CHAPTER 5
COMPONENTS

5.1 CONTEXT

The components of an air-conditioning system fall into four broad categories: source, distribution, delivery, and control elements. Source components provide primary heating and cooling effect and include chillers, boilers, cooling towers, and similar equipment. Distribution components transport heating or cooling effect from the source(s) to the spaces that require conditioning and include ductwork, fans, piping, and pumps. Delivery components introduce heating or cooling effect into conditioned spaces and include diffusers, baseboard radiators, fan-coil units, and a range of other "terminal" devices. Control components regulate the operation of equipment and systems for comfort, process, safety, and energy efficiency. In central systems these components may be spread throughout a building. In local systems these components are all packaged into a relatively small container.

The many components of air-conditioning systems are presented in depth in several chapters in the *ASHRAE Handbook—HVAC Systems and Equipment.* That resource should be a primary reference because this design manual covers only a few items of particular importance to HVAC&R system design. This chapter provides an overview of key components; subsequent chapters discuss the arrangements of components that constitute various HVAC&R system configurations.

5.2 COOLING SOURCE EQUIPMENT

5.2.1 Vapor Compression Refrigeration

The most commonly used source of building cooling is the vapor compression refrigeration thermodynamic cycle. For small capacities, reciprocating, rotary, or scroll compressors are used; medium-sized units frequently employ screw compressors; large units use centrifugal compressors. Reciprocating compressors for air conditioning applications consume approximately 1 kW/ton [0.3 kW/kW], while centrifugal units can consume as little as 0.52 kW/ton [0.16 kW/kW].

A vapor compression system for building cooling usually consists of one or more components that either directly cool air (termed "DX" for *direct expansion*) or cool water (using a *chiller*). Cooled air from a DX system is conducted to conditioned spaces; chilled water from a chiller is conducted to cooling coils where it subsequently cools air. Each refrigeration device includes a refrigerant compressor driven by a prime mover (usually an electric motor). The compressor takes its suction from a heat exchanger (the evaporator) and discharges refrigerant into a second heat exchanger (the condenser). Condensed (liquid) refrigerant passes from the condenser to the evaporator through a throttling valve, thus providing a closed refrigerant circuit.

A refrigeration system works against thermal lift, the difference between the condensing temperature and the evaporating temperature. Typical chiller condenser water temperatures are 85°F [29.4°C] supply and 95°F [35.0°C] return. Typical chilled water temperatures are 44°F [6.7°C] supply and 56°F [13.3°C] return. Raising the chilled-water temperature 1°F [0.6°C] can reduce the energy used for refrigeration by approximately 1.75%. Changing the temperature range (difference between supply and return) of either the condenser water loop or the chilled water loop will impact the flow requirements for those systems—with reduced pumping horsepower for higher temperature ranges. The advantages of using ultra-low chilled water temperatures are discussed in Section 9.3.8.

Split systems are DX systems in which the evaporator and condenser are separated and placed at different locations. (As a rule, the compressor is located near the condenser.) The upper practical limit for total distance between the evaporator and condenser is 300 ft [90 m], but the only true limit is based upon careful design of the interconnecting refrigerant piping. The compressor discharge line should be sized for a gas velocity of no less than 1000 fpm [5 m/s] under minimum load with upward gas flow and no less than 500 fpm [2.5 m/s] with horizontal flow. The maximum recommended flow velocity under full load is 4000 fpm [20 m/s] because higher velocities generally cause excessive pressure drop and noise. Packaged (pre-designed) split systems are typically restricted to substantially shorter separation distances.

The upper limit for vertical lift (difference in elevation) is 60 ft [18 m]. When the net lift is no more than 8 ft [2.4 m], no U-traps are required to ensure return of oil to the compressor (Figure 5-1). However, in these cases, the pipe must pitch away from the compressor at least 0.5 in. for every 10 ft [4.2 mm per m] of horizontal run. Suction and hot gas risers that exceed 8 ft [2.4 m] of net lift require a trap at the base of the riser and an additional trap at each 25 ft [7.6 m] interval of net lift (Figure 5-2). If the compressor is installed below the condenser, the hot gas discharge must have double risers to capture the oil in a *P-trap* and lift it to the condenser for eventual return to the compressor. In this case, always use an oil separator directly after the compressor discharge valve and an inverted P-trap entering the condenser with a liquid shutoff valve leaving the condenser. A P-trap consists of three line-size elbows arranged to trap oil in two of the elbows before the horizontal run. A smaller diameter bypass permits gas to continue flowing until sufficient pressure is built up to propel the trapped oil upward to the condenser.

Fig. 5-1 Refrigerant Line Arrangement for Low Lift.

Fig. 5-2 Refrigerant Line Arrangement for High Lift.

It is wise to include a receiver between the condenser and the evaporator. This is a horizontal tank that allows separation of refrigerant gas and liquid; the receiver serves as a reservoir for the refrigerant and accommodates changes in inventory in the refrigerant loop under different operating conditions. Alternatively, it may be possible to hold refrigerant in the condenser.

Receivers, when used, should be designed to store the entire refrigerant charge (to handle winter shut down or when high-side components are removed for repair). This avoids the need to vent refrigerant, an unacceptable practice because of its expense and harmful effect on the ozone layer. Receivers should be sized to be no more than 80% full, allowing 20% for expansion.

The *International Building Code* (ICC 2003) specifies minimum ventilation requirements for dilution of lost refrigerant in a compressor room. The code also prescribes proper sizing for mechanical room exhaust fans and limitations on the auxiliary equipment that may be installed in a mechanical room. Follow local codes if they are more restrictive than model codes. "Break-glass stations" must be provided at each exit door, to start the exhaust fan and shut down the refrigeration equipment in case of an emergency. Be sure to provide for sufficient makeup air, which need not be conditioned.

Refrigerants R-11, R-12, R-113, R-114, and R-115 have historically been used in building applications, depending upon system capacity. These refrigerants, however, are being phased out of use (and are no longer manufactured in the U.S.) in accordance with the Montreal Protocol. This international treaty, signed in 1987 and ratified since then by most industrial nations including the United States and Canada, aims to reduce the use of chlorofluorocarbons (CFCs) in order to protect the ozone layer of the stratosphere. Note that ozone protection is not identical to concerns about global warming, although both issues involve the release of gases into the atmosphere. Alternative refrigerants include R-22 for small or medium-sized systems, R-123 to replace R-11, and R-134a to replace R-12. Refrigerant properties are seen as an element of green design, and alternative refrigerants are promoted as such by equipment manufacturers.

5.2.2 Absorption Refrigeration

The absorption refrigeration cycle is primarily employed in large capacity applications where a motive force other than electricity (such as steam or suitable waste process heat) is readily available. Often the initial costs of an absorption versus a vapor compression refrigeration system are within 15 to 20% of each other, the compression cycle being less expensive (except where subsidies or incentives for absorption systems are available). The COP (coefficient of performance) of an absorption refrigeration system will be less than that of a vapor compression system—COPs for absorption systems are typically less than 1.0. Due to the greater heat rejection of absorption systems (generally 31,000 Btu/ton [2.6 kW/kW]) compared to compression systems (15,000 Btu/ton or less [1.25 kW/kW]), cooling towers, condenser water pumps, and other auxiliaries must be larger and are therefore more expensive for absorption systems than for compression systems.

Absorption machines can be direct-fired (when a heat source is directly applied to the machine) or indirect-fired by steam or hot water generated by gas or oil combustion or another process (such as solar energy or waste heat). Absorption machines also require a few electric auxiliaries. An economic comparison of refrigeration systems depends greatly upon relative energy costs. Since most air-conditioning ton-hours occur in the summer months and gas utilities may have more economical rates during summer, absorption chillers may be economically attractive under those conditions (although recent energy demands have broken down many traditional usage and economic patterns). Absorption machines do not use any CFCs, which has seen them promoted as a green alternative to mechanical refrigeration. Absorption chillers can also be quieter than vapor compression machines and produce less vibration.

5.2.3 Evaporative Cooling

The use of evaporative coolers to provide direct active cooling has decreased since the 1950s, when an emphasis on low first cost overwhelmed consideration of operating or life-cycle costs and tended to relegate evaporative coolers to niche markets. This situation persists today. Depending upon a building's operating hours and location, however, evaporative cooling can be applied effectively from both a cost and performance perspective. This is certainly true in the southwestern United States, where low outdoor wet-bulb temperatures are prevalent, in other locations where wet-bulb temperatures are below 55 to 60°F [12.8 to 15.6°C] for a substantial number of hours, or when a building is to be air conditioned at night and there is a fairly wide diurnal temperature swing.

Direct evaporative cooling requires 100% outdoor air and exhaust capability; therefore, the cost of equipment space, filter capacity, air delivery, etc., must be weighed against the energy savings of evaporative cooling versus refrigeration cooling. Indirect evaporative cooling broadens the application range of this psychrometric process into climate zones and occupancies where direct evaporative cooling is not a viable system choice.

Two general types of evaporative cooling equipment are available: spray-type air washers and wetted-media-type coolers. These options are discussed below.

5.2.3.1 Spray-Type Air Washers

Spray-type air washers are versatile in that they can use chilled water from a refrigeration system when the outdoor air wet-bulb temperature is high, but can also provide cooling without chilled water when the outdoor air is suitable for conventional evaporative cooling.

Two types of spray air washers are available. One type is the "full open" spray washer. It is generally either 7 or 11 ft [2.1 or 3.4 m] long in the direction of airflow, depending upon whether it has one or more spray banks. Air passes through the device at about 500 fpm [2.5 m/s] and passes through a spray of water. The direct contact of air with the water droplets provides an efficient heat exchange. Depending upon air velocity and spray density (gpm/ft^2 [L/s m^2]), the adiabatic efficiency (the ratio of actual dry-bulb depression to the difference between entering air dry-bulb and spray water temperature) can be as high as 98%.

Another type of air washer is the "short-coil" washer. It consists of a conventional extended finned coil bank placed in a casing with a deep water pan and an opposing spray bank. The short-coil washer has a much lower spray density than the spray washer and only requires 4 ft [1.2 m] of length in the direction of airflow. When the outdoor air wet bulb temperature is not conducive to evaporative cooling, chilled water coils provide cooling. When the outdoor air wet bulb temperature is sufficiently low, the chilled water coils are shut off and a spray of water cascading over the extended coil fins provides the cooling effect. The closed circuit chilled water loop of the short-coil washer requires less chilled water pumping energy than a full-open device. Because of lower spray capacity than the full-open approach, however, it has an adiabatic efficiency of only 50%, thereby requiring more hours of active refrigeration operation.

Water reservoirs require constant maintenance to prevent corrosion and the buildup of bacteria and fungi that would otherwise be dispersed into the air steam during the evaporative cooling process. When this is not done, air washers can become a potential source of biological contaminants and a cause of poor indoor air quality and/or sick building syndrome.

5.2.3.2 Wetted-Media-Type Coolers

Because wetted-media coolers use non-refrigerated water, they are adaptable to air-conditioning systems only in those climate zones where very low ambient wet-bulb temperatures prevail year-round. In a climate like that of Phoenix, Arizona, a dual air-conditioning system may be indicated under certain conditions—wetted media for 10 months of the year and compression refrigeration during the 2-month humid season. The extremely low operating cost of the wetted-media evaporative cooler often justifies the higher initial investment for two systems.

The systems described thus far are *direct* evaporative coolers because water is injected into the primary (supply) air stream, which is cooled adiabatically. In *indirect* evaporative coolers, secondary air (outdoor air or exhaust air) on one side of a heat exchanger is cooled evaporatively. The primary (supply) air is on the other side of the heat exchanger and is indirectly cooled through the heat exchanger walls; thus, no moisture is added to the primary air although it is sensibly cooled. The leaving dry-bulb temperature of the primary air can be no lower than the entering wet-bulb temperature of the secondary air. An indirect evaporative cooler used as a pre-cooler is illustrated in Figure 5-3. Other examples are given in the chapter on Evaporative Air Cooling in the *ASHRAE Handbook—HVAC Systems and Equipment*. That chapter also covers technological improvements to the evaporative cooling process, which broaden the scope of application through the use of two-stage (and even three-stage) indirect/direct evaporative cooling.

Fig. 5-3 Indirect Evaporative Cooler Used as a Pre-Cooler.

5.2.4 Desiccant Systems

Desiccant systems dry air by exposure to desiccants. A desiccant is a liquid or solid that has a high affinity for water—a hygroscopic substance. Desiccants are regenerated (dried after absorbing water) by heating. Such systems, used where latent loads are a serious design issue, are discussed in Section 9.1 and in Appendix E.

5.2.5 Prime Movers

Induction electric motors are the predominant prime mover for refrigeration compressors. Where local electric utility rates incorporate a penalty for low power factor, the addition of capacitors or the use of synchronous electric motors to improve power factor may be cost effective.

Where natural gas is relatively cheap (seasonally or throughout the year), a combination of gas- or steam-turbine-driven centrifugal compression and absorption refrigeration is sometimes used, particularly for large buildings. This scheme is referred to as a *piggyback system,* in which steam is generated in a boiler and first sent to a back-pressure turbine, which drives a centrifugal compressor. The exhaust from the steam turbine becomes the thermal input to an absorption chiller. When a gas turbine drives the compressor, its exhaust gases generate steam for the absorption unit in a heat recovery boiler. As a general rule, the steam-turbine-driven compressor is selected to have one-half the capacity of the absorption unit.

Example 5.1. For a total air-conditioning load of 900 tons [3,165 kW], select a 300-ton [1,055 kW] centrifugal chiller driven by a back-pressure turbine with a steam rate of 36 lb/hp [21.9 kg/kW]; couple this with a single-effect, low-pressure, 600-ton [2,110 kW] absorption chiller having a steam rate of 18 lb/ton [2.4 kg/kW]. The result is 900 tons [3,165 kW] of capacity using 10,800 lb [4,900 kg] of steam per hour or 12 lb/ton [1.6 kg/kW]. This compares with 18 lb/ton [2.4 kg/kW] for a single effect absorption system.

Where district steam is available in the summer at low cost, steam-driven turbines are sometimes used as prime movers for large centrifugal chillers. Here again, the selection of steam versus electric drive depends primarily upon the relative utility costs, and a careful analysis should be made to determine the relative merits of the prime mover options. On rare occasions and for very large buildings, gas turbines have been used as prime movers for centrifugal compressors, either with or without waste heat boilers.

There is increasing interest in the use of combined heat and power (CHP) systems (also called cogeneration systems) that produce electricity on site and provide a consistent source of “waste” heat that can be used a prime mover medium. See, for example, a discussion of CHP systems for commercial buildings in the *ASHRAE Journal* (Zogg, Roth and Brodrick 2005).

5.2.6 Variable-Speed Drives

The use of variable-speed (variable frequency) drives for compressors, air handlers, and pumps has become common because they are readily available at reasonable cost and can provide energy savings in many system types. These drives are solid-state devices that were, at one time, add-ons to constant speed drive equipment. They are now frequently part of the factory-supplied equipment for air handlers, pumps, and compressors. Variable-speed drives save energy whenever electric motors run at less than full power.

Since most HVAC&R equipment rarely runs at full capacity, substantial energy savings are available with these drives. Typical centrifugal compressor performance curves under pre-rotation vane control and under variable-speed control are shown in Figures 5-4 and 5-5, respectively. Power demands of fans under different methods of capacity control are compared in Figure 5-6. Another advantage of variable speed drives is that they provide high power factors, thereby eliminating the need to provide power factor correction for connected motors.

Fig. 5-4 Constant Speed Compressor Performance under Pre-Rotation Vane Control (Courtesy of *Heating/Piping/Air Conditioning* 1984).

Fig. 5-5 Variable-Speed Compressor Performance (Courtesy of *Heating/Piping/Air Conditioning* 1984).

Fig. 5-6 Fan Power versus Volume Characteristics for Different Control Methods (Courtesy of *Heating/Piping/Air Conditioning* 1984).

Comparing two methods of fan capacity control—inlet vane and variable-speed—the following issues arise. When reducing flow rate from full capacity, inlet vanes are initially efficient, but, as they progressively close, the reduction in flow rate is attributable more to throttling than to imparting initial spin to the air entering the impeller. Inlet vanes also impose about a 10% horsepower penalty because they obstruct the fan inlet. Variable-speed control maintains a high efficiency over the entire operating range and is therefore more energy conserving (Figure 5-6).

Variable-speed controls for centrifugal pumps have characteristics similar to those shown in Figure 5-6. The system implications of variable-speed pumping are discussed in Section 8.8. Additional information is contained in the chapters in the *ASHRAE Handbook—HVAC Systems and Equipment* dealing with Compressors; Condensers; and Motors, Motor Controls, and Variable-Speed Drives.

Like all solid-state devices, variable-speed drives are sensitive to differences in phase loads. Therefore, ensure that the phase differences are no greater than 10% on circuits incorporating such devices.

5.3 HEATING SOURCE EQUIPMENT

5.3.1 Boilers

Four principal types of boilers are commercially available to designers of heating plants for large buildings:

Type	Fuel
Sectional cast iron	Oil, gas, or dual fuel
Water tube	Oil, gas, or dual fuel
Fire tube	Oil, gas, or dual fuel

Electric resistance	Electricity

Oil, gas, and electricity are the energy sources most commonly used in new buildings. On rare occasions, coal, wood chips or hog fuel, biomass, or solid waste are used.

The principal reasons for selecting central boilers for heating are (1) to provide a plant at a convenient central location that can distribute a secondary heating medium (hot water or steam) throughout a building or a multi-building complex, (2) to take advantage of the diversity of individual zone loads, and (3) to centralize maintenance. The choice of fuel should be made on the basis of life-cycle cost, not on first cost alone. If the budget permits, in areas where the gas utility provides a firm and interruptible gas rate consider dual-fuel burners (oil/propane or gas) when selecting combustion boilers. Under normal conditions, the boilers would operate on cheaper interruptible gas. When the utility curtails this supply during periods of high demand, the operator would switch to the standby fuel (oil or propane). If recommending dual-fuel burners, inform the owner/operator of the importance of using the alternate fuel occasionally each season to ensure fuel system readiness.

The following issues should be considered when selecting a boiler type. Consider a sectional cast iron boiler if unskilled operators are anticipated. Cast iron is least susceptible to oxidation due to poor water treatment. Its principal operational concern is thermal shock. If heavy oil is the principal fuel, consider a fire tube boiler because it is easier to clear the tubes of soot (by rodding them), whereas a water tube boiler requires soot blowers in the exhaust gas stream. Use of these devices, however, is restricted by many communities. In choosing fuel burners, consider pressure-atomizing burners up to 500,000 Btu/h [145 kW]. Above this capacity, investigate air-atomizing, flame retention burners, which become favorable at 1-million Btu/h [295 kW] and above.

If electric boilers are considered, make a selection based upon more than just low first cost. Electric resistance boilers vary in cost, depending upon the power density of elements and the control steps. 50 W/in^2 is preferable to 100 W/in^2 [32 kW/mm^2 vs. 64 kW/mm^2] and a large number of control steps is also desirable. Modular boilers are often a reasonable choice in large installations. Care needs to be taken to match the boiler voltage to the available building system voltage, as actual output capacity is dependent upon operating voltage.

If a boiler plant is the heat source for a closed loop heat pump system, heat pump selection and water loop temperature controls are important. Excessive temperatures in the water loop can cause operational problems with closed loop heat pumps.

5.3.2 Furnaces

The purpose of a furnace is to directly heat air for space conditioning. Furnaces are seldom selected as the heat source for large buildings. They are often used, however, in small office buildings, residences, and industrial plants (in modular form).

5.3.3 Electric Resistance Heating

Electric resistance may provide the heat source for a boiler or furnace, as discussed above. Electric resistance heating is also often used in conjunction with local or terminal heating systems. The benefit of this particular application is the flexibility it offers in locating small units (baseboard convectors, unit heaters, or terminal duct heaters) in or near individual spaces as a means of zone control. In the case of in-duct heaters, flow sensors should be provided to disconnect the heater element whenever airflow stops in order to avoid a fire hazard.

Care must be exercised in selecting electric heaters (such as baseboard radiators) because they have a fixed output based upon their wattage. By comparison, hot water radiator output can be varied by changing operating water temperature or flow rate to increase capacity as necessary to deal with local load conditions.

While, at first glance, electric resistance heating may not appear to be economical, a careful examination of local electric utility rates can determine whether this impression is justified. Many utilities have offered incentives to all-electric buildings and rebates for thermal storage (Section 9.2) are frequently available. Since electric resistance heating is often the lowest first cost alternative, this option deserves a careful economic analysis. A guide to electric utility tariffs is given in Section 2.8.5.

5.3.4 Electric Heat Pumps

Heat pumps transfer heat extracted from one medium (air, water) to another medium via the vapor compression refrigeration cycle. Options include air-source, water-source, ground-source, and heat reclaim systems. Ground-source heat pumps are increasingly being used in green buildings because of their high efficiency. Another potential heat source for heat pumps is building exhaust air after it has first been utilized in a heat exchanger coupled with outdoor air intakes.

In large buildings, the perimeter-to-floor area ratio is relatively small compared to that in small buildings. Consequently, internal gains attributable to lights, computers, occupants, and other heat sources are high compared to transmission losses. During unoccupied periods, the building electrical capacity normally relegated to serving lighting and plug loads is available for perimeter heating by heat pumps supplemented with electric resistance heat.

Heat pumps need not meet 100% of the heating requirements of a building. Frequently, it is more practical to select only those areas easily served with heat pumps and to provide supplemental loop or point-of-use heat from another medium (electric resistance, gas) for areas such as entrance vestibules, etc. These and other heat pump uses are described in the chapter on Applied Heat Pump and Heat Recovery Systems in the *ASHRAE Handbook—HVAC Systems and Equipment*. Packaged terminal air conditioners, often employed for cooling core areas of large buildings, and water-source heat pumps, used in closed loops (see Chapter 7 of this manual), are described in the same *Handbook* volume.

5.4 HEAT TRANSFER EQUIPMENT

5.4.1 Condensers and Evaporators

These types of heat exchangers are described in more detail in the chapter on Condensers in the *ASHRAE Handbook—HVAC Systems and Equipment*. Only a few of the most important considerations relating to system design are mentioned here.

Shell-and-tube condensers are the most prevalent type in large building air conditioning applications. These are generally available in one, two, three, or four passes on the water-tube side, depending upon the flow of condenser water versus pressure drop. Frequently, a designer can choose between two-pass and three-pass or three-pass and four-pass with little difference in condenser performance and, by doing so, can select a water piping arrangement most suitable to the equipment room layout. Remember that large buildings generally require large-diameter condenser water pipes, which are both expensive and occupy a large volume of space. As a first cut, design heat exchangers with a 3 to 5°F [1.7 to 2.8°C] approach temperature. Shell-and-tube condensers require periodic cleaning of the inside of the tubes to maintain performance. Therefore, on large condensers, consider marine-type water boxes. They permit removal of condenser heads for tube cleaning without dismantling the condenser water piping. Integral mechanical cleaning systems are also an option.

In most urban atmospheres, condenser water circulated through open cooling towers can cause rapid scale build up on condenser tubes. It is inconvenient to shut down cooling systems for maintenance during the summer months, yet tube scale buildup peaks in late summer just when full capacity is needed. Consequently, consider specifying centrifugal chilling systems with a fouling factor of 0.001 rather than the typical commercial factor of 0.0005, because a 0.0005 fouling factor can be built up during the first weeks of the cooling season. Chemical water treatment systems can also be specified.

Most manufacturers of centrifugal or absorption chillers offer more than one condenser size. Due to competition, manufacturers are inclined to focus on lowest first cost in their initial product offer. Such equipment proposals often include a condenser that is marginally acceptable. A condenser with limited condenser surface is apt not to meet the equipment full-load capacity rating, which most likely will be needed at the time the condenser tubes are most fouled. Large shell-and-tube condensers are manufactured using extruded finned tubes with nearly the same ratio of primary (tube) and secondary (fin) surface. As a general rule, provide at least 6 ft^2 of total external surface (primary plus secondary) per ton [0.16 m^2 per kW] of full-load rating for conservative sizing of a condenser. Added condenser surface will pay dividends in two ways: longer tube-cleaning intervals (tube cleaning is labor intensive and therefore expensive) and lower condensing temperatures with associated reduced energy use. Life-cycle costing must be considered when specifying the size of evaporator and condenser surfaces.

In direct expansion systems, estimate about 1/15 hp per ton [0.014 kW/kW] of refrigeration for the condenser fan. Raising the evaporator temperature 1°F [0.6°C] decreases energy use by approximately 1.75%. Lowering the condensing temperature 1°F [0.6°C] produces approximately one-half of this saving.

Absorption systems are more sensitive to condenser tube fouling than compression systems, and the designer should either devote extra care to design for easy cleaning or allow for large fouling factors during analysis of equipment performance. Many systems can be retrofitted with on-line tube-cleaning equipment for low fouling factors. Some manufacturers offer such devices, which maintain a low fouling factor through the entire operating period. Sidestream sand filters have also performed well in cleaning condenser water.

5.4.2 Double-Bundle Heat Exchangers

Double-bundle condensers are an option for large heat pump and heat recovery systems. As the name implies, two water tube bundles are contained in a single shell. High-pressure refrigerant vapor discharged into the condenser shell from the compressor surrounds—and is in contact with—both tube bundles. The condensing temperature is based upon the number of active tubes. When a double-bundle condenser provides hot water (100 to 110°F [37.8 to 43.3°C]) for low-density fin-type baseboard (or unit) heaters or for hot water reheat coils, the water is circulated through the winter bundle of the condenser where it picks up the heat rejected from the condenser. When the evaporator-compressor portion of the system is called upon to provide more cooling than can be absorbed by the hot water bundle, the summer bundle is activated by circulating cooling tower water, which permits the rejection of the excess heat through the cooling tower. Close control is provided by bypassing a varied quantity of water around the cooling tower as needed to sustain the temperature of the hot water leaving the winter bundle (see Figure 5-7).

Fig. 5-7 Double-Bundle Condenser (Griffin 1974).

Double-bundle evaporators are used only rarely. An example is a double-bundle evaporator and a double-bundle condenser used with a well-water heat pump providing chilled water in summer and hot water in winter. In summer, chilled water is circulated through the summer bundle of the evaporator and well water is circulated through the summer bundle of the condenser (to reject the heat to the environment). Conversely, in winter, hot water is circulated through the winter bundle of the condenser and the well water is circulated through the winter bundle of the evaporator (to absorb heat from the well water).

Heat exchangers are often used in place of double-bundle condensers. This is less efficient, however, because it adds the penalty of an additional temperature difference. In lieu of a double-bundle condenser, an auxiliary condenser is added beside the main condenser to act as the winter bundle. With this arrangement, care should be taken in piping the compressor discharge into the condenser so that the superheat is picked up by the lead condenser, which should always be the winter bundle.

5.4.3 Plate Heat Exchangers

A plate-type heat exchanger can transfer heat from one liquid to another at high efficiency (with a 2 to 3°F [1.1 to 1.7°C] approach). This device consists of a series of gasketed, embossed metal plates (usually stainless steel) held between two rigid end frames by threaded rods and spanner nuts. The alternating plates create a heat exchanger with a cold medium on one side and a warm medium on the other side of the plate, allowing for counterflow in very narrow passages. Heat transfer rates are very high due to the large surface area that even a small plate heat exchanger provides and due to turbulence created by embossing the plates with corrugations, which forces the flow into a thin turbulent film that eliminates stagnant areas. Pressure drops are comparable to those of shell-and-tube heat exchangers because of the many parallel paths in plate heat exchangers. When cleaning is required, the exchanger is opened by unscrewing the spanner nuts and spreading the plates to permit wash-down by a hose and brushing of plate surfaces if needed. Some concerns have, however, been expressed about the collection of debris in plate heat exchangers.

The use of various types of heat exchangers in air-conditioning systems is increasing in a quest for improved energy efficiency, including the following applications:

- Cooling chilled water directly with cooling tower water in winter or whenever the temperature of the tower water is less than that of the chilled water return (see Section 9.3.9).

- Using heat exchangers in place of double-bundle condensers (or double-bundle evaporators) in heat pump applications.

- In solar thermal collector systems or other applications requiring heat exchange between glycol and water.

- Heat exchange between thermal storage systems (hot or cold) and circulating hot water or chilled water streams.

- Numerous opportunities for heat reclaim between various flow streams in a building.

5.4.4 Cooling Coils

Cooling coils are used to exchange heat (sensible and latent) between an airflow and water or refrigerant. Cooling coils are usually selected to perform within the following limits:

Entering air dry-bulb temperature	65 to 100°F [18.3 to 37.8°C]
Entering air wet-bulb temperature	60 to 85°F [15.6 to 29.4°C]

Air face velocity	300 to 600 fpm [1.5 to 3.0 m/s]
Refrigerant saturation temperature at coil outlet	30 to 50°F [-1.1 to 10.0°C]
Refrigerant superheat at coil outlet	6°F or more [3.3°C]—to prevent slugging the compressor with liquid
Refrigerant entering temperature	35 to 65°F [1.7 to 18.3°C]
Chilled-water quantity	1.2 to 2.4 gpm per ton [0.02 to 0.04 L/s per kW], equivalent to a water temperature rise of 20 to 10°F [11.1 to 5.6°C]
Chilled-water velocity	3 to 8 fps [0.9 to 2.4 m/s]
Chilled-water pressure drop	< 20 ft of water [60 kPa]

Coil face velocity is an important design consideration. In variable air volume systems, when air volume is reduced the coil face velocity will be reduced as well. Below about 300 fpm [1.5 m/s], laminar flow can occur—reducing the cooling capacity of the coil. Conversely coil velocities above 600 fpm [3.0 m/s] can cause moisture carryover. For special applications, however, the range in the above design variables may be exceeded (see the *ASHRAE Handbook—HVAC Systems and Equipment*).

5.4.5 Cooling Towers

Cooling towers are used to reject heat from a condenser to the outdoor air via the use of condenser water. For large building air-conditioning systems, vertical counterflow towers are most prevalent. They are equipped with either induced draft or forced draft fans. Condenser water systems are open systems in which air is continuously in contact with the water, and require a somewhat different approach to pump selection and pipe sizing than do closed loop water systems. Water contact with the air introduces impurities, which can result in continuous scaling and corrosion. Anticipate aging of the piping, which will result in a pressure drop that increases with time and higher-than-usual fouling factors. Condenser water systems without water-regulating valves do not necessarily require a strainer. Strainers provided in the cooling tower basin or at the pump will usually be adequate. Additional design considerations include:

- Size the tower to avoid windage drift of tower carryover, which might fall on adjacent buildings or parked cars (where cooling tower water can damage paint finishes).

- Provide water treatment and makeup systems, including water meters to permit deduction of makeup water from sanitary sewer charges that are based upon water consumption.

- Winterize the tower and allow for ready availability of freeze protection in the event of a sudden change in the weather.

- Do not locate cooling towers near outdoor air intakes to prevent excessive entering air wet-bulb temperatures and potential bacterial contamination of intake air (e.g., Legionella).

■ Do not locate cooling towers near the building facade to avoid staining it and to prevent the entry of noise into occupied spaces.

Typical design values (*per ton of refrigeration* [*per kW of refrigeration*]) include:

- 3.0 gpm condenser water for vapor-compression systems [0.05 L/s]
- 3.5 gpm condenser water for absorption systems [0.06 L/s]
- 300 cfm airflow[40 L/s]
- 1/15 fan hp for induced draft (draw-through) towers [0.014 kW]
- 1/7 fan hp for forced draft (blow-through) towers [0.03 kW].

Space requirements include:

- 8-ft [2.5-m] high towers: 1 ft^2 per 500 gross ft^2 [0.1 m^2 per 50 m^2] of building area
- higher towers: 1 ft^2 per 400 gross ft^2 [0.12 m^2 per 50 m^2] of building area.

Design of piping from the tower sump to the pump should be undertaken with caution. Place the sump level above the top of the pump casing to provide positive prime. Pitch all piping up, either to the tower or to the pump suction, to eliminate air pockets. Suction strainers should be equipped with inlet and outlet gages that indicate when cleaning is required. Normally, piping is sized to yield water velocities between 5 and 12 fps [1.5 and 3.7 m/s]. Use friction factors for aged or old pipes.

In cooling tower installations, ascertain that sufficient NPSH (net positive suction head—see Section 5.5.2) for the condenser water pump is provided, and design the connections so that the flow through each cooling tower has the same pressure drop. Install equalizing lines of proper size between towers; otherwise, one tower will fill and the other one will overflow. Winterize the tower piping if necessary or provide for drain-down when pumps shut off. In the latter case, the isolation valves must be located in a heated space. Provide valving or sump pump isolation so that any single cell can be serviced while the others are active.

5.4.6 Evaporative Condensers

Evaporative condensers are used to reject heat to the outdoor air by means of the evaporation of water. Water is sprayed on the condenser coils, and the resultant heat of vaporization, combined with any sensible heat effect, cools the liquid inside the coils. The residual spray water is usually recirculated. An evaporative condenser uses much less makeup water than a cooling tower in a conventional water-cooled condenser arrangement (approximately 1.6 to 2.0 gph/ton [0.5 to 0.6 mL/s per kW] by evaporation plus 0.5 to 0.6 gph/ton [0.15 to 0.18 mL/s per kW] for bleed water). Evaporative condensers are used primarily as refrigeration system condensers. In such applications, they should be located as close as possible to the refrigeration machinery to keep the refrigerant lines short. In the winter, the water can often be shut off with the dry coil providing adequate heat rejection. Evaporative condensers are also used to cool recirculating water in closed loop water-source heat pump systems (see Chapter 7).

Figure 5-8 shows two evaporative condenser types using city, well, or river makeup water. Piping should be sized in accordance with the principles outlined in Section 5.4.5 with flow velocities of 5 to 10 fps [1.5 to 3.0 m/s]. Where city water is used, a pump is usually not required since the water arrives under pressure. For well or river water, pumps may be necessary, in which case the procedure for pump sizing is generally that indicated below for cooling tower systems. For practical purposes, the velocity head is insignificant and can usually be ignored in these systems. Since evaporative condensers have little storage capacity, liquid receivers (see Section 5.2.1) should be installed downstream of such condensers to hold the refrigerant until it is required by the system.

Fig. 5-8 Evaporative Condenser Types.

5.5 PUMPS

5.5.1 General Characteristics

Pumps are used to circulate various liquids in an HVAC&R system. The different types of centrifugal pumps and their applications in HVAC&R systems are treated in the ASHRAE *Handbook—HVAC Systems and Equipment*. Centrifugal pumps are most prevalent in HVAC&R systems. Table 5-1 shows some of the effects of variations in system or operating parameters on pump performance. The use of pump curves for system design is explained in Section 8.8.

TABLE 5-1 Characteristics of Common Types of Pumps

	Positive Displacement Pumps			Centrifugal Pumps	
	Rotary	Piston	Radial	Mixed Flow	Axial Flow
Flow	Even	Pulsating	Even	Even	Even
Effect of Increasing Head:					
On flow	Negligible decrease	-----	Decrease	Decrease	Decrease
On bhp[a]	Increase	Increase	Decrease	Small decrease to large increase	Large increase
Effect of Closing					

Discharge Valve:					
On pressure	Can destruct unless relief valve is used	-----	Up to 30%	Consid-erable increase	Large increase
On bhp	Increases to destruction	-----	Decrease 50 to 60%	10% decrease to 80% increase	Increase 80 to 150%

[a] An increase in head with a centrifugal pump can reduce bhp, thus the reason for selecting "non-overloading" pumps that can cross higher horsepower curves on a reduction in pump head when the pump "runs out" along the pump curve.

5.5.2 Net Positive Suction Head

To eliminate cavitation, a certain minimum net positive suction head (NPSH) must be maintained at the inlet of a pump. The required NSPH for a specific pump is available from the manufacturer, either from catalog data or on request. Although usually given as a single number, NPSH increases with flow. The required NPSH can be considered as the pressure required to overcome pump inlet losses and to keep water flowing into the pump without the formation of vapor bubbles, which are the cause of cavitation. A piping system will produce an available NPSH that reflects its design and installation conditions, For satisfactory pump operation, the available NPSH must always exceed the required NPSH; if it does not, bubbles and pockets of vapor will form in the pump. The results will be a reduction in capacity, loss of efficiency, noise, vibration, and cavitation. The available NPSH in a system is expressed by:

available NPSH = $P_a + P_s + V^2/2g - h_{vpa} - h_f$ (5-1)

where

available NPSHA = net positive suction head available

P_a = atmospheric pressure at elevation of installation, ft [kPa]
P_s = water pressure at pump centerline, ft [kPa]
$V^2/2g$ = velocity pressure at pump centerline, ft [kPa]
h_{vpa} = absolute vapor pressure at pumping temperature, ft [kPa]
h = friction and entrance head losses in the suction piping, ft [kPa]

Net positive suction head is not normally a concern in closed systems. It also is not ordinarily a factor in open systems unless hot fluids are pumped, the suction lift is large, a cooling tower outlet and its pump inlet are at approximately the same elevation, or there is considerable friction in the pump suction pipe. Insufficient available NPSH can occur because of undersized piping, too many fittings, if a valve in the suction line is throttled, or if the mesh of a strainer on the suction side of a pump becomes clogged.

Example 5-2. The pump in Figure 5-9 has a NPSH of 22.4 ft [67 kPa] of water. The pressure loss in the suction piping is 4.6 ft of water [13.8 kPa], and L equals 12 ft [3.7 m]. Determine the additional lift available from the pump.

Solution: The additional lift equals the NPSH minus the suction head and piping pressure losses. Additional lift = 22.4 - 4.6 - 12 = 5.8 ft of water [17.3 kPa].

Fig. 5-9 Pump with Suction Lift.

5.6 VALVES

5.6.1 Valve Types

The following valve types are regularly used in HVAC systems.

Ball and butterfly valves operate from the fully open to the fully closed position with a 90° turn. They can be used for throttling and offer very low resistance to flow when fully open.

Balancing valves are provided for flow measurement and adjustment purposes.

Check valves prevent flow reversal. They can be spring-loaded to close before an actual reversal of flow occurs.

Gate valves are intended to be either open or fully closed and are not used for throttling. They offer low resistance to flow when fully open.

Globe valves are used for throttling and have high resistance to flow.

Pressure-reducing valves are used to maintain a constant downstream pressure under varying flow and pressure conditions.

Pressure-relief valves open, when system pressure reaches a preset level, to prevent overpressure in a system.

Control valves provide for automatic operations by means of an actuator mounted to the valve body which imparts a push-pull or a rotary motion to the valve stem. Control valves are available with electric, hydraulic (pneumatic), thermostatic, or self-operated actuators. A control valve provided with an automatic controller and sensor becomes part of an automated control system (see Chapter 10).

5.6.2 Valve Characteristics

The internal design of a valve determines its performance characteristics (percent flow versus percent stroke) as the valve is positioned from open to closed. Three typical characteristics—quick opening, linear, and equal percentage—are shown in Figure 5-10 at constant pressure drop. The *quick-opening* characteristic is usually applied where a process requires essentially on-off operation. For example, a preheat coil in an outdoor air duct may be operated by a quick-opening valve, the controller of which has a sensor in the outdoor air.

Fig. 5-10 Typical Valve Flow Characteristics.

A *linear* valve gives a uniform percentage change in flow for each percentage change in valve position. A typical application would be on a steam-heating coil operated by a controller whose sensor is on the discharge side of the coil. Since the coil output is uniform with steam flow, a linear valve provides a uniform change in temperature as the proportional controller increases or decreases the valve position to match the load to maintain a temperature setpoint.

An *equal-percentage* valve gives a very gradual change in flow for each percentage change in valve position near the closed position and a very rapid increase in flow near the fully open position. An example is the control of a hydronic heating or cooling coil. The nonlinear coil heat transfer characteristic (Figure 5-11a) is counteracted by the equal-percentage valve characteristic (Figure 5-11b), resulting in a nearly linear coil output. The flow-versus-valve-position characteristics (Figure 5-11c) assist the controller in matching the flow to the load and minimizing coil discharge fluctuations.

Fig. 5-11 Characteristics of Cooling or Heating Coil and Equal-Percentage Control Valve.

5.6.3 Valve Sizing

Sizing of control valves should be undertaken by determining the Cv (valve coefficient) required to provide the design flow conditions at full load, i.e., in the wide-open position. Manufacturers furnish values of Cv coefficients for each valve size. By definition, the Cv coefficient is the flow (in gpm) through a wide-open valve at a 1-psi pressure drop across the valve [or SI equivalents]. For water (with a specific gravity of 1), this reduces to:

$$Q = C_v (\Delta p)^{0.5} \qquad (5\text{-}2)$$

where,
Q = flow rate, gpm [m^3/s]
C_v = required valve coefficient
Δp = pressure drop across valve, psi [Pa]

For steam, this formula becomes:

$$W_s = 2.1\, C_v [(p_1 + p_2)(p_1 - p_2)]^{0.5} \qquad (5\text{-}3)$$

where

C_v = required valve coefficient
p_1 = valve inlet pressure, psia
p_2 = valve outlet pressure, psia
W_s = full-load steam flow rate, lb/h
Δp = pressure drop across valve, psi
[for SI applications: multiply g/s by 7.94 to get lb/h; multiply kPa by 0.145 to get psi]

5.6.4 Applications

Two-way valves vary the volume of water flowing through a heat transfer coil (Figure 5-12) and associated distribution circuit. Systems with two-way valve controls must provide for variable flow conditions. Constant system flow volume can be maintained by varying the volume of water flowing through the coil while allowing a bypass flow around the coil using a *three-way mixing* valve located on the leaving side of the coil (Figure 5-13). A more expensive *three-way diverting* valve, located on the entering side of the coil, varies the volume of water flowing through the coil and diverts flow around the coil as the coil flow is decreased.

The application of valves to air-and-water systems is treated in Chapter 7; their application to all-water systems is treated in Section 8.7.

Fig. 5-12 Two-Way Valve Controlling Hot Water Circuit.

Fig. 5-13 Three-Way Valve Controlling Chilled-Water Circuit.

5.7 PIPING

The design of piping systems is described in the chapter on Pipe Sizing in the *ASHRAE Handbook—Fundamentals*, while physical pipe sizes are given in the *ASHRAE Handbook—HVAC Systems and Equipment*. Assume a value of C = 100 for old or dirty pipe (i.e., in open systems), 140 for new steel pipe, and 150 for plastic or copper tube/pipe in the Hazen-Williams equation. Friction loss in hydronic systems is usually between 1 and 5 ft per 100 ft of pipe [10 to 50 kPa per 100 m], with an average of 2.5 ft per 100 ft [25 kPa per 100 m]. Typically, add 50% to the friction loss from the straight length of piping to account for fittings. The following rule is often used to account for the pressure drop in elbows: the equivalent straight length of pipe (in feet) for an elbow equals twice the nominal pipe diameter (in inches). Thus, a 1-in. [25 mm] elbow causes the same friction loss as 2 ft [0.6 m] of straight 1-in. [25 mm] pipe. Pumping power (for water, with a specific gravity of 1) is estimated from:

bhp = (gpm) (head of water, ft) / (3960) (pump efficiency) (5-4)
[kW = (L/s) (head of water, kPa) / (1002) (pump efficiency)]

with pump efficiencies ranging from 0.40 to 0.60 for small pump capacities and from 0.70 to 0.85 for larger pumps.

To calculate the appropriate size of compression/expansion tanks, use the table in the chapter on Hydronic Heating and Cooling System Design in the *ASHRAE Handbook—HVAC Systems and Equipment* that lists volumes in gallons per foot of pipe [L per m]. The water volumes in chillers, coils, heat exchangers, etc. must be added to those volumes.

Approximate parameters for air control expansion tanks are:

- 16% of total water volume for chilled water
- 20% of total water volume for hot water under standard sea level conditions and 30 to 40 ft [90 to 120 kPa] static head
- 12 psig [83 kPa] fill pressure, 30 psig [207 kPa] relief valve
- 200°F [93°C] hot water, 44°F [6.7°C] chilled water.

Tank sizes as estimated above can usually be cut in half if compression tanks, which contain a diaphragm that maintains separation between the air cushion and the water, are used.

If antifreeze liquids are used, the reduced thermal capacity (specific heat) and increased viscosity of these liquids compared to water (see the chapter on Secondary Coolants (Brines) in the *ASHRAE Handbook—Fundamentals*) must be considered. For example, the flow resistance of a 30% glycol aqueous solution is 15% greater than that of water. Drag-reducing agents may be able to offset these conditions.

The following velocities are recommended for closed (pressurized) piping systems:

Service	Velocity Range (fps) [m/s]
Pump discharge	8 to 12 [2.5 to 3.7]
Pump suction	4 to 7 [1.2 to 2.1]
Drain line	4 to 7 [1.2 to 2.1]
Header	4 to 12 [1.2 to 3.7]
Riser	3 to 10 [1.5 to 3.1]
General service	5 to 10 [1.5 to 3.1]
City water	3 to 7 [0.9 to 2.1]

Maximum recommended water velocities to minimize erosion are as follows:

Normal Annual Operating Hours	Maximum Recommended Water Velocity (fps) [m/s]
1500	12 [3.7]
2000	11.5 [3.5]
3000	11 [3.4]
4000	10 [3.1]
5000	9 [2.7]

8000	8 [2.4]

Pressure drop calculations for a typical 10,000-ft^2 [930-m^2] low-rise office building are shown in Table 5-2. The information is based upon manufacturers' data, except where noted. Both pressurized and open systems are considered.

TABLE 5-2 Sample Pressure Drop Calculations

Pressurized System

Item	Pressure drop, ft of water [kPa]
Boiler (assume 20 gpm [1.3 L/s])	1.0 [3.0]
Pump tree	6.0 [17.9] (from calculations)
Strainer (in tree)	8.0 [23.9]
Supply piping	12.0 [35.9] (from calculations, assume 400 ft [122 m] at 3 ft/100 ft [0.3 kPa/m])
Return piping	12.0 [35.9] (from calculations, assume 400 ft [122 m] at 3 ft/100 ft [0.3 kPa/m])
3-way valve	12.0 [35.9] (assume 50% of loop)
Total	51.0 [152.5]

Space heating hot water pump:

Bhp = (20 gpm) (51 ft) / (3960) (0.75) = 0.34 Bhp

[kW = (1.3 L/s) (152.5 kPa) / (1002) (0.75) = 0.26 kW]

Open System

Item	Pressure drop, ft of water [kPa]
Cooling tower return pipe	1.5 [4.5] (from calculations)
Indoor retention tank	0.5 [1.5]
Basket strainer (3.5 psi [24 kPa])	7.0 [20.9]
Pump tree	6.0 [17.9] (from calculations)
Chiller	6.0 [17.9]
Cooling tower supply pipe	1.5 [4.5] (from calculations)
3-way modulating valve	3.0 [8.9]
Cooling tower nozzle	12.0 [35.9]
Lift to tower nozzle	12.0 [35.9] (from building plans)
Total	49.5 [147.9]

Condenser water pump:

Bhp = (3 gpm per ton) (49.5 ft) / (3960) (0.75) = 0.05 Bhp/ton

[kW = (0.0538 L/s per kW) (147.9 kPa) / (1002) (0.75) = 0.0106 kW/kW]

5.7.1 Air Control and Venting

If air and other gases are not eliminated from a hydronic piping circuit, they may cause "air binding" in terminal heat transfer elements and noise and/or reduction in flow in the piping circuit. Piping can be run level, provided flow velocities in excess of 1.5 fps are maintained. Otherwise, piping should pitch up in the direction of flow to a high point containing an air vent or a runout to a room terminal unit. Vent high points in piping systems and terminal units with manual or automatic air vents. As automatic air vents may malfunction, provide valves at each vent to permit servicing without draining the system. Pipe the discharge of each vent to a point where water can be wasted into a drain or container to prevent damage to the surroundings. If a standard compression or expansion tank is used, free air contained in the circulating water can be removed from the piping circuit and trapped in the expansion tank by air-separation devices. If a diaphragm-type tank is used, vent all air from the system.

5.7.2 Drains, Shutoffs, and Strainers

Equip all low points with drains. Provide separate shutoff and drain valves for individual equipment so that an entire system does not have to be drained to service a single piece of equipment. Use strainers where necessary to protect equipment. Strainers placed in the suction of a pump must be large enough to avoid cavitation. Large separating chambers, which serve as main air-venting points and dirt strainers ahead of pumps, are available. Automatic control valves or spray nozzles operating with small clearances or openings require protection from pipe scale, welding slag, etc., which may readily pass through a pump and its protective separator. Individual fine mesh strainers are often required ahead of each control valve.

5.7.3 Thermometers and Gages

Include thermometers or thermometer wells to assist the system operator, the test and balance technician, the commissioning team, and for use in troubleshooting. Install permanent thermometers with correct scale range, easy-to-read display, and separable sockets at all points where temperature readings are regularly needed. Thermometer wells should be installed where readings will be needed only during start-up, balancing, and commissioning. Install gage cocks at points where pressure readings will be required. Gages permanently installed in a system tend to deteriorate due to vibration and pulsation, and cannot be counted on to provide reliable readings over the long haul. A single gage connected to both supply and return piping with appropriate valving will permit checking of pressure differential with an informal type of self-calibration.

5.7.4 Flexible Connectors

Flexible connectors at pumps and other machinery help reduce the transfer of vibration from equipment to piping. Flexible connectors also prevent damage caused by misalignment and thermal expansion/contraction of equipment piping. Vibration can be transmitted across a flexible connection, however, through flowing water, thereby reducing the effectiveness of the connector.

5.8 DUCTWORK

Recommended and maximum air velocities for ducts are given in Table 5-3. Three methods for the design of ductwork are in common use: equal friction, static regain, and the T-method. Refer to the chapter on Duct Design in the *ASHRAE Handbook—Fundamentals*, which explains all three methods in detail.

TABLE 5-3 Maximum and Recommended Air Velocities in Ductwork

Recommended velocities, fpm [m/s]

Location	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outdoor air intakes[a]	300 [1.5]	300 [1.5]	300 [1.5]
Filters[a]	250 [1.3]	300 [1.5]	350 [1.8]
Heating coils[a,b]	450 [2.3]	500 [2.5]	600 [3.1]
Cooling coils[a]	450 [2.3]	500 [2.5]	600 [3.1]
Air washers[a]	500 [2.5]	500 [2.5]	500 [2.5]
Fan outlets	1000–1600 [5.1-8.1]	1300–2000 [6.6-10.2]	1600–2400 [8.1-12.2]
Main ducts[b]	700–900 [3.6-4.6]	1000-1300 [5.1-6.6]	1200–1800 [6.1-9.1]
Branch ducts[b]	600 [3.1]	600–900 [3.1-4.6]	800–1000 [4.1-5.1]
Branch risers[b]	500 [2.5]	600–700 [3.1-3.6]	800 [4.1]

Maximum velocities, fpm [m/s]

Location	Residences	Schools, Theaters, Public Buildings	Industrial Buildings
Outdoor air intakes[a]	300 [1.5]	300 [1.5]	300 [1.5]
Filters[a]	300 [1.5]	500 [2.5]	500 [2.5]
Heating coils [a,b]	500 [2.5]	600 [3.1]	1000 [5.1]
Cooling coils[a]	450 [2.3]	500 [2.5]	600 [3.1]
Air washers	500 [2.5]	500 [2.5]	500 [2.5]
Fan outlets	1700 [8.6]	1500–2200 [7.6-11.2]	1700–2800 [8.6-14.2]
Main ducts[b]	800–1200 [4.1-6.1]	1100-1600 [5.6-8.1]	1300–2200 [6.6-11.2]
Branch ducts[b]	700-1000 [3.6-5.1]	800-1300 [4.1-6.1]	1000-1800 [5.1-9.1]
Branch risers[b]	650–800 [3.3-4.1]	800–1200 [4.1-6.1]	1000-1600 [5.1-8.1]

[a] These velocities are for total face area, not the net free area; other velocities in the table are for net free area.
[b] For low-velocity systems only.

The Duct Design chapter in the *ASHRAE Handbook—Fundamentals* and the *HVAC Systems—Duct Design* manual (SMACNA 1990) contain extensive tabulations of pressure losses in ducts. Manufacturers' data are also readily available. Roughness factors for other than smooth sheet metal ducts can also be found in these references. Internal linings may add 25 to 40% to air resistance, while flexible ducts may add 50%. The latter require special attention so that they remain round and do not collapse or become crushed.

Recommended cfm per square foot [L/s per m^2] values for different conditions are given in the *ASHRAE Pocket Guide* (2005). Minimum ventilation (outdoor air) requirements are given in ASHRAE *Standard 62* or in local codes, which, if they exist, supersede the standard. For supply air systems, use approximate pressure losses of:

- 0.08 in. of water per 100 1inear feet of duct [0.66 Pa per m] for quiet areas
- 0.10 in. of water per 100 1inear feet of duct [0.82 Pa per m] for ordinary areas
- 0.15 in. of water per 100 1inear feet of duct [1.21 Pa per m] for factory areas.

A pressure loss analysis for a typical 10,000-ft^2 [930-m^2] low-rise office building is shown in Table 5-4. Data are based upon manufacturers' data or are from SMACNA *HVAC Systems—Duct Design* except where noted.

TABLE 5-4 Sample Pressure Loss Calculations

Supply Air Fan

Item	Pressure drop, in. of water [Pa]
Outdoor air louver	0.05 [12.4]
Mixed-air damper plenum	0.05 [12.4]
Preheat coil	0.13 [32.3]
Filters (dirty)	0.75 [186.6]
Cooling coil (wet)	1.30 [323.4]
Heating coil	0.13 [32.3]
Fan inlet	0.50 [124.4]
VAV inlet vanes	0.75 [186.6]
Fan outlet	0.50 [124.4]
Primary ductwork	2.00 [497.6] (from calculations)
Terminal box	0.20 [49.8] (from calculations)
Air diffuser	0.10 [24.9]
Total	6.66 [1607.1]

Return Air Fan

Item	Pressure drop, in. of water [Pa]
Room/plenum/ceiling	0.04 [9.9]
Plenum drop	0.05 [12.4]
Duct inlet	0.03 [7.5]
Fan inlet	0.30 [74.6]
Ductwork	0.50 [124.4] (from calculations)
Fan inlet vanes	0.50 [124.4]
Fan outlet	0.10 [24.9]
Total	1.52 [378.1]

Exhaust Air Fan

Item	Pressure drop, in. of water [Pa]
Exhaust register	0.10 [24.9]
Fire damper	0.04 [9.9]
Ductwork	0.50 [124.4] (from calculations)
Total	0.64 [159.2]

A prudent designer will add a reasonable safety factor to these values to ensure that adequate fan capacity and power have been provided for the system.

Fan power for a ductwork system can be estimated as described in Section 4.5.2. Typical fan efficiencies range from 0.40 to 0.50 for small fans and from 0.55 to 0.60 for large fans.

Outdoor air (for ventilation or makeup) must equal exhaust air plus exfiltration at all times. Exhaust air fans will cause the infiltration of outdoor air unless supply/makeup fans supply a quantity of outdoor air equal to the exhausted plus exfiltrated air. Ignoring this fact may lead to frozen coils or sprinkler piping in cold climates. Buildings should be pressurized to reduce or prevent infiltration. It is customary, therefore, to supply approximately 5 to 10% more outdoor air than exhaust air to account for exfiltration.

Building static pressure controls need to reference outdoor ambient conditions. The outdoor air sensor must not be sensitive to weather or wind effects. Also, use a large chamber to dampen sudden variations in pressure readings.

Standard louvers, hoods, and other air intake openings are usually sized at 500 to 800 fpm [2.5 to 4.1 m/s] through the free area of the opening. Stormproof (drainable) louvers should be used to reduce the entry of rain/snow. Specification of louver free area must be coordinated with the architect. Install flashings at the exterior wall, and weep holes or a floor drain to carry away rain or melted snow entering the intake. In cold regions, a snow baffle may be required to direct fine snow particles to a low-velocity area below the dampers. At maximum airflow, overall resistance of the louver, dampers, and outdoor air intake duct should approximately equal the resistance of the return air systems.

Construct relief openings in large buildings similarly to outdoor air intakes but equip them with motorized or building pressure-operated backdraft dampers to prevent reversal of airflow caused by high wind pressures or building stack action when the automatic dampers are open. Detail self-acting dampers to prevent rattling.

5.9 REFERENCES

ASHRAE. 2005. *ASHRAE Pocket Guide for Air-Conditioning, Heating, Ventilation and Refrigeration.* American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. Atlanta, GA.

ASHRAE. 2005. *Handbook—Fundamentals.* American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. Atlanta, GA.

ASHRAE. 2004. *Handbook—HVAC Systems and Equipment.* American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. Atlanta, GA.

Griffin, C.W. 1974. *Energy Conservation in Buildings.* Construction Specifications Institute, Alexandria, VA.

Heating/Piping/Air Conditioning. 1984. May, p. 85.

ICC. 2003. *International Building Code*. Internationa1 Code Council. Falls Church, VA.

SMACNA. 1990. *HVAC Systems—Duct Design.* Sheet Metal and Air Conditioning Contractors' National Association, Inc. Chantilly, VA.

Trane. 1977. *Refrigeration Manual.* The Trane Company. LaCrosse, WI.

Zogg, R., K. Roth and J. Brodrick. 2005. "Using CHP Systems in Commercial Buildings." *ASHRAE Journal*, Vol. 47 (9), pp. 33-35.

Fig. 5-1 >>

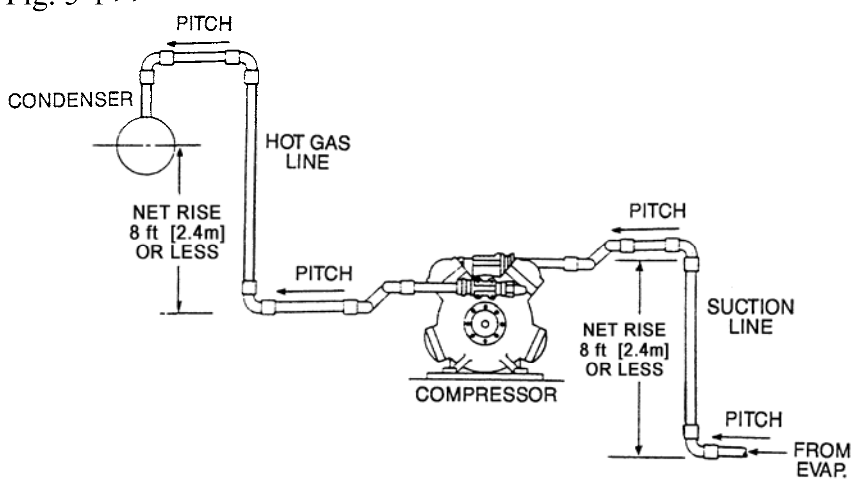


Fig. 5-2 >>

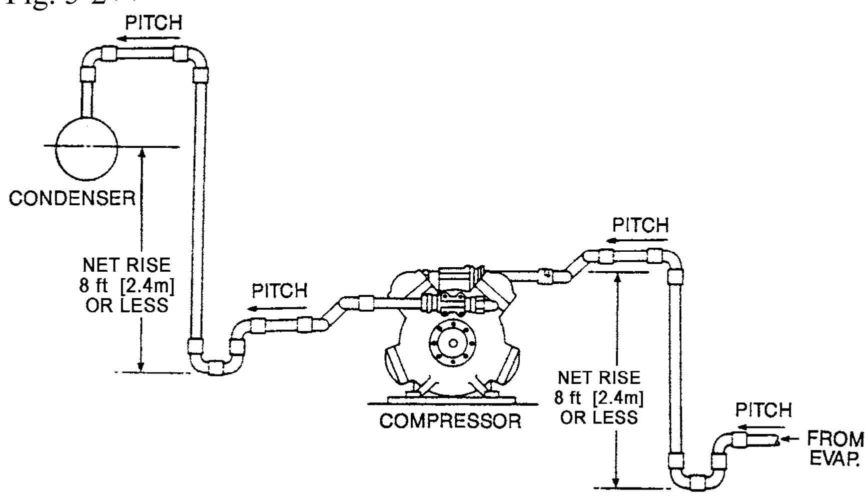

PITCH
CONDENSER
NET RISE
8 ft [2.4m]
OR LESS
PITCH
PITCH
COMPRESSOR
NET RISE
8 ft [2.4m]
OR LESS
PITCH
FROM
EVAP.

Fig. 5-3 >>

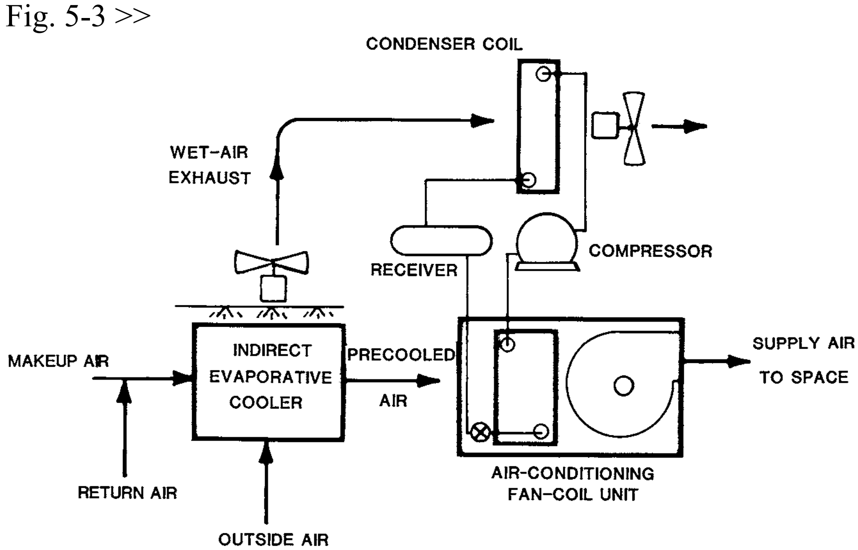

CONDENSER COIL
WET-AIR
EXHAUST
RECEIVER
COMPRESSOR
MAKEUP AIR
INDIRECT
EVAPORATIVE
COOLER
PRECOOLED
AIR
SUPPLY AIR
TO SPACE
RETURN AIR
OUTSIDE AIR
AIR-CONDITIONING
FAN-COIL UNIT

Fig. 5-4 >>

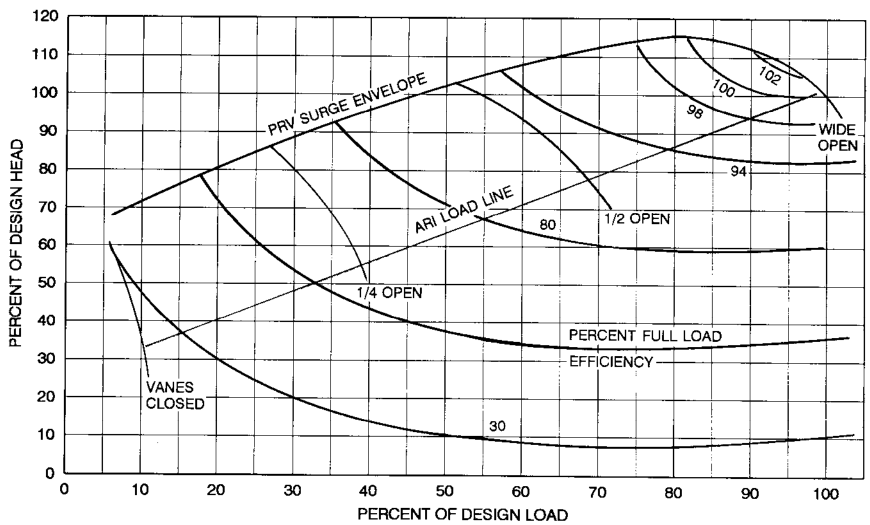


Fig. 5-5 >>

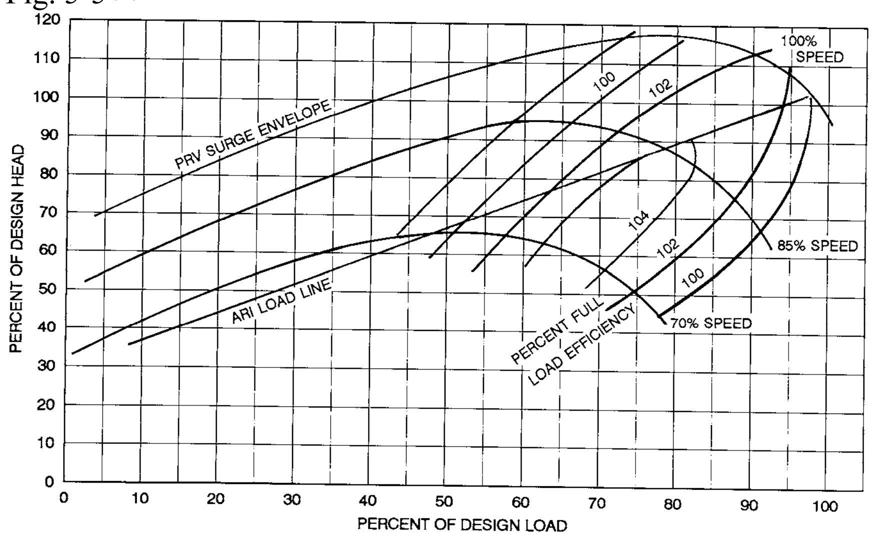


Fig. 5-6 >>

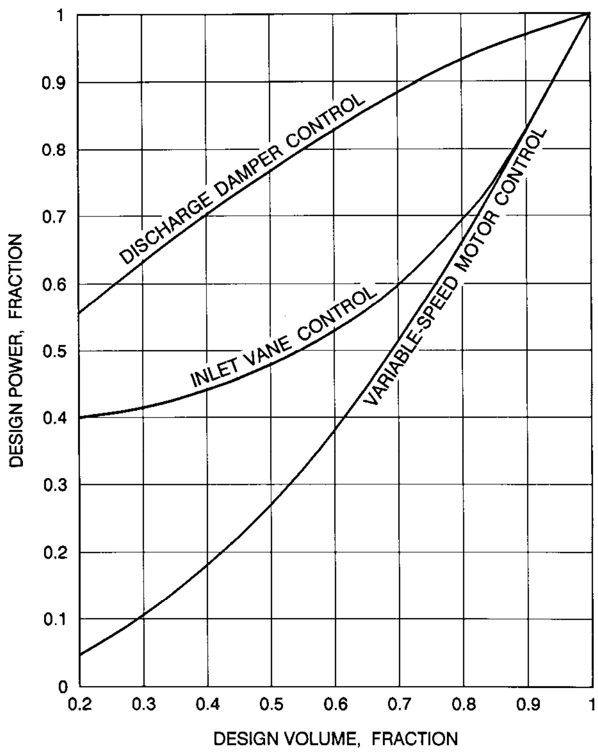

DISCHARGE DAMPER CONTROL
INLET VANE CONTROL
VARIABLE-SPEED MOTOR CONTROL
DESIGN POWER, FRACTION
DESIGN VOLUME, FRACTION
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1

Fig. 5-7 >>

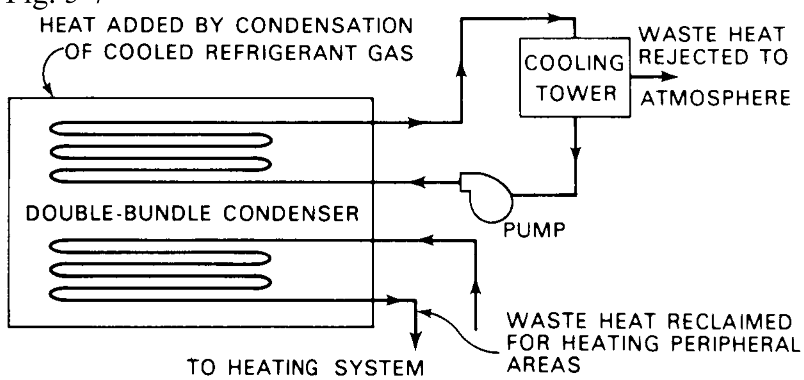

HEAT ADDED BY CONDENSATION
OF COOLED REFRIGERANT GAS
COOLING
TOWER
WASTE HEAT
REJECTED TO
ATMOSPHERE
DOUBLE-BUNDLE CONDENSER
PUMP
WASTE HEAT RECLAIMED
FOR HEATING PERIPHERAL
AREAS
TO HEATING SYSTEM

Fig. 5-8 >>

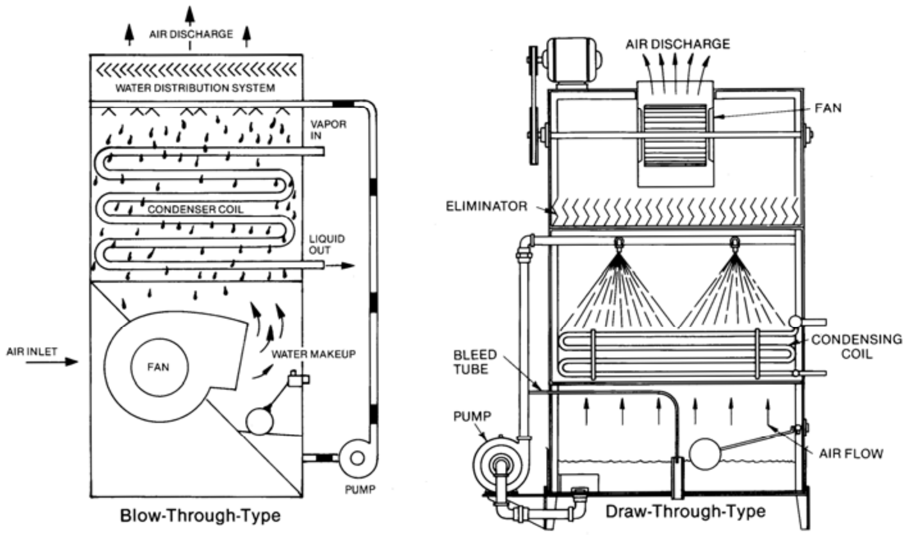


Fig. 5-9 >>

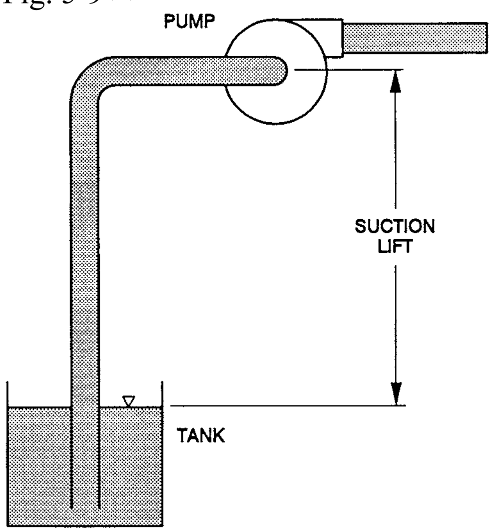


Fig. 5-10 >>

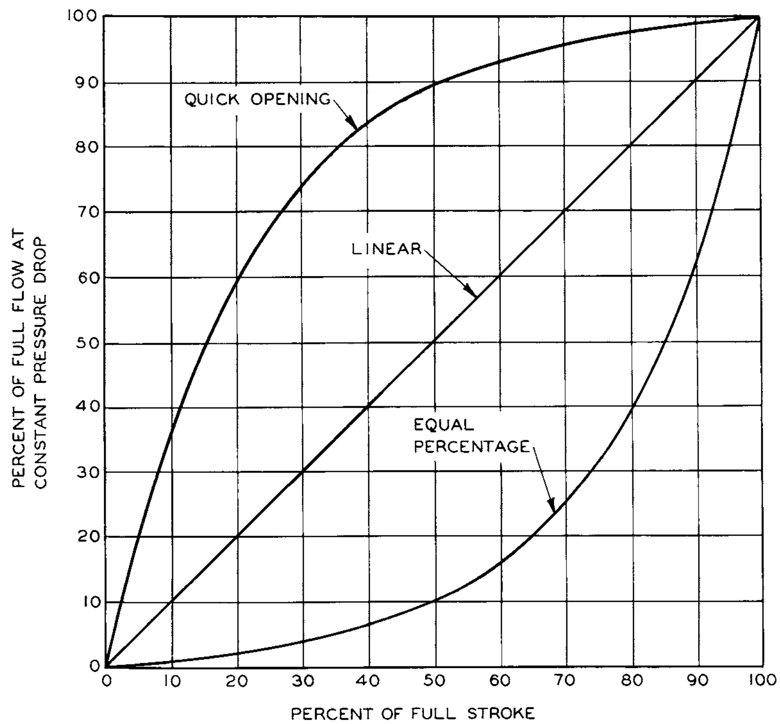


Fig. 5-11 >>

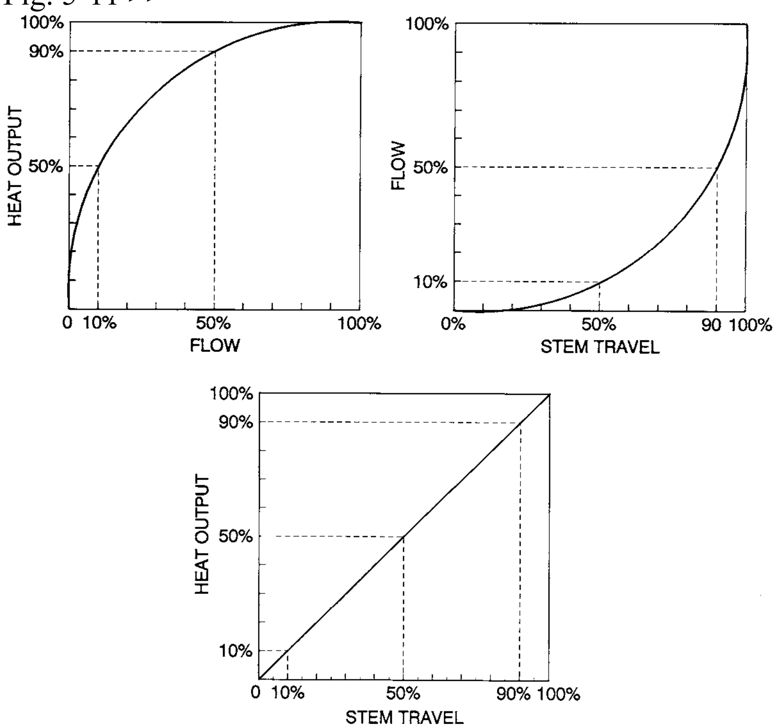


Fig. 5-12 >>

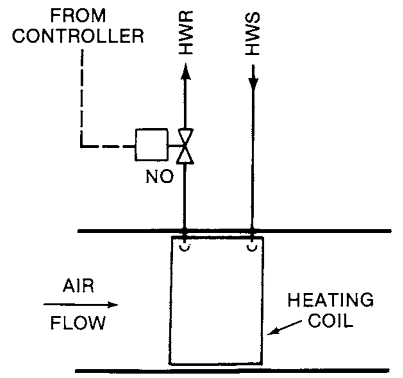
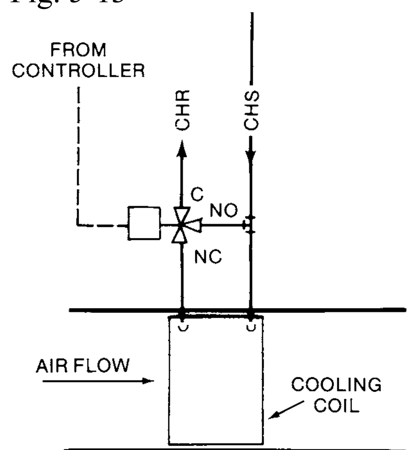

FROM
CONTROLLER
HWR
HWS
NO
AIR
FLOW
HEATING
COIL

Fig. 5-13 >>


FROM
CONTROLLER
CHR
CHS
C
NO
NC
AIR FLOW
COOLING
COIL