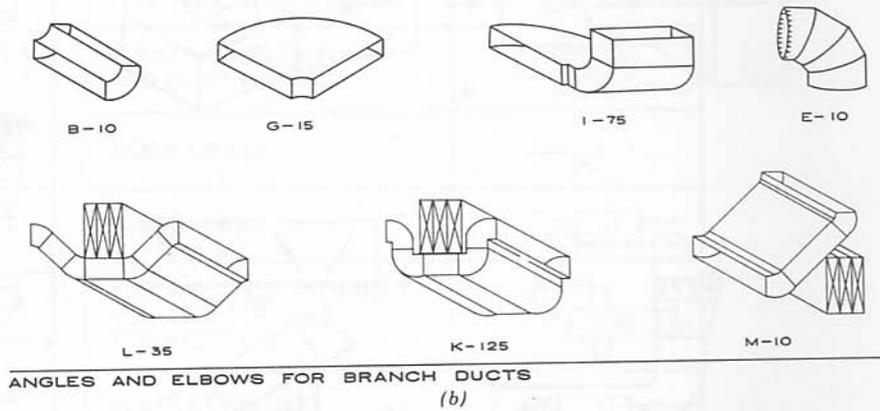


Figure 5.18 (d) Two typical trunk ducts with takeoffs are shown. At the left, the trunk reduces at each takeoff, and the takeoff connects at the transition fitting. Compare this to Figure 5.17b. At the right, the duct size reduction is one-sided, and all takeoffs are with round duct. This design is common in residential work. Both trunk designs are known as reducing trunk or reducing plenum designs.

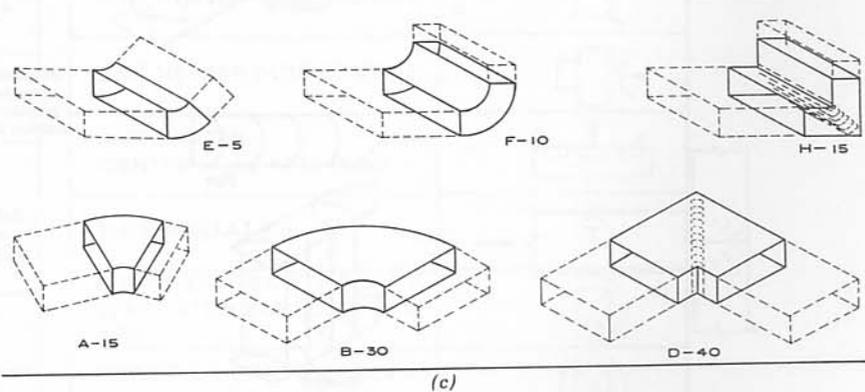
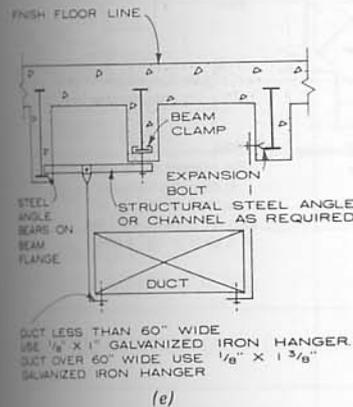
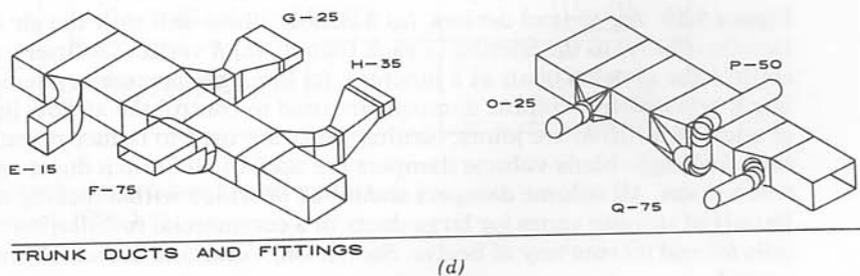


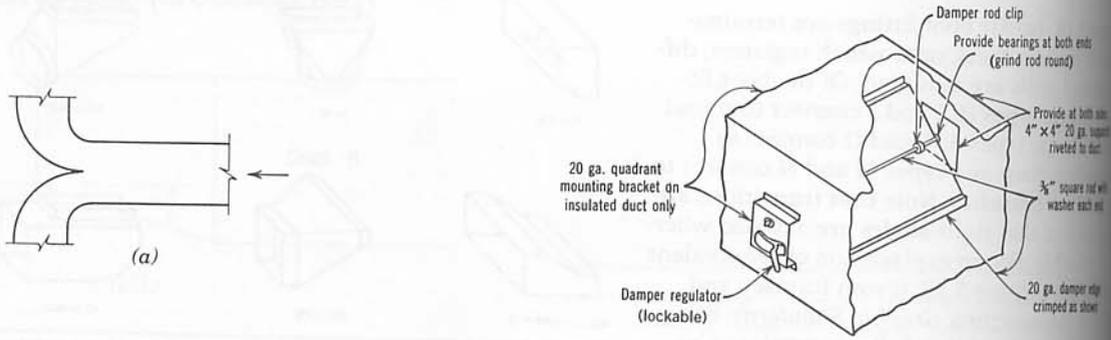
Figure 5.18 (e) Typical hanger detail for rectangular duct. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., p. 628, 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)



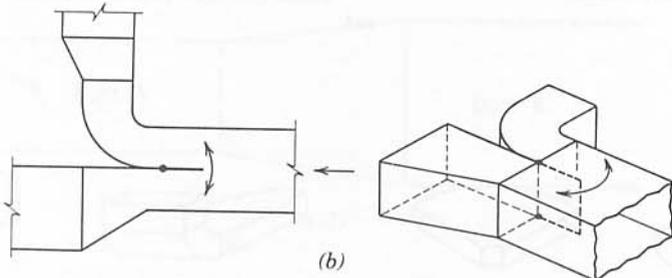
NOTE
 On ducts over 48 in. wide hangers shall turn under and fasten to bottom of duct. When cross-sectional area exceeds 8 sq ft duct will be braced by angles on all four sides.

DUCT SUPPORT DETAIL

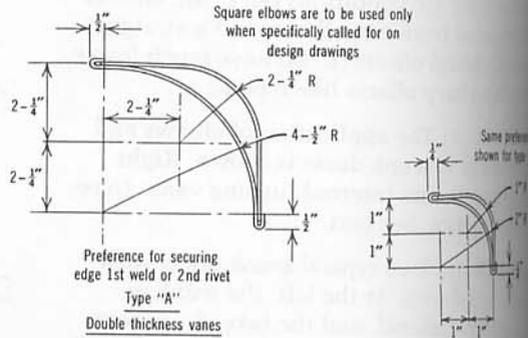




Note:
For ducts up to 29" wide
and/or up to 12" high
(e)



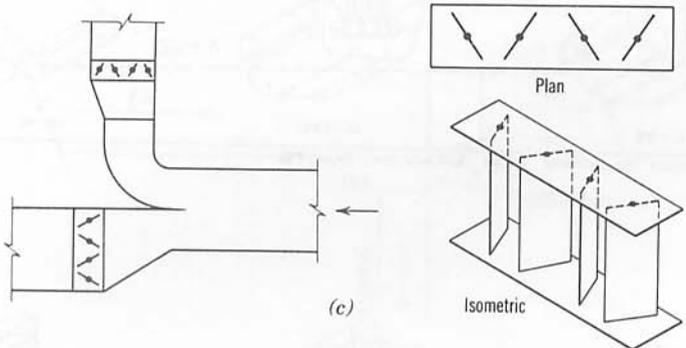
(b)



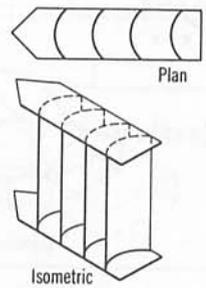
(c)

Preference for securing
edge 1st weld or 2nd rivet
Type "A"
Double thickness vanes
For use in ducts greater than
24" x 24" in size. Use same gauge
galvanized iron as duct, not to
exceed 20 gauge

Type "B"
Double thickness vanes
For use in ducts 24" x 24" and above



(d)



(d)

Use galvanized vanes for
galvanized or aluminum ducts
Plate same gauge
thickness as duct
9"
3-3/4"
Duct size
over
24" x 24"
Square elbow with type "A" double
thickness vanes

Vanes
preassembled
on runner
plates

Square elbow with type "B" double
thickness vanes

Notes:
Use galvanized steel for vanes
in either steel or aluminum
ductwork. Provide 1 stay for
ducts 72" to 120" wide & 2 stays
at 3/5 points for 120" & above.

(f)

Figure 5.19 Air control devices. (a) A double elbow will split the air flow in proportion (inversely) to the friction in each branch. (b) A vertical splitter vane is used to control the division of air at a juncture. (c) In higher pressure systems (above 0.5 in. w.g.), opposed-leaf volume dampers are used to control the airflow into each branch of a junction. (d) At tee joints, turning vanes are used to reduce turbulence and friction. (e) Single-blade volume dampers are used in all branch ducts and in some trunk ducts. All volume dampers should be provided with a locking mechanism. (f) Details of turning vanes for large ducts in a commercial installation. [Drawing details (e) and (f) courtesy of Seelye, Stevenson, Value and Knecht, Consulting Engineers.]

SYMBOL MEANING	SYMBOL	SYMBOL MEANING	SYMBOL
POINT OF CHANGE IN DUCT CONSTRUCTION (BY STATIC PRESSURE CLASS)		SUPPLY GRILLE (SG) (U.S. UNITS)	
DUCT (1ST FIGURE, SIDE SHOWN 2ND FIGURE, SIDE NOT SHOWN)		RETURN (RG) OR EXHAUST (EG) GRILLE (NOTE AT FLR OR GLG) (METRIC UNITS)	
ACOUSTICAL LINING DUCT DIMENSIONS FOR NET FREE AREA		SUPPLY REGISTER (SR) (A GRILLE + INTEGRAL VOL CONTROL)	
DIRECTION OF FLOW		EXHAUST OR RETURN AIR INLET CEILING (INDICATE TYPE)	
DUCT SECTION (SUPPLY) (U.S. UNITS)		SUPPLY OUTLET, CEILING, ROUND (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION	
DUCT SECTION (EXHAUST OR RETURN) (METRIC UNITS)		SUPPLY OUTLET, CEILING, SQUARE (TYPE AS SPECIFIED) INDICATE FLOW DIRECTION	
INCLINED RISE (R) OR DROP (D) ARROW IN DIRECTION OF AIR FLOW		TERMINAL UNIT. (GIVE TYPE AND/OR SCHEDULE)	
TRANSITIONS. GIVE SIZES NOTE F O T FLAT ON TOP OR F O B FLAT ON BOTTOM IF APPLICABLE		COMBINATION DIFFUSER AND LIGHT FIXTURE	
STANDARD BRANCH FOR SUPPLY & RETURN (NO SPLITTER)		DOOR GRILLE	
SPLITTER DAMPER		SOUND TRAP	
VOLUME DAMPER MANUAL OPERATION		FAN & MOTOR WITH BELT GUARD & FLEXIBLE CONNECTIONS	
AUTOMATIC DAMPERS MOTOR OPERATED		VENTILATING UNIT (TYPE AS SPECIFIED)	
ACCESS DOOR (AD) ACCESS PANEL (AP)		UNIT HEATER (DOWNBLAST)	
FIRE DAMPER: SHOW ◀ VERTICAL (Wall) SHOW ◆ HORIZ. (Floor)		UNIT HEATER (HORIZONTAL)	
DAMPERS: (SD) SMOKE— S FIRE SMOKE (F/SD)— ▲		UNIT HEATER (CENTRIFUGAL FAN) PLAN	
HEAT STOP FOR FIRE RATED CLG		THERMOSTAT	
TURNING VANES		POWER OR GRAVITY ROOF VENTILATOR-EXHAUST (ERV)	
FLEXIBLE DUCT FLEXIBLE CONNECTION		POWER OR GRAVITY ROOF VENTILATOR-INTAKE (SRV)	
GOOSENECK HOOD (COWL)		POWER OR GRAVITY ROOF VENTILATOR-LOUVERED	
BACK DRAFT DAMPER		LOUVERS & SCREEN	

Figure 5.20 (a) Symbols commonly used in HVAC work. (Reproduced with permission from SMACNA HVAC Systems Duct Design Manual, 1990.)

AHU	Air-handling Unit	FTR	Fin-tube Radiation
C	Condensate (Steam)	GC	General Contractor
CD	Cold Duct	HP	Heat Pump
CDR	Condensing Water Return	HPS	High Pressure Steam
CDS	Condensing Water Supply	HWR	Heating Water Return
CFM	Cubic Feet/Minute of Air	HWS	Heating Water Supply
CHWR	Chilled and Heating Water Return	LPS	Low Pressure Steam
CHWS	Chilled and Heating Water Supply	MB	Mixing Box
CWR	Chilled Water Return	RA	Return Air
CWS	Chilled Water Supply	RAD	Return Air Damper
EF	Exhaust Fan	SA	Supply Air
FA	Fresh Air	UH	Unit Heater
FAD	Fresh Air Damper	UV	Unit Ventilator
FCU	Fan Coil Unit	VD	Volume Damper
FD	Fire Damper		

Figure 5.20 (b) List of abbreviations commonly used in HVAC work.

- (d) Do not insulate perimeter heating ducts run in the concrete floor slab. These ducts help to heat the slab and contribute materially to comfort in the space.
- (e) Do insulate ducts run outside and in attics, garages, crawl spaces and any other space where the heat loss (or gain) is clearly wasted.
- (f) If at all possible, avoid installation of ducts in exterior walls and attics. If unavoidable, insulate the ducts according to the formula in the next paragraph.

As a rule, the R value of duct insulation should be

$$R = \frac{\Delta t \text{ (}^\circ\text{F)}}{15}$$

where Δt is the temperature difference between the air in the duct and surrounding ambient air temperature.

Thus, for a duct passing through an uninsulated attic and carrying cooling air at 55°F, we would first estimate the summer attic temperature to be 140°F. Then

$$R = \frac{140 - 55}{15} = \frac{85}{15} = 6, \text{ use R-6 insulation.}$$

For the same duct carrying warm air at 140°F, with a winter attic temperature of 20°F:

$$R = \frac{140 - 20}{15} = \frac{120}{15} = 8, \text{ use R-8 insulation}$$

We would, therefore, probably insulate the duct for the larger of the two requirements, that is, R-8. We

say probably because it is essentially an economic decision. As an absolute minimum, use R-2 insulation on ducts in insulated enclosed crawl spaces and in uninsulated basements and R-4 everywhere else.

Since blanket insulation is necessarily compressed when installed around a duct, even if glued, most authorities recommend doubling the normal thickness required. That assumes a 50% compression. Using this assumption, typical duct insulation would be:

- R-2: 1 in. of (compressed) glass fiber blanket or 1/2 in. of rigid glass fiber board or liner
- R-4: 2 in. of (compressed) glass fiber blanket or 1 in. of rigid glass fiber board or liner
- R-6: 3 in. of (compressed) glass fiber blanket or 1 1/2 in. of rigid glass fiber board or liner
- R-8: 2 in. (compressed) glass fiber blanket covered with 1 in. of rigid glass fiber board or liner

In all cases, a vapor barrier on the outside of the insulation should be provided if the duct is carrying cool conditioned air.

As an indication of the importance of duct insulation, we can calculate temperature gain and loss in air carried by an uninsulated duct. An uninsulated duct 50 ft long, carrying 2000 cfm of 55°F air through a 140°F attic at 700 fpm (common conditions for a trunk feeder duct) will gain about 11°F. That means that the cool air supplied at the end will be at 66°F instead of 55°F. Obviously, the

desired cooling will not be achieved. Similarly, in winter, the same duct carrying 1500 cfm of warm air at 140°F through the same attic at 20°F will lose almost 20°F. That means that the warm air will be delivered at 120°F instead of 140°F. Since the total temperature rise is about 70°F (from 70°F room temperature to 140°F supply temperature), this 20°F loss represents a loss of almost 30% of the heating capacity of the duct! Obviously then, duct insulation can be critical for ducts passing through unconditioned, uninsulated interior spaces or through exterior areas.

5.15 Air Distribution Outlets

Conditioned air originates at the warm air furnace (or heat pump/refrigeration unit) is distributed by the duct system and terminates at the room outlets. After mixing with room air, the “deconditioned” air is picked up at a return air outlet. It is then carried back to the furnace for reprocessing and recirculation. Supply air outlets are normally registers or diffusers, and occasionally grilles. Return air outlets are almost always grilles, and occasionally registers.

A *grille* is any slotted, louvered or perforated cover that fits onto a duct termination. See Figure 5.21. Louvered grilles have horizontal and/or vertical vanes. These vanes may be fixed or movable. The purpose of the grille is to permit free passage of air while generally obscuring direct vision or access into the duct. Fixed-opening grilles are normally mounted on return ducts and are wall-, ceiling- or floor-mounted. Adjustable vane units are normally used on supply ducts and are wall- or floor-mounted.

a. Definitions

A *register* is a grille that is equipped with some type of volume damper for the control of air flow. See Figure 5.22. It normally has movable vanes for directing the air stream. Registers are almost always used on supply ducts. In very special cases requiring volume damping, registers may also be used on return ducts.

A *diffuser* is a supply air outlet normally designed to distribute the air it supplies in a widespread pattern roughly parallel to the surface in which it is mounted. Most diffusers are intended for ceiling mounting, although wall diffusers are used as well. The majority of diffusers are equipped with devices that permit changing their air distri-

bution patterns. They may or may not be equipped with volume dampers, as needed. Diffusers can be round, square, rectangular or linear as required. See Figure 5.23.

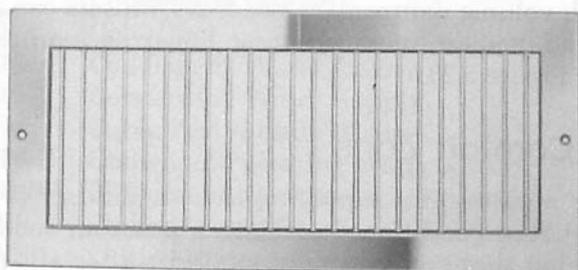
b. Comfort Zone

The purpose of a supply register or diffuser is to introduce conditioned air into a space in such a fashion that comfort is maintained. After all, the whole purpose of supplying conditioned air is to establish comfortable conditions in the room. Let us review very briefly what these comfort conditions are, as regards air temperature and air speed (see Section 5.1).

Winter: 68–74°F DB, 40 fpm maximum
 Summer: 73–79°F DB, 50 fpm maximum

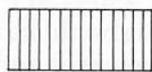
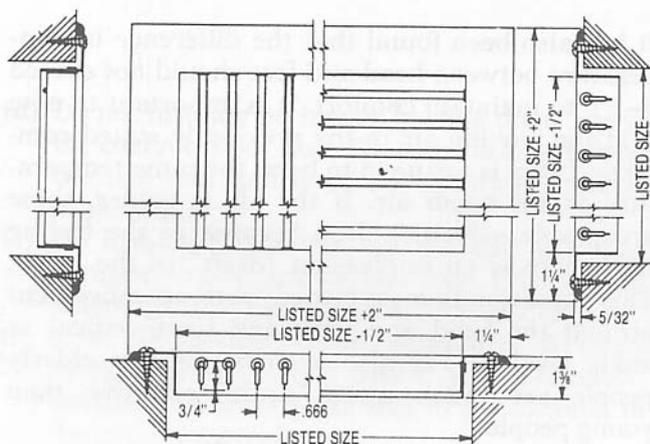
It has also been found that the difference in temperature between head and feet should not exceed 4–5°F to maintain comfort. It is important to note that the moving air in the previously stated comfort criteria is assumed to be at the same temperature as the room air. If the air is colder, these acceptable velocities drop because of the feeling that there is an unpleasant “draft” in the room. This condition is most critical with air movement around the head and neck and least critical at ankle level. It has also been found that elderly people are much more “draft” sensitive than young people.

Now consider that the air in the branch ducts strikes the supply register or diffuser at 500–600 fpm, at a winter temperature of 130–150°F or a summer temperature of 53–57°F. It should be obvious that no register or diffuser, however well designed, can instantaneously convert this air to the previously listed comfort conditions. This is one of the reasons that the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), in its standard 55, which defines comfort conditions, specifies an “occupied zone” in a room, inside which comfort conditions must be maintained. Another reason is that, in rooms with cold walls or ceilings, there is always a temperature gradient between the comfort conditions in the occupied zone and the area near cold walls. This occupied zone is shown in Figure 5.24. It is simply the area from the floor to a height of 6 ft and the room volume that is 2 ft from any wall. Establishing this occupied zone means that air from a wall, floor or ceiling supply outlet has about 2 ft within which to alter its temperature and

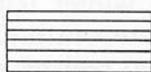


Single deflection
supply grille

(a-1)



Vertical Bars

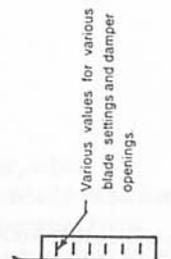


Horizontal Bars

(a-2)

Figure 5.21 (a-1) A single deflection supply grille controls the air stream flow in two directions. Units are available with horizontal or vertical bars. (a-2) Section through the grille showing construction of both designs. (a-3) Performance data of the grille. (Courtesy of Carnes Company.)

SIZE	300		400		500		600		700		CFM	Throw	NC										
	0	22 1/2	45	0	22 1/2	45	0	22 1/2	45	0				22 1/2	45								
6x6	.010	.012	.027	.014	.021	.050	.023	.034	.082	.034	.051	.120	.047	.071	.165	12x10	24x5	250	330	420	500	580	
8x4																14x8	30x4	L	18	23	26	31	
																20x6		L	L	L	L	22	
8x5																12x12	18x8	300	400	500	600	700	
10x4																14x10	24x6	Throw	15	20	25	29	34
																16x8	28x5	NC	L	L	L	24	28
8x6																16x10	40x4	CFM	330	430	540	650	760
10x5																26x6		Throw	15	21	26	30	36
12x4																30x5		NC	L	L	20	25	29
14x4																		CFM	410	540	680	820	950
																15x12	32x6	Throw	17	22	29	34	40
																20x10	34x6	NC	L	L	21	28	31
8x8																18x14	48x5	CFM	470	620	780	930	1090
10x6																18x12		Throw	18	24	31	37	43
12x5																35x6		NC	L	L	22	28	32
12x6																16x16	25x10	CFM	530	710	890	1070	1250
14x5																18x14	32x8	Throw	20	26	33	40	46
18x14																22x12	42x6	NC	L	L	23	29	33
10x8																18x16	30x10	CFM	600	800	1000	1200	1400
14x6																20x14	36x8	Throw	21	27	35	42	49
16x5																24x12	48x6	NC	L	L	24	30	34
10x10																18x18	28x12	CFM	680	900	1120	1350	1580
12x8																20x16	30x12	Throw	22	29	37	44	52
16x6																24x14	36x10	NC	L	L	26	31	35
18x6																20x20	28x14	CFM	830	1110	1390	1670	1950
28x4																22x18	34x12	Throw	24	32	41	49	57
																24x16	40x10	NC	L	21	27	33	37



Notes:

- (1) Additions and factors (listed below) have to be applied for varying blade settings and damper openings.
- (2) For sizes, CFM, blade settings or damper openings, etc., not listed, interpolate as necessary.

SINGLE DEFLECTION

SYMBOLS

- V = Duct velocity in fpm.
- CFM = Quantity of air in cubic ft./min.
- NC = Noise criteria (8 db room attenuation), re 10⁻¹² watts.
- P₁ = Total pressure inches H₂O.
- T = Throw in feet.
- L = NC less than 20

NC — Add the following db to the NC obtained from Table No. 1 for various blade settings.

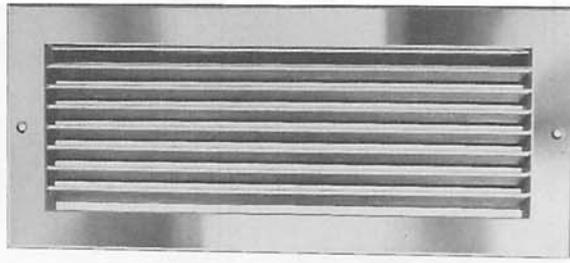
Duct Velocity	300	400	500	600	700	800	900	1,000	1,200
45°	6	6	6	6	5	5	5	5	5

T — Multiply the T listed in Table No. 1 by the following F₁ factor for various blade settings.

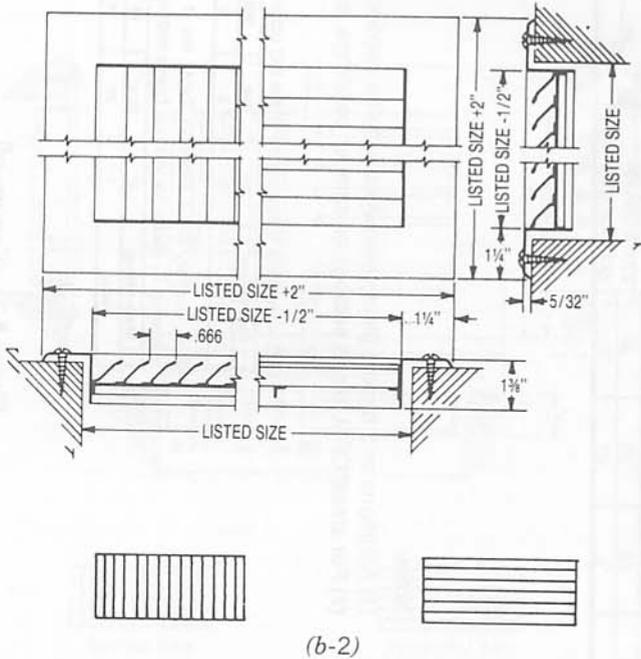
Blade Setting	0°	22 1/2°	45°
Factor	1	.89	.60

(α-3)

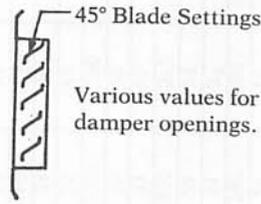
Figure 5.21 (Continued)



Fixed blade return air grille
(b-1)



Return Air Performance Data

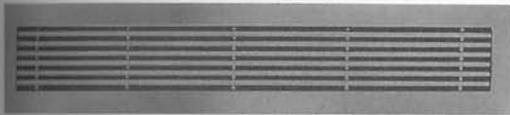


SYMBOLS:
 V = Duct velocity in fpm
 CFM = Quantity of air in cubic ft./min.
 NC = Noise criteria (8 room attenuation) re 10⁻¹² watts.
 L = NC less than 20
 P_t = Total pressure inches H₂O.

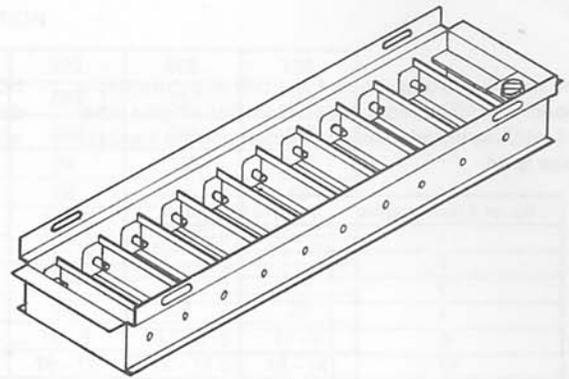
Size	V P _t	200 .03	400 .09	600 .16	800 .25
4x4	CFM	22	44	66	88
	NC	L	L	L	20
6x6	CFM	50	100	150	200
8x4	NC	L	L	L	24
8x6 12x4	CFM	65	130	200	270
10x5	NC	L	L	L	25
10x6 16x4	CFM	80	160	240	320
12x5 18x4	NC	L	L	21	27
8x8 14x5	CFM	90	180	260	350
12x6	NC	L	L	22	28
10x8 16x5	CFM	110	220	330	440
14x6 20x4	NC	L	L	23	30
10x10 20x5 12x8	CFM	140	280	400	550
28x4 18x6	NC	L	L	25	32
12x10 24x5 14x8	CFM	160	320	480	640
30x4 20x6	NC	L	L	25	33
14x10 28x5 16x8	CFM	190	380	570	760
36x4 22x6	NC	L	L	27	35
12x12 26x6 16x10	CFM	200	400	600	800
30x5 18x8 40x4	NC	L	L	28	36
14x14 24x8 16x12	CFM	270	540	820	1090
34x6 20x10	NC	L	L	30	38
16x14 48x5	CFM	310	620	930	1240
18x12 36x6	NC	L	20	31	39
16x16 24x10 30x8	CFM	360	710	1070	1420
18x14 22x12	NC	L	21	33	41
18x16 30x10 36x8	CFM	400	800	1200	1600
20x14 24x12 48x6	NC	L	22	34	42
18x18 30x12 36x10	CFM	450	900	1350	1800
20x16 24x14 40x8	NC	L	23	35	43
22x20 30x14 36x12	CFM	600	1200	1800	2400
24x18 26x16	NC	L	26	37	46
24x24 48x12	CFM	800	1600	2400	3200
30x18 36x16	NC	L	28	40	49
28x24 48x14	CFM	900	1800	2700	3600
32x20 36x18	NC	L	29	41	50
30x24 36x20	CFM	1000	2000	3000	4000
48x16	NC	L	30	42	51
36x24	CFM	1200	2400	3600	4800
48x18	NC	20	31	43	52

Figure 5.21 (b-1) Return air grille with stationary curved blades. The blades are available in vertical or horizontal position. Performance is unaffected by blade orientation. (b-2) Section through the grille showing construction of both designs. (b-3) Performance data. (Courtesy of Carnes Company.)

(b-3)

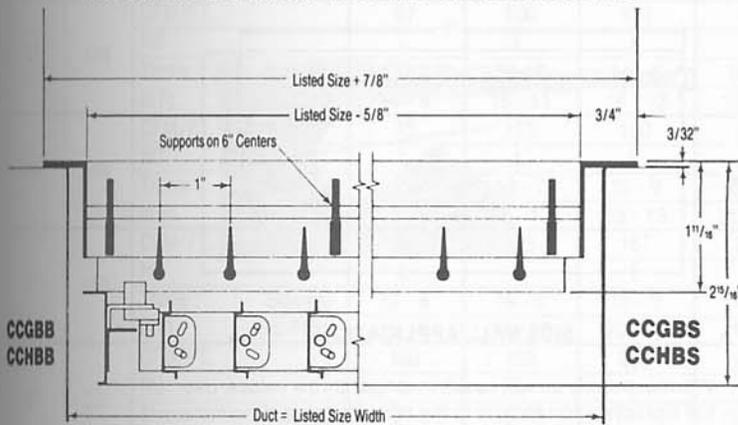


Linear grille/register
(c-1)

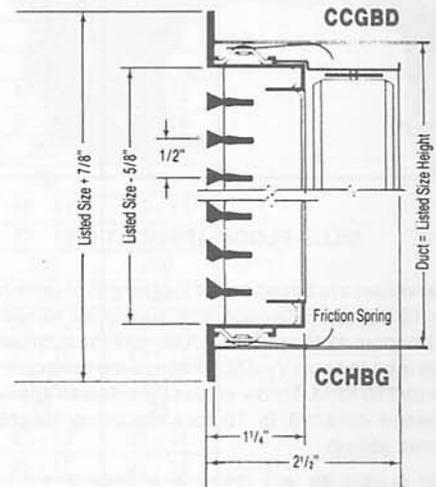


(c-2)

FLOOR APPLICATION MODELS CCGB and CCHB



DIMENSIONAL DATA



Model	Degree Blade Deflection	Damper	Straightening Vanes	Available Sizes*			
				Listed Size Height		Listed Size Width	
				Min.	Max.	Min.	Max.
CCGBG	0	NO	NO	2	12	6	72
CCHBG	15	NO	NO	2	12	6	72
CCGBD	0	YES	NO	2	12	6	72
CCHBD	15	YES	NO	2	12	6	72
CCGBS	0	NO	YES	2	12	6	72
CCHBS	15	NO	YES	2	12	6	72
CCGBB	0	YES	YES	2	12	6	72
CCHBB	15	YES	YES	2	12	6	72

*Minimum & Maximum single unit sizes. Larger list widths furnished as separate pieces.

(c-3)

Figure 5.21 (c-1) Linear grille, usable as a supply or return air grille or, when equipped with a damper (c-2), as a supply register. This design is commonly used for floor, sill, sidewall and ceiling applications. The damper actuator is operated by a screwdriver inserted through the grille face. (c-3) Section through a typical unit in a floor mount application. (Courtesy of Carnes Company.)

Sound ratings are based on a 4 foot unit with the damper full open, and 10 db room attenuation. For lengths other than 4 feet, use the table below to determine the increase in noise level.

No. of 4 foot lengths	db to be added
1	0
2	3
3	5
4	6
6	8
10	10

Tests show that drastic dampering at the grille will result in considerable db increase. Dampering at the grille should be reserved for fine balancing. Gross balancing should be provided for by dampers upstream in the supply ductwork.

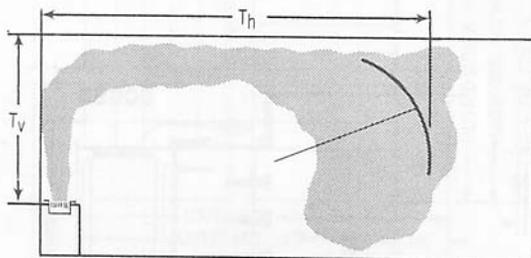
NC values shown in the performance tables are for the damper in the full open position. Partially closed dampers will increase the NC level as shown in the table below.

Effective Damper Opening %	db to be added
100	0
82	8
71	13
50	21

"L" indicates an NC value less than 20.

The total and static pressure is with damper in the full open position and is given in inches water gage (W.G.)

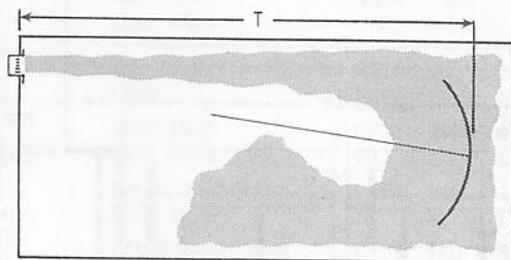
THROW



SILL & FLOOR APPLICATION

Throw values are based on a 4 foot length of grille having 0° or 15° blade deflection and supply air temperature equal to room air temperature. The maximum throw value shown is based on a V_t of 50 FPM and the minimum throw value on 150 FPM. Throw values for sidewall application are based on an 8 to 10 foot mounting height (See sketches above).

Cooler supply air will result in shorter throw values.



SIDEWALL APPLICATION

Warmer supply air will result in longer throw values. Use the multiplication factors in the table below to determine throw values depending on supply air temperature.

V_t FPM	Isothermal	$\Delta t = -20^\circ \text{ F}$	$\Delta t = +20^\circ \text{ F}$
150	1.00	1.00	1.00
50	1.00	.90	1.10

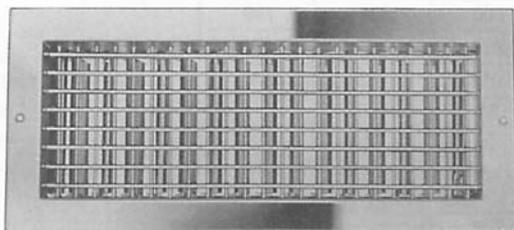
PERFORMANCE DATA – 0° BLADE DEFLECTION

List Size Height	A _k Per Ft. of Length	Duct Velocity - FPM		200	300	400	500	600	700
		Total Pressure P _t		.010	.025	.046	.073	.107	.147
		Static Pressure P _t		.008	.020	.037	.058	.085	.117
2"	.038	CFM/FT.		33	50	67	84	100	117
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	8 - 4	9 - 6	11 - 6	11 - 8	12 - 8	15 - 9
			Sill-Floor	12 - 8	13 - 10	14 - 11	15 - 12	16 - 12	18 - 13
2½"	.063	CFM/FT.		42	62	83	104	125	146
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	8 - 4	10 - 6	12 - 7	12 - 8	13 - 8	15 - 9
			Sill-Floor	13 - 9	14 - 11	15 - 12	16 - 12	17 - 12	18 - 14
3"	.089	CFM/FT.		50	75	100	125	150	175
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	8 - 4	10 - 6	12 - 7	13 - 8	14 - 8	15 - 9
			Sill-Floor	13 - 9	15 - 11	16 - 12	17 - 12	18 - 13	19 - 14
3½"	.114	CFM/FT.		58	88	117	146	175	204
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	8 - 4	11 - 7	13 - 8	14 - 9	15 - 9	16 - 10
			Sill-Floor	14 - 9	15 - 11	17 - 12	18 - 13	19 - 13	20 - 15
4"	.139	CFM/FT.		67	100	133	176	200	233
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	10 - 4	12 - 7	14 - 8	15 - 9	16 - 10	17 - 11
			Sill-Floor	14 - 9	16 - 11	18 - 12	19 - 13	20 - 14	21 - 15
4½"	.164	CFM/FT.		75	113	150	188	225	263
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	11 - 4	13 - 7	15 - 9	16 - 10	17 - 10	18 - 12
			Sill-Floor	15 - 9	16 - 12	19 - 13	20 - 14	21 - 14	22 - 16
5"	.189	CFM/FT.		83	125	167	209	250	292
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	12 - 4	14 - 8	16 - 9	17 - 10	18 - 11	20 - 13
			Sill-Floor	15 - 9	17 - 12	20 - 13	21 - 14	22 - 15	23 - 16
6"	.238	CFM/FT.		100	150	200	250	300	350
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	12 - 4	15 - 8	17 - 10	19 - 11	20 - 12	22 - 13
			Sill-Floor	16 - 9	18 - 12	20 - 13	22 - 15	23 - 16	25 - 17
8"	.322	CFM/FT.		133	200	267	334	400	467
		NC		L	L	L	L	L	L
		Throw in Ft.	Sidewall	13 - 4	16 - 9	18 - 11	21 - 12	23 - 13	25 - 14
			Sill-Floor	17 - 10	19 - 13	21 - 14	23 - 16	25 - 17	27 - 18

Note: A_k is the effective free area, in square feet

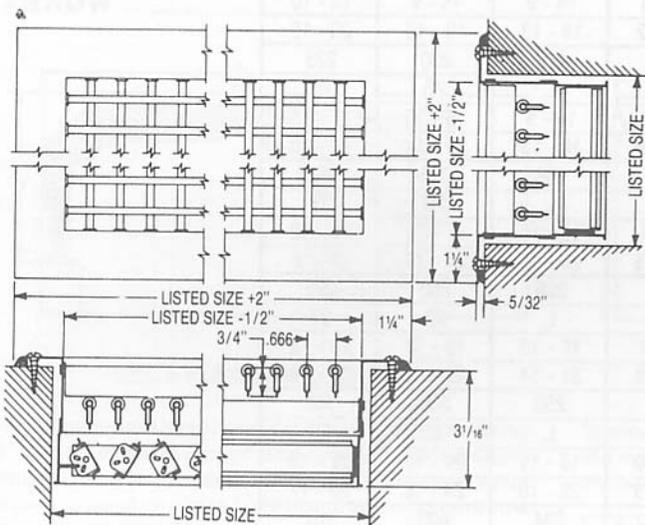
(c-5)

Figure 5.21 (c-5) Typical performance data for 0° blade deflection. A similar table is available for 15° deflection.

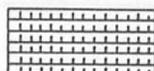


Double-deflection register

(a-1)



Vertical Bars



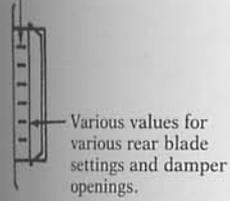
Horizontal Bars

(a-2)

Figure 5.22 (a-1) Double-deflection supply register combines a double deflection grille with an opposed blade damper. This combination provides air deflection in one, two, three or four directions, together with positive volume control. Units are available with vertical front bars and horizontal rear bars, or the reverse. (a-2) Section through the register showing construction of both designs. (Courtesy of Carnes Company.)

Performance Data

FRONT BLADE 0°



DOUBLE DEFLECTION

SYMBOLS:

- V = Duct velocity in fpm
- CFM = Quantity of air in cubic ft./min.
- NC = Noise criteria (8 db room attenuation) re 10⁻¹² watts.
- P_t = Total pressure inches H₂O.
- T = Throw in feet.
- L = NC less than 20

Size	Blade Set° P _t	V = Duct Vel.														
		300			400			500			600			700		
		0	22½	45	0	22½	45	0	22½	45	0	22½	45	0	22½	45
6x8	CFM			75			100			125			150			175
8x4	Throw			8			11			14			16			19
	NC			L			L			L			L			L
8x5	CFM			85			110			140			165			195
10x4	Throw			8			11			15			17			21
	NC			L			L			L			L			L
8x6	CFM			100			135			170			200			235
10x5	Throw			9			12			16			19			23
12x4	NC			L			L			L			L			20
14x4	CFM			115			155			195			235			270
	Throw			9			13			17			20			24
	NC			L			L			L			L			21
8x8 16x4	CFM			130			180			220			260			310
10x6	Throw			10			14			18			21			26
12x5	NC			L			L			L			L			22
12x6	CFM			150			200			250			300			350
14x5	Throw			11			15			19			22			27
18x4	NC			L			L			L			L			23
10x8 20x4	CFM			170			230			280			340			390
14x6	Throw			12			16			20			24			29
16x5	NC			L			L			L			L			24
10x10 20x5	CFM			210			280			350			400			480
12x8 24x4	Throw			14			18			22			26			31
16x6	NC			L			L			L			L			25
18x6	CFM			230			310			380			460			530
28x4	Throw			15			19			24			28			33
	NC			L			L			L			L			26
12x10 24x5	CFM			250			330			420			500			580
14x8 30x4	Throw			15			20			25			29			34
20x6	NC			L			L			L			L			27
12x12 18x8 36x4	CFM			300			400			500			600			700
14x10 24x6	Throw			16			21			27			31			36
16x8 28x5	NC			L			L			L			L			28
16x10 40x4	CFM			330			430			540			650			760
26x6	Throw			17			22			28			32			38
30x5	NC			L			L			L			L			28
14x14 24x8 40x5	CFM			410			540			680			820			950
16x12 32x6 48x4	Throw			18			24			31			36			43
20x10 34x6	NC			L			L			L			L			29
16x14 48x5	CFM			470			620			780			930			1090
18x12	Throw			19			26			33			49			46
36x6	NC			L			L			L			L			30
16x16 26x10	CFM			530			710			890			1070			1250
18x14 32x8	Throw			21			28			35			43			49
22x12 42x6	NC			L			L			L			L			32
18x16 30x10	CFM			600			800			1000			1200			1400
20x14 36x8	Throw			22			29			37			45			52
24x12 48x6	NC			L			L			L			L			32

Notes: (1) Additions and factors (listed below) have to be applied for varying blade settings and damper openings. (2) For sizes, CFM, blade settings or damper openings, etc., not listed below, interpolate as necessary.

Model RTDA—Register (Front Blade 0°)

NC—Add the following db to the NC obtained from Table for various rear blade settings.

Dual Velocity	300	400	500	600	700	800	900	1,000	1,200
Rear Blade 0°	2	2	2	1	1	1	1	1	1
Rear Blade 45°	12	12	12	11	11	10	10	10	10

NC—Add the following db to the NC obtained above for various damper openings.

Damper Opening	100%	75%	50%
db Add	0	10	22

P_t—Multiply the P_t listed in Table by the following F₂ factor for the wide open damper.

Blade Setting	0°	22½°	45°
Factor	1.70	1.50	1.10

T—Multiply the T listed in Table by the following F₁ factor for various blade settings.

Rear Blade Setting	0°	22½°	45°
Factor	1	.89	0.60

Figure 5.22 (a-3) Performance data of the double-deflection register with front blade set at 0° deflection.

Performance Data

Size	Blade Set° P _t	300			400			500			600			700	
		V = Duct Vel.			V = Duct Vel.			V = Duct Vel.			V = Duct Vel.			V = Duct Vel.	
		0	22½	45	0	22½	45	0	22½	45	0	22½	45	0	22½
		.016	.019	.034	.028	.033	.059	.043	.051	.091	.062	.074	.130	.084	.099
6x6	CFM	75			100			125			150			175	
8x4	Throw	7			10			12			14			17	
	NC	L			L			L			20			24	
8x5	CFM	85			110			140			165			195	
10x4	Throw	7			10			13			15			18	
	NC	L			L			L			21			25	
8x6	CFM	100			135			170			200			235	
10x5	Throw	8			11			14			17			20	
12x4	NC	L			L			L			22			26	
14x4	CFM	115			155			195			235			270	
	Throw	9			12			15			18			21	
	NC	L			L			L			22			26	
8x8 16x4	CFM	130			180			220			260			310	
10x6	Throw	9			12			16			19			23	
12x5	NC	L			L			L			23			27	
12x6	CFM	150			200			250			300			350	
14x5	Throw	10			13			17			20			24	
18x4	NC	L			L			L			24			28	
10x8 20x4	CFM	170			230			280			340			390	
14x6	Throw	11			14			18			21			24	
16x5	NC	L			L			20			24			28	
10x10 20x5	CFM	210			280			350			400			480	
12x8 24x4	Throw	12			16			20			23			28	
16x6	NC	L			L			20			25			29	
18x6	CFM	230			310			380			460			530	
28x4	Throw	13			17			21			24			29	
	NC	L			L			21			26			30	
12x10 24x5	CFM	250			330			420			500			580	
14x8 30x4	Throw	13			17			22			25			30	
20x6	NC	L			L			21			28			33	
12x12 18x8 36x4	CFM	300			400			500			600			700	
14x10 24x6	Throw	14			19			24			28			33	
16x8 28x5	NC	L			L			22			27			31	
16x10 40x4	CFM	330			430			540			650			760	
26x6	Throw	15			20			25			29			34	
30x5	NC	L			L			22			27			31	
14x14 24x8 40x5	CFM	410			540			680			820			950	
16x12 32x6 48x4	Throw	16			21			28			32			38	
20x10 34x6	NC	L			L			23			28			33	
16x14 48x5	CFM	470			620			780			930			1090	
18x12	Throw	17			23			29			34			41	
36x6	NC	L			L			23			28			33	
16x16 26x10	CFM	530			710			890			1070			1250	
18x14 32x8	Throw	19			25			31			38			44	
22x12 42x6	NC	L			L			24			29			34	
18x16 30x10	CFM	600			800			1000			1200			1400	
20x14 36x8	Throw	20			26			33			40			47	
24x12 48x6	NC	L			L			24			30			36	

Notes: (1) Additions and factors (listed below) have to be applied for varying blade settings and damper openings.
 (2) For sizes, CFM, blade settings or damper openings, etc., not listed below, interpolate as necessary.

Model RTDA—Register (Front Blade 22½°)

NC—Add the following db to the NC obtained from Table for various rear blade settings.

Duct Velocity	300	400	500	600	700	800	900	1,000
Rear Blade 0°	2	2	2	1	1	1	1	1
Rear Blade 45°	7	7	6	6	6	6	6	5

NC—Add the following db to the NC obtained above for various damper openings.

Damper Opening	100%	75%	50%
db Add	0	10	22

T—Multiply the T listed in Table by the following F₁ factor for various blade settings.

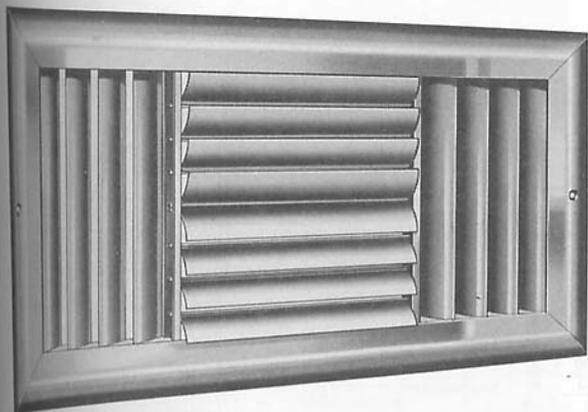
Rear Blade Setting	0°	22½°
Factor	1	.89

P_t—Multiply the P_t listed in Table by the following F₂ factor for the wide open damper.

Blade Setting	0°	22½°	45°
Factor	1.50	1.25	1.08

(a-4)

Figure 5.22 (a-4) Performance data of the register with front blade set at 22½°.



Four-way air supply register
(b-1)

REGISTER DIMENSIONS

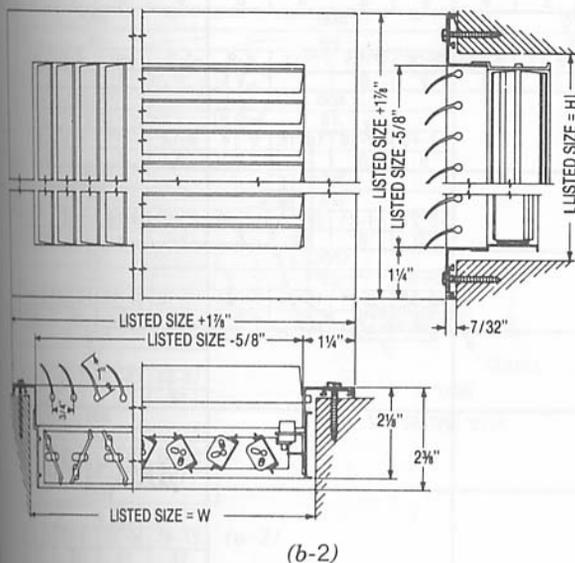


Figure 5.22 (b-1) Register with four sets of adjustable deflectors. This unit establishes a four-way air pattern and apportions equal quantities of primary air in four directions. (b-2) Section through the register showing construction of blades and volume dampers. (Courtesy of Carnes Co.)

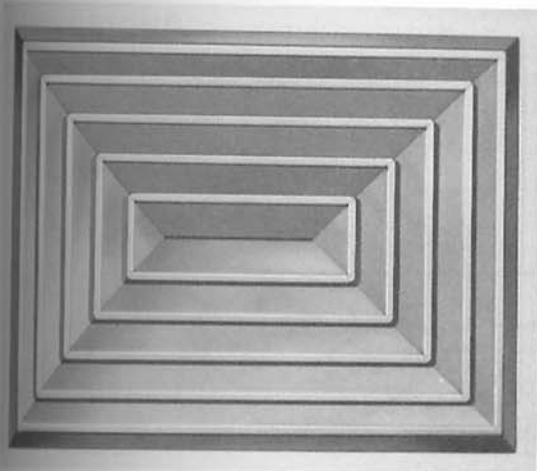
PERFORMANCE DATA

CURVED BLADE

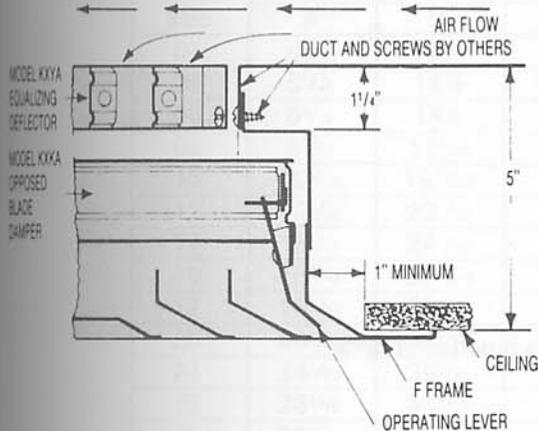
CFM	SIZE W X H (WIDE X HIGH)	10 x 4 8 x 5 6 x 6				12 x 4 10 x 5 8 x 6				14 x 5 12 x 6 8 x 8				24 x 6 18 x 8 14 x 10 12 x 12			
		1	2	3	4	1	2	3	4	1	2	3	4	1	2	3	4
50	Duct Velocity	200				165											
	Total Pressure	.02				.01											
	Throw	4-6	3-5	2-4	2-4	3-5	2-4	2-4	2-4								
	Min. Ceiling Height	8	8	8	8	8	8	8	8								
75	Duct Velocity	300				250											
	Total Pressure	.03				.03											
	Throw	5-8	5-8	4-7	4-7	4-7	3-6	3-5	3-5								
	Min. Ceiling Height	8	8	8	8	8	8	8	8								
100	Duct Velocity	400				330				200							
	Total Pressure	.05				.03				.02							
	Throw	7-11	6-10	5-8	5-8	6-10	5-8	4-6	4-6	5-7	4-6	3-5	3-5				
	Min. Ceiling Height	8	8	8	8	8	8	8	8	8	8	8	8				
125	Duct Velocity	500				415				250							
	Total Pressure	.08				.06				.02							
	Throw	9-14	8-12	7-16	6-10	7-11	6-10	6-10	5-11	5-7	4-6	4-6	3-5				
	Min. Ceiling Height	8	8	8	8	8	8	8	8	8	8	8	8				
150	Duct Velocity	600				450				300				150			
	Total Pressure	.12				.07				.03				.01			
	Throw	12-18	10-16	8-12	7-11	9-14	8-12	7-11	6-10	7-11	5-7	5-7	4-6	4-6	3-5	3-5	3-5
	Min. Ceiling Height	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8	8
200	Duct Velocity	800				670				400				200			
	Total Pressure	.18				.16				.05				.02			
	Throw	14-20	12-18	10-16	9-14	11-16	9-14	8-12	8-12	9-14	7-11	7-11	6-9	5-8	4-6	4-6	4-6
	Min. Ceiling Height	9	8	8	8	9	8	8	8	9	8	8	8	9	8	8	8
300	Duct Velocity									600				300			
	Total Pressure									.12				.03			
	Throw									13-19	12-18	10-16	9-14	8-12	7-11	6-9	5-7
	Min. Ceiling Height									9	8	8	8	9	8	8	8
400	Duct Velocity									800				400			
	Total Pressure									.23				.05			
	Throw									18-27	14-21	13-19	12-18	11-16	9-14	8-12	7-11
	Min. Ceiling Height									10	8	8	8	10	8	8	8
500	Duct Velocity									1000				500			
	Total Pressure									.37				.08			
	Throw									20-30	18-27	16-24	14-21	13-19	12-18	10-16	9-14
	Min. Ceiling Height									10	9	8	8	10	9	8	8
600	Duct Velocity													600			
	Total Pressure													.12			
	Throw													16-24	13-19	12-18	11-17
	Min. Ceiling Height													12	10	8	8
800	Duct Velocity													800			
	Total Pressure													.23			
	Throw													22-33	18-27	15-22	14-21
	Min. Ceiling Height													13	10	9	8
1000	Duct Velocity													1000			
	Total Pressure													.37			
	Throw													27-40	20-30	18-27	16-24
	Min. Ceiling Height													15	12	10	9

(b-3)

Figure 5.22 (b-3) Performance data of the four-way register.



Rectangular four-way blow ceiling diffuser
(a-1)



F FRAME

(a-2)



FOUR-WAY BLOW

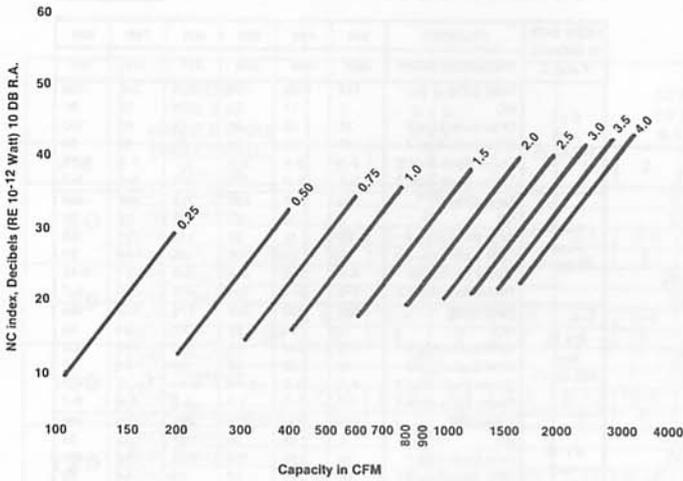
NECK SIZE IN INCHES Y and Z	VELOCITY	300	400	500	600	700	800
	PRESSURE DROP	.028	.049	.080	.111	.145	.195
6 x 9 Area .375 Sq. Ft.	Total CFM	113	150	190	225	260	300
	NC	L	14	20	25	29	34
	CFM Each Side Y	38	50	65	75	85	100
	CFM Each Side Z	19	25	30	38	45	50
	Throw Each Side Y	2-3	3-4	5-6	6-7	7-8	9-10
	Throw Each Side Z	1-2	2-3	3-4	4-5	5-6	6-7
6 x 12 Area .50 Sq. Ft.	Total CFM	150	200	250	300	350	400
	NC	L	13	20	25	29	33
	CFM Each Side Y	56	75	93	112	131	150
	CFM Each Side Z	19	25	32	38	44	50
	Throw Each Side Y	4-5	5-6	6-7	7-9	9-11	10-12
	Throw Each Side Z	1-2	2-3	3-4	4-5	5-6	6-7
6 x 15 Area .625 Sq. Ft.	Total CFM	188	250	312	375	438	500
	NC	L	15	21	27	31	35
	CFM Each Side Y	75	100	124	150	175	200
	CFM Each Side Z	19	25	32	38	44	50
	Throw Each Side Y	5-7	6-8	8-10	9-11	11-13	12-14
	Throw Each Side Z	1-2	2-3	3-4	4-5	5-6	6-7
6 x 18 Area .75 Sq. Ft.	Total CFM	225	300	375	450	525	600
	NC	L	15	21	27	31	35
	CFM Each Side Y	94	125	156	188	218	250
	CFM Each Side Z	19	25	32	38	44	50
	Throw Each Side Y	6-8	7-9	9-11	10-12	11-13	12-15
	Throw Each Side Z	1-2	2-3	3-4	4-5	5-6	6-7
6 x 21 Area .875 Sq. Ft.	Total CFM	263	350	438	525	612	700
	NC	L	16	23	28	33	37
	CFM Each Side Y	112	150	187	224	262	300
	CFM Each Side Z	19	25	32	38	44	50
	Throw Each Side Y	7-9	8-10	10-12	11-13	12-15	13-16
	Throw Each Side Z	1-2	2-3	3-4	4-5	5-6	6-7
6 x 24 Area 1.00 Sq. Ft.	Total CFM	300	400	500	600	700	800
	NC	L	17	23	28	33	37
	CFM Each Side Y	131	175	218	262	306	350
	CFM Each Side Z	19	25	32	38	44	50
	Throw Each Side Y	8-10	9-11	11-13	12-15	13-16	14-18
	Throw Each Side Z	1-2	2-3	3-4	4-5	5-6	6-7
9 x 12 Area .75 Sq. Ft.	Total CFM	225	300	375	450	525	600
	NC	L	15	21	27	31	35
	CFM Each Side Y	70	94	118	141	165	188
	CFM Each Side Z	42	56	70	84	98	112
	Throw Each Side Y	4-6	5-7	7-9	8-10	10-12	11-14
	Throw Each Side Z	2-3	3-4	4-5	6-8	7-9	8-10
9 x 15 Area .938 Sq. Ft.	Total CFM	282	375	470	563	656	750
	NC	L	16	23	28	33	37
	CFM Each Side Y	99	132	165	198	230	263
	CFM Each Side Z	42	56	70	84	98	112
	Throw Each Side Y	5-7	6-8	8-10	10-13	12-15	13-16
	Throw Each Side Z	2-3	3-4	5-7	6-8	7-9	8-10
9 x 18 Area 1.125 Sq. Ft.	Total CFM	338	450	562	675	788	900
	NC	L	16	24	29	34	37
	CFM Each Side Y	127	169	211	254	296	338
	CFM Each Side Z	42	56	70	84	98	112
	Throw Each Side Y	6-9	7-10	9-12	11-14	12-15	14-18
	Throw Each Side Z	2-3	3-4	5-7	6-8	7-9	8-10
9 x 21 Area 1.31 Sq. Ft.	Total CFM	393	524	655	786	917	1050
	NC	L	17	24	29	34	38
	CFM Each Side Y	155	206	258	309	360	413
	CFM Each Side Z	42	56	70	84	98	112
	Throw Each Side Y	6-8	7-9	10-13	12-15	13-17	15-19
	Throw Each Side Z	2-3	3-4	5-7	6-8	7-9	8-10
9 x 24 Area 1.50 Sq. Ft.	Total CFM	450	600	750	900	1050	1200
	NC	L	18	25	30	35	39
	CFM Each Side Y	183	244	305	366	427	488
	CFM Each Side Z	42	56	70	84	98	112
	Throw Each Side Y	9-12	10-13	12-15	13-16	14-18	16-20
	Throw Each Side Z	2-3	3-4	5-7	6-8	7-9	8-10
12 x 15 Area 1.25 Sq. Ft.	Total CFM	375	500	625	750	875	1000
	NC	L	18	25	30	35	39
	CFM Each Side Y	113	150	188	225	263	300
	CFM Each Side Z	75	100	125	150	175	200
	Throw Each Side Y	5-7	6-8	8-10	10-13	12-15	13-16
	Throw Each Side Z	4-6	5-7	7-9	9-11	10-13	11-14
12 x 18 Area 1.50 Sq. Ft.	Total CFM	450	600	750	900	1050	1200
	NC	L	18	25	30	35	39
	CFM Each Side Y	150	200	250	300	350	400
	CFM Each Side Z	75	100	125	150	175	200
	Throw Each Side Y	5-7	6-8	8-11	12-15	13-17	15-19
	Throw Each Side Z	4-6	5-7	7-9	9-11	10-13	11-14

(a-3)

Figure 5.23 (a-1) Typical rectangular four-way blow steel ceiling diffuser. (a-2) Mounting detail with an F-frame, in a hung ceiling. Deflector and damper are also shown. (a-3) Diffuser performance data as a function of duct air velocity. (Courtesy of Carnes Company.)

SOUND RATINGS

Figure 1: Noise rating chart for Model SK Diffusers without Damper



To use Noise Rating Chart, Figure 1, determine the square footage of the diffuser neck area from the chart in Figure 5.23 (a-1). Follow the vertical CFM line on the chart until it intersects the diagonal square footage line. Read the NC Level on the left hand side of the chart. For the effect of dampering, use chart Figure 2.

Figure 2: Effect of Damper on NC Index

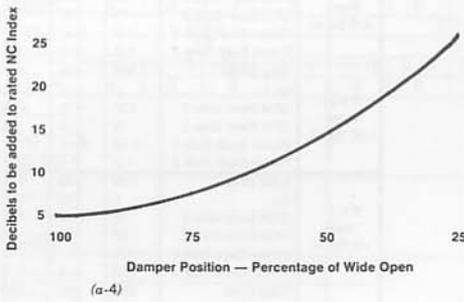


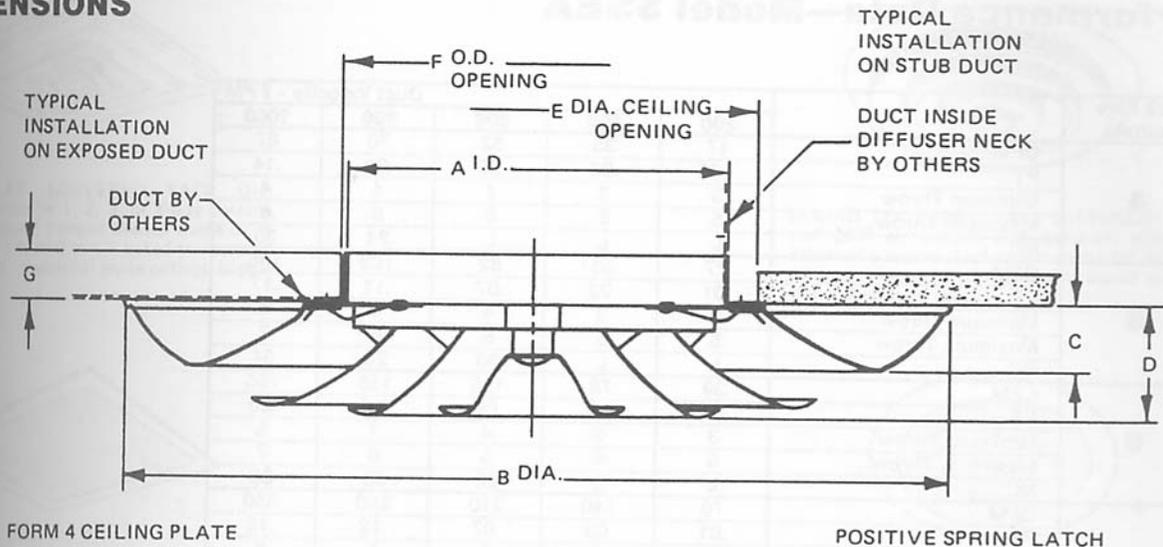
Figure 5.23 (a-4) Noise ratings of the four-way ceiling diffuser as a function of neck area.



Round steel fixed pattern ceiling diffuser (b-1)

Figure 5.23 (b-1) Fixed-pattern round steel ceiling diffuser, with four cones. Similar units are available with adjustable patterns. (Courtesy of Carnes Company.)

DIMENSIONS



Size	Form 4						
	A	B	C	D	E	F	G
04	4 ¹ / ₁₆	14 ¹ / ₂	1 ³ / ₁₆	1 ¹⁵ / ₁₆	7 ¹ / ₄	4 ¹ / ₈	3 ¹ / ₄
05	5 ¹ / ₁₆	14 ¹ / ₂	1 ³ / ₁₆	1 ¹⁵ / ₁₆	7 ¹ / ₄	6 ¹ / ₈	1 ³ / ₈
06	6 ¹ / ₁₆	14 ¹ / ₂	1 ³ / ₁₆	1 ¹⁵ / ₁₆	7 ¹ / ₄	6 ¹ / ₈	1
08	8 ¹ / ₁₆	17 ⁹ / ₁₆	1 ³ / ₈	2 ³ / ₈	9 ¹ / ₄	8 ¹ / ₈	1
10	10 ¹ / ₁₆	19 ¹³ / ₁₆	1 ³ / ₁₆	1 ⁷ / ₈	11 ¹ / ₄	10 ³ / ₈	1
12	12 ¹ / ₁₆	22 ¹ / ₁₆	1 ³ / ₈	2 ⁵ / ₁₆	13 ¹ / ₄	12 ³ / ₈	1
14	14 ¹ / ₁₆	24 ¹ / ₁₆	1 ¹ / ₂	2 ⁵ / ₁₆	15 ¹ / ₄	14 ³ / ₈	1
16	16 ¹ / ₁₆	28 ¹³ / ₁₆	2 ¹ / ₈	2 ³ / ₄	17 ¹ / ₄	16 ³ / ₈	1
18	18 ¹ / ₁₆	32 ¹ / ₁₆	2 ¹ / ₈	3 ¹ / ₈	19 ¹ / ₄	18 ³ / ₈	1
20	20 ¹ / ₁₆	33 ⁹ / ₁₆	2 ¹ / ₄	3 ¹ / ₈	21 ¹ / ₄	20 ³ / ₈	1
24	24 ¹ / ₁₆	39 ⁵ / ₁₆	2 ⁵ / ₈	3 ¹¹ / ₁₆	25 ¹ / ₄	24 ³ / ₈	1
28	28 ¹ / ₁₆	48 ⁹ / ₁₆	2 ¹¹ / ₁₆	4 ¹ / ₂	29 ¹ / ₄	28 ³ / ₈	1
32	32 ¹ / ₁₆	60 ¹³ / ₁₆	2 ⁷ / ₈	4 ¹⁵ / ₁₆	33 ¹ / ₄	32 ³ / ₈	1
36	36 ¹ / ₁₆	60 ¹³ / ₁₆	2 ⁷ / ₈	4 ¹⁵ / ₁₆	37 ¹ / ₄	36 ³ / ₈	1

Dimensions are given in inches.

(b-2)

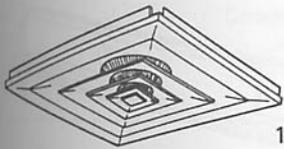
Figure 5.23 (b-2) Installation detail for diffuser mounting directly on a stub duct. Units can also be mounted on a damper, a splitter/damper, a radial deflector, an equalizing damper or a direction changer as required by the particular installation.

Performance Data—Model SSEA

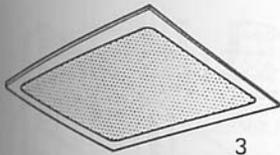
Neck Size (Inches)		Duct Velocity - FPM				
		200	400	600	800	1000
4	CFM	17	35	52	70	87
	PT	.005	.02	.05	.09	.14
	Minimum Throw	3	3	4	4	4
	Maximum Throw	5	5	5	6	6
	NC	L	L	L	24	30
5	CFM	27	55	82	109	136
	PT	.01	.03	.07	.11	.17
	Minimum Throw	3	3	4	4	4
	Maximum Throw	5	5	6	6	7
	NC	L	L	20	28	34
6	CFM	39	78	115	155	195
	PT	.01	.04	.08	.13	.20
	Minimum Throw	3	3	4	4	5
	Maximum Throw	5	6	6	6	7
	NC	L	L	22	30	36
8	CFM	70	140	210	280	350
	PT	.01	.03	.07	.12	.18
	Minimum Throw	4	4	5	5	6
	Maximum Throw	6	6	8	8	10
	NC	L	L	27	34	40
10	CFM	109	218	325	435	545
	PT	.01	.06	.12	.24	.36
	Minimum Throw	4	5	6	7	7
	Maximum Throw	7	9	11	13	14
	NC	L	L	29	37	42
12	CFM	158	315	475	630	785
	PT	.01	.04	.10	.20	.31
	Minimum Throw	5	6	6	7	10
	Maximum Throw	8	10	12	14	18
	NC	L	21	31	39	45
14	CFM	215	430	645	855	1070
	PT	.02	.05	.11	.20	.31
	Minimum Throw	6	8	11	13	18
	Maximum Throw	9	13	15	19	23
	NC	L	22	32	40	46
16	CFM	280	660	840	1120	1400
	PT	.01	.07	.14	.26	.41
	Minimum Throw	7	10	13	17	19
	Maximum Throw	10	14	17	22	25
	NC	L	22	33	41	47

(b-3)

Figure 5.23 (b-3) Typical performance data for the illustrated diffuser.



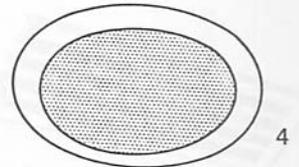
RECTANGULAR LOUVERED FACE DIFFUSER: Available in 1, 2, 3, or 4-way pattern, steel or aluminum. Flanged overlap frame or inserted in 2 X 2 ft or 2 X 4 ft baked enamel steel panel to fit tile modules of lay-in ceilings. Supply or return.



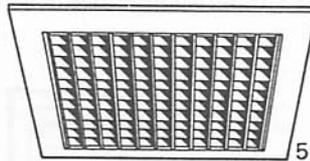
RECTANGULAR PERFORATED FACE DIFFUSER: Available in 1, 2, 3, or 4-way pattern, steel or aluminum. Flanged overlap frame or 2 X 2 ft and 2 X 4 ft for replacing tile of lay-in ceiling can be used for supply or return air.



ROUND LOUVERED FACE DIFFUSER: Normal 360° air pattern with blank-off plate for other air patterns. Surface mounting for all type ceilings. Normally of steel with baked enamel finish. Supply or return.



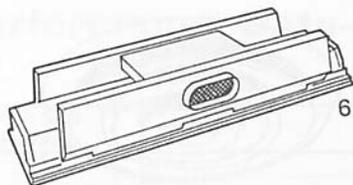
ROUND PERFORATED FACE DIFFUSER: Normal 360° air pattern with blank-off plate for other air patterns. Steel or aluminum. Flanged overlap frame for all type ceilings. Can be used for supply or return air.



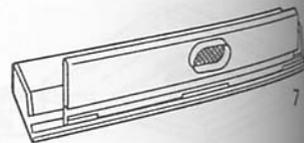
LATTICE TYPE RETURN: All aluminum square grid type return grille for ceiling installation with flanged overlap frame or of correct size to replace tile.

(c)

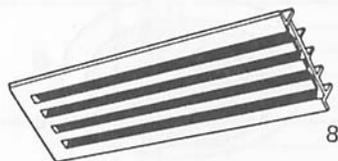
Figure 5.23 (c) Common air distribution outlets and their principal characteristics. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., p. 628, 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)



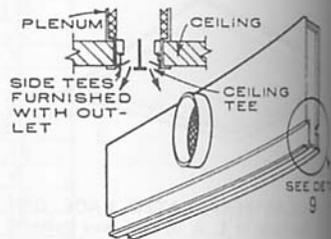
SADDLE TYPE LUMINAIRE AIR BOOT: Provides air supply from both sides of standard size luminaires. Maximum air delivery (total both sides) approximately 150 to 170 cfm for 4 ft long luminaire.



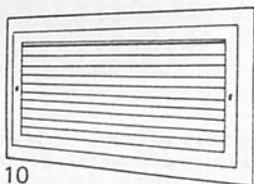
SINGLE SIDE TYPE LUMINAIRE AIR BOOT: Provides air supply from one side of standard size luminaires. Maximum air delivery approximately 75 cfm for 4 ft long luminaire.



LINEAR DIFFUSER: Extruded aluminum, anodized, duranodic, or special finishes, one way or opposite direction or vertical down air pattern. Any length with one to eight slots. Can be used for supply or return and for ceiling, sidewall, or cabinet top application.



INTEGRATED PLENUM TYPE OUTLET FOR "T" BAR CEILINGS: Slot type outlet, one or two way opposite direction air pattern. Available in 24, 36, 48, and 60 in. lengths. Register integrates with "T" bar. Approximately 150 to 175 cfm for 4 ft long, two slot unit.



SIDEWALL OR DUCT MOUNTED REGISTER: Steel or aluminum for supply or return. Adjustable horizontal and vertical deflection. Plaster frame available. Suitable for long throw and high air volume.

(c)

Comfort Conditions in the Occupied Zone:

Winter: 68–74°F, 40 Fpm max.

Summer: 73–79°F DB, 50 Fpm max.

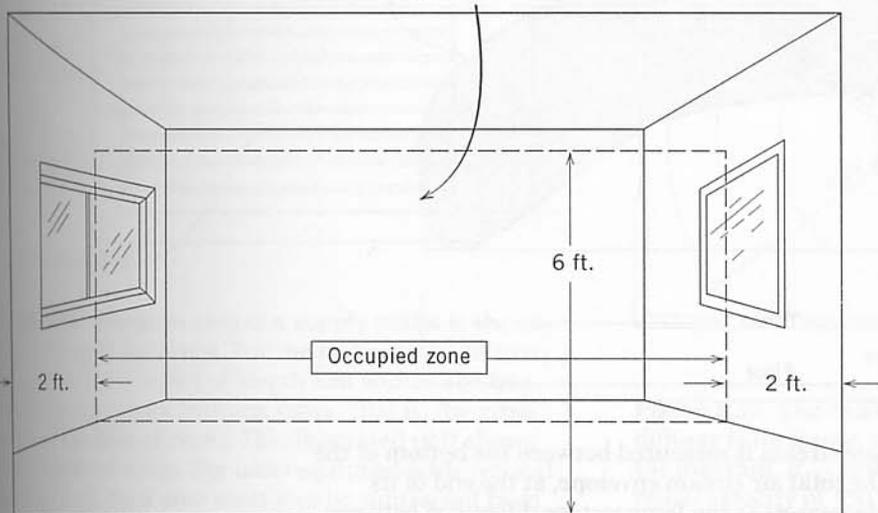
Maximum vertical ΔT is 5°F

Figure 5.24 The occupied zone of a room is defined as the volume 2 ft from each wall and 6 ft high off the floor. In this space, design comfort conditions must be maintained. A maximum vertical temperature differential of 5 F° between ankle height at 4 in. above finished floor (AFF) and neck height of 67 in. AFF should be maintained year round.

velocity to acceptable limits. Further, within this area it must at least begin to mix with room air, in order to transfer its heat (or coolness) without leaving stagnant spaces.

5.16 Outlet Characteristics

The operating characteristics of a register or diffuser describe the flow of air from the unit and how this supply air mixes with the room air. The following glossary defines the terms that describe these actions in two categories: outlet performance and air mixing.

a. Outlet Performance

(1) *Drop*. When the supply air is colder than the room air, it drops as it travels across the space. The performance criterion known as drop is the vertical distance that the lower edge of the air pattern falls, between the out-

let and the end of its throw. See Figure 5.25a. See also the term *Rise*.

- (2) *Face velocity* (outlet velocity). The average air velocity coming out of the outlet, measured in the plane of the opening. Since the velocity will vary over the face of the outlet, multiple measurements must be taken and averaged.
- (3) *Free area*. The open area of a register or grille through which air can pass, unobstructed. The free area, which varies between 60 and 90% of the gross area, determines the face velocity and pressure drop of the outlet. See Figure 5.26.
- (4) *Gross area*. The area of the inside dimensions of the frame, that is, the grille area, not including the device frame. See Figure 5.26.
- (5) *Isothermal jet*. An air jet at the same temperature as the room air.
- (6) *Noise Criterion (NC)*. An indication of the background noise level acceptable in a specific space. The NC rating of an outlet is determined by plotting the decibel values of the noise it creates on a special graph. The outlet

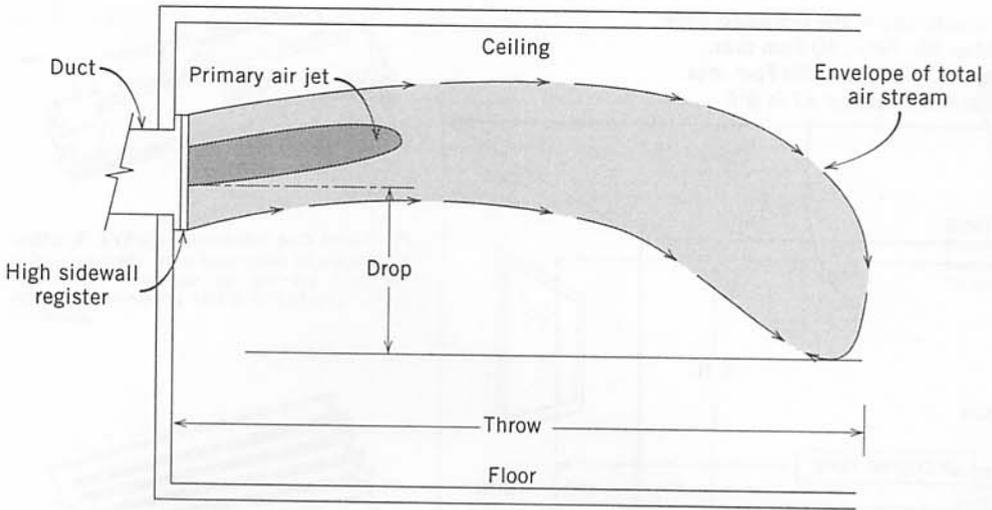


Figure 5.25 (a) The drop of a cold air stream is measured between the bottom of the primary air jet and the bottom of the total air stream envelope, at the end of its throw. Drop increases (and throw decreases) as the temperature difference between room air and incoming air increases, simply because the entering colder air is heavier. See also Figure 5.37.

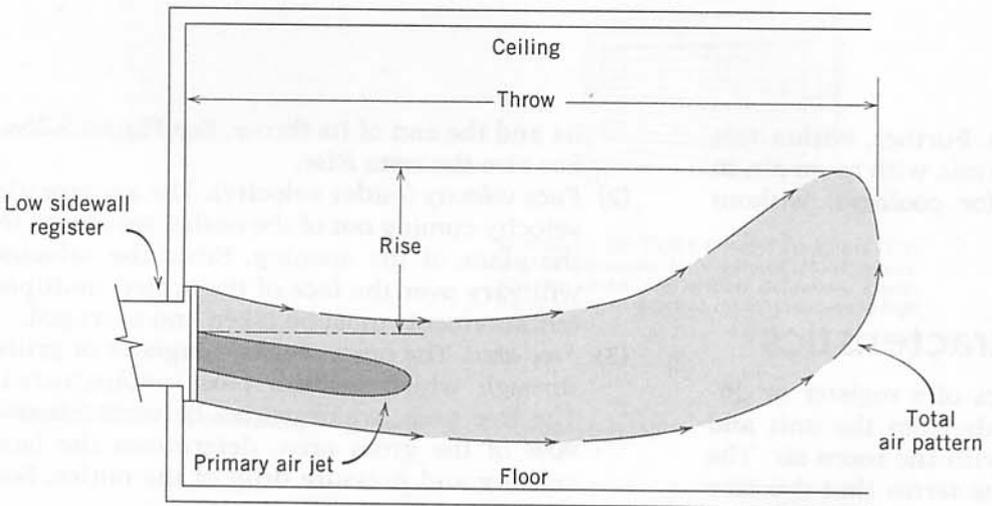


Figure 5.25 (b) The rise of a warm air stream is measured between the top of the primary air jet and the top of the total air pattern, at the end of its throw. Rise increases (and throw decreases) as the temperature difference between incoming air and room air increases, because the incoming warmer air is lighter. See also Figure 5.39.

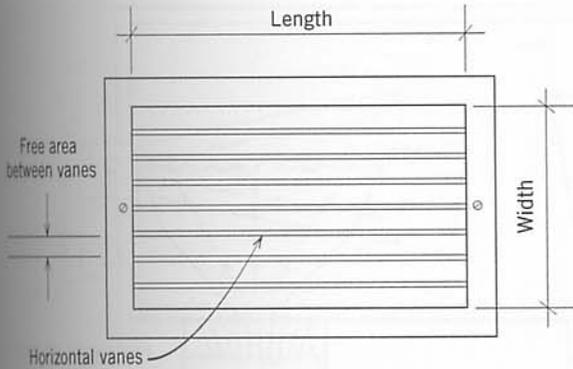
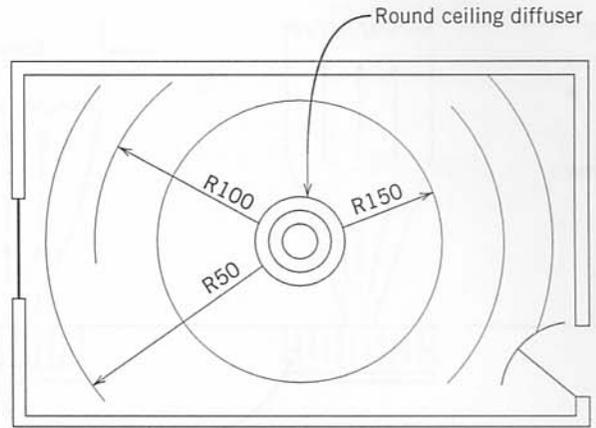


Figure 5.26 The gross area of a supply outlet is the entire area inside the frame. For the rectangular diffuser shown, it is the product of length and width. The free area is the open area between vanes, that is, the gross area less the area of vanes. The illustrated unit shows only horizontal vanes. For units equipped with vertical vanes as well, their area must also be subtracted from gross area to obtain net, or free, area.



PLAN VIEW

Figure 5.27 The radius of diffusion of a round ceiling diffuser is its throw, to a specified terminal velocity. In the diagram, R_{150} is the throw of the air stream to a terminal velocity of 150 fpm. R_{100} and R_{50} are the (radial) throws to terminal velocities of 100 and 50 fpm, respectively.

NC rating is then compared to the maximum permissible NC of that space, to determine whether the outlet is usable in that space.

- (7) *Radius of diffusion.* The horizontal distance that an air stream travels after leaving a ceiling outlet before the maximum velocity drops to a specified level, usually between 50 and 200 fpm. This is the term used for the throw of a ceiling diffuser that discharges radially. See Figure 5.27.
- (8) *Rise.* When the entering supply air is warmer than the room air, it tends to rise because it is also lighter than the room air. Rise is the vertical distance that the upper edge of the total air pattern rises, between the outlet and the end of its throw. See Figure 5.25(b). See also *Drop*.
- (9) *Spread.* A measure of the dispersion of an air stream from a wall or floor outlet caused by the vertical vanes in the face of the outlet. As the spread increases, the throw decreases. When the vanes are set for zero deflection, the air pattern has approximately the same dispersion and throw as a free discharge. See Figure 5.28.
- (10) *Surface effect.* See Figure 5.29. When an outlet is located within about 1 ft of a room surface, the motion of the air stream creates a low

pressure area between the air stream and the room surface (floor, wall or ceiling). This forces the air stream against the surface, reducing air entrainment, increasing throw, and decreasing drop. This effect is useful in perimeter heating and cooling systems, to blanket walls with conditioned air without causing drafts.

- (11) *Temperature differential.* The difference in temperature between supply air and average room DB temperature.
- (12) *Terminal velocity.* The maximum air stream velocity at the end of the throw. See Figure 5.30.
- (13) *Throw.* See Figure 5.30. The horizontal (or vertical) axial distance an air stream travels from the outlet until the maximum stream velocity drops to a specified minimum, usually between 50 and 200 fpm. Throw data listed for outlets in manufacturers' catalogs are for isothermal air streams discharging into an open, clear, unobstructed space. It, therefore, does not take into account surface effect for high sidewall outlets or floor perimeter outlets, or drop for cold air, or rise for warm air.
- (14) *Vane.* A portion of the register face designed to direct the air stream. See Figure 5.26.

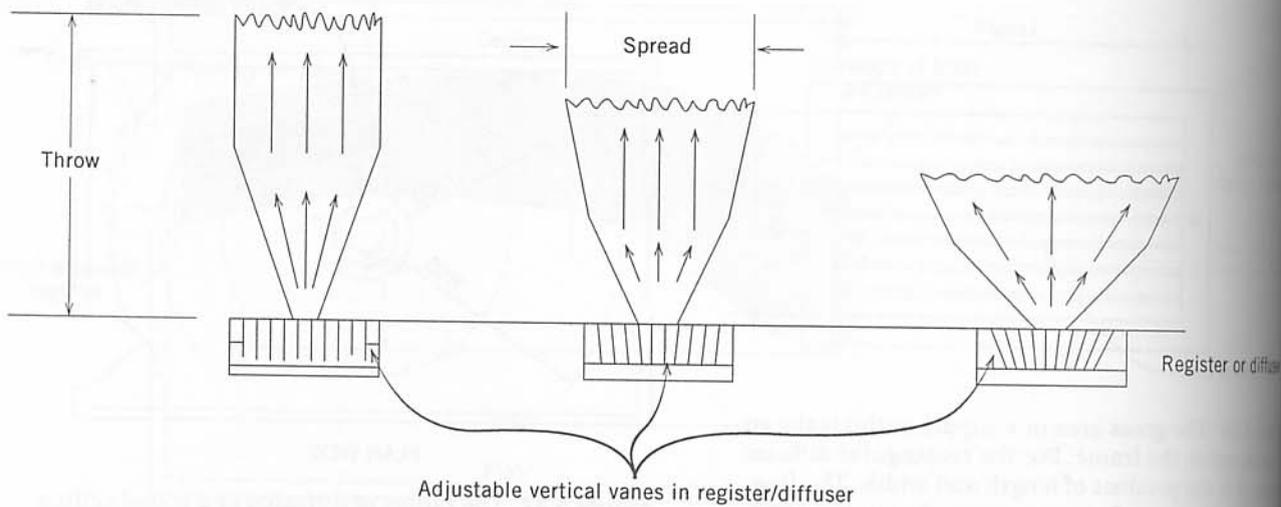


Figure 5.28 Vertical vanes in the register face can be adjusted to vary the width or spread of the air stream pattern. As the stream widens, its length, or throw, shortens.

- (15) *Velocity*. Generally refers to the maximum axial velocity of an air stream. Peripheral velocity is considerably lower.
- (16) *Vertical temperature gradient*. See Figure 5.24. Unless otherwise specified, the temperature differential between air at 4 in. above the floor (ankle height) and air at 67 in. above the floor (neck height). In spaces where people are normally seated, such as an auditorium, neck height is taken to be 42 in. AFF (above finished floor). These gradients occur when there is incomplete mixing of primary and secondary air and leads to areas of stagnant air.

b. Characteristics of the Air Stream

- (1) *Diffusion*. The distribution in a space, of supply air from an outlet, and its mixing with room air.
- (2) *Entrainment; entrained air*. The action by which room air moves into, and mixes with, the stream of primary air from the supply outlet; the air so entrained. See also *Induction*. See Figure 5.30.
- (3) *Induction*. The process by which room air is drawn to an outlet by aspiration of the primary air stream. The combined air then constitutes the air stream. See Figure 5.30.
- (4) *Envelope*. The outer boundary of the moving air stream, where motion is caused by the primary air jet. It does not include air moving from convective air currents.
- (5) *Primary air*. The air delivered by the supply duct to the outlet. See Figure 5.30.
- (6) *Primary air pattern*. The shape of the air stream from the supply outlet, where the air velocity at the outer edges of the air stream envelope is not less than 150 fpm. The air in the envelope consists of primary air, induced room air and entrained room air. See Figure 5.30.
- (7) *Secondary air; room air*. The amount of secondary (room) air in the total air pattern is usually 10–20 times the amount of primary air. Also refers to the room air drawn into the primary air stream.
- (8) *Stagnant air; stagnant zone*. Still room air that is substantially unaffected by the primary air stream. The zone or area of a space contains stagnant air. See Figure 5.31. A condition caused by insufficient mixing of primary and secondary air. Air motion in the stagnant zone is caused by convective currents only.
- (9) *Stratification*. The formation of areas of stagnant air that are unaffected by the primary air stream. The air in the room is, therefore, stratified, with a moving strata (layer) and a stagnant (still) layer. Air motion in the stagnant layer is caused only by natural convection.

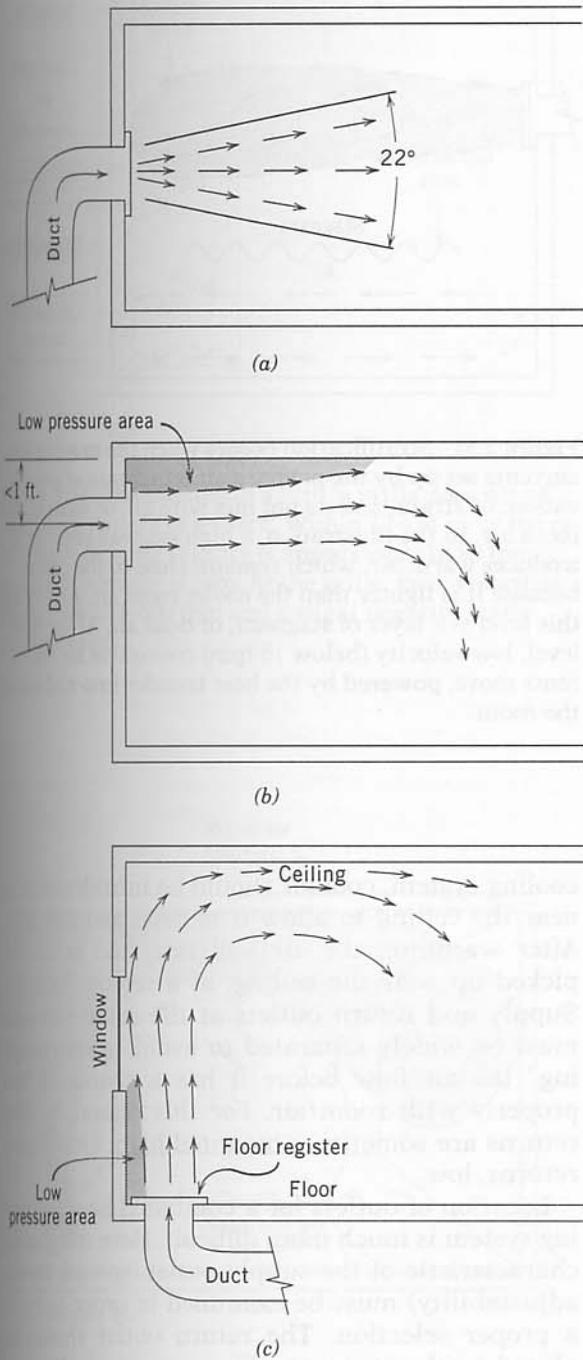


Figure 5.29 (a) When an isothermal air jet discharges into an open unobstructed space, it expands in a 22° cone. (b) If the air stream is within about 1 ft of a wall or ceiling, a low pressure area is formed between the jet and the boundary. (This area is shown shaded.) This forces the air stream against the boundary (wall, ceiling) and increases the air stream's throw. This action is called the surface effect. The surface effect can be induced in air streams from sidewall outlets somewhat farther from the ceiling by setting the register's horizontal vanes so that the air stream strikes the ceiling at a glancing angle. This will produce the effect illustrated and will lengthen the throw. Too sharp an angle of incidence will cause turbulence that will shorten the throw. (c) By placing a floor heating register within 1 ft of a window wall, the warm air jet is forced against the window by the surface effect. There it mixes with the cold air sliding down from the window. The warm mixture rises as shown.

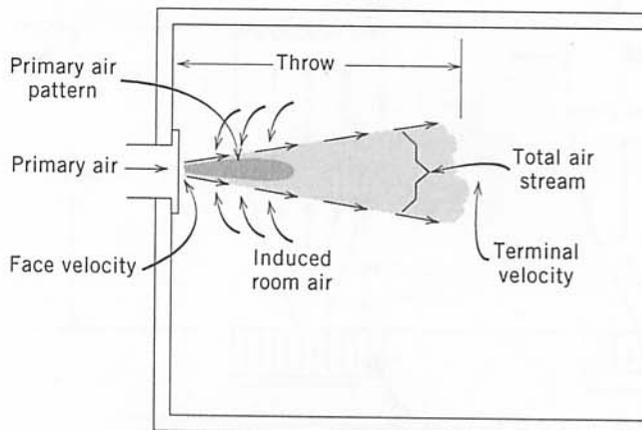


Figure 5.30 Throw length of an unobstructed air stream is measured from the register to the point at which maximum air stream velocity has dropped to a specified level—usually between 75 and 200 fpm. Primary air mixes with room air by a process of induction to form the total air stream. Room air induced into the stream is called entrained air.

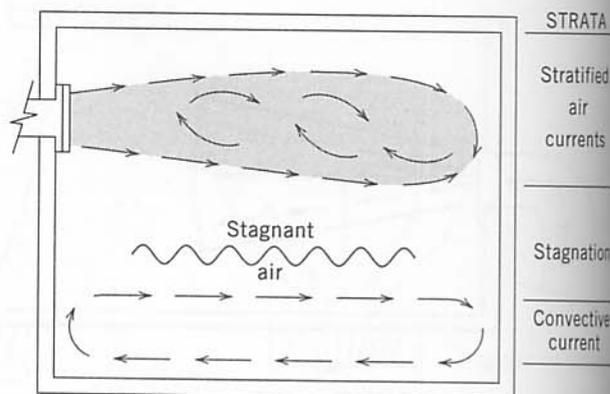


Figure 5.31 Stratification occurs when the moving air currents set up by the primary air jet remain at one elevation, or strata, and do not mix with all, or most of, the room air. In the illustration, a high sidewall register introduces warm air, which remains close to the ceiling because it is lighter than the cooler room air. Just below this level is a layer of stagnant, or dead air. At the floor level, low velocity (below 15 fpm) convective air currents move, powered by the heat transfer into and out of the room.

tion currents and is typically less than 20 fpm. See Figure 5.31.

(10) *Supply air.* See *Primary air.*

(11) *Total air pattern.* The envelope of all the air moving in a space as a result of the supply air stream. It does not include air moving in convective currents.

5.17 Outlet Selection, Location and Application

Outlets are selected and located according to application (heating, cooling or both), duct location (overhead, floor level), type of heating/cooling system (perimeter, radial, extended plenum, etc.) and the characteristics of the supply system including supply air velocity and temperature differentials. The simplest selection is for an all-heating or all-cooling system. Heated air, being lighter than the room air, tends to rise. Therefore, for an all-heating system, warm air should be supplied at or near the floor level to allow it to rise; return air, which has cooled and dropped, can be picked up near the floor level across the room. Conversely, cooled air is heavier than the room air. Therefore, in an all-

cooling system, cool air should be introduced at or near the ceiling to allow it to drop and circulate. After warming, the air will rise, and it can be picked up near the ceiling, at a remote location. Supply and return outlets at the same elevation must be widely separated to avoid “short-circuiting” the air flow before it has a chance to mix properly with room air. For this reason, heating returns are sometimes mounted high, and cooling returns, low.

Location of outlets for a combined heating/cooling system is much more difficult. Here the specific characteristic of the supply outlet (spread, throw, adjustability) must be examined in order to make a proper selection. The return outlet should be placed in the stagnant air area so as to increase circulation. (Note, however, that return outlets do not “draw” and, therefore, will not affect the supply air pattern. See Figure 5.32.) When using low perimeter supply outlets, stagnant air will concentrate near the ceiling during cooling [Figure 5.33(b)] and near the floor during heating [Figure 5.33(a)]. Since it is impossible to satisfy both conditions with a single return outlet, two choices remain. Either use a high wall return grille because the stagnant air problem is more severe during cooling or use a high and a low return register on a

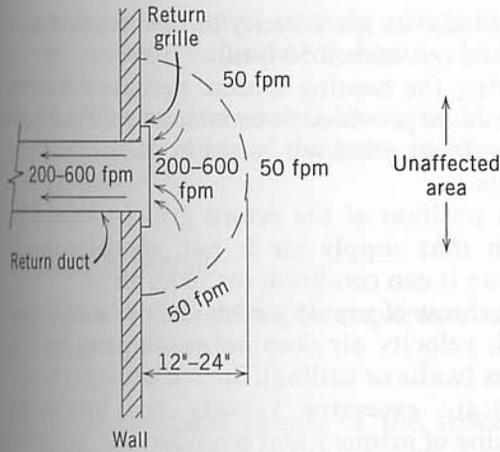


Figure 5.32 The location of a return grille does not affect the air pattern in a space. Within 12–24 in. of the return grille face, air velocity is approximately 50 fpm and static pressure is zero. At the grille face, velocities range from 200–600 fpm and a slight negative static pressure exists.

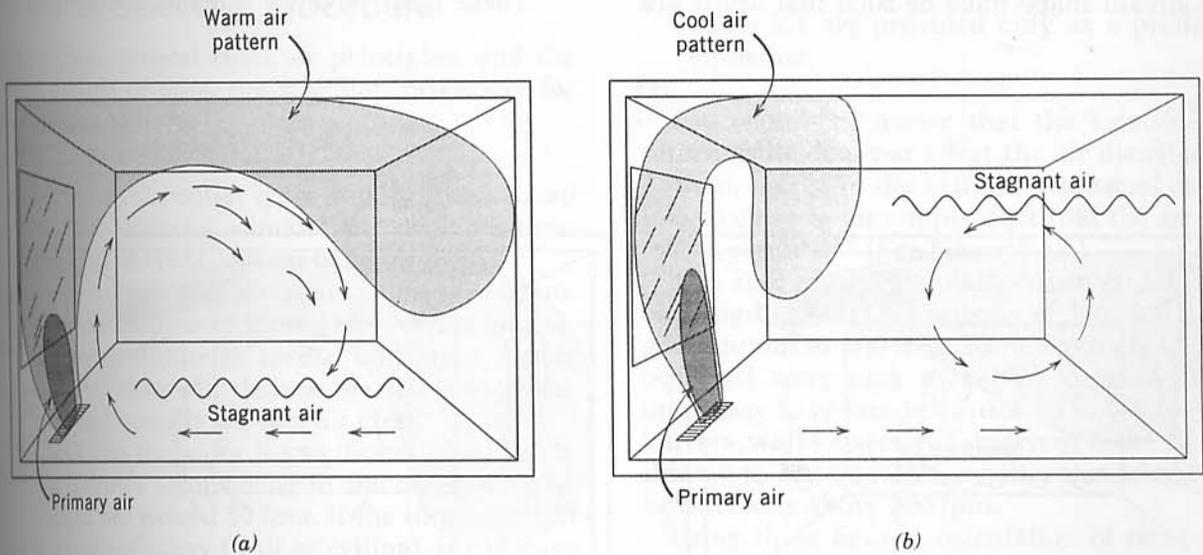


Figure 5.33 (a) Warm primary air from the spread floor register combines with cold air coming off the window and induced room air, to form a strong circular current. The room is free of drafts and uniformly heated. Linear diffusers and low sidewall diffusers will produce the same result. See Figure 5.35. (b) The cooling air pattern falls back on itself near the window, resulting in inadequate mixing of primary air and room air. The higher the throw, the better the room cooling. The throw should at least reach the ceiling to achieve satisfactory room cooling.

single duct, with volume dampers. During cooling, the upper register is opened; during the heating season, the lower register is opened and the upper is closed. Since two registers are considerably more expensive than a single grille, this solution is seldom used. Furthermore, the occupant must be sufficiently knowledgeable to perform this function. In practice, most designers would use a single high outlet. See Figure 5.1.

In combined heating/cooling systems using high sidewall outlets (Figure 5.34), the largest stagnant air pool during heating occurs at floor level. This then would be the location of the return outlet, which is also effective for cooling.

a. Principles of Outlet Selection

The general rules governing the selection of outlets are these:

- (1) The supply air stream shape should be selected to mix with room air so that no stratification and stagnant air remains in the occupied zone.
- (2) The location of the supply outlet(s) and the air stream shape must be such that drafts are avoided (i.e., air velocity in the occupied zone should not exceed 50 fpm).
- (3) During the heating season, perimeter heating should be provided to counteract cold air dropping from windows, outside doors and cold walls.
- (4) The position of the return grille(s) should be such that supply air is not "short-circuited" before it can condition the room air.
- (5) The throw of supply outlets should be such that high velocity air does not reach room boundaries (walls or ceiling). Such a situation would indicate excessive velocity and insufficient mixing of primary and secondary air. Terminal velocity should occur at the boundary. Therefore,
 - (a) The total air pattern of vertical throw outlets should reach the ceiling (Figure 5.33).
 - (b) The total air pattern of high sidewall outlets should reach the opposite wall (Figure 5.34).
 - (c) The total air pattern of ceiling diffusers should reach the room walls.
- (6) Large rooms require multiple supply outlets. These must have air streams whose total air

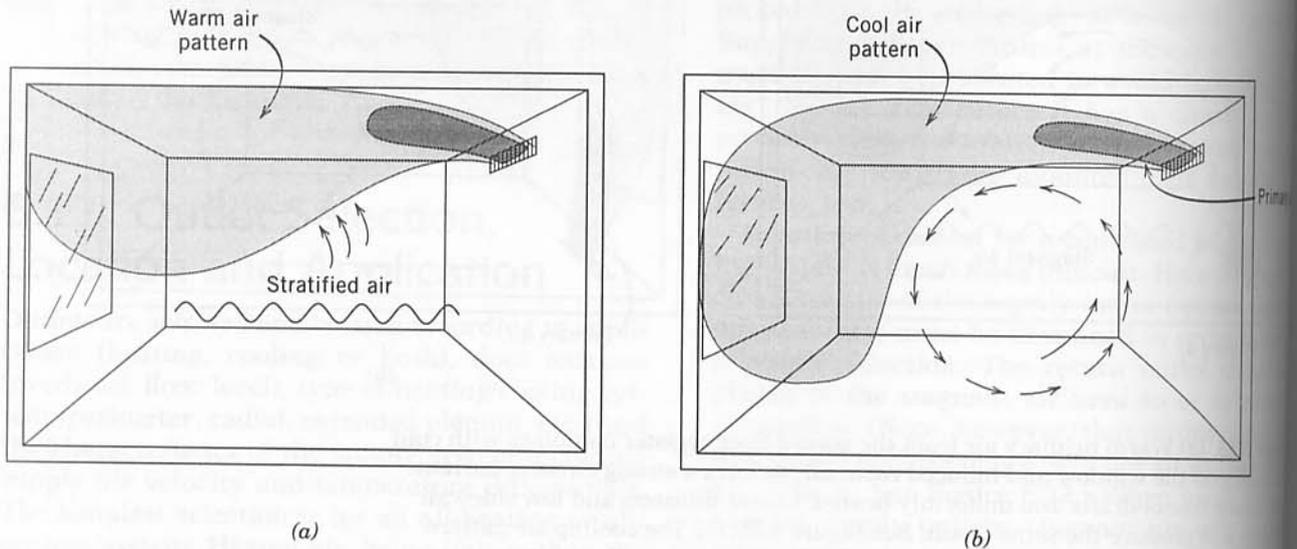


Figure 5.34 (a) The warm air will remain along the ceiling leaving a cool stagnant layer below. With a long enough throw, the warm air will reach the cool window and establish proper circulation. Excessive throw will result in undesirable drafts. (b) Cool air will readily entrain rising warm room air, and a good circulation will be established. The HSW inside wall register location is recommended for cooling and ventilation using outside air.

envelopes overlap to provide adequate coverage, without collision of high velocity air streams.

- (7) The NC rating of outlets must be no higher than the NC rating of the room in which they are installed.

b. Data Analysis

Application of these principles requires an analysis of three groups of data:

- (1) The architectural details of the space, with particular reference to the location of heat loss or gain such as windows, doors, ceiling under a roof and outside walls.
- (2) The amount (cfm) of air required and its temperature, as calculated from the heat loss or heat gain required.
- (3) The characteristics of supply registers and diffusers, as given in the technical data section of manufacturers' catalogs, with particular reference to velocity, throw, diffusion and NC.

c. Outlet Selection Procedure

Using these general rules, or principles, and the three groups of data, the selection procedure for outlets is as follows:

- (1) Decide on the outlet types and locations based on duct locations, window and door positions and type of HVAC system being designed.
- (2) Use a manufacturer's catalog that gives complete outlet data including cfm, throw for specific temperature(s), spread and noise. Select outlet(s) that will deliver the maximum cfm required (usually the cool air cfm).
- (3) Check the throw for a specific terminal velocity. Throws terminating in the occupied zone should not exceed 50 fpm. If the throw ends at a room boundary (wall or ceiling), it can have a terminal velocity up to 150 fpm, but preferably 50–100 fpm.
- (4) Check the outlet pressure drop against the system static pressure drop calculations.
- (5) Check the outlet noise level (NC value) at the face velocity being used. Table 5.1 is a short list of recommended NC values for a few common occupancies, plus maximum face velocity for most outlets that correspond to this NC criteria. In actual practice, always check face velocity data for a specific outlet. The velocities in

Table 5.1 Recommended NC Criteria for Various Occupancies and Maximum Supply Outlet Face Velocity

<i>Occupancy</i>	<i>NC Criteria</i>	<i>Maximum Face Velocity</i>
Large auditoriums	20–25	500
Small auditoriums, theaters, houses of worship, conference rooms, executive office	25–30	600
Sleeping room	25–35	700
Private office, libraries, small conference rooms	30–35	700
Living rooms, recreation rooms	35–40	800
Lobbies, drafting rooms, computer areas	40–45	800
Merchandising areas	40–50	900
Light industry, kitchen, equipment rooms	50–60	1000

Table 5.1 are provided only as a preliminary guideline.

You should be aware that the location of the return grille does not affect the air distribution in a room, except in the extremely unusual case that it is so close to the supply duct that the air flow is "short-circuited." (See Figure 5.32.)

As a rule of thumb, plain return outlets should be designed for a face velocity of 300–400 fpm and a maximum of 500 fpm. Return outlets above the occupied zone such as ceiling or high sidewall units may have face velocities up to 600 fpm. Door louvers, wall louvers and undercut doors should be limited to 300 fpm. Filter grilles should not have a face velocity above 300 fpm.

Using these figures, calculation of return grille size is straightforward.

Example 5.4 What size low sidewall plain return grille is required for a room supplied with 120 cfm of air for summer cooling and 85 cfm for winter heating?

Solution: We would size the return grille for the larger of the two airflow requirements. Using Equation (5.7), which is derived in Section 5.23.a (page 288)

$$Q = AV \quad (5.7)$$

where

Q is the airflow in cubic feet per minute,

A is the open area of the grille in square feet and

V is the air velocity in feet per minute

and trying a solution with a face velocity of 400 fpm (see rule in preceding paragraph), we have

$$120 \text{ cfm} = A \times 400 \text{ fpm}$$

$$A = \frac{120 \text{ ft}^3/\text{min}}{400 \text{ ft}/\text{min}} = 0.3 \text{ ft}^2$$

$$A = 0.3 \text{ ft}^2 \times \frac{144 \text{ in.}^2}{\text{ft}^2} = 43.2 \text{ in.}^2$$

Assuming a free area of 80% of gross area, we would calculate

$$A_{\text{gross}} = \frac{43.2}{0.8} = 54 \text{ in.}^2$$

Therefore, a 6 × 9-in. grille (or larger would be satisfactory).

5.18 Outlet Air Patterns

There are four basic air flow patterns in a space:

- Vertical, spreading
- Vertical, nonspreading
- Horizontal, at, or close to, ceiling level
- Horizontal, at, or close to, floor level

These characteristics of each pattern and its application are explained next.

a. Vertical Flow, Spreading Air Pattern

The vertical flow, spreading air pattern is produced by low sidewall diffusers, floor-mounted diffusers, and low wall linear diffusers such as the baseboard type. See Figure 5.35. Units are normally placed under windows and on cold walls to prevent cold air drafts from forming during the heating season. For heating, floor outlets below windows are preferable, particularly as part of a perimeter heating system. The total air pattern for heating appears approximately as in Figure 5.33(a). Throw should reach the ceiling in order to establish a good circular air motion pattern in the room. The return outlet is placed low on the opposite (inside) wall. This type of air pattern is not highly recommended for a combined heating and cooling system because the cooling air pattern will fall back on itself and a thorough mixing of primary and secondary air will

not occur. [See Figure 5.33(b).] Careful selection of the outlet characteristic, however, can result in an acceptable cooling system.

b. Vertical Flow, Nonspreading Pattern

The vertical flow, nonspreading distribution is similar to the spreading type except that throw is much longer due to the smaller spread. The outlet types are the same as for the spread distribution. This distribution is usable for combined heating and cooling because the longer throw will substantially eliminate the stagnant air shown in Figure 5.33(b).

c. Horizontal Flow

High Side Wall Outlets. The best position for a high sidewall (HSW) register with respect to heating is on an inside wall opposite the room window, as shown on Figure 5.36. The total air heating and cooling patterns produced are shown in Figure 5.34. As can be seen, the high sidewall register location is ideal for cooling (and ventilation), but much less effective for heating due to the tendency of hot air to rise. It is, therefore, important that the primary air throw be long enough to reach the outside wall and window. There, cool air entrainment will lower the total air temperature, causing the air to drop and establish the desired circulation. Insufficient throw will leave a blanket of hot air on the ceiling and stagnant cold air below.

If a sufficiently long throw is not possible due to the architecture of the space, an alternate (and expensive) solution is to use separate outlets for heating and cooling. A single return is usually possible. Register throw is normally adjustable by using the vertical vanes on the register face. Typical throw patterns are shown in Figure 5.28. The register's horizontal vanes can also be used to increase throw by setting them so that the primary air strikes the ceiling at a glancing angle, within 3–4 ft of the wall. This will induce the surface effect and will markedly increase the air stream throw. See Figure 5.29(b). Too great a horizontal vane angle will cause the primary air stream to collide with the ceiling. This will cause turbulence that will shorten the air stream throw.

Ceiling Diffuser. This type of outlet is used primarily for cooling. If heating is required, a diffuser with a vertical discharge characteristic is required.

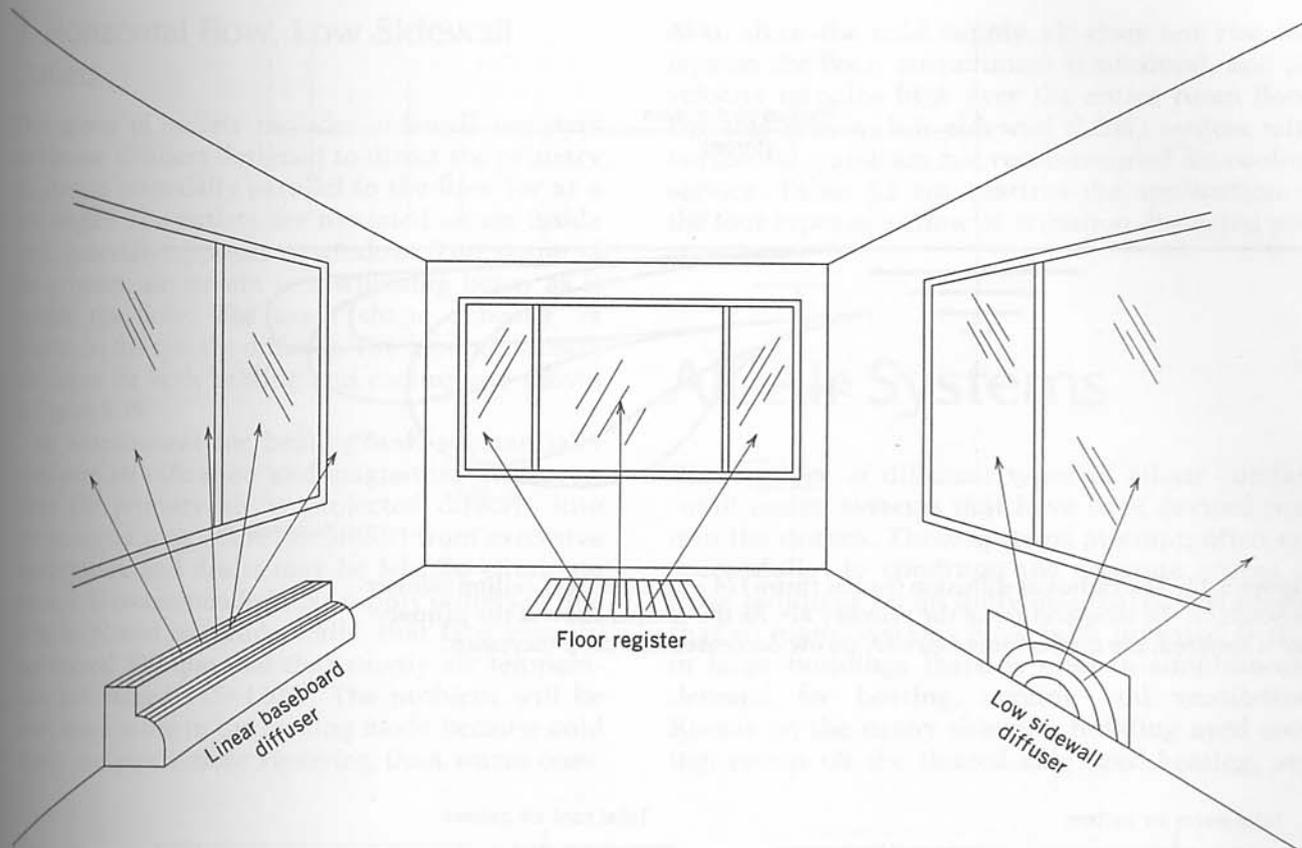


Figure 5.35 Typical vertical air stream spread-type outlets. These outlets are best located below windows. For heating duty, the spread patterns of warm air should blanket the windows. This prevents cold air drafts and increases the mixing of primary air with room air. Throw should reach the ceiling.

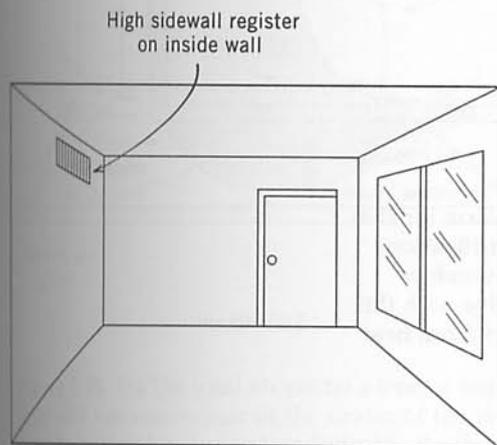


Figure 5.36 Typical horizontal flow HSW register. The HSW position is ideal for cooling and ventilation and marginally acceptable for heating if throw is adequate and the outlet is on an inside wall. See also the total air stream patterns in Figure 5.34.

Horizontal discharge diffusers are available with radial patterns and directional patterns. The former are usually round; the latter are square or rectangular. Horizontal flow ceiling diffusers have the ability to entrain a large amount of room air. As a result, and because the horizontal flow covers a large area, horizontal flow ceiling diffusers can handle large flow rates (high cfm) without producing unpleasant drafts. In cooling use, the diffuser's throw depends heavily on the temperature difference between the incoming air and the room air. See Figure 5.37. As a result, it is important for the design technologist to check the catalog throw data for a particular temperature difference. Typical cooling and heating air flow patterns are shown in Figure 5.38. The throw selected should reach the room walls, or, in large rooms, it should reach the air pattern of the adjacent diffusers. See Figure 5.27.

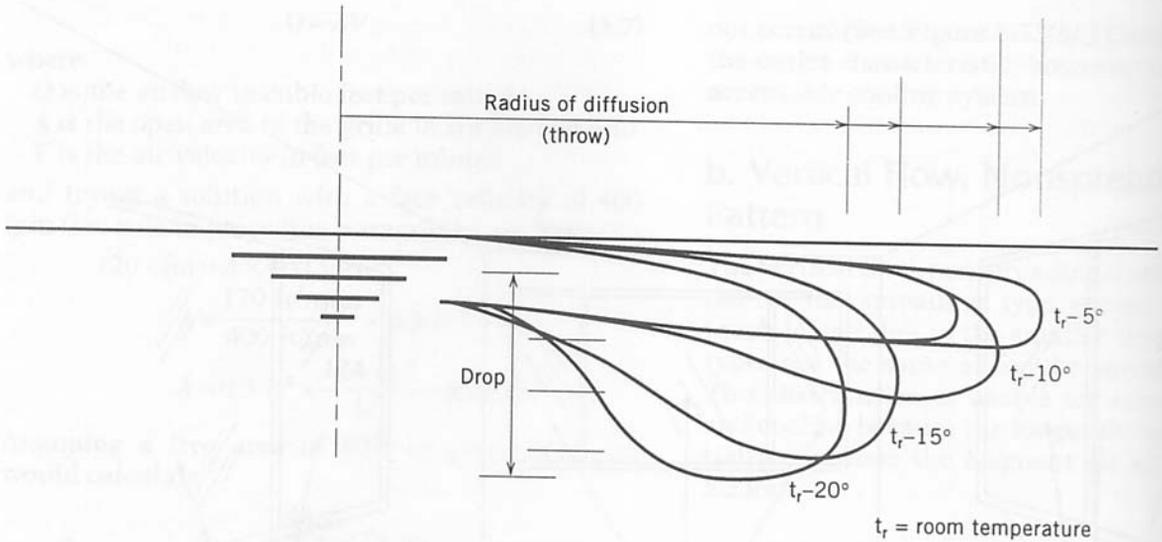


Figure 5.37 The radius of diffusion (radial throw) of a symmetrical ceiling diffuser depends on the temperature of the primary air. As the temperature of the primary air is lowered, the air becomes heavier, throw decreases and drop increases.

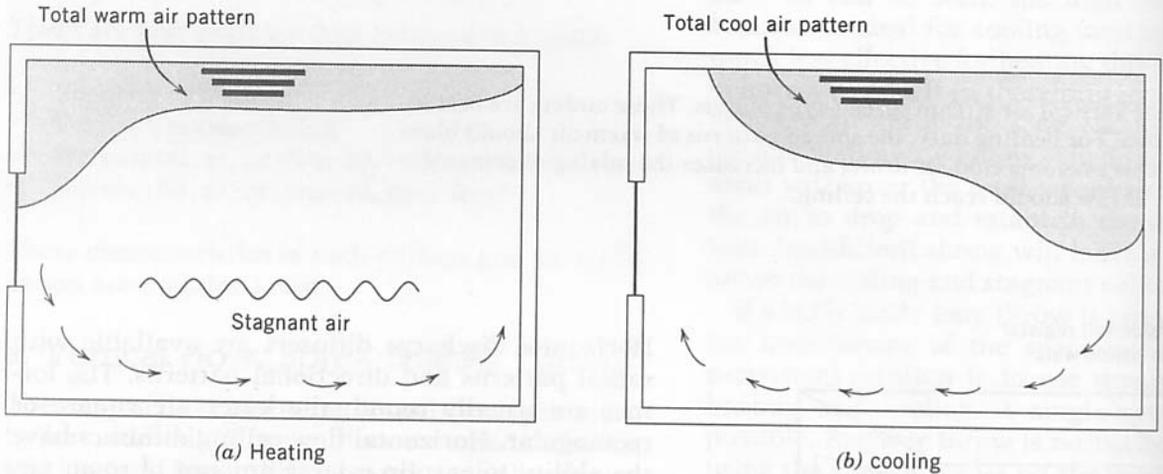


Figure 5.38 (a) A horizontal throw ceiling diffuser produces a layer of warm air that clings to the ceiling and does not mix with room air. This produces stratification, with warm air above and cool air below. The blanket is lower near the window where it mixes with cool air and, as a result, drops. (b) Cooling is effective with this distribution. The cool air drops on the inside wall and rises convectively from heat picked up at the window.

d. Horizontal Flow, Low Sidewall Outlets

This group of outlets includes sidewall registers and linear diffusers designed to direct the primary air stream essentially parallel to the floor (or at a low angle). The outlets are mounted on an inside wall, generally opposite a window. The shape of the primary air stream is a widening beam as it crosses the floor. The total air stream envelopes for both heating and cooling are shown in Figure 5.39.

Air entrainment for heating will substantially eliminate stratification and stagnation. However, since the primary air is projected directly into the occupied zone, some discomfort from excessive temperature and drafts may be felt. To eliminate these, it is recommended that supply outlets of this type be placed on inside walls, that face velocity not exceed 300 fpm and that supply air temperature not exceed 115–120°F. The problem will be even more acute in the cooling mode because cold drafts are much more annoying than warm ones.

Also, since the cold supply air does not rise but lays on the floor, entrainment is minimal, and air velocity remains high over the entire room floor. For this reason, low sidewall (LSW) outlets with horizontal throw are not recommended for cooling service. Table 5.2 summarizes the application of the four types of airflow distribution discussed previously.

All-Air Systems

The number of different types of all-air comfort conditioning systems that have been devised runs into the dozens. These systems attempt, often unsuccessfully, to condition the separate spaces in large buildings on an individual basis. The reason that so many systems have been invented is that in large buildings there is often a simultaneous demand for heating, cooling and ventilation. Rooms on the sunny side of a building need cooling, rooms on the shaded side need heating, and

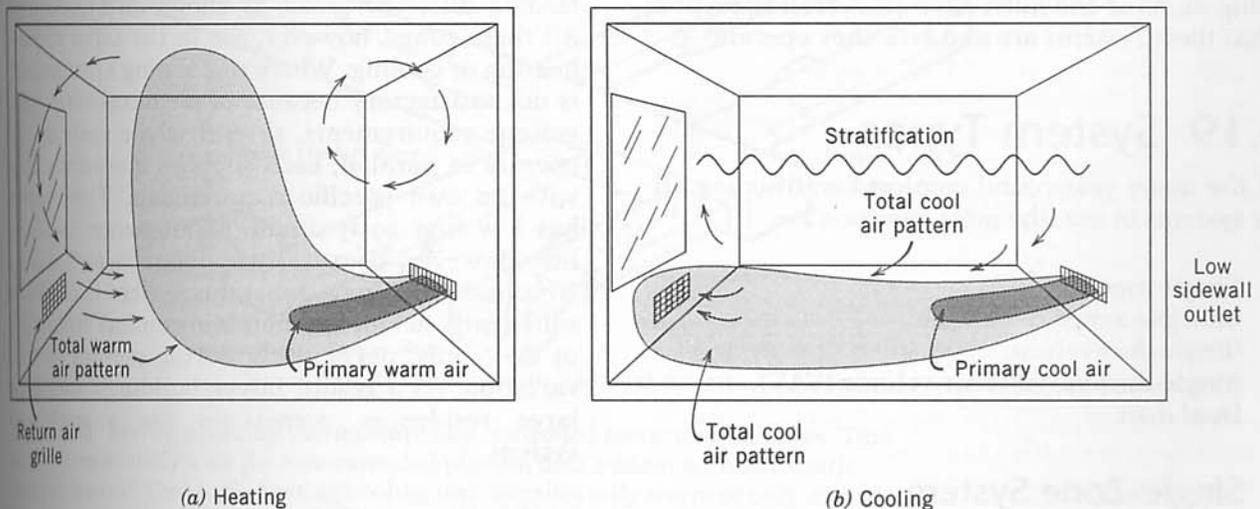


Figure 5.39 (a) The total air pattern from a horizontal discharge low sidewall diffuser will rise somewhere in the center of the room as shown. Higher air velocity must be avoided to prevent undesirable drafts. Induction and entrainment occur on both sides of the total air pattern, with very little stagnant air remaining. This system, with proper balancing, is adequate for heating. (b) Cool air will simply lie at the floor level with this distribution. Very little mixing with room air occurs, causing stratification of the entire upper section of the occupied area. A large temperature differential develops between the cold floor and the warm upper level. This arrangement is not recommended for cooling.

Table 5.2 Application of Air Supply Outlets

<i>Flow Pattern</i>	<i>Type of Outlet</i>	<i>Preferred Location</i>	<i>Recommended Application</i>
Vertical, spread	LSW diffuser, linear baseboard, spread floor register	Exterior walls, below windows	Preferably heating, also cooling
Vertical, nonspread	LSW diffuser, linear baseboard, nonspread floor register	Exterior walls, below windows	Heating and cooling
Horizontal, high sidewall	HSW register	Inside wall	Cooling; also heating if carefully designed and adjusted
Horizontal, ceiling diffuser	Ceiling diffuser	Ceiling	Cooling
Horizontal, low sidewall	LSW register, linear diffusers	Interior walls, opposite windows	Heating

rooms in the building core need primarily ventilation. These large building systems are beyond the scope of the book as far as design is concerned. However, the engineering technologist will definitely be called upon to work on such systems under the direction of an HVAC engineer. That being so, he or she must have an overall concept of what these systems are and how they operate.

5.19 System Types

Of the many year-round comfort conditioning all-air systems in use, the most common are

- Single zone.
- Multiple zones.
- Single-duct reheat.
- Single-duct variable air volume (VAV).
- Dual duct.

a. Single-Zone System

See Figure 5.40. This is the system most often used for small single-use buildings that operate as a single zone. It is also the system that will be studied most intensively in this book. These systems are always low pressure and low velocity. A single air-handling unit, which supplies fixed quantities of air to the building spaces, is used. If a change in air volume is required for seasonal changeover (heating to cooling and vice versa), the duct volume

dampers must be reset. The total air quantity can be changed at the air handler, generally by motor speed control. In modern systems, air quantities can be regulated automatically.

Multiple subzones can be established by thermostatic control of volume dampers in branch ducts feeding different areas, as shown in Figure 5.41. All these zones, however, are in the same mode—heating or cooling. Where this zoning arrangement is not satisfactory because of air temperature and volume requirements, several whole systems can operate in parallel; each supplies a separate zone with its own specific requirements. This system has low first cost, simple maintenance and long life. However, because it is designed to handle a structure as a single zone, this system is often not sufficiently flexible to maintain comfort conditions in all rooms, particularly during periods of load variation. As a result, larger buildings, including large residences, sometimes use a multizone system.

b. Multizone System

See Figure 5.42. There are several variations of the design. Essentially, a multizone system consists of a single air handler plus individual zone duct systems. Each duct system has a heating and cooling source supplied by a central heating device and a central refrigeration device. In some systems, each zone has heating and cooling coils; in other systems, hot and cold air are supplied, to be mixed

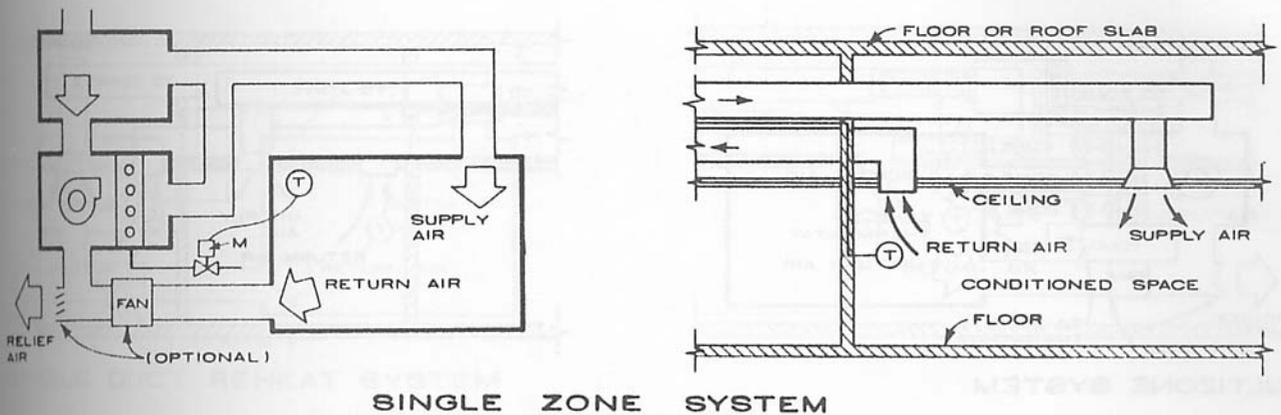


Figure 5.40 Single-zone systems are low pressure, low velocity installations, best applied to small residential and commercial buildings. The entire building is treated as a single control zone, controlled by a single thermostat. See text. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., 1988, John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

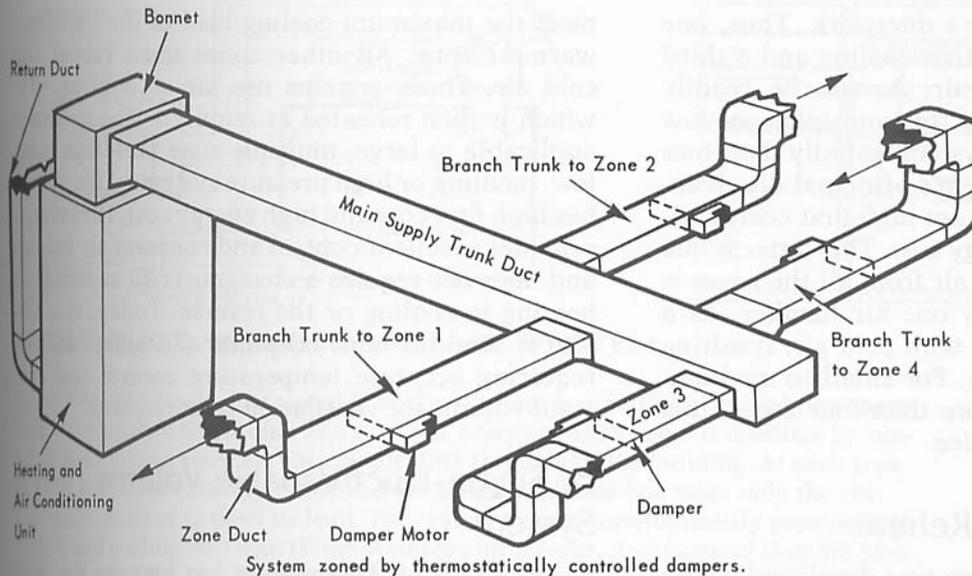


Figure 5.41 System zoned by thermostatically controlled motorized dampers. This system is essentially a single-zone extended-plenum duct system with automatic damper control. The single heating/cooling unit supplies only warm or only cool air, at a temperature suitable to supply the heaviest zone load. The other zones then throttle the air supply to satisfy their load requirements. (Reproduced with permission from *ACCA Manual C*, p. 21).

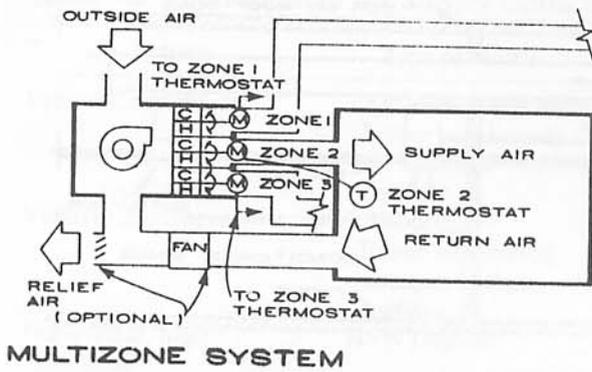
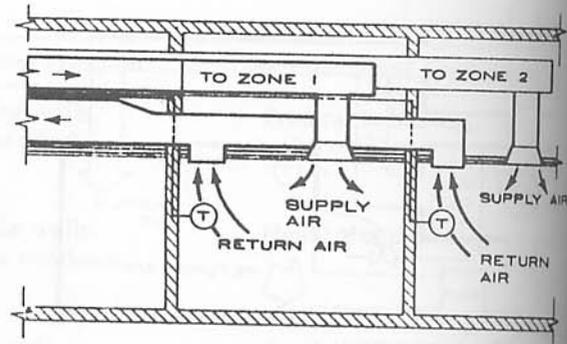


Figure 5.42 Multizone systems are used for medium-size buildings where different zones have different conditioned air requirements. Hot and cold air are produced centrally and provided to each zone. Therefore, heating and cooling can be provided simultaneously to different zones. Limiting factors are the number of zones and energy costs. See text. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)



at the entry to each zone's ductwork. Thus, one zone can use heating, another cooling and a third ventilation with outside air. As can be readily imagined, the ductwork for this multiple zone low pressure, low velocity system rapidly becomes enormous. This is the system's principal disadvantage. Other disadvantages are high first cost, difficult control and high energy cost. This latter is due to the fact that the return air from all the zones is mixed, since there is only one air handler. As a result, warm air is mixed with cold air, resulting in a large waste of energy. For small to medium-size buildings with no more than four zones, this system is a reasonable choice.

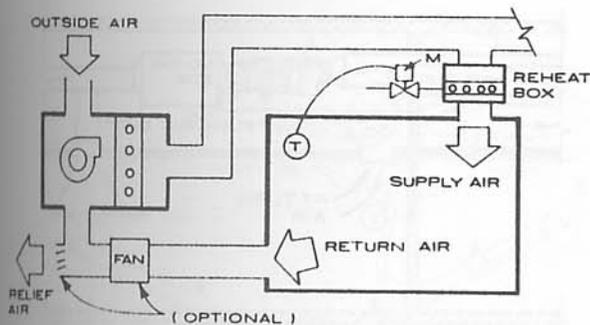
c. Single Duct with Reheat

See Figure 5.43. This system was developed before the energy crisis of 1973 made HVAC designers (and others) take a long hard look at their comfort conditioning systems. It is not commonly used today because it is notorious for energy waste. However, careful design can make it useful for some climates. The system was designed to solve the problem of massive ductwork in large multiple zone buildings using the multizone system already described. This system uses a single duct that provides air (generally very cold) to the entire building. At each zone, a small reheat coil heats this cold air to the temperature required for that zone. The central system must provide air cold enough to

meet the maximum cooling load of the building's warmest zone. All other zones must reheat this cold air. These systems use air as cold as 40°F, which is then reheated as required. The system is applicable to large, multiple zone buildings using low, medium or high pressure systems. This system has high first cost and high energy cost. However, it provides excellent control and constant air volume and does not require a changeover to switch from heating to cooling or the reverse. Today, this system is used for labs, hospitals and other facilities requiring accurate temperature control and constant volume for ventilation requirements.

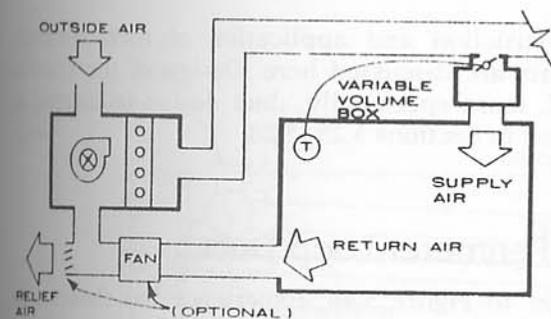
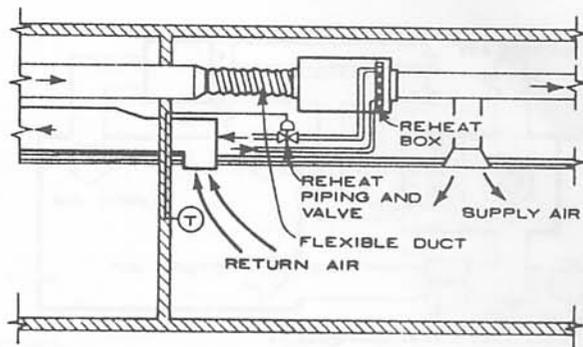
d. Single-Duct Variable Volume (VAV) System

See Figure 5.44. This system has become the most popular design for medium-size to large buildings because of low first cost, low energy cost and small ductwork. The system, as its name implies, compensates for variable loads by varying the volume of air supplied rather than its temperature. This is the "secret" of its energy economy. The air volume variation is accomplished by a thermostatically controlled variable air-volume box. This box takes main duct air from the single supply duct and modulates the air quantity supplied to a space to match its load. The central supply will furnish either cold air or warm air to the entire building depending on outdoor conditions and prevailing



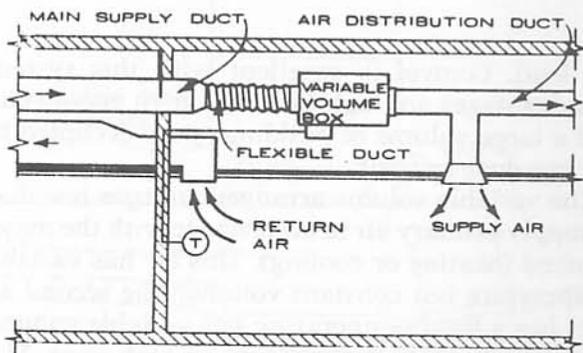
SINGLE DUCT REHEAT SYSTEM

Figure 5.43 The single-duct reheat system circulates constant temperature cold air, which is then reheated at each zone as required. This arrangement occupies little space and provides excellent control. However, it is extremely energy wasteful. As a result, it is seldom used in modern design. See text. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)



SINGLE DUCT VARIABLE VOLUME SYSTEM

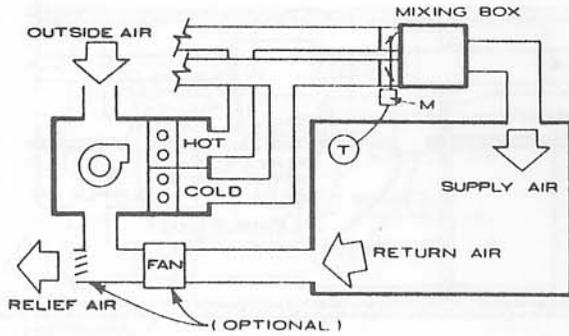
Figure 5.44 The single-duct variable volume system is economical of building space since it runs only a single duct and excels at energy conservation. It operates by supplying fixed temperature air via a single duct throughout the building. At each zone location, a thermostatically controlled variable air volume box takes only the volume of air required to meet its load. The system cannot simultaneously provide both heating and cooling. See text. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)



indoor needs. Obviously then, this system is more suited to buildings that always need cooling (large interior zone) than to buildings with perimeters requiring heating and cooling simultaneously. Also, because the volume of air supplied to a space varies with load, this system cannot be used in buildings requiring constant air changes, such as labs and medical facilities.

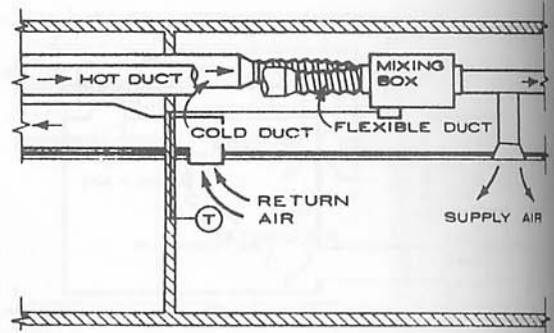
e. Dual Duct Systems

See Figure 5.45. This system comes in two designs—variable volume and constant volume. The constant-volume system consists of two complete duct distribution systems—one with hot air and one with cold air. A mixing box at each zone location provides air at the temperature required for



DOUBLE DUCT SYSTEM

Figure 5.45 The dual-duct system supplies hot and cold air in separate ducts, to be mixed as required by the load of each zone. This system is available in a constant volume and a variable volume design. The constant volume design has better control but is less economical than the variable volume design. See text. (From Ramsey and Sleeper, *Architectural Graphic Standards*, 8th ed., 1988, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)



the load. Control is excellent with this system. Disadvantages are high first cost, high energy cost and a large volume of building space occupied by the two duct systems.

The variable volume arrangement uses one duct to supply primary air in accordance with the major demand (heating or cooling). This air has variable temperature but constant volume. The second air duct has a fixed temperature but variable volume. The two air streams are mixed at each zone. This system uses smaller duct work than the constant volume system, is cheaper to install and uses less energy. Control, however, is not as rapid and accurate as with the constant volume system.

5.20 Single-Zone System Duct Arrangements

As previously noted, single-zone systems are used in buildings where the entire space can be considered as a single zone. This generally includes small to medium-size residences, repair shops, stores, small industrial buildings and the like. The duct arrangements most frequently used are:

- Perimeter loop.
- Radial (perimeter).
- Radial (overhead).
- Extended plenum.
- Reducing plenum.

Construction and application of these arrangements are discussed here. Design of the systems and, more specifically, duct design is covered in detail in Sections 5.25–5.28.

a. Perimeter Loop Duct

Refer to Figure 5.46. Experience has shown that this duct arrangement is ideal for heating slab-on-grade and crawl space structures in cold climates. The perimeter duct, installed directly in the concrete floor, heats the slab, thereby providing a large radiant heat source for the entire structure. The perimeter floor outlets, which should be located under all windows, will temper the cold air sliding down from the windows and prevent cold drafts. See Figures 5.33(a) and 5.35. Additional perimeter floor outlets are installed to supply additional warm air, as indicated by the load calculations. If the system is to be used for cooling as well as heating, nonspread floor registers should be used. See Section 5.18b and Table 5.2. The ducts themselves can be metallic (galvanized sheet metal or steel), concrete, asbestos-cement, ceramic or organic fiber. The installation is shown in Figure 5.46. The usual duct size is 6–8 in. depending on the air quantities being carried. A typical residential perimeter loop system is shown in Figure 5.47. The recommended floor outlet is shown in Figure 5.48.

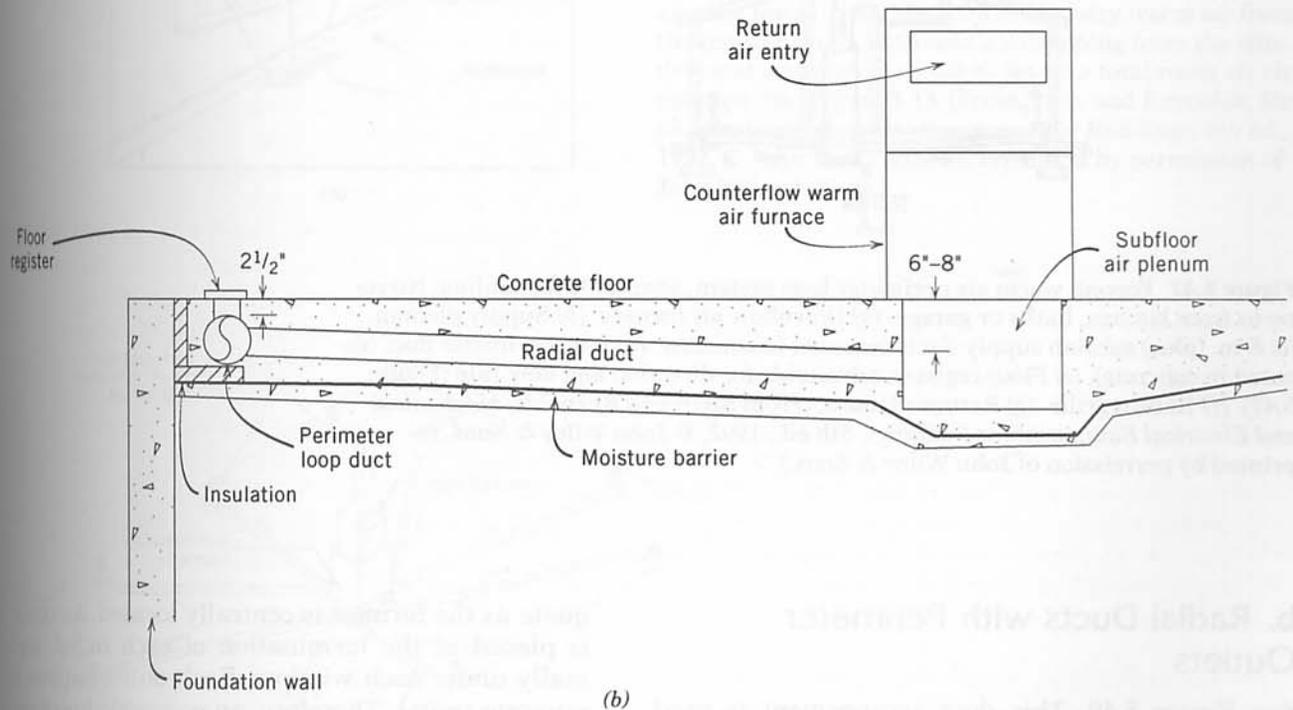
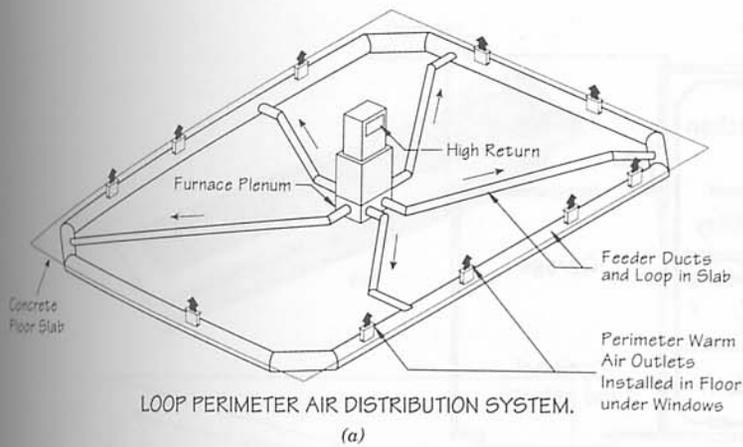


Figure 5.46 (a) Perimeter loop air distribution system. A downflow warm air furnace forces air into a subfloor plenum. Radial ducts 6–8 in. in diameter connect the perimeter loop duct to the air plenum. Floor registers around the structure supply warm air into the various rooms. The concrete floor slab is heated by the radial feeder ducts and loop perimeter ducts. (b) Installation detail of the loop duct. [(a) Bobenhausen, *Simplified Design of HVAC Systems*, 1994, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.]

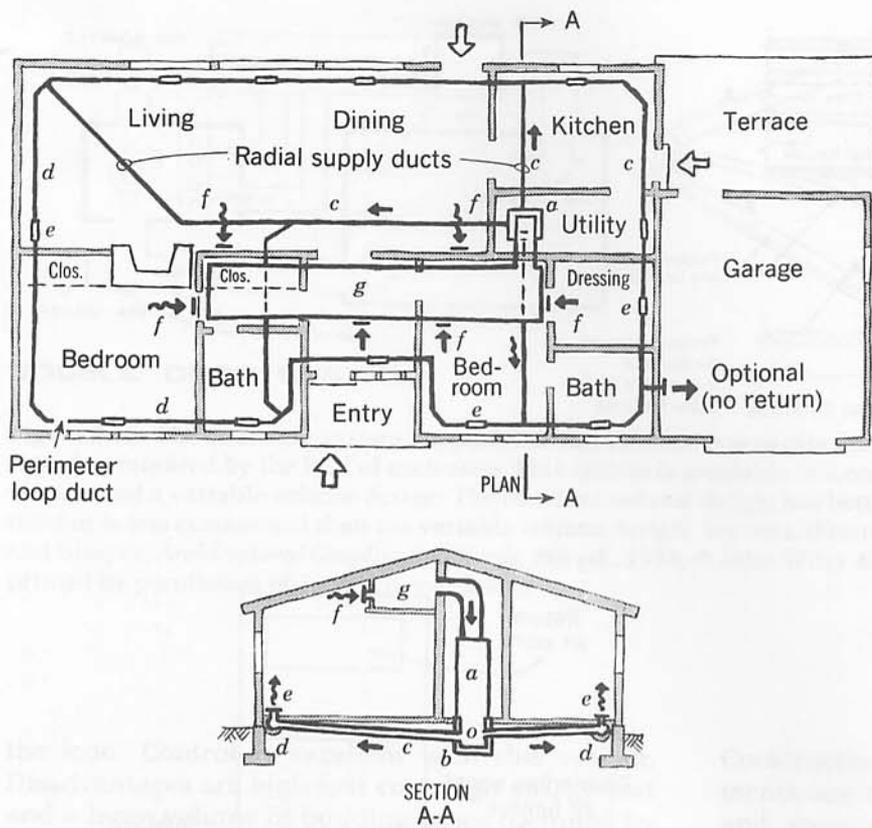


Figure 5.47 Forced, warm air perimeter loop system, adaptable for cooling. No returns from kitchen, baths or garage. (a) Downflow air furnace. (b) Supply plenum. (c) 8-in. (plus) subslab supply ducts (encased in concrete). (d) 8-in. perimeter duct (encased in concrete). (e) Floor register, adjustable for direction and flow rate (Figure 5.47). (f) Return grille. (g) Return plenum. (From Stein and Reynolds, *Mechanical and Electrical Equipment for Buildings*, 8th ed., 1992, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

b. Radial Ducts with Perimeter Outlets

See Figure 5.49. This duct arrangement is used where floor slab heating is not of primary importance. This might be in buildings with a low ceiling basement or an enclosed crawl space or in a mild climate area. In buildings with a basement, the radial ducts are run uninsulated, under the floor slab. In buildings with a crawl space, the radials can be run in or under the floor slab, as desired. When run under the floor slab in an enclosed crawl space, they are usually uninsulated; in an open crawl space, they are insulated.

As with the perimeter loop system, a downflow furnace supplies air to the radial ducts via an underfloor plenum. A single return is usually ade-

quate as the furnace is centrally located. An outlet is placed at the termination of each radial, normally under each window. Each outlet requires a separate radial. Therefore, an economic breakover point occurs where it becomes more economical to use a perimeter loop with only a few radials. This is one of the first decisions that the project mechanical engineer makes.

c. Radial Duct Arrangement (Overhead)

This system is used where the primary function of the comfort conditioning system is cooling and the air handler (furnace, heat pump, air conditioner) is centrally located so that branch duct lengths to the various building spaces are roughly equal in

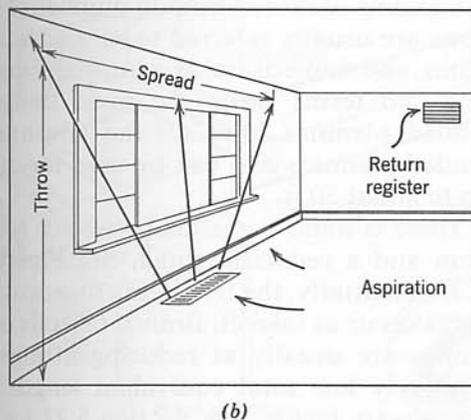
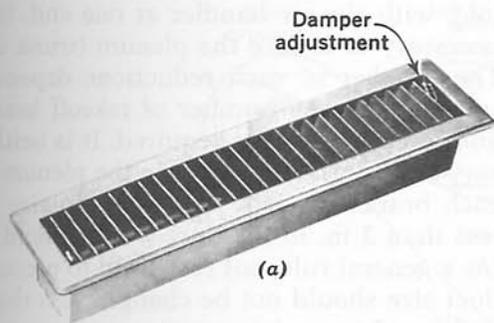
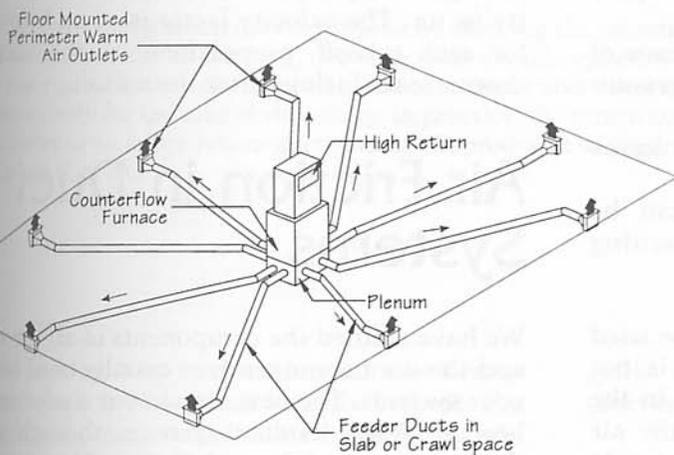


Figure 5.48 Floor register (a) and its distribution (b). The register's vanes control the spread and, thereby, also the throw. See Figure 5.28. The very warm air from the register mixes with cold air dropping from the window and warmish room air to set up a total room air circulation. See Figure 5.33. (From Stein and Reynolds, *Mechanical and Electrical Equipment for Buildings*, 8th ed., 1992, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)



RADIAL PERIMETER AIR DISTRIBUTION SYSTEM.

Figure 5.49 Radial duct system with perimeter outlets. This arrangement is used in structures with basements and with open or enclosed crawl space below the first floor (slab). Ducts are either encased in the slab (open crawl space) or run below the floor (basement or enclosed crawl space). (Bobenhausen, *Simplified Design of HVAC Systems*, 1994, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

length. An upflow air handler is used, feeding radial ducts that terminate in ceiling diffusers. The specific design of diffusers depends on whether their use is cooling only or primarily cooling and secondarily heating. (See Figure 5.38.) If heating is important, a diffuser with a downward throw would be utilized.

d. Extended Plenum Air Distribution System

An extended plenum system is simply a trunk duct extending from the supply plenum of the main air-handling unit (warm air furnace or air conditioning unit) with multiple outlets and/or branch ducts connected to it. See Figures 5.41 and 5.50. Because the trunk duct does not change in size from its connection to the supply plenum until the end of its run, it is, in effect, an extended plenum; hence its name. The extended plenum may run in a basement or crawl space as in Figure 5.50, in which case the outlets would be floor or sidewall units. Alternatively, the extended plenum can run overhead in a dropped ceiling or attic, in which case the outlets would be ceiling diffusers or sidewall registers as required. Extended plenums have a number of advantages including:

- Low first cost because of the absence of expensive duct size change fittings.
- Low operating cost because of the absence of energy-using fittings (high static pressure losses).
- Ease of balancing due to low pressure losses and few trunk pressure changes.
- Ease of making changes. Branches can be added, moved and removed without upsetting the system.

The extended plenum arrangement can be used efficiently when the overall trunk length is not more than 50 ft long with the air handler in the center or not more than 30 ft long with the air handler at one end. Longer duct lengths not only become uneconomical but also require the use of a reducing plenum to maintain air velocity. This will become clear in our duct design discussion further on in this chapter.

e. Reducing Plenum Air Distribution System

This arrangement is also known as a semi-extended plenum. See Figure 5.18(d) (page 231) and Figure B.3. When the plenum is more than about 25–30 ft

long with the air handler at one end, it becomes necessary to reduce the plenum (trunk duct) size. The number of such reductions depends on the total length, the number of takeoff branch ducts and the air velocities required. It is neither necessary nor advisable to reduce the plenum size after each branch takeoff. As a rule, no size reduction less than 2 in. in the duct width should be made. (As a general rule, not restricted to plenum design, duct size should not be changed less than 2 in. in width or 2 in. in diameter.)

Plenums with a minimum number of size reductions are usually referred to as semi-extended plenums and sometimes as semi-reducing plenums. The two terms mean the same thing. Semi-extended plenums have all the advantages of extended plenums and can be used for duct lengths up to about 50 ft.

There is some confusion between a reducing plenum and a reducing trunk. See Figures B.2 and B.3. Essentially, the trunks are identical; the differences occur at takeoff. Branch takeoffs on reducing trunks are usually at reducing fittings and have relatively low total equivalent length (TEL) and no velocity factor. See Section 5.27 for a detailed explanation of this factor and Appendix B, Figure B.3 for typical values. Reducing plenum takeoffs are right angles fittings on the trunk body. They have generally high TEL to which is added a velocity factor. The velocity factor is an additional TEL for each takeoff, proportional to the number of downstream fittings after the takeoff.

Air Friction in Duct Systems

We have studied the components of all-air systems and the duct arrangements usually used in single-zone systems. The next step in our study is to learn how air flows in duct systems, through straight duct sections and through fittings. Most important, we must learn how to calculate the friction losses of air movement in ducts. Once we have mastered this skill, we will be in position to approach overall duct system design for small to medium-size residential and commercial buildings.

Refer to Figure 5.14 (page 223), which lists all the sources of static pressure loss in a system. Note that there are two sources of static pressure loss—items of equipment and ductwork. The loss in an equipment item—evaporator coil, humidifier, filter, supply register and return grille—can be de-

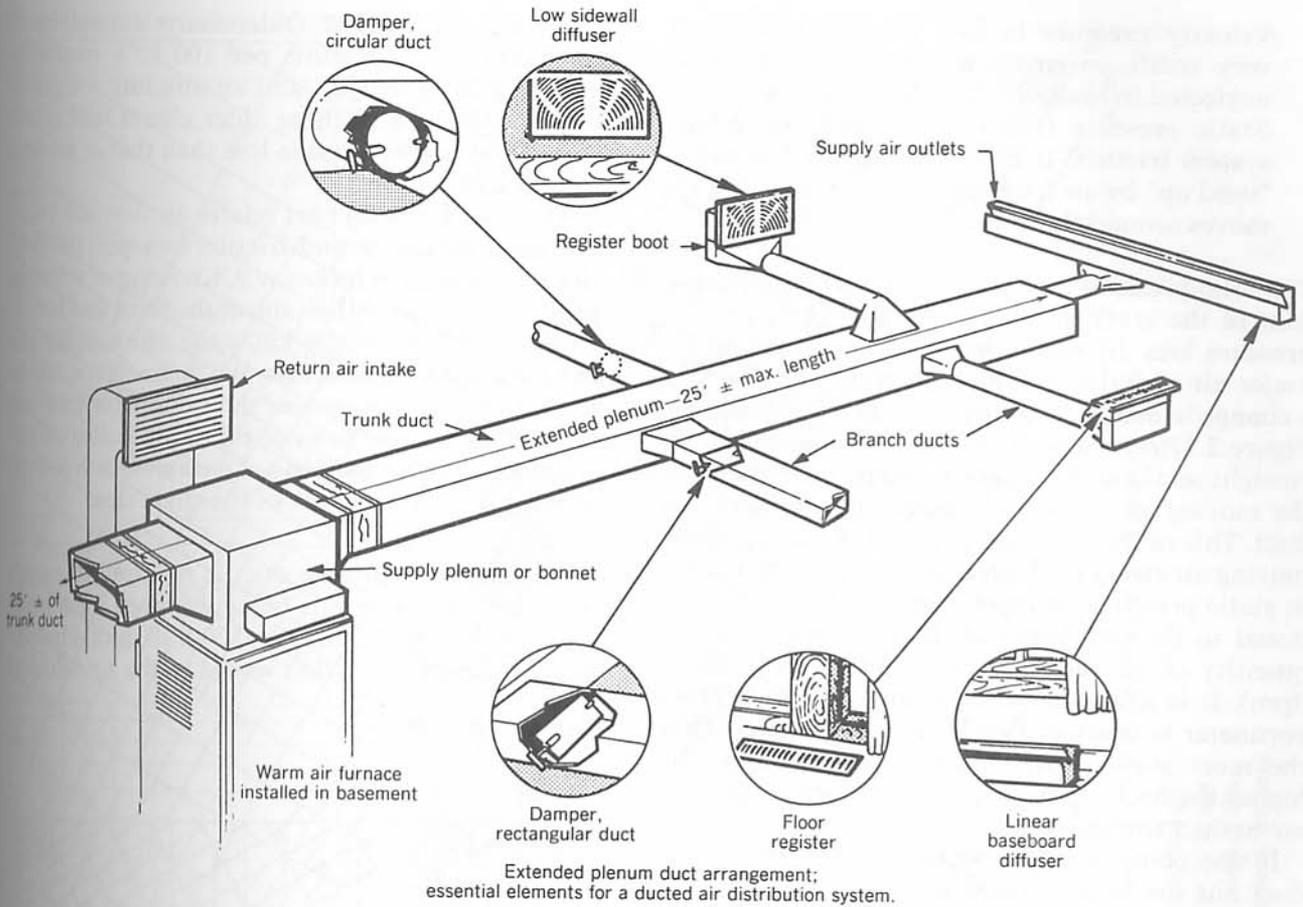


Figure 5.50 Extended plenum duct arrangement, showing the essential elements of a ducted air system. It is understood, of course, that a single system would normally use only one type of branch duct and one type of outlet. A floor-level single return intake is shown only for the sake of simplicity. In practice, the return could be a high sidewall outlet or an entire return duct system. (Reproduced with permission from *ACCA Manual C*, p. 28.)

tained by looking it up in the manufacturer's catalog. The losses in ductwork, which include the supply plenum, must be calculated. We will study that subject next.

You may have noticed that there is one large part of the overall system that is completely ignored as far as static pressure loss is concerned. That part is the section between the supply register output and the return register input, that is, the building spaces or rooms. The reason that these are neglected is that the air velocity in them is so low that the static and velocity pressure drop in them is close to zero and, therefore, negligible. The pressure at the face of the supply register is usually taken to be zero and that at the face of the return

grille is assumed to be very slightly negative (suction).

5.21 Air Friction in Straight Duct Sections

At this point, you should review Section 5.4, which introduced the subject of air pressure in ducts. Very briefly, we learned there that:

- The source of all duct pressure is the system air handler, usually the furnace blower.
- Total pressure at any point is the sum of static pressure plus velocity pressure.

- Velocity pressure in low velocity systems is very small, so small, indeed, that it is often neglected in branch duct calculations.
- Static pressure is that required to overcome system friction. It can be thought of as being "used up" by air friction in the ducts, as the air moves around the system.

The total pressure loss in any section of duct is the sum of the static pressure loss and the dynamic pressure loss. Dynamic pressure loss is caused by major air turbulence, which, in turn, is caused by a change in duct size or direction. See, for instance, Figure 5.17(c-1) (page 230). Static pressure loss in straight sections of duct is caused by the friction of the moving air "rubbing" against the walls of the duct. This rubbing results in a loss of energy in the moving air stream, which expresses itself as a drop in static pressure, or head. This friction is proportional to the roughness of the duct walls, to the quantity of air moving (cfm) and to its velocity (fpm). It is also proportional to the ratio of duct perimeter to cross-sectional area. This means that the more surface there is per unit of area, the higher the friction, simply because there is greater air-to-duct surface contact.

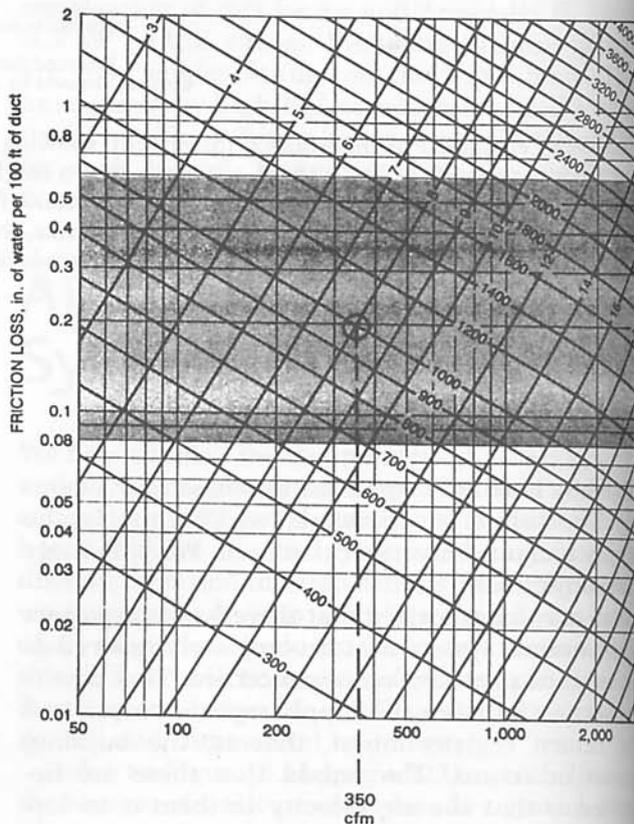
It was pointed out in Section 5.12.b that round duct has the highest ratio of area to perimeter of any shape. Conversely, round duct has the lowest perimeter to cross-sectional area ratio. This means that round duct has the lowest friction loss of any shape, for a given air flow and velocity. The reason is that its minimum perimeter means minimum contact between duct wall and moving air to cause friction. Round duct is used as the basis of all friction calculations. If other shapes are used, they are calculated as equivalents to round duct, as we will explain shortly.

Figure 5.51 is the standard duct friction chart for round galvanized steel duct. It is based on air at 70°F and sea level air pressure, weighing 0.075 lb/ft³, flowing in galvanized steel duct constructed with longitudinal seams and beaded slip couplings on 4-ft centers. This duct has an absolute roughness ϵ of 0.0003. (There are several other round galvanized steel constructions that have the same roughness. The chart in Figure 5.51 applies equally to such a duct.) This duct construction gives a roughness category of "medium smooth." No correction to the chart data is required for air at temperatures from 40 to 100°F, at elevations up to 1500 ft and at duct pressures +20 in. of water relative to the ambient pressure. Note that this

chart dates from 1987. Older charts were all based on ducts with 40 joints per 100 ft, a roughness category of "average," and an absolute roughness of $\epsilon=0.005$. Use of these older charts will give a somewhat higher friction loss than that of modern duct construction.

The duct friction chart relates air flow, air velocity, duct diameter and friction loss per 100 ft of duct, measured in inches w.g. Knowing or selecting any two of these will establish the third and fourth. In practice, we usually know the air quantity and the maximum friction loss. We then select a combination of air velocity and duct diameter from the chart. The shaded area of the chart indicates recommended combinations of parameters. A few examples should make use of the chart clear.

Example 5.5 A branch duct is required to supply 350 cfm. The system is being designed for a static friction drop of 0.2 in. w.g./100 ft of duct. Find the required duct size. What would be the air velocity?



Example 5.5

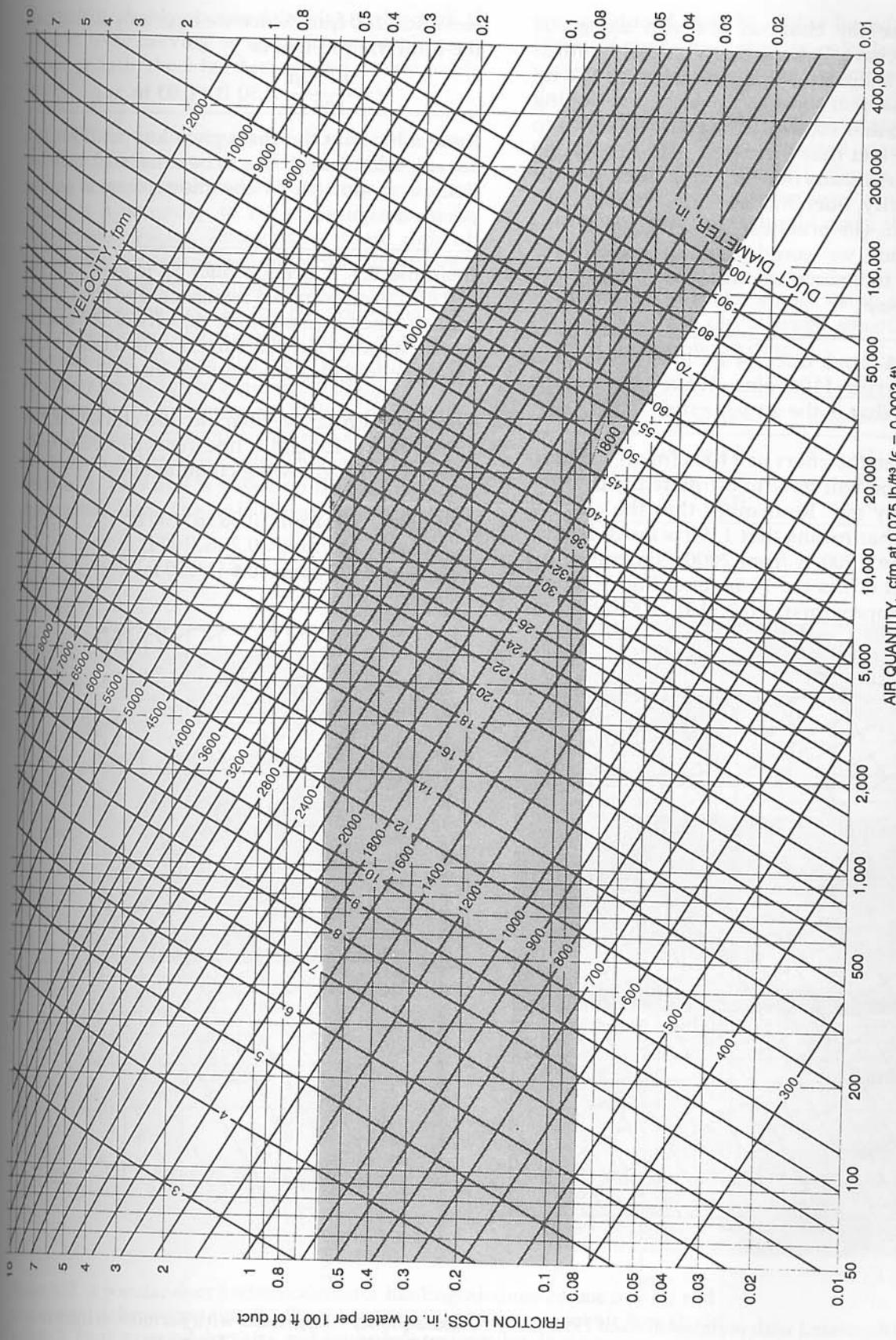
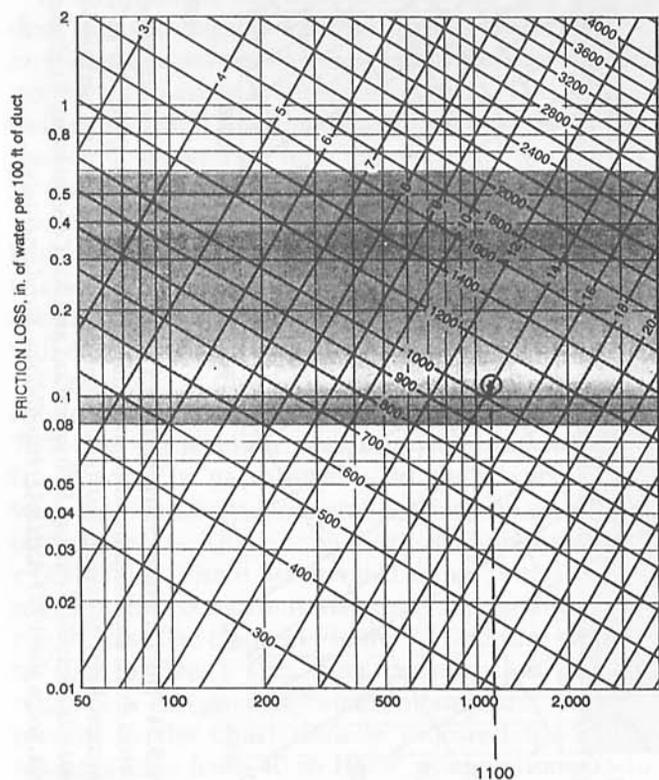


Figure 5.51 Duct friction chart, standard (English) units. Air motion in round galvanized steel duct, medium smooth, roughness coefficient $\epsilon = 0.0003$. Air at 70°F, sea level atmospheric pressure, 0.075 lb/ft³ density. (Reproduced by permission of the American Society of Heating, Refrigeration and Air Conditioning Engineers, Atlanta, Georgia, from the 1993 ASHRAE Handbook—Fundamentals.)

Solution: Enter the chart at 350 cfm along the bottom of the chart. Draw a line vertically until it intersects the horizontal line representing 0.2 in. w.g. The intersection shows 8-in. duct (line sloping up to the right) and 1000 fpm (line sloping down to the right). The fact that this point is in the shaded part of the chart means that the combination of air quantity, velocity, duct diameter and friction is an acceptable one. (In practical duct design, if this were a residence, we would probably use a larger duct in order to reduce the air velocity, so as to limit duct noise.)

Example 5.6 A trunk duct 14 in. in diameter and 30 ft long carries 1100 cfm. What is its static friction loss? What is the air velocity?

Solution: Enter the chart at 1100 cfm on the horizontal axis. (You will have to estimate the position of 1100 cfm, by eye. Remember that the chart is logarithmic. That means that 1100 is much farther from 1000 than 1900 is from 2000.) Extend a line vertically until it hits the 14-in. duct line. Read off the chart (by approximation) 0.11 in./100 ft friction



Example 5.6 (Reprinted with permission from 1993 ASHRAE Handbook—Fundamentals.)

loss and 1100 fpm. Since we have only 30 ft of duct, the total friction loss is

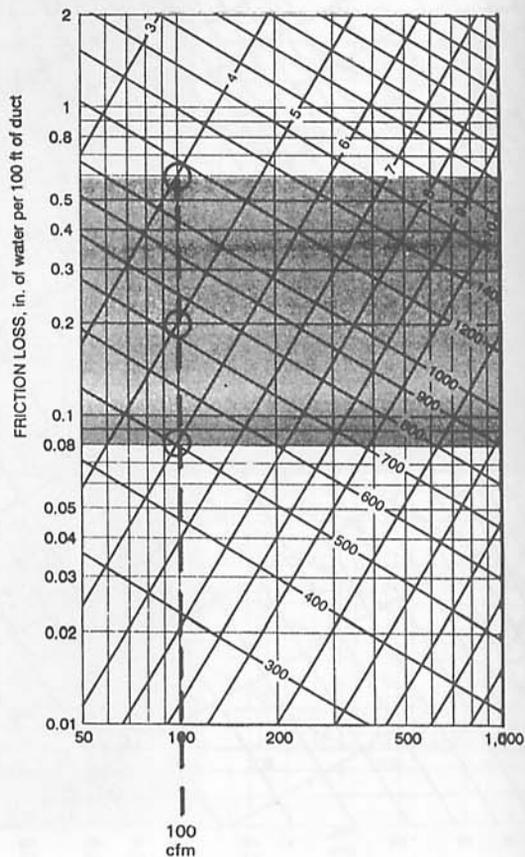
$$f = \frac{0.11 \text{ in.}}{100 \text{ ft}} \times 30 \text{ ft} = 0.03 \text{ in. w.g.}$$

Remember that the chart gives loss per 100 ft. Loss for any other length has to be calculated, as before. Failure to do this is the most common error of novice designers.

Example 5.7 A branch duct 15 ft long will carry 100 cfm to a room in a residence. The noise criteria recommends a maximum velocity of 500 fpm. Select an appropriate duct size. Friction is not critical since the length of duct is very short.

Solution: Enter the chart at 100 cfm on the horizontal scale. Extend a line vertically. Note that it intersects the following combinations:

6-in. duct	500 fpm	0.08 in./100 ft
5-in. duct	730 fpm	0.2 in./100 ft
4-in. duct	1170 fpm	0.6 in./100 ft



Example 5.7 (Reprinted with permission from 1993 ASHRAE Handbook—Fundamentals.)

We would probably choose 6-in. duct, although if the duct were serving a noisy room such as a kitchen or bath, the 5-in. duct is also a reasonable choice.

As you have surely concluded, the friction chart of Figure 5.51 is not easy or convenient to use because of its logarithmic air quantity scale and because it is necessary to make visual interpolations between values. Recognizing these difficulties, a number of companies and professional organizations have produced slide rule-type calculators that give the same data as the friction chart, plus sizes of equivalent rectangular ducts and other important data. Two of the best known of these calculators are shown in Figures 5.52 and 5.53. The calculator shown in Figure 5.53 gives the friction loss for straight duct sections on one side and duct fitting losses on the other side. This calculator was produced in 1989 and is based on the current duct construction data. The unit shown in Figure 5.52

was produced in 1976 and is based on older data. Users of these and similar calculators should always check to determine which duct roughness data was used to calibrate the calculator.

Table 5.3 gives duct roughness factors for materials other than the galvanized steel used in the friction chart of Figure 5.51. Figure 5.54 is a chart of correction factors to be used with other ducts. Illustrative examples should make their use clear.

Example 5.8 A technologist using a duct calculator based on the old average smoothness duct arrives at the following data.

Duct Section	Q , cfm	Diameter, in.	$f/100$ ft
A	1000	12	0.20
B	600	10	0.195
C	200	6	0.32

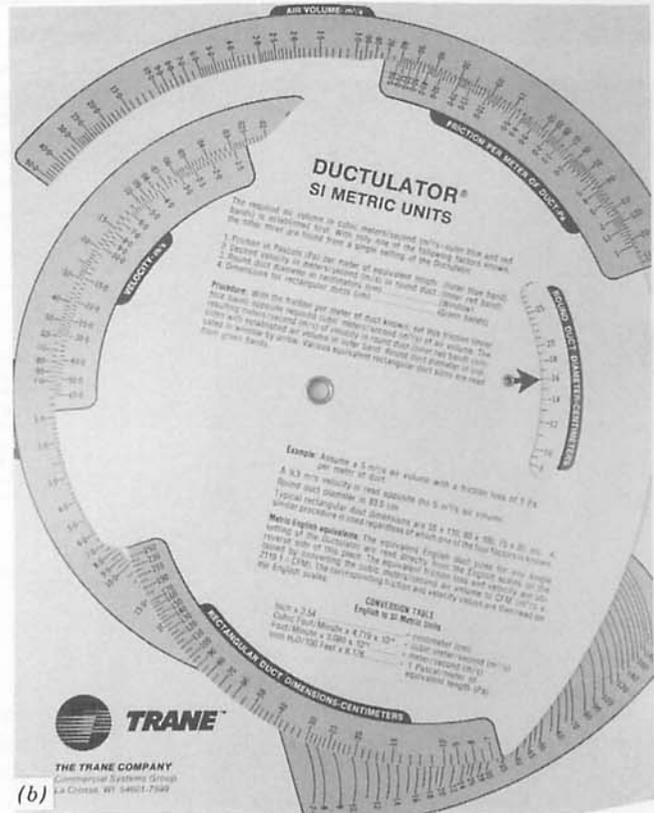
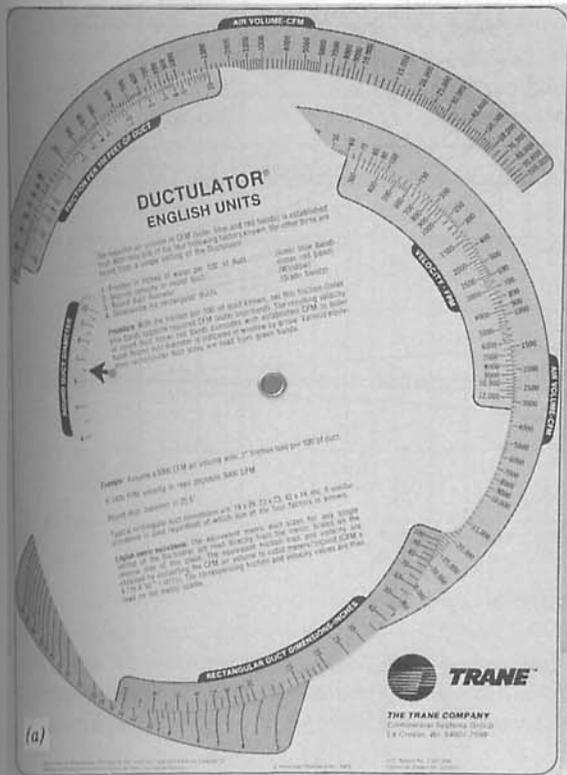


Figure 5.52 A popular duct friction calculator has English units on one side (a) and metric units on the reverse side (b). Scales on the calculator represent air flow Q , air velocity V , friction, round duct size and equivalent rectangular duct size.

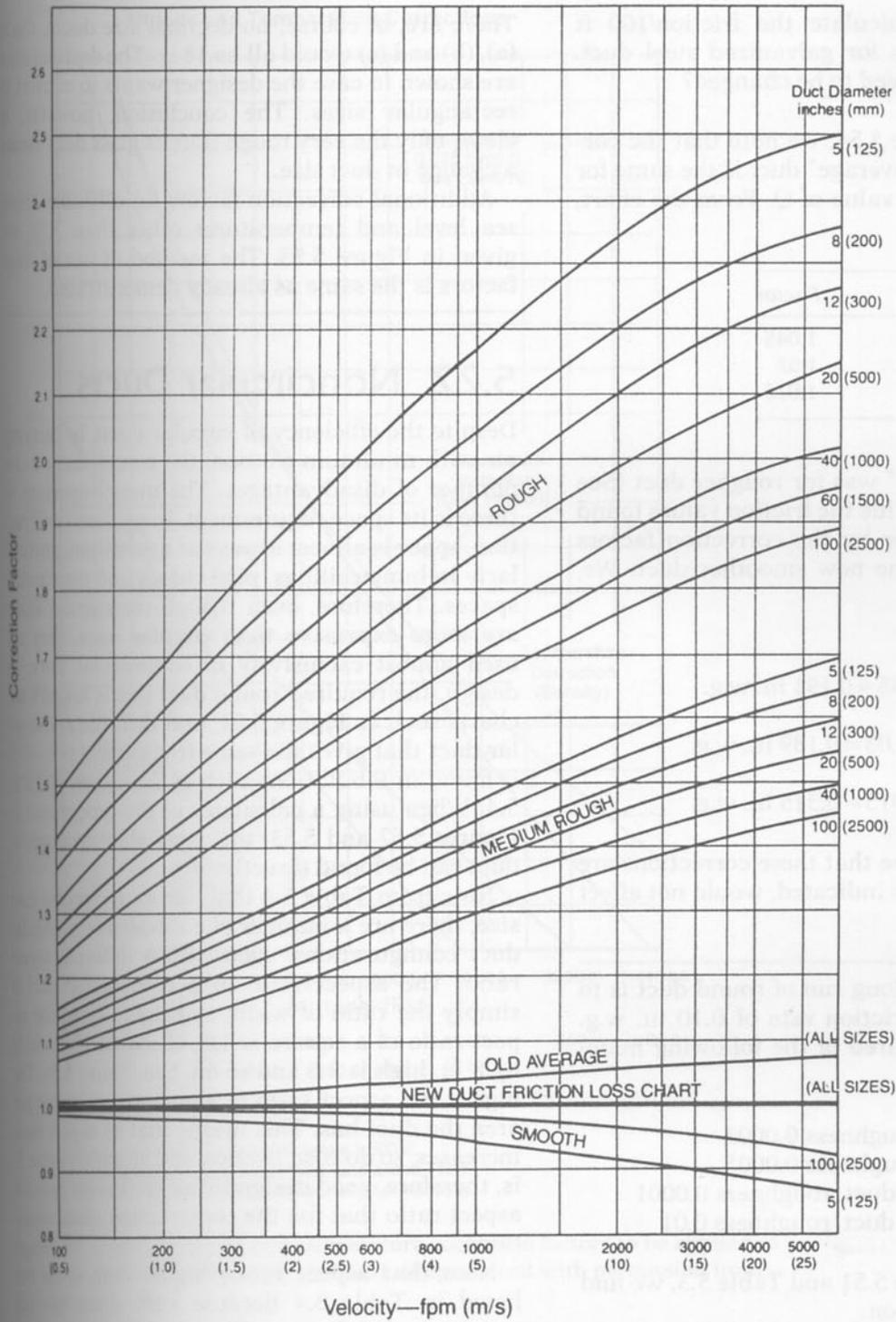


Figure 5.54 Chart of correction factors to be applied to friction data of Figure 5.51 when using ducts of roughness other than medium smooth. See Table 5.3. (Reproduced with permission from *SMACNA HVAC Systems Duct Design Manual*, 1990.)

Using Figure 5.54, calculate the friction/100 ft based on current data for galvanized steel duct. Would the duct sizes need to be changed?

Solution: From Figure 5.54, we note that the correction curve for "old average" duct is the same for all duct sizes for each value of Q . From the chart, we have:

Q, cfm	Factor
1000	1.045
600	1.03
200	1.015

Since the "old average" was for rougher duct (See Table 5.3), we must divide the friction values found with the old calculator by the correction factors to obtain friction of the new smoother duct. We, therefore, have:

Duct section A:

$$\text{friction}/100 \text{ ft} = 0.2/1.045 = 0.191 \text{ in. w.g.}$$

Duct section B:

$$\text{friction}/100 \text{ ft} = 0.195/1.03 = 0.189 \text{ in. w.g.}$$

Duct section C:

$$\text{friction}/100 \text{ ft} = 0.32/1.015 = 0.315 \text{ in. w.g.}$$

We can immediately see that these corrections are minor and, for the sizes indicated, would not affect the choice of duct size.

Example 5.9 A 100 ft long run of round duct is to carry 2000 cfm at a friction rate of 0.10 in. w.g. What size duct is required of the following materials:

- (a) galvanized steel, roughness 0.0003
- (b) galvanized steel, roughness 0.0005
- (c) PVC thermoplastic duct, roughness 0.0001
- (d) Fibrous glass-lined duct, roughness 0.01

Solution: Using Figure 5.51 and Table 5.3, we find the following information.

Duct Material	Friction Desired	Duct Size, in.	Correction Factor	Chart Friction	Duct Size, in.
(a)	0.10	18	1.0	0.095	17.9
(b)	0.10		1.06	0.094	18
(c)	0.10		0.91	0.11	17.5
(d)	0.10		1.92	0.052	20.4

There are, of course, no decimal size ducts. Ducts (a), (b) and (c) would all be 18 in. The decimal sizes are shown in case the designer wants to convert to rectangular sizes. The conclusion, however, is clear; only the very rough fibrous glass duct causes a change in duct size.

Additional correction factors for altitudes above sea level and temperatures other than 70°F are given in Figure 5.55. The method of use of these factors is the same as already demonstrated.

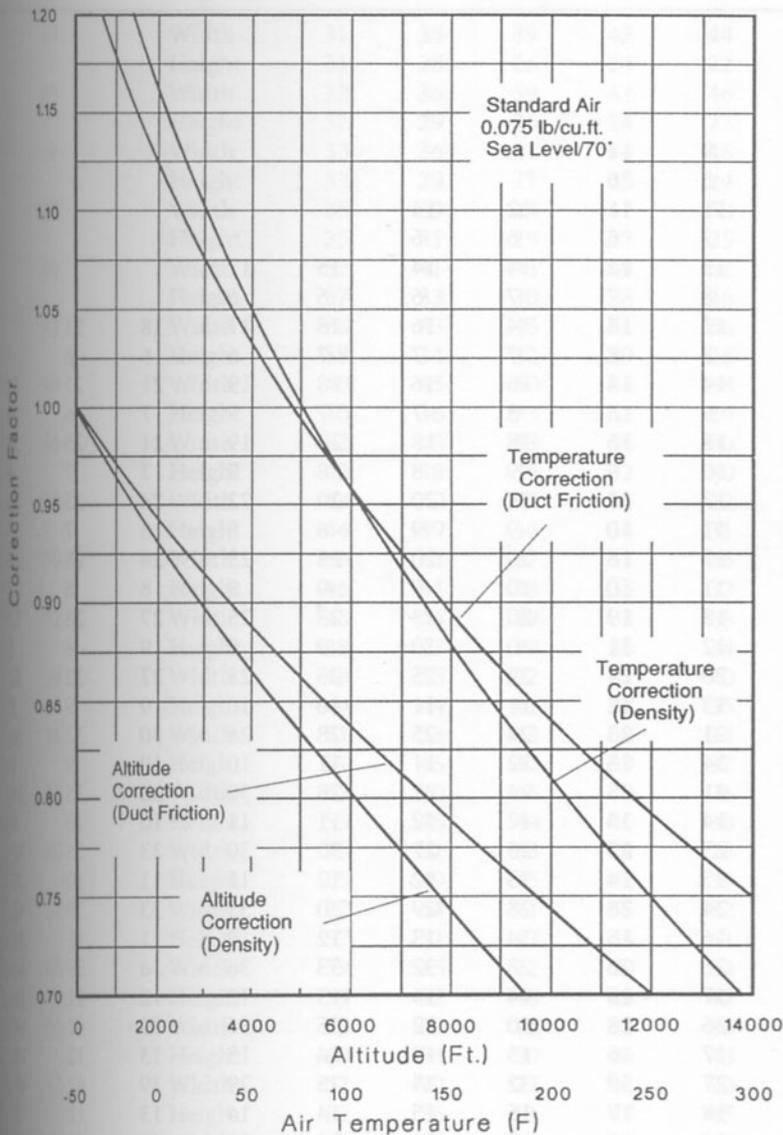
5.22 Noncircular Ducts

Despite the efficiency of circular ducts in carrying air with minimum friction, the round shape has a number of disadvantages. The most important of these is its space requirement. In modern construction, space is almost always at a premium, particularly in hung ceilings, pipe chases and mechanical spaces. Therefore, even though rectangular ducts are more expensive than circular ones, they are used almost exclusively in commercial work. In design, the required round duct size is found from charts such as Figure 5.51, and then the rectangular duct that gives the same friction loss per 100 ft is found in tabulations such as that given in Table 5.4. When using a calculator of the type shown in Figures 5.52 and 5.53, this equivalent rectangular duct can be found directly.

Note from Table 5.4 that, for each circular duct size, there are a number of equivalent rectangular duct configurations, each with a different aspect ratio. The aspect ratio of a rectangular duct is simply the ratio of width to height. Thus, the aspect ratio of a square is 1.0, of a duct 16 in. wide by 8 in. high is 2.0 and so on. See Figure 5.56. The higher the aspect ratio is, the more perimeter per area the duct has. This means that as aspect ratio increases, so do cost, friction and vibration noise. It is, therefore, good design to use the lowest possible aspect ratio that fits the construction space conditions.

Note that aspect ratios higher than 4 are not listed in Table 5.4 because such ducts are not recommended for use. They can be built and, in special cases, are used. In large sizes, internal supports must be used to keep the long dimension from sagging and vibrating. This increases costs radically. Also, fittings for high aspect ratio ducts are expensive and very inefficient (high pressure losses). Simply as a matter of interest, relative installed costs of ducts, taking square ducts (1.0 aspect ratio) as 100%, are:

Altitude and Temperature Corrections



Notes:

- Altitude correction not required below 1,500 ft.
- Temperature correction below 40 F and above 110 F
- Correction for duct friction includes affect of viscosity.

Figure 5.55 Graph of altitude and temperature correction factors to be applied to duct friction data obtained from Figure 5.51. (Reproduced with permission from ACCA Manual Q, 1990.)

Table 5.4 Equivalent Rectangular Duct Dimension

Duct Diameter, in.	Rectangular Size, in.	Aspect Ratio																			
		1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00	3.50										
6	Width	—	6																		
	Height	—	5																		
7	Width	6	8																		
	Height	6	6																		
8	Width	7	9	9	11																
	Height	7	7	6	6																
9	Width	8	9	11	11	12	14														
	Height	8	7	7	6	6	6														
10	Width	9	10	12	12	14	14	15	17												
	Height	9	8	8	7	7	6	6	6												
11	Width	10	11	12	14	14	16	18	17	18	21										
	Height	10	9	8	8	7	7	7	6	6	6										
12	Width	11	13	14	14	16	16	18	19	19	21	21									
	Height	11	10	9	8	8	7	7	7	7	7	6									
13	Width	12	14	15	16	18	18	20	19	21	21	25									
	Height	12	11	10	9	9	8	8	7	7	7	7									
14	Width	13	14	17	18	18	20	20	22	24	25	25									
	Height	13	11	11	10	9	9	8	8	8	8	7									
15	Width	14	15	17	18	20	20	23	25	24	28	28									
	Height	14	12	11	10	10	9	9	9	8	8	8									
16	Width	15	16	18	19	20	23	23	25	27	28	28									
	Height	15	13	12	11	10	10	9	9	9	9	8									
17	Width	16	18	20	21	22	25	25	28	27	32	32									
	Height	16	14	13	12	11	11	10	10	9	9	9									
18	Width	16	19	21	23	24	25	28	28	30	32	32									
	Height	16	15	14	13	12	11	11	10	10	10	9									
19	Width	17	20	21	23	24	27	28	30	30	35	35									
	Height	17	16	14	13	12	12	11	11	10	10	10									
20	Width	18	20	23	25	26	27	30	30	33	35	35									
	Height	18	16	15	14	13	12	12	11	11	10	10									
21	Width	19	21	24	26	28	29	30	33	33	39	39									
	Height	19	17	16	15	14	13	12	12	11	11	10									
22	Width	20	23	26	26	28	32	33	36	36	39	39									
	Height	20	18	17	15	14	14	13	13	12	11	11									
23	Width	21	24	26	28	30	32	35	36	39	42	42									
	Height	21	19	17	16	15	14	14	13	13	12	12									
24	Width	22	25	27	30	32	34	35	39	39	42	42									
	Height	22	20	18	17	16	15	14	14	13	12	12									
25	Width	23	25	29	30	32	36	38	39	42	46	46									
	Height	23	20	19	17	16	16	15	14	14	13	12									
26	Width	24	26	30	32	34	36	38	41	42	46	46									
	Height	24	21	20	18	17	16	15	15	14	13	12									
27	Width	25	28	30	33	36	38	40	41	45	49	49									
	Height	25	22	20	19	18	17	16	15	15	14	13									
28	Width	26	29	32	35	36	38	43	44	45	49	49									
	Height	26	23	21	20	18	17	17	16	15	14	13									
29	Width	27	30	33	35	38	41	43	44	48	53	53									
	Height	27	24	22	20	19	18	17	16	16	15	14									
30	Width	27	31	35	37	40	43	45	47	48	53	53									
	Height	27	25	23	21	20	19	18	17	16	15	14									
31	Width	28	31	35	39	40	43	45	50	51	56	56									
	Height	28	25	23	22	20	19	18	18	17	16	15									
32	Width	29	33	36	39	42	45	48	50	54	56	56									
	Height	29	26	24	22	21	20	19	18	18	17	16									
33	Width	30	34	38	40	44	47	50	52	54	60	60									
	Height	30	27	25	23	22	21	20	19	18	17	16									

Table 5.4 (Continued)

Duct Diameter, in.	Rectangular Size, in.	Aspect Ratio										
		1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00	3.50	4.00
34	Width	31	35	39	42	44	47	50	52	57	60	64
	Height	31	28	26	24	22	21	20	19	19	17	16
35	Width	32	36	39	42	46	50	53	55	57	63	68
	Height	32	29	26	24	23	22	21	20	19	18	17
36	Width	33	36	41	44	48	50	53	55	60	63	68
	Height	33	29	27	25	24	22	21	20	20	18	17
38	Width	35	39	44	47	50	54	58	61	63	67	72
	Height	35	31	29	27	25	24	23	22	21	19	18
40	Width	37	41	45	49	52	56	60	63	66	70	76
	Height	37	33	30	28	26	25	24	23	22	20	19
42	Width	38	43	48	51	56	59	63	66	69	74	80
	Height	38	34	32	29	28	26	25	24	23	21	20
44	Width	40	45	50	54	58	61	65	69	72	81	84
	Height	40	36	33	31	29	27	26	25	24	23	21
46	Width	42	48	53	56	60	65	68	72	75	84	88
	Height	42	38	35	32	30	29	27	26	25	24	22
48	Width	44	49	54	60	62	68	70	74	78	88	92
	Height	44	39	36	34	31	30	28	27	26	25	23
50	Width	46	51	57	61	66	70	75	77	81	91	96
	Height	46	41	38	35	33	31	30	28	27	26	24
52	Width	48	54	59	63	68	72	78	83	84	95	100
	Height	48	43	39	36	34	32	31	30	28	27	25
54	Width	49	55	62	67	70	77	80	85	90	98	104
	Height	49	44	41	38	35	34	32	31	30	28	26
56	Width	51	58	63	68	74	79	83	88	93	102	108
	Height	51	46	42	39	37	35	33	32	31	29	27
58	Width	53	60	66	70	76	81	85	91	96	105	112
	Height	53	48	44	40	38	36	34	33	32	30	28
60	Width	55	61	68	74	78	83	90	94	99	109	116
	Height	55	49	45	42	39	37	36	34	33	31	29
62	Width	57	64	71	75	82	88	93	96	102	112	120
	Height	57	51	47	43	41	39	37	35	34	32	30
64	Width	59	65	72	79	84	90	95	99	105	116	124
	Height	59	52	48	45	42	40	38	36	35	33	31
66	Width	60	68	75	81	86	92	98	105	108	119	128
	Height	60	54	50	46	43	41	39	38	36	34	32
68	Width	62	70	77	82	90	95	100	107	111	123	132
	Height	62	56	51	47	45	42	40	39	37	35	33
70	Width	64	71	80	86	92	99	105	110	114	126	136
	Height	64	57	53	49	46	44	42	40	38	36	34
72	Width	66	74	81	88	94	101	108	113	117	130	140
	Height	66	59	54	50	47	45	43	41	39	37	35
74	Width	68	76	84	91	98	104	110	116	123	133	144
	Height	68	61	56	52	49	46	44	42	41	38	36
76	Width	70	78	86	93	100	106	113	118	126	137	148
	Height	70	62	57	53	50	47	45	43	42	39	37
78	Width	71	80	89	95	102	110	115	121	129	140	152
	Height	71	64	59	54	51	49	46	44	43	40	38
80	Width	73	83	90	98	104	113	118	124	132	144	156
	Height	73	66	60	56	52	50	47	45	44	41	39

Source: Data extracted and reprinted by permission of the American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, Georgia, from the 1993 *ASHRAE Handbook—Fundamentals*.

Dimension	12" Diam.	11" x 11"	16" x 8"	24" x 6"	Major axis - 15" Minor axis - 9"
Aspect ratio	—	1.0	2.0	4.0	—
Perimeter (in.)	37.7	44	48	60	38.3
Area (in. ²)	113.1	121	128	144	106
Ratio of perimeter: area	0.333	0.364	0.375	0.417	0.361
Air velocity (constant CFM)	100%	93.4%	88.4%	78.5%	106.7%

Figure 5.56 Comparative characteristics of equivalent friction ducts. Duct cost is proportional to the quantity of metal used, that is, the perimeter. Note that if air volume (cfm) is held constant, the air velocity will drop as cross-sectional duct area increases. See Section 5.23. Duct equivalent sizes are taken from Tables 5.4 and 5.5.

Aspect ratio	1	2	3	4	5	5	7
Cost, %	100	115	130	145	165	185	210

In addition, since higher aspect ratio means higher friction, it also means more energy use by the blower and, therefore, higher operating costs.

Oval ducts have recently become fairly popular, particularly in residential work. This is because

the clear space in a stud construction wall is only 3⁵/₈ in., which will accept only a 3-in. round duct. However, oval ducts as large as 3 x 15 in. will fit into a stud wall with studs 16 in. on centers. This is equivalent to a 7-in. round duct. Oval equivalents to circular ducts are given in Table 5.5. Figure 5.57 shows the use of circular, rectangular and oval ducts in a typical installation situation.

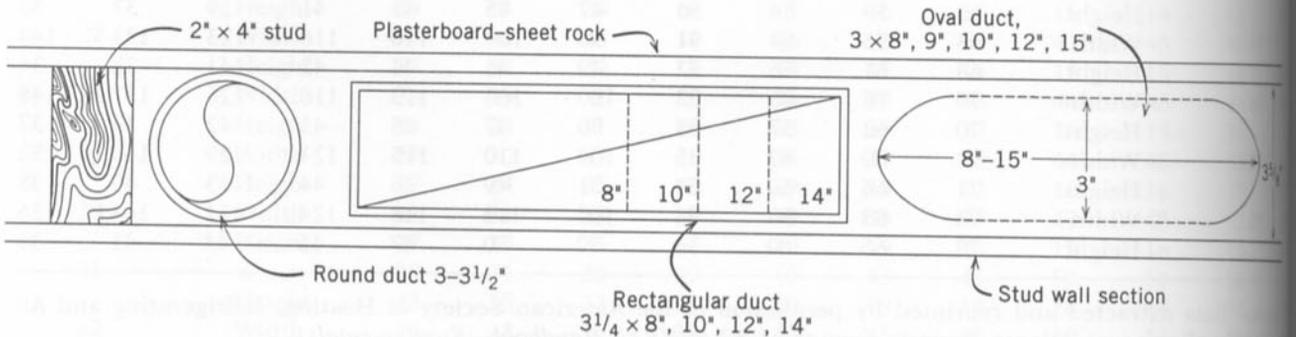


Figure 5.57 The restricted width of a stud wall (3⁵/₈ in.) requires use of rectangular or oval ducts. Rectangular ducts for use as stacks in stud walls are available in the sizes shown.

Table 5.5 Size of Equivalent^a Oval Duct to Circular Duct

Equivalent Circular Duct Diameter, in.	Minor Axis, in.												
	3	4	5	6	7	8	9	10	11	12	14	16	
5	8												
5.5	9	7											
6	11	9											
6.5	12	10	8										
7	15	12	10	8									
7.5	19	13	—	9									
8	22	15	11	—									
8.5		18	13	11	10								
9		20	14	12	—	10							
9.5		21	18	14	12	—							
10			19	15	13	11							
10.5			21	17	15	13	12						
11				19	16	14	—	12					
11.5				20	18	16	14	—					
12				23	20	17	15	13					
12.5				25	21	—	—	15	14				
13				28	23	19	17	16	—	14			
13.5				30	—	21	18	—	16	—			
14				33	—	22	20	18	17	15			
14.5				36	—	24	22	19	—	17			
15						27	23	21	19	18			
16						30	—	24	22	20	17		
17						35	—	27	24	21	19		
18						39	—	30	—	25	22	19	
19						46	—	34	—	28	23	21	19
20						50	—	38	—	31	27	24	21
21								43	—	34	28	25	21
22										48	—	31	29
23										52	—	34	30
24												45	33
25												50	36
26												56	38
27													49
28													52
29													58
30													61
31													
32													
33													

^aEquivalent duct friction.

Source: Data extracted and reprinted by permission of the American Society of Heating, Refrigerating and Air Conditioning Engineers, Atlanta, Georgia, from the 1993 ASHRAE Handbook—Fundamentals.

5.23 Air Friction in Duct Fittings

A duct system consists of straight sections and fittings. In Sections 5.21 and 5.22, we learned how to calculate the friction loss in straight sections of ducts of various shapes. We will now learn how to do this calculation for fittings. The term *fittings* applies to every part of a duct system except straight duct sections of unchanging size. Fittings, therefore, include transitions, inlets, outlets, elbows, angles, offsets, wyes, tees, dovetails, branches, exit connections and so on. Indeed, the list is so long that ASHRAE has developed a computer duct fitting data base (1993) to assist designers in duct system calculations.

On the average, pressure losses in fittings comprise at least one-half of the total pressure loss in a system and sometimes as much as 75%. It is, therefore, apparent that these losses must be carefully calculated in preparing any but the smallest and simplest duct system. There are two methods of determining the pressure loss in a fitting—equivalent length and loss coefficient calculation. Equivalent length should be used only when there is no difference in air velocity between the entrance and the exit of the fitting. Loss coefficient calculations are applicable in all situations. To understand why this is so, we must first study the effect of duct (or fitting) cross-sectional area on the velocity of airflow.

a. Air Velocity in Ducts

Refer to Figure 5.58. In the section of duct illustrated there is an air flow of Q cfm at a velocity of

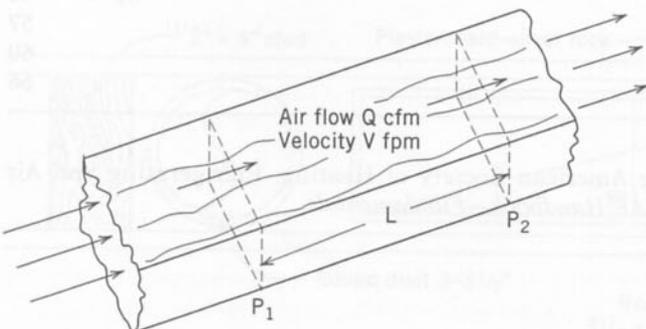


Figure 5.58 Q cfm of air volume traveling through a duct of A ft² cross-sectional area, will move at V ft/min, where $Q = AV$. See text for derivation of this relation.

V fpm. The volume of air flowing past section P_1 in 1 min, when the air velocity is 1 fpm is a column of air 1 ft long of cross-sectional area A . Numerically, the volume of this column is the area A ft² times 1 ft length, or A ft³. Since this volume flows past in 1 min, the flow Q is A ft³/min (or cfm).

If the air velocity were 2 fpm, the column would be 2 ft long and its volume would be $2A$ ft³. Since it too flows by in one minute, the flow Q would then be $2A$ ft³/min (or cfm). Therefore, if the air velocity is V , the volume of the air column passing section P_1 in 1 min would be VA ft³, and the flow rate Q would be VA ft³/min or AV cfm.

We have, therefore, developed the fundamental flow equation:

$$Q = AV \quad (5.7)$$

where

Q is the volume of air flow in cubic feet per minute,

A is the cross-sectional area of the duct in square feet and

V is the air velocity in feet per minute.

An example will help you understand this extremely useful equation.

Example 5.10 A design technologist has the choice of using a 10-in. round duct or a 12 × 7-in. rectangular duct to carry 350 cfm of conditioned air. Both give the same friction (see Table 5.4). Calculate the air velocity in each.

Solution: We will use Equation (5.7), remembering to convert duct area into square feet. For a round 10-in. duct:

$$A = \frac{\pi D^2}{4} = 0.7854D^2 = 0.7854(10)^2 \text{ in.}^2$$

$$A = 78.54 \text{ in.}^2 \times \frac{\text{ft}^2}{144 \text{ in.}^2} = 0.545 \text{ ft}^2$$

$$V = \frac{Q}{A} = \frac{350 \text{ ft}^3/\text{min}}{0.545 \text{ ft}^2} = 642 \text{ ft/min}$$

For a rectangular 12 × 7-in. duct:

$$A = 12 \text{ in.} \times 7 \text{ in.} = 84 \text{ in.}^2 \times \frac{\text{ft}^2}{144 \text{ in.}^2} = 0.583 \text{ ft}^2$$

$$V = \frac{350 \text{ ft}^3/\text{min}}{0.583 \text{ ft}^2} = 600 \text{ ft/min}$$

The numbers in Example 5.10 were chosen to demonstrate a very important point. When using the friction chart of Figure 5.51, the air velocity shown is that of air in a round duct. In this case,

an inexperienced technologist might be inclined to select a larger round duct because, in residential work, it is recommended that air velocities above 600 fpm be avoided, due to noise. He or she would then find the equivalent (oversized) rectangular duct. What should be done is first to check air velocity in the rectangular duct that is equivalent to the original (10-in.) round duct. It is always lower than that in a round duct. In this case, it is down to 600 fpm, which meets the recommendation limit. An equivalent rectangular duct of higher aspect ratio would have an even lower air velocity. (We are assuming, of course, that air flow Q is held constant at the design value.)

b. Equivalent Duct Lengths for Fitting Losses

We stated previously that one of the two methods for figuring fitting static pressure loss is by using an equivalent length of straight duct. We also stated that this method should be used only where fitting input and output velocities are the same. We will now explain why this is so. Refer to Figure 5.59(a), which shows graphically the pressure loss in a straight section of duct. Total pressure at any point, as we learned in Section 5.4, is the sum of static pressure and velocity pressure. This means that at the inlet of the duct

$$P_{\text{inlet}} = P_{S1} + P_{V1}$$

and at the outlet

$$P_{\text{outlet}} = P_{S2} + P_{V2}$$

where

- P_{S1} is static pressure at point 1,
- P_{S2} is static pressure at point 2,
- P_{V1} is velocity pressure at point 1 and
- P_{V2} is velocity pressure at point 2.

However, since the size of the duct does not change nor does the air volume Q that flows, the outlet air velocity V_2 must equal the inlet velocity V_1 , because

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

and both Q and A remain unchanged. Since we know that the velocity pressure is

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

it follows that velocity pressure is the same at the outlet as at the inlet. Therefore, all the pressure

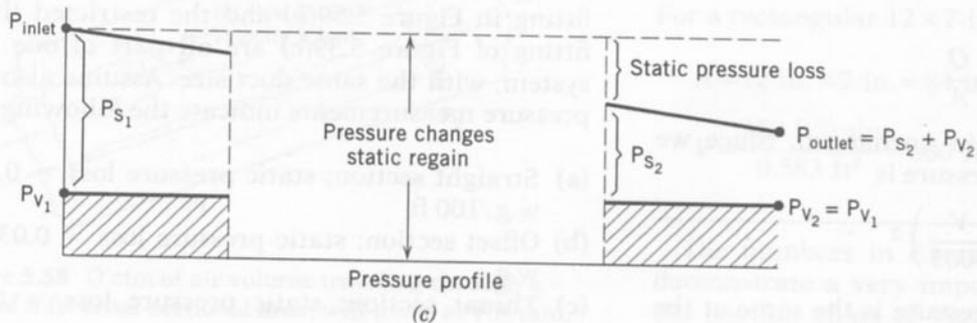
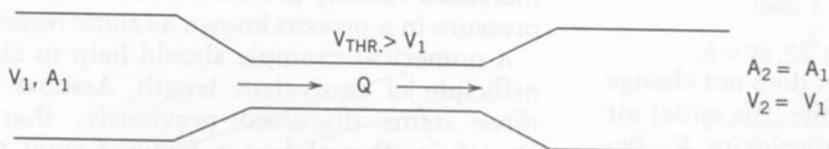
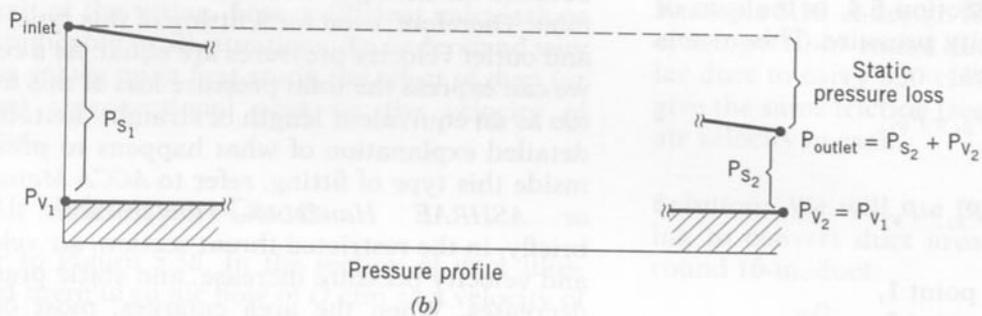
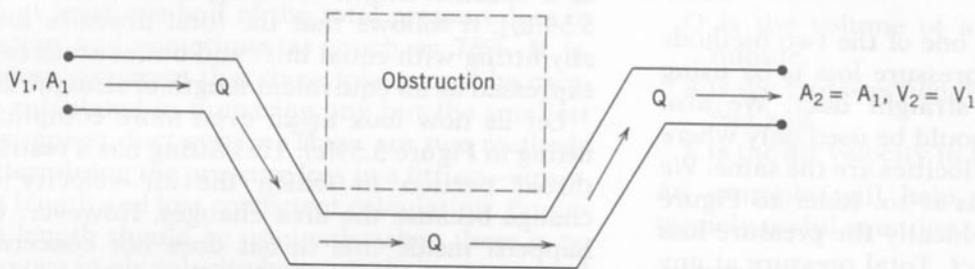
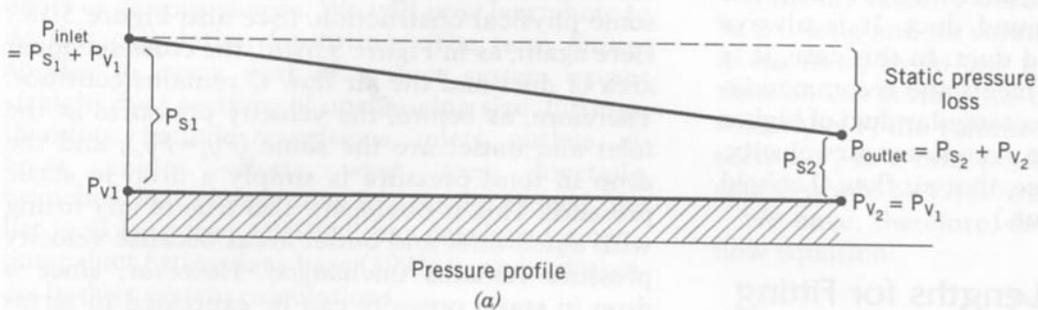
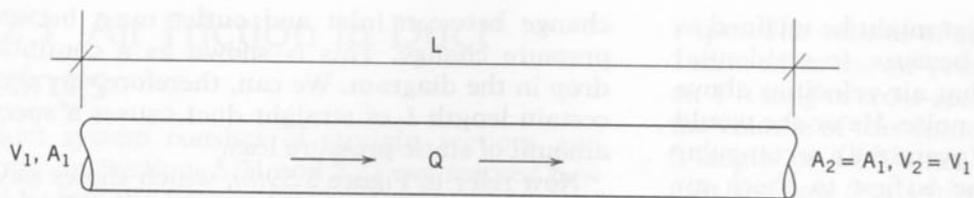
change between inlet and outlet must be static pressure change. This is shown as a continuous drop in the diagram. We can, therefore, say that a certain length L of straight duct causes a specific amount of static pressure loss.

Now refer to Figure 5.59(b), which shows a typical duct offset fitting of the type used to dip under some physical obstruction. (See also Figure 5.18.) Here again, as in Figure 5.59(a), the cross-sectional area of duct and the air flow Q remains constant. Therefore, as before, the velocity pressures at the inlet and outlet are the same ($P_{V2} = P_{V1}$), and the drop in total pressure is simply a drop in static pressure. This is obviously also true of any fitting with equal inlet and outlet areas because velocity pressure remains unchanged. However, since a drop in static pressure can be expressed in terms of a specific length of straight duct [see Figure 5.59(a)], it follows that the total pressure loss in any fitting with equal inlet and outlet areas can be expressed as an equivalent length of straight duct.

Let us now look at an even more complicated fitting in Figure 5.59(c). This fitting has a restricted throat portion in which the air velocity does change because the area changes. However, what happens inside this throat does not concern us, because the fitting outlet has the same area as the inlet. Therefore, even for a fitting of this type, inlet and outlet velocity pressures are equal. As a result, we can express the total pressure loss of this fitting too as an equivalent length of straight duct. (For a detailed explanation of what happens to pressure inside this type of fitting, refer to *ACCA Manual Q* or *ASHRAE Handbook—Fundamentals*. Very briefly, in the restricted throat section, air velocity and velocity pressure increase, and static pressure decreases. When the area enlarges, most of the increased velocity pressure is reconverted to static pressure in a process known as *static regain*.)

A numerical example should help to clarify the principle of equivalent length. Assume that the three items discussed previously, that is, the straight section of duct in Figure 5.59(a), the offset fitting in Figure 5.59(b) and the restricted throat fitting of Figure 5.59(c) are all part of one duct system, with the same duct size. Assume also that pressure measurements indicate the following:

- (a) Straight section; static pressure loss = 0.1 in. w.g./100 ft
- (b) Offset section; static pressure loss = 0.035 in. w.g.
- (c) Throat section: static pressure loss = 0.075 in. w.g.



Say that the offset fitting is the equivalent of L_1 ft of straight duct, then:

$$\frac{0.035 \text{ in.}}{L_1 \text{ ft}} = \frac{0.1 \text{ in.}}{100 \text{ ft}}$$

$$L_1 = 35 \text{ ft}$$

Similarly, the throat section is equivalent to L_2 ft of straight duct:

$$\frac{0.075 \text{ in. w.g.}}{L_2} = \frac{0.1 \text{ in. w.g.}}{100 \text{ ft}}$$

$$L_2 = 75 \text{ ft}$$

Now refer to Figure 5.18 (page 231). Adjacent to each fitting sketch is a number. It represents the equivalent length of straight duct that will cause the same static friction loss as the fitting. For fittings with equal inlet and outlet areas, these equivalent lengths are accurate. For fittings with different areas, these equivalent lengths are only an approximation and should only be used in small to medium low velocity, low pressure systems.

A comprehensive listing of equivalent lengths for residential fittings is given in Appendix B. Note that some of these fittings have different inlet and outlet areas. This means that for accurate work, equivalent length should not be used. In residential work, most ducts are sized for noise criteria, which makes them larger than would be required by friction calculations. Also, at the low velocities used in residential systems, velocity pressures are almost negligible. Finally, all ducts have volume dampers that allow balancing the system after installation. As a result, the inaccuracies introduced by using equivalent length for all fittings do not result in an unworkable design for a small-to-medium-size low velocity residential type design.

c. Loss Coefficients

The loss coefficient method of figuring pressure loss in a duct fitting is always applicable because

it considers total pressure loss. This is different from the equivalent length technique, which considers only static pressure loss (because velocity pressure remains constant). Consider a common transition fitting such as shown in Figure 5.60. Note that the outlet velocity pressure is higher than the inlet velocity pressure because the air velocity is higher at the outlet than at the inlet. This is so because the outlet area is smaller than the inlet area. Remembering that $Q=AV$ and, therefore, $V=Q/A$, it follows that with Q constant, a drop in area means a corresponding rise in velocity. And, since $P_V=(V/4005)^2$, outlet velocity pressure must be higher than inlet velocity pressure. The pressure loss in this fitting is, therefore, a combination of static pressure loss and velocity pressure gain. For this reason, the equivalent length method, which considers only static pressure drop, is not accurate.

An abbreviated list of loss coefficients for common fittings appears in Appendix C. A complete list of 228 different types is available in electronic form, in a data base, from ASHRAE. In this form, the data can be used directly in any of the major duct design programs. Alternatively, the data can be used for manual calculation. Somewhat shorter lists are printed in *SMACNA Duct Design Manual*, *ACCA Manual Q* and the 1993 *ASHRAE Handbook—Fundamentals*. See the bibliography at the end of Chapter 7 for additional sources.

Pressure loss in a fitting is calculated, using its loss coefficient, as follows:

$$P_{\text{LOSS}} = P_V \times C \quad (5.8)$$

where

- P_{LOSS} is the fitting pressure loss in inches w.g.,
- P_V is the velocity pressure at the fitting in inches w.g. and
- C is the loss coefficient.

When calculating the pressure loss in a fitting with different inlet and outlet areas, as for instance a

Figure 5.59 Outlet velocity of any fitting or duct section is the same as inlet velocity provided flow is constant, and inlet and outlet areas are equal. (a) In a straight section of duct, velocity pressure is constant, and static pressure drops off uniformly due to friction. Total pressure is always the sum of static and velocity pressure. (b) Regardless of the fitting shape—in this case an underpass type—outlet velocity will equal inlet velocity if flow is constant and cross-sectional inlet and outlet areas are equal. The pressure profile does not show the pressure changes in the underpass. The net overall result is a loss of static pressure. (c) In the throat section of this fitting, velocity is high and so is velocity pressure. However, because outlet area equals inlet area, there is a partial static pressure regain and overall a net static pressure loss. See text for a full explanation.

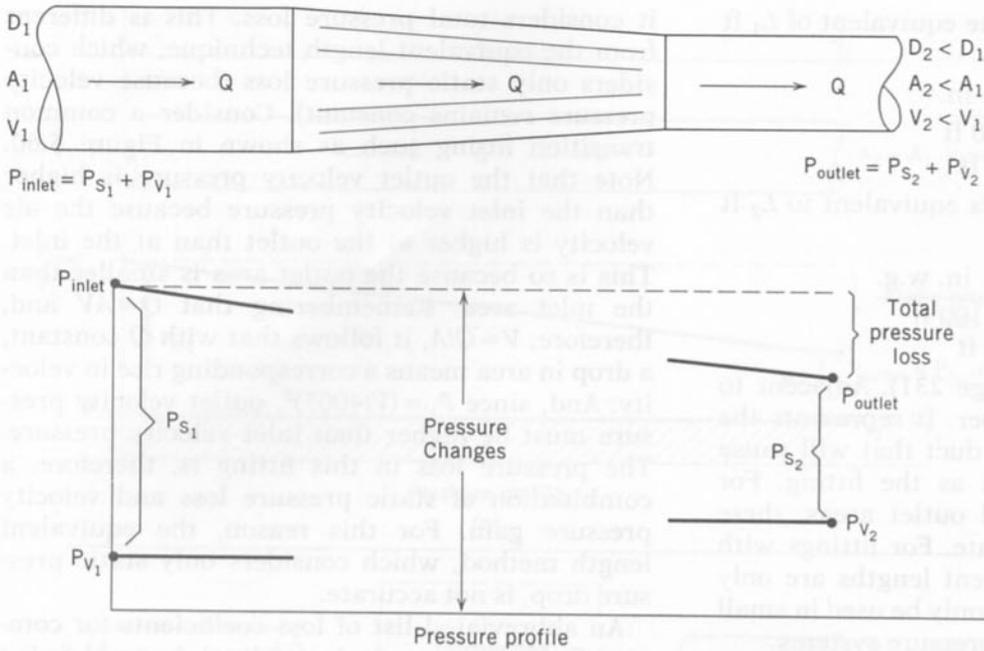
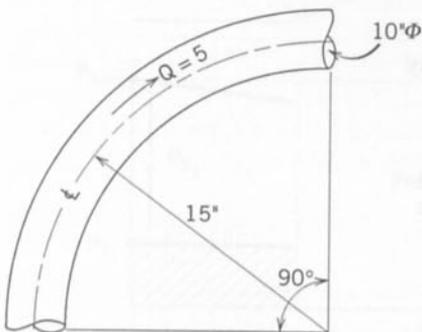


Figure 5.60 Transition fitting between round ducts of different diameters. Because the outlet diameter is smaller than the inlet diameter its area is also smaller. Therefore, the outlet velocity pressure is higher ($P_{V2} > P_{V1}$). As a result, the total fitting pressure loss is not a static pressure loss as in Figure 5.59 but a combination of static pressure loss and velocity pressure gain.

transition fitting, use the velocity pressure at the smaller opening in Equation 5.8. This will be, of course, the higher of the two pressures. Examples will make the calculation method clear.

Example 5.11 A smooth radius 10 in. diameter 90° elbow, with a radius of 15 in., carries 500 cfm. Calculate its pressure loss.



Example 5.11

Solution: We note in the Appendix C data for the fitting that an additional piece of information is required before we can select the loss coefficient, the R/D ratio. In our case, $R = 15$ in. and $D = 10$ in., so

$$\frac{R}{D} = \frac{15}{10} = 1.5$$

We then select from the table, for a 90° elbow with $R/D = 1.5$, a loss coefficient of 0.15. Using Equation 5.8, we have

$$P_{Loss} = P_V \times C = 0.15P_V$$

We know that

$$P_V = (V/4005)^2;$$

$$V = Q/A;$$

and

$$A = \frac{\pi}{4} D^2$$

We, therefore, calculate

$$A = \frac{\pi}{4} D^2 = \frac{\pi}{4} (10)^2 = 78.54 \text{ in.}^2$$

$$V = \frac{Q}{A} = \frac{500 \text{ ft}^3/\text{min}}{78.54 \text{ in.}^2 \times \text{ft}^2/144 \text{ in.}^2} = 917 \text{ ft/min}$$

$$P_v = (V/4005)^2 = (917/4005)^2 = 0.05 \text{ in.w.g.}$$

$$P_{\text{loss}} = 0.15 (.05) = 0.008 \text{ in. w.g.}$$

This is a very small loss, as we would expect from a smooth, large radius elbow.

Example 5.12 A round conical transition from 8 to 12-in. diameter duct carries 500 cfm. The cone angle is 45°. Find the fitting pressure loss.

Solution: The pressure loss is calculated using equation (5.8)

$$P_{\text{LOSS}} = P_v \times C$$

The solution procedure is, therefore,

- Step 1. Using tabular data, find loss coefficient C .
- Step 2. Calculate P_v .
- Step 3. Calculate fitting pressure loss P_{LOSS} .

We will now perform the calculation using this three-step procedure.

Step 1: In Appendix C, Section 6, we find the loss

coefficient data for the fitting. It is reproduced here for ease of reference.

From the illustration we see that the one piece of data still required is the ratio A_1/A where A_1 and A are the upstream and downstream fitting areas, respectively. (The other required data—cfm and cone angle—are given.)

$$\frac{A_1}{A} = \frac{A_{\text{upstream}}}{A_{\text{downstream}}} = \frac{(\pi/4)(D_1)^2}{(\pi/4)(D)^2} = \frac{12^2}{8^2} = \frac{144}{64} = 2.25$$

Using the three pieces of data:

$$\text{cfm} = 500$$

$$\Phi = 45^\circ$$

$$A_1/A_2 = 2.25$$

we find from the table by interpolation:

A_1/A	Coefficients
2	0.33
2.25	C
4	0.61

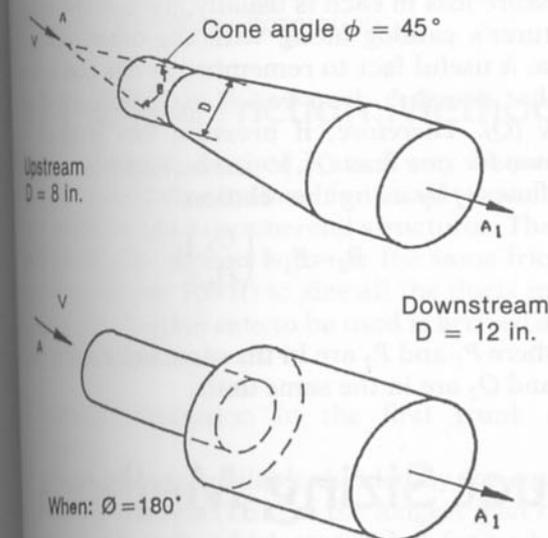
Appendix C - Section 6

Loss Coefficients, Transitions (Diverging Flow)

Use the velocity (V_c) in the upstream section to determine the reference velocity pressure (P_v)

$$P_t = C \times P_v \text{ (In. Wg.)}$$

A) Transition, Round, Conical (Upstream P_v)



Coefficient C

CFM	A_1/A	Coefficient C							
		16'	20'	30'	45'	60'	90'	120'	180'
<2400	2	0.14	0.19	0.32	0.33	0.33	0.32	0.31	0.30
	4	0.23	0.30	0.46	0.61	0.68	0.64	0.63	0.62
	6	0.27	0.33	0.48	0.66	0.77	0.74	0.73	0.72
	10	0.29	0.38	0.59	0.76	0.80	0.83	0.84	0.83
	>16	0.31	0.38	0.60	0.84	0.88	0.88	0.88	0.88
2400 to 12000	2	0.07	0.12	0.23	0.28	0.27	0.27	0.27	0.26
	4	0.15	0.18	0.36	0.55	0.59	0.59	0.58	0.57
	6	0.19	0.28	0.44	0.90	0.70	0.71	0.71	0.69
	10	0.20	0.24	0.43	0.76	0.80	0.81	0.81	0.81
	>16	0.21	0.28	0.52	0.76	0.87	0.87	0.87	0.87
>12000	2	0.05	0.07	0.12	0.27	0.27	0.27	0.27	0.27
	4	0.17	0.24	0.38	0.51	0.56	0.58	0.58	0.57
	6	0.16	0.29	0.46	0.60	0.69	0.71	0.70	0.70
	10	0.21	0.33	0.52	0.60	0.76	0.83	0.84	0.83
	>16	0.21	0.34	0.56	0.72	0.79	0.85	0.87	0.89

$$C = 0.33 + \frac{2.25 - 2}{4 - 2} \times (0.61 - 0.33)$$

$$C = 0.365$$

Step 2: Calculate P_v .

We know from Equation (5.6) that $P_v = (V/4005)^2$, and $V = Q/A$. As stated previously, the air velocity is calculated at the smaller opening. Therefore,

$$V = \frac{Q}{A} = \frac{500 \text{ cfm}}{(\pi/4)D^2}$$

Since Q is stated in cubic feet per minute, and V is required in feet per minute, we must convert the fitting diameter from inches to feet.

$$8 \text{ in.} \times \frac{1 \text{ ft}}{12 \text{ in.}} = 0.667 \text{ ft}$$

Then

$$V = \frac{Q}{A} = \frac{500 \text{ cfm}}{(\pi/4)D^2} = \frac{500 \text{ cfm}}{0.785(0.667)^2 \text{ ft}^2}$$

$$V = 1432 \text{ fpm}$$

Therefore,

$$P_v = (V/4005)^2 = (1432/4005)^2 = 0.128 \text{ in. w.g.}$$

Step 3: Calculate P_{LOSS} .

$$P_{\text{LOSS}} = P_v \times C = 0.128 \text{ in. w.g.} \times 0.364$$

$$P_{\text{LOSS}} = 0.047 \text{ in. w.g.}$$

This is an appreciable pressure loss.

The preceding calculations serve two purposes. The first is to demonstrate the loss coefficient method of pressure loss calculation for fittings. The second is to give you an appreciation of the value of a fitting data base and a computer program that performs all of these laborious arithmetic calculations in the twinkling of an eye.

5.24 Sources of Duct System Pressure Loss

There are five principal sources of pressure loss in a duct system. They are:

- Straight sections of duct.
- Duct fittings.
- Supply outlets and return inlets.
- Blower inlet and outlet structures.
- Air system devices.

We have studied the first three items in some detail, and you should be able, at this point, to determine the pressure losses in each category.

The blower (or fan) inlet and outlet structures are the plenums that are used to connect ductwork to the air-handling unit. In residential work, where

velocities are low and plenums are simple, losses are very low, rarely exceeding 0.05 in. w.g. In commercial work, losses can be as high as 0.4 in. w.g. for inlets with sharp changes of direction. The inlet and outlet losses are frequently referred to in the literature as the "system effect." Loss coefficients for various inlet and outlet configurations are given in the previously referenced SMACNA, ACCA and ASHRAE publications. They should be consulted for all commercial work and for large residential designs.

The final category of pressure loss sources are known as air side devices. These include:

- Filters of all types, including air washers
- Humidifiers
- Heat exchangers, including energy recovery devices and duct heaters
- Dampers, air-flow controls and smoke control devices
- Louvers and screens
- Sound traps and acoustic linings
- Heating and cooling coils, including DX cooling coils, steam and hot water coils and electrical heating elements
- Air distribution equipment, including mixing boxes of all types and valves
- Monitoring devices and measuring equipment permanently installed in the airflow

Here, too, most of these items are found only in commercial equipment. Since it is assumed that an HVAC technologist will work on many projects, including large commercial ones, he or she should be aware of these pressure loss sources. The actual pressure loss in each is usually given in the manufacturer's catalog along with the other technical data. A useful fact to remember in this connection is that pressure drop varies as the square of air flow (Q). Therefore, if pressure loss is given or known for one flow Q_1 , it can be found for another air flow Q_2 by using the relation

$$P_2 = P_1 \times \left(\frac{Q_2}{Q_1} \right)^2 \quad (5.9)$$

where P_1 and P_2 are in the same units and Q_1 and Q_2 are in the same units.

Duct Sizing Methods

There are two extremely detailed and time-consuming procedures in the design of an HVAC sys-

tem. The first is the determination of heat losses and gains, as we studied in Chapter 2. The second, when designing an all-air system, is the duct-sizing procedure. Before the advent of computers, forms and schedules were used (and still are) in an effort to systematize and simplify these complex and wearying calculations. The advantage of using a prepared form or schedule is that it forces you to plug in the numbers in the right places, making it difficult (but far from impossible) to make a mistake. A computer program does exactly the same thing, but with the great additional advantage that it does all the calculating. The disadvantage of both, at least for a beginner, is that they make the design procedure mechanical, and this can lead to errors. Since most of you are novice designers, we will avoid extensive design examples that may overwhelm you with numbers in favor of small sectional designs that clearly demonstrate the methods involved.

There are four duct-sizing methods in common use for design of single-zone, low velocity systems. They are:

- Equal friction method
- Modified equal friction method
- Extended plenum method
- Semi-extended plenum (reducing plenum) method

For larger, complex and/or high velocity systems, duct-sizing methods include static regain, constant velocity and the T-method, among others. These methods require considerable experience, are usually done by computer and are beyond the scope of this text. Refer to the bibliography at the end of this chapter for more information.

5.25 Equal Friction Method

The equal friction method is used very frequently in the design of duct systems for small to medium-size residences and commercial structures. The basic idea of this method is to use the same friction rate (friction per 100 ft) to size all the ducts in the system. The friction rate to be used is arrived at by one of three methods:

- Velocity limitation in the first trunk duct section
- Total pressure available divided by the equivalent total length (TEL) of the longest duct run
- Rule of thumb, which states that for such systems the friction rate should be between 0.08 and 0.12 in. w.g. per 100 ft.

An example of the use of the equal friction method should help to make its application clear.

Example 5.13 Use the equal friction method to size the ducts shown in Figure 5.61(a). Justify all assumptions.

Solution: The example could represent the supply duct layout for a medium-size ranch-style residence or a single-level commercial building. We will assume that this is a residential installation. The design steps that precede preparation of the duct diagram will be detailed later in this chapter, in the discussion of overall design procedure.

- (1) *Determine friction rate to be used.* Consult Table 5.6, which lists maximum velocities permitted for ductwork and outlets in residential installations. (Table 5.7 gives the same data for other types of buildings.) We will use the maximum trunk velocity permitted, because the duct is insulated and is relatively small. Both of these factors help reduce the noise generated and transmitted by the ductwork. Using either the chart in Figure 5.51 or one of the calculators shown in Figures 5.52 or 5.53 we determine that a flow of 1050 cfm at 900 fpm gives a friction rate of 0.08 in. w.g./100 ft. This friction rate is just at the edge of the shaded recommended design area of Figure 5.51 and complies with the lower value of the rule-of-thumb range (0.08–0.12) listed previously.

As a further check on this proposed friction rate, we should work out the third method listed, that is, divide the pressure available by the TEL of the longest duct run. Referring to Figure 5.61(a), this would be duct run ABCDEFG. In order to do this accurately, we would need a detailed duct diagram showing all fittings. We would also need to calculate the net external pressure available from the furnace data such as is given in Figure 5.6(b). (The furnace data given in Figure 5.12(b) is not sufficient. The technologist would have to contact the manufacturer for static pressure data corresponding to the cfm figures tabulated.) The net supply duct static pressure available is only a portion of the total external static pressure. That calculation will be explained in the following duct system design procedure.

The TEL is calculated by adding all the straight duct lengths to the sum of the TEL

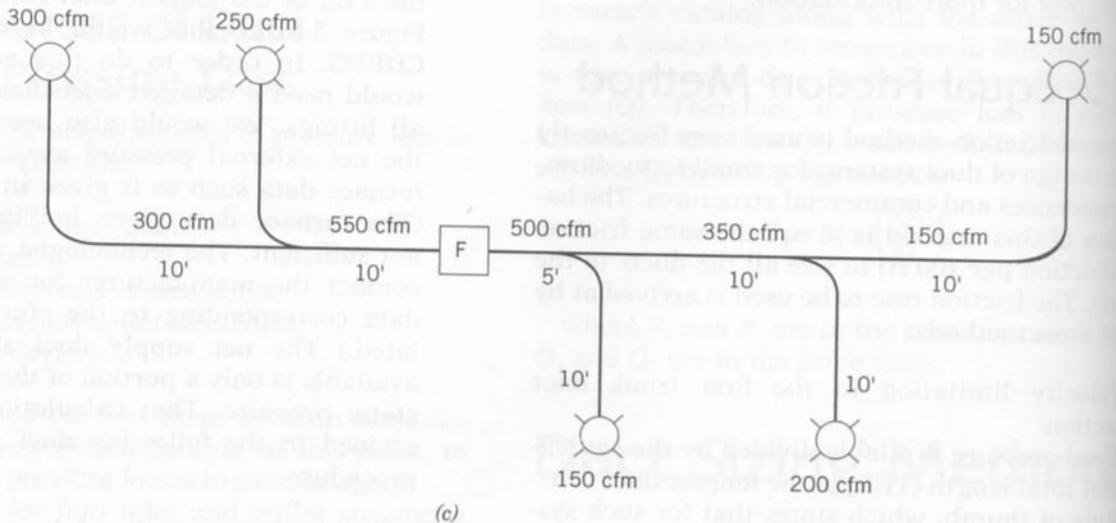
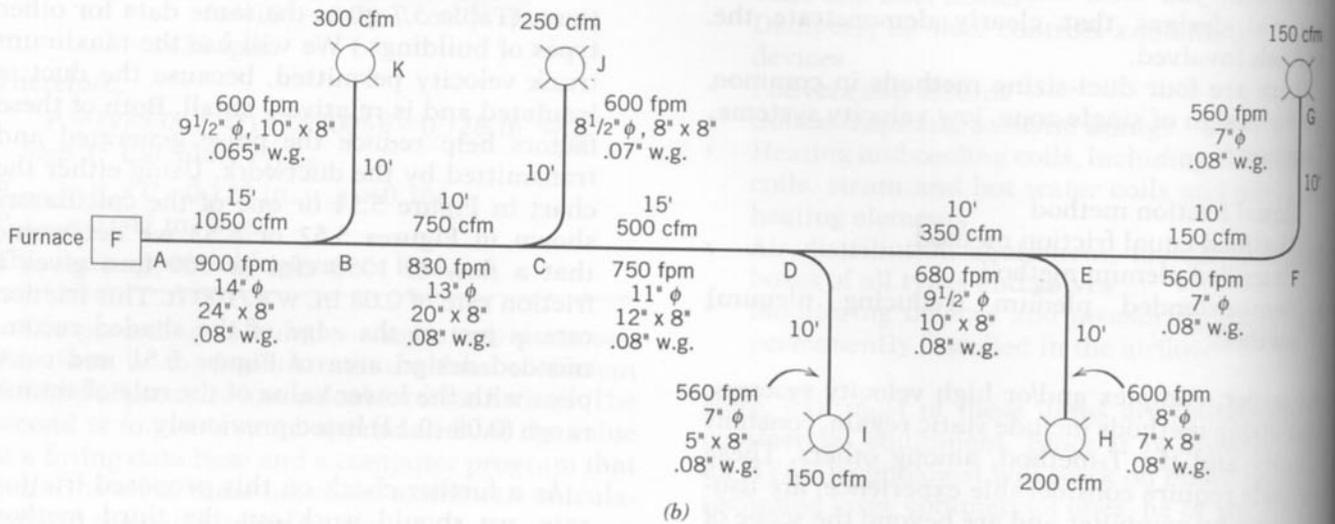
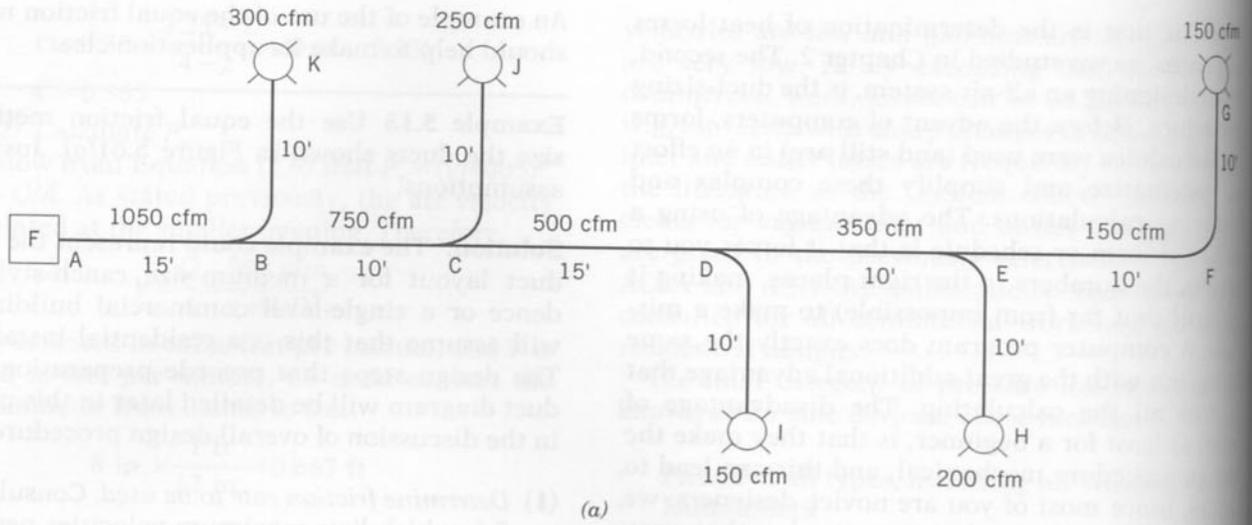


Table 5.6 Recommended Air Velocities (fpm) for Noise Limitation in Residences

Location	Supply Side				Return Side			
	Recommended		Maximum		Recommended		Maximum	
	Rigid	Flex	Rigid	Flex	Rigid	Flex	Rigid	Flex
Trunk ducts	700	600	900	600	600	600	700	600
Branch ducts	600	600	900	600	400	600	600	600
Supply outlet face velocity	Size for Throw		700		—		—	
Return grille face velocity	—		—		—		500	
Filter grille face velocity	—		—		—		300	

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Table 5.7 Maximum Velocities for Low Velocity Systems (fpm)

Application	Controlling Factor— Noise Generation, Main Ducts	Controlling Factor—Duct Friction			
		Main Ducts		Branch Ducts	
		Supply	Return	Supply	Return
Residences	See Table 5.6	1000	800	600	600
Apartments, hotel bedrooms, hospital bedrooms	1000	1500	1300	1200	1000
Private offices, director's rooms, libraries	1200	2000	1500	1600	1200
Theaters, auditoriums	800	1300	1100	1000	800
General offices, high-class restaurants, high-class stores, banks	1500	2000	1500	1600	1200
Average stores, cafeterias	1800	2000	1500	1600	1200
Industrial	2500	3000	1800	2200	1500

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Figure 5.61 (a) Single-line diagram of a duct arrangement, showing duct lengths and required air quantities. (b) Duct sizes, air velocities and friction rate for all sections of the system, as calculated by the equal friction rate method. (c) Placement of the air handler (furnace) in the center of the duct system makes balancing the system much simpler.

values of all the fittings as obtained from Appendix B. Since, for our example, we have neither the fan pressure nor the system TEL, we will assume that this calculation gives a friction rate close to the 0.08 in. w.g. we have already obtained. If an actual calculation were to show a value much higher, a lower motor speed would be selected to drop the pressure. If the friction rate is too low, a higher motor speed is needed. Too high a friction rate means excessive velocity and noise; too low a friction rate means excessively large ducts.

(2) *Sizing all sections of trunk duct.* Using the friction rate of 0.08 in. w.g./100 ft, we now proceed to size all sections of trunk duct, again using either Figure 5.51 or a duct calculator. The round duct sizes obtained for duct sections AB, BC, CD, DE, and EF are shown on Figure 5.61*b*. Air velocity and the friction rate are also indicated for each trunk section. Depending on the architectural layout, the technologist might choose to use either round ducts or rectangular ducts. The equivalent rectangular sizes are also shown on Figure 5.61*b*. Note that all the ducts are 8 in. deep. This is the depth commonly used in residential work, because this size duct will fit in the space between floor joists. If the duct is run across the joists, any reasonable depth is usable. The entire main duct assembly is constructed as a reducing trunk duct. See Figure 5.18(*d*) and Appendix B, Figure B.2.

(3) *Branch ducts.* Again, using the friction chart or a duct friction calculation, duct sizes for branches can be determined using the same friction rate. Maximum velocity should not exceed 600 fpm to limit noise levels and friction rate adjusted accordingly. Round branch ducts are commonly used in residences, although rectangular ducts may be preferable to fit the architecture. Note that section FG is simply an extension of trunk section EF. For this reason, the velocity in EF was also held below 600 fpm. This essentially completes the design of the duct system.

The equal friction method gives best results when the TELs of all runs are approximately equal. This would be the case, for instance, if the furnace were in the center of the duct run as in Figure 5.61(*c*) and not at the end, as is Figure 5.61(*a*). A glance at Figure 5.61(*a*) shows why. The total pressure drop between the furnace and outlet G (duct run ABCDEFG) is much higher than the pres-

sure drop to outlet K (duct run ABK). That means that the pressure at outlet K is much higher than that at outlet G. The result will be too much air at K and too little at G, in other words, an unbalanced system. (A good rule of thumb to follow is that the pressures at all outlets in a system should not vary one from another, by more than 0.05 in. w.g.)

To compensate for this unbalance, it is standard practice to install volume dampers in all branches and runouts. Field adjustment of these dampers adds friction to short runs, permitting the system to be balanced. The problem with this "fix" is that dampers cause noise, which is exactly what we want to avoid. A much more satisfactory solution, from an engineering point of view, is to design the required additional friction into the system. This, indeed, is exactly what is done in the modified equal friction method.

5.26 Modified Equal Friction Method

The modified equal friction method is more accurately called the equal pressure loss method because it attempts to give all runs approximately the same TEL. This makes the system (almost) self-balancing.

The procedure for this method is:

- (1) Prepare a detailed duct layout showing all fittings.
- (2) Using the required air quantities in each section, calculate the pressure drop in all straight duct sections. Use the velocity limitations imposed by noise criteria.
- (3) Find the TEL of all fittings using Appendix B or calculate their pressure drops using Appendix C.
- (4) Find the pressure drop of the longest run.
- (5) Redesign the friction in other branches and fittings so that the total pressure drop from the furnace to each supply outlet is approximately equal.

This last step is the most difficult. Refer to Figure 5.61*a*. In order to make the pressure drop to outlet K the same as that to outlet G, a large pressure drop must be introduced into runout BK. One way to do this is to reduce the size of the duct from 9½ in. to 4–5 in. This, however, will increase the air velocity enormously, causing noise, vibration and severe drafts in the room being served. A much

better technique is to use a high resistance takeoff such as types P or Q in Figure B.2 or types A or F in Figure B.3. Even these, however, are not sufficient in short runs, and balancing dampers will still be required in all branches and runouts.

In more complex systems, with several main branches, the calculation becomes one of trial and error. We begin with an educated guess at friction rates for the various branches. Then TELs are found, and pressure drops are calculated and compared. At that point, friction rates, fitting types, air velocities and duct sizes are all juggled in an attempt to balance the system. True balance is almost always impossible, which is one reason that volume dampers are almost always used. Another important reason is that seasonal changeover between heating and cooling modes always requires changes in air flow. These changes are accomplished, in part, with dampers.

The advantage of the modified equal friction method is that it will give a nearly balanced system that will require only slight field adjustment. The disadvantage of the method is that to design it correctly, for anything but a small system, is tedious and time-consuming. Actually, to perform the calculations accurately, loss coefficients should be used for fittings rather than TEL. This is because almost all the fittings involve velocity changes between inlet and outlet. As explained in Section 5.23, the loss in such fittings cannot be calculated accurately using TEL. And, as we saw in that section, manual calculation of fitting pressure loss, using its loss coefficient, is an involved and time-consuming operation. Fortunately, in modern engineering offices, computers have relieved designers of these burdensome calculations.

5.27 Extended Plenum Method

See Figures 5.50 and B.3. An extended plenum, as explained in Section 5.20d, is simply a relatively short straight section of trunk duct that feeds a number of branch outlets, usually not exceeding six. The trunk cross section does not change throughout its length. It is called an *extended plenum* because, like a plenum, its size is constant, and it extends over a length of 25–30 ft. In small systems, it may represent the entire duct system. In large systems, such trunk ducts are found at the discharge of a fan, VAV box, mixing box and the like. They too are known as extended plenums.

The extended plenum method is not so much a specific calculation method, as it is the characteristics of air flow in this type of trunk duct. The air flow in an extended plenum is governed by principles that we have already learned. This air flow can be summarized as follows.

- (a) As we proceed along the trunk duct, air velocity and friction rate will decrease after each takeoff. This is necessarily so, since $Q = AV$, that is, air volume is the product of duct cross-sectional area and air velocity. (See Section 5.23, Equation 5.7, page 288.) Since air volume Q decreases after each takeoff and area A remains constant, velocity V must also decrease proportionately.
- (b) Since trunk velocity decreases after each takeoff, it will frequently occur that runout (branch) velocity is higher than trunk velocity. This will cause a small conversion pressure loss. This loss is not appreciable at trunk velocities below 600 fpm.
- (c) At takeoffs where trunk velocity is much higher than branch velocity, the pressure loss at the takeoff will be high. This can result in a "starved" takeoff, if the takeoff is not properly designed. This will become clear in the following calculations.

The design criteria usually applied to an extended plenum are:

- The trunk duct is sized for volume and design velocity. This velocity is usually friction-limited rather than noise-limited.
- Branch takeoffs are velocity-limited by noise criteria.
- Maximum trunk length should not exceed 30 ft.
- Takeoffs are usually round duct, but rectangular is also acceptable.
- Takeoff connections for round or rectangular duct should be made with a 45° angle connection. (A 90° connection can have very high pressure drop if trunk velocity is higher than branch velocity.)
- Extractors or scoops at takeoff points should be avoided. They cause turbulence and high pressure losses.
- Since the system is not self-balancing, balancing dampers should be installed in each branch.
- Since the friction rate changes after each takeoff, the friction loss in each section of trunk duct must be calculated individually.

A numerical example should clarify the preceding system characteristics and design principles.

Example 5.14 Refer to the extended plenum shown in Figure 5.62. An extended trunk of constant size feeds five branches with outlets at their ends. (For the sake of simplicity, we have made all the branches identical.) Each branch feeds an outlet with a 400-cfm design air quantity. Design this extended plenum duct system. Do not exceed an air velocity of 1000 fpm in any section. Calculate friction rates and pressure drops throughout the system.

Is the system balanced or nearly so?
What conclusions can be drawn about the balance of an extended plenum?

Solution:

- (1) The first step is to size the main trunk. Totaling all the air volumes required, we note them on the drawing:

Each branch	400 cfm
Duct section AB	2000 cfm
Duct section BC	1600 cfm
Duct section CD	1200 cfm
Duct section DE	800 cfm
Duct section EF	400 cfm

Using the specified maximum velocity of 1000 fpm, we find that duct section AB must be 19 in. in diameter. The equivalent rectangular section is 20 × 16 in., giving a friction rate of 0.075 in. w.g./100 ft for this section. These data are marked on the drawing.

- (2) Since we know the air quantity in each section of the plenum, we can calculate the air velocity in each section very simply as follows:

$$Q_{AB} = A \times V_{AB}$$

and

$$Q_{BC} = A \times V_{BC}$$

Therefore,

$$\frac{Q_{AB}}{Q_{BC}} = \frac{V_{AB}}{V_{BC}}$$

since A remains constant.

or

$$V_{BC} = V_{AB} \times \frac{Q_{BC}}{Q_{AB}} = 1000 \text{ fpm} \times \frac{1600 \text{ cfm}}{2000 \text{ cfm}} = 800 \text{ fpm}$$

Similarly,

$$V_{CD} = V_{AB} \times \frac{Q_{CD}}{Q_{AB}} = 1000 \times \frac{1200 \text{ cfm}}{2000 \text{ cfm}} = 600 \text{ fpm}$$

and

$$V_{DE} = 400 \text{ fpm}$$

and

$$V_{EF} = 200 \text{ fpm}$$

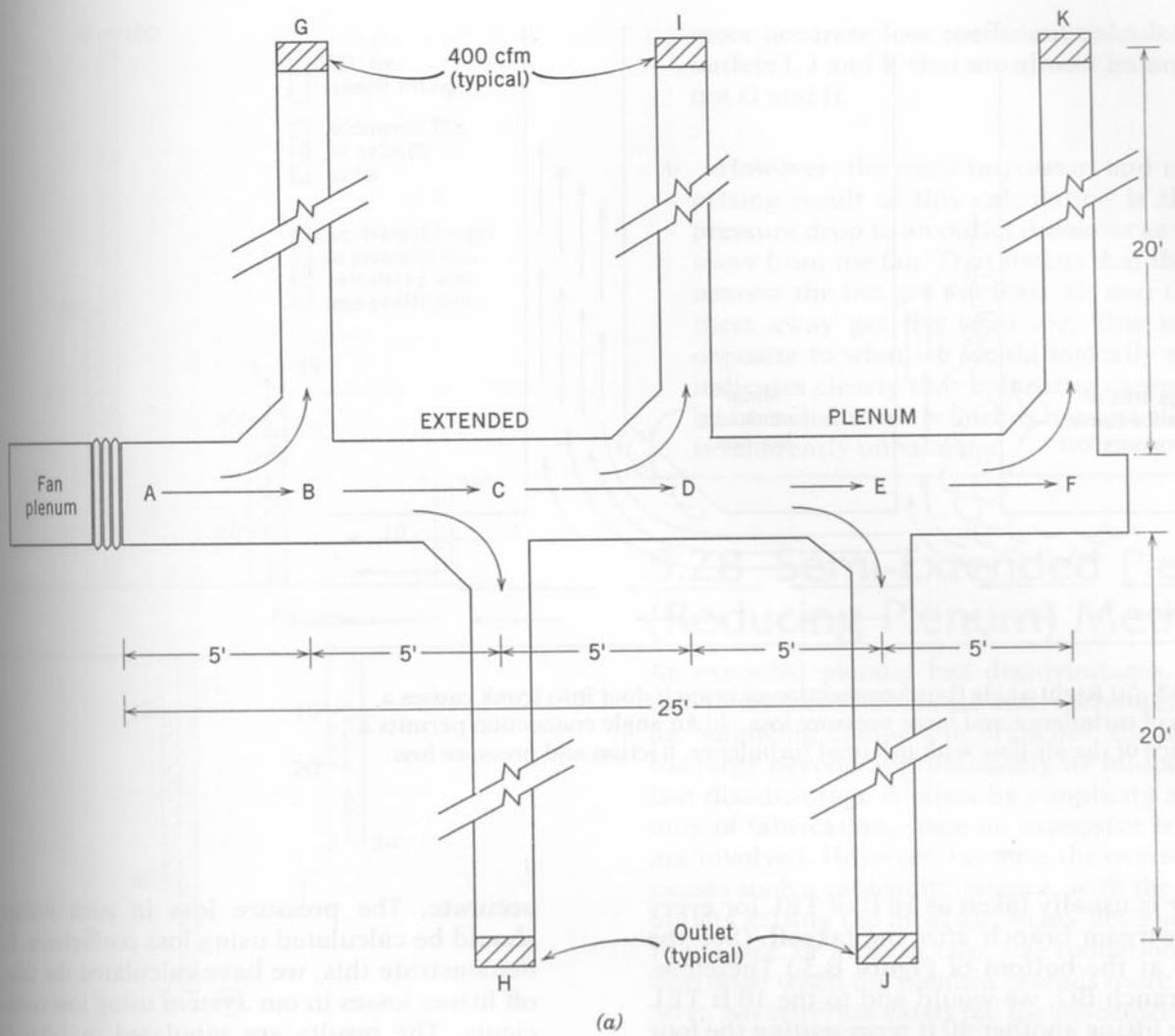
These values are marked on the duct diagram.

- (3) Since we know the Q , and V and size of each section, we can use the chart to find the friction rate. This too is marked on the diagram for each section. Note that the air velocity in the last section, EF, is so low (200 fpm) that the friction rate and, therefore, also the actual friction, are negligible. The friction rates for sections BC, CD and DE are 0.048, 0.03 and 0.013 in. w.g., respectively.
- (4) We select a velocity of 500 fpm for the branches to avoid noise problems. Knowing Q and V for the branches, we find that a 12 in. round or 10 × 12-in. rectangular duct is required. The friction rate is 0.035 in. w.g. These data apply to all branches, and this is marked on the duct drawing. At this point, we can calculate the pressure losses in all the straight duct sections of the system. These are summarized in Table A.

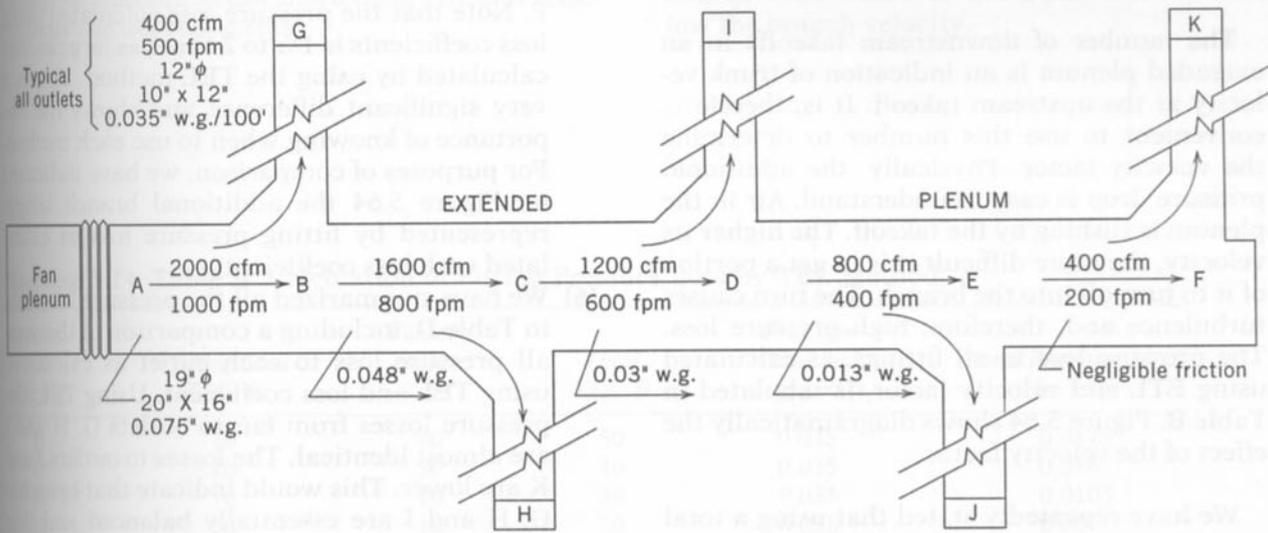
(Example 5.14) Table A Pressure Loss in Straight Duct Sections

Duct Section	Length, ft	Friction Rate, in. w.g./100 ft	Pressure Loss, in. w.g.
AB	5	0.075	0.00375
BC	5	0.048	0.0024
CD	5	0.03	0.0015
DE	5	0.013	0.00065
EF	5	0	0
BG, CH, DI, EJ, FK	20	0.035	0.007

- (5) Next we need to find the pressure drop of the takeoff fitting at each branch connection. Note that we used a 45° takeoff to reduce pressure loss. Figure 5.63 shows why the loss in such a takeoff is less than that in a right angle takeoff. Referring to Appendix B, Figure B.3, we find that a type D fitting has an TEL of 10 ft. To this TEL must be added an additional length to reflect the high takeoff fitting pressure loss due to the difference in air velocities between main and branch. The greater this difference is, the higher the pressure drop will be. This velocity



(a)



(b)

Figure 5.62 (a) Extended plenum layout for Example 5.14. (b) Extended plenum of Example 5.14 with all duct sizes, airflows and sectional friction losses indicated.

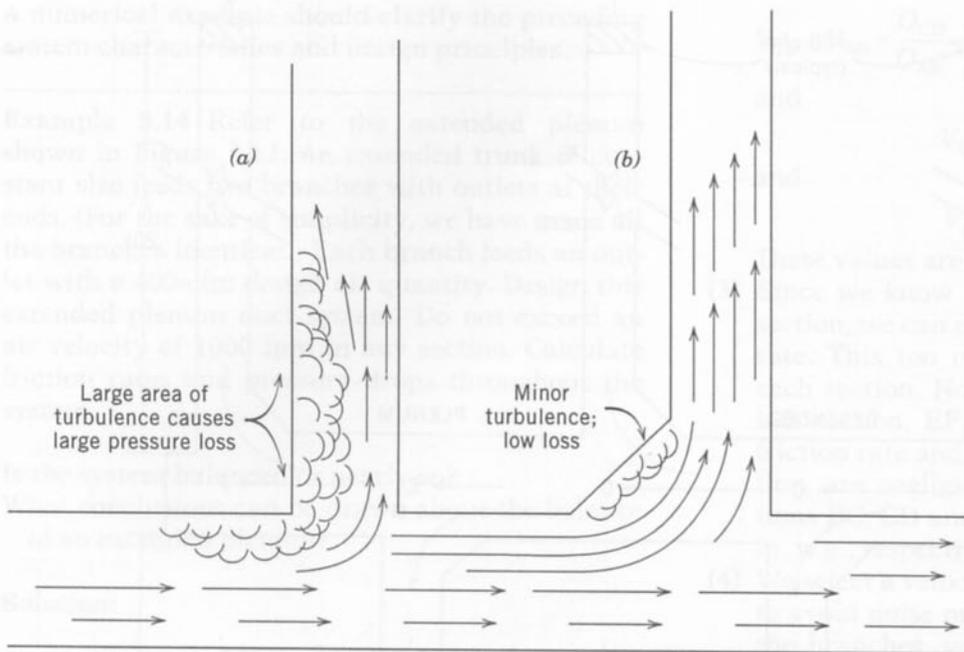


Figure 5.63 (a) Right angle (butt) connection of branch duct into trunk causes a large area of turbulence and large pressure loss. (b) An angle connection permits a smooth turn of the air flow with minimal turbulence, friction and pressure loss.

factor is usually taken as 10 ft of TEL for every downstream branch *after* the takeoff. (See the table at the bottom of Figure B.3.) Therefore, for branch BG, we would add to the 10 ft TEL of the fitting another 40 ft representing the four downstream takeoffs.

The number of downstream takeoffs in an extended plenum is an indication of trunk velocity at the upstream takeoff. It is, therefore, convenient to use this number to determine the velocity factor. Physically, the additional pressure drop is easy to understand. Air in the plenum is rushing by the takeoff. The higher its velocity, the more difficult it is to get a portion of it to turn off into the branch. The turn causes turbulence and, therefore, high pressure loss. The pressure loss in all fittings, as calculated using ETL and velocity factor, is tabulated in Table B. Figure 5.64 shows diagrammatically the effect of the velocity factor.

We have repeatedly stated that using a total equivalent length (TEL) for fittings through which a velocity change occurs, is not entirely

accurate. The pressure loss in such fittings should be calculated using loss coefficients. To demonstrate this, we have calculated the take-off fitting losses in our system using loss coefficients. The results are tabulated in Table C. The data were taken from Appendix C, fitting P. Note that the pressure loss calculated using loss coefficients is $1\frac{1}{2}$ to 2 times as large as that calculated by using the TEL method. This is a very significant difference and shows the importance of knowing when to use each method. For purposes of comparison, we have indicated on Figure 5.64 the additional branch length represented by fitting pressure loss as calculated with loss coefficients.

- (6) We have summarized all the pressure loss data in Table D, including a comparison of the overall pressure loss to each outlet as calculated using TEL and loss coefficient. Using TEL, the pressure losses from fan to outlets G, H and I are almost identical. The losses to outlets J and K are lower. This would indicate that branches G, H and I are essentially balanced and that J and K need to be throttled slightly, using dampers. Notice, however, that when using the

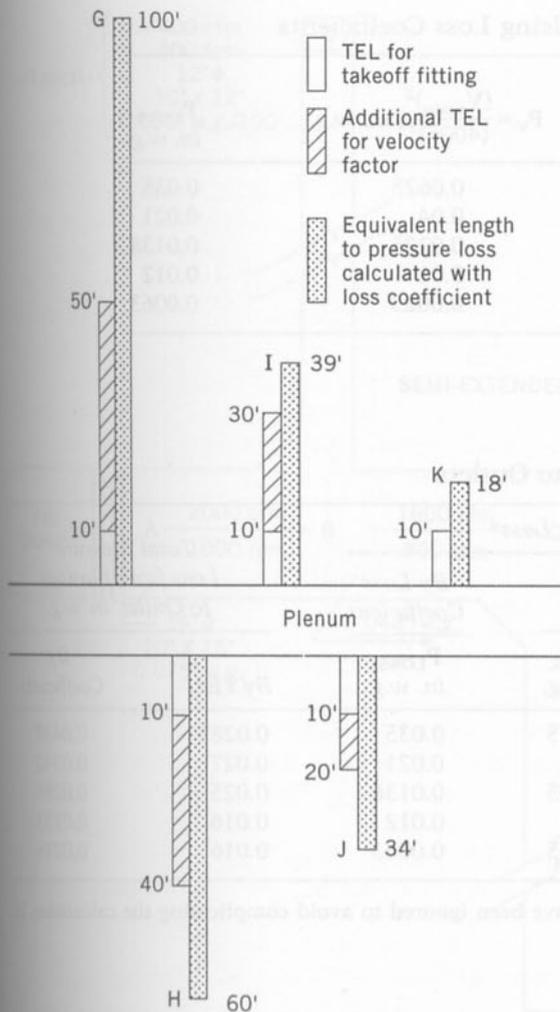


Figure 5.64 Example 5.14. Equivalent lengths of takeoff fittings for branches, as calculated using TEL plus velocity factor and loss coefficients.

more accurate loss coefficient calculation it is outlets I, J and K that are almost balanced, and not G and H.

However, the most important and most surprising result of this calculation is that total pressure drop to an outlet *decreases* as we move away from the fan. That means that the outlets nearest the fan get the least air and those farthest away get the most air. This is exactly opposite to what we would logically expect. It indicates clearly that balancing dampers must be installed in all branches because the system is inherently unbalanced.

5.28 Semi-Extended Plenum (Reducing Plenum) Method

An extended plenum has disadvantages. In addition to inherent imbalance and, therefore, the need for balancing dampers in branches, the duct is far too large beyond approximately its midpoint. This last disadvantage is offset by simplicity and economy of fabrication, since no expensive transitions are involved. However, because the oversized duct causes such a radical imbalance, with the remotest outlets getting the most air, many designers prefer to use one, or at most two, size reductions. This is also done when the plenum extends more than 25–30 ft. No criterion exists for the placement of these transitions. Some designers suggest that a transition be used wherever the trunk velocity falls below the branch velocity.

Example 5.14, Table B Calculation of Fitting Pressure Loss by TEL Method

Location of Fitting	Fitting Loss TEL, ft	Velocity Factor, ft	Total TEL, ft	Friction Rate, in. w.g./100 ft	Total Pressure Loss, in. w.g.
B	10	40	50	0.035	0.0175
C	10	30	40	0.035	0.014
D	10	20	30	0.035	0.0105
E	10	10	20	0.035	0.007
F	10	0	10	0.035	0.0035

Example 5.14, Table C Fitting Pressure Loss Calculation Using Loss Coefficients

Location of Fitting	$\frac{Q_{\text{branch}}}{Q_{\text{main}}}$	$\frac{V_{\text{branch}}}{V_{\text{main}}}$	Loss Coefficient	$P_v = \frac{(V_{\text{main}})^2}{(4005)^2}$, in. w.g.	P_{LOSS} , in. w.g.
B	0.2	0.5	~0.55	0.0625	0.035
C	0.25	0.625	0.514	0.04	0.021
D	0.33	0.83	0.615	0.0225	0.0138
E	0.5	1.25	1.19	0.01	0.012
F	1.0	~2.5	~2.5	0.0025	0.0063

Example 5.14, Table D Total Pressure Loss from Furnace to Outlets

Outlet Location	Pressure Loss in Straight Ducts ^a	Fitting Loss ^a				Total Pressure Loss from Furnace to Outlet, in. w.g.	
		Loss, in. w.g.	By TEL		By Loss Coefficient	By TEL	By Coefficient
			Path	TEL, ft	P_{LOSS} , in. w.g.		
G	ABG	0.01075	50	0.0175	0.035	0.0283	0.0458
H	ABCH	0.01315	40	0.014	0.021	0.0272	0.0342
I	ABCDI	0.01465	30	0.0105	0.0138	0.025	0.0285
J	ABCDEJ	0.0153	20	0.007	0.012	0.016	0.0273
K	ABCDEFK	0.0153	10	0.0035	0.0063	0.016	0.0216

^aThe pressure losses in the trunk duct caused by the takeoff fittings have been ignored to avoid complicating the calculation. In actual practice, they should be included.

To illustrate this semi-extended plenum method, we have placed a transition in the duct of Example 5.14 (see Figure 5.62) at point D, beyond the third takeoff. At this point velocity drops to 400 fpm in the original duct, as compared to a branch velocity of 500 fpm. We have resized the plenum for 1000 fpm and recalculated all the losses based on these revised data. See Figure 5.65. Table A1 shows the pressure losses in the straight sections of duct; Table B1 lists the fitting losses, including both takeoff fittings and the duct transition fitting. Note that the loss for takeoff fittings at B, C and D are calculated with loss coefficients. The loss coefficient table does not cover the flow and velocity conditions at points E and F. As a result, we were forced to use the TEL method for these fittings. This is indicated in Tables B1 and C1.

Table D1 summarizes the results. Notice that the total pressure losses to all outlets are much more uniform than for the full extended plenum. Balanc-

Table A1 Pressure Loss in Straight Duct Sections, Semi-extended Plenum

Duct Section	Length, ft	Friction Rate, in. w.g./100 ft	Pressure Loss, in. w.g.
AB	5	0.075	0.00375
BC	5	0.048	0.0025
CD	5	0.03	0.0015
DE	5	0.125	0.0063
EF	5	0.03	0.0015
All branches	20	0.035	0.007

ing dampers would still be needed in all runouts for minor field adjustment. The disadvantage of this method is the additional cost due to the addition of a duct transition(s). This cost is at least partially offset by the cheaper smaller duct section after each transition.

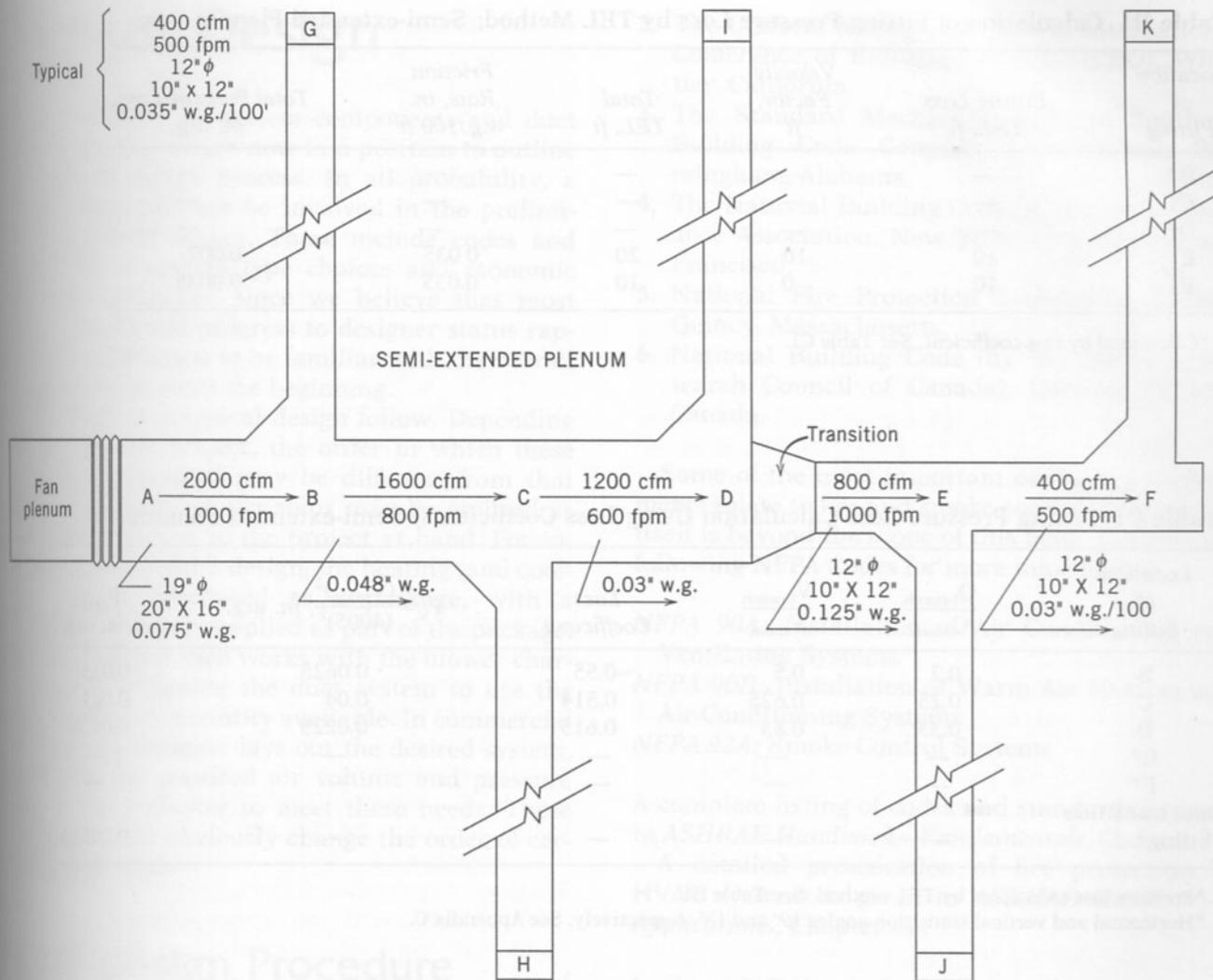


Figure 5.65 Semi-extended plenum layout. This trunk duct arrangement is also called a reducing plenum.

Table B1 Calculation of Fitting Pressure Loss by TEL Method; Semi-extended Plenum

Location of Fitting	Fitting Loss TEL, ft	Velocity Factor, ft	Total TEL, ft	Friction Rate, in. w.g./100 ft	Total Pressure Loss, in. w.g.
B ^a	—	—	—	—	—
C ^a	—	—	—	—	—
D ^a	—	—	—	—	—
E	10	10	20	0.035	0.007
F	10	0	10	0.035	0.0035

^aCalculated by loss coefficient. See Table C1.

Table C1 Fitting Pressure Loss Calculation Using Loss Coefficients; Semi-extended Plenum

Location of Fitting	$\frac{Q_{\text{branch}}}{Q_{\text{main}}}$	$\frac{V_{\text{branch}}}{V_{\text{main}}}$	Loss Coefficient	$P_V = \frac{(V_{\text{main}})^2}{(4005)^2}$, in. w.g.	P_{Loss} , in. w.g.
B	0.2	0.5	~0.55	0.0625	0.035
C	0.25	0.625	0.514	0.04	0.021
D	0.33	0.83	0.615	0.0225	0.0138
E ^a	—	—	—	—	—
F ^a	—	—	—	—	—
Duct transition fitting ^b	—	—	—	—	0.00313

^aPressure loss calculated by TEL method. See Table B1.

^bHorizontal and vertical transition angles 19° and 15°, respectively. See Appendix C.

Table D1 Total Pressure Loss from Furnace to Outlets; Semi-extended Plenum

Outlet Location	Path	Pressure Loss in Straight Ducts ^a Loss, in. w.g.	Fitting Loss ^a			Total Pressure Loss from Furnace to Outlet, in. w.g.
			By TEL		By Loss Coefficient	
			TEL, ft	P_{Loss} , in. w.g.	P_{Loss} , in. w.g.	
G	ABG	0.01075	—	—	0.035 ^c	0.0458
H	ABCH	0.01315	—	—	0.021 ^c	0.0342
I	ABCDI	0.01465	—	—	0.0138 ^c	0.0285
J	ABCDEJ	0.021	—	0.007 ^b	0.00313 ^d	0.0311
K	ABCDEFK	0.022	—	0.0035 ^b	0.00313 ^d	0.0286

^aThe pressure losses in the trunk duct caused by the takeoff fittings have been ignored to avoid complicating the calculation. In actual practice they should be included.

^bFrom Table B1.

^cFrom Table C1.

^dLoss in the duct transition.

System Design

Having studied air system components and duct system design, we are now in a position to outline the overall design process. In all probability, a technologist will not be involved in the preliminary stages of design. These include codes and jurisdictions, system type choices and economic decisions. However, since we believe that most technologists will progress to designer status rapidly, it is important to be familiar with the overall design process, from the beginning.

The steps in a typical design follow. Depending on the specific project, the order in which these steps are performed may be different from that listed. Too, some of the steps may be omitted as not being relevant to the project at hand. For instance, in residential design, the heating (and cooling) unit is purchased as a package, with a multispeed blower supplied as part of the package. The technologist then works with the blower characteristics, designing the duct system to use the pressure and air quantity available. In commercial projects, the designer lays out the desired system, calculates the required air volume and pressure and selects a blower to meet these needs. These differences will obviously change the order of certain design steps.

5.29 Design Procedure

Keeping in mind that each project has its own peculiarities that may require variations, the typical warm air system design procedure is described next.

a. Codes and Ordinances

Every construction project is subject to local construction ordinances and codes. Most of these refer to national codes. Some of the larger cities have their own codes in addition to requiring adherence to national codes. Determining which codes apply is the responsibility of the project engineer. Fulfilling code requirements is the designer's responsibility. One or more of the following codes will apply to any duct system:

1. The BOCA Basic Mechanical Code of Building Officials and Code Administrators International, Inc., Homewood, Illinois.

2. The Uniform Mechanical Code of International Conference of Building Officials (ICBO), Whittier, California.
3. The Standard Mechanical Code of Southern Building Code Congress International, Birmingham, Alabama.
4. The National Building Code of American Insurance Association, New York, Chicago and San Francisco.
5. National Fire Protection Association (NFPA), Quincy, Massachusetts.
6. National Building Code (by the National Research Council of Canada), Ottawa, Ontario, Canada.

Some of the most important ordinance requirements relate to fire and smoke control. The subject itself is beyond the scope of this book. Refer to the following NFPA Codes for more information:

NFPA 90A: Installation of Air Conditioning and Ventilating Systems

NFPA 90B: Installation of Warm Air Heating and Air Conditioning Systems

NFPA 92A: Smoke Control Systems

A complete listing of codes and standards appears in *ASHRAE Handbook—Fundamentals*, Chapter 38.

A detailed presentation of fire protection in HVAC systems is given in *ASHRAE Handbook—Applications*, Chapter 47.

b. Load Calculation

In this step, the heating (and cooling) loads for each space in the structure are calculated. They are then summarized to determine the total building load for heating, and for cooling if required. A stripped architectural drawing should be prepared at this stage. This is simply an architectural plan showing walls, doors and windows. Dimensional data (including ceiling heights) should be indicated. On this plan, the calculated loads for each space are indicated. The next stage of the load calculation procedure is to calculate air quantities required for each space, again for heating and cooling, using the relations developed in Section 5.3:

$$\text{cfm} = \frac{\text{Heating load in Btuh}}{1.08 \Delta T}$$

and

$$\text{cfm} = \frac{\text{Cooling load in Btuh}}{1.1 \Delta T}$$

The temperature difference ΔT is in the range of 45–75 F° for heating and 17–21 F° for cooling. The difference figures are based on a winter room return-heating air temperature of 70°F and heated air entering the room at 115–145°F. It is recommended that the entering air not exceed 135°F if a person can stand next to the supply register, since 145°F air is uncomfortably hot.

Similarly, the cool air temperature differential is based on a return-air temperature of 75°F and cool room air entering at 54–58°F. Here, also, if entering air can strike an occupant, 58°F is a better choice. Air at 54°F is uncomfortably cold, particularly when blown across the skin at velocities up to 300 fpm. The larger of the two air quantities (heating or cooling) is obviously the one that will determine the required duct sizes to all spaces and trunk duct sizes.

c. System Type

At this point, a study is made of the building operation in order to decide whether a multizone system is required. If zoning is required, the type of zoning arrangement is the next decision. See Section 5.19. Once the system is decided upon, a single-line duct diagram can be drawn on the working drawing(s) showing air quantities in all sections. At this stage, if the technologist is working with calculation forms or computer input forms, all the air quantity (and temperature) data can be entered. The dimensional data for all spaces was entered on the load forms at the load calculation stage.

d. Furnace Selection

On the basis of the calculated building load, a furnace of sufficient capacity (MBH) is selected. The furnace rating must include spare capacity as described in Section 5.8.b. Having determined the furnace MBH and the total cfm required for the building, the furnace temperature rise requirement should be calculated:

$$\text{Temperature rise} = \frac{\text{Furnace rating in MBH}}{1.08 (\text{Total cfm})}$$

With these three items of data—the furnace capacity, temperature rise and cfm required—a specific unit can be chosen from manufacturers' data tables, similar to those of Figure 5.6*b* and Figure 5.12.

From these same tables dimensional information can be taken. These data are now used to determine the space requirements for the furnace, its plenum,

duct connections and so on. Of course, the decision as to the type of furnace to be used, that is, upflow high-boy, upflow low-boy, downflow or horizontal, has already been made, based on the building architecture and the duct plan to be used.

If the furnace is a residential type, the external static friction available for different motor speeds and air flow quantities is also available from the manufacturer's data. If the unit is a commercial furnace, the blower will be selected at a later stage.

e. Supply and Return Outlets

On the basis of air quantities calculated in Section 5.29*b*, and considering the selection criteria detailed in Sections 5.15–5.18, supply outlets and return outlets can be selected, located and sized. Use manufacturers' data such as that shown in Figures 5.21, 5.22 and 5.23. Show locations and sizes on the working drawings. Record the static pressure drops at each outlet, for future use. Except in unusual installations, the total pressure drop in supply registers or diffusers should not exceed 0.07 in. w.g. and 0.04 in. w.g. in a return grille.

A convenient rule of thumb for determining the number of outlets required in a space is that one outlet is required for every 8000 Btuh of heating load and every 4000 Btuh of cooling load. It is also convenient to translate these figures into cfm. Using a common heating temperature rise of 55°F (125°F supply) and a cooling temperature drop of 19°F (56°F supply), we can calculate the cfm per outlet. For heating:

$$Q = \frac{8000}{1.08 (55^\circ\text{F})} = 135 \text{ cfm/outlet}$$

For cooling:

$$Q = \frac{4000}{1.1 (19^\circ\text{F})} = 190 \text{ cfm/outlet}$$

These figures can vary $\pm 15\%$ depending on air temperatures, outlet locations and outlet face velocity. However, as a guide and as a quick check on calculations, the figures given are reliable.

f. Duct Design

The duct design stage begins with establishing a target friction rate as described in Section 5.25. Assuming that we are dealing with a package unit, the external pressure available can be determined as described in Section 5.8.c. As a rule this pressure will be somewhere between 0.1 and 0.4 in. w.g. Pressures above 0.4 in. w.g. are necessary only for very long duct runs or runs with many fittings and

turns. Pressures below 0.1 are suitable only for straight trunk runs 25 ft long or less, with short branches. The duct design procedures described in Sections 5.21–5.28 can now be applied. A detailed duct plan is required showing all fittings, turns, junctions and the like to enable accurate calculation of pressure losses in the various system branches.

For low velocities, the use of TEL figures for fitting losses will not introduce large errors. In all cases, loss coefficients will give more accurate pressure loss data. If pressure loss calculations result in an overall duct system pressure loss below 0.1 in. w.g. or above 0.4 in. w.g., the friction rate should be altered (if possible), so that the pressure loss falls in this range. Pressures above 0.4 in. w.g. are available at high blower speeds but are preferably avoided because of high air velocity and the attendant noise problems.

Return-air ducts are included in the preceding calculation. Some designers prefer to divide the available pressure between the supply and return duct systems on an estimated basis. In our opinion, this is justified only if the return system is not ducted but consists of door undercuts, door and wall louvers and the like.

The particular duct design procedure selected depends on the duct layout. In most cases, the modified equal friction procedure as described in Section 5.26 will be adequate. Once the design is complete, the technologist can decide whether to use round ducts or convert to rectangular ducts. Of course, the decision regarding the duct material was made at the beginning of the duct design stage. Once all the ducts are sized, the entire system should be rechecked for velocity and noise problems. When the designer is satisfied that the system is workable, it can be drawn as a two line duct layout on the plans. At this stage, physical problems of installation and coordination with other trades will arise. Such problems will frequently lead to minor (and sometimes major) changes in the duct design.

g. Additional Design Items

As the technologist gains experience, he or she will almost intuitively know where to place fittings, turning vanes, balancing dampers and the like. A few useful rules in duct design are these:

- (1) Use the longest possible radii in turn fittings.
- (2) Where sharp turns are unavoidable, use elbows with turning vanes. A sharp turn is one where

the inside radius of the duct is less than one-third of the duct width.

- (3) When calculating pressure drop, use dampers in their open position as a fitting pressure loss.
- (4) Do not use register dampers for balancing. They are a source of unacceptable noise. Instead, place a balancing damper in the branch, as far upstream as possible.
- (5) In residential work, remember that the smallest stack readily available is $3\frac{1}{4} \times 10$ in. This is equivalent to a 6 in. round duct, which is often too large for the air requirement in a heating system. A damper is, therefore, always required. See Figure 5.1 for location of stacks.
- (6) Most residential air-conditioning contractors will use joist spaces for return air. Since this space is 16 in. wide and 8–12 in. deep, it is equivalent to a 11–15 in. duct, even with the additional roughness of the wood. As a result, additional pressure is available in the supply system.

5.30 Designer's Checklist

Most designers use a checklist to ensure that the design includes all the required items, properly applied. Such checklists are developed over years of design experience and vary from one designer to another. In many offices, such checklists are standardized for use by all designers. One such checklist is reprinted here with permission from *ACCA Manual Q—Commercial Low Pressure, Low Velocity Duct System Design*.

- Make sure that all duct velocities are in the correct range.
- Make sure that the velocity through each air-side component (such as filters, coils, louvers and dampers) is in the correct range.
- Branch takeoffs should not be close to the fan.
- A branch runout fitting should not be installed behind an elbow or upstream runout (leave six diameters).
- A branch runout fitting should not be installed behind an upstream extractor.
- The proportions or sizes of "split flow" fittings should be based on the cfm requirements of each resulting branch.
- Turning vanes should be installed with leading and trailing edges that are perpendicular to the air flow.
- Fan inlet and outlet fittings should minimize system effect losses.

- Fans should be selected to operate near the middle of the recommended operating range.
- Economizer cycle requires a return fan or an exhaust fan.
- A return fan is recommended if the pressure drop in the return duct system exceeds 0.10 in. w.g.
- Relief air dampers are required if the outdoor air cfm that is introduced into the space exceeds the cfm that is exhausted from the space and there is no return or exhaust fan (space pressurized).
- Corrections for surface roughness are required. Use the appropriate friction chart or use a sheet metal friction chart and apply the required correction factor.
- Corrections for elevation and temperature are required.
- Allowances (in the load calculations) should be made for duct losses and duct leakage.
- Fan motor should be selected for the largest power requirement that will be experienced during start-up or during normal operation.
- Fan curve and system curves should be checked to ensure that the fan operating point will remain within the recommended operating range during all possible operating conditions.
- Variable pitch (adjustable) pulleys should be specified to provide a way to adjust fan performances at the job site.
- Air distribution outlets and return inlets should be selected and sized according to the manufacturer's recommendations.
- Air outlets should have integral dampers (registers) that can be used for making minor adjustments.
- Splitter dampers should be used as diverters (only), and they should not be used to control air volume.
- A balancing damper should be installed at each branch duct takeoff from the main (supply or return) duct.
- A balancing damper should be installed in each runout or duct drop from a main duct or branch duct to a supply outlet or return inlet.
- Balancing dampers should be installed in each zone duct of a multizone duct system.
- Show all dampers, including fire dampers, in their proper locations on the plans.
- Provide open/closed-type dampers in outside and return air duct entrances.
- Show damper locations at accessible points and, whenever possible, at an acceptable distance from a duct transition or fitting.
- Avoid attaching diffusers, registers or grilles directly onto the bottom or sides of a duct.
- Provide (SMACNA-approved) boots, necks and extractors at all 90° branch duct connections to sidewall registers or ceiling diffusers.
- Short discharge ducts between mixing boxes and supply registers may cause excessive discharge velocities and air noise at face of register.
- Do not allow return air from one space or zone to pass through another space or zone to reach a return air register.
- Door louvers do not provide an acceptable return air path when the return air system operates at low pressure (e.g., ceiling return plenum).
- A perforated static pressure plate downstream from the fan discharge may be required if the fan discharges through a "blow through" coil.
- Screens are required on outdoor air intake and exhaust openings.
- Provide access doors of adequate size within working distance of all coils, volume dampers, fire dampers, pressure-reducing valves, reheat coils, mixing boxes, blenders, constant volume regulators and the like.
- Provide access for making pressure, temperature and tachometer readings at all critical points.
- All duct seams, duct connections, casing and plenum connections should be sealed to minimize leakage.
- Avoid "line of sight" installation of opposing supply outlets or return inlets (unless they serve the same room).
- Sound attenuation should be provided downstream from air-side devices that generate excessive noise.
- Merging air streams should be thoroughly mixed before they enter any type of air-side device or component.
- Provide a change of filters just prior to balancing.
- Make sure that there is no "short-circuiting" of discharge air from cooling towers, condensing units, relief exhausts, roof exhausters and the like to the inlet of any outside air intake.

Key Terms

Having completed the study of this chapter, you should be familiar with the following key terms. If any appear unfamiliar or not entirely clear, you should review the section in which these terms appear. All key terms are listed in the index to assist you in locating the relevant text.

ACCA

Air stream diffusion
 Air stream drop
 Air stream envelope
 Air stream spread
 Air stream terminal velocity
 Air stream throw
 All-air systems
 Annual fuel utilization efficiency (AFUE)
 Aspect ratio
 Back-pressure
 Balancing damper
 Comfort zone
 Condensing furnaces
 Counterflow furnace
 Diffuser
 Downflow furnace
 Draft gauge
 Dual-duct system
 Duct transition
 Dynamic pressure loss
 Electronic air cleaners
 Entrained air
 Entrainment
 Equal friction method
 Equal pressure loss method
 Equivalent length
 Evaporator coil
 Extended plenum
 Extended plenum method
 External static pressure
 Flexible ducts
 Grille
 Heat exchanger
 High-boy furnace
 High sidewall outlets
 Horizontal furnace
 Humidistat
 Inclined tube manometer
 Induction
 Isothermal jet
 Lateral furnace
 Loss coefficient (fitting)
 Low-boy furnace
 Low sidewall outlets

MBH

Make-up air
 Mixing boxes
 Modified equal friction method
 Multileaf damper
 Net static pressure
 Noise criterion (NC)
 Occupied zone
 Outlet face velocity
 Oval ducts
 Perimeter loop
 Pitot tube
 Primary air
 Primary air pattern
 Pulse combustion furnace
 Radius of diffusion
 Reducing plenum
 Register
 Register free area
 Register gross area
 Roughness Index
 Semi-extended plenum
 SMACNA
 Secondary air
 Single-leaf damper
 Single-zone system
 Spring pressure
 Stagnant air
 Stagnant zone
 Static
 Static head
 Static pressure
 Static regain
 Stratification
 Supply air
 Supply air rise
 Supply plenum
 Surface effect
 System effect
 Takeoff fittings
 Total air pattern
 Upflow furnace
 Variable air volume (VAV)
 Vane

Velocity factor
Velocity pressure
Vertical temperature gradient
Viscous impingement filter

Volume dampers
Water column
Water gauge

Supplementary Reading

American Society of Heating, Refrigeration and,
Air Conditioning Engineers (ASHRAE)
1791 Tullie Circle, N.E.
Atlanta, Ga. 30329 Tel.404-636-8400

Handbook—Fundamentals, 1993

Air Conditioning Contractors of America (ACCA)
1513 16th Street, N.W.
Washington, D.C. 20036

Manual 4—Perimeter Heating and Cooling, 1990

Manual B—Principles of Air Conditioning, 1970

Manual C—What Makes a Good A/C System

*Manual D—Duct Design for Residential Winter
and Summer Air Conditioning and Equipment
Selection*, 1984

Manual G—Selection of Distribution Systems

Manual J—Load Calculation, 1986

*Manual Q—Commercial Low Pressure Low Velocity
Duct System Design*, 1990

Manual S—Residential Equipment Selection

Manual T—Air Distribution Basics

Sheet Metal and Air Conditioning Contractors Na-
tional Association, Inc. (SMACNA)

8224 Old Courthouse Road

Tysons Corner, Vienna, Va. 22180

HVAC System Duct Design, 1990

HVAC Systems Applications, 1987

Air Movement and Control Association, Inc.
(AMCA)

30 West University Drive

Arlington Heights, Ill. 60004

Publication 200, Air Systems, 1987

6. Air Systems, Heating and Cooling, Part II

Refrigeration

In Chapter 5, we learned the design principles of all-air systems and concentrated on warm air heating. In this chapter, we will learn the principles of mechanical refrigeration as applied to comfort cooling systems. Particular emphasis will be placed on the operation of the heat pump, which has found extensive use in providing both cooling and heating in a wide variety of buildings. Finally, we will apply the knowledge gained in this chapter and in Chapter 5 in design exercises for specific buildings. Study of this chapter will enable you to:

1. Understand the operation of the vapor compression refrigeration cycle and all the components in such a system.
2. Understand the operating theory of a heat pump and distinguish between the various types of heat pumps.
3. Compare heat pump performance in the heating mode on the basis of coefficient of performance (COP) and heating season performance factor (HSPF) figures.
4. Evaluate air conditioner and heat pump performance on the basis of energy efficiency ratio (EER) and seasonal energy efficiency ratio (SEER) ratings.
5. Calculate the seasonal operating cost of a heat pump based on its SEER.
6. Find a system's heat pump balance point using a graphical construction method.
7. Determine the proper size DX coil to use based on sensible and latent cooling loads.
8. Understand the application of centralized systems with terminal units and decentralized systems with package, incremental units.
9. Understand the use of packaged terminal air conditioner (PTAC) and packaged terminal

heat pump (PTHP) equipment and the application of split air conditioners and heat pumps.

10. Draw and assist in the design of complete heating/cooling residential and small commercial buildings. This includes selection and location of equipment and ductwork design.

6.1 Unit of Mechanical Refrigeration

A practical mechanical refrigeration cycle was developed in 1902 by Dr. Willis Carrier, although several decades would pass before it began to be practically applied for comfort cooling. Most modern air conditioning units use this same compressive cycle, although in a much improved form. Prior to World War II, comfort cooling was primarily used in theaters and was based not on mechanical refrigeration but rather on ice. Large fans would blow air across blocks of ice, picking up cool damp air. This air, when blown into the theater, cooled the space, although the increased humidity often led to an uncomfortable feeling of clamminess in humid climates. To this day, the unit of refrigeration in common use is based on this original application of ice for cooling.

As we learned in Chapter 1, the latent heat of fusion of water is 144 Btu/lb. That is the amount of heat that must be extracted from a pound of water to turn it into ice. Conversely, when ice melts, each pound of ice absorbs from its surroundings 144 Btu (at 32°F). Therefore, when 1 ton of ice melts, it absorbs

$$2000 \text{ lb} \times 144 \text{ Btu/lb} = 288,000 \text{ Btu}$$

When a ton of ice melts over a 24-hr period, the rate of heat absorption from the surroundings (refrigeration) is

$$\frac{288,000 \text{ Btu}}{24 \text{ hr}} = 12,000 \text{ Btu/h or } 12,000 \text{ Btuh}$$

Therefore, cooling at a rate of 12,000 Btuh is called one ton of cooling.

Example 6.1 What is the cooling capacity in Btuh of the following air conditioning units:

- (a) $\frac{3}{4}$ ton
- (b) 2 tons
- (c) 5 horsepower

Solution:

- (a) $\frac{3}{4} \text{ ton} \times 12,000 \text{ Btuh/ton} = \frac{3}{4} (12,000) = 9000 \text{ Btuh}$
- (b) $2 \text{ tons} (12,000 \text{ Btuh/ton}) = 24,000 \text{ Btuh}$
- (c) 5 horsepower: Insufficient data. Many nontechnically trained people equate compressor horsepower to tons, simply because in small air conditioning units there is a rough correspondence between horsepower and tonnage. This is, however, a mere coincidence, and using it as a rule is incorrect. In large units, more than 1 ton of refrigeration is produced for each horsepower of compressor motor size. In all air conditioners, the motor size depends on the efficiency of the entire assembly, and this varies with each unit. The only accurate way of determining the refrigeration capacity and the electrical requirements of an air conditioning unit is to consult the manufacturer's published data.

6.2 Cooling by Evaporation

The compressive refrigeration cycle is based on the cooling effect of evaporation. In Chapter 1, we learned that when a liquid vaporizes it absorbs heat. The quantity of heat absorbed per unit weight is called the latent heat of vaporization. For water this quantity is 970 Btu/lb. Note that this is almost seven times as large as the latent heat of fusion of water (144 Btu/lb). This heat-absorbing characteristic is used very effectively by the body for cooling by perspiration evaporation, as was pointed out in Chapter 2.

Vaporization of a liquid will occur in two ways: slowly by evaporation or rapidly by boiling, which is essentially forced evaporation. Since water is the liquid that is most common in our environment, we will use it in our explanation, although the same principles apply to any liquid. Refer to Figure 6.1, which represents a bowl of water exposed to the atmosphere. Some of the water molecules at the surface will escape into the air and become water vapor. This process is called *evaporation*. It takes place constantly at every exposed body of water on earth, including all ponds, rivers, lakes, and oceans. It is precisely this evaporation action, caused by the heat of the sun, that drives the earth's weather systems.

The rate at which evaporation occurs depends on

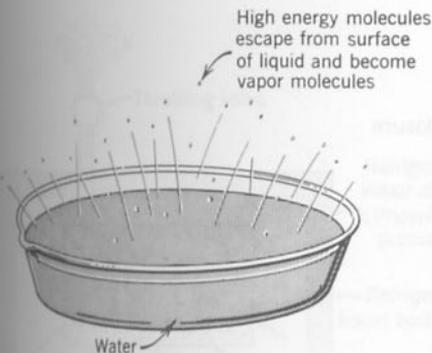


Figure 6.1 A liquid at any temperature will evaporate spontaneously into the atmosphere. The rate of evaporation increases with water temperature and surrounding air temperature. It also increases as the surrounding ambient pressure drops. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

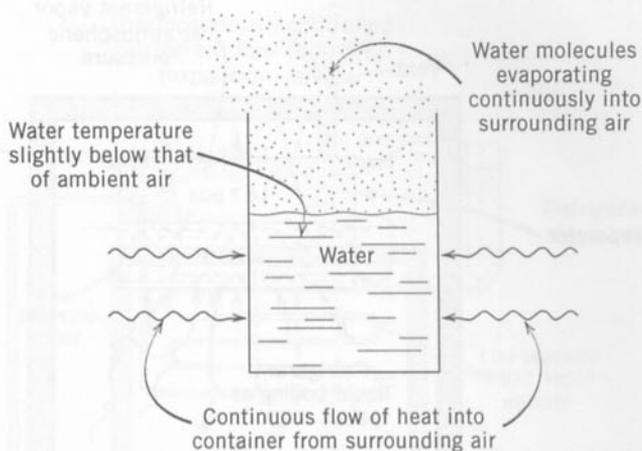


Figure 6.2 As water evaporates, it takes heat from the water in the container, thus lowering its temperature. This heat is replaced by heat flow from the surrounding space.

the water temperature and air temperature and pressure. The evaporation rate increases with increasing water and air temperature and decreasing air pressure. As the water temperature increases, it will eventually boil. The temperature at which this occurs is called the liquid's boiling point, or more accurately its *saturation temperature*. At this temperature, all the water will change to gas (water vapor) and in so doing will absorb 970 Btu/lb from the heat source.

The cooling effect of evaporation is caused by this heat absorption characteristic. Refer to Figure 6.2. Water evaporating from an open container absorbs heat from the water in the container, thus lowering its temperature slightly. This heat is replaced, through the container walls, from the heat in the surrounding air. This process will continue until all the water in the container evaporates. In the case of the evaporation of perspiration, the body supplies the continuous heat being absorbed, thus cooling itself very effectively. Obviously, an open system such as that seen in Figures 6.1 and 6.2 is not commercially practical since the refrigerant (in this case water) is constantly being used up. What is required for a practical mechanical refrigeration system is a closed arrangement that will reuse the refrigerant continuously. That is the essential idea behind the compressive refrigeration cycle.

6.3 Refrigeration Using a Closed Vapor Compression System

Water is not a practical refrigerant because of its high saturation temperature (boiling point). What is required in a closed system is a liquid that will reach its saturation temperature (boil) at a low temperature. Various fluids have been used as refrigerants, including ammonia and a group of fluorinated hydrocarbons generally known by the commercial name Freon. Unfortunately, these materials have been found to be environmentally unsuitable and are now replaced with other, environmentally neutral fluids. The exact chemical composition of modern refrigerants is not important to us. What is important is to understand how refrigerants like Freon, with a boiling point of about -20°F , are used in the refrigeration cycle. A thorough understanding of this compressive cycle is very important to the technologist. For this reason, it is developed in detail here.

a. Refrigerant Vaporization

Refer to Figure 6.3. An insulated space such as a refrigerator box can be rapidly cooled by simply

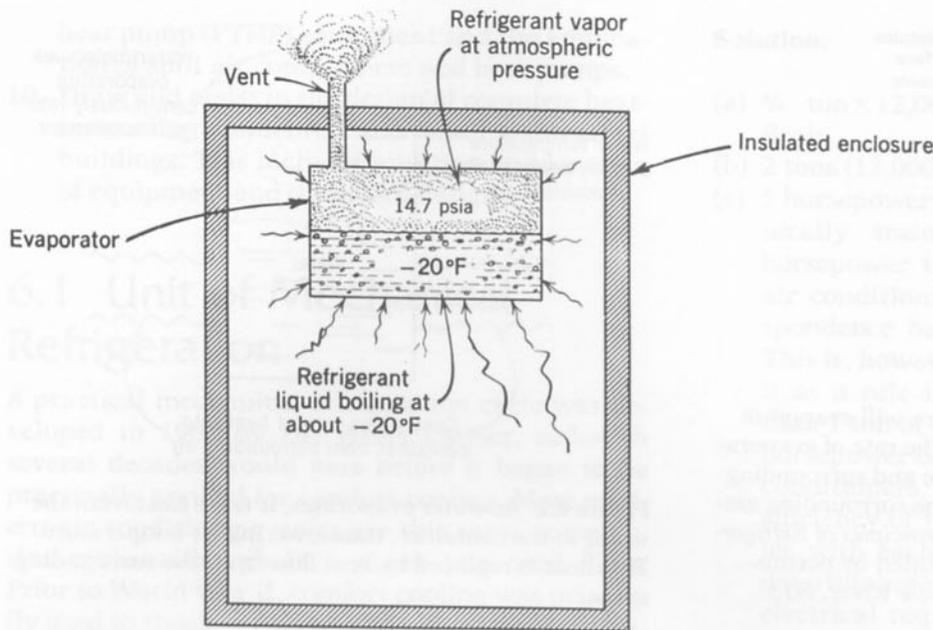


Figure 6.3 The refrigerant in the evaporator boils at about -20°F , at atmospheric pressure. It will, therefore, rapidly cool the inside of the container. Cooling will continue until all the refrigerant is exhausted into the surrounding air. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

allowing a low boiling point refrigerant liquid to flash (boil) into vapor at atmospheric pressure. In so doing, the evaporating vapor absorbs large amounts of heat from the surroundings, thus cooling the interior of the refrigerator box and its contents. Of course, we would quickly lose the refrigerant, and the cooling would be rapid, severe and of short duration. By placing a throttling device in the vent, as in Figure 6.4, we can control the flow of vapor. This, in turn, controls its pressure and therefore also its boiling point. The valve is, therefore, effectively a cooling temperature control. The device containing the boiling refrigerant is called the system *evaporator* because it is there that the refrigerant evaporates.

b. Refrigerant Flow Control

Refer to Figure 6.5. A valve is required at the input of the evaporator. It must control the flow of refrigerant into the evaporator from a storage container, to correspond exactly to the flow out of the evaporator. The specific design of this device is not of concern to us; only its operation is important. It is called the *refrigerant flow control de-*

vice, and in most modern systems it is a thermostatically controlled expansion valve.

c. Recycling the Refrigerant

Obviously, the refrigerant should not be exhausted to free air, as is schematically shown in Figures 6.1 and 6.2. It must somehow be recycled. Since it evaporated by absorbing heat, all that is required to return it to its liquid state is to remove the same amount of heat. This will condense the vapor back into a liquid. The device used to perform this function is, therefore, called a *condenser*. See Figure 6.6. The problem that immediately arises is how the condenser will draw off this heat. The condenser is simply a coil through which the saturated vapor passes. It is cooled by blowing ambient air over it or by running it through a heat exchanger that uses cooling water. Since the temperature of the vapor at the evaporator output is somewhere between 30 and 50°F , ambient air at 80 – 110°F or cooling water at 60 – 90°F are useless as cooling mediums. (Cooling water comes from municipal lines, cooling ponds or cooling towers.) Heat will not flow "uphill," that is, from the cool refrigerant

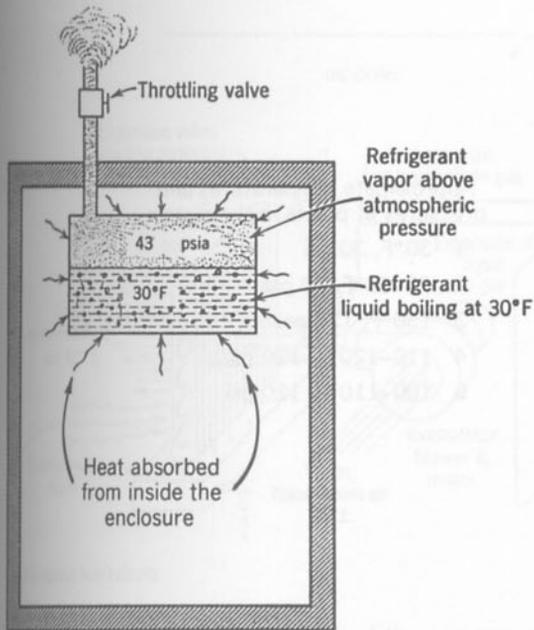


Figure 6.4 A throttling valve in the evaporator outlet line can be used to vary the evaporator pressure and, therefore, its boiling point. Here the valve is almost completely closed. This raises the boiling point from -20°F at atmospheric pressure to 30°F at about 3 atmospheres. Shutting the valve would raise the pressure until the vapor boiling point equalled the container temperature. At that point, all heat transfer stops. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

vapor to the warmer condenser cooling medium (ambient air or cooling water). To overcome this problem, the vapor coming out of the evaporator is compressed, raising its temperature to about 130°F and its pressure to about 8 atmospheres (120 psi). This hot compressed gas then enters the condenser coils where it is cooled by air or water to about 100°F and partially condensed into liquid. Pressure remains at 7–8 atmospheres. This warm, high pressure liquid and gas refrigerant mixture then travels past the refrigerant tank and on to the thermostatically controlled valve at the input of the evaporator, to begin the cycle all over again.

d. A Closed Recycling System

See Figure 6.6, which is a flow diagram of a simple vapor compression closed-cycle system. The principal parts of the system, shown schematically, are:

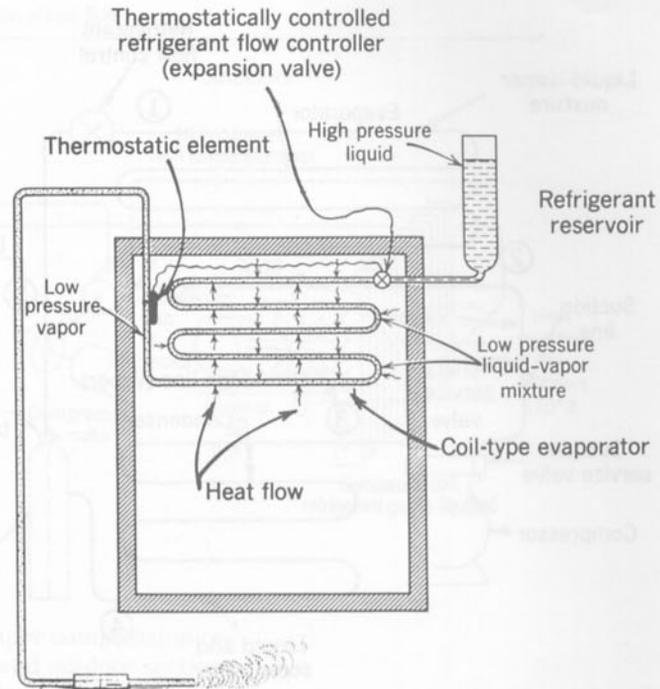


Figure 6.5 The refrigerant flow control valve supplies refrigerant to the evaporator at exactly the rate at which it is evaporated. The thermostat bulb senses the outlet (boiling) temperature. In passing through the expansion valve, the liquid refrigerant expands, thereby reducing pressure and creating a liquid-vapor mixture at relatively low pressure. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

- (1) **Evaporator.** The function of an evaporator is to provide a heat-absorbing surface. It is usually a coil of pipe, inside which the refrigerant is vaporizing and absorbing heat. Air blown over the surface of this pipe is cooled. This cool air is the end product of the refrigeration process. In a common household refrigerator, the cool air is confined to a closed box containing perishables. In a comfort air conditioning system, recirculated room air is blown over the evaporator coil. Since the room air is warmer than the evaporator coil, it will cause a 10–20 °F temperature rise in the vapor temperature. This is shown on Figures 6.7 and 6.8.
- (2) **Suction line.** The suction line is the line through which the slightly warmed refrigerant vapor passes on its way to the compressor.
- (3) **Compressor.** The function of the compressor is to change the low temperature, low pressure

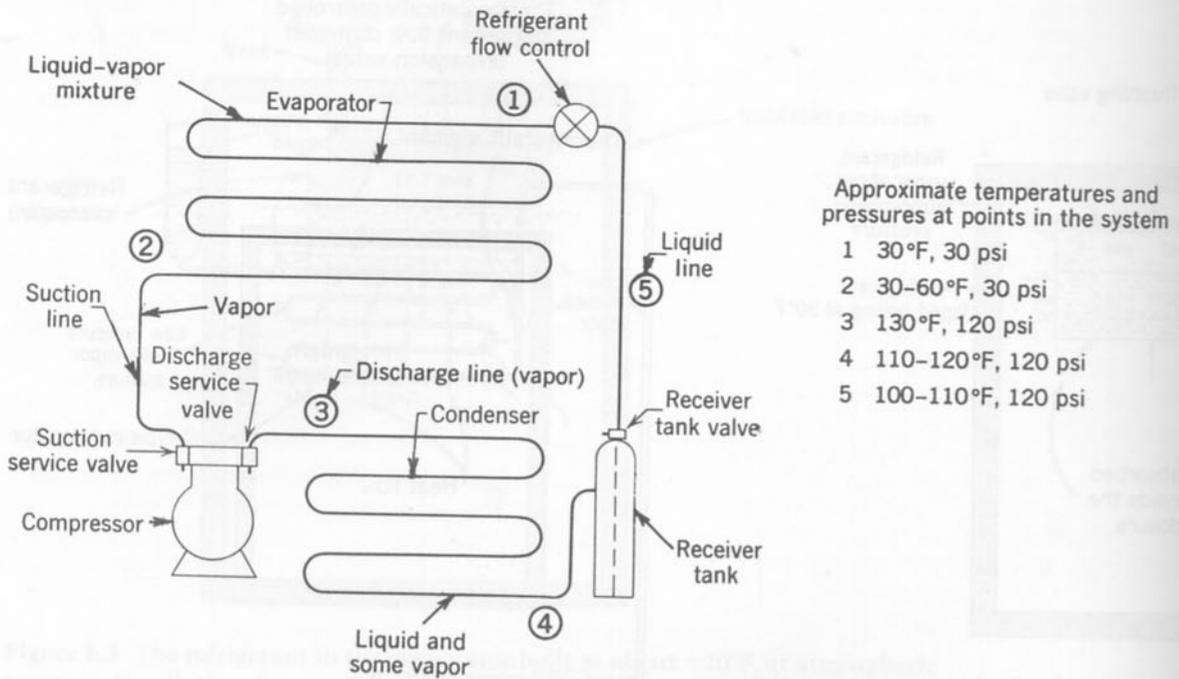


Figure 6.6 Schematic diagram showing the major components of a closed-cycle vapor compression refrigeration system. Typical temperatures and pressures at various points in the system are also shown. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

vapor coming out of the evaporator to a high temperature, high pressure gas.

- (4) **Condenser.** The condenser receives hot vapor from the compressor and condenses it to a gas-liquid mixture by cooling. It is usually constructed as a long folded pipe over which cooling air passes when air is used as the cooling medium. Alternatively, cool water and a heat exchanger can be used to perform the required cooling and vapor condensation.
- (5) **Receiver tank.** The receiver tank is used to store a quantity of liquid refrigerant.
- (6) **Flow control device.** Located at the input of the evaporator, the flow control device reduces the temperature and pressure of the high temperature liquid refrigerant and supplies a low temperature, low pressure liquid-gas mixture to the evaporator.

Figure 6.7 shows the basic closed-cycle vapor compression refrigeration system as applied to comfort cooling. Two blowers (fans) are added to

the basic equipment described previously. The indoor blower recirculates room air. The exterior fan cools the condenser with outside air. Remember that the terms *evaporator* and *condenser* refer to actions performed on the refrigerant. Some confusion arises because warm humid air condenses on the evaporator (see Figure 6.7) and is drained off. This condensate has no relation to the system condenser, which condenses refrigerant vapor. Another source of confusion arises because the evaporator is frequently called a cooling coil due to its cooling action. This is particularly common when the evaporator is installed in a warm air furnace and connected to a remote compressor and condenser. See for instance Figures 5.8, 5.9, 5.10, 5.11 and 5.13. Finally, the evaporator is also frequently referred to as a DX (direct expansion) coil in systems such as that shown in Figure 6.7. A DX coil absorbs heat by the direct expansion of the refrigerant liquid to a gas. The desired cooling action occurs when recirculated room air is blown over the coil. When the cold surface of an evaporator

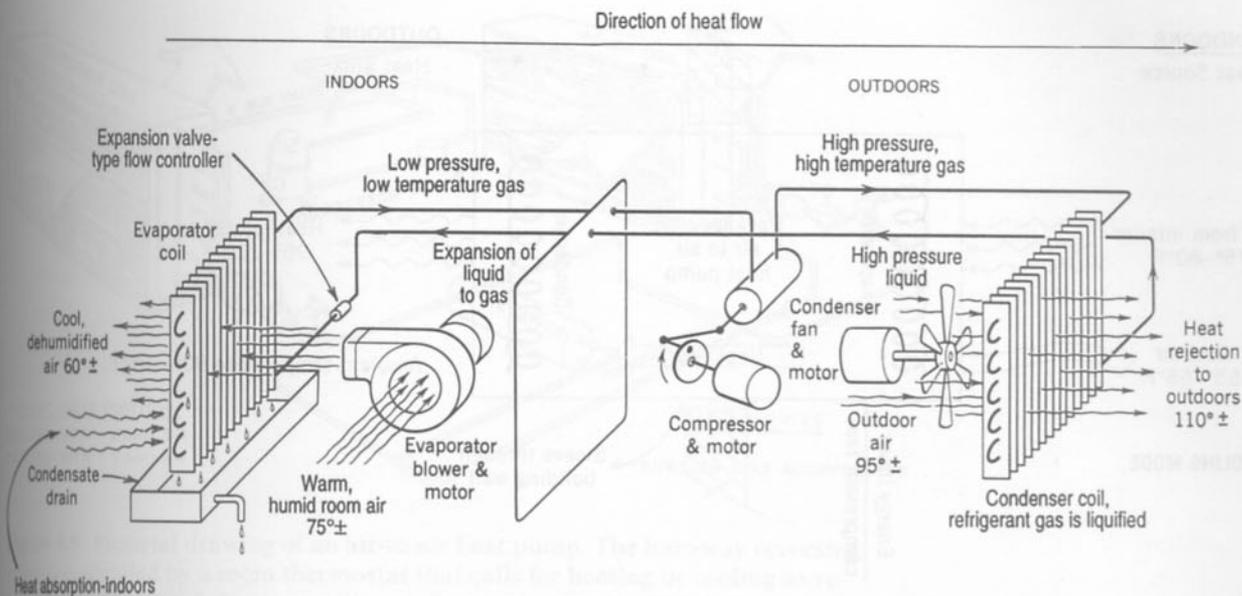


Figure 6.7 Pictorial representation of the components of a vapor compression refrigeration system as applied to comfort cooling. The indoor and outdoor sections can be installed in a single enclosure, as in the common window unit or through-the-wall package unit. Alternatively, the two sections can be separated by up to 100 ft with only insulated refrigerant pipes connecting the two sections. See for instance Figure 5.6(a).

used to cool water that will then be circulated through the building as a cooling medium, the entire assembly is known as a chiller rather than an air conditioner.

Heat Pumps

6.4 Basic Heat Pump Theory

In Section 6.3, we explained the basic theory of the vapor compression cooling cycle. Refer to Figure 6.7. Notice that the entire complex system—evaporator, compressor, condenser, piping and valving—accomplishes only one simple task. That task is to transfer heat from one place to another. The system does not create heat (except for machinery friction); it transfers heat. In the air-to-air system shown in Figure 6.7, the system takes heat from the warm indoor space and dumps it outside. Since the temperature outdoors far exceeds the indoor

temperature, the heat is being transferred (pumped) “uphill.” We learned in Section 1.7 that, according to the basic laws of thermodynamics, heat naturally flows “downhill,” that is, from a point of higher temperature to a point of lower temperature. To reverse this process and pump heat from a lower to a higher temperature, we must add energy. That is what the compressor does—it adds energy to the system, which acts using the system machinery, to pump the heat “uphill.” The two air movers shown (condenser fan and evaporator blower) are simply devices that aid in the heat transfer. The actual heat transfer is accomplished by phase changes of the refrigerant (liquid to gas to liquid). To simplify matters somewhat, we can redraw Figure 6.7 as the block diagram of Figure 6.8(a), showing only energy considerations. There we see that heat is pumped from a lower temperature to a higher temperature, using electrical energy input to do so.

Look at Figure 6.8(a), and forget for the moment that it is a block diagram of an air conditioner

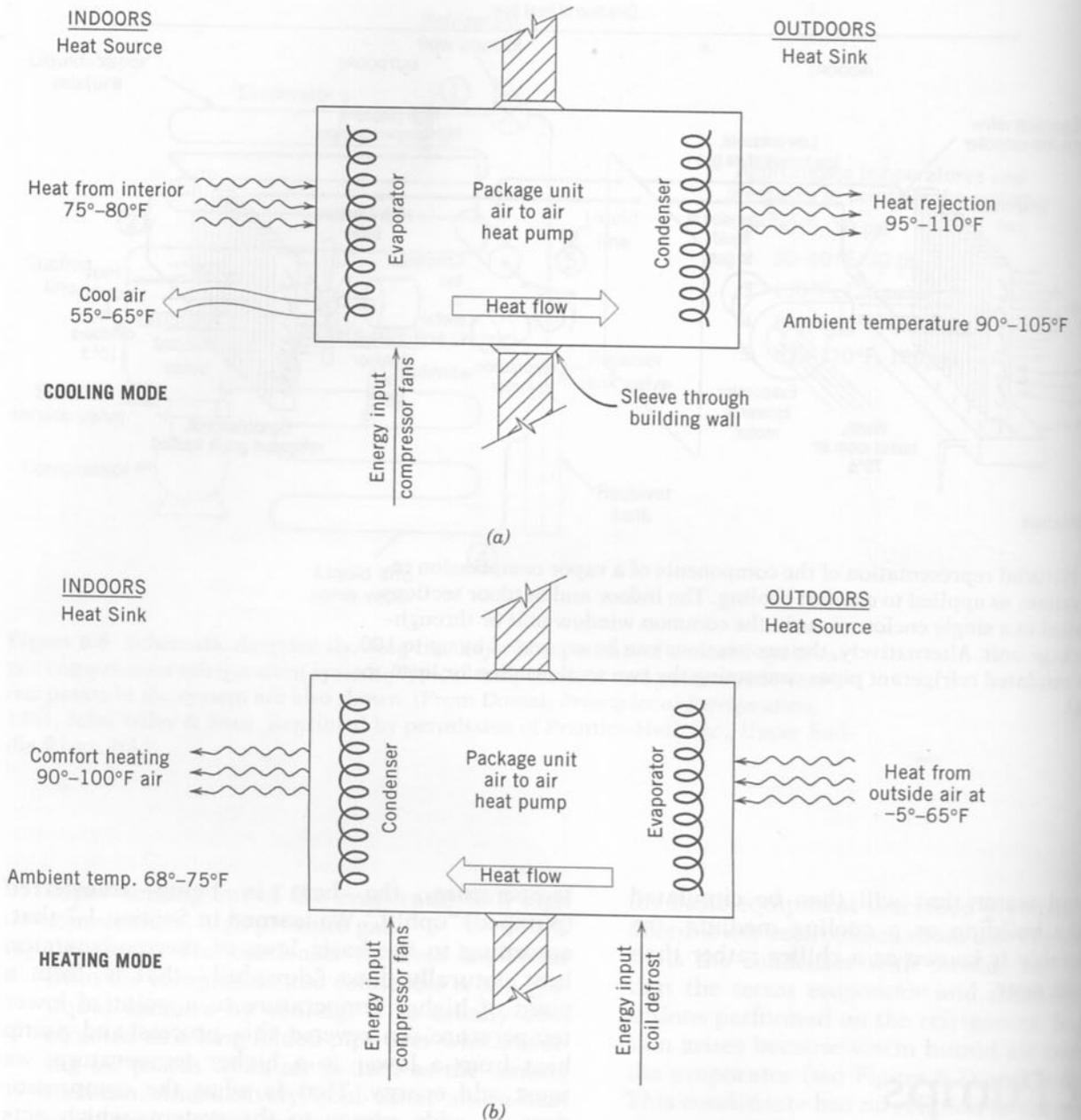


Figure 6.8 Heat flow diagrams for the two modes of operation of an air-to-air heat pump. (a) In the cooling mode, operation is identical to that of an air-to-air through-the-wall package-type air conditioner. Some of the heat in warm (humid) inside air is picked up by the indoor evaporator, carried through the unit and rejected outdoors. The exterior condenser is cooled by ambient outside air. Energy for the heat pumping is supplied by the compressor plus the evaporator and condenser fans. (b) In heating mode, heat is extracted from outside air and is pumped into the building interior. It is rejected inside in the form of warm air for comfort heating. Energy for the heat-pumping operation is supplied by the compressor. Additional energy is taken by fans and by electric defrosting heaters for the exterior evaporator.

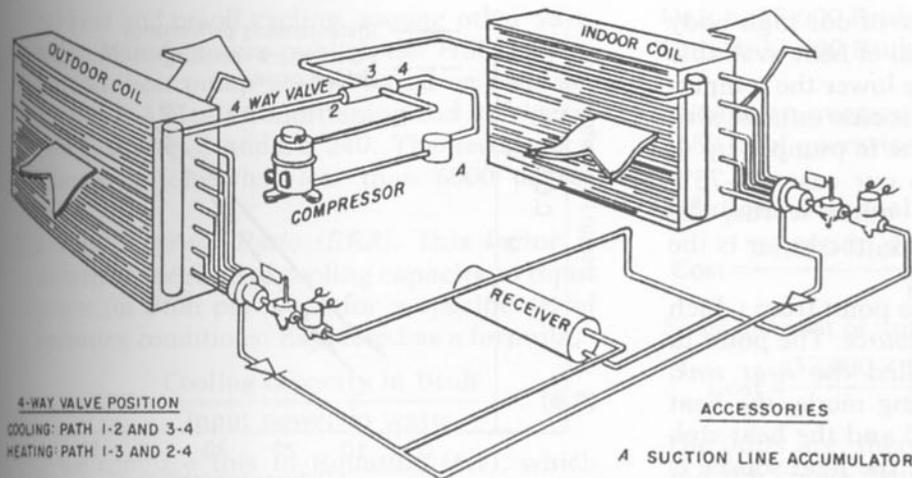


Figure 6.9 Pictorial drawing of an air-to-air heat pump. The four-way reversing valve is controlled by a room thermostat that calls for heating or cooling as required. For the sake of clarity, auxiliary valves and control/safety devices are not shown. (From Ambrose, *Heat Pumps and Electric Heating*, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

intended for cooling. It should be obvious that the vapor compression system can also be used for heating by simply turning it around, as is shown schematically in Figure 6.8(b). In actual practice, it is not necessary to turn the unit around physically. All that is needed is a four-way valve that will reverse the flow of refrigerant. Since the evaporator and condenser are both coils, they can act as either one or the other, depending on refrigerant flow. That means that reversing flow into the evaporator changes it into a condenser. Similarly, reversing flow through a condenser changes it into an evaporator. Such a complete assembly, including the reversing valve, is called a *heat pump*. It is capable of heating or cooling, depending on its control valve position. In the cooling mode, it is identical to what is commonly called an air conditioner. In its heating mode, it is always referred to as a heat pump. Figure 6.9 schematically shows the refrigerant flow and flow valve position in both modes. The position of the valve is controlled by the indoor thermostat. When heating is required, it sets the four-way flow valve into heating position; when cooling is required, it resets the valve into the cooling position.

Another attractive characteristic of the heat pump is its very efficient operation in the heating mode. This will be discussed in Section 6.5. The

question that arises at this point is something like, If the heat pump can supply both cooling and heating, and the latter very efficiently, why would anyone buy separate heating and cooling systems, as is so often done? The answer to that excellent question has to do with both economics and engineering. It will become clear in the following discussion.

6.5 Heat Pump Performance (Heating Mode)

a. Heat Extraction

Refer to Figure 6.8(b). The question that usually arises when referring to heating mode performance is how usable heat can be extracted from cold outside air. A moment's thought, however, will answer this question. A complete absence of heat occurs at absolute zero temperature, which corresponds to 0° Rankine or -460°F . At any temperature above absolute zero, air contains heat. The specific heat of dry air is $0.24 \text{ Btu/lb}\cdot^\circ\text{F}$ or $^\circ\text{R}$. That means that at 100°F (560°R) air contains 134 Btu/lb . At a typical winter temperature of 40°F (500°R), air contains 120 Btu/lb or 90% of the heat content at 100°F ! Therefore, despite the fact that we tend to

think of 40°F air as cold because of our high body temperature, there is a good deal of heat available for use in such air. Of course, the lower the temperature of the heat source is (in this case outside air), the more work that must be done to pump it up to the heat sink temperatures (in our case 68–75°F indoor room temperature). In other words, the lower the outside air temperature, the lower is the efficiency of a heat pump.

A word about terminology: the point from which heat is taken is called the *heat source*. The point to which heat is delivered is called the *heat sink*. Thus, for a heat pump in cooling mode, the heat source is the room being cooled and the heat sink is outside air. In heating mode, the heat source is outside air, and the room is the heat sink.

b. Heat Pump Heating Efficiency

The law of conservation of energy states that energy cannot be created or destroyed. Refer again to Figure 6.8*b*; the energy (heat) extracted from the outside air plus the energy input to the compressor appear as heat input into the space. In other words, the heat produced by the heat pump is greater than its energy input. The ratio of heat output to energy input is called its coefficient of performance (COP). The COP of a well-designed heat pump varies between 1.5 and 3.0, depending on the outside temperature. It is defined as

$$\text{COP} = \frac{\text{Heat delivered in Btuh}}{\text{Energy supplied in Btuh}} \quad (6.1)$$

The difficulty with heat pumps is that heat output and, therefore, also COP drop as outside temperature drops. That means that as the weather turns colder and the demand for heat increases, the heat pump output decreases, simply because the heat must be pumped over a larger temperature difference. This is shown graphically in Figure 6.10(*a*).

To make things just a bit more complicated, three other heat pump performance factors are in common use. The informed technologist should understand their meaning.

(1) Heating Seasonal Performance Factor (HSPF).

This factor is more meaningful than COP because it considers the heat pump's performance over the entire heating season. COP is a measure of what the heat pump is doing at a particular instant and, in turn, depends on indoor and outdoor temperatures. HSPF is an indicator of seasonal efficiency, including supplement-

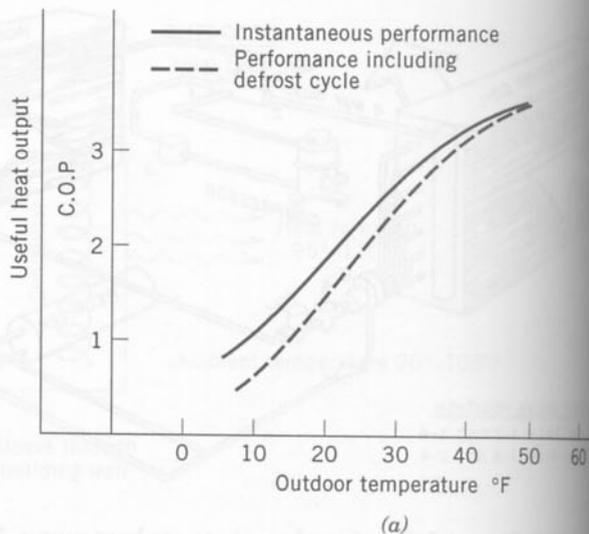


Figure 6.10 (a) Heat pump heating cycle performance. Note that useful output and coefficient of performance both drop sharply as the outdoor temperature drops. Outside air is used as the heat source for this air-to-air heat pump characteristic.

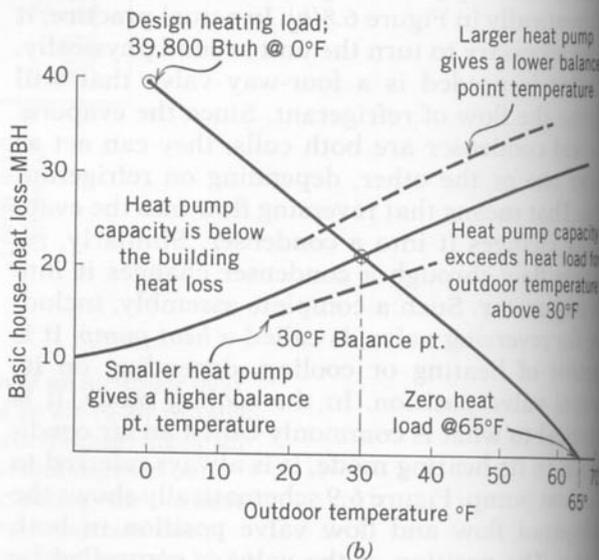


Figure 6.10 (b) The intersection of the building heat loss curve (straight line) and the heating characteristic of a proposed heat pump (curved line) determines the system balance point (see text). The balance point temperature can be raised by using a smaller unit or lowered by using a larger unit.

tal heat and on-off cycling, among other variables. Manufacturers publish the HSPF figure for each heat pump model, based on tests according to ARI (Air Conditioning and Refrigeration Institute) Standard 240. The tests use a climate of somewhat less than 6000 degree days.

- (2) *Energy Efficiency Ratio (EER)*. This factor is defined as the ratio of cooling capacity to input power (in Btuh per watt) for a specific set of operating conditions. Expressed as a formula,

$$EER = \frac{\text{Cooling capacity in Btuh}}{\text{Input power in watts}} \quad (6.2)$$

If we compare this to Equation (6.1), which defines COP, we see that the two factors are related by the conversion factor of 3.412 Btuh/w, that is

$$EER = 3.412 \times COP \quad (6.3)$$

therefore, a COP of 2.0 is the same as an EER of 6.8 and so on. EER factors are most frequently applied to consumer refrigeration products such as room air conditioners and household refrigerators.

- (3) *Seasonal Energy Efficiency Ratio (SEER)*. This factor is an indication of the heat pump's performance over the entire cooling season. It is determined by testing a unit with up to four separate tests, each at varying indoor and outdoor conditions that simulate varying weather conditions. Therefore, it is to cooling what HSPF is to heating—an attempt to simulate efficiency over an entire season.

The EER and SEER ratings of two heat pumps can be compared directly only if their Btuh capacities are identical. If this is not true, the seasonal electrical energy bill for each unit can be calculated and the results compared. The seasonal electrical energy cost is

$$\text{Electrical cost} = \frac{\text{Btuh} \times \text{Hours} \times \text{Rate}}{(\text{EER or SEER}) (1000)} \quad (6.4)$$

where

Btuh is the unit's rating,

Hours is the anticipated number of operating hours per season and

Rate is the local electrical rate in dollars per kilowatt-hour.

Example 6.2 Compare the seasonal cost of two heat pump units:

Unit A: 28,000 Btuh, SEER = 7.4

Unit B: 32,000 Btuh, SEER = 8.6

Assume an average electrical cost of 8.5¢/kwh and 1680 hr of operation (4 months at 14 hr per day).

Solution: Seasonal cost of unit A:

$$\text{Cost} = \frac{28,000 \text{ Btuh} \times 1680 \times 0.085 \text{ \$/kwh}}{7.4 (1000)} = \$540.00$$

Seasonal cost of unit B:

$$\text{Cost} = \frac{32,000 \times 1680 \times 0.085 \text{ \$/kwh}}{8.6 (1000)} = \$531.00$$

The units are, therefore, equal in energy expense, and the choice between them would be made on some other basis.

Typical performance figures for residential heat pump performance are:

COP	2.0–3.5
HPSF	6.0–8.5
EER	6.8–11.9
SEER*	10–12.5

All air source heat pumps will build up a layer of frost on the outdoor evaporator as the outside temperature drops below 45°F. Since this ice layer will seriously degrade heat pump performance, heat pumps are provided with automatic defrost cycles. This requires energy and serves to reduce overall efficiency further as the outdoor temperature falls. The heat pump capacity rating, when considering the defrost heat "penalty," is referred to as the unit's *integrated capacity*. It too is shown on the graph of Figure 6.10(a).

Most heat pumps use electricity as fuel. Since electricity is generally more expensive than fossil fuels in dollars per Btuh, it is generally not economical to operate a heat pump for comfort heating when the COP falls below about 2.0. This usually occurs at an outdoor temperature between 20 and 30°F for air-to-air units. The subject of the economics of heat pump operation is complex and is not normally the concern of a technologist. It is touched on here to give the technologist an appreciation of the considerations involved in equipment and system selection.

*The National Appliance Energy Conservation Act of 1987 prohibits the manufacture of any air conditioner with a SEER less than 10.

6.6 Equipment Sizing Considerations

a. Heat Pump Balance Point

In order to determine the theoretical ideal size for a heat pump in heating mode, a graphical solution method is used. The method consists of plotting the building heating load characteristic and a heat pump curve together and finding their intersection. Refer to Figure 6.10(b). It is assumed that a building's heating load varies linearly with temperature. This simply means that a change in outdoor temperature will cause a proportional change in building heat loss (load). This is substantially true if we consider only the heat loss through the building envelope and ignore infiltration and deliberate ventilation losses. For the purpose of heat pump sizing, when designing residential systems, this assumption can be made without introducing an excessive error.

To draw the heat load characteristic, plot two points: the calculated design heating load at the design temperature and the no-load (no heat) point at 65°F. If we take The Basic House plan from Figure 3.32(c) as an example (page 131) we see that the calculated heating load at 0°F design temperature is 39,800 Btuh. Plotting the two points on Figure 6.11 and connecting them with a straight line gives us the theoretical building heat load at every outside temperature from 0 to 65°F. If we now plot on the same sheet the performance curve of the heat pump being considered, the intersection point of the two lines is called the *system balance point*. (The heat pump performance curve data are available from the equipment manufacturer's published technical material.) At the balance point, the heat pump output exactly matches the building's heat loss. At outdoor temperatures above the balance point, the heat pump has excess capacity. At temperatures below the balance point, the heat pump output is insufficient, and additional heat is required. This is indicated on Figure 6.10(b).

For commercial buildings, the concept of a system balance point is still valid. However, the building load line is not as easily determined as for a residence. In determining the heat load, the ventilation portion of the heat load cannot be ignored. Furthermore, the no-load condition cannot be set simply at 65°F outdoor temperature. When the building is occupied, it has internal heat gain from occupants, machinery and solar load. These will reduce the no-load outside temperature to 45°F or

even lower, depending on the building details. Since most commercial building cooling calculations are done today by computer, the design technologist can easily calculate the building heat load at 5–10 F° intervals from the outside design temperature up to no-load. These points can then be used to plot the building heat loss characteristic. In commercial buildings with multiple zones, separate calculations have to be made for each zone. Design considerations for such buildings are beyond the scope of this book.

A balance point at 30°F or lower will ensure excellent heating efficiency from the heat pump for all of the building's needs. A balance point between 30 and 35°F will give good heating efficiency except on the few days every year or so when the outdoor temperature actually drops to the design temperature figure. On those few days, the house may be cooler than the design indoor condition (68–75°F), and some additional heat may be desirable. However, the actual operating balance point is almost always lower than the one calculated because no allowance for internal heat gains and solar gain is taken in residential heat load calculations. As a result, a balance point between 30 and 35°F will seldom require the use of supplemental heating equipment. A balance point above 35°F means either a very cold climate, an undersized heat pump or both. The solution to that situation will require one of the following:

1. Separate supplemental heating, either electric or fossil fuel.
2. A larger heat pump unit.
3. A modified heat pump system that will supply the required additional heat. This may be a hybrid heat pump—fossil fuel system or simply a heat pump with auxiliary electric resistance heating elements. This latter is the cheapest alternative. As a result, most heat pumps intended for residential use are provided with supplementary electric resistance heating elements.

An additional consideration in the sizing a heat pump is its functioning in cooling mode. This is discussed next.

b. Heat Pumps in Cooling Mode and Air Conditioners

As already stated, the heat pump in cooling mode is identical to what is commonly known as an air

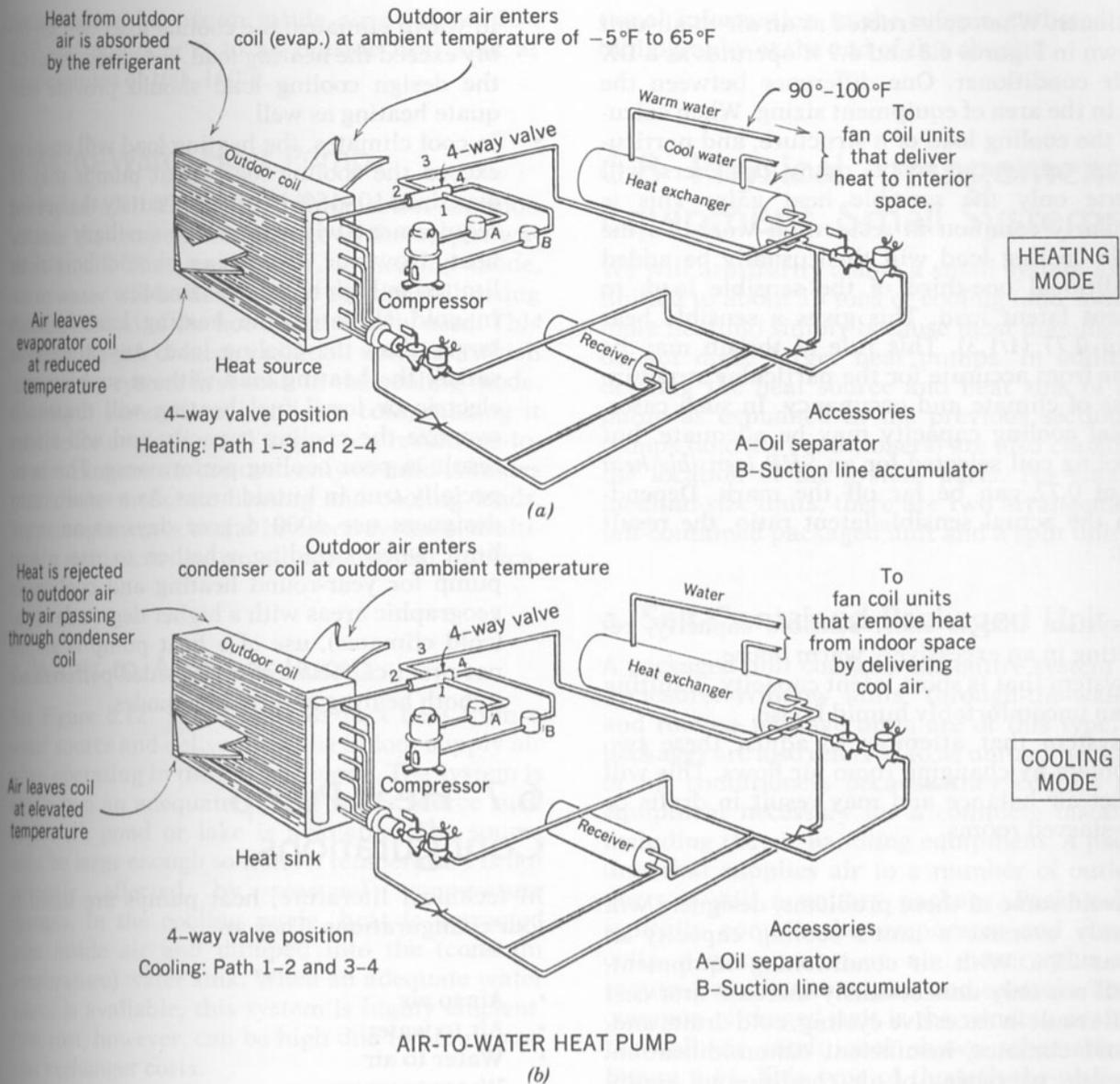


Figure 6.11 Air-to-water heat pump energy flow. (a) In the heating mode, heat flows into the refrigerant at the outdoor coil and is rejected to the water in the heat exchanger. The heated water is then circulated to fan coil units to heat the indoor space. (b) In the cooling mode, heat is picked up from the indoor space by circulating room air over the (cool) water coils in the fan coil unit. This heat is rejected at the outdoor coil acting as a refrigerant condenser. (From Ambrose, *Heat Pumps*, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

conditioner. When constructed as an air-to-air unit as shown in Figures 6.8 and 6.9 it operates as a DX coil air conditioner. One difference between the two is in the area of equipment sizing. When calculating the cooling load of a structure, and particularly for residential work, many designers will calculate only the sensible heat gain. This is particularly common in residential work. To the sensible cooling load will then usually be added an additional one-third of the sensible load, to represent latent load. This gives a sensible heat ratio of 0.77 (1/1.3). This rule of thumb may be very far from accurate for the particular structure because of climate and occupancy. In such cases, the total cooling capacity may be adequate, but the cooling coil selected for an *SHR* (*sensible heat ratio*) of 0.77 can be far off the mark. Depending on the actual sensible/latent ratio, the result can be:

- A system that is short sensible capacity, resulting in an excessively warm house.
- A system that is short latent capacity, resulting in an uncomfortably humid house.
- A system that attempts to adjust these two problems by changing room air flows. This will upset air balance and may result in drafts or air-starved rooms.

To avoid some of these problems, designers will frequently oversize a unit's cooling capacity as much as 25%. With air conditioning equipment, this will not only unnecessarily increase first cost but will result in excessive cycling, cold drafts and, in humid climates, insufficient dehumidification. As a result, oversizing of air conditioning equipment is discouraged. Indeed, some designers recommend deliberate undersizing, particularly in humid climates. This ensures almost continuous unit operation and adequate dehumidification. The solution to these problems, of course, is to calculate both sensible and latent loads as accurately as possible. A cooling coil matching the calculated requirements can then be selected from manufacturers' published data.

When sizing a heat pump for cooling capacity, oversizing the cooling capacity will result in better heating performance and less need to use supplemental electric or fossil fuel heat. The following suggestions, based on experience, should be helpful in sizing heat pumps for heating and cooling service:

- In warm climates, the cooling load will probably exceed the heating load. Selecting a unit for the design cooling load should provide adequate heating as well.
- In cool climates, the heating load will equal or exceed the cooling load. Heat pumps may be oversized 10–15% in order to satisfy the heating requirement without use of auxiliary electric heat. However, oversizing should be strictly limited in high humidity climates.
- In cold climates, the heating load is much larger than the cooling load. Any attempt to satisfy the heating load without supplemental electric or fossil fuel heating will drastically oversize the cooling capacity and will always result in poor cooling performance. This is especially true in humid areas. As a result, many designers use 6000 degree days as an upper limit when deciding whether to use a heat pump for year-round heating and cooling. In geographic areas with a higher degree day total (cold climates), use of a heat pump is simply not practical because of degraded performance in both heating and cooling modes.

6.7 Heat Pump Configurations

In technical literature, heat pumps are listed in four configurations. They are

- Air to air
- Air to water
- Water to air
- Water to water

The first word refers to the heat source and the second to the heat sink. These names refer to the heat pump in its heating mode.

a. Air-to-Air Heat Pump

See Figures 6.8 and 6.9. In the heating mode, this arrangement extracts heat from outside air at temperatures of -5° – 65° F and delivers it to inside spaces at temperatures of 90 – 100° F. This type of unit is the most common arrangement for residential work, as it requires only an electrical connection and no special piping. In cooling mode [Figure 6.8 (a)], the unit operates like a common air condi-

tioner, taking heat from inside air at about 75–80°F and pumping it to an outside heat sink at temperatures of 100–110°F.

b. Air-to-Water Heat Pump

See Figure 6.11. This configuration is not common because the temperature change in the output water is small. That means that, in heating mode, warm water will be delivered to a hydronic heating system, instead of the hot water usually used. This requires the use of large radiation surfaces, which increase the system first cost. In the cooling mode, the output water is cool and not cold, making it suitable for use in fan coil units. As a result, air-to-water arrangements are used only in mild climates with low to moderate heating and cooling loads. The arrangement is useful in zoned systems, multi-family residences and small commercial applications.

c. Water-to-Air Heat Pump

See Figure 6.12. These units extract heat from a water source and deliver it to the indoor supply air when operating in the heating mode. The system is used when an adequately large water source such as a well, pond or lake is available. The source must be large enough so that its temperature is not seriously affected by seasonal temperature changes. In the cooling mode, heat is extracted from inside air and dumped into the (constant temperature) water sink. When an adequate water source is available, this system is highly efficient. First cost, however, can be high due to piping and heat exchanger costs.

d. Water-to-Water Heat Pump

See Figure 6.13. These units use heat exchangers at both ends of the system. Like the water-to-air system, a large reliable water source/sink is also required. As a result, this arrangement is most often used in commercial applications. Efficiency of the system is high because no defrosting cycle is required.

In addition to these four standard arrangements, there are designs using coils buried in the earth, hybrid designs using solar collectors and others. These types are highly specialized and must be designed for a specific application. Technologists interested in these special designs will find addi-

tional information in the references listed in the bibliography at the end of this chapter.

6.8 Physical Arrangement of Equipment, Small Systems

We will arbitrarily define a small system as being limited to about 15 tons of cooling (and somewhat more heating) simply because most manufacturers do not make larger heat pumps. In addition to defining the heat source and heat sink of a heat pump as explained in the previous section, heat pumps (and air conditioners) are also classified by the location of the system parts. For small and medium-size units, there are two arrangements: a self-contained packaged unit and a split unit.

a. Self-Contained Packaged Unit

A packaged unit contains the entire system in one enclosure. Window units, through-the-wall units and rooftop package units are of this type. These packages are also referred to as unitary heat pumps or air conditioners because they contain all the equipment necessary for a complete installation, including the air-handling equipment. A packaged unit that supplies air to a number of outlets via ducts is still a unitary package. Packaged units typically contain the evaporator and condenser coils, a compressor, an air mover, plus all the required auxiliaries, piping and controls. The most common packaged unit is the window or through-the-wall air conditioner shown schematically in Figure 6.14. This type of through-the-wall unit is referred to as an incremental unit, a PTAC (packaged terminal air conditioner) or a PTHP (packaged terminal heat pump). The same equipment can be arranged for rooftop mounting with air discharging at the ceiling level of the space to be cooled. Since cold air is heavier than warm air, a rooftop arrangement is ideal for cooling. It is less effective in the heating mode, because the warm air output will simply stay at the ceiling level, and circulation will be poor. See Figure 5.38 (page 000). For this reason, packaged rooftop units are most commonly used for cooling only.

Figure 6.15 shows a typical through-the-wall climate control package consisting of an electric air conditioner and a gas heater in a single enclosure. Warm or cool air, as called for by the interior

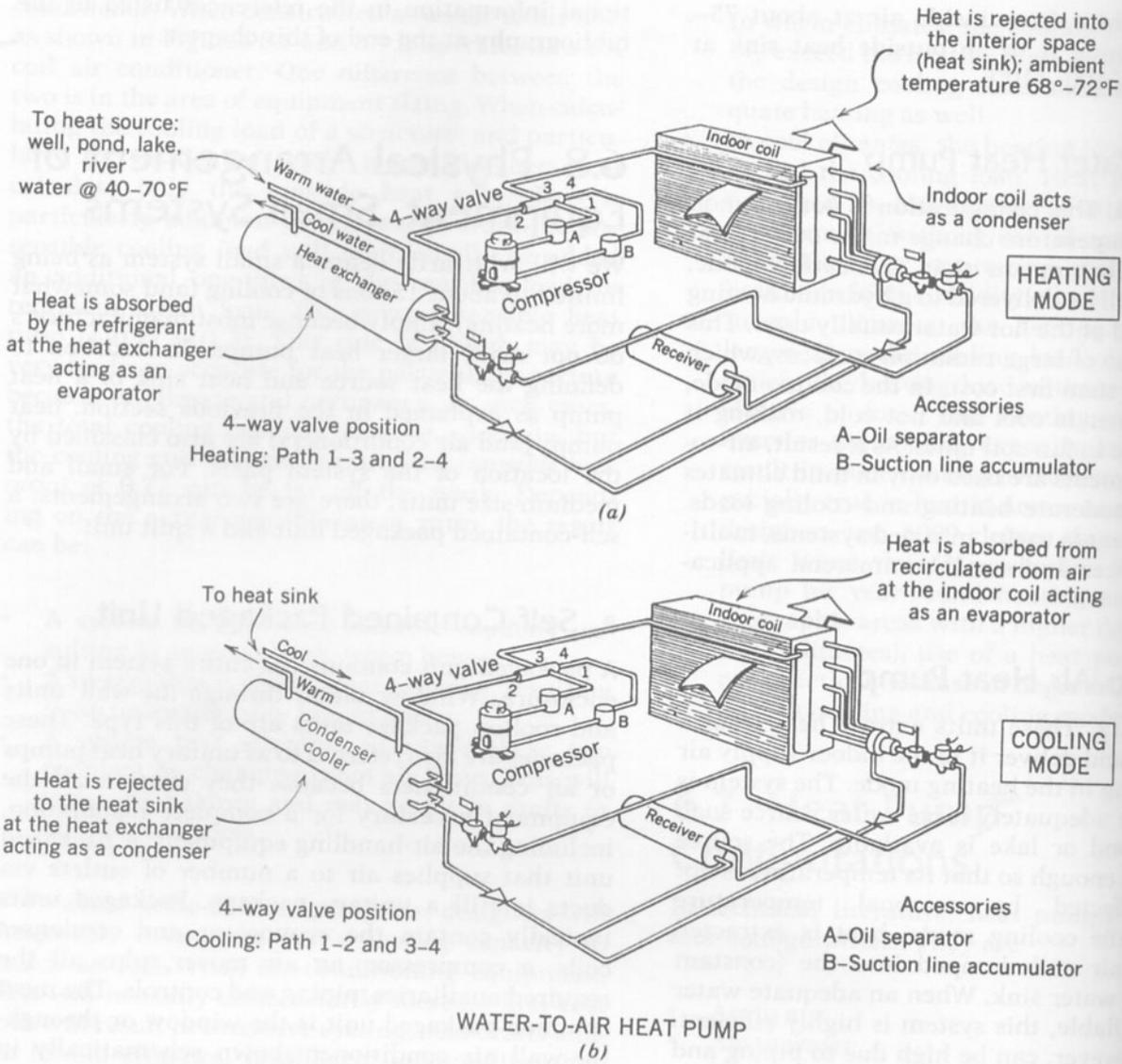


Figure 6.12 Water-to-air heat pump. (a) In the heating mode, water pumped from the water source (heat source) gives up heat at the heat exchanger. This is transferred by the refrigerant to the indoor coil. There, the indoor coil acting as a condenser rejects the heat into the indoor space (heat sink). (b) In the cooling mode, heat is absorbed from the indoor space air by circulation over the indoor coil, which acts as an evaporator. The heat is transferred to the heat exchanger where it is picked up by circulating water from the water source heat sink. (From Ambrose, *Heat Pumps*, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

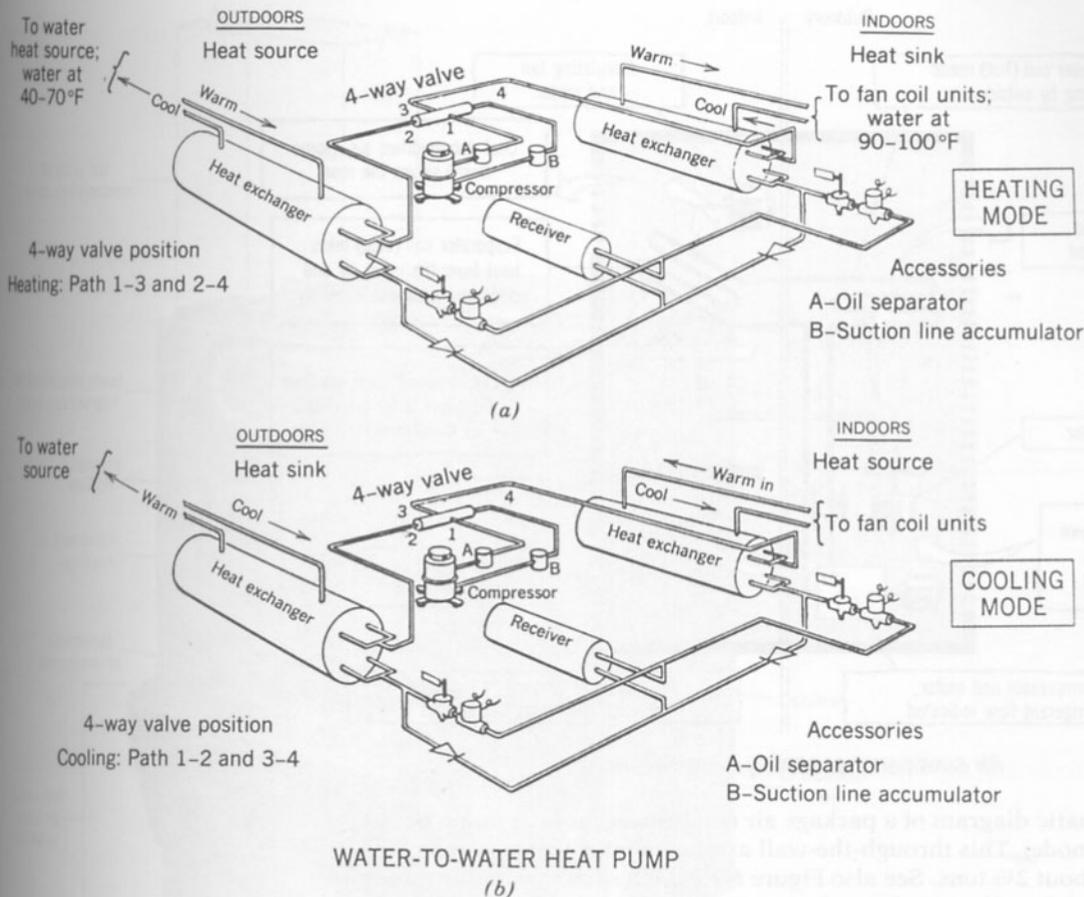


Figure 6.13 Water-to-water heat pump. These units require a body of water to act as both a heat source and heat sink. The indoor section of the heat pump supplies fan coil units or similar terminals that will operate on relatively small temperature differentials in water. (a) In the heating mode, source water flows through the heat exchanger acting as an evaporator. Heat is drawn into the refrigerant source and delivered to the indoor heat exchanger, which acts as a condenser. Heat is rejected there, warming the interior space. (b) In the cooling mode, heat picked up from the fan coil water circuit by the indoor coil acting as an evaporator is transferred to the outdoor heat exchanger, which acts as a condenser. From the heat exchanger, heat is transferred to the water source, which acts as a heat sink. (From Ambrose, *Heat Pumps*, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

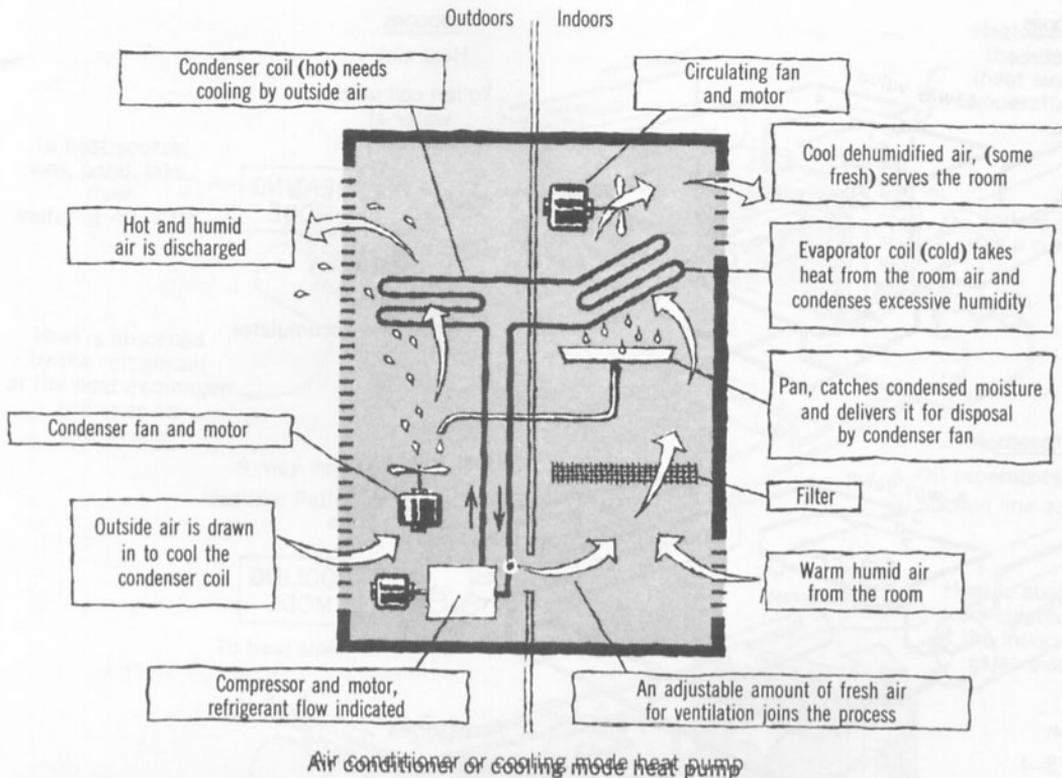
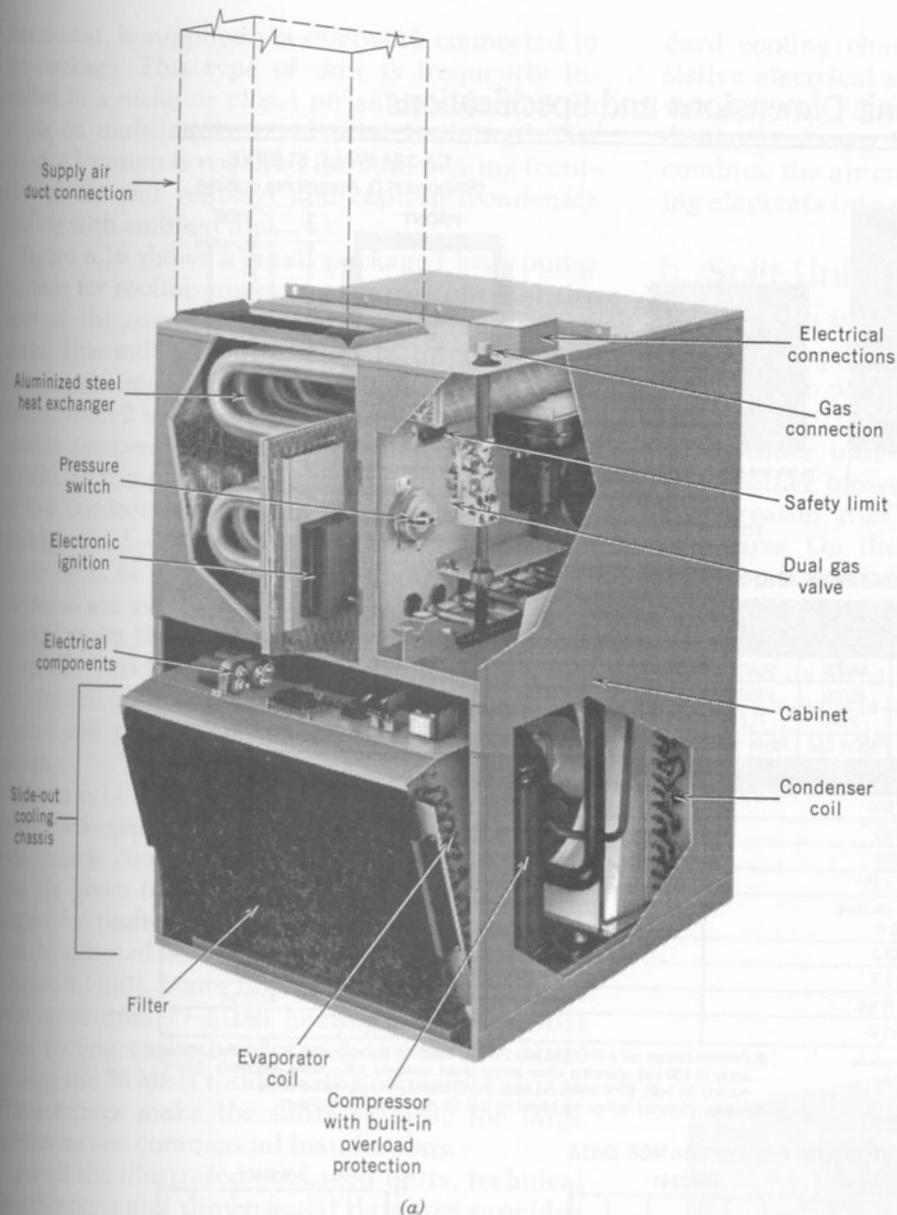


Figure 6.14 Schematic diagram of a package air conditioner (or heat pump operating in cooling mode). This through-the-wall arrangement is very common for small units up to about 2½ tons. See also Figure 6.7, which shows a similar through-the-wall (or window) installation of a package unit. Note that the evaporator and room air blower are always inside and that the compressor, condenser and condenser fan are always outside.

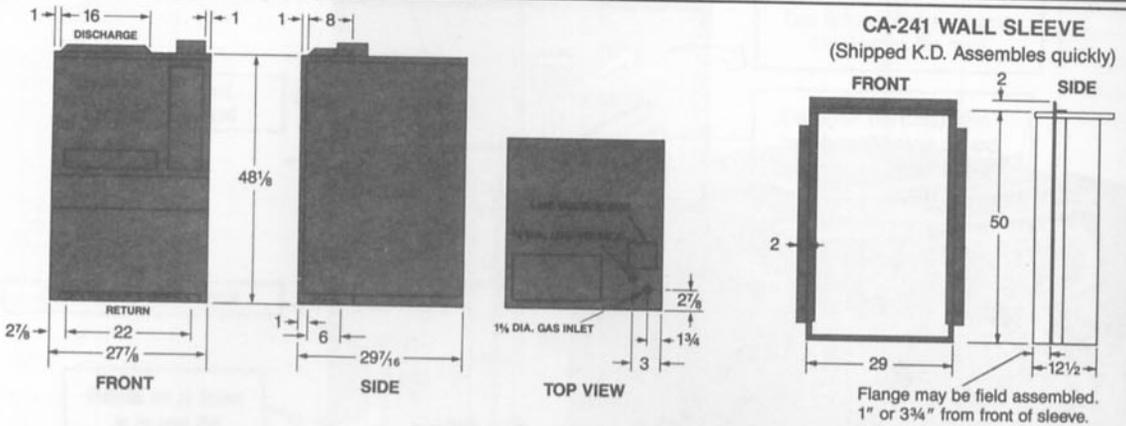


(a)

Figure 6.15 (a) Combination through-the-wall gas heating and electric cooling (air conditioner) unit. The cool air supply duct is connected to the top of the unit, as shown. Combustion air is drawn in through the exterior exposed surface of the unit (the rear in this photo). A built-in power vent eliminates the need for a chimney. Similar units are available as all-electric heat pumps and as electric heating, electric cooling (air conditioner) units. (b) Dimensional and technical data for the units shown in (a). Btuh capacities vary from 27 to 50 MBH in heating and 1½ tons (17,600 Btuh) to 2½ tons (29,200 Btuh) in cooling at a seasonal energy efficiency ratio of 8. (Courtesy of Armstrong Air Conditioning, a Lennox International company.)

HWC Series

Higher Efficiency Cooling Unit Dimensions and Specifications



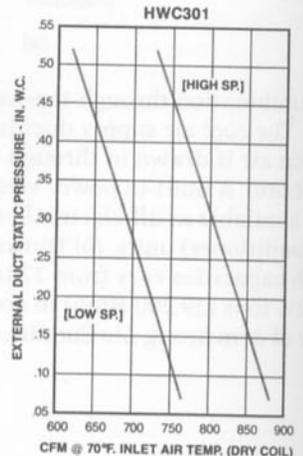
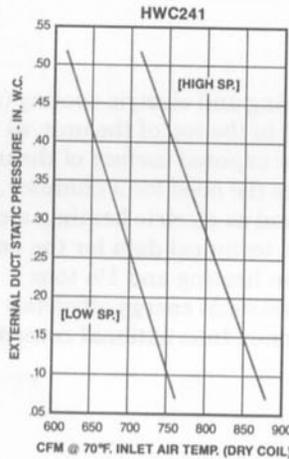
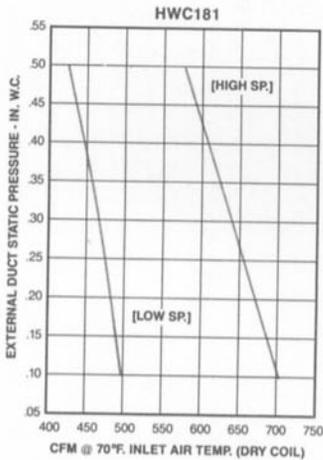
RATINGS AND SPECIFICATIONS

HEATING SECTION											
	36,000	48,000	60,000	66,000	36,000	48,000	60,000	66,000	60,000	66,000	
Rated Input - Btu/hr.	36,000	48,000	60,000	66,000	36,000	48,000	60,000	66,000	60,000	66,000	
Capacity - Btu/hr. ①	27,000	36,000	45,000	50,000	27,000	36,000	45,000	50,000	45,000	50,000	
Efficiency - A.F.U.E. ●	70.8	70.8	71.6	71.2	70.8	70.8	71.6	71.2	71.6	71.2	
CFM @ Heating Speed	500 Lo Spd	500 Lo Spd	700 Hi Spd	700 Hi Spd	750 Lo Spd	750 Lo Spd	750 Lo Spd	750 Lo Spd	750 Lo Spd	750 Lo Spd	
COOLING SECTION											
	230-250 VOLTS — SINGLE PHASE				208-230 VOLTS — THREE PHASE						
Capacity - Btu/hr.	17,600				23,800			29,200			
Efficiency - S.E.E.R.	8.00				8.05			8.05			
CFM (Hi-Spd.)	650				800			780			
Filter Size	13" x 25"				18" x 25"			18" x 25"			
Compressor	P.S.C. Hermetic				P.S.C. Hermetic			P.S.C. Hermetic			
R.L.A.	8.0				12.8			15.5			
L.R.A.	43.3				60.0			70			
Min. Circuit Ampacity	13.3				21			23.6			
Max. Circuit Fuse Size	20 Amps				30 Amps			35 Amps			
Approx. Ship. Weight	400 lb.				415 lb.			430 lb.			

Provision for condensate drain must be provided inside structure.
 ① Capacity ratings are based on D.O.E. standard tests.

● Certified ratings per A.R.I. Standard 210 and ASHRAE Standard 116. BTUH ratings shown apply to 230-volt operation when tested under standard A.R.I. rating conditions of 95°F outside dry bulb, 80°F inside dry bulb and 67° inside wet bulb temperatures.
 ● Energy efficiency ratings are based on U.S. Government standard tests.

BLOWER PERFORMANCE DATA



(b)

Figure 6.15 (Continued)

thermostat, is supplied via ductwork connected to the package. This type of unit is frequently installed in a niche or closet on an outside wall, in single or multifamily residential buildings. The exterior location is required for both heating (combustion air and venting) and cooling (condenser cooling with ambient air).

Figure 6.16 shows a small packaged heat pump suitable for rooftop mounting, or mounting at the level of the conditioned space, using side entry ducts. This unit would be used in an all-electric installation. Models in this design have cooling ratings from 2 to 3½ tons. Heat output varies with outside temperature and is tabulated in Figure 6.16(b). To compensate for the drop in heat output at low outdoor temperatures, resistance heating packages of 5–20 KW are available. Other heat pump units of this design and slightly larger dimensions are available up to 5 tons cooling. These ratings make the units suitable for small to medium-size residences, stores and offices. Figure 6.16 (c) and (d) give electrical data, blower performance and dimensional data for models of the design.

Figure 6.17 shows a medium-size, commercial-grade packaged unit that also supplies gas heating and electric cooling. Heating and cooling capacities are given in Figure 6.17(b). This unit is designed for rooftop mounting [Figure 6.17(c)] but can be adapted for side entry of ducts as well [Figure 6.17(d)]. Units in this design have ratings that range from 77 MBH heating and 35.6 MBH total cooling capacity (3 tons) up to 150 MBH heating and 58 MBH total cooling capacity (5 tons). These ratings make the units suitable for large residences and commercial installations.

For all the illustrated packaged units, technical specifications and dimensional data are provided with the illustration. These should assist you in obtaining a “feel” for the relation between HVAC rating and the physical size of equipment. A very common application of a small package through-the-wall heat pump is shown in Figure 6.18. Here the noise generated by the heat pump can be useful in masking traffic noise and noise from adjoining rooms.

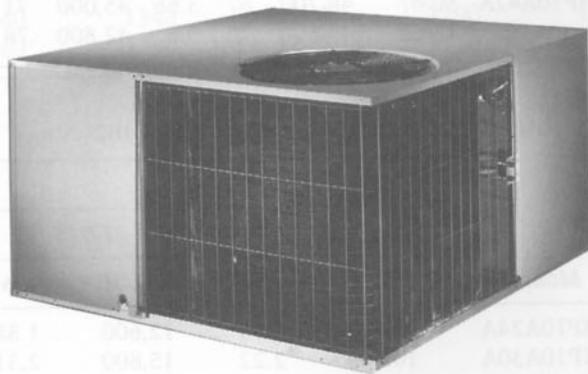
Another type of packaged unit is the PTAC mentioned previously. Although a through-the-wall heat pump is properly referred to as a PTHP, it is frequently (incorrectly) lumped with packaged air conditioners as a PTAC. Most through-the-wall PTAC units in speculative construction use a stan-

dard cooling chassis (air conditioner) plus a resistive electrical section (chassis), as shown schematically in Figure 6.19(a). A typical unit of this design is shown in Figure 6.19(b). Other designs combine the air conditioner and the resistive heating elements into one compact chassis.

b. Split Unit

If we look at Figures 6.7 and 6.15, we immediately see that the package air conditioner or heat pump is really two parts connected by a couple of refrigerant pipes. The indoor unit has a coil and blower; the outdoor unit contains the compressor, filter and another blower or fan. There is no good technical reason that would prevent separating the two parts. On the contrary, there are very good reasons, discussed previously, and listed next, to do so.

- *Noise.* As already noted, a package unit is noisy, even when it is installed outside, and air is ducted to the interior. Ducts are excellent noise channels.



(a)

Figure 6.16 (a) Package heat pump, suitable for rooftop mounting, with vertical supply and return, or mounting adjacent to the conditioned space, with horizontal duct connections [see (d)]. (Courtesy of Armstrong Air Conditioning, a Lennox International Company.)

Performance Data

Model	Cooling				HSPF	Heating				CFM
	BTUH	SEER	EER	S/T		47° F.		17° F.		
						BTUH	COP	BTUH	COP	
PHP10A24A	23000	10.00	9.20	.72	6.80	22600	3.10	12600	2.00	800
PHP10A30A	29000	10.00	9.10	.72	7.00	28600	3.10	15800	2.00	1000
PHP10A36A	34000	10.00	9.00	.74	7.20	34000	3.15	21000	2.20	1200
PHP10A36B	34000	10.00	9.00	.74	7.20	34000	3.15	21000	2.20	1200
PHP10A42A	40000	10.00	9.00	.74	7.20	40000	3.20	22600	2.20	1400

Cooling Performance—Extended Ratings

Model	Indoor Temp DBWB	Outdoor Temp.—DB														
		65 Deg.			82 Deg.			95 Deg.			105 Deg.			115 Deg.		
		BTUH	S/T	KW	BTUH	S/T	KW	BTUH	S/T	KW	BTUH	S/T	KW	BTUH	S/T	KW
PHP10A24A	85/72	26,400	.61	2.03	27,600	.65	2.43	24,000	.66	2.52	22,600	.68	2.72	19,600	.71	2.88
	80/67	25,200	.67	2.00	26,000	.70	2.40	23,000	.72	2.50	21,200	.76	2.68	18,000	.80	2.85
	75/62	23,800	.74	1.98	24,600	.78	2.38	20,000	.81	2.47	19,200	.85	2.65	15,800	.90	2.80
PHP10A30A	85/72	33,000	.60	2.55	33,600	.64	3.03	30,000	.64	3.20	28,200	.67	3.38	26,000	.69	3.58
	80/67	31,600	.67	2.54	32,200	.70	3.01	29,000	.72	3.18	27,000	.76	3.35	23,500	.80	3.49
	75/62	30,200	.74	2.52	30,600	.78	2.98	25,400	.81	3.06	24,000	.85	3.30	20,200	.91	3.37
PHP10A36A	85/72	37,000	.60	3.20	38,800	.65	3.63	36,000	.65	3.99	34,200	.68	4.25	32,000	.70	4.51
	80/67	36,200	.67	3.18	38,000	.71	3.60	34,000	.74	3.77	32,600	.77	4.03	28,200	.79	4.36
	75/62	34,800	.73	3.14	36,200	.78	3.55	31,000	.82	3.75	28,400	.86	3.98	23,800	.92	4.19
PHP10A36B	85/72	37,000	.60	3.20	38,800	.65	3.63	36,000	.65	3.99	34,200	.68	4.25	32,000	.70	4.51
	80/67	36,200	.67	3.18	38,000	.71	3.60	34,000	.74	3.77	32,600	.77	4.03	28,200	.79	4.36
	75/62	34,800	.73	3.14	36,200	.78	3.55	31,000	.82	3.75	28,400	.86	3.98	23,800	.92	4.19
PHP10A42A	85/72	49,600	.61	3.71	46,400	.65	4.20	44,800	.66	4.61	41,000	.70	4.80	36,800	.74	5.07
	80/67	48,200	.67	3.66	45,000	.71	4.16	40,000	.74	4.44	36,800	.78	4.65	32,000	.84	4.85
	75/62	44,600	.76	3.6	42,800	.78	4.08	38,500	.81	4.25	34,400	.86	4.46	30,000	.90	4.62

Heating Performance—Extended Ratings

Model	Outdoor Temp DB/WB									
	0/0		17/15		35/33		47/43		62/56	
	BTUH	KW	BTUH	KW	BTUH	KW	BTUH	KW	BTUH	KW
PHP10A24A	9,400	1.75	12,600	1.85	17,600	2.15	22,600	2.14	27,000	2.20
PHP10A30A	10,600	2.22	15,800	2.31	22,200	2.52	28,600	2.70	35,000	2.85
PHP10A36A	12,400	2.60	21,000	2.80	26,600	3.02	34,000	3.16	41,000	3.33
PHP10A36B	12,400	2.60	21,000	2.80	26,600	3.02	34,000	3.16	41,000	3.33
PHP10A42A	14,000	2.93	22,600	3.01	27,800	3.36	40,000	3.66	48,000	3.91

(b)

Figure 6.16 (b) Performance data, including extended ratings for high and low temperatures. Cooling ratings of models in this configuration vary from 23,000 Btuh (2 tons) to 40,000 Btuh (3½ tons). Auxiliary electric resistance heaters are available in ratings of 5 kw to maintain heating capacity at low outside temperatures.

Physical and Electrical Data

Model	Voltage Hz Phase	Normal Voltage Range	Min. Circuit Ampacity	Max. Fuse/ HACR Brkr.	Compressor		Outside Fan			Indoor Blower		Refrig. Charge (oz.)	Weight (lbs)	
					Rated Load (amps)	Locked Rotor (amps)	Dia. (in.)	Nom. RPM	Rated Load (amps)	Rated Watts HP	Wheel d x w (in.)			Rated Watts HP/AMP
PHP10A24A	208-230/60/1	197-253	15.8	25	9.8	56.0	18	1075	.90	1/8	10 x 6	1/2 2.6	75	260
PHP10A30A	208-230/60/1	197-253	20.6	30	13.7	75.0	18	1075	.90	1/8	10 x 8	1/2 2.6	73	280
PHP10A36A	208-230/60/1	197-253	21.7	30	13.8	78.8	18	1075	1.80	1/4	10 x 8	1/2 2.6	93	300
PHP10A36B	200-230/60/3	187-253	17.3	25	10.3	75.0	18	1075	1.80	1/4	10 x 8	1/2 2.6	93	300
PHP10A42A	208-230/60/1	197-253	26.6	35	17.1	105.0	18	1075	1.80	1/4	10 x 9	1/2 3.4	102	330

Blower Performance Data

Model	Blower Speed	CFM @ Ext. Static Pressure—in. W.C. w/o Filter(s)*						
		0.2	0.3	0.4	0.5	0.6	0.7	0.8
PHP10A24A	Hi	1100	1060	1000	940	880	800	720
	Med	940	890	870	840	800	720	660
	Low	850	800	790	770	750	670	600
PHP10A30A	Hi	1400	1350	1280	1200	1120	1030	920
	Med	1160	1120	1080	1030	980	900	780
	Low	1050	1020	1000	950	910	840	750
PHP10A36A	Hi	1400	1350	1280	1200	1120	1030	920
	Med	1160	1120	1080	1030	980	900	780
	Low	1050	1020	1000	950	910	840	750
PHP10A36B	Hi	1400	1350	1280	1200	1120	1030	920
	Med	1160	1120	1080	1030	980	900	780
	Low	1050	1020	1000	950	910	840	750
PHP10A42A	Hi	1640	1560	1500	1400	1300	1260	1160
	Med	1570	1500	1440	1340	1270	1200	1100
	Low	1480	1430	1360	1290	1230	1170	1050

*Add .10 to duct static for downflow CFM equivalent.

(c)

Figure 6.16 (c) Physical, electrical and blower data.

Dimensions

Model	A	B	C
PHP10A24	25 ¹ / ₄	17 ⁷ / ₁₆	20 ¹¹ / ₁₆
PHP10A30	25 ¹ / ₄	17 ⁷ / ₁₆	20 ¹¹ / ₁₆
PHP10A36	25 ¹ / ₄	17 ⁷ / ₁₆	20 ¹¹ / ₁₆
PHP10A42	29 ¹ / ₄	21 ⁷ / ₁₆	24 ¹¹ / ₁₆

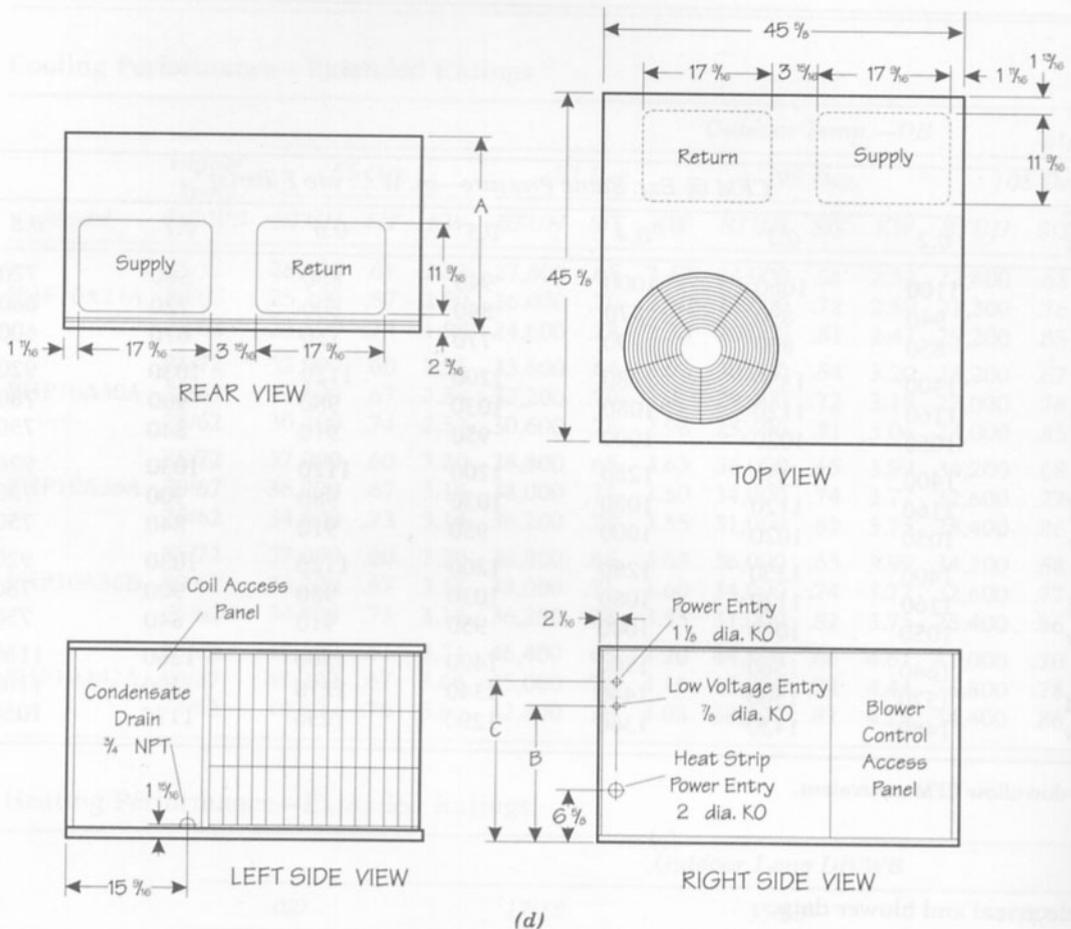
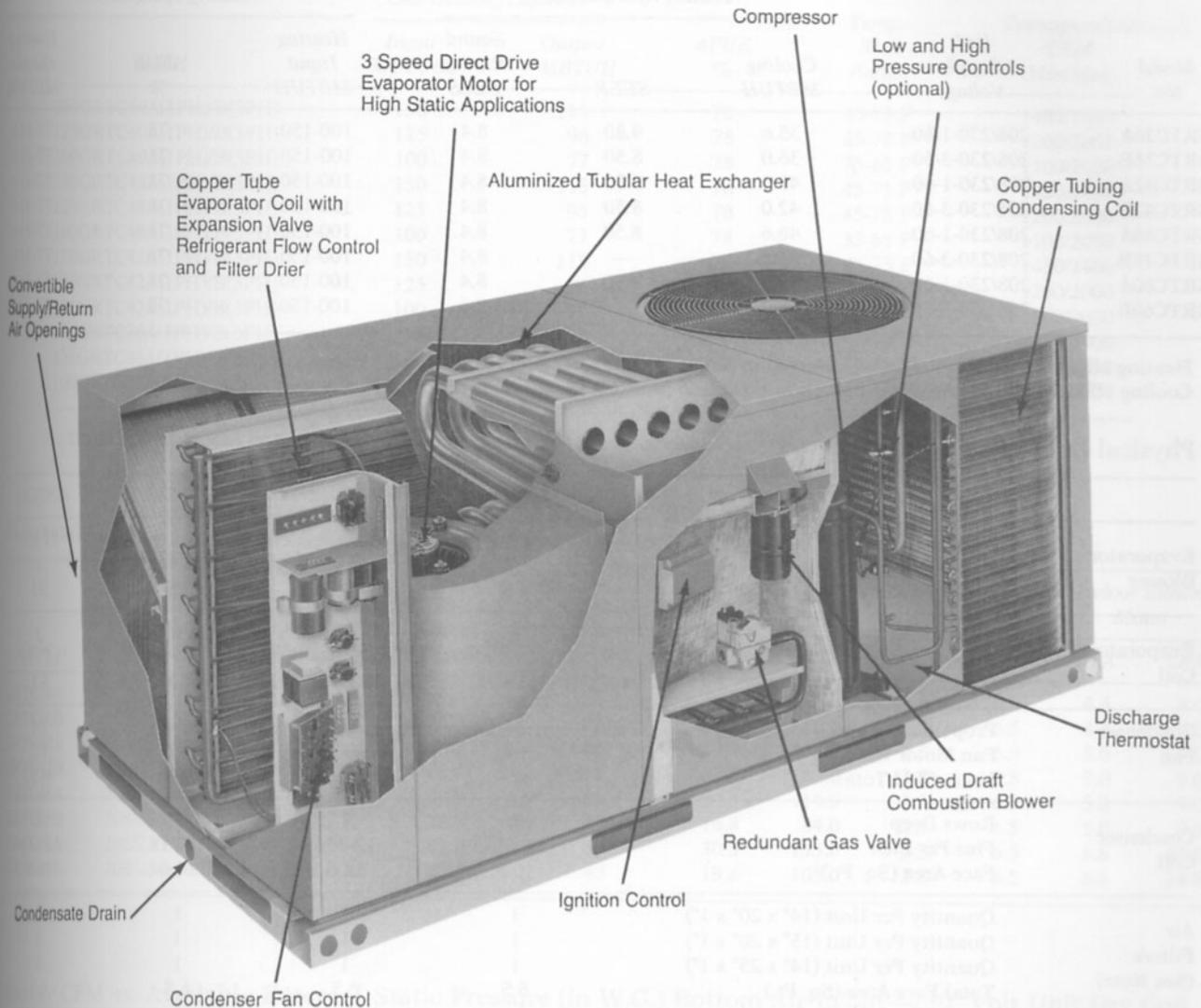


Figure 6.16 (d) Dimensional data.



(a)

Figure 6.17 (a) Package electric cooling, gas heating unit for rooftop mounting. The air conditioner section is available in 3–5 ton rating in this configuration. (Courtesy of Armstrong Air Conditioning, a Lennox International company.)

Ratings and Specifications

Model No.	Unit Supply Voltage	Cooling Specifications			Heating Specifications		Heating Output MBTUH
		Cooling MBTUH	SEER	Sound Rating BELS	Heating Input MBTUH	AFUE %	
GRTC36A	208/230-1-60	35.6	9.80	8.4	100-150	78	77-118
GRTC36B	208/230-3-60	36.0	8.50	8.4	100-150	78	77-118
GRTC42A	208/230-1-60	40.0	9.80	8.4	100-150	78	77-118
GRTC42B	208/230-3-60	42.0	8.50	8.4	100-150	78	77-118
GRTC48A	208/230-1-60	46.6	8.50	8.4	100-150	78	77-118
GRTC48B	208/230-3-60	46.6	—	8.4	100-150	78	77-118
GRTC60A	208/230-1-60	58.0	8.50	8.4	100-150	78	77-118
GRTC60B	208/230-3-60	58.0	—	8.4	100-150	78	77-118

Heating efficiency for all models is expressed in AFUE. (Annualized Fuel Utilization Efficiency).
Cooling efficiency for single phase models is expressed in SEER (Seasonal Energy Efficiency Ratio).

Physical Data—Basic Units

		GRTC36	GRTC42	GRTC48	GRTC60
Evaporator Blower	Centrifugal Blower (Dia. x Wd. in.)	11 x 9	11 x 9	11 x 9	11 x 9
	Fan Motor HP	1/2	3/4	3/4	1
	Max. Ext. SP @ Nom. CFM	.85	.85	.85	.85
Evaporator Coil	Rows Deep	3	3	3	3
	Fins Per Inch	11 - 14	11 - 14	11 - 14	11 - 14
	Face Area (Sq. Ft.)	3.6	3.6	4.3	5.1
Condenser Fan	Propeller Dia. (in.)	24	24	24	24
	Fan Motor HP	1/2	1/2	1/2	1/2
	Nom. CFM Total	4050	4150	4650	4650
Condenser Coil	Rows Deep	1	1	1	1
	Fins Per Inch	12 - 16	12 - 16	12 - 16	12 - 16
	Face Area (Sq. Ft.)	10.6	14.0	14.0	16.9
Air Filters (See Note)	Quantity Per Unit (14" x 20" x 1")	1	1	1	1
	Quantity Per Unit (15" x 20" x 1")	1	1	1	1
	Quantity Per Unit (14" x 25" x 1")	1	1	1	1
	Total Face Area (Sq. Ft.)	6.5	6.5	6.5	6.5
Charge	Refrigerant 22 (oz.)	87	83	90	105

Note: Filter racks will accept 1" or 2" thick filters.

Cooling Capacity—MBTUH

Model No.	Total Capacity	Sensible Capacity	Nominal CFM	KW
GRTC36A	35.6	25.2	1300	3.95
GRTC36B	36.0	25.6	1300	4.46
GRTC42A	40.0	27.4	1400	4.37
GRTC42B	42.0	29.0	1400	5.22
GRTC48A	46.6	31.6	1600	5.81
GRTC48B	46.6	31.6	1600	—
GRTC60A	58.0	40.2	2000	6.71
GRTC60B	58.0	40.2	2000	—

Note: Blower motor heat has been deducted from all of the above capacity ratings. The KW ratings include the KW of both the supply air blower motor and the condenser fan motor.

(b)

Figure 6.17 (b) HVAC and electrical data for 3-, 4- and 5-ton units (35.6–58 MBH).

Heating Data

Model	Gas Heating Capacity			Temp. Rise Range	Corresponding CFM Min/Max
	Input MBTUH	Output MBTUH	AFUE %		
150GRTC60A(1PH)/B(3PH)	150	115	78	45-75 F	1450/2400
125GRTC60A(1PH)/B(3PH)	125	96	78	45-75 F	1200/2000
100GRTC60A(1PH)/B(3PH)	100	77	78	35-65 F	1100/2050
150GRTC48A(1PH)/B(3PH)	150	115	78	45-75 F	1450/2400
125GRTC48A(1PH)/B(3PH)	125	96	78	45-75 F	1200/2000
100GRTC48A(1PH)/B(3PH)	100	77	78	35-65 F	1100/2050
150GRTC42A(1PH)/B(3PH)	150	115	78	45-75 F	1450/2400
125GRTC42A(1PH)/B(3PH)	125	96	78	45-75 F	1200/2000
100GRTC42A(1PH)/B(3PH)	100	77	78	35-65 F	1100/2050
150GRTC36A(1PH)/B(3PH)	150	115	78	45-75 F	1450/2400
125GRTC36A(1PH)/B(3PH)	125	96	78	35-65 F	1400/2550
100GRTC36A(1PH)/B(3PH)	100	77	78	35-65 F	1100/2050

The gas-furnaces can be converted to propane at ratings shown above.

Electrical Data

Model	Unit Supply Voltage	Min. Operating Voltage	Min. Circuit Ampacity	Max. Overcurrent Device	Compressor		Condenser Fan Motor		Indoor Blower Motor	
					RLA	LRA	RLA	LRA	RLA	LRA
GRTC36A	208/230-1-60	197	29.1	35	17.9	90.5	2.3	6.5	4.4	9.0
GRTC36B	208/230-3-60	187	22.3	30	12.5	66.0	2.3	6.5	4.4	9.0
GRTC42A	208/230-1-60	197	32.2	40	19.9	107.0	2.3	6.5	5.0	9.0
GRTC42B	208/230-3-60	187	26.4	35	15.3	82.0	2.3	6.5	5.0	9.0
GRTC48A	208/230-1-60	197	37.9	50	24.5	114.0	2.3	6.5	5.0	9.0
GRTC48B	208/230-3-60	187	28.3	35	16.8	84.0	2.3	6.5	5.0	9.0
GRTC60A	208/230-1-60	197	47.0	60	30.5	135.0	2.3	6.5	6.6	14.8
GRTC60B	208/230-3-60	187	33.4	40	19.6	105.0	2.3	6.5	6.6	14.8

Blower CFM vs. Available External Static Pressure (in W.G.) Bottom Air in/out—230-Volt Unit Dry Coil with Air Filter Only

Model	Motor Speed	External Static Pressure - IWG								
		.20	.30	.40	.50	.60	.70	.80	.90	1.0
GRTC36A	Hi	1680	1630	1560	1480	1410	1320	1240	1110	1000
	Med	1440	1400	1360	1300	1220	1160	1060	960	820
	Low	1320	1280	1240	1200	1140	1060	980	860	710
GRTC42A	Hi	1860	1780	1710	1630	1540	1420	1340	1220	1080
	Med	1780	1740	1670	1600	1510	1400	1320	1200	1060
	Low	1700	1640	1580	1500	1410	1340	1300	1120	1020
GRTC48A	Hi	1940	1890	1820	1740	1680	1560	1460	1360	1240
	Med	1840	1760	1700	1640	1560	1460	1360	1240	1100
	Low	1610	1580	1520	1460	1360	1280	1160	1040	860
GRTC60A	Hi	2460	2380	2300	2230	2160	2070	1960	1840	1680
	Med	2440	2340	2280	2200	2140	2050	1950	1800	1640
	Low	2340	2260	2180	2100	2000	1890	1760	1640	1500

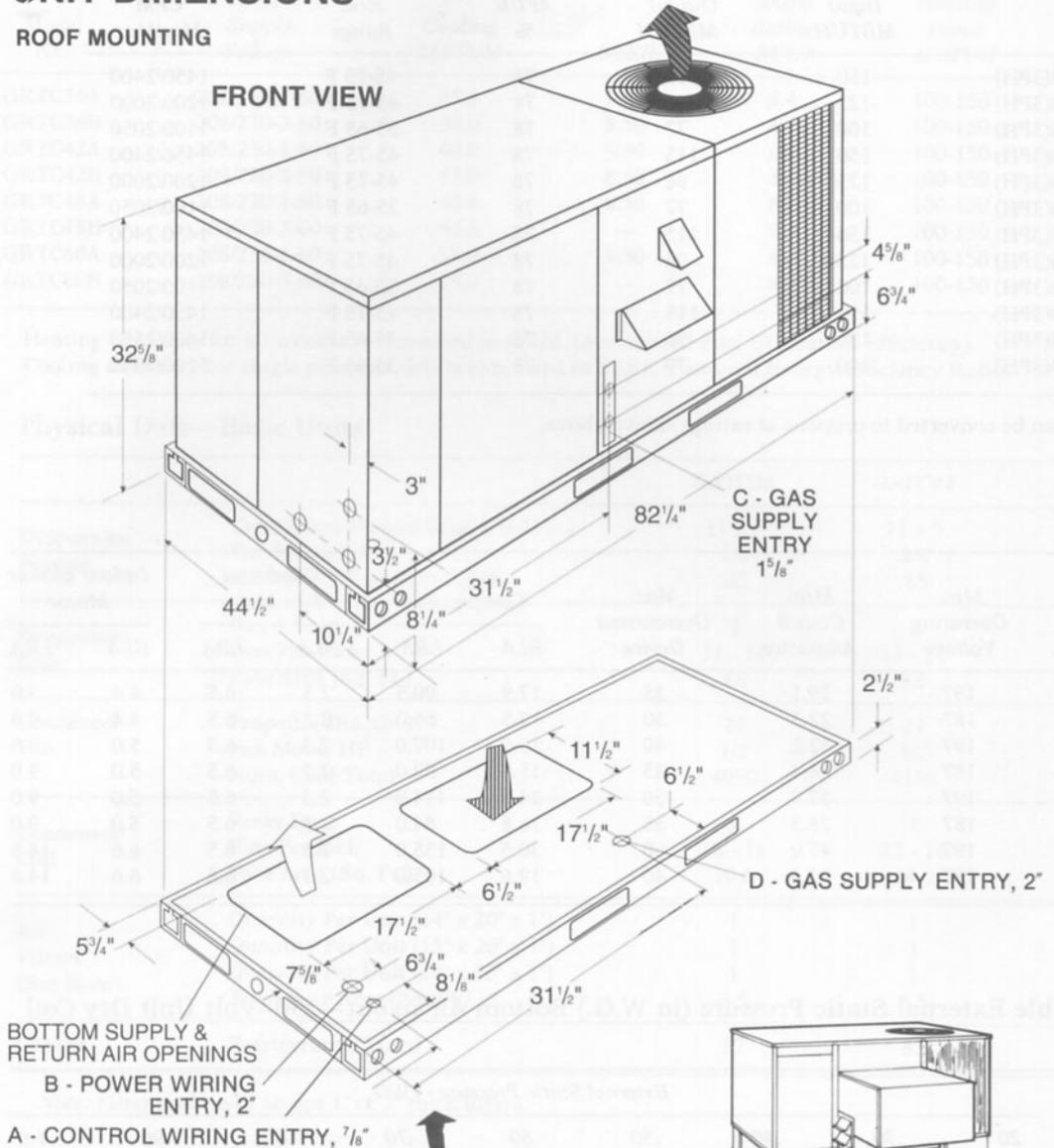
(b)

Figure 6.17 (b) (Continued)

UNIT DIMENSIONS

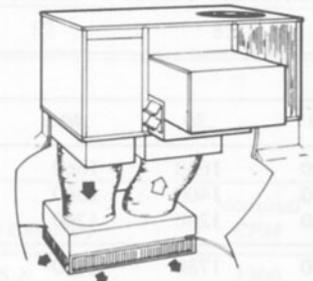
ROOF MOUNTING

FRONT VIEW



UNIT BASE WITH RAILS

Shown separately to illustrate bottom duct openings, power and gas piping connection locations.

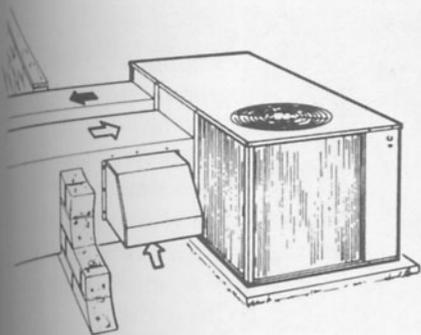
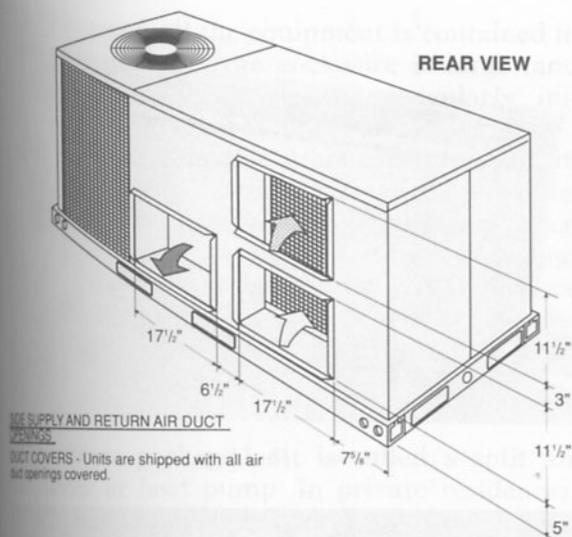


Downflow Supply and Return Air Installation with Roof Mounting Frame, Economizer Dampers and Ceiling Diffuser.

TYPICAL INSTALLATION

(c)

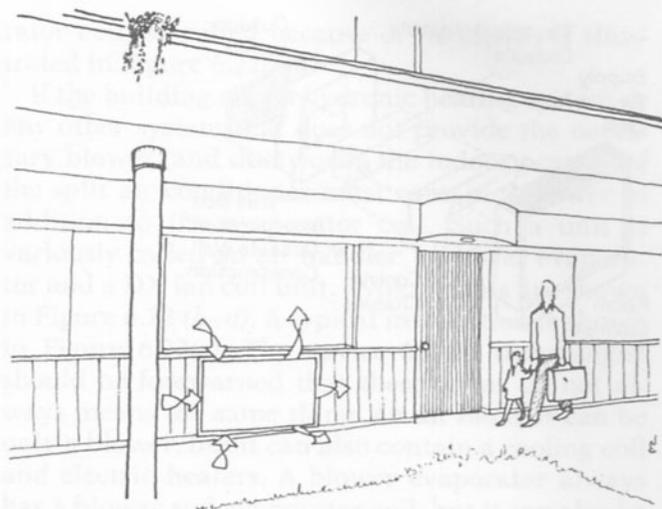
Figure 6.17 (c) Dimensional data for roof mounting.



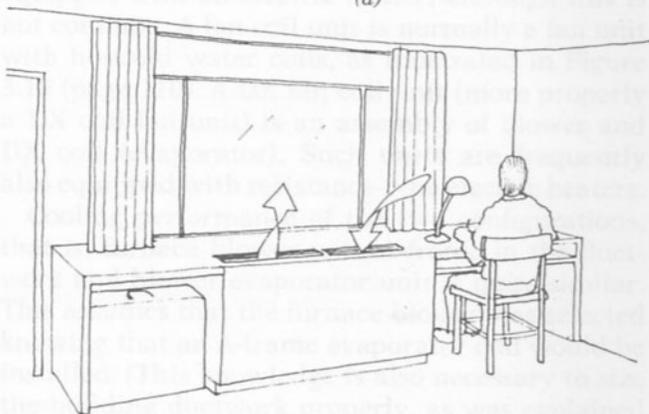
Horizontal (side) Supply and Return Air Installation with Outdoor Air Dampers.

(d)

Figure 6.17 (d) Dimensional data for duct side entry.



(a)



(b)

Figure 6.18 Motel rooms are a common application of through-the-wall heat pumps. The exterior coil is exposed to outside ambient air behind a panel set slightly forward of the outside wall (a). The interior cabinets recirculate room air through a top grille (b). All such units have a dampered opening that permits fresh air to be admitted and stale room air to be exhausted.

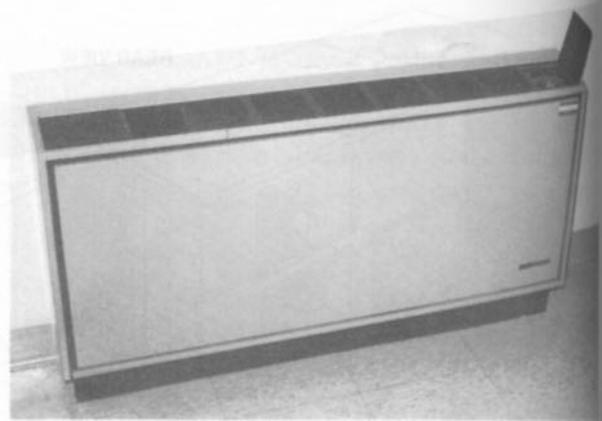
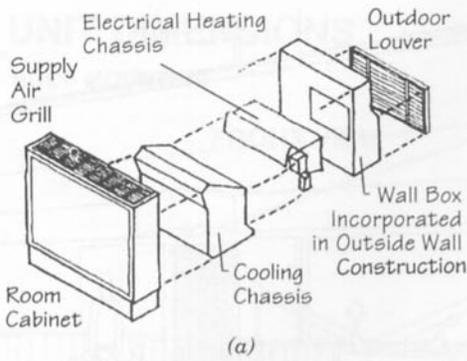


Figure 6.19 (a) Through-the-wall nonducted package air conditioner with separate heating chassis. (b-1) Inside view of a typical unit. (b-2) Outside view of the same unit. [(a) From Bobenhausen, *Simplified Design of HVAC Systems*, 1994, © and (b-1, b-2) from Bradshaw, *Building Control Systems*, 2nd ed., 1993, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.]

- **Size.** Because all the equipment is contained in a single package, the enclosure is large (and heavy). The size factor is particularly important where space is at a premium.
- **Maintenance.** Manufacturers are interested in making package units as small as possible. These small, densely packed units are often installed in small, out-of-the-way niches and closets. Both of these facts make maintenance difficult and often impossible without removing and dismantling the unit. This leads to time delay and high costs.

A two-piece package unit is called a split air conditioner or heat pump. In private residences, this arrangement for air conditioning has become almost standard. See Figure 5.6(a) (page 205) and Figure 5.12(b) (page 214). The noisy compressor and the outside coil (condenser) can be placed up to 100 ft from the house. An A-frame evaporator placed in a warm air furnace bonnet is connected to the outside (condenser) unit by two small-diameter pipes. This arrangement reduces noise, improves air flow on the condenser coil and saves space inside the house. A similar arrangement with the condenser unit on the roof is very common in low rise multifamily dwellings.

A schematic drawing showing the two alternative arrangements—a complete roof package and a split unit with roof condenser and indoor air handler—are shown in Figure 6.20(a) and (b). The same arrangement with a slab-on-grade condenser is shown in Figure 6.20(c).

The physical arrangement of a split system depends on whether the equipment is an air conditioner providing cooling only or a heat pump providing both cooling and heating.

Split Air Conditioner. The outdoor unit, called the outdoor condenser unit [see Figure 6.21(a)], always consists of the compressor and the condenser coil and its fan. It can be installed on a slab at grade level [Figure 6.21(c)] or installed on the roof [Figure 6.21(d)]. The indoor (evaporator) portion of the system has two configurations, depending on whether the building heating system is an air system or not. With an air system, we can use the heating system blower and ducts by simply placing an A-frame evaporator coil in the warm air furnace bonnet. This is the arrangement most frequently used in residences. It is illustrated in Figures 5.6(a) (page 205) and 5.7 (page 206). The A-frame evapo-

erator coil, so called because of its shape, is illustrated in Figure 6.22(a).

If the building uses a hydronic heating system or any other system that does not provide the necessary blower (and ductwork), the indoor portion of the split air conditioner must contain a blower in addition to the evaporator coil. Such a unit is variously called an air handler, a blower evaporator and a DX fan coil unit. Typical units are shown in Figure 6.22 (b–d). A typical installation is shown in Figure 6.22(e). The novice HVAC technologist should be forewarned that these terms do not always mean the same thing. An air handler can be only a blower, but it can also contain a cooling coil and electric heaters. A blower evaporator always has a blower and evaporator coil, but it can also be equipped with an electric heater, although this is not common. A fan coil unit is normally a fan unit with hot/cold water coils, as illustrated in Figure 3.19 (page 110). A DX fan coil unit (more properly a DX coil fan unit) is an assembly of blower and DX coil (evaporator). Such units are frequently also equipped with resistance-type electric heaters.

Cooling performance of the two configurations, that is, furnace blower with A-frame in the ductwork and blower evaporator unit is quite similar. This assumes that the furnace blower was selected knowing that an A-frame evaporator coil would be installed. (This knowledge is also necessary to size the building ductwork properly, as was explained in detail in Chapter 5.) Performance data for both arrangements are given in Figure 6.21(b).

Split Heat Pump. In this arrangement the exterior unit is called an outdoor heat pump. It consists of the system compressor plus a coil and fan. The coil acts as a condenser in the cooling mode and as an evaporator during the heating mode. Units are very similar in construction to the outdoor condensing unit seen in Figure 6.21. Since there is obviously no warm air heating system in the building, the heat pump indoor unit, called the heat pump air handler, always consists of a coil and blower, such as those shown in Figure 6.22. The coil acts as an evaporator during cooling and as a condenser during heating.

Split air conditioners and heat pumps up to about 15 tons of cooling, and somewhat higher heating ratings are most often used in residential and small commercial applications, including stores. Because of noise and space considerations, better than 90% of residential systems are of the split design.

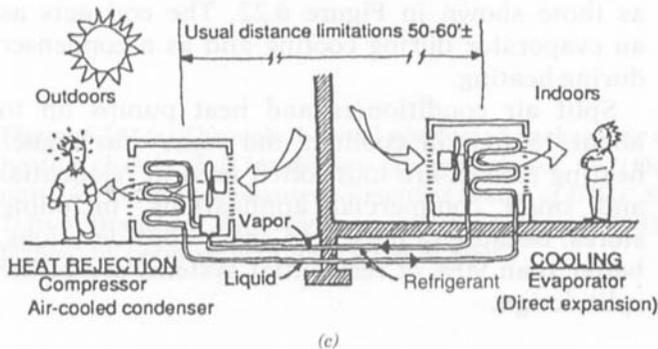
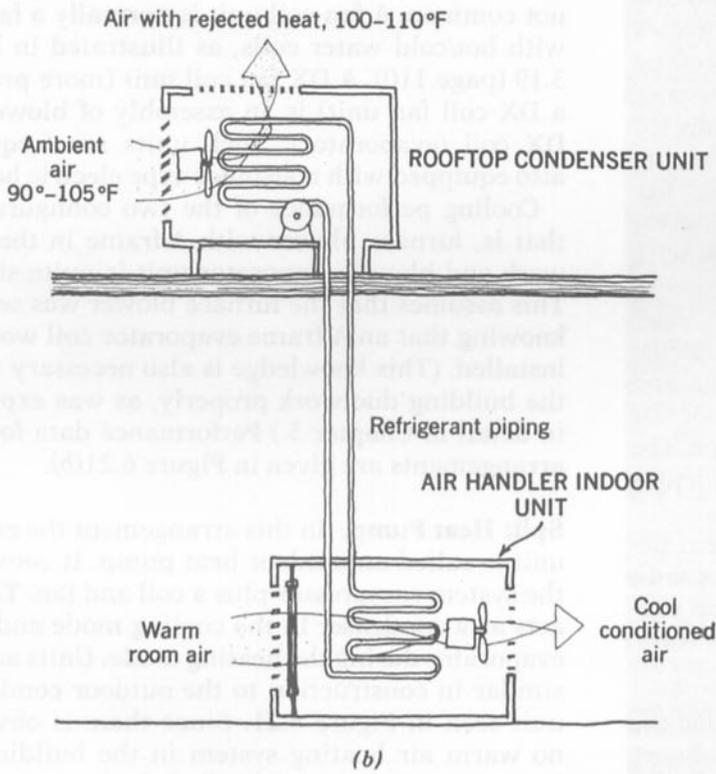
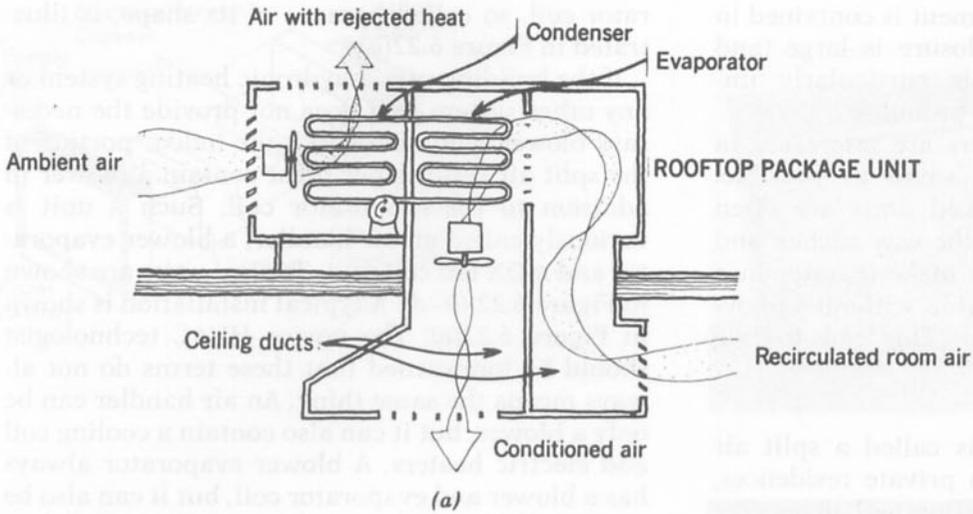


Figure 6.20 An air conditioner or a heat pump can be mounted as a single package on the roof (a), or it can be split into two units. The split unit may be a rooftop condenser and indoor evaporator/air handler (b) or a slab-on-grade outdoor condenser and an indoor evaporator and air handler (c). See the text for a discussion of the advantages and disadvantages of the three configurations.

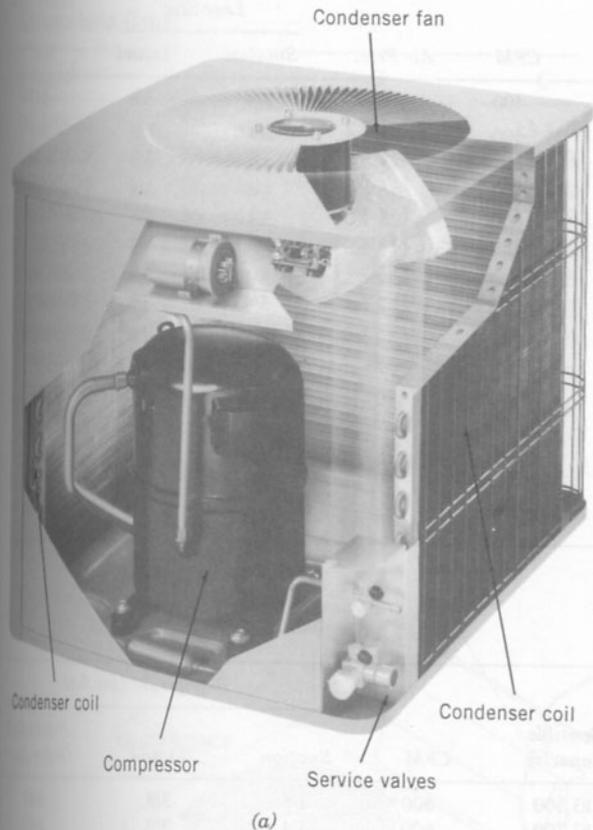


Figure 6.21 (a) Outdoor condenser unit. Models of this design have a total cooling capacity of 18,000 Btuh (1½ tons) to 57,000 Btuh (5 tons). Performance data with evaporator coils or blower evaporators is shown in (b), along with electrical data. Dimensional data, plus a schematic of a typical physical arrangement for a slab-on-grade installation, are shown in (c). A typical installation of a roof-mounted condensing unit and ceiling-mounted indoor evaporator unit is shown in (d). Also shown are the piping, wiring and some of the control items. [(a)–(c) Courtesy of Armstrong Air Conditioning, a Lennox International company. (d) Courtesy of Carrier Corp., a subsidiary of United Technologies.]

Cooling Performance with Evaporator Coils

Basic Cond. Unit Model	Indoor Model	SEER	ARI Capacity	Sensible Capacity	CFM	Air Fric.	Refrigerant Line Size		Indoor Coil Orifice Size*
							Suction	Liquid	
SCU10A12A-1	CAU,CAC18	11.0	13,000	9,100	400	.10	3/4	3/8	.041
SCU10A18A-1	CAU,CAC24	10.1	17,800	12,700	600	.22	3/4	3/8	.049
	CSH24	10.0	17,800	13,200	600	.15	3/4	3/8	.053
SCU10A24A-1	CAU,CAC24	10.0	23,600	17,300	800	.30	3/4	3/8	.057
	CAU,CAC30	10.1	23,600	17,300	800	.26	3/4	3/8	.059
SCU10A30A-1	CSH24	10.0	23,600	17,300	800	.18	3/4	3/8	.057
	CAU,CAC30	10.1	30,000	22,500	1,000	.27	3/4	3/8	.063
SCU10A36A-1	CSH36	10.0	29,600	22,500	1,000	.22	3/4	3/8	.063
	CAU,CAC36	10.1	35,400	24,800	1,200	.30	7/8	3/8	.065
SCU10A42A-1	CAU,CAC42	10.1	35,400	24,800	1,200	.28	7/8	3/8	.065
	CAU,CAC36	10.1	39,500	28,900	1,300	.30	7/8	3/8	.074
SCU10A48A-1	CAU,CAC42	10.1	39,500	28,900	1,300	.29	7/8	3/8	.074
	CAU49	10.1	47,500	35,200	1,600	.28	7/8	3/8	.084
SCU10A60A-1	MC48	10.1	47,500	35,200	1,600	.28	7/8	3/8	.084
	CSH60	10.0	57,000	42,100	1,800	.24	7/8	3/8	.090

* Required to achieve ARI rating.

Cooling Performance with Blower Evaporators

Basic Cond. Unit Model	Indoor Model	SEER	ARI Capacity	Sensible Capacity	CFM	Refrigerant Line Size		Indoor Coil Orifice Size
						Suction	Liquid	
SCU10A18A-1	AH24	10.10	18,000	13,500	600	3/4	3/8	.049
	MB08/MC24	10.10	17,800	12,700	600	3/4	3/8	.049
SCU10A24A-1	AH24	10.10	23,800	17,600	800	3/4	3/8	.059
	MB08/MC24	10.00	23,600	17,300	800	3/4	3/8	.057
SCU10A30A-1	AH36	10.10	30,000	22,500	1,000	3/4	3/8	.063
	MB12/MC29	10.10	30,000	22,500	1,000	3/4	3/8	.063
SCU10A36A-1	AH36	10.10	35,200	25,700	1,200	7/8	3/8	.068
	MB12/MC36	10.10	35,400	24,800	1,200	7/8	3/8	.065
	AH36	10.00	40,500	29,500	1,300	7/8	3/8	.074
SCU10A42A-1	MB12/MC36	10.00	39,500	28,900	1,300	7/8	3/8	.074
	MB14/MC42	10.05	40,500	30,000	1,400	7/8	3/8	.074
SCU10A48A-1	MB16/MC48	10.10	47,500	35,200	1,600	7/8	3/8	.084
	MB16/MC42	10.00	47,000	34,300	1,600	7/8	3/8	.084
SCU10A60A-1	MB20/MC60	10.00	57,000	42,200	1,800	7/8	3/8	.090

Physical and Electrical

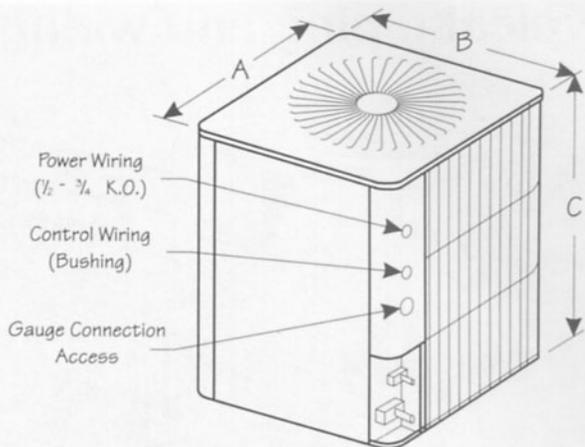
Model	Voltage Hz/Phase	Nominal Voltage Range	Min. Circuit Ampacity	Max. Overcurrent Device (amps)	Compressor (amps)		Fan Dia. (in.)	Fan Motor			Refrig. Charge (oz.)
					Rated Load	Locked Rotor		Rated HP	Nominal RPM	Full Load (amps)	
SCU10A18A-1	208/230-60-1	197-253	11	15	8	45	18	1/5	1075	0.75	72
SCU10A24A-1	208/230-60-1	197-253	16	25	12	60	18	1/5	1075	0.75	72
SCU10A30A-1	208/230-60-1	197-253	17	25	12.5	76.1	18	1/8	1075	0.9	79
SCU10A36A-1	208/230-60-1	197-253	21	30	15	78.8	18	1/3	1075	1.6	89
SCU10A42A-1	208/230-60-1	197-253	24	35	17.5	105	18	1/3	1075	1.6	102
SCU10A48A-1	208/230-60-1	197-253	30	45	22.5	119	24	1/3	1075	1.6	130
SCU10A60A-1	208/230-60-1	197-253	35	50	26.5	141	24	1/3	1075	1.6	133
SCU10A12A-1	208/230-60-1	197-253	7	15	5	26.3	18	1/5	1075	0.75	76

(b)

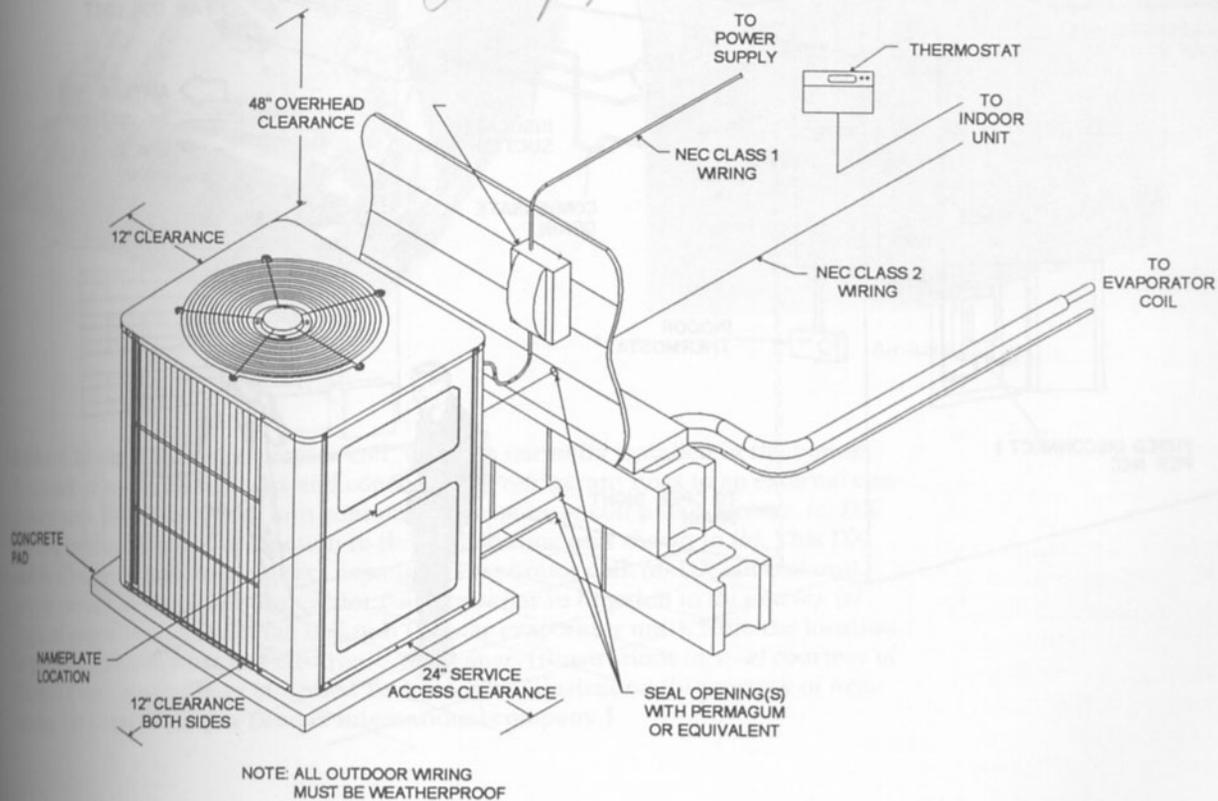
Figure 6.21 (b) Performance data.

Dimensions (in.)

Model	A	B	C
SCU10A12	28 1/4	22 1/4	23 1/8
SCU10A18	28 1/4	22 1/4	23 1/8
SCU10A24	28 1/4	22 1/4	23 1/8
SCU10A30	28 1/4	22 1/4	23 1/8
SCU10A36	28 1/4	22 1/4	25 1/8
SCU10A42	30 1/4	26 1/4	25 1/8
SCU10A48	38 1/4	34 1/4	29 1/8
SCU10A60	38 1/4	34 1/4	29 1/8



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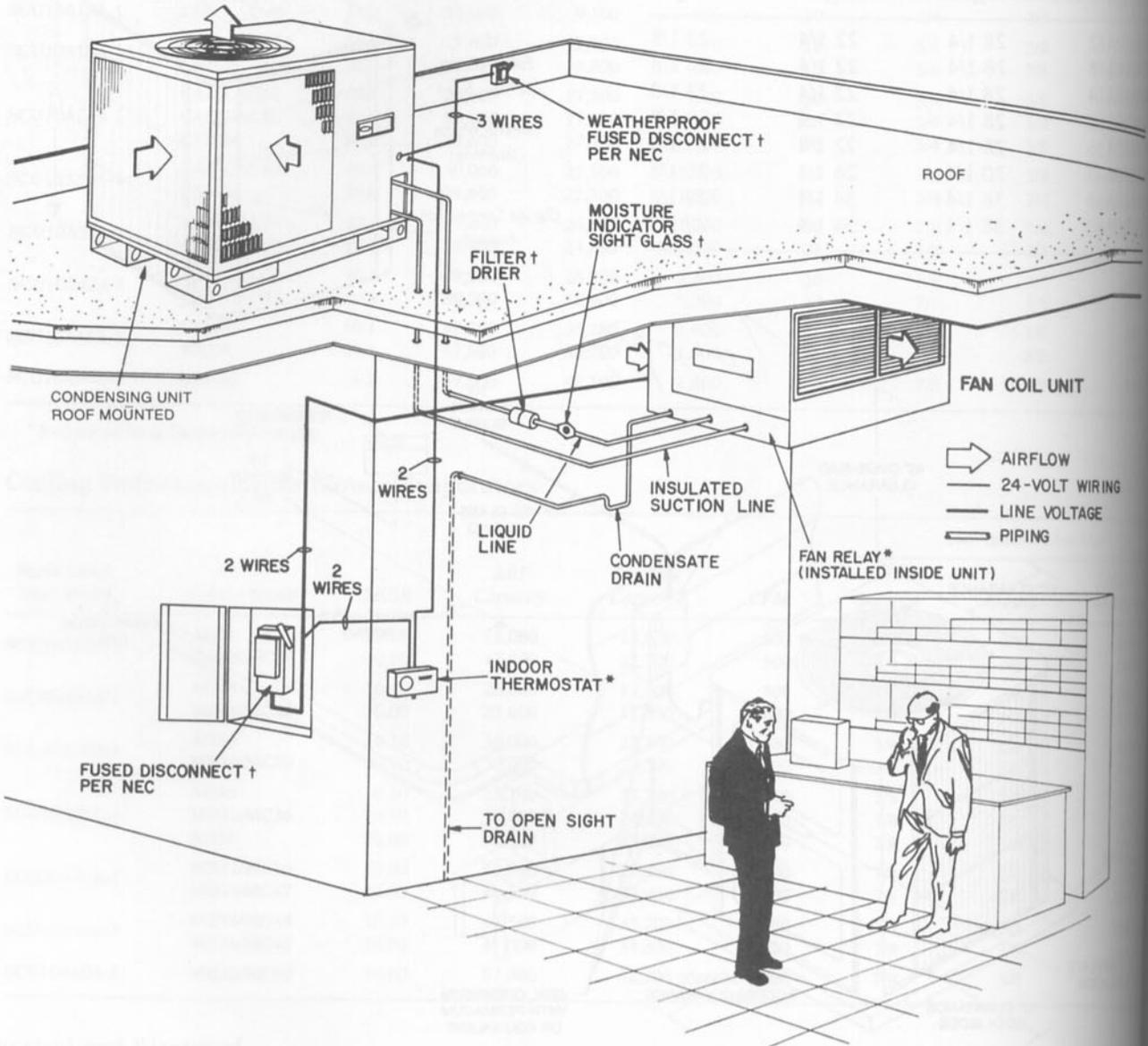


(c)

Figure 6.21 (c) Installation data.

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Typical piping and wiring

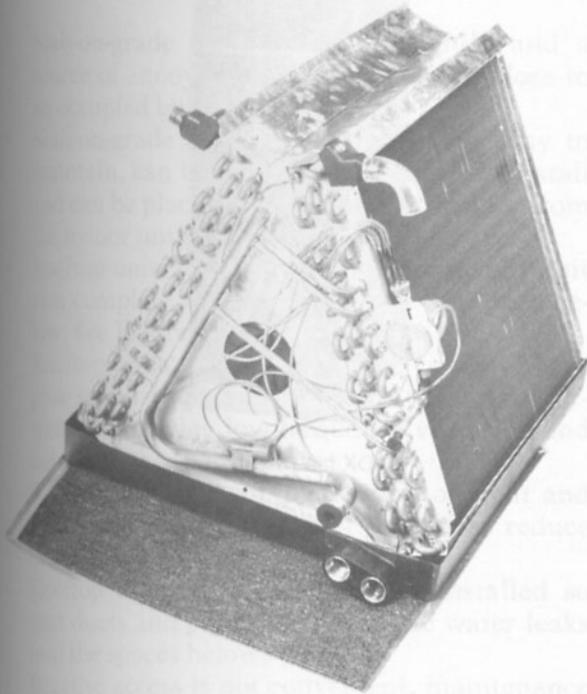


*Accessory item. †Field supplied.

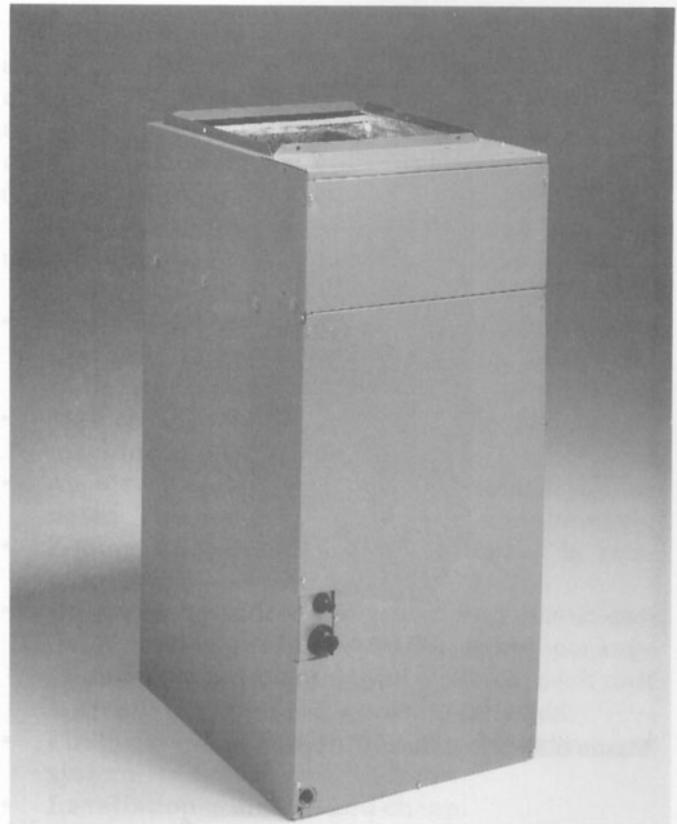
NOTES:

1. All piping must follow standard refrigerant piping techniques. Refer to Carrier System Design Manual for details.
2. All wiring must comply with the applicable local and national codes.
3. Wiring and piping shown are general points-of-connection guides only and are not intended for or to include all details for a specific installation.

Figure 6.21 (d) Typical roof installation.



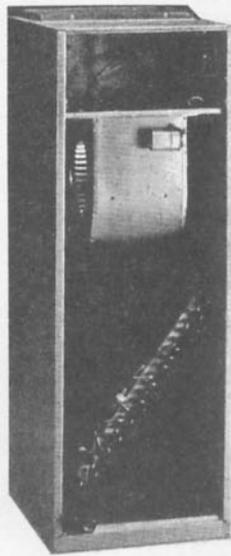
(a)



Air-handling unit

(b)

Figure 6.22 (a) A-frame evaporator coil, which is normally installed in the bonnet ductwork of warm air furnaces and connected by refrigerant lines to an external condenser unit. (b) Air-handling unit contains an evaporator coil plus a blower. (c) DX fan coil unit is identical in function to the air-handling unit shown in (b). This DX coil is a longitudinal design direct expansion evaporator coil. (d) DX fan coil unit, which uses an A-frame DX evaporator coil, is similar in function to (b) and (c). (e) Typical application of a DX fan coil unit (blower evaporator unit). Note the location of the system humidifier and electronic air cleaner. [Illustrations (a, c–e) courtesy of Carrier Corp., a subsidiary of United Technologies. Illustration (b) courtesy of Armstrong Air Conditioning, a Lennox International company.]

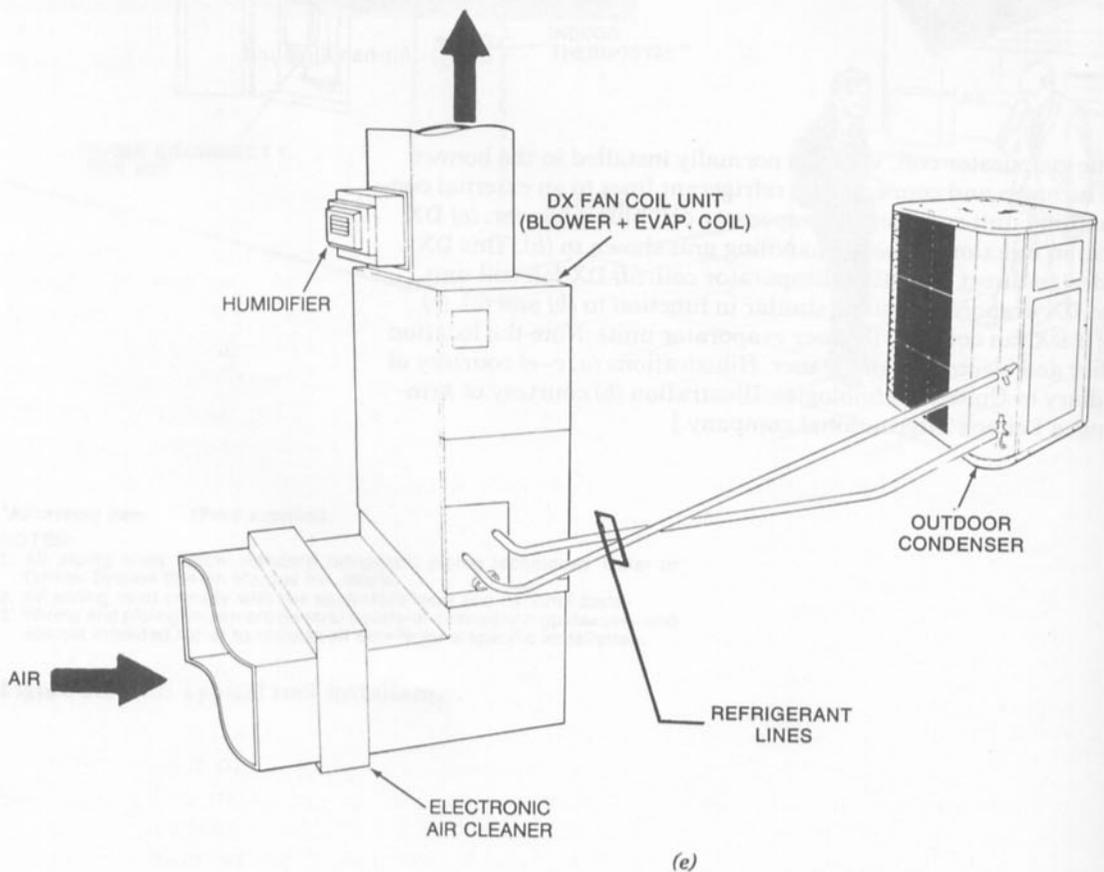


DX fan coil unit
(c)



DX fan coil unit
(d)

Figure 6.22 (Continued)



(e)

Figure 6.22 (Continued)

When selecting a split unit, the choice of whether to use a slab-on-grade outdoor unit or a rooftop unit is usually decided on architectural grounds. Some of the other considerations are:

- Slab-on-grade units can be unsightly and a source of annoying noise when placed close to an occupied building.
- Slab-on-grade outdoor units are very easy to maintain, can be screened, are cheap to install and can be placed a considerable distance from the indoor unit, if required.
- Rooftop units can be either part of a split unit or a complete package ducted to the space below. See Figure 6.20.
- Rooftop units require a massive concrete base plus sound traps and isolation to prevent extremely annoying low frequency vibration and noise into the space below.
- Rooftop units are completely out of sight and have short piping and duct runs that reduce installation costs.
- Rooftop units must be carefully installed so that ducts and piping do not cause water leaks into the spaces below.
- Because access is not convenient, maintenance of rooftop equipment is generally poor and is more expensive than that for grade-level equipment.

6.9 Physical Arrangement of Equipment, Large Systems

Cooling (and heating) systems larger than about 15 tons are classed as large systems. Here also two arrangements are common: the unitary or incremental package system and the central system. In the former, which is also called a distributed system, complete package units are used to provide cooling and heating to individual space units. In other words, a large system is broken up into many small systems. Thus, in a large multistory apartment house with, say, 100 apartments, each apartment might have a closet-mounted package like that shown in Figure 6.15(a). If the building were a complex of garden apartments, then roof-mounted packages like that of Figure 6.16 could be used. In a large motel, each room can be satisfactorily handled with a unit like that shown in Figure 6.18. These installations are all called *distributed systems* because the climate control equipment is distributed throughout the building(s).

Alternatively, a *central system* furnishing hot and/or cold water (or air) to each space can be designed. With such an arrangement, the individual spaces have only terminal units such as fan coil units, induction units and the like. See Figure 3.19 (page 110) and Figures 5.42–5.45 (pages 268–270).

The advantages of using distributed package units (rooftops, PTACs or PTHPs) follow:

- Mechanical breakdown of a unit affects only that unit. Other individual package units continue to operate.
- Maintenance is simplified and affects only one machine at a time.
- An air or water distribution system is eliminated. This reduces first cost and maintenance.
- Zoning and individual unit control is simplified.
- Units can be added, subtracted and moved easily. For this reason, systems using package units are called incremental systems; each unit is an increment of the whole installation.
- First cost is lower than that of a ducted central system.
- Installation is simple and cheap.
- Sizing and selection of equipment is much simpler than for a single package. See the design example in Section 6.17.

The disadvantages of using incremental (distributed package) units follow:

- Control of temperature, humidity, fresh air intake and air distribution are relatively crude.
- An economizer cycle (use of fresh air for cooling) is generally not provided.
- Each unit requires access to an outside wall. This restricts use of these units to perimeter zones, or buildings with open spaces, if ducting is to be avoided. Incremental units like those in Figure 6.15 are ducted. Units of the type shown in Figure 6.19 are not ducted.
- Because the compressor and condenser fan are included in a package (incremental) unit, it will be much noisier than a terminal device, such as a fan coil unit, fed from a remote, central chiller. In some installations, this disadvantage can be turned into an advantage by using the compressor and fan noise to mask or blanket unwanted noise, such as from traffic or nearby industry. (This type of application is based on the well-known fact that the noise created by your equipment is much less disturbing than that coming from a neighbor's equipment.)

These are only a few of the many considerations involved. Refer to Section 6.17 for a design example using incremental units.

Refrigeration Equipment

By this point, you should have a firm grasp of the principal components in a vapor compression refrigeration cycle and their function. A rapid survey of the operation of these components plus some of the other equipment often encountered in larger systems is in order.

6.10 Condensers

As we have seen, the function of the condenser is to condense the refrigerant vapor by removing heat from it. There are three principal types of condensers:

- Air-cooled
- Water-cooled
- Evaporative-cooled

a. Air-Cooled Condensers

Air-cooled condensers reject heat to the atmosphere by blowing ambient (outside) air over the condenser coil. Obviously then, the higher the outdoor temperature is, the more air will have to be passed over the coil to reject its heat and condense the refrigerant vapor. Air-cooled condensers, such as those illustrated in Figures 6.17 and 6.21, almost always use a propeller fan because of its high air quantity, low static pressure characteristic. Fan motors are about 0.1 to 0.2 hp per ton of cooling. This type of condenser is low in cost and reliable. However, the large amount of air that must be moved to reject heat limits this condenser to systems up to about 50 tons. Larger systems normally use water-cooled condensers. (Specially designed air-cooled condensers are available for loads up to 100 tons.)

b. Water-Cooled Condensers

Water-cooled condensers reject their heat into a heat exchanger that is cooled by water. The prob-

lem then is how to arrange a continuous supply of cooling water. In the early days of air conditioning, city water was used as the cooling agent. The water was passed through the heat exchanger once and then discarded. Because of the wastefulness of this procedure and the load it places on the municipal water and sewer systems, most local authorities prohibit this practice today. If no lake, river, pond, or wells are available to supply the required cooling water, a cooling tower must be used.

A *cooling tower* is a device that uses air to cool water, which is then recirculated to the condenser heat exchanger. Although cooling towers are available in many designs, they all operate on the principle of evaporative cooling. Water is pumped to the top of a structure (tower), where it is sprayed inside the tower. As it falls, some of it evaporates. In so doing, it absorbs 1000 Btu/lb of water vapor, as we have already learned. This heat can only come from the falling water. (Actually, the evaporating water cools the air around it, which in turn draws heat from the warm falling water.) Cooled water collects at the bottom of the tower, from which point it is recirculated to the condenser heat exchanger. See Figure 6.23. Since the water that evaporates is lost to the atmosphere, make-up water must be supplied from the city water mains.

There are three principal types of cooling tower designs: natural draft, forced draft and induced draft. See Figure 6.24. In the *natural draft tower* (Figure 6.23(a)), air circulation through the tower is by natural convection (stack effect). The sprayed water simply falls inside the tower and is partially evaporated by natural convective air currents. To assist evaporation, most towers contain a "fill" that slows the flow of water, allowing more to evaporate. In *mechanical draft towers* [Figures 6.23(b) and (c)] (forced draft or induced draft), a large fan or blower greatly increases the motion of air through the tower. This serves to increase the amount of water evaporation and, thereby, increases the cooling effect. Total water temperature drop in most towers is about 10–20°F.

Cooling towers have a number of disadvantages. Because they depend on evaporative action for their cooling, their efficiency is affected by humidity in the surrounding air. The higher the humidity (wet bulb temperature), the lower is the tower's efficiency. They require a good deal of make-up water, which in many areas is expensive. They also require frequent cleaning and corrosion protection. The noise created by their fans can be a source of annoyance to neighbors, even when towers are installed on roofs. In winter, air and water flow

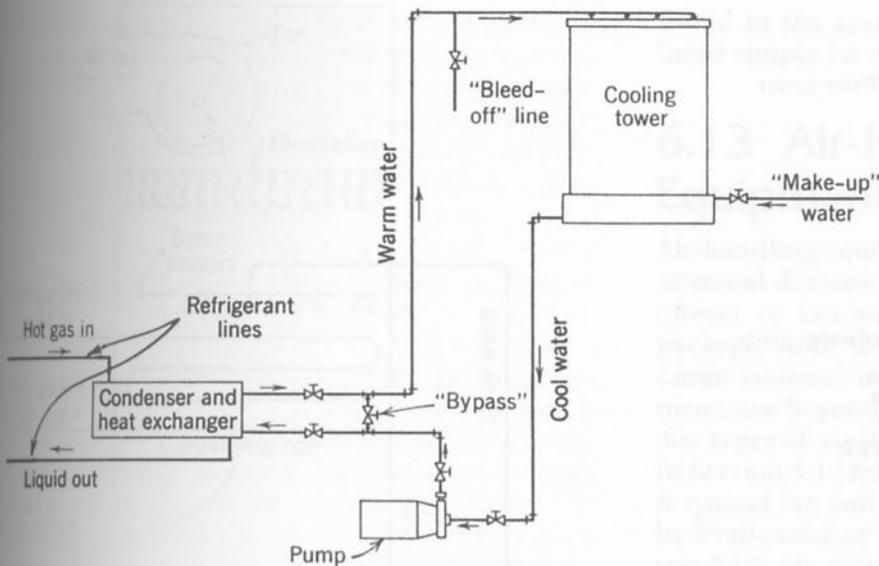


Figure 6.23 Schematic diagram of a recirculating water system between a water-cooled condenser and a cooling tower. Make-up water comes from the municipal water supply. The bypass valve permits removing the cooling tower and its accessories for maintenance. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

must be carefully regulated to prevent freezing. Occasionally, electric heaters must be installed for this purpose. (Large buildings frequently require cooling for inside zones, even in winter.) Finally, towers not in use must be drained, cleaned and repaired.

c. Evaporative Condensers

The evaporative condenser uses the cooling effect of water evaporating directly on the condenser coil. See Figure 6.25. Water from a local tank is sprayed directly on to the hot condenser coils. An induced draft fan above or to the side of the condenser coil increases the draft and, thereby, the evaporation and cooling rate. Evaporative condensers are more efficient than either air-cooled or water tower-cooled condensers. Unlike cooling towers, they must be installed close to the compressor. Depending on the hardness of spray water, scale accumulation on the condenser coils can be a problem.

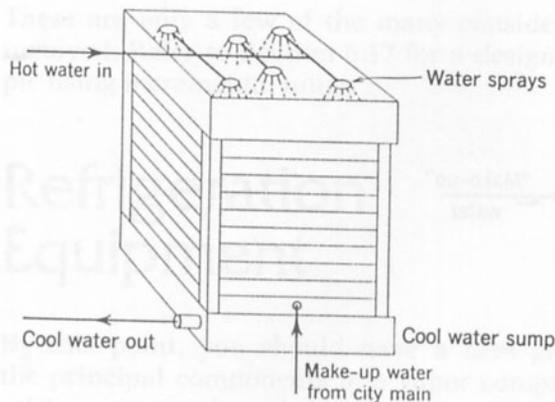
6.11 Evaporators

It is important to keep in mind that the terms *evaporator* and *condenser* refer to processing of the

refrigerant. Otherwise, terms such as *evaporative condenser* (Section 6.10.c) will lead to considerable confusion. The evaporator is the piece of equipment that performs the functional space cooling, by absorbing heat from the conditioned space. The principal type of heat-transfer-to-air evaporator in use today is simply a coil of pipe equipped with fins to aid in heat transfer. See Figure 6.26(c).

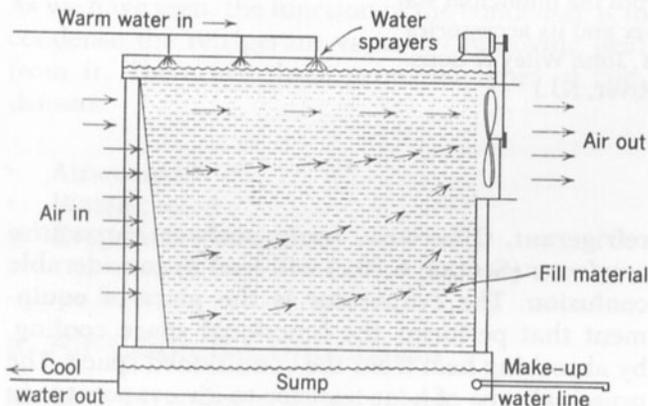
Years ago there were two types of such evaporators in common use: flooded [Figure 6.26(a)] and dry expansion [Figure 6.26(b)]. The flooded evaporator, which was always filled with liquid refrigerant, is no longer in common use. The dry expansion (DX) evaporator uses a thermostatically controlled expansion valve that meters the flow of liquid refrigerant into the evaporator coil. The flow rate is controlled such that all the liquid refrigerant is vaporized by the time it reaches the end of the coil. The flow rate varies with the variation of heat load on the evaporator. (Although DX stands for dry expansion most industry people today refer to DX as direct expansion.)

The second principal type of evaporator in common use is the shell-and-tube, water-cooled evaporator. The refrigerant passes through a series of pipes encased in a shell. Water circulating through the shell is chilled by the evaporating refrigerant.



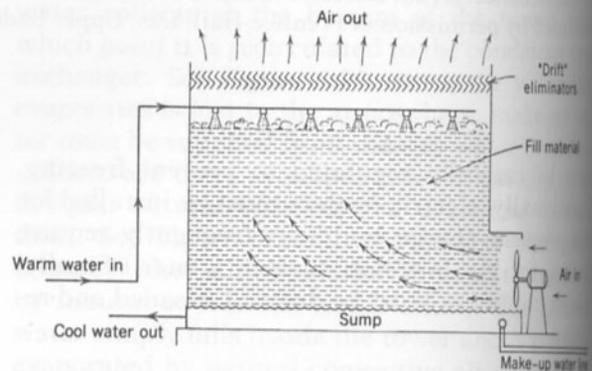
NATURAL DRAFT-COOLING TOWER

(a)



INDUCED DRAFT COOLING TOWER

(b)



FORCED DRAFT COOLING TOWER

(c)

Figure 6.24 (a) Schematic of a natural draft cooling tower. In actual construction, these towers have hyperbolic curved sides that improve the reliability and the predictability of their stack effect. (b) Induced draft cooling tower. The purpose of the fill material is to expose the falling water droplets to as large a surface area as possible, in order to increase vaporization of the warm water. (c) Forced draft cooling tower. The "drift" eliminators reduce the quantity of water that is lost as it "drifts" away. This in turn will reduce the amount of make-up water required. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

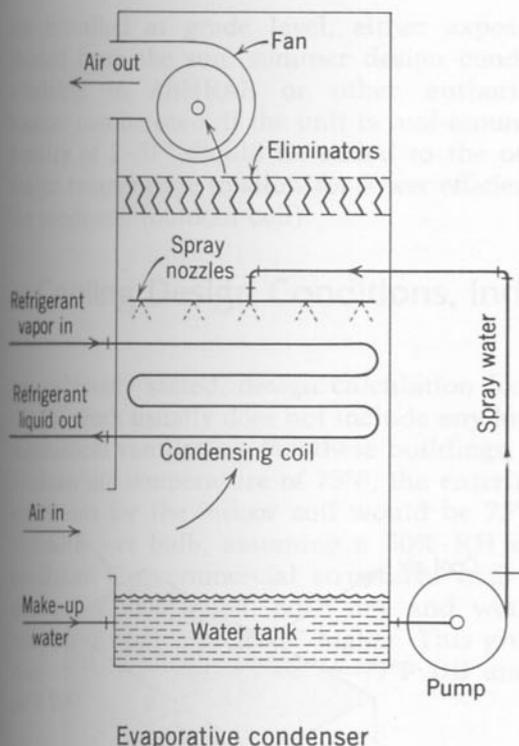


Figure 6.25 The evaporative condenser uses direct water evaporation to provide the required refrigerant cooling. Recirculated water is sprayed directly onto the hot condenser coils. The purpose of the eliminators at the top of the unit is twofold: to prevent water being carried by the air stream from entering (and damaging) the blower and to reduce the amount of water lost by drift. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

The chilled water is then used in a fan coil or similar terminal unit to cool the building interior space. In large versions, shell-and-tube evaporators are referred to as *chillers*.

6.12 Compressors

There are five types of compressors in use for various sizes and types of refrigeration equipment. They are reciprocating (piston type), rotary, helical (screw type), scroll (orbital) and centrifugal. Construction details are not the concern of the technologist or, in general, the HVAC engineer. Most air conditioning equipment is packaged by the manufacturer who selects the type of compressor best

suitable to the system requirements. The types are listed simply for information purposes.

6.13 Air-Handling Equipment

Air-handling equipment includes fans, blowers and terminal devices. As with compressors, the type of blower or fan supplied with an evaporator in a package unit is selected by the manufacturer. Large systems, using separate air-handling equipment, are beyond the scope of our study. The various types of air distribution systems were covered in Section 5.19 and are shown in Figures 5.40–5.45. A typical fan coil unit, which is the most common hydronic cooling terminal device, is shown in Figure 3.19. Air distribution equipment is covered in Chapter 5.

Design

6.14 Advanced Design Considerations

Equipment-sizing considerations are discussed in some detail in Section 6.6. The material in this section is more technical. It is intended for technologists who have acquired the necessary background knowledge and are engaged in actual design work.

Remember that, to calculate the air quantities required in heating and cooling, we used the following formulas. For heating:

$$\text{cfm} = \frac{(\text{Sensible}) \text{ heating load}}{1.08 \times \Delta t}$$

For cooling:

$$\text{cfm} = \frac{\text{Sensible cooling load}}{1.1 \times \Delta t}$$

(Rereading Sections 5.3 and 5.29 at this point would be helpful.)

The information that follows should assist you in determining the loads and the temperature differences required for these formulas.

a. Cooling Load Calculation

When using split unit air conditioners or heat pumps, the location of the air-cooled condenser or outdoor heat pump unit is important. When units

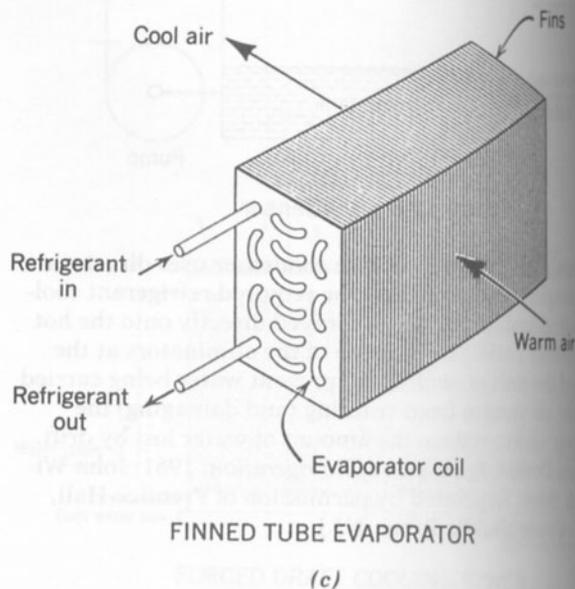
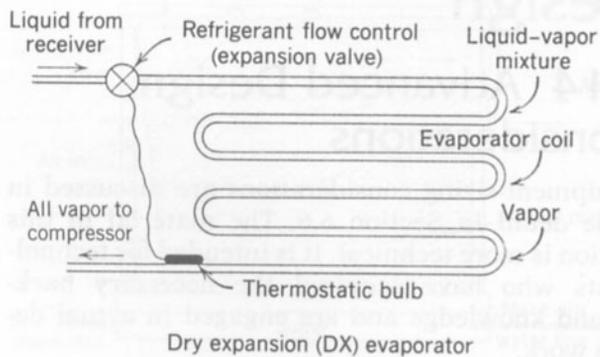
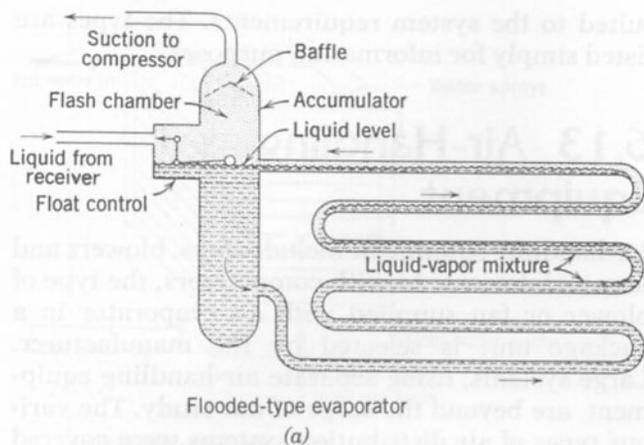


Figure 6.26 Evaporators of the flooded type (a) are no longer widely used. The dry expansion (DX) type (b) uses a thermostatically controlled expansion valve to control the flow of hot refrigerant into the evaporator coil. Fins are added to the evaporator coil (c) to aid in heat transfer. (From Dossat, *Principles of Refrigeration*, 1961, John Wiley & Sons. Reprinted by permission of Prentice-Hall, Inc., Upper Saddle River, NJ.)

are installed at grade level, either exposed or shaded from the sun, summer design conditions available in ASHRAE or other authoritative sources can be used. If the unit is roof-mounted, a penalty of 2–5F° should be added to the outside design temperature to allow for lower efficiency at the condenser (outdoor coil).

b. Cooling Design Conditions, Indoor Coil

As previously stated, design calculation for residential work usually does not include any load for mechanical ventilation. For these buildings, using a return air temperature of 75°F, the entering air conditions for the indoor coil would be 75°F DB and 62°F wet bulb, assuming a 50% RH design condition. For commercial structures with some mechanical ventilation, both dry and wet bulb temperatures are about 2F° higher. This gives indoor coil air conditions of 76–77°F DB and 63–64°F WB.

c. Heating Design Conditions, Heat Pump Indoor Coil

In the heating mode, with no ventilation considered (residential calculation), the return air temperature (entering air, indoor coil) can be taken at 70°F. If ventilation is considered, this temperature drops from 2 to 5F° depending on the outdoor design temperature.

d. Cooling Temperature Differential (Δt)

As stated in Section 6.6, the preferred procedure in cooling design is to calculate the latent as well as the sensible heat load. This permits an accurate calculation of the sensible heat ratio (SHR). A typical small structure (residence or other) might have a sensible heat load of 40,000 Btuh and a latent heat load of 8000 Btuh. This would give a total load of 48,000 Btuh and a sensible heat ratio of

$$\text{SHR} = \frac{40,000 \text{ Btuh}}{48,000 \text{ Btuh}} = 83\%$$

Having determined the SHR, we can determine the temperature difference to use in our air quantity calculations. A high SHR means a small latent heat load, less required dehumidification and, therefore, a warmer coil and a smaller Δt . Similarly, a low SHR means a high latent load, more required de-

humidification and, therefore, a colder indoor coil and a high Δt . As a guide, use the following figures:

Calculated SHR	Temperature Difference (Δt), F°
less than 0.8	19.6–21
0.8–0.85	18.1–19.5
above 0.85	17–18

6.15 Design Example—The Basic House

At this point, we have considered all the factors necessary to perform a year-round, all-air climate control design for a residence. We select for our first design example The Basic House. The architectural plan is found in Figure 3.32 (page 130). Design heat loads are also found in that illustration. Cooling loads, both sensible and latent, were calculated. We do not see any value in reproducing the calculation sheets here since the forms we used are proprietary. It is, therefore, very doubtful that you will use this particular form. Instead, the results are tabulated in Table 6.1. The total calculated latent load, assuming no mechanical ventilation, comes to 5522 Btuh.

Therefore,

$$\begin{aligned} \text{Total cooling load} &= \text{Sensible load} + \text{Latent load} \\ &= 24,487 \text{ Btuh} + 5,522 \text{ Btuh} \\ &= 30,009 \text{ Btuh} = 2\frac{1}{2} \text{ tons} \end{aligned}$$

Table 6.1 Design Data for Heating/Cooling Loads, The Basic House

Space	Heat Loss, Btuh ^{a,b}	Sensible Heat Gain, Btuh	Cooling, cfm ^b
Living room	9300	6726	330
Dining room	4300	3408	167
BR #1	5200	4108	201
BR #2	7500	5020	246
Bath	4800	Negligible	—
Kitchen	4900	5225	256
Subtotal	34,000	24,487	1199
Basement	4900	Negligible	—
Total	38,900	24,487	1,200

^aFrom Figure 3.32.

$$^b \text{cfm for heating} = \frac{38,900 \text{ Btuh}}{(1.08 \times 55\text{F}^\circ)} = 655 \text{ cfm}$$

The sensible heat ratio is

$$\text{SHR} = \frac{\text{Sensible load}}{\text{Total load}} = \frac{24,487 \text{ Btuh}}{30,009 \text{ Btuh}} = 0.816$$

From the tabulation in the previous section we can find the temperature differential required, by interpolation:

SHR	Δt
0.8	18.1 F°
0.816	?
0.85	19.5 F°

$$\begin{aligned} \Delta t &= 18.1\text{F}^\circ + \frac{0.016}{0.05} \times (19.5 - 18.1) \\ &= 181\text{F}^\circ + 0.448 \\ &= 18.55\text{F}^\circ \end{aligned}$$

Therefore, the cooling (cfm) figures in the third column of Table 6.1 are all calculated by using the expression

$$\begin{aligned} \text{cfm} &= \frac{\text{Sensible load}}{1.1 (18.55)} \\ &= \frac{\text{Sensible load in Btuh}}{20.41} \end{aligned}$$

The total cfm required for cooling is 1200 (see Table 6.1).

Calculating the air flow required for heating, and using a temperature difference of 55F° (125F° supply air and 70°F return air), we have

$$\text{cfm for heating} = \frac{38,900 \text{ Btuh}}{1.08 (55\text{F}^\circ)} = 655 \text{ cfm}$$

Duct sizes will, therefore, be calculated using the cooling air requirements as listed in Table 6.1, since the cooling air requirements are greater than the heating air requirements.

The procedure that we would use for a building of this type and size follows:

- Step 1 Select a system type based on the building architecture.
- Step 2 Locate and size supply registers and return grilles.
- Step 3 Select and locate heating unit, cooling unit and thermostats.
- Step 4 Make a duct layout including all sizing.

Following this procedure, we will develop the design, explaining each design decision.

Steps 1 and 2. In these steps, we select a system type and locate and size supply registers and return grilles.

Although a heat pump can be selected to serve this structure, we will go by a more conventional route, using a gas- or oil-fired furnace and a split air conditioner. The partial basement and crawl space are ideally suited to a basement furnace and duct system. The supply air terminals will be floor registers below exterior glass (windows) and adjacent to exterior doors. A single central return duct will be centrally located in this small house. We will equip it with high and low return grilles to receive return cooling and heating air, respectively. All doors will be undercut to permit passage of return air.

Since the return duct is large and close to the furnace blower, it will be furnished with a duct liner that will serve to reduce vibration, sound transmission and heat loss. All ducts in the unexcavated crawl space will be insulated. Although the basement has a calculated heat loss of 4900 Btuh (see Table 6.1), the furnace losses should keep it at a comfortable 65–68°F without additional heating. However, as a “safety net” a small register (8×5 in.) will be provided, tapped from one of the two main feeder ducts. Register and grille locations are shown in Figure 6.27.

We have chosen to use heavy-duty floor registers that deliver about 100–175 cfm each and to use multiple registers in each room rather than a single large register. See Figure 5.21 (c). The reason for this is the need for good circulation and coverage, particularly in cooling mode, where floor registers are problematic. See Figure 5.33 (b). In the heating mode, placement of registers below each window is ideal, since it will prevent annoying cold drafts at the floor level. See Figures 5.29 (c) and 5.33 (a). The increased cost of multiple registers is, therefore, justified.

Air velocity in all ducts is kept considerably below the noise limits given in Tables 5.6 and 5.7. The cfm figures shown in Figure 6.27 at all registers are the final flow figures after balancing. All branch ducts are equipped with dampers in the duct for this purpose. It is important to remember that great precision in calculation of air quantities is not necessary. A common rule of thumb calls for 450–500 cfm per ton total air for cooling. For the 2½ tons required here (30,000 Btuh total load), this would come to 1125–1250 cfm, which matches closely the 1200 cfm calculated (Table 6.1). Note that this is almost double the 655 cfm required for heating. This indicates the requirement for a multispeed blower motor in the furnace.

Steps 3 and 4. In these steps, we select and locate the heating and cooling equipment and design the duct system.

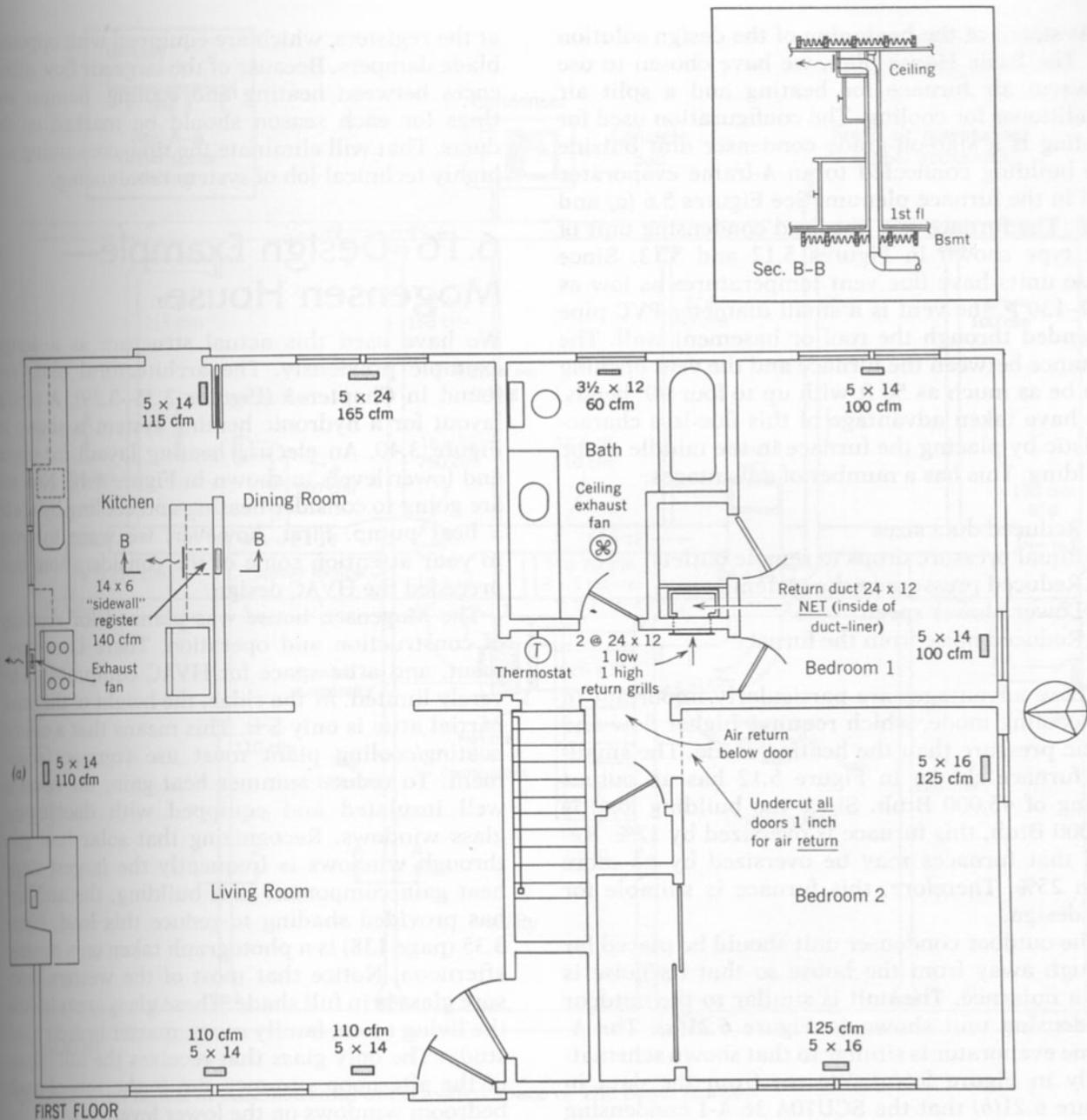


Figure 6.27 Layout of supply air registers and return air grilles for The Basic House plan.

As stated at the beginning of the design solution for The Basic House plan, we have chosen to use a warm air furnace for heating and a split air conditioner for cooling. The configuration used for cooling is a slab-on-grade condenser unit outside the building connected to an A-frame evaporator coil in the furnace plenum. See Figures 5.6 (a) and 6.28. The furnace is a gas-fired condensing unit of the type shown in Figures 5.12 and 5.13. Since these units have flue vent temperatures as low as 100–130°F, the vent is a small diameter PVC pipe extended through the roof or basement wall. The distance between the furnace and the vent opening can be as much as 50 ft with up to four 90° bends. We have taken advantage of this flue-less characteristic by placing the furnace in the middle of the building. This has a number of advantages:

- Reduced duct sizes
- Equal pressure drops to remote outlets
- Reduced pressure in the system
- Lower blower speed
- Reduced noise from the furnace

These advantages are particularly important in the cooling mode, which requires higher flow and static pressure than the heating mode. The smallest furnace shown in Figure 5.12 has an output rating of 45,000 Btuh. Since the building load is 40,000 Btuh, this furnace is oversized by 12%. Recall that furnaces may be oversized by no more than 25%. Therefore, this furnace is suitable for our design.

The outdoor condenser unit should be placed far enough away from the house so that its noise is not a nuisance. The unit is similar to the outdoor condensing unit shown in Figure 6.21(a). The A-frame evaporator is similar to that shown schematically in Figure 5.6(a). We see from the data in Figure 6.21(b) that the SCU10A 36 A-1 condensing unit with CAU indoor A-frame evaporator coil can supply the load. The duct system has been designed for low velocity, with a static head below 0.1 in. w.g. The duct sizing, pressure and velocity calculations are left as an exercise. See Problem 6.39.

One duct on the suction (return side) of the air system pulls in fresh air. A damper in a convenient place (near an access opening) can partially or entirely close this duct. It can be fully open for ventilation or fully closed for the greatest fuel economy. Balance of air flow between north and south ends of the house may be adjusted by the splitter damper where the main duct divides. Balancing the system is accomplished using dampers in all the branch ducts. Final adjustments can be made

at the registers, which are equipped with opposed blade dampers. Because of the large air flow differences between heating and cooling, damper settings for each season should be marked on the ducts. That will eliminate the time-consuming and highly technical job of system rebalancing.

6.16 Design Example—Mogensen House

We have used this actual structure as a design example previously. The architectural plans are found in Chapter 3 (Figures 3.35–3.39). A piping layout for a hydronic heating system is shown in Figure 3.40. An electric heating layout for upper and lower levels is shown in Figure 4.12. Now we are going to consider heating and cooling by use of a heat pump. First, however, we want to bring to your attention some of the considerations that preceded the HVAC design.

The Mogensen house was planned for economy of construction and operation. There is no basement, and attic space for HVAC equipment is severely limited. At the ridge, the height of the small partial attic is only 5 ft. This means that a central heating/cooling plant must use compact equipment. To reduce summer heat gain, the house is well insulated and equipped with double-pane glass windows. Recognizing that solar heat gain through windows is frequently the largest single heat gain component in a building, the architect has provided shading to reduce this load. Figure 3.35 (page 138) is a photograph taken on a summer afternoon. Notice that most of the western exposure glass is in full shade. These glass areas include the living room, family room, master bedroom and study. The only glass that receives the full impact of the afternoon summer sun is the row of small bedroom windows on the lower level.

It is sometimes a good decision to cool only part of a house. In the house we are now studying, the actual choice was to cool only the upper-level rooms. The lower level has only a very small heat gain. The east wall of that story is below grade against the cool earth. The north and south walls have no glass. The west glass (in the family room) is in shade. Finally, the windows and sliding glass doors can be opened to the cooling breezes from Long Island Sound.

Adding strength to the decision for cooling only the upper level rooms, was the planned occupancy and use of the house. Most of the family living is in the upper level. It was decided to place a heat pump in the attic since, as mentioned, space for

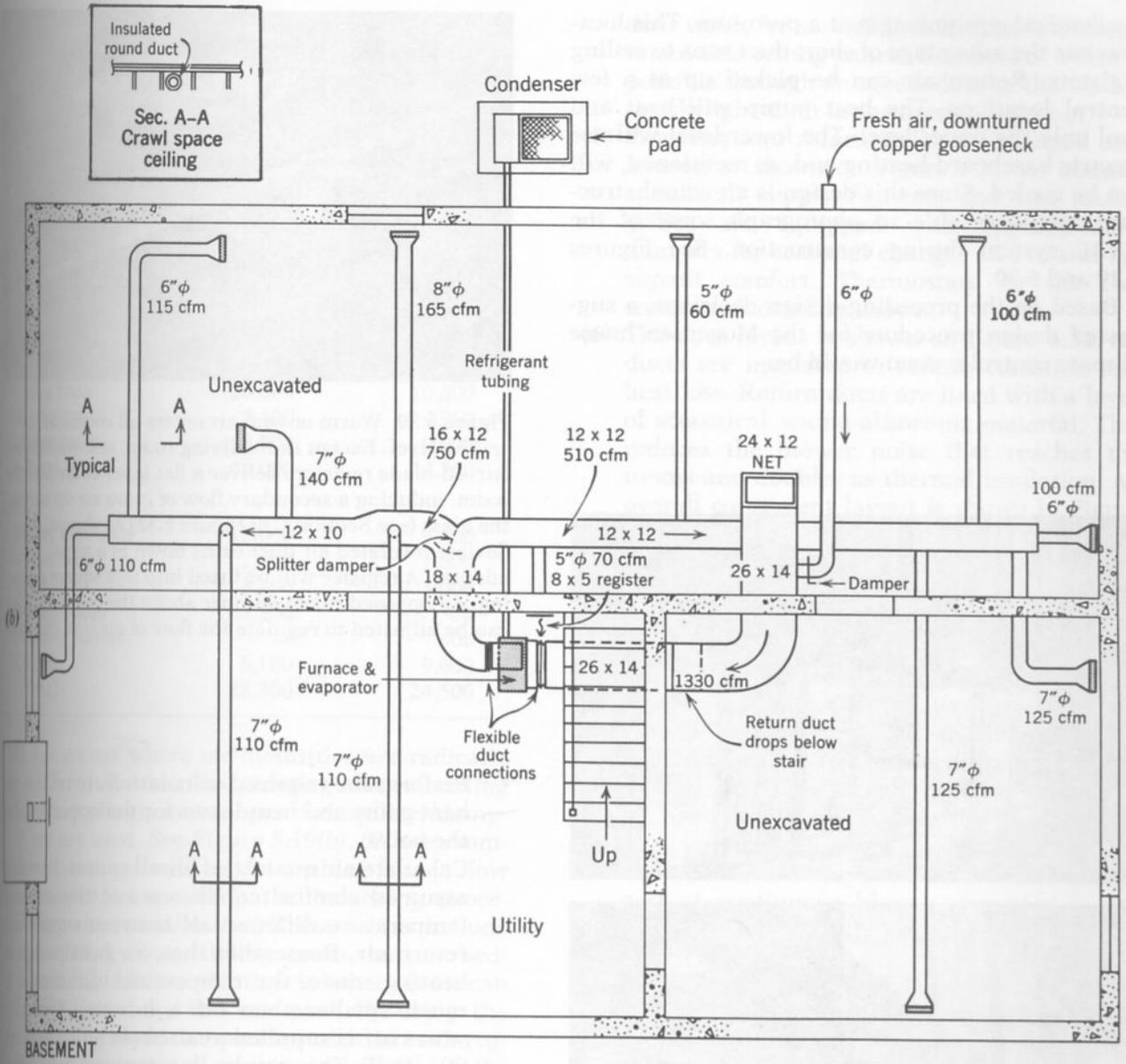


Figure 6.28 Duct layout in basement and unexcavated space of The Basic House.

mechanical equipment is at a premium. This location has the advantage of short duct runs to ceiling registers. Return air can be picked up at a few central locations. The heat pump will heat and cool only the upper level. The lower level will use electric baseboard heating and, as mentioned, will not be cooled. Since this design is an actual structure, we were able to photograph some of the HVAC system during construction. See Figures 6.29 and 6.30.

Based on the preceding design decisions, a suggested design procedure for the Mogensen house climate control system would be:

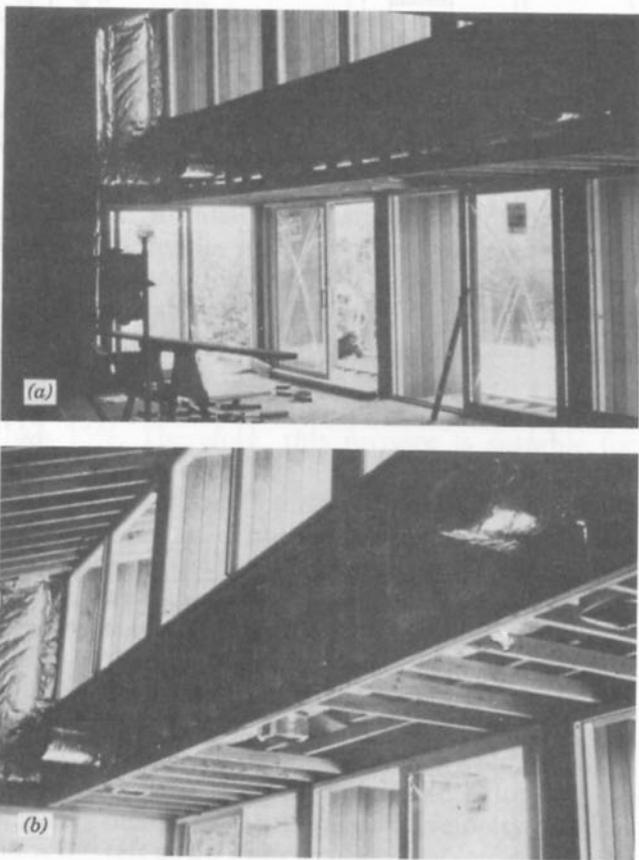


Figure 6.29 Construction photographs, Mogensen house. (a) View of living room looking south. (b) Close-up of air distribution system. Three two-way throw, curved-blade registers will deliver a warm or cool air blanket in the region of the glass doors. Two double-deflection wall registers will deliver air horizontally to effect good circulation in the room. Flexible insulated ducts will be trimmed flush with wall before installation of the two wall registers. See Figure 6.32, Section B.



Figure 6.30 Warm or cool air enters all rooms at the ceiling level. Except in the living room, one-way throw, curved-blade registers deliver a flat layer of air into the room, inducing a secondary flow of room air up across the glass (see Section C of Figure 6.32) As shown, the flexible insulated air duct turns down to a metal adapter. A register will be fitted into this square opening. An opposed-blade damper above the curved blades can be adjusted to regulate the flow of air into the room.

- Perform the required calculations to obtain the heat gains and heat losses for the upper level of the house.
- Calculate air quantities for all spaces, based on assumed comfort conditions and the required temperature differentials between supply and return air. Remember that, for heat pumps in heating mode, the temperature differential is much smaller than for a furnace. Typically, warm air is supplied at a temperature between 90–100°F. This results in a temperature difference (Δt) of 15–20 F°, assuming a return temperature of 75°F. Modern design frequently uses a lower room and return temperature, in the interest of economy and energy conservation. A return temperature of 68°F would give a temperature rise of 27 F° for 95°F entering air. This means that air quantities required for heating with heat pumps are much larger than those required by heating furnaces. These latter usually supply air at 125–140°F, giving a temperature rise (over 68°F return air) of 57 F° to 72°F.
- Select heat pump equipment.
- Select air outlet locations and types.
- Make a duct layout.
- Size ducts and registers.
- Consider ventilation requirements.

The following design considerations and procedures for the heat pump climate control of this building were evaluated.

- (1) Due to the architecture and exposure of this structure, it was decided to use two zones. The zoning makes possible short duct runs and better control. The heat losses and gains for the rooms in each zone follow.

	<i>Winter Heat Loss, Btuh</i>	<i>Summer Heat Gain, Btuh</i>
Zone 1		
Living room	28,600	10,800
Hall/foyer	9,400	4,900
Dining room	8,000	9,000
Total	46,000	24,700
Zone 2		
Master bedroom	15,000	9,400
Master bath	2,800	1,600
Study	3,900	3,000
Powder room	1,500	300
Dressing rooms	—	600
Kitchen	5,100	9,600
Total	28,300	24,500

- (2) For rooms where air quantities are radically different for the two operating modes (heating and cooling), motor-operated splitter/dampers are used. See Figure 5.19(b). (Winter/summer settings will be determined by air flow measurements during the balancing procedure after installation.) These rooms include the living room, dining room, master bedroom and kitchen. Air flow in other rooms can be adjusted by the occupant by using the opposing blade dampers installed at each register. Ducts to all rooms are sized for the larger of the heating/cooling air flow requirements. Duct dampers cannot be used here because all of the duct work is enclosed in the building wall and ceilings, making it inaccessible.
- (3) Sizing the heat pumps to supply the heat load would oversize the cooling capacity to the point that it would not dehumidify satisfactorily. As a result, a smaller unit was used, equipped with a resistance heating element to pick up the additional heat load on very cold days.
- (4) The indoor air handlers and coils are suspended in the attic. See Figures 6.22(c) and 6.32. They are connected to the two exterior slab-on-grade mounted outdoor units with insulated refrigeration lines. Access to these in-

door units, for servicing, is available through removable ceiling panels. Removal of a unit is possible through an access door in the wall of the skylight shaft in the master bedroom.

- (5) Registers and return grills are selected and located as shown on Figure 6.31. As already stated, all supply registers and two return grilles are equipped with opposed-blade dampers for seasonal changes and to suit occupant comfort. Thermostats for the two zones are located as shown.
- (6) The duct layout is shown on Figure 6.32. All ducts are insulated to reduce vibration and heat loss. Return ducts are lined with a layer of acoustical, sound-absorbing material. This reduces the blower noise that reaches the rooms and doubles as thermal insulation. An overall equipment layout is shown in Figure 6.33.
- (7) Thermostats T_1 and T_2 control the operation of the two heat pumps independently. Each unit will operate in heat or cool mode as required by the thermostat. All modern thermostats have a continuous run setting for blower operation. Continuous operation of the system blower, even after the heat pump is shut off, makes for temperature uniformity in the space and very gradual temperature changes over time. Too, continuous air motion, particularly in the cooling season, definitely adds to occupant comfort. The energy cost of continuous blower operation is not high and is generally considered to be well worth the expense.
- (8) Figure 6.33 is an overall equipment layout showing both the indoor and outdoor heat pump components and the piping and wiring connections schematically.
- (9) Sources of odor include the kitchen range, laundry dryer, laundry room, garage and bathrooms. By exhausting air from these spaces to the outdoors, these odors are reduced, and some humidity is eliminated. See Figure 6.34. The air that is drawn out of the house during seasons of heating or cooling must be replaced by outdoor air drawn in and conditioned by the central equipment. Figure 6.32 shows how this is done. In both zones, fresh air is admitted to the suction side of the blower coil unit. Its rate of flow may be adjusted by volume dampers in the fresh air duct near each unit.
- (10) Figure 6.32 is an engineering layout. Before installation, the contractor is required to submit, for approval of the engineer and archi-

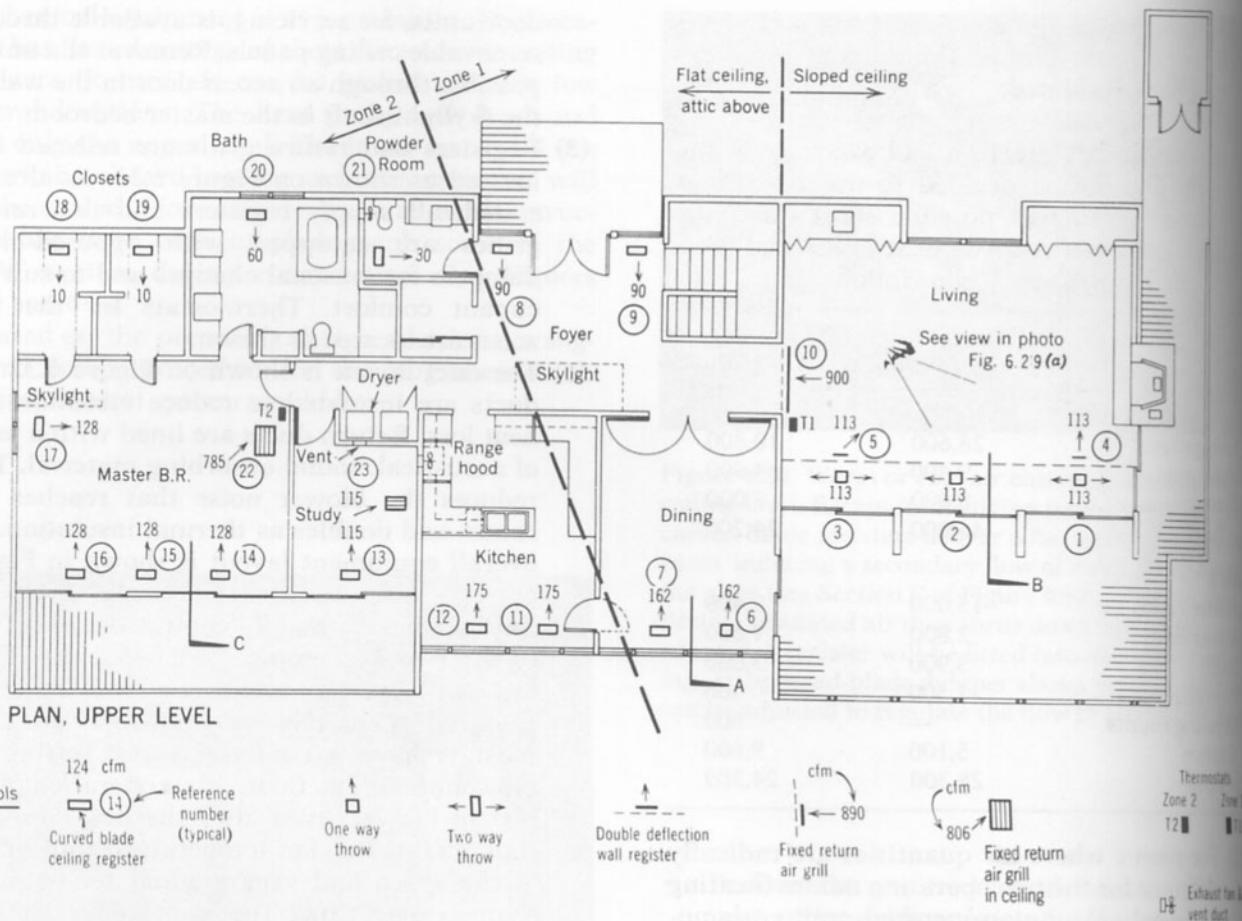


Figure 6.31 Air quantities and the layout of registers and grilles for the upper-level climate control system of the Mogensen house.

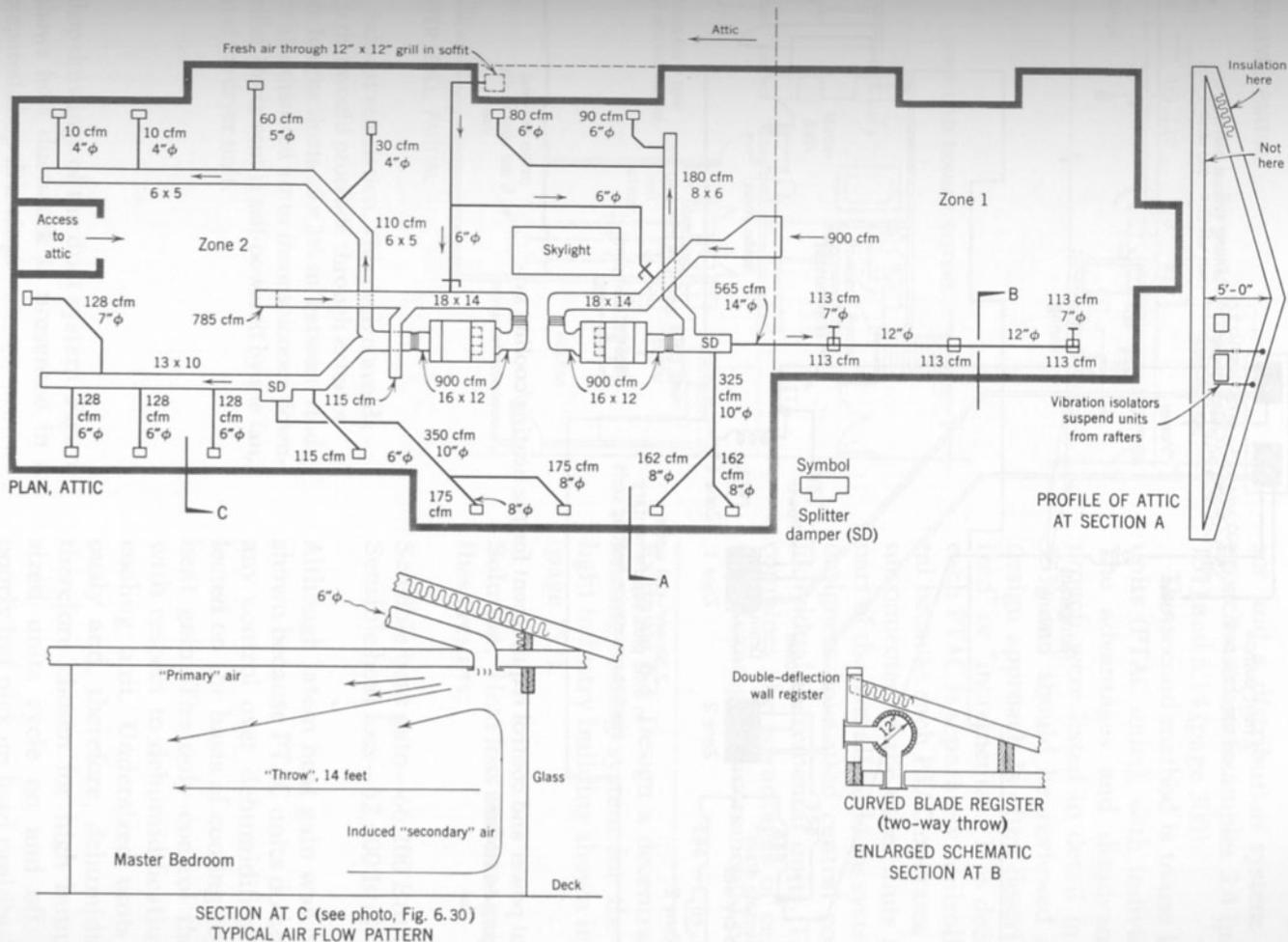


Figure 6.32 Equipment and duct layout, upper level, Mogensen house.

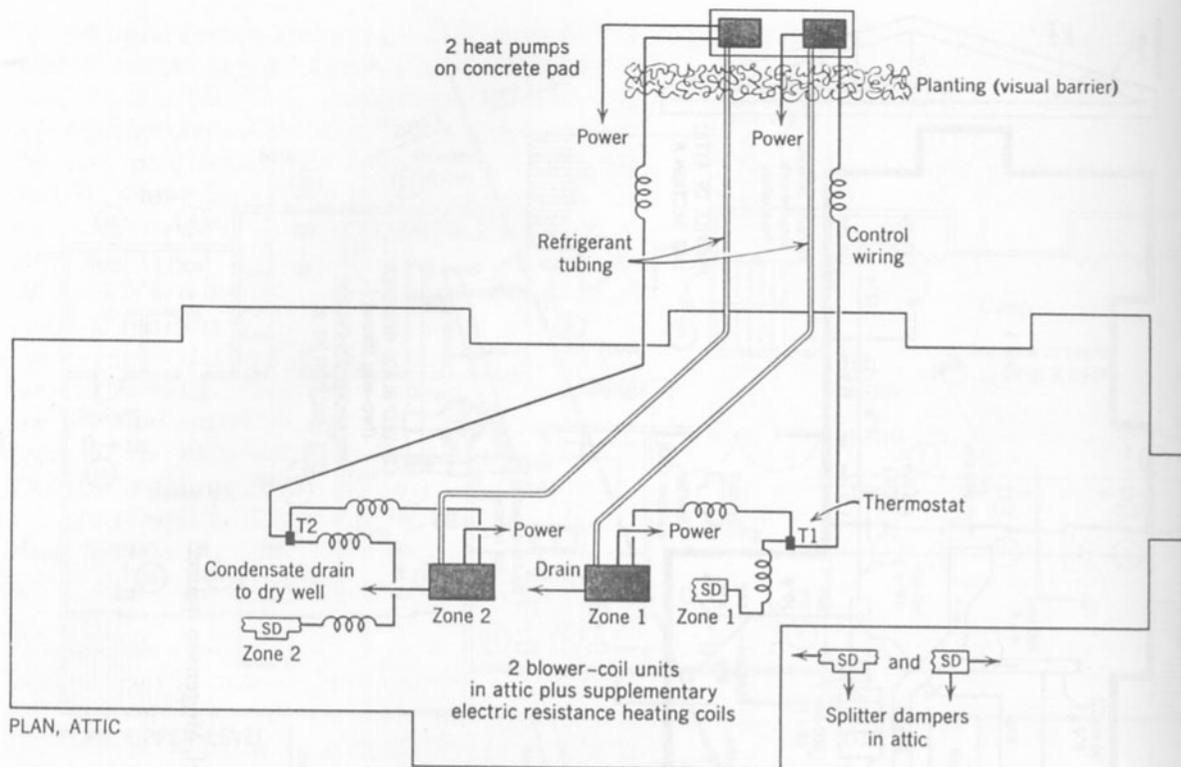


Figure 6.33 Electrical power and control requirement for the heating/cooling system, upper-level Mogensen house.

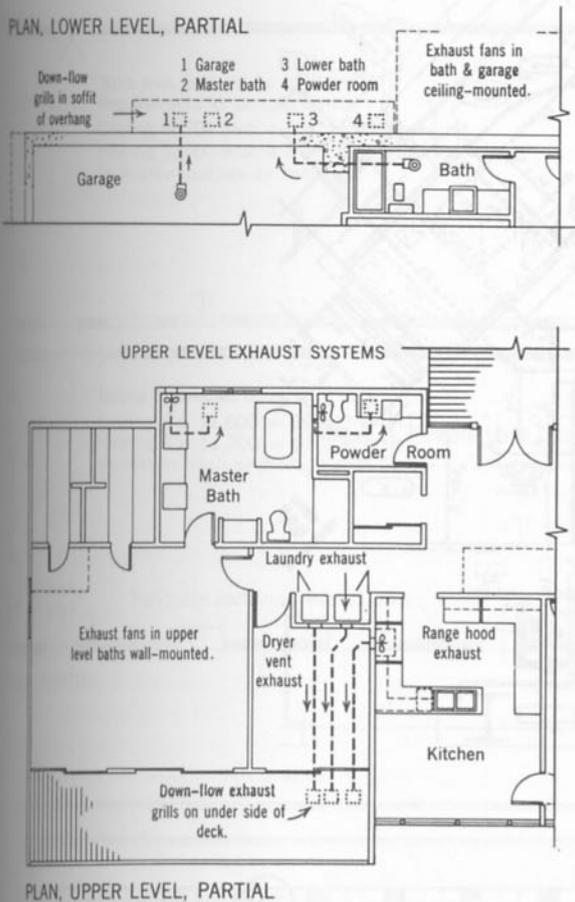


Figure 6.34 Exhaust ventilation. This design avoids units or ducts that would protrude through exterior walls or roofs. Interior ducts $10 \times 3\frac{1}{4}$ in. between studs or joists carry all exhausted air to inconspicuous down-flow soffit grilles. Dryer vent is self-powered by the fan, which is part of the dryer unit.

tect, shop drawings of the duct system. Figure 6.35 shows how ductwork is presented in a well-prepared shop drawing.

6.17 Design Example—Light Industry Building

In the previous two (residential) design examples, we used a furnace/split air conditioner combination and a split heat pump, both with ductwork. In this example, we will use nonducted through-the-wall incremental units of the type shown in Figure 6.19. As explained in Section 6.9, there are two methods of supplying the HVAC requirements of a large space. One method is to use a central proces-

sor and a distribution system. That was the approach used in Examples 3.4 (page 142), 3.5 (page 151) and 5.14 (page 300).

The second method is to use individual package units (PTAC units), with individual local control. The advantages and disadvantages of this approach were listed in detail in Section 6.9 (page 353) and should be reviewed at this time. This design approach is often described as “decentralized” or “incremental.” It is decentralized because each PTAC is separately controlled. It is incremental because each PTAC operates as a separate unit, unconnected to the other units in the building yet part of the overall building system. In recent years, designers have used central computer control of individual incremental units. This type of system combines the advantages of central control with the advantages of incremental package units.

Example 6.3 Design a decentralized, incremental heating-cooling system for the work area of the light industry building shown in Figures 3.48–3.50 (page 153).

Solution: Heat loss and heat gain calculations give these results:

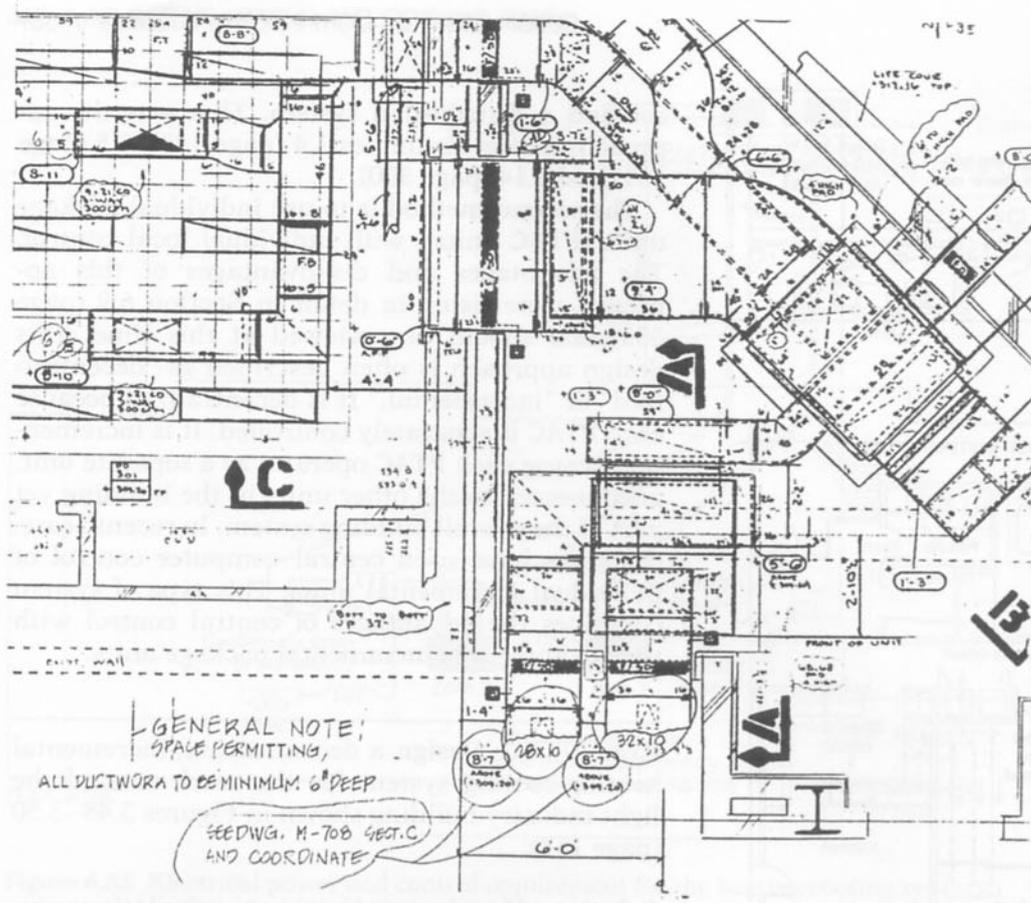
Sensible heat gain—66,200 Btuh
Sensible heat loss—52,600 Btuh

Although latent heat gain was calculated, it is not shown because PTAC units do not give the designer any control over dehumidification. Units are selected on the basis of cooling capacity for sensible heat gain. The only control that a designer has with respect to dehumidification is by size of the cooling unit. Undersized units operate continuously and, therefore, dehumidify well. They are therefore chosen for high humidity areas. Oversized units cycle on and off. They dehumidify poorly but pick up load rapidly. They are therefore more applicable to dry (hot) climates.

The following tabulation shows typical data for PTAC units that are physically suitable.

Refer to Figure 3.53, which shows the location of six hydronic heating units for the same space. A designer could alternatively have elected to use PTAC units with hydronic heating coils and electric cooling. However, we have chosen to make this an all-electric design using package units.

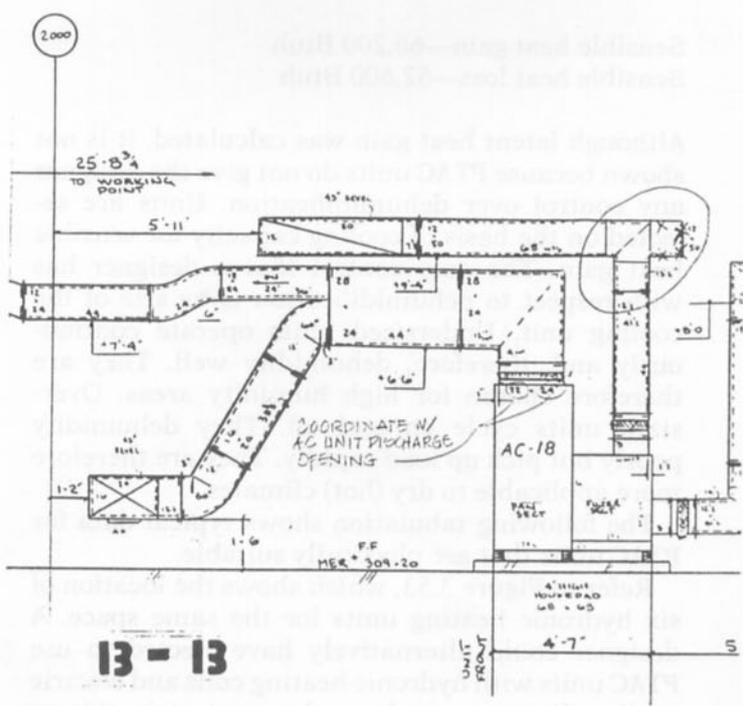
The hydronic units shown in Figure 3.53 give good coverage. We will, therefore, use a similar layout for six PTAC units. They are shown in Figures 6.36 and 6.37. Using six units, the required



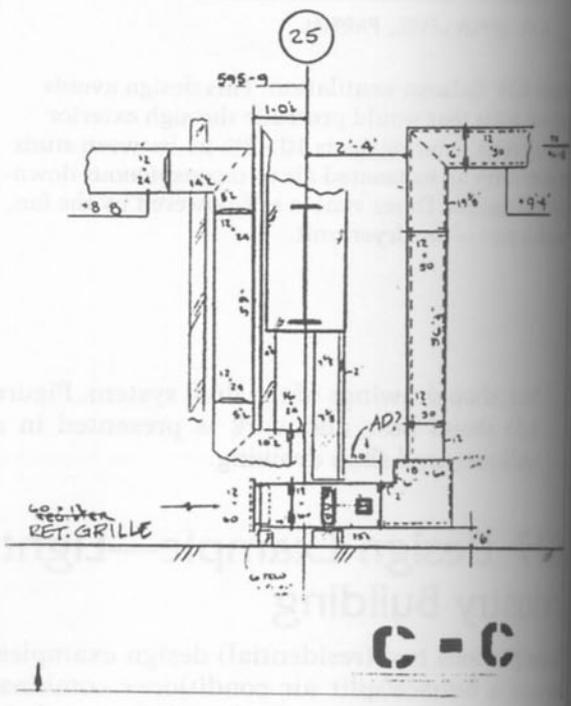
GENERAL NOTE:
SPACE PERMITTING,
ALL DUCTWORK TO BE MINIMUM 6" DEEP

SEEDWG. M-708 SECT. C
AND COORDINATE

(a)



(b)



(c)

Figure 6.35 (a) Part of a large ductwork shop drawing. Since the ducts will be installed in layers, one above the other, vertical sections (b) and (c) are drawn to show details. (Courtesy of Cool Sheet Metal Co.)

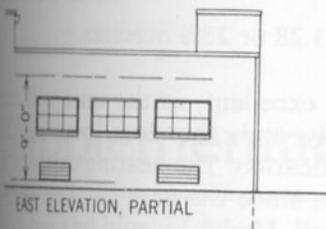
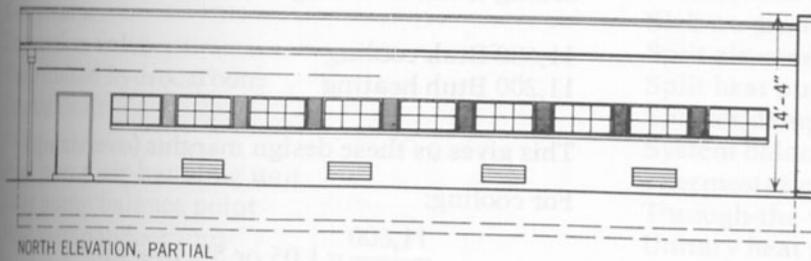
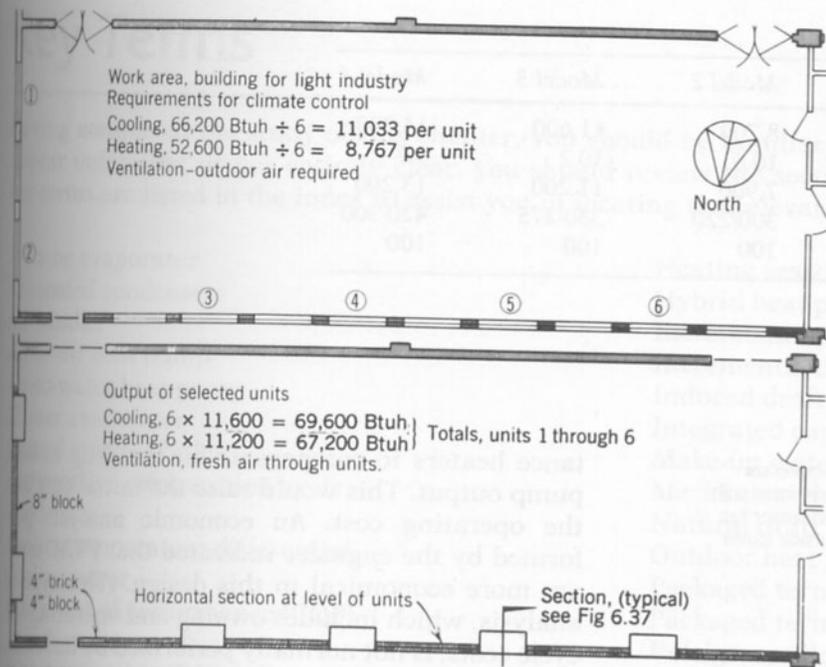


Figure 6.36 Layout of the solution to Example 6.3. The space is heated and cooled using through-the-wall all-electrical PTAC units.

ABC Manufacturing Company

Item	Model 1	Model 2	Model 3	Model 4
Cooling capacity, Btuh	6,600	8,700	11,600	14,200
EER	10.0	10.6	10.2	9.7
Heating capacity, Btuh	6,200	7,900	11,200	13,200
CFM—2-speed blower	250/180	300/220	380/275	420/300
CFM—ventilation	100	100	100	100

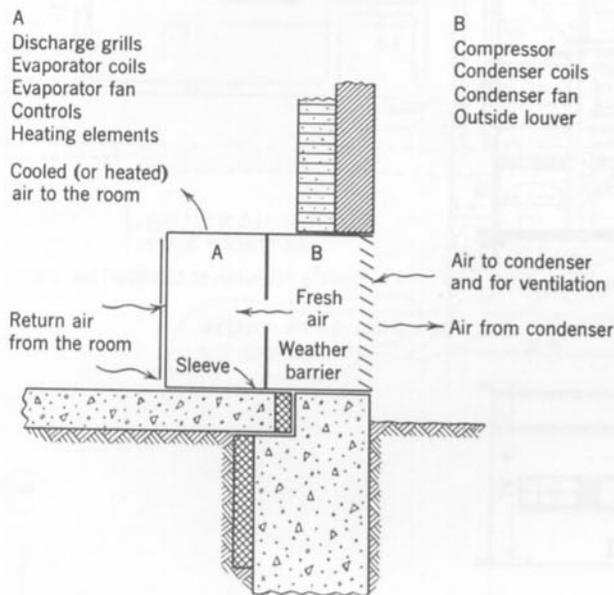


Figure 6.37 Through-the-wall PTAC unit installed.

heating and cooling capacity for each unit are:

Cooling:

$$\frac{\text{Total load}}{6} = \frac{66,200 \text{ Btuh}}{6} = 11,033 \text{ Btuh}$$

Heating:

$$\frac{\text{Total load}}{6} = \frac{52,600 \text{ Btuh}}{6} = 8,767 \text{ Btuh}$$

PTAC units are chosen rather than PTHP (heat pump) units, because the winter design temperature of 0°F would require large supplemental resis-

tance heaters to compensate for the drop in heat pump output. This would raise the initial cost and the operating cost. An economic analysis performed by the engineer indicated that PTAC units are more economical in this design. (This type of analysis, which includes owning and operating life cycle costs, is not normally performed by technologists. For more information, consult the bibliography for a more technical book by this author.)

The Model 3 PTAC unit is chosen from the preceding table. Its ratings are:

11,600 Btuh cooling
11,200 Btuh heating

This gives us these design margins (oversizing).

For cooling:

$$\frac{11,600}{11,033} = 1.05 \text{ or } 5\% \text{ oversize}$$

For heating:

$$\frac{11,200}{8,767} = 1.28 \text{ or } 28\% \text{ oversize}$$

The cooling size is excellent, as the unit will run almost continuously and, therefore, provide the required dehumidification. The heating capacity is excessive. However, since the next smaller unit—Model 2—is too small, Model 3 would be used. It is frequently difficult to meet both the heating and cooling requirements without some oversizing or undersizing. In this case, since the majority of PTAC units with resistance heaters have high low settings in addition to thermostatic control, the heating capacity oversize can be reduced by use of the low heat setting.

Key Terms

Having completed the study of this chapter, you should be familiar with the following key terms. If any appear unfamiliar or not entirely clear, you should review the section in which these terms appear. All key terms are listed in the index to assist you in locating the relevant text.

A-frame evaporator
 Air-cooled condensers
 Air handler
 Air-to-air heat pump
 Air-to-water heat pump
 Blower evaporator
 Central system chiller
 Compressive refrigeration cycle
 Condenser
 Cooling temperature differential
 Cooling tower
 Coefficient of performance (COP)
 Distributed system
 Direct expansion (DX) coil
 DX fan coil unit
 Energy efficiency ratio (EER)
 Evaporative condensers
 Evaporator
 Expansion valve
 Fluorinated hydrocarbons
 Forced draft
 Freon
 Heat pump air-handling unit
 Heat pump balance point
 Heat pump indoor unit
 Heat sink
 Heat source
 Heating mode
 Heating seasonal performance factor (HSPF)
 Hybrid heat pump
 Incremental package system
 Incremental units
 Induced draft
 Integrated capacity
 Make-up water
 Mechanical draft tower
 Natural draft tower
 Outdoor heat pump
 Packaged terminal air conditioner (PTAC)
 Packaged terminal heat pump (PTHP)
 Refrigerant flow controller
 Rooftop units
 Saturation temperature
 Seasonal energy efficiency ratio (SEER)
 Sensible heat ratio (SHR)
 Slab-on-grade units
 Split air conditioners
 Split heat pump
 Splitter/dampers
 System balance point
 Thermostatic expansion valve
 Through-the-wall unit
 Unitary heat pump
 Water-cooled condensers
 Water-to-air heat pump
 Water-to-water heat pump

Supplementary Reading

B. Stein and J. Reynolds *Mechanical and Electrical Equipment for Buildings*, 8th ed., John Wiley & Sons, New York, 1992.

ASHRAE, American Society of Heating, Refrigeration and Air Conditioning Engineers

1791 Tullie Circle, N.E.

Atlanta, Ga. 30329 Tel. 404-636-8400

1993 *Handbook—Fundamentals*

ACCA, Air Conditioning Contractors of America

1513 16th Street, N.W.

Washington, D.C. 20036

Manual CS—Commercial Applications, Systems and Equipment, 1993

Manual S—Residential Equipment Selection

SMACNA, Sheet Metal and Air Conditioning Contractors National Association, Inc.

8224 Old Courthouse Road

Tysons Corner, Vienna, Va. 22180

HVAC Systems Applications, 1987

Problems

1. How many tons of refrigeration are required to cool a space with a sensible heat gain of 65,000 Btuh?
2. Will an undersized or oversized refrigeration unit provide better dehumidification? Why?
3. A gas-fired stowaway furnace in an attic provides full winter and summer air conditioning. It is served by an outdoor compressor-condenser.
 - a. Name five connections to the attic unit other than electricity.
 - b. Draw a sketch showing the units and their connections.
4. a. Is the fresh (outdoor) air supply duct connected to the return duct or to the supply air duct?
b. Give the reason for your answer.
5. In a heat pump, the compressor operates whenever heating or cooling is needed.
 - a. Where does evaporation of the refrigerant take place in summer: outdoors or indoors?
 - b. Where does condensing of the refrigerant take place in winter: outdoors or indoors?
6. Name four locations in a residence from which it is desirable to have exhaust ventilation.
7. In a house with an air heating-cooling system, how is the air that is drawn out of the house by exhaust fans replaced?
8. When a heat pump is in use for heating and the outdoor temperature drops from 50 to 30°F, does the heat pump become less efficient or more efficient? Explain.
9. Explain briefly why COP applies to heat pumps and not to air conditioners.
10. a. Why is calculation of latent cooling load more important for nonresidential buildings than for residential ones?
b. In the same climate, which requires more cooling per square foot, a residence or a department store? Why?
c. Which require more heating? Why?
11. Define briefly the following terms:
 - a. Compressive refrigeration cycle—condenser, evaporator
 - b. Air-to-water heat pump
 - c. Heat source
 - d. Heat sink
 - e. Evaporative cooling
 - f. COP
 - g. EER, SEER
 - h. Balance point
 - i. DX, dry expansion, direct expansion
 - j. PTAC, PTHP
 - k. Cooling tower
12. Does outside humidity affect the performance of a cooling tower? How? Why?
13. What is the function of a four-way flow switch in a heat pump?
14. Is a heat pump more efficient in the heating cycle or the cooling cycle? Explain.
15. Why is a heat pump called that? What is being pumped? Explain.
16. a. What is it that a condenser condenses? How?
b. What is it that an evaporator evaporates? How?
17. How can a heat pump in its heating cycle deliver more (heat) energy than is taken from the electrical input? Doesn't this contradict the law of conservation of energy?
18. How much heat per pound of dry air is contained in air at 0°F? 32°F? 100°F?
19. In an air-to-air PTHP, what are the heat sources and heat sinks in the heating mode? Cooling mode?
20. What heat sources and sinks can be used with a water-coupled heat pump?
21. What is the EER of a heat pump that operates in heat mode with a COP of 3.2?
22. What limitations does the National Appliance Energy Conservation Act place on air conditioner and heat pump performance?
23. Why are defrosters necessary on heat pumps?
24. A store has a 0°F design condition winter heat load of 44,000 Btuh. Using the heating performance data of heat pump models 10A given in Figure 6.18(b), plot the building and heat pump curve and find the balance point.
 - a. Which model heat pump will supply all the heating required (if any)?
 - b. Will auxiliary heat be required? When?
25. Use the same heat pump characteristics as plotted in Problem 24. What is the balance point for a residence with a 10°F design condition heating load of 36 MBH. Which heat pump model is best? Explain.
26. List four types of heat pumps, with different heat source and heat sinks. Draw a block diagram of each and label the parts in heating and

cooling modes. Show the heat flow through the evaporator and condenser, with approximate temperatures. Justify all assumptions.

27. List five advantages and five disadvantages of using unitary, incremental units.
28. How does latent heat load affect the choice of an evaporator coil? (*Hint: Read Section 6.22.*)
29. The heat losses and gains for The Basic House design problem are based on a 15°F winter design condition, and 89°F DB, 75°F WB summer conditions.
 - a. Assuming that heat loss varies linearly with outside temperature, recalculate the heat loss for all spaces using a winter design temperature of 0°F.
 - b. In the interest of energy conservation, we

- are changing the summer inside design temperature from 75 to 78°F. Outside design conditions remain the same. Recalculate the cooling load for each space. Assume that cooling load, like heating load, is linearly proportional to the required temperature difference. (This is not strictly correct.)
- c. A conventional gas furnace will be used instead of a condensing unit. This requires moving the furnace (and the A-frame evaporator) to a location near the chimney. Redraw the duct layout of Figure 6.28, and recalculate all duct sizes. Be specific about all assumptions. Show pressures and velocities being used for each duct section.

Key Terms

Having completed this chapter, you should be familiar with the following key terms. If any appear unfamiliar or not entirely clear, you should review the section in which these terms appear. All key terms are listed in the index to assist you in locating the relevant text.

Anemometers
Balancing
Bimetallic element
Bourdon gauge
Bourdon tube
Capillary tubing
Capillary tube thermometer
Deflecting vane anemometer
Dial thermometer
Differential pressure
Draft gauges
Flow straightener
Hot-wire anemometer

Magnetic pressure gauge
Manometers
Pitot tube
Pyrometers
Resistance temperature device (RTD)
Rotating vane anemometer
TAB
Thermal anemometers
Thermocouple
Traverse measurements
Velocity pressure
Velometer

Supplementary Reading

B. Stein and J. S. Reynolds, *Mechanical and Electrical Equipment for Buildings*, 8th ed., John Wiley & Sons, New York, 1992. This book covers the same areas of study as the present book, but in greater detail and scope. It is very useful for further study.

American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE)
1791 Tullie Circle, N.E.
Atlanta, GA 30329

Handbook—HVAC Applications, Chapter 34, 1991
Sheet Metal and Air Conditioning Contractors National Association, Inc. (SMACNA)
8224 Old Courthouse Road
Tysons Corner, Vienna, VA 22180
HVAC Systems; Testing, Adjusting and Balancing, 1983

E. G. Pita, *Air Conditioning and Systems: An Energy Approach*, Chapter 16, John Wiley & Sons, New York, 1981.

Problems

- A manometer will be used to test the pressures in an air system. Maximum blower pressure is 0.6 in. w.g. Would you use a water manometer or a mercury manometer? Why?
- The following temperature measurements must be made in a TAB project. What type of thermometer would you use? Why?
 - Motor bearing temperature.
 - Oil reservoir temperature.
 - Air stream temperature.
 - Pipe surface temperature.
- What is a capillary tube? What does it contain when used to connect a temperature sensing bulb to a thermometer dial?
- What is a thermocouple? How is it used to measure temperature?
 - What is a bimetallic element? How is it used to measure temperature?
- Is an inclined scale manometer more accurate

than a vertical unit? Is it more precise? Explain.

6. A Pitot tube traverse in a duct gives the following velocity pressure readings for 16 equal areas of duct cross section. The pressure units are in. w.g. Find the average duct velocity in cfm.

0.26 0.29 0.29 0.25

0.27 0.32 0.33 0.27

0.29 0.33 0.34 0.28

0.27 0.29 0.31 0.25

7. A TAB technician wants to make a four-point Pitot tube traverse of a 4-in. round duct. Show where the Pitot tube should be positioned on a diameter to accomplish the traverse accurately.
8. An 8 × 14-in. register is designed to deliver 400 cfm. What should be the average velocity over its face? Explain.

9. The following air velocity readings are obtained over the face of a 8 × 14-in. register that has a K factor of 0.7. What is the average velocity over the face of the register? What is the flow in cfm?

620 650 660 630

640 680 700 635

625 650 630 620

10. Three draft gauges are to be used with a Pitot tube to simultaneously measure total pressure, static pressure and velocity pressure in a duct. Show how the gauges are connected for
- A supply air stream.
 - An exhaust air stream with positive pressure.
 - An exhaust air stream with negative pressure.

Explain.

7. Testing, Adjusting and Balancing (TAB)

All HVAC systems, regardless of size, require adjustment after installation. This does not mean that the installation was incorrect. It simply means that, even in a small simple system, all sorts of field adjustments must be made to achieve the design intention. These adjustments include motor speeds, pressure adjustments, valve settings, fuel supply control, liquid-level controls, temperature settings, damper positions and so on. This work is completely separate from the work of a field inspector. An inspector's task is to see to it that the system is installed according to plans and specifications. Once the installation is complete and approved by the inspection team, the work of testing, adjusting and balancing (abbreviated TAB) begins. Strictly speaking, the three portions of TAB are different from each other. *Testing* is the procedure that checks that equipment operates as it is supposed to. Motors turn, pumps deliver liquids, manual and automatic controls perform as required and so on. *Adjusting* is the work of setting and regulating variables such as flow, pressure, speed and temperature. *Balancing* is closely connected to adjusting because it is concerned with the flow quantities of air and water. The individual TAB processes are so interrelated that the whole procedure is simply referred to as *system balancing*. The work is very technical and highly specialized. It requires a good hands-on working knowledge of all HVAC systems, an equally thorough familiarity with a whole range of TAB instruments plus, of course, an ability to understand HVAC plans. TAB specialists very often start their careers as HVAC technologists. It is for this reason that a chapter on TAB is included here. Study of this chapter will enable you to:

1. Understand the purpose and function of testing and balancing of HVAC systems.
2. Be familiar with the functioning and applica-

tion of instruments used in HVAC testing and balancing work.

3. Perform traverse measurements in ducts using Pitot probes and manometers.
4. Understand how to use various types of anemometers in air velocity measurements.
5. Calculate average air velocities and air flow in ducts.
6. Perform the necessary preparations for balancing an air system.
7. Accomplish the balancing of straightforward limited-size air systems and assist in the balancing of large complex air systems.
8. Make the many necessary preparations for balancing a hydronic system.
9. Balance a residential hydronic heating system and assist in balancing large hydronic heating/cooling systems.
10. Prepare the report forms containing all the balancing data for air and hydronic HVAC systems.

Instrumentation

TAB work is possible only with adequate instrumentation. The physical quantities that require measurement include temperature, humidity, pressure, flow velocity and quantity, rotational speeds and electrical power and energy. For each of these physical quantities, instruments are available to suit the range and the physical accessibility of the quantity being measured. Before use, the requirement for calibration of each instrument should be checked. Some instruments maintain their accuracy for long periods of time or do not require calibration at all. Others require frequent recalibration. The manufacturers' instructions on this point should be carefully observed in order to ensure accurate measurements. There are so many instruments on the market today that a comprehensive survey would fill an entire volume. In particular, new electronic instruments appear almost daily. They offer such desirable and time-saving features as auto-ranging, digital readout, memories, and programmability. In the material that follows, we will review the basics of HVAC instrumentation, leaving the details of a specific instrument to the ability and intelligence of the technologist.

7.1 Temperature Measurement

a. Glass Tube Thermometer

See Figure 7.1(a). The simplest and most common type of thermometer is the glass tube design. All such units operate on the same principle. A reservoir at the base of the tube contains a liquid. The liquid expands and contracts according to the temperature of its surroundings—generally air or water—forcing liquid up through a calibrated glass tube. The liquid most frequently used is mercury. Mercury-filled glass tube thermometers have a useful temperature range of -40 to 1000°F , and the tubes are calibrated accordingly. Glass tube thermometers have the advantages of accuracy, indefinite life, no need for calibration and accuracies of up to 0.5% (or one-third of a scale division depending on the scale).

Their principal disadvantage is that the entire bulb (liquid reservoir) must be immersed in the fluid whose temperature is being measured. If the fluid is a liquid, full immersion can be seen. If the fluid is air, the technician must be careful to shield the bulb from surrounding surfaces at substantially different temperatures. An ambient air temperature reading will be highly inaccurate if an unshielded reading is taken near a boiler or furnace. Since glass tube thermometers take a while to achieve their final reading, several readings should be taken, a few minutes apart, each one lasting several minutes. When the same reading occurs at least twice in succession it can be recorded as the correct temperature.

Most TAB technicians use a range of glass tube thermometers with different scale graduations and different physical size. Each type is useful for a limited range of applications. Some technicians use bulb-type glass tube thermometers to measure the temperature of a pipe by placing the bulb against the pipe and wrapping the two with insulating tape. This procedure should be avoided because it is inaccurate. The line contact between the thermometer bulb and the pipe is inadequate for proper measurement. Furthermore, the insulated wrapping will prevent heat radiation from the pipe causing an artificially high reading. When surface temperature measurement is required, a special type of thermometer, called a *pyrometer*, should be used. This instrument is discussed in Section 7.1.d.

b. Dial Thermometer with Bimetallic Element

This type of thermometer uses a bimetallic element similar to that in a simple thermostat to measure temperature changes. When two metals that have different coefficients of expansion are joined together, a change in temperature causes the combination to bend or twist, depending on how they are joined. This motion is transmitted to a circular dial by a mechanical linkage. The dial is graduated in degrees of temperature. These thermometers are made in a wide variety of temperature ranges and physical designs. When used to measure liquid temperatures, the bimetallic element is mounted in a hollow metal stem attached to the dial. This stem is then immersed in the liquid whose temperature is to be measured. Domestic baking/meat thermometers are made in this design. When used to measure moving air temperature, the bimetallic element is installed inside the meter case, and so arranged that the air to be checked passes over the element. This type is illustrated in Figure 7.1(b-1). A chart-recording unit of this design is shown in Figure 7.1(b-2). These units have the advantages of ruggedness (unlike the glass tube type) and indefinite life. Although they should not require recalibration, they should be checked against a mercury glass tube unit periodically because the linkage can be damaged. This would result in an incorrect reading. These units have limited accuracy (± 5 – 10%) and are useful for quick checks.

c. Capillary Tube Thermometer

See Figure 7.1(c). One design of this type of thermometer uses a Bourdon tube, which is identical to that found in a pressure gauge. (See Figure 8.4, p. 410.) The tube is connected at one end to a long flexible fluid-filled capillary tube that ends in a relatively large sensing bulb. The other end is connected to a Bourdon gauge that is graduated in temperature degrees. The capillary tube and bulb are filled with a liquid or gas. Changes in temperature cause the fluid to expand or contract. This, in turn, changes the pressure in the Bourdon tube and, by means of a mechanical linkage, moves the dial pointer. This design is highly accurate ($\pm 1/2\%$), fairly rugged and needs infrequent calibration. Its principal advantage is the ability to read temperatures remotely. The standard length of capillary tubing is 6 ft. Units are available to measure tem-

perature of -60 to 500°F in individual ranges of about 100°F .

Another design of capillary tube thermometer uses the liquid in the tubing to read directly in a graduated glass tube. This type is essentially the same as the glass tube thermometer described previously, except that the liquid reservoir, in the form of a sensing bulb, is connected to the glass tube by a long capillary tube. This permits remote sensing. Remote sensing is very useful when measuring temperatures at locations that are difficult or hazardous to get at. A recording unit of this type is illustrated in Figure 7.1(c-2).

d. Pyrometers

These units are normally used to measure surface temperature of pipes, ducts and equipment. The unit's sensing element contains a bimetallic thermocouple. This thermocouple generates a small voltage, which is proportional to its temperature. This voltage can be measured by a millivoltmeter that is calibrated in degrees. The sensing element is connected to the instrument by wires and can, therefore, be remote at almost any distance. The great advantage of this type of instrument is that a single meter can be used to monitor as many thermocouples as desired by simply switching between the wires. Thermocouples are frequently permanently installed in equipment to permit continuous or periodic temperature checking (and alarm functions). A digital electronic pyrometer that can display temperature in either $^\circ\text{F}$ or $^\circ\text{C}$ at selectable precision is illustrated in Figure 7.1(d). Pyrometers are highly accurate, require calibration once or twice a year and cover a huge range of temperatures, according to the type of thermocouple used. When used for surface temperature sensing of a pipe or duct, the manufacturer's directions should be carefully followed, because incorrect readings can result from improper probe use.

e. Thermal Anemometers

These devices, described in Section 7.3(b), are primarily intended to measure air velocity. As a secondary function, some of these units also measure temperature. Two such units are illustrated in Figure 7.5(c) and (d). They are mentioned here because of their auxiliary temperature measuring capability. However, because their principal function is to measure air velocity, they are discussed in detail in Section 7.3.

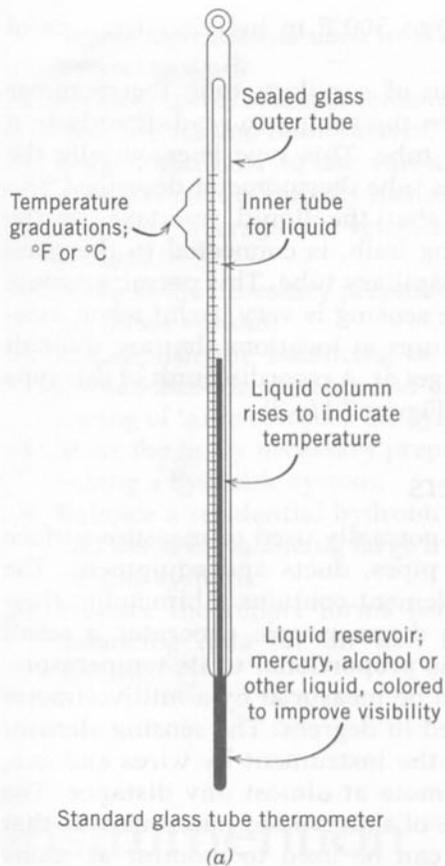


Figure 7.1 (a) Typical glass tube thermometer. They are available in a very wide range of physical sizes and temperature ranges and graduation precision.

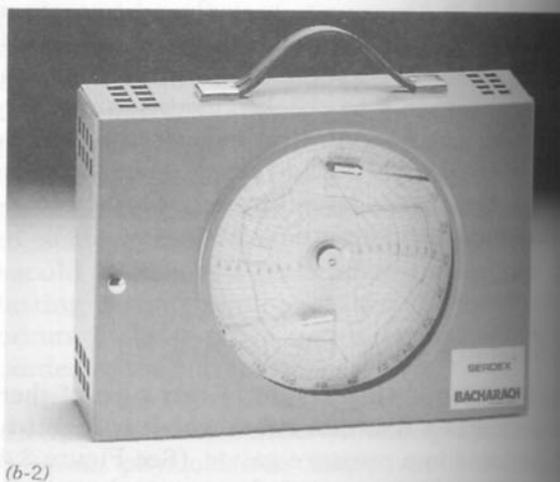
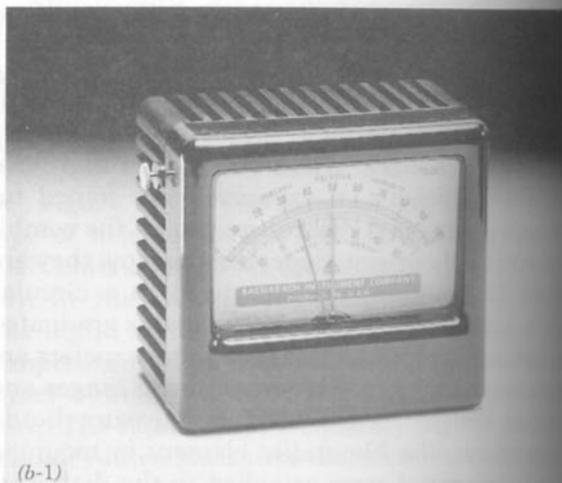


Figure 7.1 (b-1) Temperature/humidity indicator. Temperature of air passing through and around the unit is measured by a bimetallic coil-type sensing element within the unit. The dial is calibrated from 0 to 130°F. The meter also measures relative humidity with a membrane diaphragm that responds rapidly to humidity changes. The humidity range is 0–100% RH. (b-2) Chart recorder that measures temperature and humidity as described for (b-1) and records the measurements on a 6-in. diameter, 24-hr circular chart. (Photos courtesy of Bacharach, Inc.)

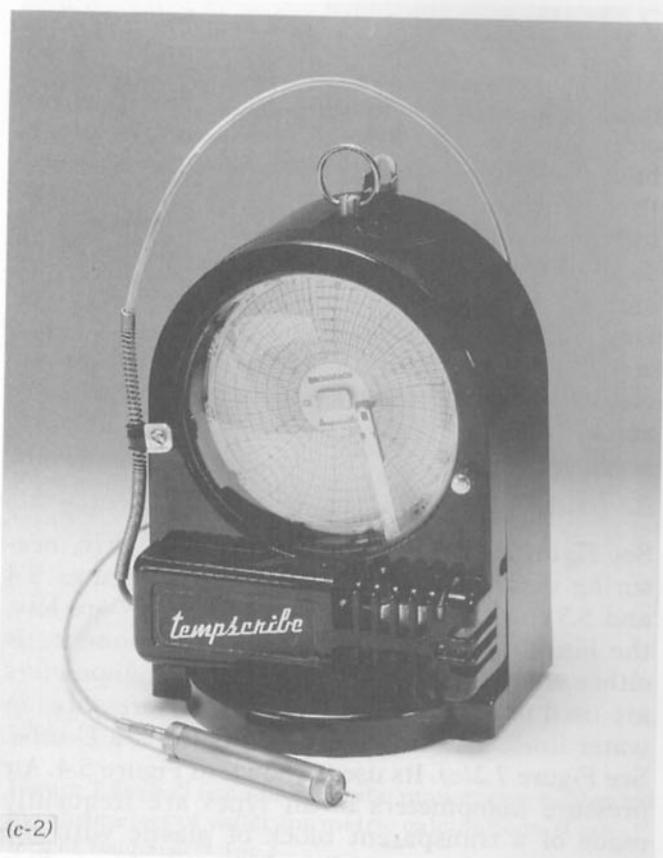
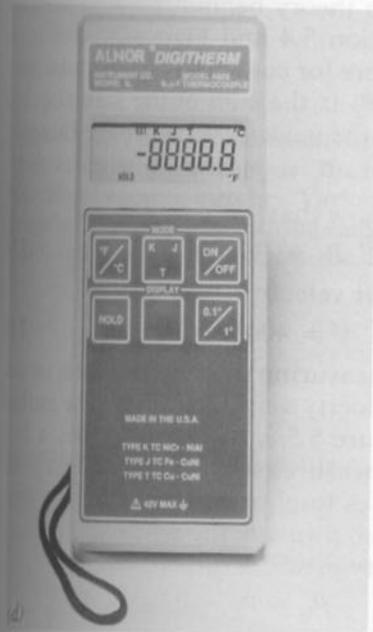
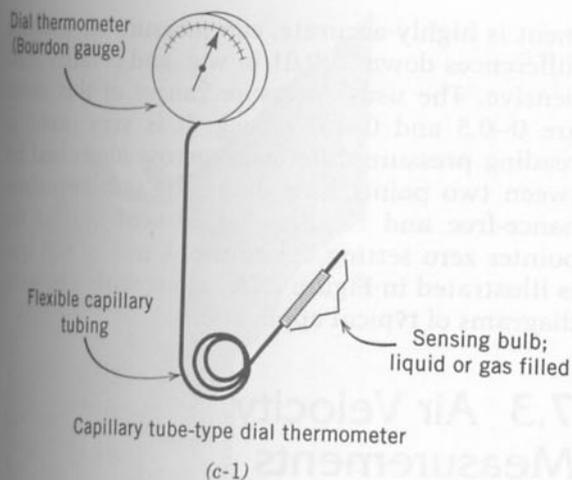


Figure 7.1 (c-1) Capillary tube-type dial thermometer. Expansion and contraction of the fluid in the sensing bulb causes a change in pressure in the Bourdon gauge. This pressure change causes movement of the dial pointer on a scale that is graduated in temperature degrees. (c-2) Capillary tube-type recording thermometer. The temperature-sensing bulb (in the lower foreground of the photo) senses temperature and transmits the signal through 6 ft of capillary tubing to the 24-hr or 7-day chart recorder. Temperature ranges are available in different models from -30 to 120°F and -35 to $+50^{\circ}\text{C}$. (d) Modern digital electronic hand-held pyrometer, with a range of thermocouple probes and sensors (not shown). Primarily intended for surface temperature measurements, although usable for measuring immersion temperatures of gases as well. The illustrated unit is programmable, will hold and retain readings and will measure in a number of ranges over an extremely wide range with good accuracy. [(c-2) Courtesy of Bacharach, Inc. (d) Courtesy of Alnor Instruments Company.]

7.2 Pressure Measurement

Air and water are both fluids. Pressure and its measurement in water piping and vessels is discussed in Sections 8.5–8.7 (pp. 406–410). If you have already studied those sections, a review at this point would be useful. Otherwise, after studying this section, we would advise you to read those sections in order to appreciate both the similarities and the differences. In either case, you should review Section 5.4, which explains in detail the concept of air pressure in ducts and the use of manometers of various types to measure duct air pressure and air velocity.

a. Manometers

See Figure 7.2. The action of a manometer in measuring duct air pressure is shown in Figures 5.4 and 5.5 (p. 202). Because these pressures are low, the liquid used in an air pressure manometer is either colored water or oil. Mercury manometers are used to measure the much higher pressures in water lines. The simplest manometer is a U tube. See Figure 7.2(a). Its use is shown in Figure 5.4. Air pressure manometers of all types are frequently made of a transparent block of plastic with the manometer tubing cast directly into the block. This makes the instrument almost indestructible yet capable of excellent accuracy. See Figure 7.2(b).

The units are calibrated directly in inches of water. Since manometers have no moving parts, they maintain their accuracy without recalibration almost indefinitely. These manometers, also referred to as draft gauges, are standard in the industry. The inclined scale type shown in Figure 7.2(b) can be read to an accuracy of 0.03 in. w.g. Most units have an adjusting piston in the liquid tube that permits setting the liquid's meniscus (level) at the zero pressure line. The unit is set level on a stable surface and read directly. Connection to ducts is made with flexible tubing and a probe that is introduced into the duct being tested through a test hole.

b. Magnetic (Differential) Pressure Gauge

This gauge measures the difference in pressure between two sealed compartments. This difference causes a diaphragm between the two compartments to move. The motion is transmitted through a magnetic linkage to a dial pointer. The instru-

ment is highly accurate, can measure air pressure differences down to 0.01 in w.g. and is fairly inexpensive. The usual pressure ranges of this meter are 0–0.5 and 0–1.0 in w.g. It is very useful in reading pressure differences across filters and between two points in a duct. The unit is maintenance-free and requires adjustment only of the pointer zero setting before use. A unit of this type is illustrated in Figure 7.2(c) along with schematic diagrams of typical applications.

7.3 Air Velocity Measurements

a. Pitot Tube and Manometer

The simplest and most common field technique for measuring air flow is by using a Pitot tube and a manometer. The theory behind this measurement is given in Section 5.4 and Figure 5.5 and is repeated briefly here for convenience. The total pressure in a duct, P_T is the sum of the static (spring) pressure P_S and the velocity pressure P_V . That is,

$$P_T = P_T + P_V$$

Since we also know that

$$P_V = (V/4005)^2 \quad (5.6)$$

it follows that air velocity

$$V = 4005 \sqrt{P_V} \quad (7.1)$$

Therefore, by measuring velocity pressure, we can calculate air velocity very accurately. The method is shown in Figure 5.5(c) using two probes: a duct wall type that measures static pressure and a probe tip that measures total pressure. Connecting them as shown there to measure the differential pressure gives the velocity pressure directly since

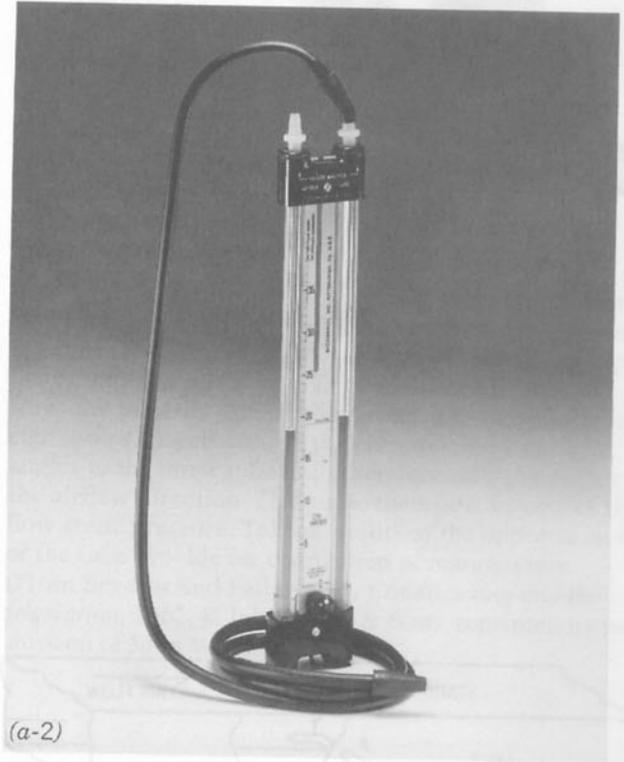
$$P_V = P_T - P_S$$

The same differential pressure measurement can be made with a single hole in a duct or pipe, by using a Pitot tube. See Figure 7.3(a). A Pitot tube is simply two concentric tubes. The inner tube has an opening that is placed facing the air stream. It measures total pressure P_T as shown in Figures 5.5(b) and 7.3(b). The outer tube has holes drilled into the circumference of the tube. These holes are, therefore, at right angles to the air stream. They measure static pressure P_S as shown in Figures 5.5(a) and 7.3(b). When the two output points are connected to opposite ends of a manometer, it reads the difference between total and static pressure, that is, velocity pressure P_V . See Figures 5.5.



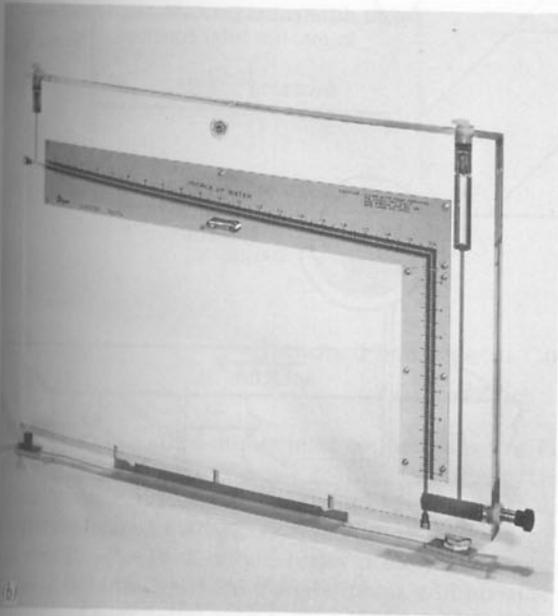
(a-1)

Figure 7.2 (a-1) U-Type manometer made with flexible tubing for carrying convenience. The tube can be filled with colored water or mercury. Various models in this design have a pressure measuring range up to 60 in. w.g. (Courtesy of Dwyer Instruments, Inc.)



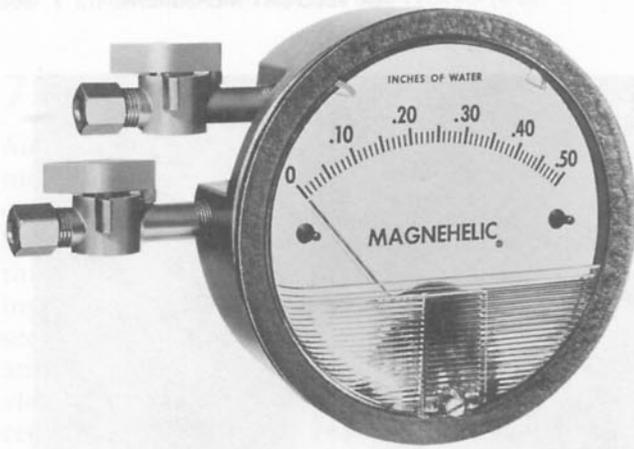
(a-2)

Figure 7.2 (a-2) Standard U-tube manometer calibrated in centimeters of water for metric calculations. (Courtesy of Bacharach, Inc.)



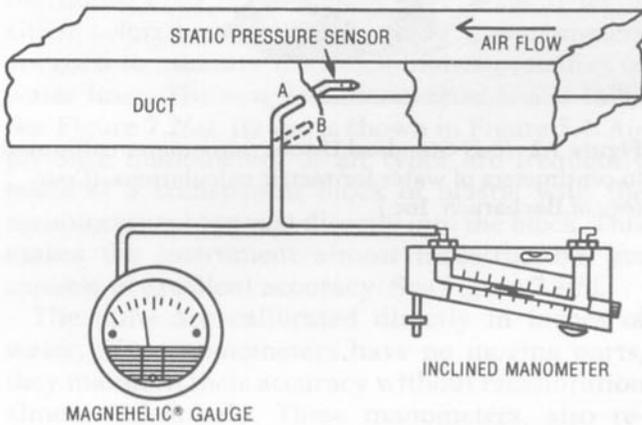
b)

Figure 7.2 (b) Inclined/vertical manometer. The inclined portion is used for pressures up to 2 in. of water and can be read to an accuracy of $\pm 1/4\%$. The vertical section is graduated from 2 to 10 in. w.g. The entire manometer is cast into a thick block of clear acrylic plastic measuring 16 \times 25 in. (Courtesy of Dwyer Instruments, Inc.)



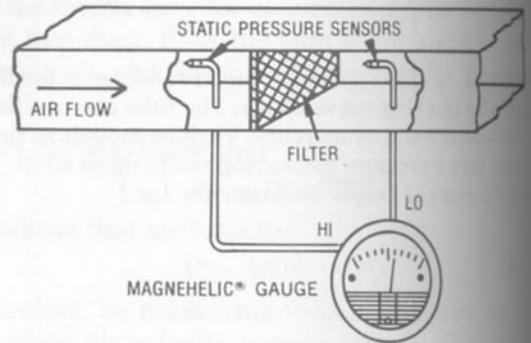
(c-1)

Figure 7.2 (c-1) Magnetic differential pressure meter measures the difference in pressure between lines connected to its two outlets. When the low pressure connection is left open as in (c-2), the meter will read gauge pressure, that is, pressure above atmospheric pressure. The inclined manometer shown in (c-2) is simply another means of measuring the same gauge pressure. (c-3) Velocity pressure is measured with a magnetic gauge by connecting the center tube of a Pitot probe to the high pressure inlet and the outer tube to the low pressure inlet. The difference is velocity pressure. The same measurement technique using an inclined tube manometer is shown for information only. See also Figure 7.4. (c-4) A differential pressure gauge can be used to measure directly the pressure drop across a filter, when set up as shown. Any appreciable change in this reading indicates a change in the condition of the filter. (c-5) Closure of the duct damper will cause an immediate increase in the upstream pressure and will indicate on the meter. This connection is useful to monitor and check the operation of fire dampers. (Courtesy of Dwyer Instruments, Inc.)



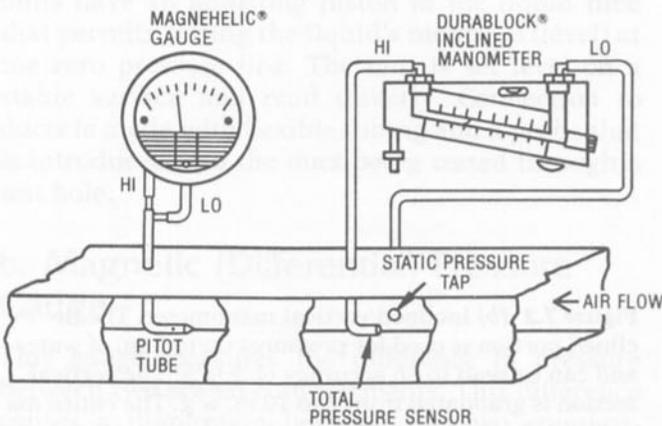
Measuring static pressure in an air duct or plenum.

(c-2)



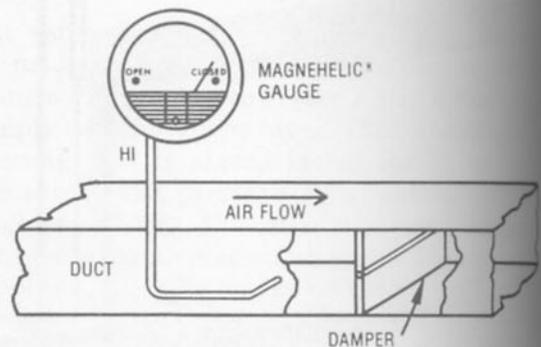
Dwyer differential pressure gauges used to monitor filter condition.

(c-4)



Differential pressure gauge used to measure velocity pressure.

(c-3)



Differential pressure gauge used for pressure sensing.

(c-5)

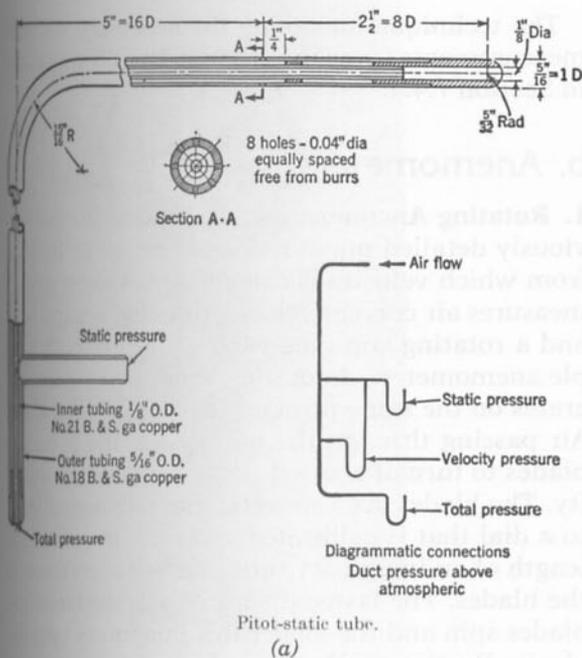


Figure 7.3 (a) Construction details of a standard Pitot tube. The tube consists of two concentric tubes. The inner tube terminates in a "nose" opening, which is placed facing into the air stream. This tube, therefore, measures the total air pressure in a duct. The outer tube has eight holes spaced around the circumference at right angles to the inner tube and, therefore, at right angles to the airflow direction. This tube, therefore, measures the flow static pressure. Takeoff points at the opposite end of the tube provide for connection of manometers. (From Severns and Fellows, *Air Conditioning and Refrigeration*, 1962, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)

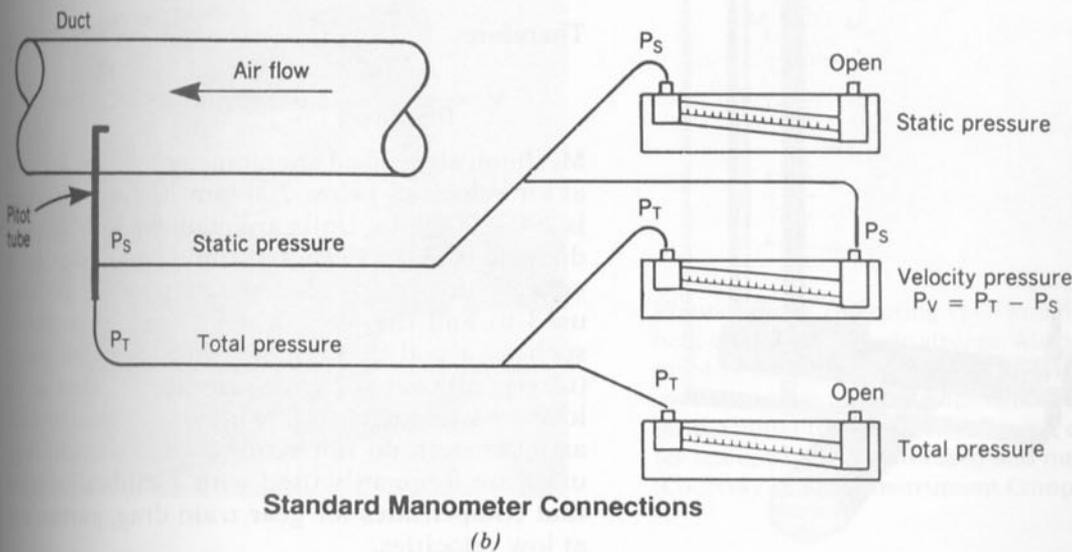


Figure 7.3 (b) Standard manometer connections to a Pitot tube probe will give static, velocity and total pressure readings as shown. Use of three manometers simultaneously permits measuring all three pressures without reconnecting. Elimination of the center manometer will still permit measurement of all three pressure quantities, since $P_v = P_t - P_s$. If only velocity pressure is required, a single manometer connected as shown for the center manometer will provide the desired measurement. (Reproduced with permission from SMACNA HVAC Systems, Testing, Adjusting and Balancing, 1983.)

and 7.4. Although only one ordinary manometer is required to take all readings, in practice two are frequently used to save time, since a series of readings must be taken over the cross section of a duct. The third reading is easily obtained from the equation $P_T = P_S + P_V$. A manometer graduated in velocity as well as pressure is shown in Figure 7.4. It is connected as shown for the center manometer in Figure 7.3(b).

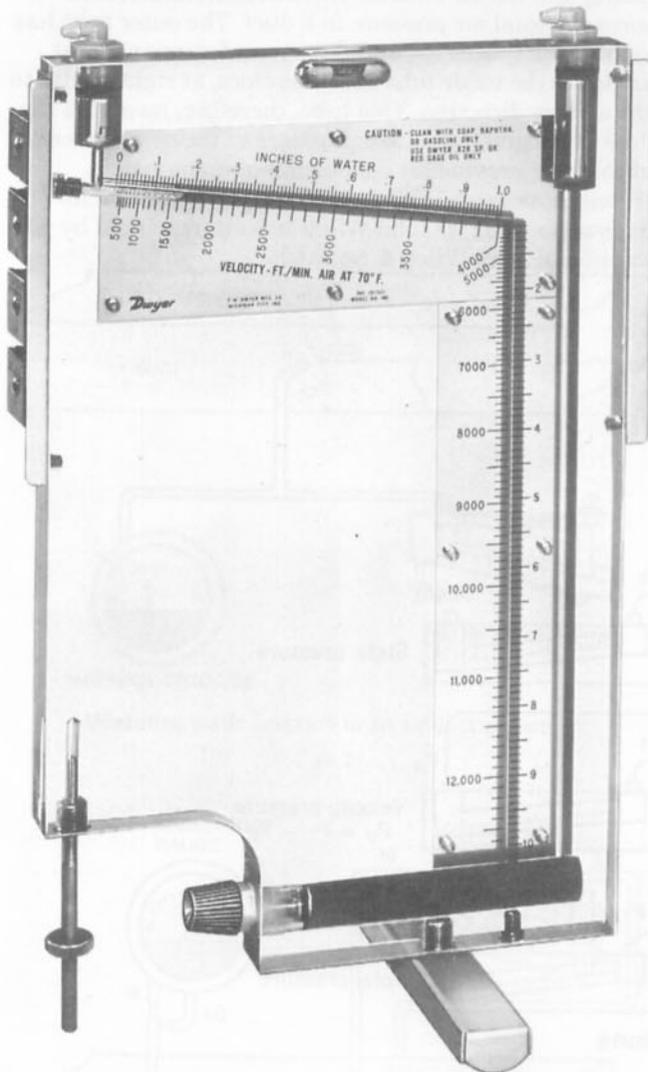


Figure 7.4 Combination inclined/vertical manometer and air velocity meter. The unit, which measures $16\frac{1}{2} \times 11$ in., is encased in a block of clear acrylic plastic, making it ideal for field work. When used with a Pitot tube and connected as shown in Figure 7.3(b), it will measure pressure from 0 to 10 in. w.g. and air velocity from 400 to 12,000 fpm. (Courtesy of Dwyer Instrument, Inc.)

The technique for taking the necessary traverse measurements (over the cross section) is explained in Section 7.4.a.

b. Anemometers

1. Rotating Anemometers. In contrast to the previously detailed pressure measurement technique from which velocity is calculated, an anemometer measures air current velocity directly. A windmill and a rotating cup-type wind gauge are both simple anemometers. A rotating vane anemometer operates on the same principle as a child's pinwheel. Air passing through the unit causes the propeller blades to turn at a speed proportional to air velocity. The blades are connected through a gear train to a dial that is calibrated in feet. It measures the length of an imaginary tube of air passing through the blades. The faster the air blows, the faster the blades spin and the longer this imaginary cylinder of air. By timing the flow, for a $\frac{1}{2}$ minute or a minute, the air velocity is easily calculated.

For instance, a dial reading of 600 ft timed in 1 min means an air velocity of 600 ft/min. Similarly, a dial reading of 600 ft in $\frac{1}{2}$ min means a velocity of 1200 fpm, since

$$\text{Velocity} = \frac{\text{Distance}}{\text{Time}}$$

Therefore,

$$V = \frac{600 \text{ ft}}{0.5 \text{ min}} = 1200 \text{ ft/min or } 1200 \text{ fpm}$$

Mechanically linked anemometers are not accurate at air velocities below 200 fpm. Their useful range is 200–2000 fpm. Units are made in 3-, 4- and 6-in. diameters. A modern self-timing unit that reads velocity directly is shown in Figure 7.5(a). When used to find the air velocity over a large surface such as a coil or filter, the unit must be moved (traversed) over the entire surface, because air velocities vary over these surfaces. Although these anemometers do not require recalibration, older units are frequently used with a calibration curve that compensates for gear train drag, particularly at low velocities.

2. Deflecting Vane Anemometer. A deflecting vane anemometer is illustrated in Figure 7.5(b). This unit contains a movable vane that is deflected (pushed) by the air current. The amount of deflection is proportional to the air speed. The vane is connected to a pointer that reads air speed directly on its scale. This device is not highly accurate, but



Figure 7.5 (a) A modern rotating vane anemometer with built-in timer that averages air flow every few seconds. This permits direct reading of average air velocity without external timer and calculator. It also permits rapid sweeping of large area grilles and registers. The rotating vane shown transmits its rotational speed electrically. This maintains accuracy over the entire range and eliminates the need for calibration curves at low air speeds. The illustrated unit has a useful range of 50–6000 fpm. It can also be used to measure volumetric flow rate. (Courtesy of Alnor Instrument Company.)



Figure 7.5 (b) Deflecting vane anemometer. The unit is held directly in the air stream. Air pressure causes the pivoted vane in the meter to deflect, moving the pointer over the meter face. The amount of deflection, which is proportional to air velocity, is read directly on the meter scale, which is calibrated and marked in air velocity. (Courtesy of Alnor Instrument Company.)



Figure 7.5 (c) Modern digital thermal anemometer and associated microprinter. The unit is autoranging with a total velocity range of 20–3000 fpm. Accuracy is $\pm 3\%$ or better on all scales. Automatic averaging of readings permits rapid scanning of grilles, registers and other surfaces with variable air volumes. The unit is also arranged to read air temperature over a range of 0–70°C (32–158°F). (Courtesy of Alnor Instrument Company.)



Figure 7.5 (d) This thermal anemometer is a multifunction instrument capable of measuring, storing and printing (with the illustrated printer) air velocity, volumetric flow, temperature and relative humidity, by use of different probes. The air velocity probe gives the unit a range of 20–6000 fpm and 0–50°C (32–122°F) in various ranges and accuracies. The RH probe measures relative humidity from 0 to 100% and temperatures from 0 to 60°C (32–140°F). (Courtesy of Alnor Instrument Company.)

it will quickly give an approximate air velocity reading. When used to measure air velocity over a large surface, a "profile" or traverse must be made, and the readings, averaged. It is most useful for measuring velocity over a small (3-in. square) specific area. Meters are available in single and multiple ranges from 0 to 3000 fpm.

3. Thermal Anemometers. A third type of anemometer operates on the principle that the electrical resistance of a hot wire will change with temperature. If air is passed over such a wire, it will be cooled in proportion to air velocity. This instrument is called, logically, a *hot wire anemometer*. The wire resistance is measured in an extremely sensitive electrical bridge circuit, making the instrument highly sensitive although not very accurate ($\pm 10\%$). It is, therefore, particularly useful in detecting low velocity air movement such as leaks or drafts.

Modern units that operate on the principle of measurement of the cooling effect of moving air are known by the more general name of *thermal*

anemometers. These units may use the traditional "hot wire," known today as a resistance temperature device (RTD). Alternatively, they can use a thermistor sensor or sensitive thermocouple junction. The principle of operation, however, remains unchanged. Two modern, electronic, digital read-out units of this type are shown in Figure 7.5(c) and (d). In addition to their primary function as anemometers, they are also usable as thermometers to measure the air current temperature.

4. Velometer. See Figure 7.6. A velometer is an anemometer that operates on the principle of a swinging vane. Sampled air passes through a Pitot tube-type circular tunnel in which the vane is mounted. The vane motion and the corresponding pointer motion are proportional to air velocity. Unlike many of the instruments discussed, and especially modern digital units, the velometer is a purely mechanical instrument. Nevertheless, it is highly accurate and is very widely used for TAB work. The meter has scales of 0–300, 0–1250, 0–2500, 0–5000 and 0–10,000 fpm. Three probes are

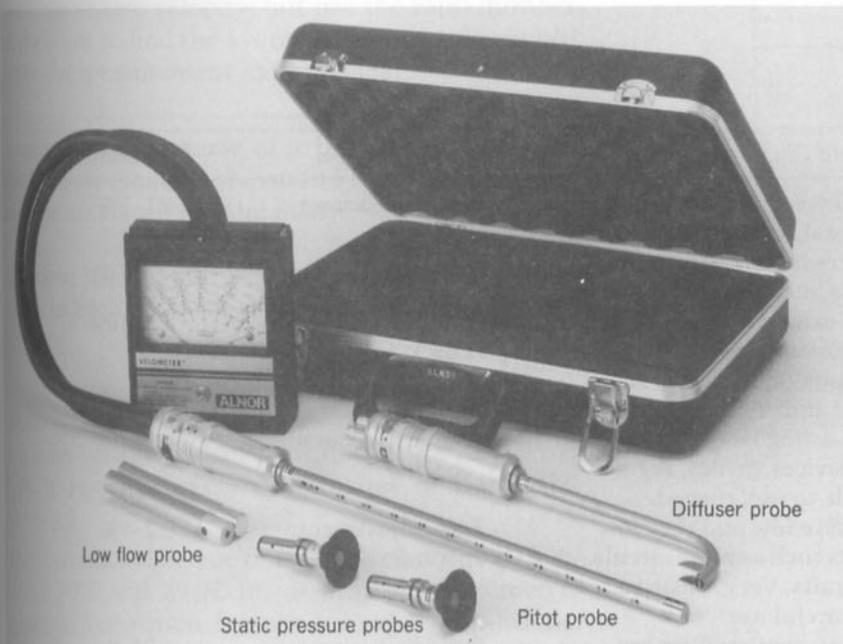


Figure 7.6 Velometer kit includes low flow probe for velocities up to 300 fpm, Pitot probe for measuring air velocities in ducts; diffuser probe for measuring air velocity at diffusers, registers and grilles; and two static pressure probes. These components are sufficient to perform a complete balancing procedure for an all-air system. (Courtesy of Alnor Instrument Company.)

provided with the meter: a low velocity probe useful for measuring in-room terminal velocities, a Pitot tube-type probe for medium air velocities as in ducts and a high velocity probe. The instrument will also measure static pressure when used with a static pressure probe. The velometer should have periodic accuracy checks, although it maintains its calibration for extended periods, depending on usage. Accuracy of readings depends on the scale and is normally better than $\pm 2\%$. Table 7.1 summarizes the uses and characteristics of anemometers commonly used in HVAC work.

All the instruments that we have discussed for measuring air velocity do so over a very limited area. Pitot tubes and velometers measure velocity at a point. Vane-type anemometers measure velocity over an area equal to their face area. This varies between 3 and 30 in.². Furthermore, the air velocity in a duct or at a register is not constant. It varies over the cross section of a duct and over the face of a register or grille. Therefore, what is needed is a measuring technique, using the instruments just discussed, that will give up an average air velocity over the entire area of the item being measured. Whether the averaging is done manually by the TAB technique or automatically by the instrument is not important. These measuring procedures are discussed in Section 7.4.

Measurements

7.4 Velocity Measurement Techniques

Air velocity in a duct varies with position in a duct. It is slower near the duct walls because of friction (drag). Therefore, to obtain average air flow velocity, it is necessary to perform a traverse or profile over the cross section of the duct. In order to make the air flow as linear as possible (without turbulence) and to make the measurements as accurate as possible, a good TAB technician will perform the following before taking any readings.

- Insert an egg-crate type of flow straightener (or other type) into the duct at least five duct diameters (or duct diagonals for rectangular duct) upstream of the Pitot tube entry.
- Perform the test in a straight section of duct, as far as possible from elbows, fittings of all types, size changes and the like. Minimum distances should be eight diameters upstream and two diameters downstream from the Pitot tube.

Table 7.1 Anemometers

Type	Use and Characteristics	Calibration	Accuracy
Manometer U, vertical, inclined	Use with probes and Pitot tube to measure total, static and velocity pressures in ducts and across filters, coils, etc. Very rugged.	Zero adjustment only	$\pm 1-5\%$
Rotating vane	Supply and exhaust air velocity and flow measurement. Also useful for terminal device face velocity. Simple and rugged.	Periodic accuracy check	$\pm 3-10\%$
Deflecting vane	Use for measuring face velocity of terminal devices, grilles, registers. Simple to use, rugged.	Frequent accuracy check	$\pm 5-10\%$
Hot-wire	Use to measure low and very low air currents such as room circulation and drafts. Very sensitive. Requires careful use.	Zero adjustment; periodic accuracy check	$\pm 2-5\%$
Velometer	All types of air motion measurements; duct (with probe), face velocity, supply and exhaust air velocity. Requires careful use, in accordance with manufacturer's recommendations.	Periodic accuracy checks are recommended	$\pm 2\%$ depending on range

- Duct diameter should be at least 30 times the diameter of the Pitot tube.
- Minimum duct dimension for a rectangular duct should be at least 30 times the diameter of the Pitot tube.

a. Rectangular Duct Traverse

If the minimum duct dimension is not at least 8 in., use Figure 7.7(a). For larger ducts, use Figure 7.7(b). Divide the cross section evenly into rectangular areas, not less than 16 and not more than 64. Minimum dimensions of a single test area should be 3 in. square. (In small ducts such as 10 × 14 in., it will be necessary to use a 2½-in. dimension for one side of a test rectangle in order to have a minimum of 16 readings). Position the Pitot tube carefully at the center of each test rectangle and take a velocity pressure reading. Number each test area and record the readings on a chart or on a numbered tabulation. Do not average the pressure readings. Convert all pressure readings to velocity, and then take the average. This is then the average air velocity in the duct.

If a duct air flow straightener cannot be used or if a fitting is close by, considerable turbulence will be present in the air stream. This may result in negative readings in the traverse. Record these readings as zero velocity, but use the total number of divisions to find the average. An example should make the measurement method clear.

Example 7.1 A traverse of a 10 × 18-in. duct gives the pressure readings shown in Figure 7.8. Find the average air velocity in the duct.

Solution: The pressures are tabulated, and the velocity in each test area is calculated using the expression

$$V = 4005 \sqrt{P_v}$$

where

V is air velocity in feet per minute and

P_v is velocity pressure in inches of water (in. w.g.).

This formula holds true only for dry air at 0.075 lb/ft³, at 70°F and 29.92 in. of mercury barometric pressure. Correction factors for other conditions, particularly humidity and altitude (atmospheric pressure), can be found in the manufacturer's literature that accompanies the Pitot tube and the manometer. After calculating all velocities, an arithmetic average is taken. This then is the duct air velocity. It is shown on Figure 7.8.

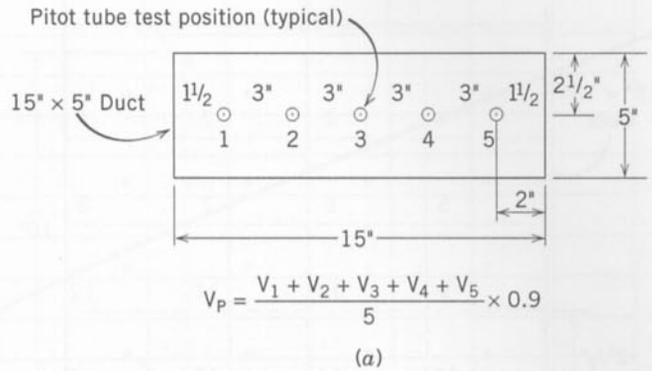


Figure 7.7 (a) For narrow ducts, it is sufficient to make a linear traverse with a Pitot tube, along the center line. Each position should represent the same proportion of total duct area. In this case the area is 3 × 5 in. The calculated average velocity must be multiplied by an arbitrary factor of about 0.9, to compensate for lower air velocity at the duct walls.

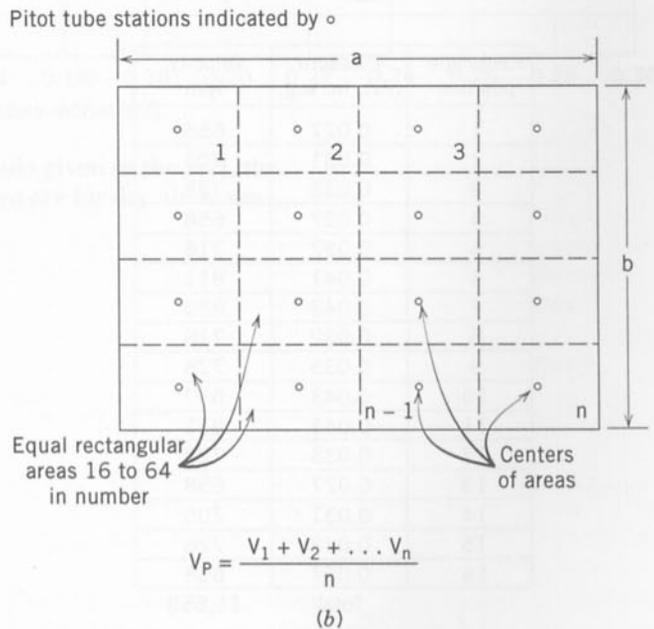


Figure 7.7 (b) Rectangular ducts are divided into 16–64 equal areas for a Pitot tube traverse. The average velocity is the arithmetic average of the individual area velocities. Accuracy increases as the size of individual rectangles decreases. Sides of measurement rectangles should be between 2½ and 6 in. Rectangles should be as nearly square as possible. (See also Appendix D, Form D.7.)

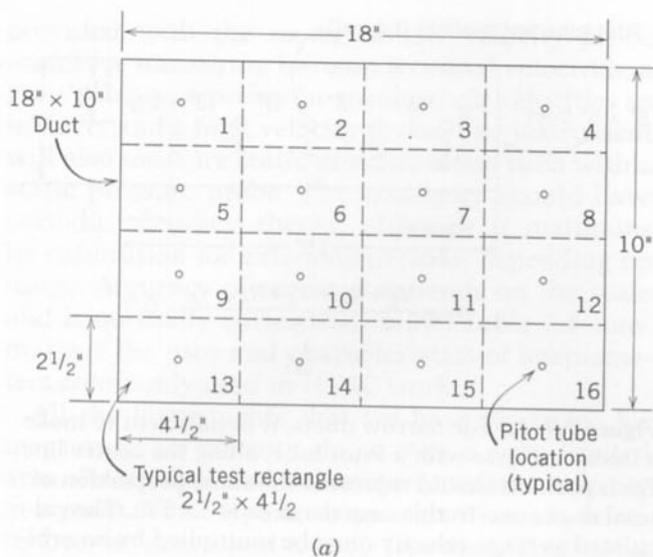


Figure 7.8 (a) Solution to Example 7.1. (a) The 18 × 10-in. duct is divided into 16 equal area sections, each measuring 2½ × 4½ in. A measurement of velocity pressure is taken with a Pitot tube at the center of each rectangle and recorded in table (b).

Rectangle position	Pressure (Pv), in. w.g.	Velocity, fpm
1	0.027	658
2	0.031	705
3	0.033	728
4	0.027	658
5	0.032	716
6	0.041	811
7	0.043	830
8	0.032	716
9	0.033	728
10	0.043	830
11	0.041	811
12	0.033	728
13	0.027	658
14	0.031	705
15	0.033	728
16	0.027	658

Total 11,558

$$\text{AVG } V = \frac{11558}{16} = 722 \text{ fpm}$$

(b)

Figure 7.8 (b) The tabulated pressures are converted to velocity using the relationship given in the text. The 16 velocities are then averaged arithmetically to give the overall average duct air velocity.

Manufacturers also publish tables and distribute slide rules that will perform the required calculation and make any corrections for air at other conditions. One such curve for dry air at 70°F is given in Figure 7.9.

b. Circular Duct Traverse

A traverse in a circular duct is done following the same principles. Readings are taken on two diameters, at right angles to each other. Since we want each reading to represent the same (annular) area, the test points get closer together as they proceed from the center outward. In very small ducts, say 3–4 in., a single reading at the duct center, multiplied by 0.9 to account for low peripheral velocity, will give a usable velocity figure. In ducts from 6 to 9 in., take six readings across. For ducts 10 and 12 in. in diameter, use eight readings. For all larger diameters, use ten readings. The positioning of Pitot tube points for these ducts is shown in Figure 7.10.

c. Face Velocity of a Register

Since the velocity of air exiting from a register is not uniform over the register face area, a traverse of some type must be made, and the readings averaged. When using a vane type anemometer, it is placed against the face of the register and covers a certain area depending on its size. This procedure should be repeated over the entire face of the register, taking care not to measure the same face area twice. The resultant arithmetic average is usable as the device's face velocity. When using a point-type anemometer such as a velometer, a traverse of the type shown in Figure 7.8 should be made. No less than 12 readings should be taken. For large registers, up to 48 point readings can be taken and averaged. Return grill air flow is generally more uniform over its face area, and a smaller number of measurements is possible.

7.5 Air Flow Measurement

Field measurement of air flow can be accomplished by two methods: direct measurement and calculation.

a. Direct Measurement

See Figure 7.11. The illustrated device is used by placing it over a ceiling or wall register or grille so

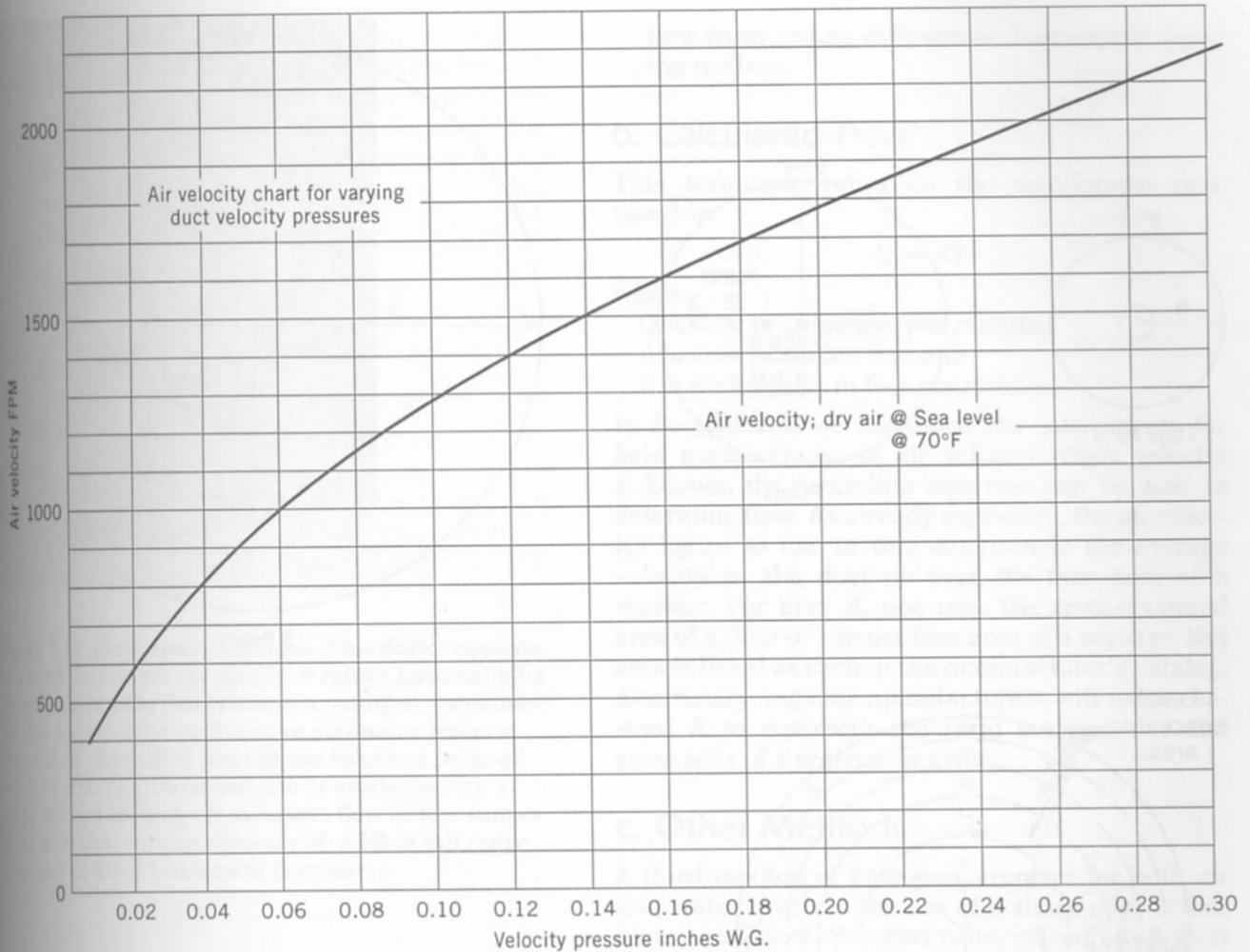
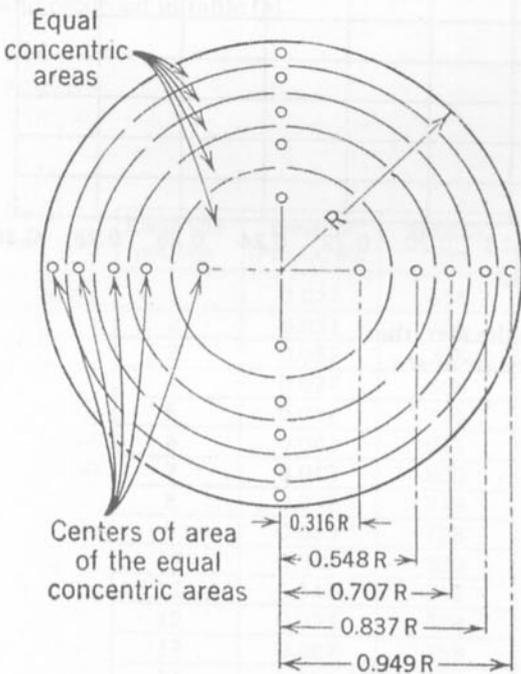
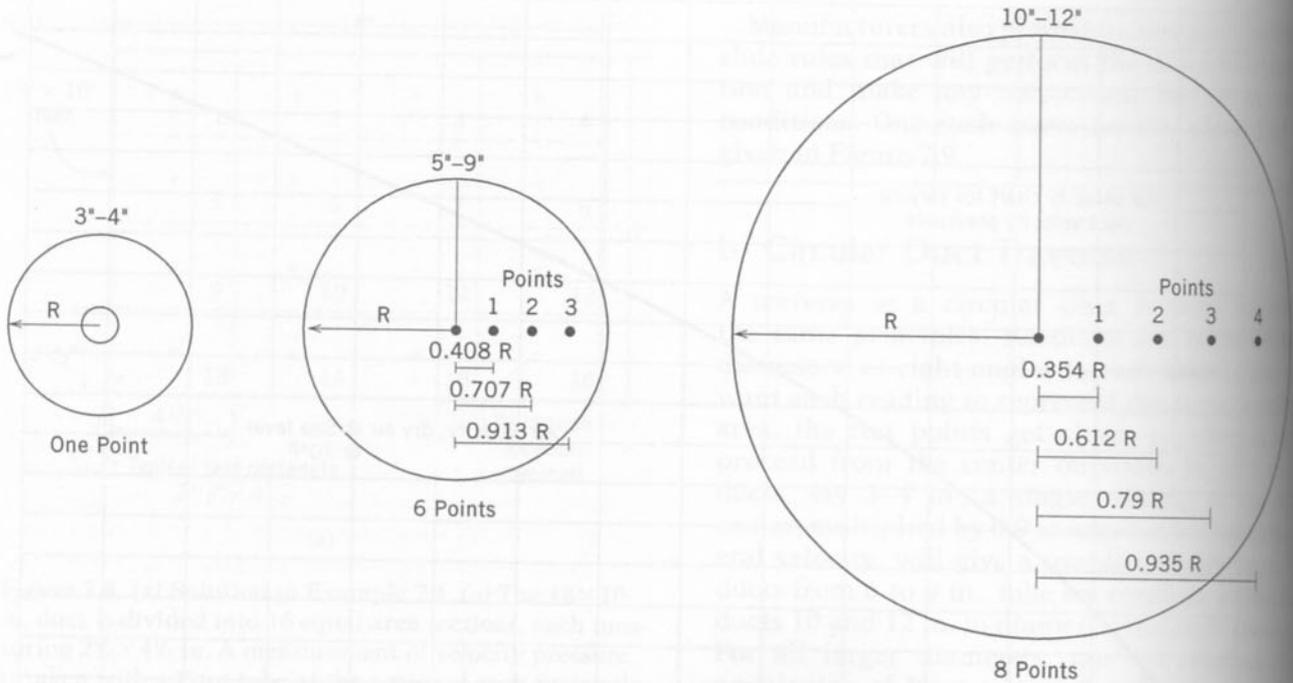


Figure 7.9 Instead of calculating air velocity using the formula given in the text, the velocity can be picked off the chart directly. The values shown are for dry air at sea level and 70°F.



Locations of a Pitot tube for a 10-point duct traverse

Figure 7.10 Traverse points on round ducts. For 3- and 4-in. ducts, a single measurement in the center will give a satisfactory velocity when multiplied by 0.9. For 5- to 9-in. ducts, two 6-point traverses at right angles (one horizontal and one vertical) are required. For 10- and 12-in. diameter ducts, use two 8-point traverses; for larger ducts, use two 10-point traverses on diameters at right angles to each other. The spacing shown for test points will ensure that each test point represents the same percentage of the total cross-sectional area. (Ten-point traverse diagram from *Severns and Fellows, Air Conditioning and Refrigeration*, 1962, © John Wiley & Sons, reprinted by permission of John Wiley & Sons.)



Figure 7.11 Flow-measuring hood. This device consists of a hood that covers the supply or return terminal plus an instrument base that contains a modified anemometer. The anemometer performs an automatic traverse over the air channelled through the base and reads directly in cfm. The illustrated unit is available with a range of hood openings. It measures flow in four ranges, up to 2000 cfm, with an accuracy of $\pm 3\%$ of full range. (Courtesy of Alnor Instrument Company.)

that the entire supply or return terminal is covered. Air flow is channelled through the base that is instrumented with a modified anemometer. The anemometer samples air velocity at 16 points over its area, determines average velocity and converts this to air flow quantity, which is then indicated on the instrument's meter. In effect, this device performs automatically the traverse that is described in Section 7.4.c. These flow-measuring hoods have several limitations.

1. They lose accuracy as register velocities increase and should not be used for velocities over 2000 fpm.
2. The hood and its instrumented base must be held manually over the register or grille. Care must be taken that the entire air supply device is covered, with no leakage. This is frequently difficult when measuring large area devices because of the bulk and weight of the instrument. This is particularly true when measuring air

flow from ceiling diffusers and air-supply lighting troffers.

b. Calculated Flow

This technique relies on the well-known relationship

$$Q = AV$$

where

- Q is flow in cubic feet per minute,
- A is area in square feet and
- V is air velocity in feet per minute.

In Section 7.4, we discussed the instruments for field measurement of air velocity. Once velocity is known, the preceding equation can be used to determine flow. As already explained, the air velocity figure to use in this equation is the average velocity in the duct or over the face area of a register. For area A , one uses the cross-sectional area of a duct or the net face area of a register. Net area is listed as such in the manufacturer's catalog. Alternately, register manufacturers will use a constant K to represent the ratio between net and gross area of a register or grille.

c. Other Methods

A third method of flow measurement for both air and water involves the use of a sharp edge orifice plate or a smooth Venturi tube, placed into a duct or pipe. Pressure measurements made on both sides of either of these devices can be related to flow by a series of calculations and graphic plots. These are complex, advanced techniques that are beyond our scope here. Refer to the bibliography at the end of this chapter for more information.

Balancing Procedures

7.6 Preparation for Balancing an Air System

Before starting a TAB procedure, a number of preliminary steps are advisable. They will help to make the actual TAB work smooth, rapid, accurate and efficient. They are:

- (a) Obtain a complete set of as-built HVAC drawings. These will be either contractor prepared field drawings or as-built-corrected contract drawings. In addition, shop drawings for

equipment and ductwork must be readily available. If drawings for any part of the system do not show as-built conditions, prepare a simple single-line drawing showing all equipment and outlets.

- (b) Mark on these drawings design air velocities and flow rates for each duct and outlet.
- (c) At each fan or blower, mark design cfm, rpm, pressure and motor data including running current. Show speed controls and interlocking.
- (d) For each filter, show type, cfm, pressure loss, area and air flow.
- (e) For coils, show pressure drop, cfm, area, temperatures and capacity.
- (f) Show location and type of all dampers.
- (g) Record any special equipment information that will be checked during the TAB procedure.
- (h) Prepare TAB report forms for recording test data. (A few sample forms are given in Appendix D.)
- (i) Select the instruments that will be needed for all tests.
- (j) Mark on the drawings where all measurements will be taken. If special access fittings are required, such as those for Pitot tubes, make sure that they are in place.
- (k) Check with the field inspector that all systems are operative including all controls. The field inspector should also have the required data on all damper positions. If not, these must be ascertained before any TAB work can begin.
- (l) Coordinate the TAB work with the contractor. It is necessary to have a contractor's representative available during TAB work. A TAB technologist is authorized to perform testing and balancing only. Any procedures, work or changes required to accomplish this TAB work must be performed by the HVAC contractor. This includes starting and operating all systems in all the design modes for which they are intended. Actual operation of the equipment by the TAB technologist can create problems of responsibility for malfunctioning. This is because TAB work is almost always performed before the system is turned over to the owner, that is, while the contractor is still fully responsible.

7.7 Balancing an Air System

The actual balancing procedure can be very complex if the system is large. Before going out into the field, the TAB technologist must plan the work

precisely. Large systems are always made up of subsystems. Proper TAB procedure would be first to balance subsystems that are independent of the overall system. This demands a complete understanding of, and familiarity with, the design. If anything is unclear, check the design intent with the project engineer. Out on the job site, the TAB technologist is expected to know exactly what he or she wants to do, how to do it and what the results are supposed to be. A brief listing of the TAB procedure in the field follows. In some jobs, several steps can be combined or done in a different order. Remember that client satisfaction depends on an adequate TAB job.

- (a) Check that all the preparatory steps listed in the preceding section have been taken.
- (b) Turn on all fans. Measure fan speeds and adjust to design values. Check motor running-current. If running current is above or more than 10% below the design value, shut down the fan until the cause is determined.
- (c) Measure and record initial cfm at supply fans. A Pitot tube traverse is the preferred method. If this is not possible for some reason, use anemometer readings across coils in the air-handling units. The cfm must be within $\pm 10\%$ of the design value before proceeding with the next step. Adjustments of fan cfm is normally made by adjustment of the drive speed. Rotational speeds are most easily checked with a simple hand-held tachometer.

An air quantity of more than $\pm 10\%$ from the design value indicates one of the following problems. They should be checked in the order listed:

- Incorrect damper positions—probably closed
 - Incorrect filter
 - Equipment malfunction
 - Incorrect installation (This should not be possible if proper inspection of the installation has been performed. The job inspector should be called in, if this seems to be the problem.)
 - Incorrect design (Consultation with the design engineer is required.)
- (d) Measure the flow (cfm) in major duct branches and adjust to within $\pm 10\%$ of design. Adjustment is normally made with splitter dampers. Dampers should be fixed in position, and the positions marked.
 - (e) Measure and adjust air flow to all air outlets to $\pm 10\%$ of design requirement. Some TAB technicians start at the last outlet (farthest

from the fan), and some start at the first outlet (nearest to the fan). Our recommendation is to use the latter method; it seems to require less readjustment. Use a velometer or vane anemometer to measure outlet air velocity. Take profile (traverse) readings to arrive at average air velocity. Calculate cfm using average velocity and net face areas of outlets. Record all flows on the appropriate TAB form.

- Measure and adjust the cfm in multiple outlet branches before adjusting the flow at each outlet.
 - Measure and adjust air flow at all air outlets.
- (f) After all terminal outlets are adjusted, repeat the entire procedure. This is necessary because each adjustment affects the pressures and flow in the entire system. As a result, the quantities previously measured in main and branch ducts will have changed. Keep repeating the procedure until flow readings remain the same when remeasured. Record the velocity and flow at each outlet for each round of adjustments. The TAB form should contain space for three sets of entries. It should not be necessary to repeat the sequence of measurements more than twice. In small systems, one repetition is frequently sufficient.
- (g) Measure and record performance of all equipment. This includes:
- Static pressure at fans, filters and coils
 - Motor currents
 - All motor and fan speeds
- (h) Measure and record WB and DB temperature at all coils along with the load condition. It may not be possible to operate the equipment at design loads. If this is so, record the operating conditions (partial load). This will enable the project engineer to determine, using the manufacturers' published data, whether the equipment is operating correctly at part load. It should also then be possible to extrapolate, to determine if the equipment will operate satisfactorily at full load.
- (i) Perform air velocity and flow checks on the return air system.

The preceding description of TAB procedures for all air systems is brief but covers all the important aspects of the work. In practice, an experienced TAB technologist will make a quick survey after the system is up and running during which he or she will detect any major deviations from the desired operating conditions. These are usually not

malfunctions. Instead, simple oversights, such as an open window, door or duct access panel or a blocked return grille or duct, can play havoc with system pressures and cfm quantities. This ability to locate trouble spots quickly comes with experience and a sharp eye for detail. For the novice TAB technologist, a very detailed step-by-step procedure list is the best course to follow.

7.8 Preparation for Balancing a Hydronic System

Before beginning any TAB work, the following preparatory steps should be taken. Thorough preparation always results in time savings in the field.

- (a) On a set of as-built drawings, mark pressures, flow rates, temperatures and motor data. Clearly mark the actual field location of valves and other controls. Check with the field inspector on any special conditions that arose during construction. Familiarize yourself with the control system. A single-line diagram of the control schemes will be extremely helpful, particularly if the system has automatic controls and interlocks that are not field adjustable.
- (b) Prepare appropriate TAB test forms for field use. (See Appendix D.)
- (c) Mark on the drawings all points of measurements and the items to be measured. This will prevent anything from being overlooked. Have orifice plates and/or Venturi tubes installed at points where flow is to be read.
- (d) Coordinate the TAB work with the construction contractor. A contractor's representative must be available to perform all hands-on system operation.

7.9 Balancing a Hydronic System

Having accomplished the preparatory work just outlined, proceed with the actual balancing procedure as detailed next.

- (a) On a preliminary visit to the site with the field inspector and a contractor's representative, check that all systems, controls and safety devices are functional and that all hydronic systems have been drained, flushed, refilled and vented as required.

- (b) Before any testing begins, confirm that manual valves are open, controls are set in their proper operating position and any seasonal controls have been properly set. This may involve overriding some automatic controls, in order, for instance, to test a heating system in the summer or a cooling system in the winter.
- (c) With the pumps off, measure (and record) static pressure at each pump outlet.
- (d) Start all systems. Immediately check the operating currents of all motor-driven equipment. If motors are drawing excessive currents, shut down the system to determine the cause.
- (e) At each pump, perform the following test:
 - (1) With the pump discharge valve wide open, record the operating characteristics—flow, discharge and suction head, speed and electric motor data.
 - (2) Gradually close the discharge line valve to shutoff point. At several points during this closing procedure, take full measurements. Use one gauge to measure all pressures; this avoids introducing a metering error. Do not permit the pump to run with the discharge line closed for any length of time, because it may overheat. Using the data recorded, plot a pump characteristic, and compare it to the manufacturer's published data. If there is any significant difference, clarify the reason with the pump manufacturer.
 - (3) Gradually open the valve, take head and flow readings and check that they fall on the curve just plotted. If not, repeat these steps until an accurate pump curve is obtained. Record the total head and flow in full-open valve position. A total head higher than design means a maximum flow lower than design and vice versa. If flow is greater than design, close down the output valve until flow is about 110% of the design value. At this point, record pressures, flow and motor data. All these readings should be within system tolerances.
- (f) Some hydraulic systems use automatic balanc-

ing valves. For such systems, manual balancing of flow rates in mains and branches is unnecessary. Where manual balancing is to be done, adjust manual-balancing valves with all systems operating. Read flow rates at orifice plates and/or Venturi tubes that were installed previously. Water flow rates (as with air flow rates) within $\pm 10\%$ of design are considered to be on target. Using balancing valves, adjust flow rates to terminal units to $\pm 10\%$ of design.

Note: Keep in mind that flow rates in hydronic heating are not critical. Terminal units will deliver about 90% of their rated output with 50% flow, because the heat output of a hydronic terminal unit (radiator or baseboard) depends primarily on the difference between ambient air temperature and hot water temperature. Chilled water-cooling systems are not so forgiving with inaccuracies in liquid flow. There a drop in flow will cause a serious drop in cooling effect.

- (g) Repeat the balancing process for chillers, large coils and terminal units until the values remain unchanged. This may require two or three repetitions.
- (h) Make a final check of pump flow and pressures and of pump electrical data. Record this information. It represents the balanced system data and can be used in the future, if any parts of the system are repaired or replaced.
- (i) Mark and record the position of all valves and balancing cocks and the readings of all gauges and thermometers. This, too, is data for future reference.

We have not discussed the TAB work required on condenser water systems, cooling towers, large chillers, heat exchangers and other parts of large systems because they are beyond our scope here. Technologists will begin TAB work on small projects. After gaining experience with the design and field aspects of small systems, many will go on to similar work on large complex systems.

Key Terms

Having completed this chapter, you should be familiar with the following key terms. If any appear unfamiliar or not entirely clear, you should review the section in which these terms appear. All key terms are listed in the index to assist you in locating the relevant text.

Anemometers
Balancing
Bimetallic element
Bourdon gauge
Bourdon tube
Capillary tubing
Capillary tube thermometer
Deflecting vane anemometer
Dial thermometer
Differential pressure
Draft gauges
Flow straightener
Hot-wire anemometer

Magnetic pressure gauge
Manometers
Pitot tube
Pyrometers
Resistance temperature device (RTD)
Rotating vane anemometer
TAB
Thermal anemometers
Thermocouple
Traverse measurements
Velocity pressure
Velometer

Supplementary Reading

B. Stein and J. S. Reynolds, *Mechanical and Electrical Equipment for Buildings*, 8th ed., John Wiley & Sons, New York, 1992. This book covers the same areas of study as the present book, but in greater detail and scope. It is very useful for further study.

American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE)

1791 Tullie Circle, N.E.

Atlanta, GA 30329

Handbook—HVAC Applications, Chapter 34, 1991
Sheet Metal and Air Conditioning Contractors
National Association, Inc. (SMACNA)
8224 Old Courthouse Road
Tysons Corner, Vienna, VA 22180
HVAC Systems; Testing, Adjusting and Balancing,
1983

E. G. Pita, *Air Conditioning and Systems: An Energy Approach*, Chapter 16, John Wiley & Sons, New York, 1981.

KN 133

TYPICAL STORAGE	TEMPERATURE
ICE CREAM	-50°
FROZEN FOOD	-10°
MEATS	+33°
POULTRY	+33° TO 34°
DAIRY FOODS	+36°
PRODUCE	+56° TO 58°

GENERAL NOTES
DOORS

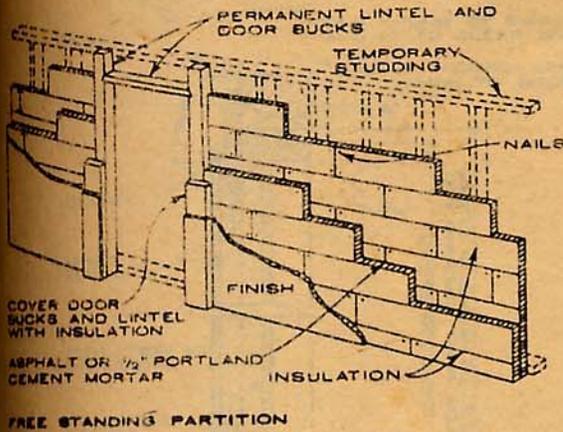
Insulation thicknesses vary with the box temperature, the type of insulation and conditions in surrounding areas and outside. In temperate climates cooler boxes at temperatures above 32° usually do not require floor insulation. Penetration of insulation by pipes, conduit, and hangers should be kept to an absolute minimum. Rods or pipes through ceiling insulation should be insulated 3'-0" above ceiling. Protection of insulation from damage from trucks and abrasion by stored goods is extremely important. Punctures in insulation finish allow moisture penetration with resulting drop in insulating efficiency and destruction of insulation structure.

INSULATION

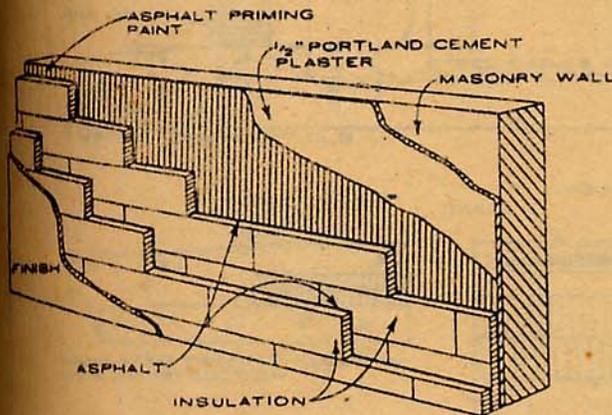
Refrigerator doors are available in a wide variety of types and finishes including sliding, overhead types and special vestibule doors to minimize refrigeration losses where long periods of opening will prevail. Consult manufacturers for door selection.

VENTILATION

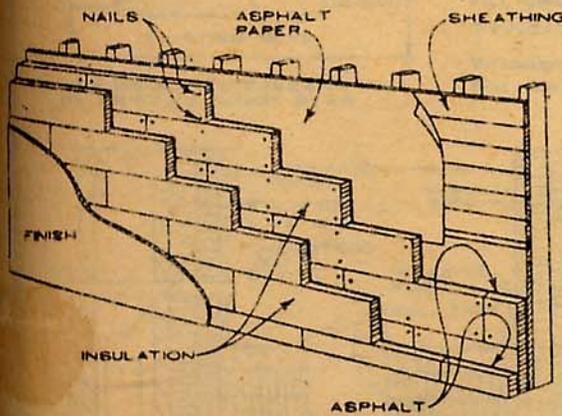
All spaces above suspended ceilings must be well ventilated. Freezers on slab on grade must be vented or heated below the slab.



FREE STANDING PARTITION

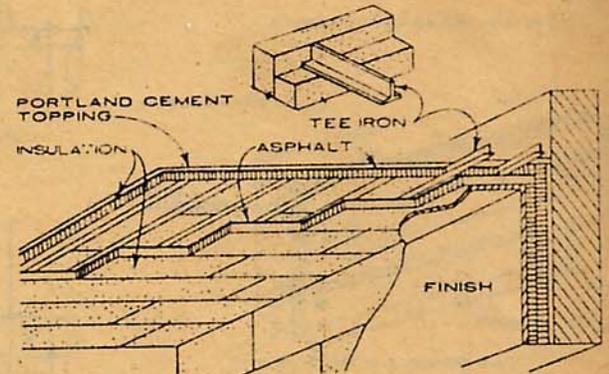


ON MASONRY WALLS

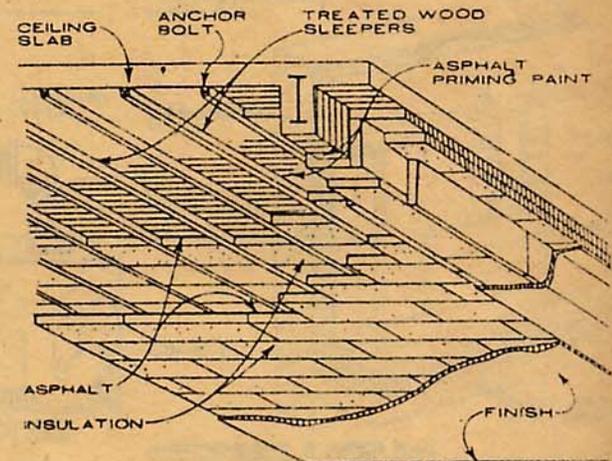


ON WOOD WALLS

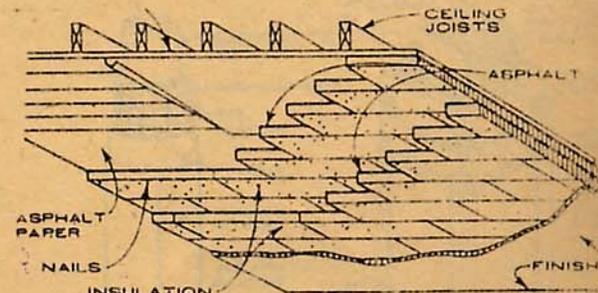
TYPICAL WALL DETAILS



ON SUSPENDED STEEL CONSTRUCTION

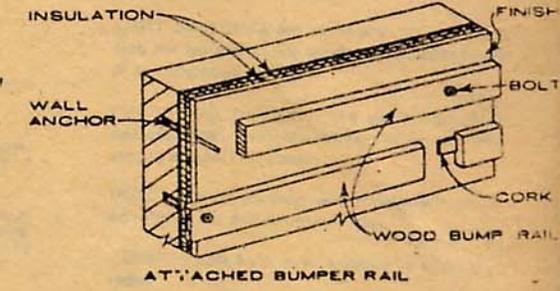
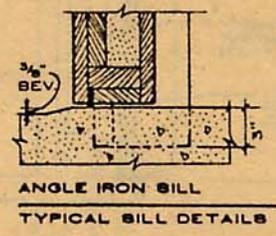
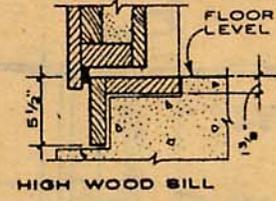
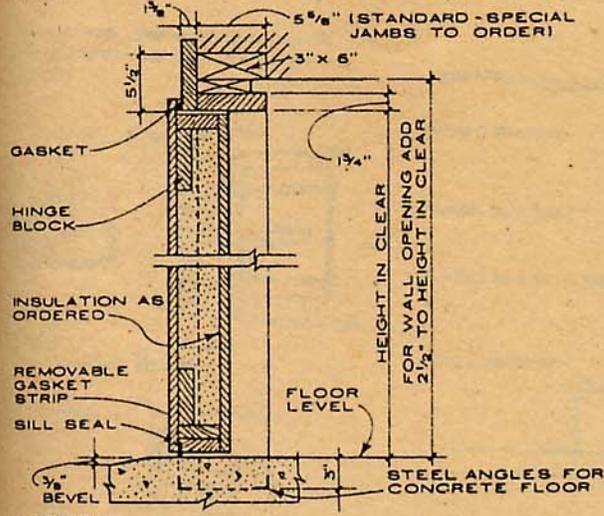
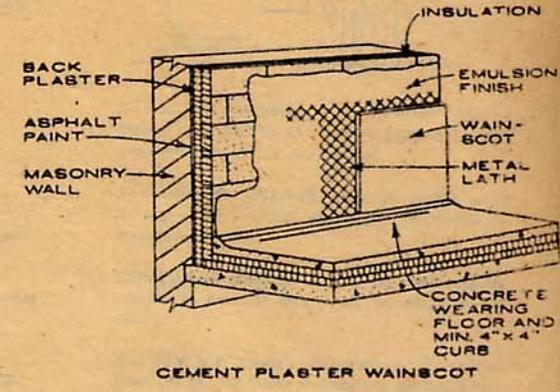
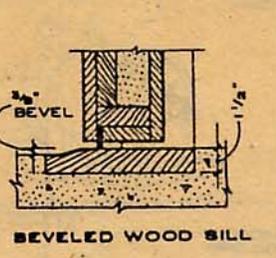
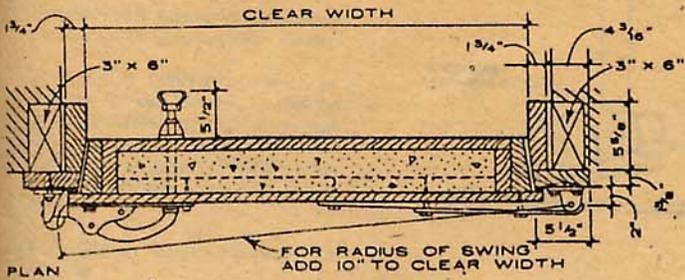


ON CONCRETE CONSTRUCTION



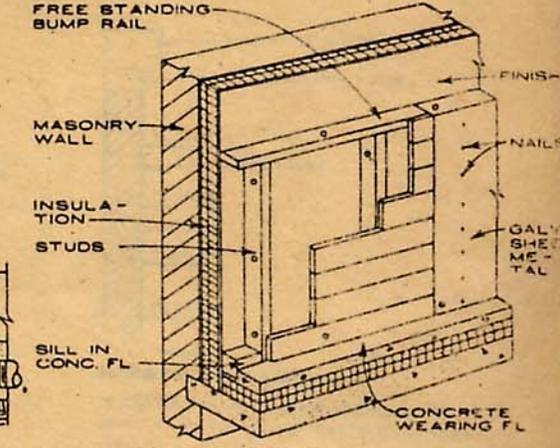
ON WOOD CONSTRUCTION

TYPICAL CEILING DETAILS

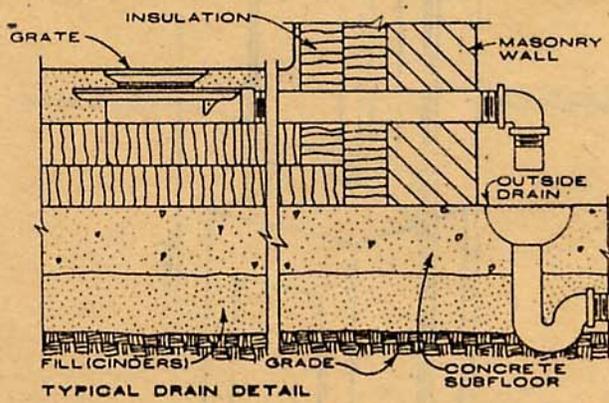
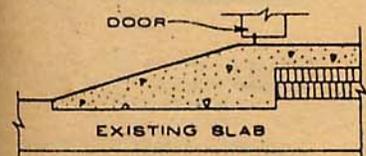
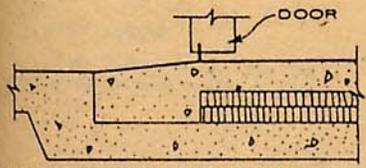


TYPICAL DOOR DETAILS

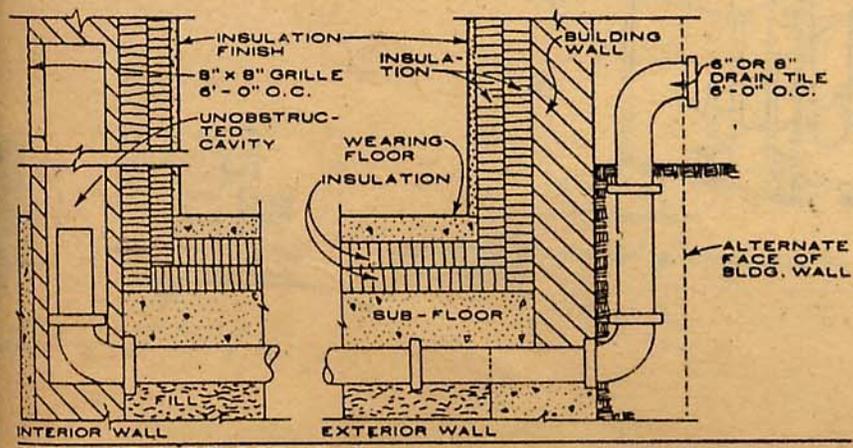
TYPICAL SILL DETAILS



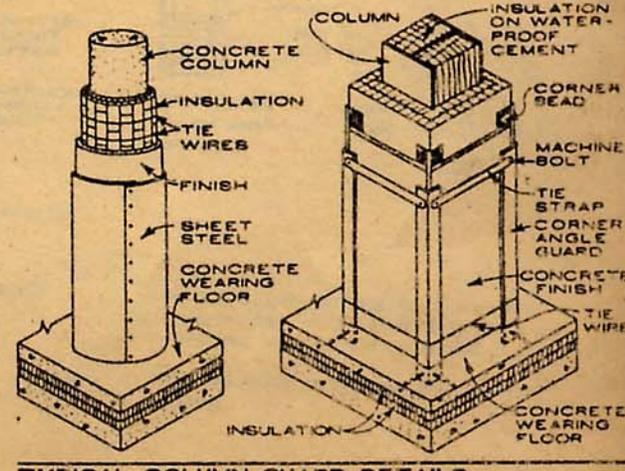
INDEPENDENT BUMPER RAIL GUARD RAIL DETAILS & WAINSCOT



Well-designed drain installed in cold room. Drip drain outside wall meets health code requirements.



DETAIL OF SUBFLOOR VENT



TYPICAL COLUMN GUARD DETAILS