and light commercial cooling equipment operates with a coil SHF of 0.75–0.8 with the air entering the coil at about 80°F or 27°C dry bulb and 67°F or 19°C wet bulb temperature. This equipment usually has a capacity of less than 10 tons or 35 kW. When the peak cooling load and latent heat requirements are appropriate, this less expensive type of equipment is used. In this case the air quantity is determined in a different way. The peak cooling load is first computed as 1.3 times the peak sensible cooling load for the structure to match the coil SHF. The equipment is then selected to match the peak cooling load as closely as possible. The air quantity is specified by the manufacturer for each unit and is about 400 cfm/ton or 0.0537 $m^3/sec \cdot kW$. The total air quantity is then divided among the various rooms according to the cooling load of each room.

64.3.5 Fuel Requirements

The only reliable methods available for estimating cooling equipment energy requirements require hour by hour predictions of the cooling load and must be done using a computer and representative weather data. This is mainly because of the great importance of thermal energy storage in the structure and the complexity of the equipment used. This approach is becoming much easier due to the development of personal computers. This complex problem is discussed in Ref. 3.

There has been recent work related to residential and light commercial applications that is adaptable to hand calculations. The analysis assumes a correctly sized system. Figure 64.15 summarizes the results of the study of compressor operating time for all locations inside the contiguous 48 states. With the compressor operating time it is possible to make an estimate of the energy consumed by the equipment for an average cooling season. The Air-Conditioning and Refrigeration Institute (ARI) publishes data concerning the power requirements of cooling and dehumidifying equipment and most manufacturers can furnish the same data. For residential systems it is generally best to cycle the circulating fan with the compressor. In this case fans and compressors operate at the same time. However, for light commercial applications the circulating fan will probably operate continuously, and this should be taken into account.

64.4 AIR-CONDITIONING EQUIPMENT

64.4.1 Central Systems

When the requirements of the system have been determined, the designer can select and arrange the various components. It is important that equipment be adequate, accessible for easy maintenance, and no more complex in arrangement and control than necessary to produce the conditions required.

Figure 64.16 shows the air-handling components of a central system for year-round conditioning. It is a built-up system, but most of the components are available in subassembled sections ready for bolting together in the field or completely assembled by the manufacturer. Other components not shown are the water heater or boiler, the chiller, condensing unit or cooling tower, pumps, piping, and controls.

All-Air Systems

An all-air system provides complete sensible heating and cooling and latent cooling by supplying only air to the conditioned space. In such systems there may be piping between the refrigerating and heat-producing devices and the air-handling device. In some applications heating is accomplished by a separate air, water, steam, or electric heating system. The term zone implies a provision or the need for separate thermostatic control, whereas the term room implies a partitioned area that may or may not require separate control.

All-air systems may be classified as (1) single-path systems and (2) dual-path systems. Single-path systems contain the main heating and cooling coils in a series flow air path using a common duct distribution system at a common air temperature to feed all terminal apparatus. Dual-path systems contain the main heating and cooling coils in a parallel flow or series-parallel flow air path using either (1) a separate cold and warm air duct distribution system that is blended at the terminal apparatus (dual-duct system), or (2) a single supply duct to each zone with a blending of warm and cold air at the main supply fan.

The all-air system is applied in buildings requiring individual control of conditions and having a multiplicity of zones such as office buildings, schools and universities, laboratories, hospitals, stores, hotels, and ships. Air systems are also used for many special applications where a need exists for close control of temperature and humidity.

The reheat system is to permit zone or space control for areas of unequal loading, or to provide heating or cooling of perimeter areas with different exposures, or for process or comfort applications where close control of space conditions is desired. The application of heat is a secondary process, being applied to either preconditioned primary air or recirculated room air. The medium for heating may be hot water, steam, or electricity.

Conditioned air is supplied from a central unit at a fixed temperature designed to offset the maximum cooling load in the space. The control thermostat activates the reheat unit when the temperature falls below the upper limit of the controlling instrument's setting. A schematic arrangement

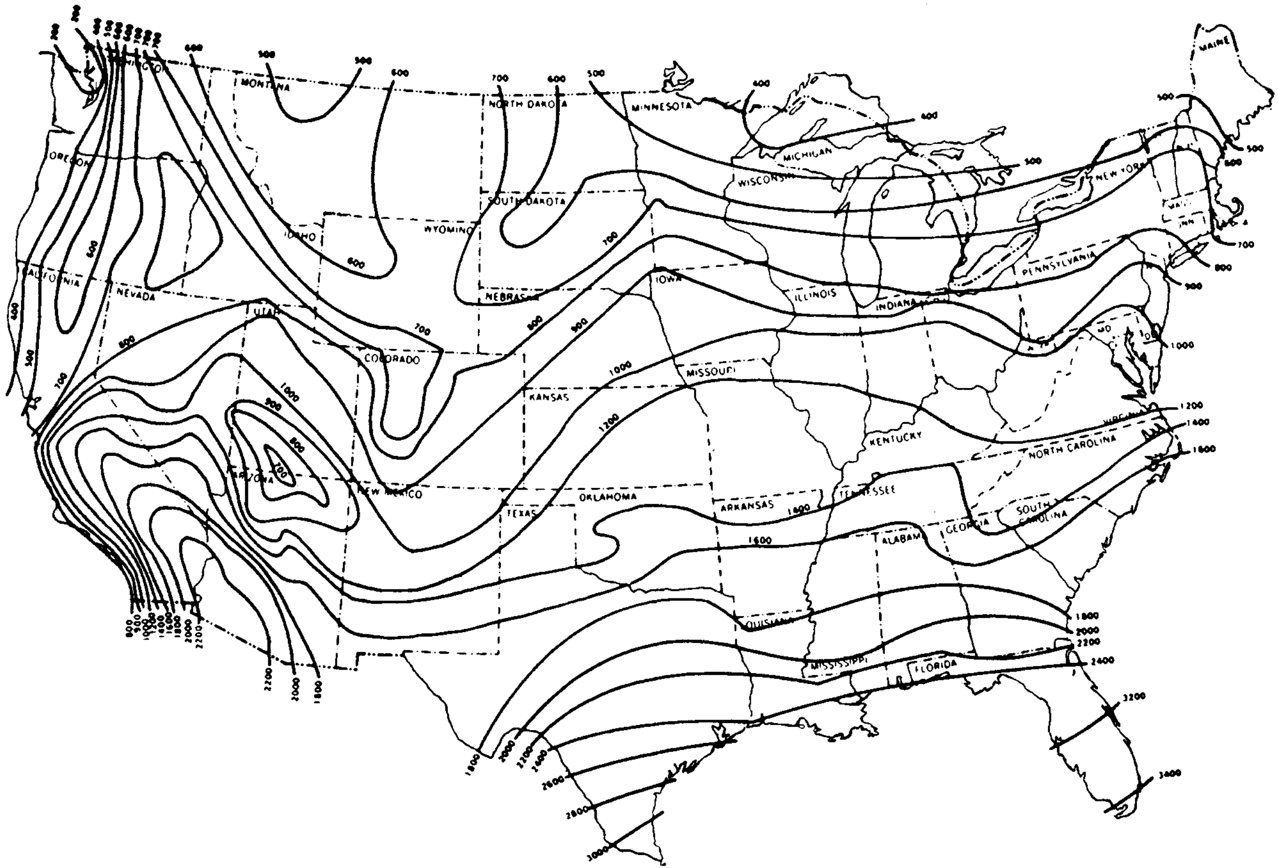


Fig. 64.15 Hours of compressor operation for residential systems. (Reprinted by permission from *ASHRAE*.)

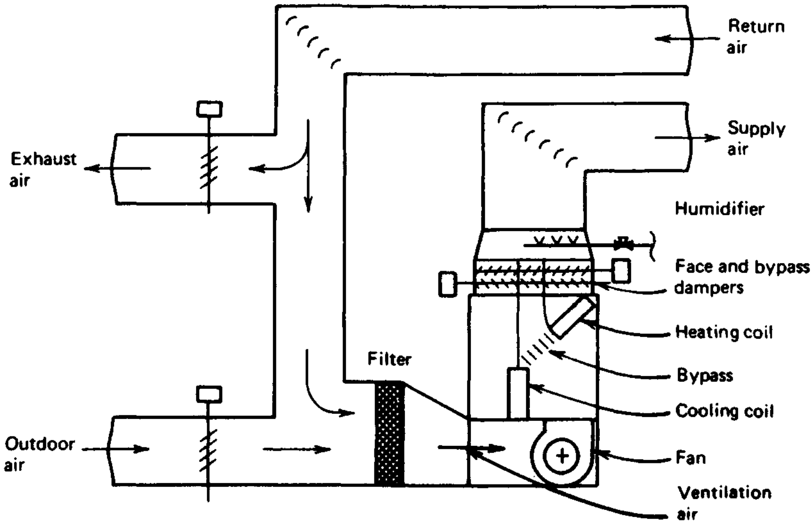


Fig. 64.16 Typical central air system.

of the components for a typical reheat system is shown in Fig. 64.17. To conserve energy reheat should not be used unless absolutely necessary. At the very least, reset control should be provided to maintain the cold air at the highest possible temperature to satisfy the space cooling requirement.

The variable-volume system compensates for varying load by regulating the volume of air supplied through a single duct. Special zoning is not required because each space supplied by a controlled outlet is a separate zone. Figure 64.18 is a schematic of a true variable-air-volume (VAV) system.

Significant advantages are low initial cost and low operating costs. The first cost of the system is low because it requires only single runs of duct and a simple control at the air terminal. Where diversity of loading occurs, smaller equipment can be used and operating costs are generally the lowest among all the air systems. Because the volume of air is reduced with a reduction in load, the refrigeration and fan horsepower follow closely the actual air-conditioning load of the building. During intermediate and cold seasons, outdoor air can be used for economy in cooling. In addition, the system is virtually self-balancing.

Until recently there were two reasons why variable-volume systems were not recommended for applications with loads varying more than 20%. First, throttling of conventional outlets down to 50–60% of their maximum design volume flow might result in the loss of control of room air motion with noticeable drafts resulting. Second, the use of mechanical throttling dampers produces noise, which increases proportionally with the amount of throttling.

With improvements in volume-throttling devices and aerodynamically designed outlets, this system can now handle interior areas as well as building perimeter areas where load variations are

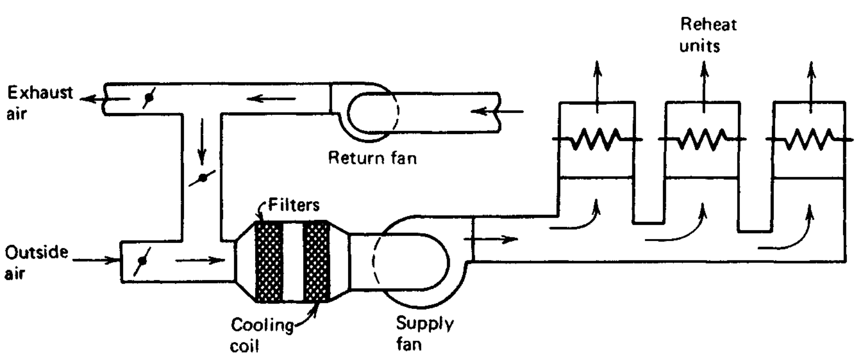


Fig. 64.17 Arrangement of components for a reheat system.

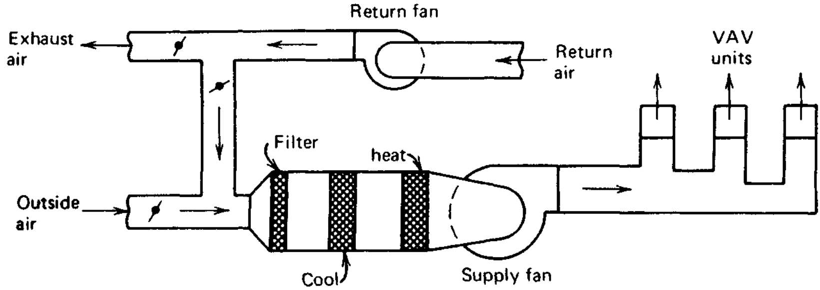


Fig. 64.18 Variable-air-volume system.

greatest, and where throttling to 10% of design volume flow is often necessary. It is primarily a cooling system and should be applied only where cooling is required the major part of the year. Buildings with internal spaces with large internal loads are the best candidates. A secondary heating system should be provided for boundary surfaces. Baseboard perimeter heat is often used. During the heating season, the VAV system simply provides tempered ventilation air to the exterior spaces.

An important aspect of VAV system design is fan control. There are significant fan power savings where fan speed is reduced in relation to the volume of air being circulated.

In the dual-duct system the central station equipment supplies warm air through one duct run and cold air through the other. The temperature in an individual space is controlled by a thermostat that mixes the warm and cool air in proper proportions. One form is shown in Fig. 64.19.

From the energy-conservation viewpoint the dual-duct system has the same disadvantage as reheat. Although many of these systems are in operation, few are now being designed and installed.

The multizone central station units provide a single supply duct for each zone, and obtain zone control by mixing hot and cold air at the central unit in response to room or zone thermostats. For a comparable number of zones this system provides greater flexibility than the single-duct and involves lower cost than the dual-duct system, but it is physically limited by the number of zones that may be provided at each central unit.

The multizone, blow-through system is applicable to locations and areas having high sensible heat loads and limited ventilation requirements. The use of many duct runs and control systems can make initial costs of this system high compared to other all-air systems. To obtain very fine control this system might require larger refrigeration and air-handling equipment.

The use of these systems with simultaneous heating and cooling is now discouraged for energy conservation.

Air and Water Systems

In an air and water system both air and water are distributed to each space to perform the cooling function. In virtually all air–water systems both cooling and heating functions are carried out by changing the air or water temperatures (or both) to permit control of space temperature during all seasons of the year.

The quantity of air supplied can be low compared to an all-air system, and less building space need be allocated for the cooling distribution system.

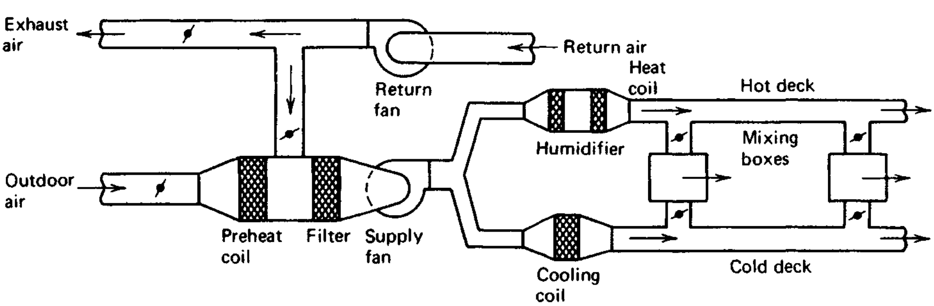


Fig. 64.19 Dual-duct system.

The reduced quantity of air is usually combined with a high-velocity method of air distribution to minimize the space required. If the system is designed so that the air supply is equal to the air needed to meet outside air requirements or that required to balance exhaust (including exfiltration) or both, the return air system can be eliminated for the areas conditioned in this manner.

The pumping power necessary to circulate the water throughout the building is usually significantly less than the fan power to deliver and return the air. Thus not only space but also operating cost savings can be realized.

Systems of this type have been commonly applied to office buildings, hospitals, hotels, schools, better apartment houses, research laboratories, and other buildings. Space saving has made these systems beneficial in high-rise structures.

Air and water systems are categorized as two-pipe, three-pipe, and four-pipe systems. They are basically similar in function, and all incorporate both cooling and heating capabilities for all-season air conditioning. However, arrangements of the secondary water circuits and control systems differ greatly.

All-Water Systems

All-water systems are those with fan-coil, unit ventilator, or valance-type room terminals. with unconditioned ventilation air supplied by an opening through the wall or by infiltration. Cooling and dehumidification are provided by circulating chilled water or brine through a finned coil in the unit. Heating is provided by supplying hot water through the same or a separate coil using two-, three-, or four-pipe water distribution from central equipment. Electric heating or a separate steam coil may also be used. Humidification is not practical in all-water systems unless a separate package humidifier is provided in each room.

The greatest advantage of the all-water system is its flexibility for adaptation to many building module requirements.

64.4.2 Unitary Systems

Unitary Air Conditioners

Unitary air-conditioning equipment consists of factory-matched refrigerant cycle components for inclusion in air-conditioning systems that are field designed to meet the needs of the user. They may vary in:

1. Arrangement: single or split (evaporator connected in the field)
2. Heat rejection: air cooled, evaporative condenser, water cooled
3. Unit exterior: decorative for in-space application, functional for equipment room and ducts, weatherproofed for outdoors
4. Placement: floor standing, wall mounted, ceiling suspended
5. Indoor air: vertical upflow, counterflow, horizontal, 90° and 180° turns, with fan, or for use with forced air furnace
6. Locations: indoor—exposed with plenums or furred in ductwork, concealed in closets, attics, crawl spaces, basements, garages, utility rooms, or equipment rooms; wall—built in, window, transom; outdoor—rooftop, wall mounted, or on ground
7. Heat: intended for use with upflow, horizontal, or counterflow forced air furnace, combined with furnace, combined with electrical heat, combined with hot water or steam coil

Unitary air conditioners as contrasted to room air conditioners are designed with fan capability for ductwork, although some units may be applied with plenums.

Heat pumps are also offered in many of the same types and capacities as unitary air conditioners.

Packaged reciprocating and centrifugal water chillers can be considered as unitary air conditioners particularly when applied with unitary-type chilled water blower coil units. Consequently, a higher level of design ingenuity and performance is required to develop superior system performance using unitary equipment than for central systems, since only a finite number of unitary models is available. Unitary equipment tends to fall automatically into a zoned system with each zone served by its own unit.

For large single spaces where central systems work best, the use of multiple units is often an advantage because of the movement of load sources within the larger space, giving flexibility to many smaller independent systems instead of one large central system.

A room air conditioner is an encased assembly designed as a unit primarily for mounting in a window, through a wall, or as a console. The basic function of a room air conditioner is to provide comfort by cooling, dehumidifying, filtering or cleaning, and circulating the room air. It may also provide ventilation by introducing outdoor air into the room, and by exhausting the room air to the outside. The conditioner may also be designed to provide heating by reverse cycle (heat pump) operation or by electric resistance elements.

64.4.3 Heat Pump Systems

The heat pump is a system in which refrigeration equipment is used such that heat is taken from a heat source and given up to the conditioned space when heating service is wanted and is removed from the space and discharged to a heat sink when cooling and dehumidification are desired. The thermal cycle is identical with that of ordinary refrigeration, but the application is equally concerned with the cooling effect produced at the evaporator and the heating effect produced at the condenser. In some applications both the heating and cooling effects obtained in the cycle are utilized.

Unitary heat pumps are shipped from the factory as a complete preassembled unit including internal wiring, controls, and piping. Only the ductwork, external power wiring, and condensate piping are required to complete the installation. For the split unit it is also necessary to connect the refrigerant piping between the indoor and outdoor sections. In appearance and dimensions, casings of unitary heat pumps closely resemble those of conventional air-conditioning units having equal capacity.

Heat Pump Types

The air-to-air heat pump is the most common type. It is particularly suitable for factory-built unitary heat pumps and has been widely used for residential and commercial applications. Outdoor air offers a universal heat-source, heat-sink medium for the heat pump. Extended-surface, forced-convection heat-transfer coils are normally used to transfer the heat between the air and the refrigerant.

Figure 64.20 shows typical curves of heat pump capacity versus outdoor dry bulb temperature. Imposed on the figure are approximate heating and cooling load curves for a building. In the heating mode it can be seen that the heat pump capacity decreases and the building load increases as the temperature drops. In the cooling mode the opposite trends are apparent. If the cooling load and heat pump capacity are matched at the cooling design temperature, then the balance point, where heating load and capacity match, is then fixed. This balance point will quite often be above the heating design temperature. In such cases supplemental heat must be furnished to maintain the desired indoor condition.

The most common type of supplemental heat for heat pumps in the United States is electrical-resistance heat. This is usually installed in the air-handler unit and is designed to turn on automatically, sometimes in stages, as the indoor temperature drops. In some systems the supplemental heat is turned on when the outdoor temperature drops below some preset value. Heat pumps which have fossil-fuel-fired supplemental heat are referred to as hybrid or bivalent heat pumps.

If the heat pump capacity is sized to match the heating load, care must be taken that there is not excessive cooling capacity for summer operation, which could lead to poor summer performance, particularly in dehumidification of the air.

Air-to-water heat pumps are commonly used in large buildings where zone control is necessary and are also sometimes used for the production of hot or cold water in industrial applications as well as heat reclaiming. Heat pumps for hot water heating are commercially available in residential sizes.

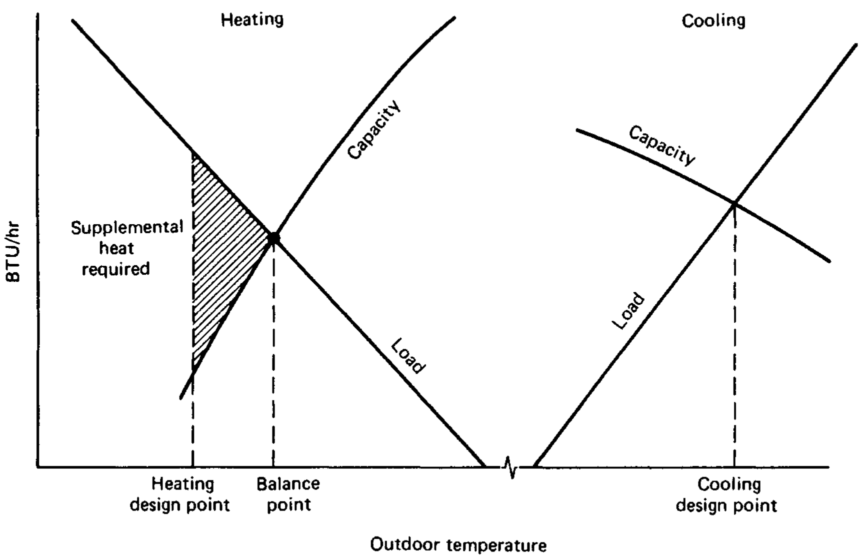


Fig. 64.20 Comparison of building heat loads with heat pump capacities.

A water-to-air heat pump uses water as a heat source and sink and uses air to transmit heat to or from the conditioned space.

A water-to-water heat pump uses water as the heat source and sink for both cooling and heating operation. Heating-cooling changeover may be accomplished in the refrigerant circuit, but in many cases it is more convenient to perform the switching in the water circuits.

Water may represent a satisfactory and in many cases an ideal heat source. Well water is particularly attractive because of its relatively high and nearly constant temperature, generally about 50°F or 10°C in northern areas and 60°F or 16°C and higher in the south. However, abundant sources of suitable water are not always available, and the application of this type of system is limited. Frequently, sufficient water may be available from wells, but the condition of the water may cause corrosion in heat exchangers or it may induce scale formation. Other considerations to be made are the costs of drilling, piping, and pumping, and the means for disposing of used water.

Surface or stream water may be used, but under reduced winter temperatures the cooling spread between inlet and outlet must be limited to prevent freeze-up in the water chiller, which is absorbing the heat.

Under certain industrial circumstances waste process water such as spent warm water in laundries and warm condenser water may be a source for specialized heat pump operations.

A building may require cooling in interior zones while needing heat in exterior zones. The needs of the north zones of a building may also be different from those of the south. In many cases a closed-loop heat pump system is a good choice. Closed-loop systems may be solar assisted. A closed-loop system is shown in Fig. 64.21.

Individual water-to-air heat pumps in each room or zone accept energy from or reject energy to a common water loop, depending on whether that area has a call for heating or for cooling. In the ideal case the loads will balance, and there will be no surplus or deficiency of energy in the loop. If cooling demand is such that more energy is rejected to the loop than is required for heating, the surplus is rejected to the atmosphere by a cooling tower. In the other case, an auxiliary furnace furnishes any deficiency.

The ground has been used successfully as a source–sink for heat pumps with both vertical and horizontal pipe installation. Water from the heat pump is pumped through plastic pipe and exchanges heat with the surrounding earth before being returned back to the heat pump, Fig. 64.22. Tests and analyses have shown rapid recovery in earth temperature around the pipe after the heat pump cycles off. Proper sizing depends on the nature of the earth surrounding the pipe, the water table level, and the efficiency of the heat pump.

Although still largely in the research stage, the use of solar energy as a heat source either on a primary basis or in combination with other sources is attracting increasing interest. Heat pumps may be used with solar systems in either a series or a parallel arrangement, or a combination of both.

64.5 ROOM AIR DISTRIBUTION

64.5.1 Basic Considerations

The object of air distribution in warm air heating, ventilating, and air-conditioning systems is to create the proper combination of temperature, humidity, and air motion in the occupied portion of the conditioned room. To obtain comfort conditions with this space, standard limits for an acceptable

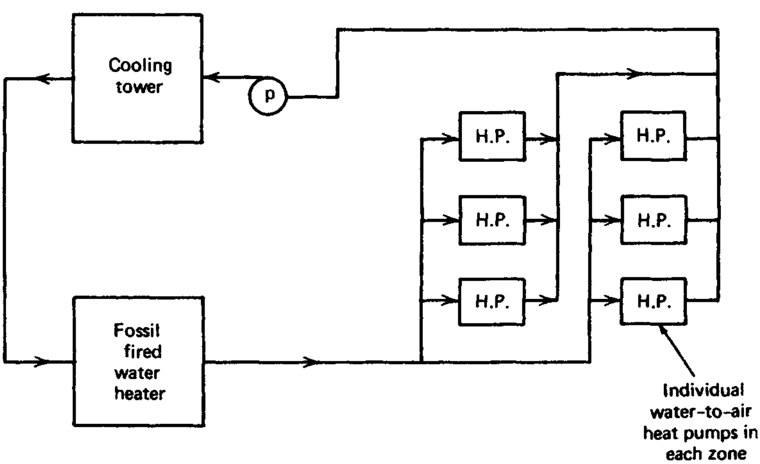


Fig. 64.21 Schematic of a closed-loop heat pump system.[7]

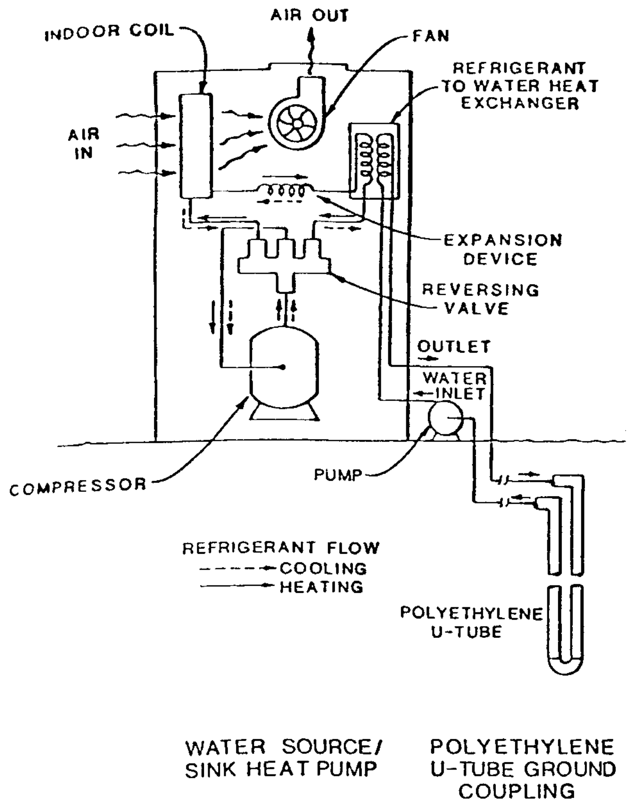


Fig. 64.22 Schematic of a ground-coupled heat pump system.

effective draft temperature have been established. This term comprises air temperature, air motion, relative humidity, and their physiological effect on the human body, any variation from accepted standards of one of these elements may result in discomfort to the occupants. Discomfort also may be caused by lack of uniform conditions within the space or by excessive fluctuation of conditions in the same part of the space. Such discomfort may arise owing to excessive room air temperature variations (horizontally, vertically, or both), excessive air motion (draft), failure to deliver or distribute the air according to the load requirements at the different locations, or rapid fluctuation of room temperature or air motion (gusts).

64.5.2 Jet and Diffuser Behavior

Conditioned air is normally supplied to air outlets at velocities much higher than would be acceptable in the occupied space. The conditioned air temperature may be above, below, or equal to the temperature of the air in the occupied space. Proper air distribution therefore causes entrainment of room air by the primary air stream and reduces the temperature differences to acceptable limits before the air enters the occupied space. It also counteracts the natural convection and radiation effects within the room.

When a jet is projected parallel to and within a few inches of a surface, the induction or entrainment is limited on the surface side of the jet. A low-pressure region is created between the surface and the jet, and the jet attaches itself to the surface. This phenomenon results if the angle of discharge between the jet and the surface is less than about 40° and if the jet is within about 1 ft of the surface. The jet from a floor outlet is drawn to the wall, and the jet from a ceiling outlet is drawn to the ceiling.

Room air near the jet is entrained and must then be replaced by other room air into motion. Whenever the average room air velocity is less than about 50 ft/min or 0.25 m/sec, buoyancy effects may be significant. In general, about 8–10 air changes per hour are required to prevent stagnant regions (velocity less than 15 ft/min or 0.08 m/sec). However, stagnant regions are not necessarily a serious condition. The general approach is to supply air in such a way that the high-velocity air from the outlet does not enter the occupied space. The region within 1 ft of the wall and above about 6 ft from the floor is out of the occupied space for practical purposes.[7]

Perimeter-type outlets are generally regarded as superior for heating applications. This is particularly true when the floor is over an unheated space or a slab and where considerable glass area exists in the wall. Diffusers with a wide spread are usually best for heating because buoyancy tends to increase the throw. For the same reason the spreading jet is not as good for cooling applications because the throw may not be adequate to mix the room air thoroughly. However, the perimeter outlet with a nonspreading jet is quite satisfactory for cooling. Diffusers are available that may be changed from the spreading to nonspreading type according to the season.

The high sidewall type of register is often used in mild climates and on the second and succeeding floors of multistory buildings. This type of outlet is not recommended for cold climates or with unheated floors. A considerable temperature gradient may exist between floor and ceiling when heating; however, this type outlet gives good air motion and uniform temperatures in the occupied zone for cooling application. These registers are generally selected to project air from about three-fourths to full room width.

The ceiling diffuser is very popular in commercial applications and many variations of it are available. Because the primary air is projected radially in all directions, the rate of entrainment is large, causing the high momentum jet to diffuse quickly. This feature enables the ceiling diffuser to handle larger quantities of air at higher velocities than most other types. The ceiling diffuser is quite effective for cooling applications but generally poor for heating. However, satisfactory results may be obtained in commercial structures when the floor is heated.

The return air intake generally has very little effect on the room air motion. But the location may have a considerable effect on the performance of the heating and cooling equipment. Because it is desirable to return the coolest air to the furnace and the warmest air to the cooling coil, the return air intake should be located in a stagnant region.

Noise produced by the air diffuser and air can be annoying to the occupants of the conditioned space. Noise criteria (NC) curves are used to describe the noise in HVAC systems.[5]

The selection and placement of the air outlets is ideally done purely on the basis of comfort. However, the architectural design and the functional requirements of the building often override comfort. When the designer is free to select the type of air-distribution system based on comfort, the perimeter type of system with vertical discharge of the supply air is to be preferred for exterior spaces when the heating requirements exceed 2000 degree (F) days. This type system is excellent for heating and satisfactory for cooling when adequate throw is provided. When the floors are warmed and the degree (F) day requirement is between about 3500 and 2000, the high sidewall outlet with horizontal discharge toward the exterior wall is acceptable for heating and quite effective for cooling. When the heating requirement falls below about 2000 degree (F) days, the overhead ceiling outlet or high sidewall diffuser is recommended because cooling is the predominant mode. Interior spaces in commercial structures are usually provided with overhead systems because cooling is required most of the time.

Commercial structures often are constructed in such a way that ducts cannot be installed to serve the desired air-distribution system. Floor space is very valuable and the floor area required for outlets may be covered by shelving or other fixtures, making a perimeter system impractical. In this case an overhead system must be used. In some cases the system may be a mixture of the perimeter and overhead type.

The Air Distribution Performance Index (ADPI) is defined as the percentage of measurements taken at many locations in the occupied zone of a space which meet a -3 to 2°F effective draft temperature criteria. The objective is to select and place the air diffusers so that an ADPI approaching 100% is achieved. ADPI is based only on air velocity and effective draft temperature, a local temperature difference from the room average, and is not directly related to the level of dry bulb temperature or relative humidity. These effects and other factors such as mean radiant temperature must be accounted for. The ADPI provides a means of selecting air diffusers in a rational way. There are no specific criteria for selection of a particular type of diffuser except as discussed above, but within a given type the ADPI is the basis for selecting the throw. The space cooling load per unit area is an important consideration. Heavy loading tends to lower the ADPI. However, loading does not influence design of the diffuser system significantly. Each type of diffuser has a characteristic room length. Table 64.3, the ADPI selection guide, gives the recommended ratio of throw to characteristic length that should maximize the ADPI. A range of throw-to-length ratios are also shown that should give a minimum ADPI. Note that the throw is based on a terminal velocity of 50 ft/min for all diffusers except the ceiling slot type. The general procedure for use of Table 64.3 is as follows:

1. Determine the airflow requirements and the room size.
2. Select the type of diffuser to be used.
3. Determine the room characteristic length.
4. Select the recommended throw-to-length ratio from Table 64.3.
5. Calculate the throw.

Table 64.3 ADPI Selection Guide[a]

Terminal Device	Room Load W/m^2	Room Load Btu/hr · ft^2	$T_{0.25}/L(T_{50}/L)$ for Max. ADPI	Maximum ADPI	For ADPI Greater Than	Range of $T_{0.25}/L(T_{50}/L)$
High sidewall grilles	250	80	1.8	68	—	—
	190	60	1.8	72	70	1.5–2.2
	125	40	1.6	78	70	1.2–2.3
	65	20	1.5	85	80	1.0–1.9
Circulr ceiling diffusers	250	80	0.8	76	70	0.7–1.3
	190	60	0.8	83	80	0.7–1.2
	125	40	0.8	88	80	0.5–1.5
	65	20	0.8	93	90	0.7–1.3
Sill grille straight vanes	250	80	1.7	61	60	1.5–1.7
	190	60	1.7	72	70	1.4–1.7
	125	40	1.3	86	80	1.2–1.8
	65	20	0.9	95	90	0.8–1.3
Sill grille spread vanes	250	80	0.7	94	90	0.8–1.5
	190	60	0.7	94	80	0.6–1.7
	125	40	0.7	94	—	—
	65	20	0.7	94	—	—
Ceiling slot diffusers[b]	250	80	0.3[b]	85	80	0.3–0.7
	190	60	0.3[b]	88	80	0.3–0.8
	125	40	0.3[b]	91	80	0.3–1.1
	65	20	0.3[b]	92	80	0.3–1.5
Light troffer diffusers	190	60	2.5	86	80	<3.8
	125	40	1.0	92	90	<3.0
	65	20	1.0	95	90	<4.5
Perforated and louvered ceiling diffusers	35–160	11–51	2.0	96	90	1.4–2.7
					80	1.0–3.4

Characteristic Room Length for Several Diffuser Types

Diffuser Type	Characteristic Length, L
High sidewall grille	Distance to wall perpendicular to jet
Circular ceiling diffuser	Distance to closest wall or intersecting air jet
Sill grille	Length of room in the direction of the jet flow
Ceiling slot diffuser	Distance to wall or midplane between outlets
Light troffer diffusers	Distance to midplane between outlets plus distance from ceiling to top of occupied zone
Perforated, louvered ceiling diffusers	Distance to wall or midplane between outlets

[a]Reprinted by permission from *ASHRAE Handbook of Fundamentals,* 1997.
[b]Given for $T_{0.50}/L(T_{100}/L)$.

6. Select the appropriate diffuser from catalog data.
7. Make sure any other specifications are met (noise, total pressure, etc.).

64.6 BUILDING AIR DISTRIBUTION

This section discusses the details of distributing the air to the various spaces in the structure. Proper design of the duct system and the selection of appropriate fans and accessories are essential. A poorly designed system may be noisy, inefficient, and lead to discomfort of occupants. Correction of faulty design is expensive and sometimes practically impossible.

64.6.1 Fans

The fan is an essential component of almost all heating and air-conditioning systems. Except in those cases where free convection creates air motion, a fan is used to move air through ducts and to induce air motion in the space. An understanding of the fan and its performance is necessary if one is to design a satisfactory duct system.

The centrifugal fan is the most widely used because it can effectively move large or small quantities of air over a wide range of pressures. The principle of operation is similar to the centrifugal pump in that rotating impeller mounted inside a scroll type of housing imparts energy to the air or gas being moved.

The vaneaxial fan is mounted on the centerline of the duct and produces an axial flow of the air. Guide vanes are provided before and after the wheel to reduce rotation of the air stream.

The tubeaxial fan is quite similar to the vaneaxial fan but does not have the guide vanes.

Axial flow fans are not capable of producing pressures as high as those of the centrifugal fan but can move large quantities of air at low pressure. Axial flow fans generally produce higher noise levels than centrifugal fans.

Fan efficiency may be expressed in two ways. The total fan efficiency is the ratio of total air power to the shaft power input:

$$\eta_t = \frac{\dot{W}_t}{\dot{W}_{sh}}$$

where Q = volume flow rate, ft^3/min or m^3/sec
$P_{01} - P_{02}$ = change in total pressure, lbf/ft^2 or Pa
$\dot{W}_{sh}$ = shaft power, ft · lbf/min or W

It has been common practice in the United States for Q to be in ft^3/min, $P_{01} - P_{02}$ to be in inches of water, and for W_{sh} to be in horsepower. In this special case,

$$\eta_t = \frac{\dot{Q}(P_{01} - P_{02})}{6.350\dot{W}_{sh}}$$

The static fan efficiency is

$$\eta_s = \frac{\dot{Q}(P_1 - P_2)}{6350\dot{W}_{sh}}$$

Figure 64.23 illustrates typical performance curves for centrifugal fans. Note the differences in the pressure characteristics and in the point of maximum efficiency with respect to the point of maximum pressure.

Table 64.4 compares some of the more important characteristics of centrifugal fans.

The noise emitted by a fan is important in many applications. For a given pressure the noise level is proportional to the tip speed of the impeller and to the air velocity leaving the wheel. Fan noise is roughly proportional to the pressure developed regardless of the blade type; however, backward-curved fan blades generally have the better (lower) noise characteristics.

The pressure developed by a fan is limited by the maximum allowable speed. If noise is not a factor, the straight radial blade is superior. Fans may be operated in series to develop higher pressures, and multistage fans are also constructed. When fans are used in parallel, surging back and forth between fans may develop, particularly if the system demand is changing. Forward-curved blades are particularly unstable when operated at the point of maximum efficiency.

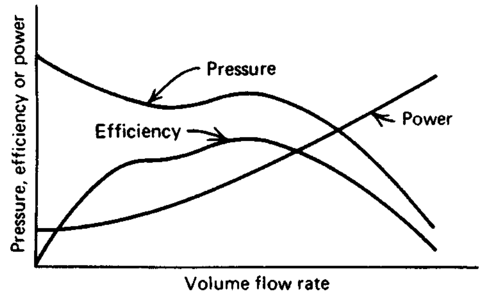
Combining both the system and fan characteristics on a plot is very useful in matching a fan to a system and to ensure fan operation at the desired conditions. Figure 64.24 illustrates the desired operating range for a forward-curved blade fan. The range is to the right of the point of maximum efficiency. The backward-curved blade fan has a selection range that brackets the range of maximum efficiency and is not so critical to the point of operation; however, this type should always be operated to the right of the point of maximum pressure. For a given system the efficiency does not change with speed; however capacity, total pressure, and power all depend on the speed. Changing the fan speed will not change the relative point of intersection between the system and fan characteristics. This can only be done by changing fans.

There are several simple relationships between fan capacity, pressure, speed, and power, which are referred to as the fan laws. The most useful fan laws are:

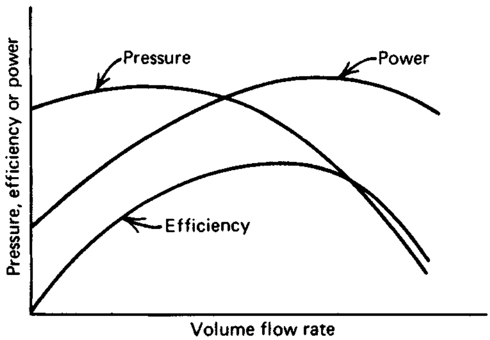
1. Capacity is directly proportional to fan speed.
2. Pressure (static, total, or velocity) is proportional to the square of the fan speed.
3. Power required is proportional to the cube of fan speed.

Three other fan laws are useful.

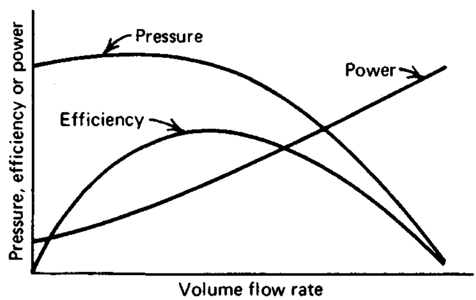
4. Pressure and power are proportional to the density of the air at constant speed and capacity.



Forward-tip fan characteristics.



Backward-tip fan characteristics.



Radial-tip fan characteristics.

Fig. 64.23 Performance curves for centrifugal fans.[7]

Table 64.4 Comparison of Centrifugal Fan Types[7]

Item	Forward-Curved Blades	Radial Blades	Backward-Curved Blades
Efficiency	Medium	Medium	High
Space required	Small	Medium	Medium
Speed for given pressure rise	Low	Medium	High
Noise	Poor	Fair	Good

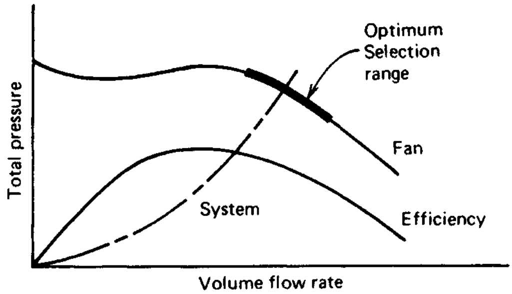


Fig. 64.24 Optimum match between system and forward-curved blade fan.[7]

5. Speed, capacity, and power are inversely proportional to the square root of the density at constant pressure.
6. Capacity, speed, and pressure are inversely proportional to the density, and the power is inversely proportional to the square of the density at a constant mass flow rate.

In a variable-air-volume system it is desirable to reduce fan speed as air-volume flow rate is reduced under part load conditions to reduce the fan power.

Fan Selection

To select a fan it is necessary to know the capacity and total pressure requirement of the system. The type and arrangement of the prime mover, the possibility of fans in parallel or series, nature of the load (variable or steady), and the noise constraints must also be considered. After the system characteristics have been determined, the main considerations in the actual fan selection are efficiency, reliability, size and weight, speed, noise, and cost.

To assist in the actual fan selection, manufacturers furnish graphs with the areas of preferred operation shown. In many cases manufacturers present their fan performance data in the form of tables. The static pressure is often given but not the total pressure. The total pressure may be computed from the capacity and the fan outlet dimensions. Data pertaining to noise are also available from most manufacturers.

It is important that the fan be efficient and quiet. Generally, a fan will generate the least noise when operated near the peak efficiency. Operation considerably beyond the point of maximum efficiency will be noisy. Forward-curved blades operated at high speeds will be noisy and straight blades are generally noisy, especially at high speed. Backward-curved blades may be operated on both sides of the peak efficiency at relatively high speeds with less noise than the other types of fans.

Fan Installation

The performance of a fan can be drastically reduced by improper connection to the duct system. In general, the duct connections should be such that the air may enter and leave the fan as uniformly as possible with no abrupt changes in direction or velocity. The designer must rely on good judgment and ingenuity in laying out the system. Space is often limited for the fan installation, and a less than optimum connection may have to be used. In this case the designer must be aware of the penalties (loss in total pressure and efficiency). Some manufacturers furnish application factors from which a modified fan curve can be computed.

The Air Movement and Control Association, Inc. (AMCA) has published system effect factors in their *Fan Applications Manual* that express the effect of various fan connections on system performance.[8]

64.6.2 Variable-Volume Systems

In variable-air-volume systems the total amount of circulated air may vary between some minimum and the full load design quantity. Normally, the minimum is about 20–25% of the maximum. The volume flow rate of the air is controlled independent of the fan by the terminal boxes, and the fan must respond to the system. The fan speed should be decreased as volume flow rate decreases. Variable speed electric motors have very low efficiency that offsets the benefit of lowering fan speed. Fan drives that make use of magnetic couplings have been developed and are referred to as eddy current drives. These are excellent devices with almost infinite adjustment of fan speed. Their only disadvantage is high cost. A change may be made in the fan speed by changing the diameter of the V-belt drive pulley by adjusting the pulley shives. This requires a mechanism that will operate while the drive is turning. The main disadvantage of this approach is maintenance. The eddy current and variable pulley drives appear to be the most practical at present.

Another approach to control of the fan is to throttle and introduce a swirling component to the air entering the fan that alters the fan characteristic in such a way that less power is required at the lower flow rates. This is done with variable inlet vanes that are a radial damper system located at the inlet to the fan. Gradual closing of the vanes reduces the volume flow rate of air and changes the fan characteristic. This approach is not as effective in reducing fan power as fan speed reduction, but the cost and maintenance are low.

Airflow in Ducts

The steady-flow energy equation applies to the flow of air in a duct. Neglecting the elevation head terms, assuming that the flow is adiabatic, and no fan is present,

$$\frac{g_c}{g}\frac{P_1}{\rho} + \frac{V_1^2}{2g} = \frac{g_c}{g}\frac{P_2}{\rho} + \frac{V_2^2}{2g} + l_f$$

and in terms of the total head

$$\frac{g_c}{g}\frac{P_{01}}{\rho} = \frac{g_c}{g}\frac{P_{02}}{\rho} + l_f$$

where V = average air velocity at a duct cross section, ft/min or m/sec
l_f = lost head due to friction, ft or m

The static and velocity head terms are interchangeable and may increase or decrease in the direction of flow depending on the duct cross-sectional area. Because the lost head must be positive, the total pressure always decreases in the direction of flow, as in Fig. 64.25.

For duct flow the units of each term are usually inches of water because of their small size. The equations may be written

$$H_{s1} + H_{v1} = H_{s2} + H_{v2} + l_f$$

and

$$H_{01} = H_{02} + l_f$$

For air at standard conditions

$$H_v = \left(\frac{V}{4005}\right)^2 \text{ in. } H_2O$$

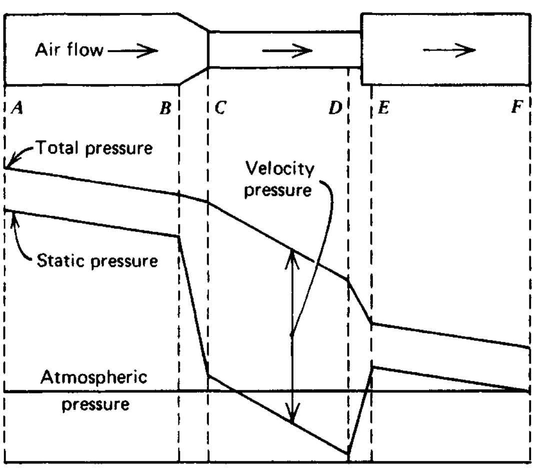


Fig. 64.25 Pressure changes during flow in ducts.[7]

where V is in ft/min,

$$P_v = \left(\frac{V}{1.29}\right)^2 \text{ Pa}$$

where V is in m/sec.

The lost head due to friction, l_f, in a straight, constant area duct may be determined by use of a friction factor. Because this approach becomes tedious when designing ducts, special charts have been prepared. Figure 64.26 is such a chart for air flowing in ducts. The chart is based on standard air and fully developed flow. For the temperature range of 50°F or 10°C to about 100°F or 38°C there is no need to correct for viscosity and density changes. Above 100°F or 38°C, however, a correction should be made. The density correction is also small for moderate pressure changes. For elevations below about 2000 ft or 610 m the correction is small. The correction for density and viscosity will normally be less than 1. The effect of roughness is an important consideration and difficult to assess.

A common problem to designers is determination of the roughness effect of fibrous glass duct liners and fibrous ducts. This material is manufactured in several grades with various degrees of absolute roughness. The usual approach to account for this roughness effect is to use a correction factor that is applied to the pressure loss obtained for galvanized metal duct.

The head loss due to friction is greater for a rectangular duct than for a circular duct of the same cross-sectional area and capacity. For most practical purposes ducts of aspect ratio not exceeding 8:1 will have the same lost head for equal length and mean velocity of flow as a circular duct of the same hydraulic diameter. When the duct sizes are expressed in terms of hydraulic diameter D and when the equations for friction loss in round and rectangular ducts are equated for equal length and capacity, an equation for the circular equivalent of a rectangular duct is obtained:

$$D_e = 1.3 \frac{(ab)^{5/8}}{(a + b)^{1/4}}$$

where a and b are the rectangular duct dimensions in any consistent units and D_e is the equivalent diameter. A table of equivalent diameters is given in the *ASHRAE Handbook.*[2]

Air Flow in Fittings

Whenever a change in area or direction occurs in a duct or when the flow is divided and diverted into a branch, substantial losses in total pressure may occur. These losses are usually of greater magnitude than the losses in the straight pipe and are referred to as dynamic losses.

Dynamic losses vary as the square of the velocity and are conveniently represented by

$$H_0 = (C)(H_v)$$

where the loss coefficient C is a constant. When different upstream and downstream areas are involved as in an expansion or contraction, either the upstream or downstream value of H_v may be used but C will be different in each case.

Fittings are classified as either constant flow, such as an elbow or transition, or as divided flow, such as a wye or tee. Tables give loss coefficients for many different types of constant flow fittings.[2] It should be kept in mind that the quality and type of construction may vary considerably for a particular type of fitting. Some manufacturers provide data for their own products.

Duct Design—General Considerations

The purpose of the duct system is to deliver a specified amount of air to each diffuser in the conditioned space at a specified total pressure. This is to ensure that the space load will be absorbed and the proper air motion within the space will be realized. The method used to lay out and size the duct system must result in a reasonably quiet system and must not require unusual adjustments to achieve the proper distribution of air to each space. A low noise level is achieved by limiting the air velocity, by using sound-absorbing duct materials or liners, and by avoiding drastic restrictions in the duct such as nearly closed dampers. Figure 64.26 gives recommended duct velocities for low- and high-velocity systems. A low-velocity duct system will generally have a pressure loss of less than 0.15 in. H_2O per 100 ft (1.23 Pa/m), whereas high-velocity systems may have pressure losses up to about 0.7 in. H_2O per 100 ft (5.7 Pa/m). Fibrous glass duct materials are very effective for noise control. The duct, insulation, and reflective vapor barrier are all the same piece of material. Metal ducts are usually lined with fibrous glass material in the vicinity of the air-distribution equip-

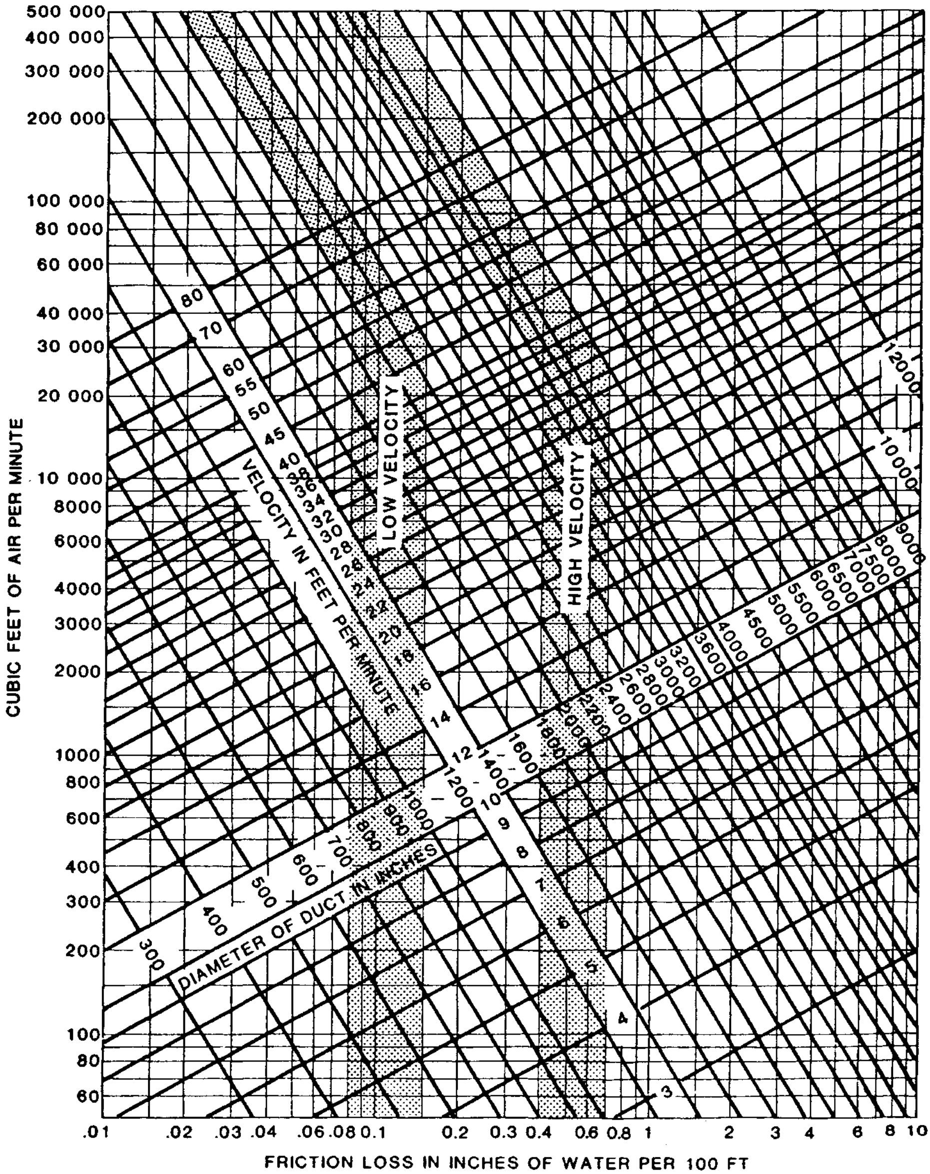


Fig. 64.26 Lost head due to friction for air flowing in ducts. (Reprinted by permission from *ASHRAE Handbook of Fundamentals.*)

ment. The remainder of the metal duct is then wrapped or covered with insulation and a vapor barrier. Insulation on the outside of the duct also reduces noise. The duct system should be relatively free of leaks, especially when the ducts are outside the conditioned space.

Generally, the location of the air diffusers and air-moving equipment is first selected with some attention given to how a duct system may be installed. The ducts are then laid out with attention given to space and ease of construction. It is very iinportant to design a duct system that can be constructed and installed in the allocated space. If this is not done, the installer may make changes in the field that lead to unsatisfactory operation.

From the standpoint of first cost, the ducts should be small; however, small ducts tend to give high air velocities, high noise levels, and large losses in total pressure. Therefore, a reasonable compromise between first cost, operating cost, and practice must be reached. A number of computer programs are available for this purpose.

For residential and light commercial applications all of the heating, cooling, and air-moving equipment is determined by the heating and/or cooling load. Therefore, the fan characteristics are known before the duct design is begun. The pressure losses in all other elements of the system except the supply and return ducts are known. The total pressure available for the ducts is then the difference between the total pressure characteristic of the fan and the sum of the total pressure losses of all of the other elements in the system excluding the ducts. Figure 64.27 shows a typical total pressure profile for a residential or light commercial system. In this case the fan is capable of developing 0.6 in. H_2O at the rated capacity. The return grille, filter, coils, and diffusers have a combined loss in total pressure of 0.38 in. H_2O. Therefore, the available total pressure for which the ducts must be designed is 0.22 in. H_2O. This is usually divided for low-velocity systems so that the supply duct system has about twice the total pressure loss of the return ducts.

Large duct systems are usually designed using velocity as a criterion, and the fan requirements are determined after the design is complete. For these systems the fan characteristics are specified and the correct fan is installed in the air handler.

Element	Total pressure loss, in. of water
Return grille	0.04
Return duct	0.08
Filter	0.08
Heat and cool coils	0.23
Supply ducts	0.14
Diffusers	0.03
Fan total pressure in. of water	0.60

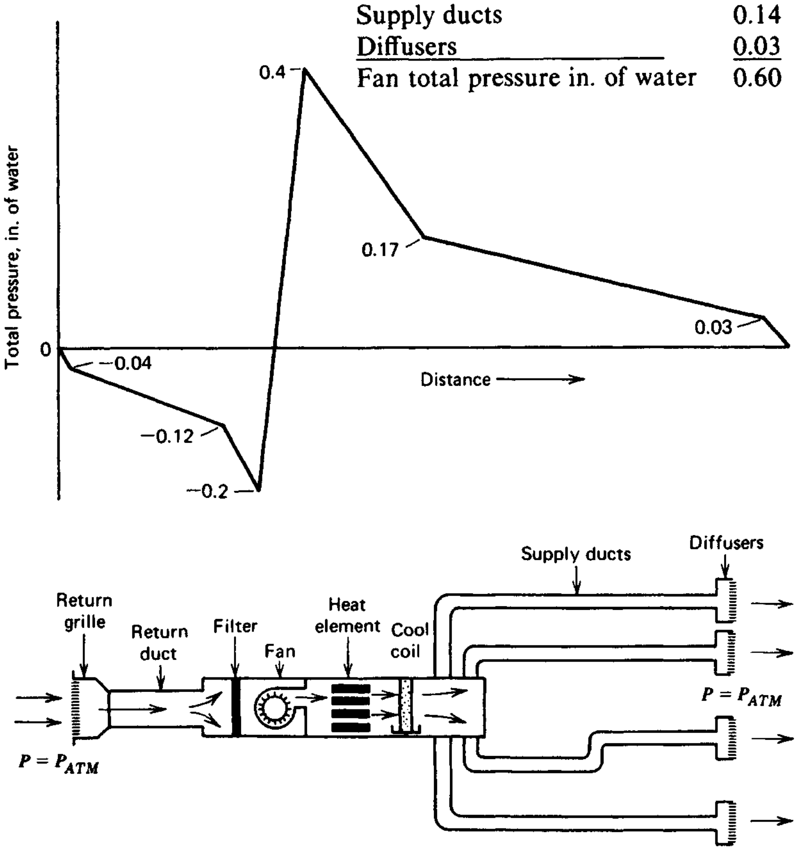


Fig. 64.27 Total pressure profile for a typical residential or light commercial system.[7]

Some designers neglect velocity pressure when dealing with low-velocity systems. This does not simplify the duct design procedure and is unnecessary. When the air velocities are high, the velocity pressure must be considered to achieve reasonable accuracy. If static and velocity pressure are computed separately, the problem becomes very complex. It is best to use total pressure in duct design because it is simpler and accounts for all of the flow energy.

Design of Low-Velocity Duct Systems

The methods described in this section pertain to low-velocity systems where the average velocity is less than about 1000 ft/min or S m/sec. These methods can be used for high-velocity system design, but the results will not be satisfactory in most cases.

Equal Friction Method

This method makes the pressure loss per foot of length the same for the entire system. If all runs from fan to diffuser are about the same length, this method will produce a well balanced design. However, most duct systems have duct runs ranging from long to short. The short runs will have to be dampered, which can cause considerable noise.

The usual procedure is to select the velocity in the main duct adjacent to the fan and to provide a satisfactory noise level for the particular application. The known flow rate then establishes the duct size and the lost pressure per unit of length. This same pressure loss per unit length is then used throughout the system. A desirable feature of this method is the gradual reduction of air velocity from fan to outlet, thereby reducing noise problems. After sizing the system the designer must compute the total pressure loss of the longest run (largest flow resistance), taking care to include all fittings and transitions. When the total pressure available for the system is known in advance, the design loss value may be established by estimating the equivalent length of the longest run and computing the low pressure per unit length.

Balanced Capacity Method

This method of duct design has been referred to as the "balanced pressure loss method." However, it is the flow rate or capacity of each outlet that is balanced and not the pressure. As previously discussed, the loss in total pressure automatically balances regardless of the duct sizes. The basic principle of this method of design is to make the loss in total pressure equal for all duct runs from fan to outlet when the required amount of air is flowing in each. For a given equivalent length the diameter can always be adjusted to obtain the necessary velocity that will produce the required loss in total pressure. There may be cases, however, when the required velocity may be too high to satisfy noise limitations and a damper or other means of increasing the equivalent length will be required.

The design procedure for the balanced capacity method begins the same as the equal friction method in that the design pressure loss per unit length for the run of longest equivalent length is determined in the same way depending on whether the fan characteristics are known in advance. The procedure then changes to one of determining the required total pressure loss per unit length in the remaining sections to balance the flow as required. The method shows where dampers may be needed and provides a record of the total pressure requirements of each part of the duct system. Both the equal friction method and the balanced capacity method are described in Ref. 7.

Return Air Systems

The design of the return system may be carried out using the methods described above. In this case the air flows through the branches into the main duct and back to the plenum. Although the losses in constant-flow fittings are the same regardless of the flow direction, divided-flow fittings behave differently and different equivalent lengths or loss coefficients must be used. Reference 2 gives considerable data for converging-type fittings of both circular and rectangular cross section. For low-velocity ratios the loss coefficient can become negative with converging-flow streams. This behavior is a result of a high-velocity stream mixing with a low-velocity stream. Kinetic energy is transferred from the higher- to the lower-velocity air, which results in an increase in energy or total pressure of the slower stream. Low-velocity return systems are usually designed using the equal friction method. The total pressure loss for the system is then estimated as discussed for supply duct systems. Dampers may be required just as with supply systems. In large commercial systems a separate fan for the return air may be required.

High-Velocity Duct Design

Because space allocated for ducts in large commercial structures is limited owing to the high cost of building construction, alternatives to the low-velocity central system are usually sought. One approach is to use hot and cold water, which is piped to the various spaces where small central units or fan coils may be used. However, it is sometimes desirable to use rather extensive duct systems without taking up too much space. The only way this can be done is to move the air at much higher velocities. High-velocity systems may use velocities as high as 6000 ft/min or about 30 m/sec. The use of high velocities reduces the duct sizes dramatically, but introduces some new problems.

Noise is probably the most serious consequence of high-velocity air movement. Special attention must be given to the design and installation of sound-attenuating equipment in the system. Because the air cannot be introduced to the conditioned space at a high velocity, a device called a terminal box is used to throttle the air to a low velocity, control the flow rate, and attenuate the noise. The terminal box is located in the general vicinity of the space it serves and may distribute air to several outlets.

The energy required to move the air through the duct system at high velocity is also an important consideration. The total pressure requirement is typically on the order of several inches of water. To partially offset the high fan power requirements, variable-speed fans are sometimes used.

Because the higher static and total pressures required by high-velocity systems aggravate the duct leakage problem, an improved duct fabrication system has been developed for high-velocity systems. The duct is generally referred to as spiral duct and has either a round or oval cross section. The fittings are machine formed and are especially designed to have low pressure losses and close fitting joints to prevent leakage.

The criterion for designing the high-velocity duct systems is somewhat different from that used for low-velocity systems. Emphasis is shifted from a self-balancing system to one that has minimum losses in total pressure.

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