James B. Bradford et al. “HVAC Equipment and Systems”
Handbook of Heating, Ventilation, and Air Conditioning
Ed. Jan F. Kreider
Boca Raton, CRC Press LLC. 2001

4

HVAC Equipment and Systems

James B. Bradford
Schiller Associates, Inc.

John A. Bryant
Texas A&M University

Ellen M. Franconi
Schiller Associates, Inc.

Moncef Krarti
University of Colorado

Jan F. Kreider
Kreider & Associates, LLC

Dennis L. O'Neal
Texas A&M University

4.1 Heating Systems

Jan F. Kreider

This chapter discusses equipment used for producing heat from fossil fuels, electricity, or solar power. The emphasis is on design-oriented information, including system characteristics, operating efficiency, the significance of part load characteristics, and criteria for selecting from among the vast array of heat producing equipment available.

The heating plants discussed in this chapter are often called the *primary systems*. Systems intended to distribute heat produced by the primary systems are called *secondary systems* and include ducts and pipes, fans and pumps, terminal devices, and auxiliary components. Such secondary systems for heating and cooling are described in Chapter 4.3. The terms *primary* and *secondary* are equivalent to the terms *plant* and *system* used by some building analysts and HVAC system modelers.

The goal of this chapter is to have the reader understand the operation of various heat generation or transfer systems and their performance:

- Furnaces
- Boilers
- Heat pumps
- Heat exchangers
- Part load performance and energy calculations for each

The primary sources of heat for building heating systems are fossil fuels — natural gas, fuel oil, and coal. Under certain circumstances electricity is used for heat in commercial buildings although the economic penalties for so doing are significant. Solar radiation power can be converted to heat for commercial building applications, including perimeter zone heating and service water heating.

4.1.1 Natural Gas and Fuel Oil-Fired Equipment

This section describes fossil fuel-fired *furnaces and boilers* — devices which convert the chemical energy in fuels to heat. Furnaces are used to heat air streams that are in turn used for heating the interior of buildings. Forced air heating systems supplied with heat by furnaces are the most common type of residential heating system in the U.S. Boilers are pressure vessels used to transfer heat, produced by burning a fuel, to a fluid. The most common heat transfer fluid used for this purpose in buildings is water, in the form of either liquid or vapor. The key distinction between furnaces and boilers is that air is heated in the former and water is heated in the latter.

The fuels used for producing heat in boilers and furnaces include natural gas (i.e., methane), propane, fuel oil (at various grades numbered from 1 through 6), wood, coal, and other fuels including refuse-derived fuels. It is beyond the scope of this handbook to describe the design of boilers and furnaces or how they convert chemical energy to heat in detail. Rather we provide the information needed by HVAC designers for these two classes of equipment. Since boilers and furnaces operate at elevated temperatures (and pressures for boilers), they are hazardous devices. As a result, a body of standards has been developed to assure the safe operation of this equipment.

Furnaces

Modern furnaces use forced convection to remove heat produced within a furnace's firebox. There are many designs to achieve this; four residential classifications based on airflow type are shown in Figure 4.1.1. The *upflow* furnace shown in Figure 4.1.1a has a blower located below the firebox heat exchanger with heated air exiting the unit at the top. Return air from the heated space enters this furnace type at the bottom. The upflow design is used in full-sized mechanical rooms where sufficient floor-to-ceiling space exists for the connecting ductwork. This is the most common form of residential furnace.

Downflow furnaces (Figure 4.1.1b) are the reverse with air flowing downward as it is heated by passing over the heat exchanger. This design is used in residences without basements or in upstairs mechanical spaces in two-story buildings. *Horizontal* furnaces of the type shown in Figure 4.1.1c use a horizontal air flow path with the air mover located beside the heat exchanger. This design is especially useful in applications where vertical space is limited, such as in attics or crawl spaces of residences.

A combination of upflow and horizontal furnaces is available and is named the *basement* or low-boy furnace (Figure 4.1.1d). With the blower located beside the firebox, air enters the top of the furnace, is heated, and exits from the top. This design is useful in applications where head room is restricted.

The combustion side of the heat exchanger in gas furnaces can be at either atmospheric pressure (the most common design for small furnaces) or at super-atmospheric pressures produced by combustion air blowers. The latter are of two kinds, forced draft (blower upstream of combustion chamber) or induced draft (blower downstream of combustion chamber); furnaces with blowers have better control of parasitic heat losses through the stack. As a result, efficiencies are higher for such *power combustion furnaces.*

In addition to natural gas, liquefied propane gas (LPG) and fuel oil can be used as energy sources for furnaces. LPG furnaces are very similar to natural gas furnaces. The only differences between the two are energy content (1000 Btu/ft^3 for natural gas and 2500 Btu/ft^3 for propane) and supply pressure to the burner. Gas furnaces can be adapted for LPG use and vice versa in many cases. Fuel oil burner systems differ from gas burner systems owing to the need to atomize oil before combustion. The remainder of the furnace is not much different from a gas furnace, except that heavier construction is often used.

Other furnaces for special applications are also available. These include (1) unducted space heaters located within the space to be heated and relying on natural convection for heat transfer to the space;

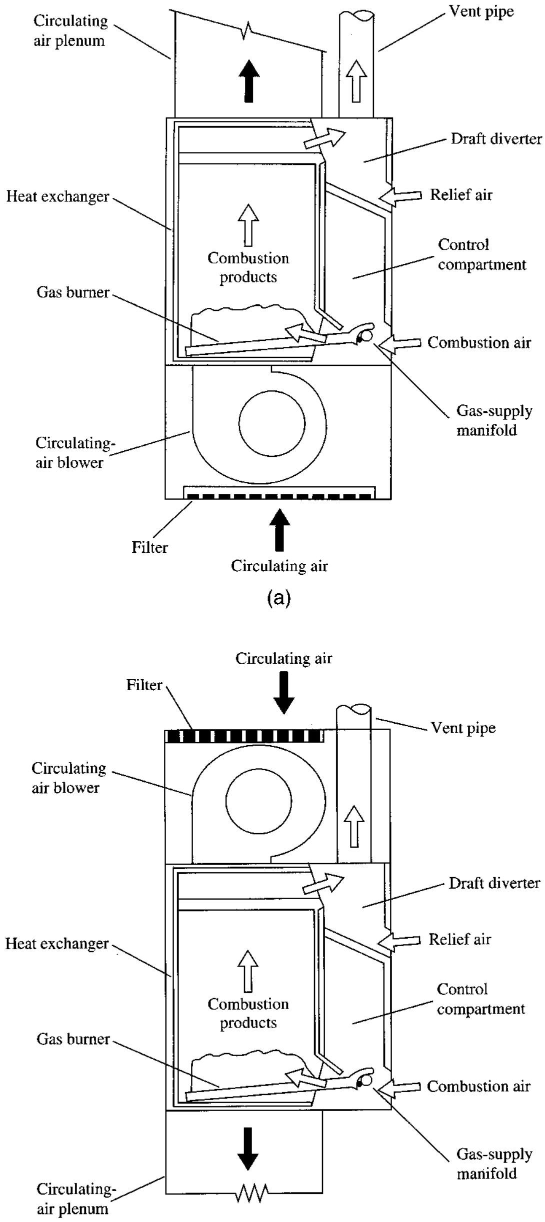

FIGURE 4.1.1 Examples of furnaces for residential space heating: (a) vertical, (b) downflow, (c) horizontal, and (d) low boy. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

(2) wall furnaces attached to walls and requiring very little space; and (3) direct fired unit heaters used for direct space heating in commercial and industrial applications. Unit heaters are available in sizes between 25,000 and 320,000 Btu/hr (7 to 94 kW).

On commercial buildings, one often finds furnaces incorporated into *package units* (or "rooftop units") consisting of air conditioners and gas furnaces (or electric resistance coils). Typical sizes of these units range from 5 to 50 tons of cooling (18 to 175 kW) with a matched to 50% over sized furnace. Smaller

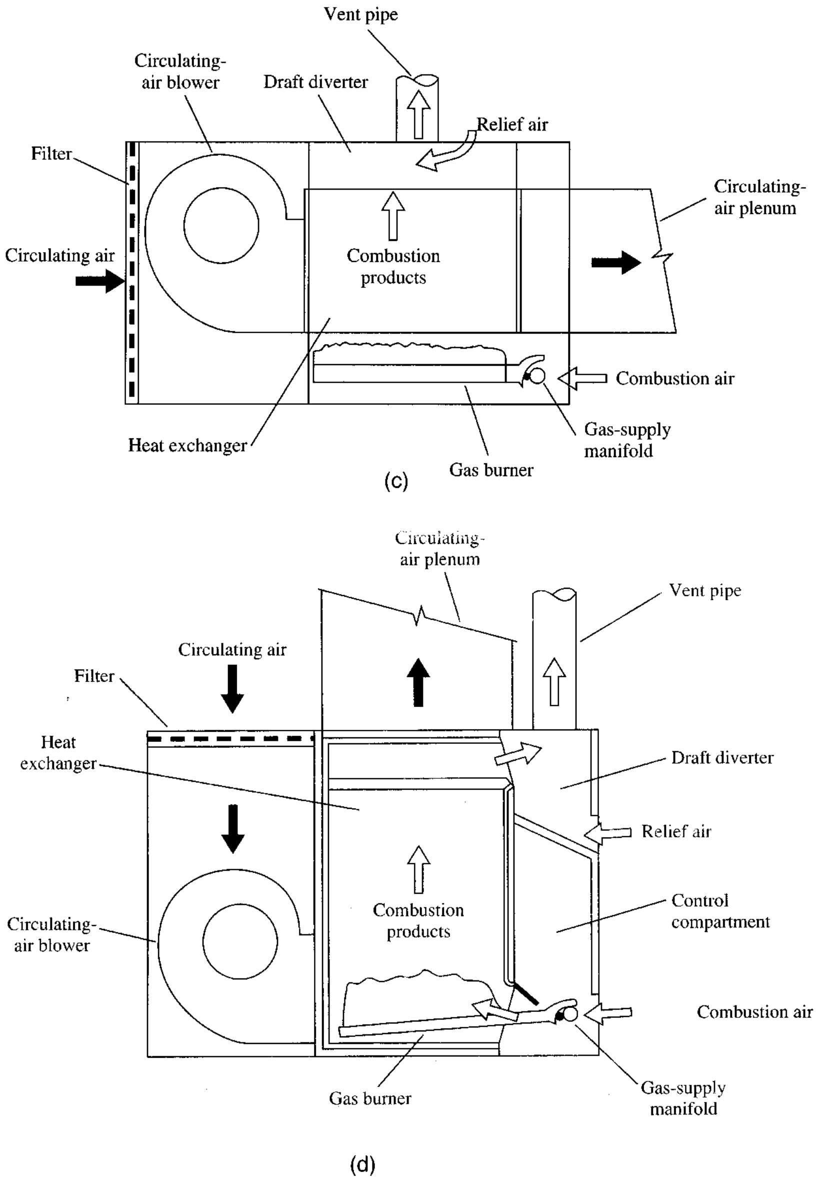

FIGURE 4.1.1 (*continued*)

units are designed to be used for a single zone in either the heating only or cooling only mode. Larger units above 15 tons (53 kW) can operate simultaneously in heating and cooling modes to condition several zones. In the heating mode, these commercial-sized package units operate with an air temperature rise of about 85°F (47 K).

Furnace Design and Selection for HVAC Applications

Selection of a furnace is straightforward once the fuel source and heat load (see Chapter 6.1) are known. The following factors must be accounted for in furnace sizing and type selection:

- *Design heat loss* of area to be heated — Btu/hr or kW
- Morning recovery capacity from night setback
- Constant *internal gains or waste heat recovery* that reduce the needed heat rating of a furnace

- *Humidification load* (see Chapters 4 and 7)
- *Fan and housing size* sufficient to accommodate air conditioning system
- *Duct heat losses* if heat so lost is external to the heated space
- *Available space* for furnace location

Residential furnaces are available in sizes ranging from 35,000 to 175,000 Btu/hr (10 to 51 kW). Commercial sizes range upward to 1,000,000 Btu/hr (300 kW).

Economic criteria including initial cost and life cycle operating cost must be considered using the techniques of Chapter 3.2 to make the final selection. Although high efficiency may cost more initially, it is often worthwhile to make the investment when the overall economic picture is considered. However, in many cases first cost is the primary determinant of selection. In these cases, the HVAC engineer must point out to the building owner or architect that the building lifetime penalties of using inexpensive but inefficient heating equipment are considerable, many times the initial cost difference.

The designer is advised to avoid the customary tendency to oversize furnaces. An oversized furnace operates at less efficiency than a properly sized one due to the penalties of part load operation. If a proper heat load calculation is done (with proper attention to the recognized uncertainties in infiltration losses and warm-up transients), only a small safety factor should be needed, for example, 10%. The safety factor is applied to account for heat load calculation uncertainties and possible future, modest changes in building load due to usage changes. Oversizing of furnaces also has other penalties, including excessive duct size and cost, along with poorer control of comfort due to larger temperature swings in the heated space.

Furnace Efficiency and Energy Calculations

The steady state efficiency η_{furn} is defined as the ratio of fuel supplied less flue losses, all divided by the fuel supplied:

$$\eta_{\text{furn}} = \frac{\dot{m}_{\text{fuel}} h_{\text{fuel}} - \dot{m}_{\text{flue}} h_{\text{flue}}}{m_{\text{fuel}} h_{\text{fuel}}} \tag{4.1.1}$$

in which the subscripts identify the fuel input and flue gas exhaust mass flow rates and enthalpies h. Gas flows are usually expressed in ft^3/hr (l/s). To find the mass flow rate, one must know the density which in turn depends on the gas main pressure. The ideal gas law can be used for such calculations. Efficiency values are specified by the manufacturer at a single value of fuel input rate.

This instantaneous efficiency is of limited value in selecting furnaces owing to the fact that furnaces often operate in a cyclic, part load mode where instantaneous efficiency may be lower than that at peak operating conditions. Part load efficiency is low since cycling causes inefficient combustion, cyclic heating and cooling of furnace heat exchanger mass, and thermal cycling of distribution ductwork. A more useful performance index is the *Annual Fuel Utilization Efficiency* (AFUE) which accounts for other loss mechanisms over a season. These include stack losses (sensible and latent), cycling losses, infiltration, and pilot losses (ASHRAE Equipment, 1996). An ASHRAE standard (103-1982R) is used for finding the AFUE for residential furnaces.

Table 4.1.1 shows typical values of AFUE for residential furnaces. The table shows that efficiency improvements can be achieved by eliminating standing pilots, by using a forced draft design, or by condensing the products of combustion to recover latent heat normally lost to the flue gases. Efficiency can also be improved by using a vent damper to reduce stack losses during furnace off periods. Although this table is prepared using residential furnace data, it can be used for commercial-sized furnaces as well. Few data have been published for commercial systems because it has not been mandated by law as it has been for residential furnaces. The AFUE has the shortcoming that a specific usage pattern and equipment characteristics are assumed. The next section discusses a more accurate method for finding annual performance of heat-producing primary systems.

The AFUE can be used to find annual energy consumption directly from its definition below. The fuel consumption during an average year Q_{yr} is given by

TABLE 4.1.1 Typical Values of AFUE for Furnaces

Type of gas furnace	AFUE, %
1. Atmospheric with standing pilot	64.5
2. Atmospheric with intermittent ignition	69.0
3. Atmospheric with intermittent ignition and automatic vent damper	78.0
4. Same basic furnace as type 2, except with power vent	78.0
5. Same as type 4 but with improved heat transfer	81.5
6. Direct vent with standing pilot, preheat	66.0
7. Direct vent, power vent, and intermittent ignition	78.0
8. Power burner (forced-draft)	75.0
9. Condensing	92.5
Type of oil furnace	**AFUE, %**
1. Standard	71.0
2. Same as type 1 with improved heat transfer	76.0
3. Same as type 2 with automatic vent damper	83.0
4. Condensing	91.0

Source: From ASHRAE. With permission.

$$Q_{\text{fuel, yr}} = \frac{Q_{yr}}{\text{AFUE}} \qquad \text{MMBtu/yr (GJ/yr)} \tag{4.1.2}$$

Where Q_{yr} is the annual heat load. Using this approach, it is a simple matter to find the savings one might expect, on the average, by investing in a more efficient furnace.

Example 1 Energy saving using a condensing furnace

A small commercial building is heated by an old atmospheric type, gas furnace. The owner proposes to install a new pulse type (condensing) furnace. If the annual heat load Q_{yr} on the warehouse is 200 GJ, what energy saving will the new furnace produce?

Assumptions: AFUE is an adequate measure of seasonal performance and furnace efficiency does not degrade with time.

Find: $\Delta Q_{fuel} = Q_{fuel,old} - Q_{fuel,new}$

Lookup values: AFUEs from Table 4.1.1

$$\text{AFUE}_{old} = 0.645 \qquad \text{AFUE}_{new} = 0.925$$

Solution

Equation 4.1.1 is used to find the solution. The energy saving is given by

$$\Delta Q_{fuel} = Q_{yr}\left(\frac{1}{\text{AFUE}_{old}} - \frac{1}{\text{AFUE}_{new}}\right)$$

Substituting the tabulated values for AFUE we have

$$\Delta Q_{fuel} = 200\left(\frac{1}{0.645} - \frac{1}{0.925}\right) = 93.9 \text{ GJ/yr}$$

The saving of energy using the modern furnace is substantial, almost equivalent to 50% of the annual heating load.

In addition to energy consumption, the designer must also be concerned with a myriad of other factors in furnace selection. These include:

- Air side temperature rise — affects duct design and air flow rate
- Air flow rate — affects duct design
- Control operation — for example, will night or unoccupied day/night setback be used or not? Is fan control by thermal switch or time delay relay?
- Safety issues — combustion gas control, fire hazards, high temperature limit switch

4.1.2 Boilers

A boiler is a device made from copper, steel, or cast iron to transfer heat from a combustion chamber (or electric resistance coil) to water in either the liquid phase, vapor phase, or both. Boilers are classified both by the fuel used and by the operating pressure. Fuels include gas, fuel oils, wood, coal, refuse-derived fuels, or electricity. This section focuses on fossil fuel fired boilers.

Boilers produce either hot water or steam at various pressures. Although water does not literally boil in hot water "boilers," they are called boilers, nevertheless. Steam is an exceptionally effective heat transport fluid due to its very large heat of vaporization and coefficient of heat transfer, as noted in Chapter 2.1.

Pressure classifications for boilers for buildings are

- *Low Pressure:* Steam boilers with operating pressures below 15 psig (100 kPa). Hot water boilers with pressures below 150 psig (1000 kPa); temperatures are limited to 250°F (120°C).
- *High Pressure:* Steam boilers with operating pressures above 15 psig (100 kPa). Hot water boilers with pressures above 150 psig (1000 kPa); temperatures are above 250°F (120°C).

Heat rates for steam boilers are often expressed in lb_m of steam produced per hour (or kW). The heating value of steam for these purposes is rounded off to 1000 Btu/lb_m. Steam boilers are available at heat rates of 50 to 50,000 lb_m of steam per hour (15 to 15,000 kW). This overlaps the upper range of furnace sizes noted in the previous section. Steam produced by boilers is used in buildings for space heating, water heating, and absorption cooling. Water boilers are available in the same range of sizes as are steam boilers: 50 to 50,000 MMBtu/hr (15 to 15,000 kW). Hot water is used in buildings for space and water heating.

Since the energy contained in steam and hot water within and flowing through boilers is very large, an extensive codification of regulations has evolved to assure safe operation. In the U.S. the ASME Boiler and Pressure Vessel Code governs construction of boilers. For example, the Code sets the limits of temperature and pressure on low pressure water and steam boilers listed above.

Large boilers are constructed from steel or cast iron. Cast iron boilers are modular and consist of several identical heat transfer sections bolted and gasketed together to meet the required output rating. Steel boilers are not modular but are constructed by welding various components together into one assembly. Heat transfer occurs across tubes containing either the fire or the water to be heated. The former are called *fire-tube boilers* and the latter *water-tube boilers*. Either material of construction can result in equally efficient designs. Small, light boilers of moderate capacity are sometimes needed for use in buildings. For these applications, the designer should consider the use of copper boilers.

Figure 4.1.2 shows a cross-section of a steam boiler of the type used in buildings.

Boiler Design and Selection for Buildings

The HVAC engineer must specify boilers based on a few key criteria. This section lists these but does not discuss the internal design of boilers and their construction. Boiler selection is based on the following criteria:

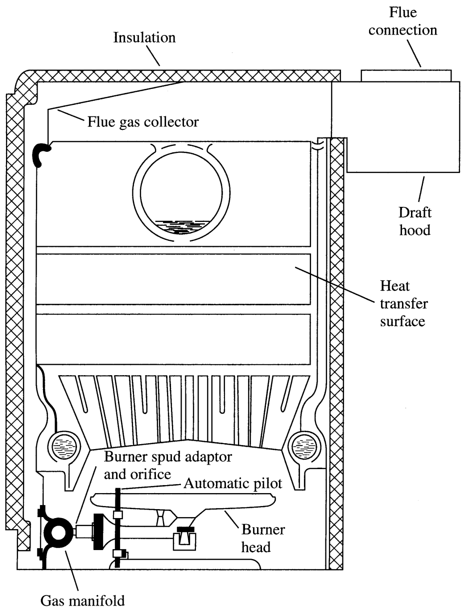

FIGURE 4.1.2 Boiler cross-sectional drawing showing burner, heat exchanger, and flue connection. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

- *Boiler fuel* — type, energy content, heating value including altitude effects if gas fired (no effect for coal or fuel oil boilers).
- *Required heat output* — net output rating in MMBtu/hr (kW)
- Operating pressure and working fluid
- Efficiency and part load characteristics
- *Other* — space needs, control system, combustion air requirements, safety requirements, ASME code applicability

The boiler heat output required for a building is determined by summing the *maximum heating requirement* of all zones or loads serviced by the boiler during peak demand for steam or hot water and adding to that (1) parasitic losses including piping losses and (2) initial loop fluid warm-up. Simply adding all of the *peak heating unit capacities* of all the zones in a building can result in an oversized boiler since the zones do not all require peak heating simultaneously. The ratio of the total of all zone loads under peak conditions to the total heating capacity installed in a building is called the *diversity.*

Additional boiler capacity may be needed to recover from night setback in massive buildings. This transient load is called the *pickup* load and must be accounted for in both boiler and terminal heating unit sizing.

Boilers are often sized by their *sea-level input* fuel ratings. Of course, this rating must be multiplied by the applicable efficiency to determine the gross output of the boiler. In addition, if a gas boiler is not to be located at sea level, the effect of altitude must be accounted for in the rating. Some boiler designs use a forced draft burner to force additional combustion air into the firebox to offset part of the effect of altitude. Also, enriched or pressurized gas may be provided at high altitude so that the heating value per unit volume is the same as at sea level. If no accommodation to altitude is made, the output of a gas boiler drops by approximately 4% per 1000 ft (13% per km) of altitude above sea level. For example, a gas boiler located in Denver, Colorado (5000 ft, 1500 m) will have a capacity of only 80% of its sea level rating.

Table 4.1.2 shows the type of data provided by manufacturers for the selection of boilers for a specific project. Reading across the table, the fuel input needs are first tabulated for the 13 boiler models listed. The fifth column is the sea level boiler output at the maximum design heat rate. The next four columns

TABLE 4.1.2 Example of Manufacturer's Boiler Capacity Table

	IBR burner capacity				Net IBR ratings								
Boiler unit number, steam, or water (1)	Light oil, gal/h (2)	Gas, kBtu/h (3)	Min. gas press. req'd., in WG (4)	Gross IBR output, Btu/h (5)	Steam, ft²/h (6)	Steam, Btu/h (7)	Water, Btu/h (8)	Net heat transfer area, ft² H_2O (9)	Boiler hp (10)	Net firebox volume, ft³ (11)	Stack gas volume, ft³/min (12)	Positive pressure in firebox in WG (13)	IBR chimney size vent dia., in (14)
▲486°F●	6.30	882	5.5	720,000	2,250	540,100	626,100	4,175	21.5	11.02	395	0.34	10
▲586°F●	8.25	1,155	7.0	940,000	2,940	705,200	817,400	5,450	28.1	14.45	517	0.35	10
▲686°F●	10.20	1,428	5.5	1,160,000	3,625	870,200	1,008,700	6,725	34.6	18.08	640	0.35	10
▲786°F●	12.15	1,701	6.0	1,380,000	4,355	1,044,700	1,200,000	8,000	41.2	21.61	762	0.36	12
▲886°F●	14.10	1,974	5.0	1,600,000	5,115	1,227,900	1,391,300	9,275	49.6	25.14	884	0.37	12
▲986°F●	16.05	2,247	6.0	1,820,000	5,875	1,409,800	1,582,600	10,550	54.3	28.67	1,006	0.38	14
▲1086°F●	18.00	2,520	6.5	2,040,000	6,600	1,583,900	1,773,900	11,825	60.9	32.20	1,128	0.39	14
▲1186°F●	19.95	2,793	7.0	2,260,000	7,310	1,754,700	1,965,200	13,100	67.5	35.73	1,251	0.40	14
▲1286°F●	21.95	3,073	7.0	2,480,000	8,025	1,925,500	2,156,500	14,375	74.1	39.26	1,376	0.41	14
▲1386°F●	23.90	3,346	6.5	2,700,000	8,735	2,096,300	2,347,800	15,650	80.6	42.79	1,498	0.42	14
▲1486°F●	25.90	3,626	7.5	2,920,000	9,445	2,267,100	2,539,100	16,925	87.2	46.32	1,623	0.43	16
▲1586°F●	27.85	3,899	7.5	3,140,000	10,160	2,437,900	2,730,400	18,200	93.8	49.85	1,746	0.44	16
▲1686°F●	29.75	4,165	8.5	3,350,000	10,835	2,600,900	2,913,000	19,420	100.1	53.38	1,865	0.45	16

Note: 1 bhp = 33,475 Btu/h = 9.8 kW.
Source: From Rabl, A. and Kreider, J.F., *Heating and Coling of Buildings*, McGraw-Hill, New York, 1994. With permission.

convert the heat rate to steam and hot water production rates. The following column expresses heat rate in still a different way, using units of boiler horsepower (= 33,475 Btu/hr or 9.81 kW). The final four columns provide information needed for designing the combustion air supply system and the chimney.

A rule of thumb to check boiler selection in heating climates in the U.S. is that the input rating (columns two and three of the table, for example) in Btu/hr expressed on the basis of *per heated square foot* of building is usually in the range of one third to one fifth of the design temperature difference (difference between indoor and outdoor, winter design temperature). For example, if the design temperature difference for a 100,000 square foot building is 80°F, the boiler input would be expected to be in the range between 1.6 MMBtu/hr ([80/5] × 100,000 ft²) and 2.7 MMBtu/hr ([80/3] × 100,000 ft²). The difference between the two depends on the energy efficiency of the building envelope and its infiltration controls. Boiler efficiency also has an effect on this design check.

Proper control of boilers in response to varying outdoor conditions can improve efficiency and occupant comfort. A standard feature of boiler controls is the *boiler reset* system. Since full boiler capacity is needed only at peak heating conditions, better comfort control results if capacity is reduced with increasing outdoor temperature. Capacity reduction of zone hot water heating is easy to accomplish by simply reducing the water temperature supplied by the boiler. An example reset schedule might specify boiler water at 210°F at an outdoor temperature of –20°F and at 70°F a water temperature of 140°F. This schedule is called a 1:1 schedule since for every degree rise in outdoor temperature the boiler output drops by 1.0°F.

Auxiliary Steam Equipment

Steam systems have additional components needed to provide safety or adequate control in building thermal systems. This section provides an overview of the most important of these components including steam traps and relief valves.

Steam traps are used to separate both steam condensate and noncondensable gases from live steam in steam piping systems and at steam equipment. Steam traps "trap" or confine steam in heating coils, for example, while releasing condensate to be revaporized again in the boiler. The challenge in trap selection is to assure that the condensate and gases are removed promptly and with little to no loss of live steam. For example, if condensate is not removed from a heating coil, it will become waterlogged and have

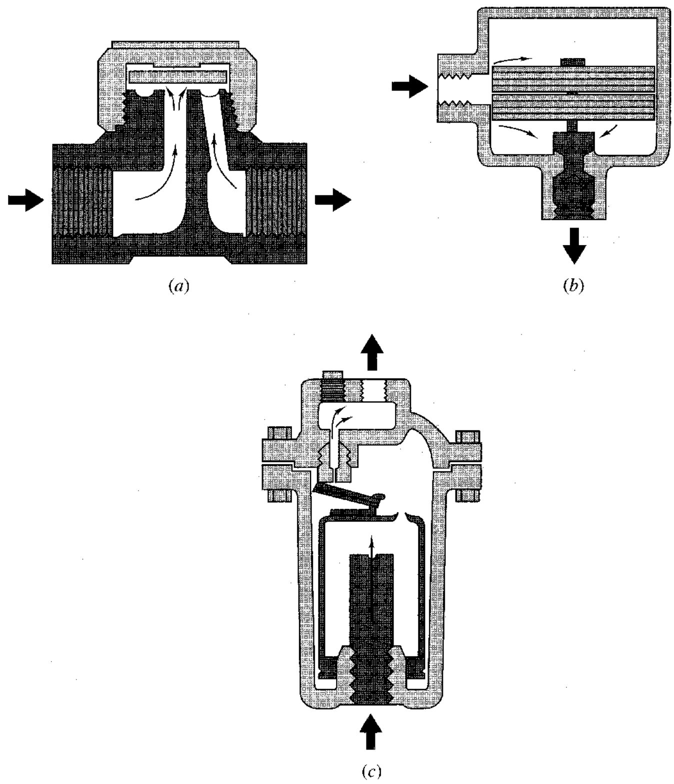

FIGURE 4.1.3 Steam traps. (a) Disc trap; (b) thermostatic trap; (c) mechanical, inverted bucket trap. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

much reduced heating capacity. A brief description of the most common types of traps which should be used in HVAC applications follows.

Thermodynamic traps are simple and inexpensive. The most common type, the *disc trap*, is shown in Figure 4.1.3a. This type of trap operates on kinetic energy changes as condensate flows through and flashes into steam within the trap. Steam flashed (i.e., converted from hot liquid to vapor) from hot condensate above the disc holds the trap closed until the disc is cooled by cooler condensate. Steam line pressure then pushes the disc open. It remains open until all cool condensate has been expelled and hot condensate is again present and flashes again to close the valve. The disc action is made more rapid by the flow of condensate beneath the disc; the high velocities produce a low pressure area there in accordance with Bernoulli's equation and the disc slams shut. These traps are rugged and make a characteristic clicking sound, making operational checking easy. They can stick open if a particle lodges in the seat. This design has relatively high operating cost due to its live steam loss.

Thermostatic traps use the temperature difference between steam and condensate to control condensate flow. One type of thermostatic trap is shown in Figure 4.1.3b. The bimetal unit within the housing opens the valve as condensate cools, thereby allowing condensate to exit the trap. Significant subcooling of the condensate is needed to open the valve and operation can be slow. Other more complex designs have more rapid response and reduced need for subcooling. The bimetal element can be replaced with a bellows filled with an alcohol/water mixture permitting closer tracking of release setting as steam temperature changes. A trap that has a temperature/pressure characteristic greater than the temperature/pressure of saturated steam will lose live steam, whereas a trap with a *T*/*p* characteristic lying below the steam curve will build up condensate. The ideal trap has an opening *T*/*p* characteristic identical with the *T*/*p* curve of saturated steam.

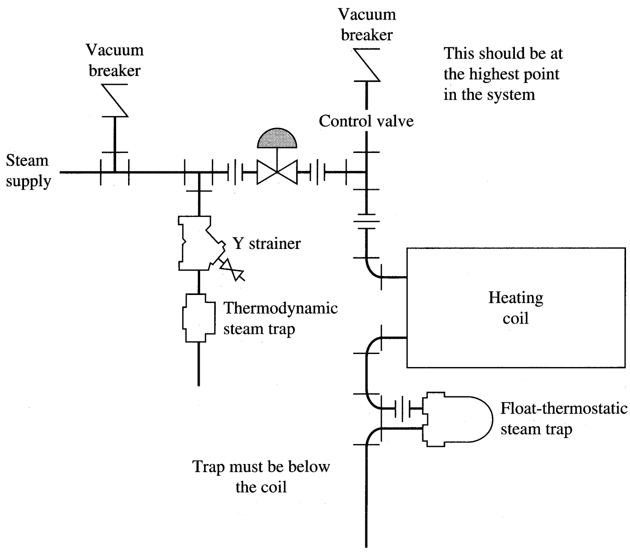

FIGURE 4.1.4 Piping arrangement for heating coil steam trap application. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

Mechanical traps operate on the density difference between condensate and live steam to displace a float. Figure 4.1.3c shows one type of mechanical trap — the *inverted-bucket trap* — that uses an open, upside down bucket with a small orifice. Steam flowing with the condensate (that fills the housing outside of the bucket) fills the inverted bucket and causes it to float since the confined steam is less dense than the liquid water surrounding the bucket. Steam bleeds through the small hole in the bucket and condenses within the trap housing. As the bucket fills with condensate it becomes heavier and eventually sinks and opens the valve. Steam pressure forces condensate from the trap. The design of this trap continuously vents noncondensable gases, although the capacity for noncondensable gas flow (mostly air) rejection is limited by the size of the small hole in the top of the bucket. This hole is limited in size by the need to control parasitic steam loss through the same hole. Dirt can block the hole causing the trap to malfunction. The trap must be mounted vertically. An inverted bucket trap has significantly smaller parasitic live steam losses than the thermodynamic disc trap.

Steam traps are used to drain condensate from steam headers and from equipment where condensing steam releases its heat to another fluid. Steam piping is sloped so that condensate flows to a collecting point where it is relieved by the trap. At equipment condensate collection points, the trap is placed below the equipment where the condensate drains by gravity. Figure 4.1.4 shows a typical trap application for both purposes. The left trap drains the header, and the right trap drains the condensate produced in the heating coil.

Selection of traps requires knowledge of the condensate rejection rate (lb_m/hr, kg/s) and the suitability of various trap designs to the application. Table 4.1.3 summarizes the applications of the three types of traps discussed above (Haas, 1990).

The operating penalties for malfunctioning steam traps (clogged, dirty, or corroded) can dwarf the cost of a trap because expensive heat energy is lost if live steam is lost from malfunctioning traps. One of the first things to inspect in an energy audit of a new or existing steam system is the condition of the traps. For example, if steam is produced in a gas fired boiler of typical efficiency, a 0.25 in (0.64 cm) orifice in a steam trap will lose about $2000 worth of steam in a year. (The cost of gas in this example is $3.00/thousand ft^3 [$0.11/m^3], the usual units used by utilities; this converts to approximately $3.00/million Btu or $2.84/GJ).

A pressure *relief valve* is needed to control possible overpressure in boilers for safety reasons. Valves are specified by their ability to pass a given amount of steam or hot water at the boiler outlet condition. This

TABLE 4.1.3 Operating Characteristics of Steam Traps

System needs	Thermodynamics	Float-thermostatic	Inverted bucket
Maximum pressure (psig)	1740	465	2755
Maximum capacity (lb/h)	5250	100,000	20,500
Discharge temperature, °F	Hot	Hot	Hot
	(Close to saturated-steam temperature)		
Discharge	On/off	Continuous	On/off
Air venting	Good	Excellent	Fair
Dirt handling	Fair	Good	Good
Freeze resistance	Good	Poor	Poor
Superheat	Excellent	Poor	Fair
Waterhammer	Excellent	Fair	Excellent
Varying load	Good	Excellent	Good
Change in psi	Good	Excellent	Fair
Backpressure	Maximum 80%	Good	Good
Usual failure	Open	Closed/air vent open	Open

Source: From Rabl, A. and Kreider, J.F., *Heating and Coling of Buildings*, McGraw-Hill, New York, 1994. With permission.

dump rate can either be specified in units of mass per time or in units of energy flow per time. Pressure relief valves must be used wherever heat can be added to a confined volume of water. Water could become confined in the piping of an HVAC system, for example, if automatic control valves failed closed or if isolation valves were improperly closed by a system operator. Not only boilers must be protected, but also heat exchangers and water pipe lengths that are heated externally by steam tracing or solar heat. The volume expansion characteristics of water can produce tremendous pressures if heat is added to confined water. For example, water warmed by only 30°F (17°C) will increase in pressure by 1100 psi (7600 kPa). The method for sizing boiler relief valves is outlined in Wong (1989). The discharge from boiler relief valves must be piped to a drain or other location where injury from live steam will be impossible.

Combustion Calculations — Flue Gas Analysis

The combustion of fuel in a boiler is a chemical reaction and as such is governed by the principles of stoichiometry. This section discusses the combustion of natural gas (for our purposes assumed to be 100% methane) in boilers as an example of fuel burning for heat production. It also outlines how the flue gas from a boiler can be analyzed to ascertain the efficiency of the combustion process. Continuous monitoring of flue gases by a building's energy management system can result in early identification of boiler combustion problems. In a new building, one should test a boiler to determine its efficiency as installed and to compare output to that specified by the designer.

Combustion analysis involves using the basic chemical reaction equation and the known composition of air to determine the composition of flue gases. The inverse problem, finding the precombustion composition, is also of importance when analyzing flue gases. The chemical reaction for stoichiometric combustion of methane is

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \tag{4.1.3}$$

Recalling that the molecular weights are

Hydrogen (H_2): 2
Methane (CH_4): 16
Oxygen (O_2): 32
Carbon Dioxide (CO_2): 44
Water (H_2O): 18

we can easily determine that 4.0 lb of oxygen per lb of methane are required for complete combustion. Since air is 23% oxygen by weight, 17.4 lb_m (or kg) of air per lb_m (or kg) of fuel are required, theoretically.

It is easy to show that on a volumetric basis (recall Avogadro's law which states that one mole of any gas at the same temperature and pressure occupies the same volume) the equivalent requirements are 2.0 ft^3 of oxygen per ft^3 of methane for complete combustion. This oxygen requirement is equivalent to 8.7 ft^3 of air per ft^3 of methane. A rule of thumb to check the preceding calculation is that 0.9 ft^3 of air are required for 100 Btu of fuel heating value (about 0.25 m^3 of air per MJ of heating value). For example, the heating value of natural gas is about 1000 Btu/ft^3 requiring 9 ft^3 of air according to the above rule. This compares well with the value of 8.7 ft^3 previously calculated.

Combustion air is often provided in excess of this amount to guarantee complete combustion. Incomplete combustion yields toxic carbon monoxide (CO) in the flue gas. This incomplete combustion is to be avoided not only as energy waste but as air pollution. The amount of excess air involved in combustion is usually expressed as the *excess air fraction* $f_{\text{exc air}}$:

$$f_{\text{exc air}} = \frac{\text{air supplied} - \text{stoichiometric air}}{\text{stoichiometric air}} \tag{4.1.4}$$

In combustion calculations for gaseous fuels, the air amounts in Equation 4.1.4 are usually expressed on a volumetric basis, whereas for all other fuels a mass basis is used.

The amount of excess air provided is critical to the efficiency of a combustion process. Excessive air both reduces combustion temperature (reducing heat transfer rate to the working fluid) and results in excessive heat loss through the flue gases. Insufficient excess air results in incomplete combustion and loss of chemical energy in the flue gases. The amount of excess air provided varies with the fuel and with the design of the boiler (or furnace). Recommendations of the manufacturer should be followed. The optimum excess air fraction is usually between 10 and 50%.

Flue gas analysis is a method of determining the amount of excess air in a combustion process. This information can be used to find an approximate value of boiler efficiency. Periodic, regular analysis can provide a trend of boiler efficiency with time, indicating possible problems with the burner or combustion equipment in a boiler or furnace. Flue gas analysis is often expressed as the volumetric fraction of flue gases — oxygen, nitrogen, and carbon monoxide. If these three values are known, the excess air (%) can be found from (ASHRAE, 1997)

$$f_{\text{exc air}} = \frac{O_2 - 0.5CO}{0.264N_2 - (O_2 - 0.5CO)} \tag{4.1.5}$$

in which the chemical symbols represent the volume fractions in the flue gas analysis expressed in %. The following example indicates how this expression is used.

Example 2 Flue gas analysis

The volumetric analysis of flue gas from combustion of methane in a gas boiler is measured to be

10.5% carbon dioxide
3.2% oxygen
86.3% nitrogen
0% carbon monoxide

Find the amount of excess air. Is it within the recommended range suggested above?

Equation 4.1.5 will be used as follows:

$$f_{\text{exc air}} = \frac{3.2\% - (0.5 \times 0\%)}{0.264(86.3\%) - [3.2\% - (0.5 \times 0\%)]} = 0.163$$

The excess air is 16.3%, within the 10 - 50% range above.

TABLE 4.1.4 Stoichiometric and Excess Air Values of CO_2 for Combustion of Common Fossil Fuels

Type of fuel	Theoretical or maximum CO_2, %	CO_2 at given excess-air values 20%	40%	60%
Gaseous fuels				
Natural gas	12.1	9.9	8.4	7.3
Propane gas (commercial)	13.9	11.4	9.6	8.4
Butane gas (commercial)	14.1	11.6	9.8	8.5
Mixed gas (natural and carbureted water gas)	11.2	12.5	10.5	9.1
Carbureted water gas	17.2	14.2	12.1	10.6
Coke oven gas	11.2	9.2	7.8	6.8
Liquid fuels				
No. 1 and No. 2 fuel oil	15.0	12.3	10.5	9.1
No. 6 fuel oil	16.5	13.6	11.6	10.1
Solid fuels				
Bituminous coal	18.2	15.1	12.9	11.3
Anthracite	20.2	16.8	14.4	12.6
Coke	21.0	17.5	15.0	13.0

Source: From Rabl, A. and Kreider, J.F., *Heating and Coling of Buildings*, McGraw-Hill, New York, 1994. With permission.

The efficiency of a steam boiler can be found from field measurements by

$$\eta_{boil} = \frac{\dot{Q}_{steam}}{\dot{m}_{fuel}(HHV)} \tag{4.1.6}$$

where

$\dot{Q}_{steam}$ is the steam output rate, Btu/hr (kW)
$\dot{m}_{fuel}$ is the fuel supply rate, lbm/hr (kg/s)
HHV is the higher heating value of the fuel, Btu/lb (kJ/kg)

The previous discussion described the combustion of methane and at what rate air is to be supplied for proper combustion. Of course, many other fuels are used to fire boilers. Table 4.1.4 contains data which can be used to quickly estimate the excess air from a flue gas analysis for other fuels.

Coal and fuel oil contain carbon and hydrogen along with sulfur, the combustion of all of which produce heat. However, sulfur oxide formed during combustion is a corrosive acid if dissolved in liquid water. In order to avoid corrosion of boilers and stacks, liquid water must be avoided anywhere in a boiler by maintaining sufficiently high stack temperatures to avoid condensation. (Stainless steel stacks and fireboxes provide an alternative solution since they are not subject to corrosion, but they are very costly.) In addition, sulfur oxides are one of the sources of acid rain. Therefore, their emissions must be carefully controlled.

Boiler Efficiency and Energy Calculations

A simpler, overall efficiency equation can be used for boiler energy estimates if the steam rate required for using Equation 4.1.6 is not measurable:

$$\eta_{boil} = \frac{HHV - losses}{HHV} \tag{4.1.7}$$

The loss term includes five parts:

1. Sensible heat loss in flue gases
2. Latent heat loss in flue gases due to combustion of hydrogen
3. Heat loss in water in combustion air
4. Heat loss due to incomplete combustion of carbon
5. Heat loss from unburned carbon in ash (coal and fuel oil)

Boilers can be tested for efficiency in laboratories and rated in accordance with standards issued by the Hydronics Institute (formerly IBR, the Institute of Boiler and Radiator Manufacturers, and the SBI, the Steel Boiler Institute), the American Gas Association (AGA), and other industry groups. In addition to cast iron boiler ratings, IBR ratings are industry standards for baseboard heaters and finned-tube radiation. The ratings of the Hydronics Institute apply to steel boilers (IBR and SBI are trademarks of the Institute). SBI and IBR ratings apply to oil- and coal-fired boilers while gas boilers are rated by the AGA.

As noted earlier, efficiency under specific test conditions has very limited usefulness in calculating the annual energy consumption of a boiler due to significant drop off of efficiency under part load conditions. For small boilers (up to 300 MBtu/hr [90 kW]), the U.S. Department of Energy has set a method for finding the AFUE (defined in the earlier section on furnaces). Annual energy consumption must be known in order to perform economic analyses for optimal boiler selection.

For larger boilers, data specific to a manufacturer and an application must be used in order determine annual consumption. Efficiencies of fossil fuel boilers vary with heat rate depending on their internal design. If the boiler has only one or two firing rates, the continuous range of heat inputs needed to meet a varying heating load is achieved by cycling the boiler on and off. However, as load decreases, efficiency decreases since the boiler spends progressively more and more time in transient warm-up and cool-down modes during which relatively little heat is delivered to the load. At maximum load, the boiler cycles very little and efficiency can be expected to be near the rated efficiency of the boiler. Part load effects can reduce average efficiency to less than half of the peak efficiency. Of course, for an oversized boiler, average efficiency is well below the peak efficiency since it operates at part load for the entire heating season. This operating cost penalty persists for the life of a building, long after the designer who oversized the system has forgotten the error.

To quantify part load effects we define the *part-load ratio*, PLR (a quantity between 0 and 1), as

$$\mathrm{PLR} \equiv \frac{\dot{Q}_o}{\dot{Q}_{o,\mathrm{full}}} \tag{4.1.8}$$

where

$\dot{Q}_o$ is the boiler heat output at part load; Btu/hr, kW
$\dot{Q}_{o,\mathrm{full}}$ is the rated heat output at full load; Btu/hr, kW

It is not practical to calculate from basic principles how boiler *input* depends on the value of PLR since the processes to be modeled are very complex and nonlinear. The approach used for boilers (and other heat producing equipment in this chapter) involves using test data to calculate the boiler input needed to produce an output $\dot{Q}_o$. If efficiency were constant and if there were no standby losses, the function relating input to output would merely be a constant, the efficiency. For real equipment the relationship is more complex. A common function used to relate input to output (i.e., to PLR) is a simple polynomial (at least for a boiler) such as

$$\frac{\dot{Q}_i}{\dot{Q}_{i,\mathrm{full}}} = A + B(\mathrm{PLR}) + C(\mathrm{PLR})^2 + \ldots \tag{4.1.9}$$

where

$\dot{Q}_i$ is the heat input required to meet the part load level quantified by PLR
$\dot{Q}_{i,\text{full}}$ is the heat input at rated full load on the boiler

The first term of Equation 4.1.9 represents standby losses, for example, those resulting from a standing pilot light in a gas boiler. Since the part load characteristic is not far from linear for most boilers, a quadratic or cubic expression is sufficient for annual energy calculations. Part load data are not as readily available as are standard peak ratings. If available, the data may often be in tabular form. The designer will need to make a quick regression of the data to find A, B, and C, using commonly available spreadsheet or statistical software in order to be able to use the tabular data for annual energy calculations as described in the following.

The remainder of this section examines a particularly simple application of a boiler — building space heating — to see how important part load effects can be. The *annual* energy input $Q_{i,\text{yr}}$ of a space heating boiler can be calculated from the following basic equation:

$$Q_{i,\text{yr}} = \int_{yr} \frac{\dot{Q}_o(t)}{\eta_{\text{boil}}(t)} dt \tag{4.1.10}$$

where

η_{boil} is the boiler efficiency — a function of time since the load on the boiler varies with time
$\dot{Q}_o(t)$ is the heat load on the boiler which varies with time as well, Btu/hr (kW)

The argument of the integral is just the instantaneous, time-varying energy input to the boiler. However, since the needed output — not the input — is usually known as a result of building load calculations, the form in Equation 4.1.10 is that practically used by designers. The time dependence in this expression is determined in turn by the temporal variation of load on the boiler as imposed by the HVAC system in response to climatic, occupant, and other time-varying loads.

A simple case is a boiler used solely for space heating. As described in Chapter 6.1, the heating load is determined first to order by the difference between indoor and outdoor temperature; all characteristics of the building's load and use remaining fixed. Therefore, the heat rate in Equation 4.1.10 is determined by outdoor temperature if the interior temperature remains constant. In this very simple case one could replace the integral in Equation 4.1.10 with a sum utilizing the bin approach as follows:

$$Q_{i,\text{yr}} = \sum_{j=0}^{N} = \frac{\dot{Q}_o(T_j) n_j(T_j)}{\eta_{\text{boil}}(T_j)} \tag{4.1.11}$$

where

$\eta_{\text{boil}}(T_j)$ is the efficiency of the boiler in a given ambient temperature bin j; the efficiency depends strongly but indirectly on ambient temperature T_j since the load, which determines PLR, depends on temperature.
$\dot{Q}_o(T_j)$ is the boiler load (i.e., building heat load) which depends on ambient temperature as described above.
$n_j(T_j)$ is the number of hours in the temperature bin j for which the value of efficiency and heat input apply.

This expression assumes that the sequence of hours during the heating season is of no consequence. The following example illustrates how bin weather data can be used to take proper account of part load efficiency of a boiler used for space heating.

TABLE 4.1.5 Summary of Solution for Example 3 — Boiler Energy Analysis

Calculating Annual Boiler Energy Use							
Bin range, °F	Bin size, h	Heating load, kBtu/h	PLR	$\dot{Q}_i$, kBtu/h	Boiler effic.	Fuel used, MBtu	Net output, MBtu
55 to 60	762	0	0.00	875	0.000	667	0
50 to 55	783	500	0.07	1844	0.271	1444	391
45 to 50	716	1000	0.14	2750	0.364	1969	716
40 to 45	665	1500	0.21	3594	0.417	2390	997
35 to 40	758	2000	0.29	4375	0.457	3316	1516
30 to 35	713	2500	0.36	5094	0.491	3632	1782
25 to 30	565	3000	0.43	5750	0.522	3249	1695
20 to 25	399	3500	0.50	6344	0.552	2531	1396
15 to 20	164	4000	0.57	6875	0.582	1127	656
10 to 15	106	4500	0.64	7344	0.613	778	477
5 to 10	65	5000	0.71	7750	0.645	504	325
0 to 5	80	5500	0.79	8094	0.680	647	440
–5 to 0	22	6000	0.86	8375	0.716	184	132

Example 3 Annual energy consumption of a gas boiler accounting for part load effects

A gas boiler is used to supply space heat to a building. The load varies linearly with ambient temperature as shown in Table 4.1.5 below. If the efficiency of the boiler is 80% at peak, rated conditions, find the seasonal average efficiency, annual energy input, and annual energy output using the data in the table. The boiler input at rated conditions is 8750 MBtu/hr corresponding to –12.5°F temperature bin at which the load is 7000 MBtu/hr.

This boiler is turned off in temperature bins higher than 57.5°F roughly corresponding to the limit of the heating season; therefore, the standby losses above this temperature are zero.

The values of the coefficients in the part load characteristic Equation 4.1.9 are

$$A = 0.1$$

$$B = 1.6$$

$$C = -0.7$$

Load data shown in the third column of Table 4.1.5 exhibit the linearity of load with ambient temperature.

The part load characteristic equation is

$$\frac{\dot{Q}_i}{\dot{Q}_{i,\text{full}}} = 0.1 + 1.6(\text{PLR}) - 0.7(\text{PLR})^2 \tag{4.1.12}$$

The key equation for the solution is Equation 4.1.11. Since we are given the part load energy input equation instead of the efficiency at part load, this expression takes a somewhat simpler form

$$Q_{i,\text{yr}} = \sum_{j=0}^{j=N} = \dot{Q}_i(T_j)n_j(T_j)$$

To find the total energy used by the boiler, one sums the "fuel used" column of Table 4.1.5 to find that 22,439 MMBtu are used to meet the annual load of 10,525 MMBtu. The ratio of these two numbers is the overall annual boiler efficiency, 47%. This value is 41% less than the peak efficiency of 80%. Clearly, one must take part load effects into account in annual energy calculations.

One method of avoiding the poor efficiency of this system would be to use two (or more) smaller boilers, the combined capacity of which would total the needed 7000 kBtu/h. Properly chosen, the smaller boilers would have operated more nearly at full load more of the time resulting in higher seasonal efficiency. However, smaller boilers cost more than one large boiler with the capacity equal to the total of the smaller boilers. Multiple boiler systems also offer standby security; if one boiler should fail, the other could carry at least part of the load. A single boiler system would entirely fail to meet the load.

The final decision must be made based on economics, giving proper account to the increased reliability of a system composed of several smaller boilers. Constraints are imposed on such decisions by initial budget, fuel type, owner and architect decisions, and available space.

4.1.3 Service Hot Water

Heated water is used in buildings for various purposes, including basins, sinks for custodial service, showers, and specialty services including kitchens in restaurants and the like. This section overviews service (or domestic) water heating methods for buildings. For details refer to ASHRAE (1999).

Water is heated by equipment that is either part of the space heating system, i.e., the boiler, or by a standalone water heater. The standalone equipment is similar to a small boiler except that water chemistry must be accounted for by use of anodic protection for the tank and by water softening in geographic areas where hardness can cause scale (lime) deposits in the water heater tank.

Two types of systems are used for water heating — *instantaneous* or *storage*. The former heats water on demand as it passes through the heater which uses either steam or hot water. Output temperatures can vary with this system unless a control valve is used on the heated water (not the heat supply) side of the water heater (usually a heat exchanger). Instantaneous water heaters are best suited to relatively uniform loads. They avoid the cost and heat losses of the storage tank but require larger and more expensive heating elements.

Storage type systems are used to accommodate varying loads or loads where large peak demands make it impractical to use instantaneous systems. Water in the storage tank is heated by an immersion steam coil, by direct firing, or by an external heat exchanger. In sizing this system, the designer must account for standby losses from the tank jacket and connected hot water piping. For any steam-based system cold supply water can be preheated using the steam condensate.

In order to size the equipment two items must be known:

- Hourly peak demand for the year — gal/hr, l/hr
- Daily consumption — gal/day, l/d

Of course, the volumetric usage rates must be converted to energy terms by multiplying by the specific heat and water temperature rise.

$$\dot{Q}_{water} = \dot{m}_{water} c_{water} (T_{set} - T_{source}) \tag{4.1.13}$$

where

$\dot{Q}_{water}$ is the water heat rate, either on a daily or hourly basis; Btu/d or Btu/hr, kJ/d or W
$\dot{m}_{water}$ is the water mass flow rate, either on a daily or hourly basis, calculated from the volumetric flow listed above
$(T_{set} - T_{source})$ are the required hot water supply temperature and water source temperatures, respectively.
c_{water} is the specific heat of water

Table 4.1.6 summarizes water demands for various types of buildings, and Table 4.1.7 lists nominal set points of water heaters for several end uses. When using the lower settings in the table, the designer must be aware of the potential for *Legionella pneumophila* (Legionnaire's Disease). This microbe has been

TABLE 4.1.6 Hot Water Demands and Use for Various Types of Buildings

Type of building[a]	Maximum hour	Maximum day	Average day
Men's dormitories	3.8 gal (14.4 L)/student	22.0 gal (83.4 L)/student	13.1 gal (49.7 L)/student
Women's dormitories	5.0 gal (19 L)/student	26.5 gal (100.4 L)/student	12.3 gal (46.6 L)/student
Motels: No. of units[a]			
20 or less	6.0 gal (22.7 L)/unit	35.0 gal (132.6 L)/unit	20.0 gal (75.8 L)/unit
60	5.0 gal (19.7 L)/unit	25.0 gal (94.8 L)/unit	14.0 gal (53.1 L)/unit
100 or more	4.0 gal (15.2 L)/unit	15.0 gal (56.8 L)/unit	10.0 gal (37.9 L)/unit
Nursing homes	4.5 gal (17.1 L)/bed	30.0 gal (113.7 L)/bed	18.4 gal (69.7 L)/bed
Office buildings	0.4 gal (1.5 L)/person	2.0 gal (7.6 L)/person	1.0 gal (3.8 L)/person
Food service establishments:			
Type A: full-meal restaurants and cafeterias	1.5 gal (5.7 L)/max meals/h	11.0 gal (41.7 L)/max meals/h	2.4 gal (9.1 L)/average meals/h[c]
Type B: drive-ins, grilles, luncheonettes, sandwich and snack shops	0.7 gal (2.6 L)/max meals/h	6.0 gal (22.7 L)/max meals/h	0.7 gal (2.6 L)/average meals/day[c]
Apartment houses: No. of apartments			
20 or less	12.0 gal (45.5 L)/apt.	80.0 gal (303.2 L)/apt.	42.0 gal (159.2 L)/apt.
50	10.0 gal (37.9 L)/apt.	73.0 gal (276.7 L)/apt.	40.0 gal (151.6 L)/apt.
75	8.5 gal (32.2 L)/apt.	66.0 gal (250 L)/apt.	38.0 gal (144 L)/apt.
100	7.0 gal (26.5 L)/apt.	60.0 gal (227.4 L)/apt.	37.0 gal (140.2 L)/apt.
200 or more	5.0 gal (19 L)/apt.	50.0 gal (195 L)/apt.	35.0 gal (132.7 L)/apt.
Elementary schools	0.6 gal (2.3 L)/student	1.5 gal (5.7 L)/student	0.6 gal (2.3 L)/student[b]
Junior and senior high schools	1.0 gal (3.8 L)/student	3.6 gal (13.6 L)/student	1.8 gal (6.8 L)/student[b]

[a] The average usage of a U.S. residence is 60 gal/day (227 L/h) with a peak usage of 6 gal/h (22.7 L/h) (ASHRAE, 1987).
[b] Interpolate for intermediate values.
[c] Per day of operation. Temperature basis: 140°F.
Source: From ASHRAE. With permission.

TABLE 4.1.7 Representative Hot Water Use Temperatures

Use	Temperature °F	Temperature °C
Lavatory		
Handwashing	105	40
Shaving	115	45
Showers and tubs	110	43
Therapeutic baths	95	35
Commercial and institutional laundry	180	82
Residential dishwashing and laundry	140	60
Surgical scrubbing	110	43
Commercial spray-type dishwashing		
Single or multiple tank hood(s) or rack(s)		
Wash	150 min	65 min
Final rinse	180–195	82–90
Single tank conveyor		
Wash	160 min	71 min
Final rinse	180–195	82–90

Note: Table values are water use temperatures, not necessarily water heater set points.
Source: From ASHRAE. With permission.

traced to infestations of shower heads; it is able to grow in water maintained at 115°F (46°C). This problem can be limited by using domestic water temperatures in the 140°F (60°C) range.

Hot water can be supplied from a storage type system at the maximum rate

$$\dot{V}_{water} = \dot{V}_r + \frac{f_{useful} V_{tank}}{\Delta t} \tag{4.1.14}$$

where

$\dot{V}_{water}$ is the volumetric hot water supply rate; gal/hr, l/s
$\dot{V}_r$ is the water heater recovery rate; gal/hr, l/s
f_{useful} is the useful fraction of the hot water in the tank before dilution lowers temperature excessively; 0.60–0.80
V_{tank} is the tank volume; gal, (L)
Δt is the duration of peak demand, h, (s)

Jacket losses are assumed to be small.

4.1.4 Electric Resistance Heating

Electricity can be used as the heat source in both furnaces and boilers. Electric units are available in the full range of sizes from small residential furnaces (5 to 15 kW) to large boilers for commercial buildings (200 kW to 20 MW). Electric units have four attractive features:

- Relatively lower initial cost
- Efficiency near 100%
- Near zero part load penalty
- Flue gas vents are not needed

The cost of electricity (both energy and demand charges, see Chapters 3.1 and 3.2) diminishes the apparent advantage of electric boilers and furnaces, however. Nevertheless, they continue to be installed where first cost is a prime concern. However, the prudent designer should consider the overwhelming life cycle costs of electric systems. Electric boiler and furnace sizing follows the methods outlined above for fuel-fired systems. In many cases, the thermodynamic and economic penalties of pure resistance heating can be reduced by using electric heat pumps, the subject of the next section.

Environmental concerns must also be considered when considering electric heating. Low conversion and transmission efficiencies (relative to direct combustion of fuels for water heating) result in relatively higher CO_2 emissions. SO_2 emissions from coal power plants are also an environmental concern.

4.1.5 Electric Heat Pumps

A *heat pump* extracts heat from environmental or other medium temperature sources (such as the ground, groundwater, or building heat recovery systems), raises its temperature sufficiently to be of value in meeting space heating or other loads, and delivers it to the load. This chapter emphasizes heat pumps used for space heating with outdoor air or groundwater as the heat source.

Figure 4.1.5 shows a heat pump cycle on the *T-s* diagram; Figure 4.1.6 shows it on the more frequently used p-h diagram. It is exactly the refrigeration cycle discussed in Chapter 2. Vapor is compressed in step 3-4 and heat is extracted from the condenser in step 4–1. This heat is used for space heating in the systems discussed in this section. In step 1-2, isenthalpic throttling takes place to the low side pressure. Finally, heat extracted from the environment, or other low temperature heat source, is used to boil the refrigerant in the evaporator in step 2-3.

An ideal Carnot heat pump would appear as a rectangle in the *T-s* diagram. The coefficient of performance (COP) of a Carnot heat pump is given in Chapter 2.1 which shows there to be inversely proportional to the difference between the high and low temperature reservoirs. The same result applies generally to heat pumps using real fluids. Although the high side temperature (T_1) remains essentially fixed (ignoring for now the effect of night thermostat setback), the low side temperature closely tracks the widely varying outdoor temperature. As a result, the *capacity and COP of air source heat pumps are strong functions of outdoor temperature*. This feature of heat pumps must be accounted for by the designer since heat pump capacity diminishes as the space heating load on it increases. Heat pumps can be

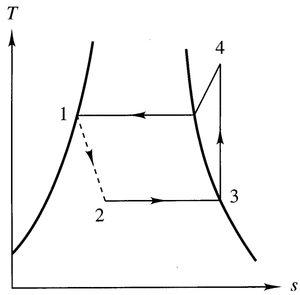

FIGURE 4.1.5 Heat pump T-s diagram showing four steps of the simple heat pump process. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

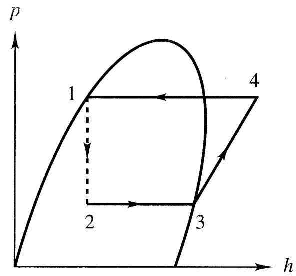

FIGURE 4.1.6 Heat pump p-h diagram showing four steps of the simple heat pump process. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

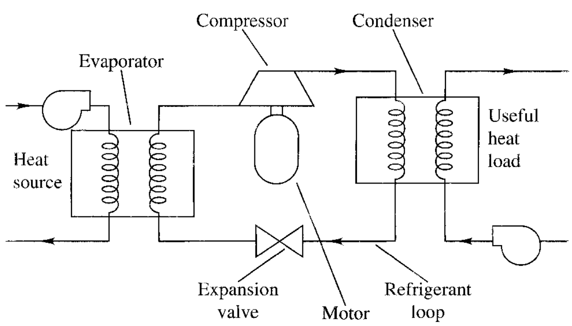

FIGURE 4.1.7 Liquid source heat pump mechanical equipment schematic diagram showing motor driven centrifugal compressor, condenser, and evaporator. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

supplemented by fuel heat or electric resistance heating depending on the cost of each. Figure 4.1.7 shows a water source heat pump system that is not subject to outdoor temperature variations if groundwater or a heat recovery loop is used as the heat source.

The attraction of heat pumps is that they can deliver more thermal power than they consume electrically during an appreciable part of the heating season. In moderate climates requiring both heating and cooling, the heat pump can also be operated as an air conditioner, thereby avoiding the additional cost of a separate air conditioning system. Figure 4.1.8 shows one way to use a heat pump system for both heating and cooling by reversing flow through the system.

Typical Equipment Configurations

Heat pumps are available in sizes ranging from small residential units (10 kW) to large central systems (up to 15 MW) for commercial buildings. Large systems produce heated water at temperatures up to

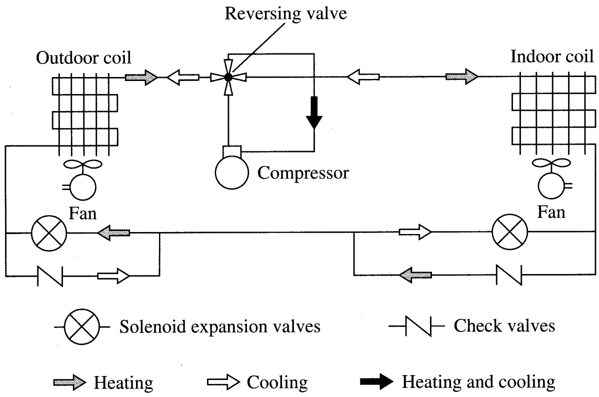

FIGURE 4.1.8 Air-to-air heat pump diagram. A reciprocating compressor is used. This design allows operation as a heat pump or an air conditioner by reversing the refrigerant flow. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

220°F (105°C). Central systems can use both environmental and internal building heat sources. In many practical circumstances the heat gains in the core zones of a commercial building could satisfy the perimeter heat losses in winter. A heat pump could be used to efficiently condition both types of zones simultaneously.

Heat pumps require a compressor and two heat exchangers. In the energy bookkeeping that one does for heat pumps, the power input to the compressor is added to the heat removed from the low temperature heat source to find the heat delivered to the space to be heated. Increased heating capacity at low air source temperatures can be achieved by oversizing the compressor. To avoid part load penalties in moderate weather, a variable speed compressor drive can be used.

The outdoor and indoor heat exchangers use forced convection on the air side to produce adequate heat transfer coefficients. In the outdoor exchanger, the temperature difference between the boiling refrigerant and the air is between 10 and 25°F (6–14°C). If the heat source is internal building heat, water is used to transport heat to the heat pump evaporator and smaller temperature differences can be used.

A persistent problem with air source heat pumps is the accumulation of frost on the outdoor coil at coil surface temperatures just above the freezing point. The problem is most severe in humid climates; little defrosting is needed for temperatures below 20°F (–7°C) where humidities are below 60%. *Reverse cycle defrosting* can be accomplished by briefly operating the heat pump as an air conditioner (by reversing the flow of refrigerant) and turning the outdoor fan off. Hot refrigerant flowing through the outside melts the accumulated frost. This energy penalty must be accounted for in calculating the COP of heat pumps. Defrost control can be initiated either by time clock or, better, by a sensor measuring either the refrigerant condition (temperature or pressure) or, ideally, by the air pressure drop across the coil.

The realities of heat pump performance, as discussed above, reduce the capacity of real systems from the Carnot ideal. Figure 4.1.9 shows ideal Carnot COP values as a function of source temperature for a high side temperature of 70°F (21°C). The intermediate curve shows performance for a Carnot heat pump with real (i.e., finite temperature difference) heat exchangers. Finally, the performance of a real heat pump is shown in the lower curve. Included in the lower curve are the effects of heat exchanger losses, use of real fluids, compressor inefficiencies, and pressure drops. The COP of real machines is much lower (about 50%) than that for an ideal Carnot cycle with heat exchanger penalties.

The *energy efficiency ratio* EER is the ratio of heating capacity (Btu per hour) to the electric input rate (watts). EER thus has the units of Btu per watt-hour. The dimensionless COP is found from the EER by dividing it by the conversion factor 3.413 Btu/W · h.

Heat Pump Selection

The strong dependence of heat pump output on ambient temperature must be accounted for when selecting central plant equipment. If outdoor air is used as the heat source, peak heating requirements

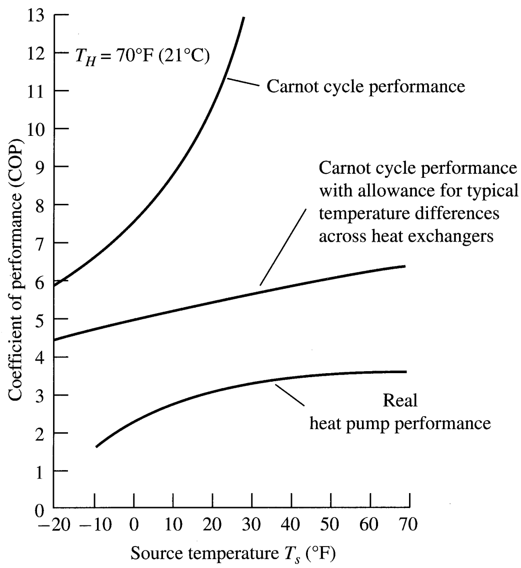

FIGURE 4.1.9 COP of ideal Carnot, Carnot (with heat exchanger penalty), and real heat pumps. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

will invariably exceed the capacity of any economically feasible unit. Therefore, auxiliary heating is needed for such systems. Supplemental heat should always be added downstream of the heat pump condenser to ensure that the condenser operates at as low a temperature as possible, thereby improving the COP.

The amount of auxiliary heat needed and the type (electricity, natural gas, oil, or other) must be determined by an economic analysis and fuel availability (heat pumps are often used when fossil fuels are unavailable). The key feature of such analysis is the combined effect of part load performance and ambient source temperature on system output and efficiency. The following section shows how the temperature bin approach can be used for such an assessment. Figure 4.1.10 shows the conflicting characteristics of heat pumps and buildings in the heating season. As ambient temperature drops, loads increase, but heating capacity drops. The point at which the two curves intersect is called the *heat pump balance point*. To the left, auxiliary heat is needed; to the right, the heat pump must be modulated since excess capacity exists.

Recovery from night thermostat setback must be carefully thought out by the designer if an air source heat pump is used. A step change up in the thermostat setpoint on a cold winter morning will inevitably cause the auxiliary heat source to come on. If this heat source is electricity, high electrical demand charges may result, and the possible economic advantage of the heat pump will be reduced. One approach to avoid activation of the electric resistance heat elements uses a longer setup period with gradually increasing thermostat setpoint. A smart controller could control the start-up time based on known heat pump performance characteristics and outdoor temperature. Alternatively, fuel could be used as the auxiliary heat source. During building warmup, all outside air dampers remain closed as is common practice for any commercial building heating system.

Heat pump efficiency is greater if lower high side temperatures can be used. In order to produce adequate space heat in such conditions, a larger coil may be needed in the air stream. However, if the coil is sized for the cooling load, it will nearly always have adequate capacity for heating. In such a case, adequate space heat can be provided at relatively low air temperatures, 95–110°F (35–43°C). Table 4.1.8 summarizes advantages and disadvantages of air and water source heat pumps.

Controls for heat pumps are more complex than for fuel-fired systems since outdoor conditions, coil frosting, and heat load must all be considered. In addition, to avoid excessive demand charges, the controller must avoid *coincident* operation of resistance heat and the compressor at full capacity (attempting to meet a large load on a cold day).

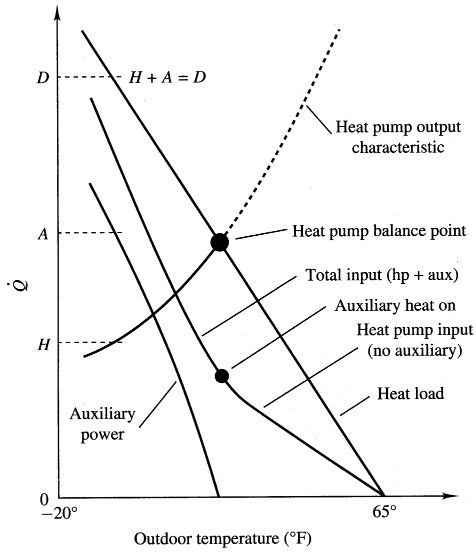

FIGURE 4.1.10 Graph of building heat load, heat pump capacity, and auxiliary heat quantity as a function of outdoor air temperature for a typical, air source residential heat pump. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

Table 4.1.8 Advantages and Disadvantages of Air and Water Source Heat Pumps

Type	Advantages	Disadvantages
Air source	Indoor distribution permits air conditioning and humidity control Outdoor air source readily available Simple installation Least expensive Established commercial technology	Defrost required Low capacity at cold outdoor temperature Lower efficiency because of large evaporator $\Delta T \approx 30°F$ Indoor air distribution temperature must be high for comfort reasons Reliability at low temperature is only fair, due to frosting effects Must keep evaporator clear of leaves, dirt, etc.
Water source	Multiple family and commercial installation as central system In commercial installations, good coupling to cooling towers No refrigerant reversal needed; reverse water flow instead	Needs water source at useful temperature Efficiency penalty due to space heat exchanger ΔT

Part Load Performance

As discussed in detail above, air source heat pumps are particularly sensitive to the environment. Earlier we examined how the performance of boilers changed when the heating load changed with outdoor temperature. The COP of an air source heat pump has an even greater dependence on environmental conditions. This section provides an example to illustrate the magnitude of the effect. Since air source heat pumps are often used on residences, it provides a residential scale example.

Example 4 Seasonal Heat Pump Performance Using the Bin Method

A residence in a heating climate has a total heat transmission coefficient K_{tot} = 650 Btu/(h · °F) (343 W/K). An air source heat pump with a capacity of 39,900 Btu/h (11.7 kW) at 47°F (8.3°C) (standard rating point in the U.S.) is to be evaluated. Find the heating season electrical energy usage, seasonal COP (often called the *seasonal performance factor*, SPF), and energy savings relative to electric

TABLE 4.1.9 Heat Pump and Building Load Data – Example 4

Bin temp. °F	Heating load Btu/h	Heat pump COP	Heat pump output Btu/h	Heat pump input Btu/h	Auxiliary power Btu/h	Heating system COP
62	1,950	2.64	1,950	739	0	2.64
57	5,200	2.68	5,200	1,940	0	2.68
52	8,450	2.64	8,450	3,201	0	2.64
47	11,700	2.63	11,700	4,449	0	2.63
42	14,950	2.50	14,950	5,980	0	2.50
37	18,200	2.39	18,200	7,615	0	2.39
32	21,450	2.23	21,450	9,619	0	2.23
27[a]	24,700	2.07	24,700	11,932	0	2.07
22	27,950	1.97	25,100	12,741	2,850	1.79
17	31,200	1.80	22,400	12,444	8,800	1.47
12	34,450	1.70	19,900	11,706	14,550	1.31
7	37,700	1.54	17,600	11,429	20,100	1.20
2	40,950	1.39	15,400	11,079	25,550	1.12
–3	44,200	1.30	13,500	10,385	30,700	1.08
–8	47,450	1.17	11,700	10,000	35,750	1.04

[a] Heat pump balance point.

resistance heating. Use the bin data and heat pump performance data given in Table 4.1.9. The house heating base temperature is 65°F (18.3°C) accounting for internal gains. Figure 4.1.11 shows the energy flows as a function of outdoor temperature.

The preceding table includes these data in order:

1. Center point of temperature bin, T_{bin}
2. Heating demand, $\dot{Q} = K_{tot}(65°\text{F} - T_{bin})$
3. COP from manufacturer's data, a function of temperature, including defrost
4. Heat pump output; above the balance point, $\dot{Q}$; below the heat pump balance point, manufacturer's data
5. Heat pump input, the heat pump output divided by COP
6. Auxiliary power; the positive difference, if any, between $\dot{Q}$ and heat pump output
7. *Heating system* COP given by $\dot{Q}$ divided by the sum of auxiliary power and heat pump input

The energy calculations are summarized in Table 4.1.10. The bottom line in the table contains energy totals. With the heat pump, the total electricity requirement is 48.7 MBtu/yr (51.4 GJ/yr). If pure resistance heating were used, the total electricity requirement would be 98.36 MBtu/yr (103.8 GJ/yr).

The SPF for the heat pump is the seasonal output divided by the seasonal input to the heat pump:

$$\mathrm{SPF_{hp}} = \frac{Q_{o,\mathrm{yr}}}{Q_{i,\mathrm{yr}}} = \frac{90.33\ \mathrm{MBtu}}{40.66\ \mathrm{MBtu}} = 2.22$$

The SPF for the heating system is the seasonal heat load divided by the seasonal input to the heat pump and the auxiliary heater:

$$\mathrm{SPF_{sys}} = \frac{Q_{o,\mathrm{yr}}}{Q_{i,\mathrm{yr}} + Q_{i,\mathrm{aux,yr}}} = \frac{98.36\ \mathrm{MBtu}}{(40.66 + 8.03)\ \mathrm{MBtu}} = 2.02$$

The advantage of a constant temperature heat source is apparent from this example. If ground water or building exhaust air (both essentially at constant temperature) were used as the heat source rather than outdoor air, there would not be a drop off in capacity as with the outdoor air source device just when heat is most needed.

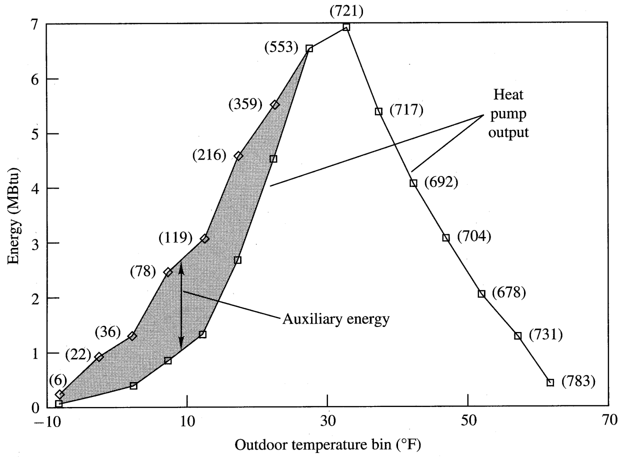

FIGURE 4.1.11 Heat pump energy use for the bin method example. The numbers at each bin temperature indicate the number of hours of occurrence in each bin. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

TABLE 4.1.10 Heat Pump Energy Calculations – Example 4

Bin temp. °F	Bin hours h	Heating energy MBtu	Heat pump output MBtu	Heat pump input MBtu	Aux. heat input MBtu	Total input MBtu
62	783	1.53	1.53	0.58	0.00	0.58
57	731	3.80	3.80	1.42	0.00	1.42
52	678	5.73	5.73	2.17	0.00	2.17
47	704	8.24	8.24	3.13	0.00	3.13
42	692	10.35	10.35	4.14	0.00	4.14
37	717	13.05	13.05	5.46	0.00	5.46
32	721	15.47	15.47	6.94	0.00	6.94
27[a]	553	13.66	13.66	6.60	0.00	6.60
22	359	10.03	9.01	4.57	1.02	5.60
17	216	6.74	4.84	2.69	1.90	4.59
12	119	4.10	2.37	1.39	1.73	3.12
7	78	2.94	1.37	0.89	1.57	2.46
2	36	1.47	0.55	0.40	0.92	1.32
–3	22	0.97	0.30	0.23	0.68	0.90
–8	6	0.28	0.07	0.06	0.21	0.27
Total		98.36	90.33	40.66	8.03	48.70

[a] Heat pump balance point.

4.1.6 Low Temperature Radiant Heating

Heating systems in many parts of the world use warmed floors and/or ceilings for space heating in buildings. Although this system is unusual in the U.S., the good comfort and quiet operation provided by this approach make it worth considering for some applications. In Europe it is far more common. Radiant systems are well suited to operation with heat pump, solar, and other low temperature systems. This section discusses the principles of low temperature space heating. This form of heating is distinct from high temperature radiant heating using either electricity or natural gas to provide a high temperature source from which radiation can be directed for localized heating.

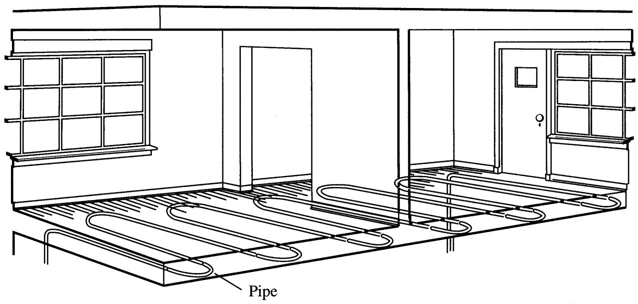

FIGURE 4.1.12 Residential radiant floor heating system. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

Figure 4.1.12 shows how a radiant floor might be configured in a residence. The same concept can also be used in the ceiling in both residential and commercial buildings. The term *radiant* is a misnomer since between 30% (ceilings) and 50% (floors) of heat transferred from radiant panels is actually by convection. However, we will use the industry's nomenclature for this heating system.

The radiation heat output of radiant panels is given by the Stefan-Boltzmann equation as discussed in Chapter 2.1.

$$\dot{Q} = \in_{eff} F_{h,u} \sigma(T_h^4 - T_u^4) \tag{4.1.15}$$

where

$\in_{eff}$ is $1/[(1/\in_h + 1/\in_u) - 1]$, the effective emittance of the space, and the subscripts h and u refer to the unheated and heated (by radiant panels) surfaces of the space; the effective emittance is approximately 0.8.
$F_{h,u}$ is the view factor between the heating surface and the unheated surfaces; its value is 1.0 in the present case.
T_h is the heating surface temperature.
T_u is the mean of the unheated surface temperatures.
σ is the Stefan-Boltzmann constant (see Chapter 2).

Convection from the heating surface can be found using the standard free convection expressions in Chapter 2.1.

The designer's job is to determine the panel area needed, its operating temperature, the heating liquid flow rate, and construction details. The panel size is determined based on standard heat load calculations (Chapter 6.1). Proper account should be made of any losses from the back of the radiant panels to unheated spaces. Panel temperatures should not exceed 85°F (29.5°C) for floors and 115°F (46°C) for ceilings.

Water temperatures are typically 120°F (49°C) for floors and up to 155°F (69°C) for ceilings. Panels can be piped in a series configuration if pipe runs are not excessively long (the final panels in a long series run will not perform up to specifications due to low fluid temperatures). Long series loops also have excessively high pressure drops. If large areas are to be heated, a combination of series and parallel connections can be used. Manufacturers can advise regarding the number of panels that can be connected in series without performance penalties.

If radiant floors are to be built during building construction rather than using prefabricated panels in ceilings, the following guidelines can be used. Tubing spacing for a system of the type shown above should be between 6 and 12 in (15 and 30 cm). The tubing diameter ranges between 0.5 and 1.0 in (0.6 and 2.5 cm). Flow rates are determined by the rate of heat loss from the panel, which in turn depends on the surface temperature and hence the fluid temperature. This step in the design is iterative. Panel design follows this process:

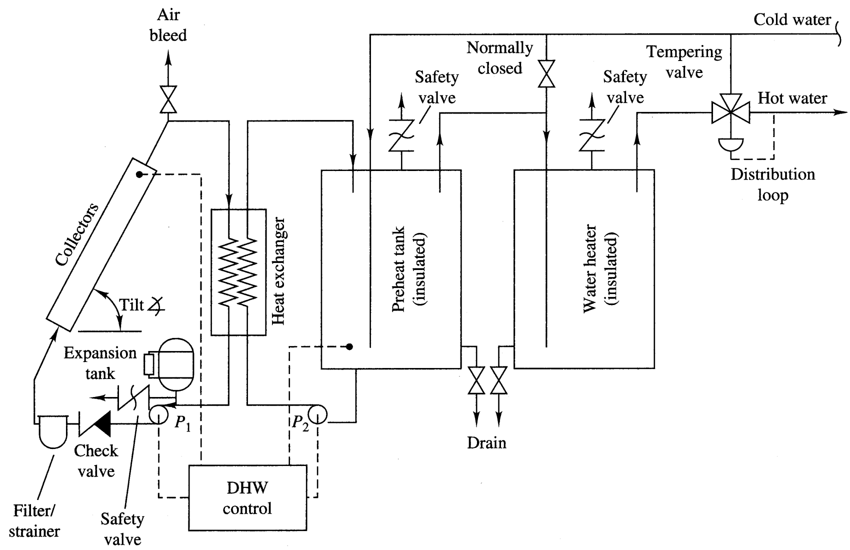

FIGURE 4.1.13 Solar water heating system including collectors, pumps, heat exchanger, and storage tanks along with piping and ancillary fittings. Collectors are tilted up from the horizontal at a fixed angle roughly equal to the local latitude. (From Rabl, A. and Kreider J.F., *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY, 1994. With permission.)

- Determine room heat load.
- Decide on location of panels (roof or floor).
- Find panel heat flux including both radiation and convection contributions at 80°F (27°C) for floor panels and 110°F (43°C) for ceiling panels.
- Divide heat load by heat flux to find needed panel area.
- If panel area exceeds available floor or ceiling area, raise panel temperature (not exceeding temperatures noted earlier) and repeat steps 3 and 4.
- If the panel area is still insufficient, consider both floor and ceiling panels.

Control of radiant heating systems has proven to be a challenge in the past due to the large time constant of these systems. Both under- and overheating are problems. If the outdoor temperature drops rapidly, this system will have difficulty responding quickly. On the other hand, after a morning warmup followed by high solar gains on a sunny winter day, the radiant system may overshoot. The current generation of "smart" controls should help improve the comfort control of these systems.

4.1.7 Solar Heating

Solar energy is a source of low temperature heat that has selected applications to buildings. Solar water heating is a particularly effective method of using this renewable resource since low to moderate temperature water (up to 140°F, 60°C) can be produced by readily available, flat plate collectors (Goswami, Kreider, and Kreith, 2000).

Figure 4.1.13 shows one system for heating service water for residential or commercial needs using solar collectors. The system consists of three loops; it is instructive to describe the system's operation based on these three.

First, the collector loop (filled with a nonfreezing solution if needed) operates whenever the DHW controller determines that the collector is warmer, by a few degrees, than the storage tank. Heat is transferred from the solar-heated fluid by a counterflow or plate heat exchanger to the storage tank in the second loop of the system. Storage is needed since the availability of solar heat rarely matches the water heating load. The check valve in the collector loop is needed to prevent reverse flow at night in systems where the collectors (which are cold at night) are mounted above the storage tank.

The third fluid loop is the hot water delivery loop. Hot water drawn off to the load is replaced by cold water supplied to the solar preheat tank, where it is heated as much as possible by solar heat. If solar energy is insufficient to heat the water to its setpoint, conventional fuels can finish the heating in the water heater tank, as shown on the right of Figure 4.1.13. The tempering valve in the distribution loop is used to limit the temperature of water dispatched to the building if the solar tank should be above the water heater setpoint in summer.

The energy delivery of DHW systems can be found using the *f*-chart method described in Duffie and Beckman (1992). As a rough rule of thumb, one square foot of collector can provide one gallon of hot water per day (45 L/m^2) on the average in sunny climates. Design pump flows are to be 0.02 gal/min per square foot of collector [0.01 $L/(s \cdot m^2)$], and heat exchanger effectivenesses of at least 0.75 can be justified economically. Tanks should be insulated so that no more than 2% of the stored heat is lost overnight.

Solar heating should be assessed on an economic basis. If the cost of delivered solar heat, including the amortized cost of the delivery system and its operation, is less than that of competing energy sources, an incentive exists for using the solar resource. The collector area needed on commercial buildings can be large; if possible, otherwise unused roof space can be used to hold the collector arrays. See Chapter 6.4 for a complete and detailed description of solar system analysis and design.

References

ASHRAE, 1999, *HVAC Applications*, ASHRAE, Atlanta, GA.

ASHRAE, 2000, *Equipment*, ASHRAE, Atlanta, GA.

ASHRAE, 2001, *Fundamentals*, ASHRAE, Atlanta, GA.

Haas, J. H., 1990, Steam traps — key to process heating, *Chemical Engineering*, vol. 97, pp. 151–156, January.

Goswami, Y., Kreider, J. F., and Kreith, F., 2000, *Principles of Solar Engineering*, Taylor and Francis, New York, pp. 694.

Rabl, A. and Kreider, J.F., 1994, *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY.

Wong, W. Y., 1989, Safer relief valve sizing, *Chemical Engineering*, vol. 96, pp. 137–140, May.

4.2 Air Conditioning Systems

Dennis L. O'Neal and John A. Bryant

Air conditioning has rapidly grown over the past 50 years, from a luxury to a standard system included in most residential and commercial buildings. In 1970, 36% of residences in the U.S. were either fully air conditioned or utilized a room air conditioner for cooling (Blue, et al., 1979). By 1997, this number had more than doubled to 77%, and that year also marked the first time that over half (50.9%) of residences in the U.S. had central air conditioners (Census Bureau, 1999). An estimated 83% of all new homes constructed in 1998 had central air conditioners (Census Bureau, 1999). Air conditioning has also grown rapidly in commercial buildings. From 1970 to 1995, the percentage of commercial buildings with air conditioning increased from 54 to 73% (Jackson and Johnson, 1978, and DOE, 1998).

Air conditioning in buildings is usually accomplished with the use of mechanical or heat-activated equipment. In most applications, the air conditioner must provide both cooling and dehumidification to maintain comfort in the building. Air conditioning systems are also used in other applications, such as automobiles, trucks, aircraft, ships, and industrial facilities. However, the description of equipment in this chapter is limited to those commonly used in commercial and residential buildings.

Commercial buildings range from large high-rise office buildings to the corner convenience store. Because of the range in size and types of buildings in the commercial sector, there is a wide variety of equipment applied in these buildings. For larger buildings, the air conditioning equipment is part of a total system design that includes items such as a piping system, air distribution system, and cooling tower.

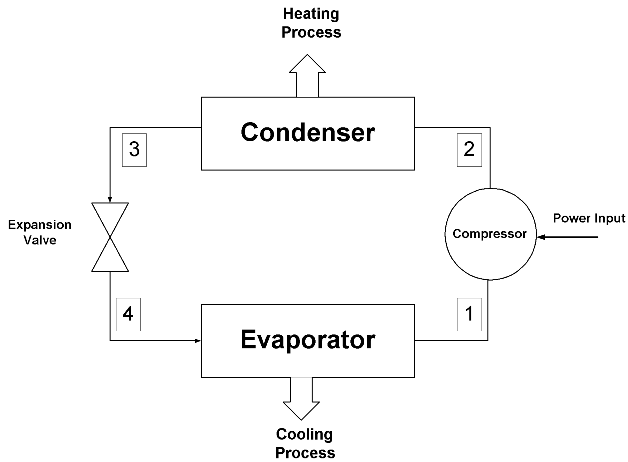

FIGURE 4.2.1 Simplified schematic and pressure/enthalpy diagram of the vapor compression cycle used in many air conditioners.

Proper design of these systems requires a qualified engineer. The residential building sector is dominated by single family homes and low-rise apartments/condominiums. The cooling equipment applied in these buildings comes in standard "packages" that are often both sized and installed by the air conditioning contractor.

The chapter starts with a general discussion of the vapor compression refrigeration cycle then moves to refrigerants and their selection. Chillers and their auxiliary systems are then covered, followed by packaged air conditioning equipment.

4.2.1 Vapor Compression Cycle

Even though there is a large range in sizes and variety of air conditioning systems used in buildings, most systems utilize the vapor compression cycle to produce the desired cooling and dehumidification. This cycle is also used for refrigerating and freezing foods and for automotive air conditioning. The first patent on a mechanically driven refrigeration system was issued to Jacob Perkins in 1834 in London, and the first viable commercial system was produced in 1857 by James Harrison and D.E. Siebe (Thevenot 1979).

Besides vapor compression, there are two less common methods used to produce cooling in buildings: the absorption cycle and evaporative cooling. These are described later in the chapter. With the vapor compression cycle, a working fluid, which is called the refrigerant, evaporates and condenses at suitable pressures for practical equipment designs.

The four basic components (Figure 4.2.1) in every vapor compression refrigeration system are the compressor, condenser, expansion device, and evaporator. The compressor raises the pressure of the refrigerant vapor so that the refrigerant saturation temperature is slightly above the temperature of the cooling medium used in the condenser. The type of compressor used depends on the application of the system. Large electric chillers typically use a centrifugal compressor while small residential equipment uses a reciprocating or scroll compressor.

The condenser is a heat exchanger used to reject heat from the refrigerant to a cooling medium. The refrigerant enters the condenser and usually leaves as a subcooled liquid. Typical cooling mediums used in condensers are air and water. Most residential-sized equipment uses air as the cooling medium in the condenser, while many larger chillers use water.

After leaving the condenser, the liquid refrigerant expands to a lower pressure in the expansion valve. The expansion valve can be a passive device, such as a capillary tube or short tube orifice, or an active device, such as a thermal expansion valve or electronic expansion valve. The purpose of the valve is to regulate the flow of refrigerant to the evaporator so that the refrigerant is superheated when it reaches the suction of the compressor.

At the exit of the expansion valve, the refrigerant is at a temperature below that of the medium (air or water) to be cooled. The refrigerant travels through a heat exchanger called the evaporator. It absorbs energy from the air or water circulated through the evaporator. If air is circulated through the evaporator, the system is called a *direct expansion system*. If water is circulated through the evaporator, it is called a *chiller*. In either case, the refrigerant does not make direct contact with the air or water in the evaporator.

The refrigerant is converted from a low quality, two-phase fluid to a superheated vapor under normal operating conditions in the evaporator. The vapor formed must be removed by the compressor at a sufficient rate to maintain the low pressure in the evaporator and keep the cycle operating.

All mechanical cooling results in the production of heat energy that must be rejected through the condenser. In many instances, this heat energy is rejected to the environment directly to the air in the condenser or indirectly to water where it is rejected in a cooling tower. With some applications, it is possible to utilize this waste heat energy to provide simultaneous heating to the building. Recovery of this waste heat at temperatures up to 65°C (150°F) can be used to reduce costs for space heating.

Capacities of air conditioning are often expressed in either tons or kilowatts (kW) of cooling. The ton is a unit of measure related to the ability of an ice plant to freeze one short ton (907 kg) of ice in 24 hr. Its value is 3.51 kW (12,000 Btu/hr). The kW of thermal cooling capacity produced by the air conditioner must not be confused with the amount of electrical power (also expressed in kW) required to produce the cooling effect.

4.2.2 Refrigerants Use and Selection

Up until the mid-1980s, refrigerant selection was not an issue in most building air conditioning applications because there were no regulations on the use of refrigerants. Many of the refrigerants historically used for building air conditioning applications have been chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). Most of these refrigerants are nontoxic and nonflammable. However, recent U.S. federal regulations (EPA 1993a; EPA 1993b) and international agreements (UNEP, 1987) have placed restrictions on the production and use of CFCs and HCFCs. Hydrofluorocarbons (HFCs) are now being used in some applications where CFCs and HCFCs were used. Having an understanding of refrigerants can help a building owner or engineer make a more informed decision about the best choice of refrigerants for specific applications. This section discusses the different refrigerants used in or proposed for building air conditioning applications and the regulations affecting their use.

The American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) has a standard numbering system (Table 4.2.1) for identifying refrigerants (ASHRAE, 1992). Many popular CFC, HCFC, and HFC refrigerants are in the methane and ethane series of refrigerants. They are called halocarbons, or halogenated hydrocarbons, because of the presence of halogen elements such as fluorine or chlorine (King, 1986).

Zeotropes and azeotropes are mixtures of two or more different refrigerants. A zeotropic mixture changes saturation temperatures as it evaporates (or condenses) at constant pressure. The phenomena is called temperature glide. At atmospheric pressure, R-407C has a boiling (bubble) point of –44°C (–47°F) and a condensation (dew) point of –37°C (–35°F), which gives it a temperature glide of 7°C (12°F). An azeotropic mixture behaves like a single component refrigerant in that the saturation temperature does not change appreciably as it evaporates or condenses at constant pressure. R-410A has a small enough temperature glide (less than 5.5°C, 10°F) that it is considered a near-azeotropic refrigerant mixture.

ASHRAE groups refrigerants (Table 4.2.2) by their toxicity and flammability (ASHRAE, 1994). Group A1 is nonflammable and least toxic, while Group B3 is flammable and most toxic. Toxicity is based on the upper safety limit for airborne exposure to the refrigerant. If the refrigerant is nontoxic in quantities less than 400 parts per million, it is a Class A refrigerant. If exposure to less than 400 parts per million is toxic, then the substance is given the B designation. The numerical designations refer

TABLE 4.2.1 Common Refrigerants with Their Applications and Characteristics

Refrigerant	Typical or Proposed Applications	Normal boiling point, °C	Safety Group
	Methane Series		
11	Low pressure centrifugal chillers	24	A1
12	Refrigeration, medium pressure chillers, auto A/C	–30	A1
22	Package A/C, heat pumps	–41	A1
32	Component of R-407C and R-410A	–52	A2
	Ethane Series		
123	Low pressure chillers	27	B1
125	Component of R-407C and R-410A	–49	A1
134a	Chillers, refrigeration, auto A/C	–26	A1
	Propane Series		
290	Proposed replacement for R-22	–42	A3
	Zeotropes		
407C	Package A/C, heat pumps	–44	A1
410A	Package A/C, heat pumps	–53	A1
	Azeotropes		
500	Medium pressure centrifugal chillers	–33	A1
502	Refrigeration, low temperature heat pumps	–45	A1
	Hydrocarbons		
600	Refrigeration	0	A3
600a	Refrigeration	–12	A3
	Inorganic Compounds		
717	Industrial Refrigeration	–33	B2
744	Proposed automotive A/C	–78	A1

Source: From ASHRAE (1997); Smit et al. (1996).

TABLE 4.2.2 Toxicity and Flammability Rating System

Flammability	Group	Group
High	A3	B3
Moderate	A2	B2
Non	A1	B1
Threshold Limit Value (parts per million)	<400	>400

Source: From ASHRAE (1994).

to the flammability of the refrigerant. The last column of Table 4.2.1 shows the toxicity and flammability rating of common refrigerants.

Refrigerants in the A1 group usually fulfill the basic requirements for an ideal refrigerant for comfort air conditioning because they are nontoxic and nonflammable. Common refrigerants in the A1 group used in building air conditioning applications include R-11, R-12, R-22, R-134a, and R-410A.

R-11, R-12, R-123, and R-134a are refrigerants commonly used in centrifugal chiller applications. Both R-11, a CFC, and R-123, an HCFC, have low-pressure high-volume characteristics ideally suited for use in centrifugal compressors. Before the ban on production of CFCs, R-11 and R-12 were the refrigerants

TABLE 4.2.3 Comparative Performance of Commonly Used Refrigerants at +2°C Evaporating Temperature and +40°C Condensing Temperature[a]

Refrigerant Number	Evaporator Pressure (kPa)	Condenser Pressure (kPa)	Net Cooling Effect (kJ/kg)	Refrigerant Circulated (kg/min)	Compressor Displacement (L/min)	Coefficient of Performance
11	44	173	155	3.9	144	6.5
12	329	960	114	5.3	27	6.0
22	531	1533	157	3.8	17	5.9
123	36	155	142	4.2	173	6.3
134a	315	1017	143	4.2	27	5.9
290	508	1368	270	2.2	20	5.8
407C	488	1738	151	4.0	20	4.8
410A	848	2417	157	3.8	12	5.5
717	464	1559	1072	0.6	13	7.1

[a] Refrigerant circulation rate is based on 10 kW cooling capacity.

of choice for chiller applications. The use of these two refrigerants is currently limited to maintenance of existing systems. Both R-123 and R-134a are now being used extensively in new chillers. R-123 provides an efficiency advantage (Table 4.2.3) over R-134a. However, R-123 has a B1 safety classification, which means it has a lower toxicity threshold than R-134a. If an R-123 chiller is used in a building, ASHRAE Standard 15 (ASHRAE, 1992) provides guidelines for safety precautions when using this or any other refrigerant that is toxic or flammable.

Refrigerant 22 is an HCFC, is used in many of the same applications, and is still the refrigerant of choice in many reciprocating and screw chillers as well as small commercial and residential packaged equipment. It operates at a much higher pressure than either R-11 or R-12. Restrictions on the production of HCFCs will start in 2004. In 2010, R-22 cannot be used in new air conditioning equipment. R-22 cannot be produced after 2020 (EPA, 1993b).

R-407C and R-410A are both mixtures of HFCs. Both are considered replacements for R-22. R-407C is expected to be a drop-in replacement refrigerant for R-22. Its evaporating and condensing pressures for air conditioning applications are close to those of R-22 (Table 4.2.3). However, replacement of R-22 with R-407C should be done only after consulting with the equipment manufacturer. At a minimum, the lubricant and expansion device will need to be replaced. The first residential-sized air conditioning equipment using R-410A was introduced in the U.S. in 1998. Systems using R-410A operate at approximately 50% higher pressure than R-22 (Table 4.2.3); thus, R-410A cannot be used as a drop-in refrigerant for R-22. R-410A systems utilize compressors, expansion valves, and heat exchangers designed specifically for use with that refrigerant.

Ammonia is widely used in industrial refrigeration applications and in ammonia water absorption chillers. It is moderately flammable and has a class B toxicity rating but has had limited applications in commercial buildings unless the chiller plant can be isolated from the building being cooled (Toth, 1994, Stoecker, 1994). As a refrigerant, ammonia has many desirable qualities. It has a high specific heat and high thermal conductivity. Its enthalpy of vaporization is typically 6 to 8 times higher than that of the commonly used halocarbons, and it provides higher heat transfer compared to halocarbons. It can be used in both reciprocating and centrifugal compressors.

Research is underway to investigate the use of natural refrigerants, such as carbon dioxide (R-744) and hydrocarbons in air conditioning and refrigeration systems (Bullock, 1997, and Kramer, 1991). Carbon dioxide operates at much higher pressures than conventional HCFCs or HFCs and requires operation above the critical point in typical air conditioning applications. Hydrocarbon refrigerants, often thought of as too hazardous because of flammability, can be used in conventional compressors and have been used in industrial applications. R-290, propane, has operating pressures close to R-22 and has been proposed as a replacement for R-22 (Kramer, 1991). Currently, there are no commercial systems sold in the U.S. for building operations that use either carbon dioxide or flammable refrigerants.

Table 4.2.3 shows a comparative performance of refrigerants at evaporating and condensing temperatures typical of cooling applications. The data show the relatively large cooling effect produced with ammonia. For the specific conditions, R-11 and R-123 have the lowest evaporating pressures. The coefficient of performance (COP) listed in the far right column of Table 4.2.3 is a measure of the thermodynamic efficiency of an air conditioner (or chiller) with that particular refrigerant. It is defined as the cooling output (kW) divided by the power input (kW) to the compressor. The actual COP in a system will not only depend on the refrigerant, but on the design of the compressor, heat exchangers, and expansion device. At cooling conditions, there is substantial drop off in efficiency between R-123 and R-134a. Manufacturers who have replaced their R-11 chillers with R-134a chillers have had to make substantial modifications in the chiller designs to maintain comparable efficiencies. Two R-22 replacement refrigerants, R-407C and R-410A, both have lower COPs than R-22.

Rowland and Molina (1974) hypothesized that CFCs were responsible for destroying ozone in the stratosphere. By the late 1970s, the U.S. and Canada had banned the use of CFCs in aerosols. In the mid 1980s, a 40% depletion in the ozone layer was measured (Salas and Salas, 1992). In September 1987, forty-three countries signed an agreement called the Montreal Protocol in which the participants agreed to freeze CFC production levels by 1990 and then to decrease production by 20% by 1994 and 50% by 1999. The protocol was ratified by the U.S. in 1988 and subjected the air conditioning industry to major CFC restrictions. Title IV of the Clean Air Act of November 1990 required elimination of the production of CFCs by 2000 (EPA, 1993a) and placed a schedule on the phasing out of the production of HCFCs by 2030.

Two ratings were developed to classify the harmful effects of a refrigerant on the environment (EPA, 1993b). The first, the ozone depletion potential (ODP), quantifies the potential damage that the refrigerant molecule has in destroying ozone in the stratosphere. The estimated atmospheric life of a given CFC or HCFC is an important factor in determining the value of the ODP.

The second rating is known as the halocarbon global warming potential (HGWP). It relates the potential for a refrigerant in the atmosphere to contribute to the greenhouse effect. Like CO_2, refrigerants such as CFCs, HCFCs, and HFCs can block energy from the earth from radiating back into space. One molecule of R-12 can absorb as much energy as almost 5000 molecules of CO_2. Both the ODP and HGWP are normalized to the value of R-11.

Table 4.2.4 shows the ODP and HGWP for a variety of refrigerants. As a class of refrigerants, the CFCs have the highest ODP and HGWP. Because HCFCs tend to be more unstable compounds and therefore have much shorter atmospheric lifetimes, their ODP and HGWP values are much smaller than those of the CFCs. All HFCs and their mixtures have zero ODP because fluorine does not react with ozone. However, some of the HFCs, such as R-125 and R-134a, do have HGWP values that are as large as or larger than some of the HCFCs. Hydrocarbons provide zero ODP and HGWP.

In recent years, attempts have been made to develop an alternate criteria, called the total equivalent warming impacts (TEWI) for evaluating the global warming impact of different refrigerants (Sand, Fischer, and Baxter, 1999). TEWI includes the total energy use of the equipment over its expected lifetime as well as the global warming caused by release of the refrigerant charge in the system. The TEWI depends on assumptions about the usage and efficiency of the system. Sand, Fischer, and Baxter (1999) estimated that energy usage in a system accounts for over 90% of the global warming potential of the system. A high efficiency system using R-22 could have a lower TEWI than a lower efficiency system using a zero HGWP refrigerant. The TEWI represents a more systems approach to global warming impact than does HGWP by itself.

4.2.3 Chilled Water Systems

Chilled water systems were used in less than 4% of commercial buildings in the U.S. in 1995. However, because chillers are usually installed in larger buildings, chillers cooled over 28% of the U.S. commercial building floor space that same year (DOE, 1998). Five types of chillers are commonly applied to commercial buildings: reciprocating, screw, scroll, centrifugal, and absorption. The first four utilize the vapor

TABLE 4.2.4 Ozone Depletion Potential and Halocarbon Global Warming Potential of Popular Refrigerants and Mixtures

Refrigerant Number	Ozone Depletion Potential (ODP)	Halogen Global Warming Potential (HGWP)
	Chlorofluorocarbons	
11	1.0	1.0
12	1.0	3.05
22	0.051	0.37
123	0.016	0.019
	Hydrofluorocarbons	
32	0	0.13
125	0	0.58
134a	0	0.285
	Hydrocarbons	
50	0	0
290	0	0
	Zeotropes	
407C	0	0.22
410A	0	0.44
	Azeotropes	
500	0.74	2.4
502	0.23	5.1

Source: Compiled from Salas and Salas (1992), NR (1995), and Didion (1996).

compression cycle to produce chilled water. They differ primarily in the type of compressor used. Absorption chillers utilize thermal energy (typically steam or combustion source) in an absorption cycle with either an ammonia-water or water-lithium bromide solution to produce chilled water.

Overall System

Figure 4.2.2 shows a simple representation of a dual chiller application with all the major auxiliary equipment. An estimated 86% of chillers are applied in multiple chiller arrangements like that shown in the figure (Bitondo and Tozzi, 1999). In chilled water systems, return water from the building is circulated through each chiller evaporator where it is cooled to an acceptable temperature (typically 4 to 7°C) (39 to 45°F). The chilled water is then distributed to water-to-air heat exchangers spread throughout the facility. In these heat exchangers, air is cooled and dehumidified by the cold water. During the process, the chilled water increases in temperature and must be returned to the chiller(s).

The chillers shown in Figure 4.2.2 are water-cooled chillers. Water is circulated through the condenser of each chiller where it absorbs heat energy rejected from the high pressure refrigerant. The water is then pumped to a cooling tower where the water is cooled through an evaporation process. Cooling towers are described in a later section. Chillers can also be air cooled. In this configuration, the condenser would be a refrigerant-to-air heat exchanger with air absorbing the heat energy rejected by the high pressure refrigerant.

Chillers nominally range in capacities from 30 to 18,000 kW (8 to 5100 ton). Most chillers sold in the U.S. are electric and utilize vapor compression refrigeration to produce chilled water. Compressors for these systems are either reciprocating, screw, scroll, or centrifugal in design. A small number of centrifugal chillers are sold that use either an internal combustion engine or steam drive instead of an electric motor to drive the compressor.

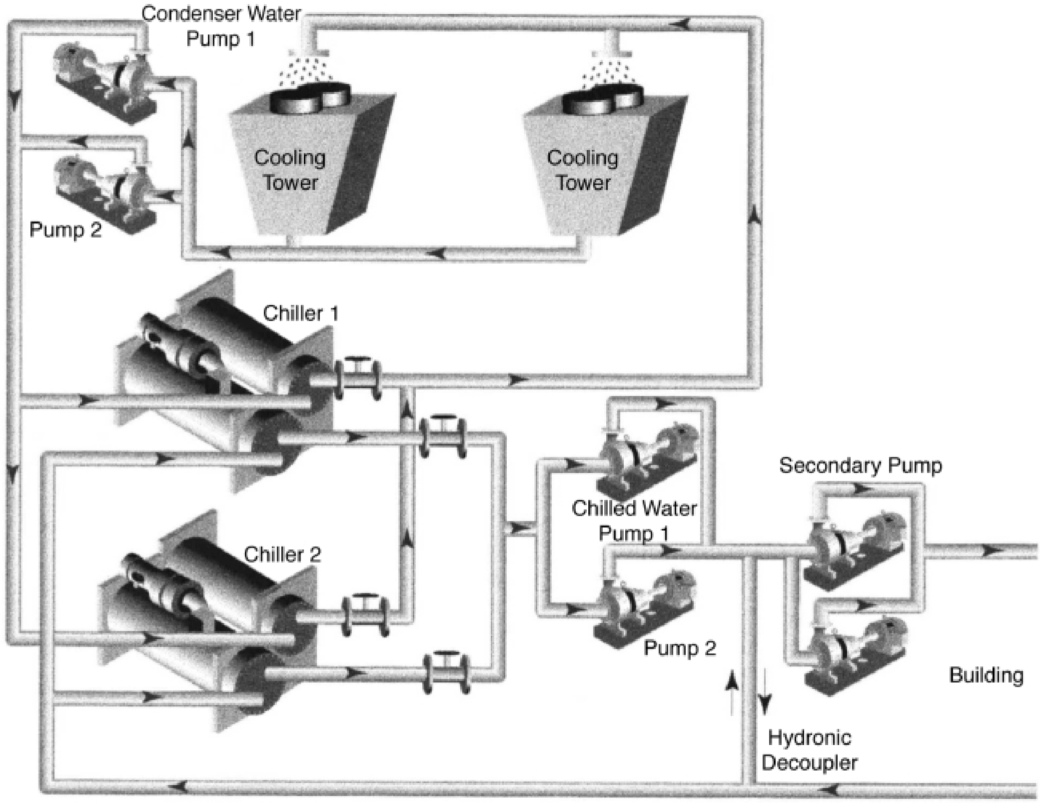

FIGURE 4.2.2 A dual chiller application with major auxiliary systems (courtesy of Carrier Corporation).

The type of chiller used in a building depends on the application. For large office buildings or in chiller plants serving multiple buildings, centrifugal compressors are often used. In applications under 1000 kW (280 tons) cooling capacities, reciprocating or screw chillers may be more appropriate. In smaller applications, below 100 kW (30 tons), reciprocating or scroll chillers are typically used.

Vapor Compression Chillers

Table 4.2.5 shows the nominal capacity ranges for the four types of electrically driven vapor compression chillers. Each chiller derives its name from the type of compressor used in the chiller. The systems range in capacities from the smallest scroll (30 kW; 8 tons) to the largest centrifugal (18,000 kW; 5000 tons). Chillers can utilize either an HCFC (R-22 and R-123) or HFC (R-134a) refrigerant. The steady state efficiency of chillers is often stated as a ratio of the power input (in kW) to the chilling capacity (in tons). A capacity rating of one ton is equal to 3.52 kW or 12,000 btu/h. With this measure of efficiency, the smaller number is better. As seen in Table 4.2.5, centrifugal chillers are the most efficient; whereas, reciprocating chillers have the worst efficiency of the four types. The efficiency numbers provided in the table are the steady state full-load efficiency determined in accordance to ASHRAE Standard 30 (ASHRAE, 1995). These efficiency numbers do not include the auxiliary equipment, such as pumps and cooling tower fans that can add from 0.06 to 0.31 kW/ton to the numbers shown (Smit et al., 1996).

Chillers run at part load capacity most of the time. Only during the highest thermal loads in the building will a chiller operate near its rated capacity. As a consequence, it is important to know how the efficiency of the chiller varies with part load capacity. Figure 4.2.3 shows a representative data for the efficiency (in kW/ton) as a function of percentage full load capacity for a reciprocating, screw, and scroll chiller plus a centrifugal chiller with inlet vane control and one with variable frequency drive (VFD) for the compressor. The reciprocating chiller increases in efficiency as it operates at a smaller percentage of

TABLE 4.2.5 Capacity Ranges and Efficiencies of Vapor Compression Chillers Used for Commercial Building Air Conditioning

Type of Chiller	Nominal Capacity Range (kW)	Refrigerants Used in New Systems	Range in Full Load Efficiency (kW/ton)
Reciprocating	50 to 1750	R-22	0.80 to 1.00
Screw	160 to 2350	R-134a, R-22	0.60 to 0.75
Scroll	30 to 200	R-22	0.81 to 0.92
Centrifugal	500 to 18,000	R-134a, R-123	0.50 to 0.70

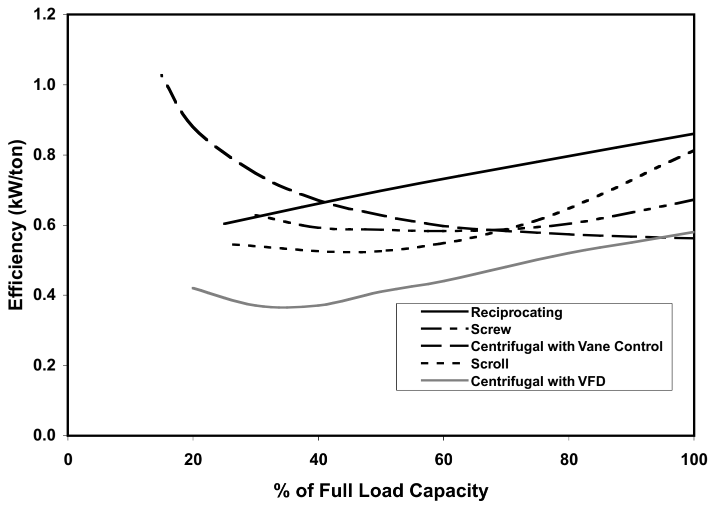

FIGURE 4.2.3 Chiller efficiency as a function of percentage of full load capacity.

full load. In contrast, the efficiency of a centrifugal with inlet vane control is relatively constant until the load falls to about 60% of its rated capacity and its kW/ton increases to almost twice its fully loaded value.

In 1998, the Air Conditioning and Refrigeration Institute (ARI) developed a new standard that incorporates into their ratings part load performance of chillers (ARI 1998c). Part load efficiency is expressed by a single number called the integrated part load value (IPLV). The IPLV takes data similar to that in Figure 4.2.3 and weights it at the 25%, 50%, 75%, and 100% loads to produce a single integrated efficiency number. The weighting factors at these loads are 0.12, 0.45, 0.42, and 0.01, respectively. The equation to determine IPLV is:

$$\mathrm{IPLV} = \frac{1}{\frac{0.01}{A} + \frac{0.42}{B} + \frac{0.45}{C} + \frac{0.12}{D}}$$

where,

A = efficiency at 100% load
B = efficiency at 75% load
C = efficiency at 50% load
D = efficiency at 25% load

Most of the IPLV is determined by the efficiency at the 50% and 75% part load values. Manufacturers will provide, on request, IPLVs as well as part load efficiencies such as those shown in Figure 4.2.3.

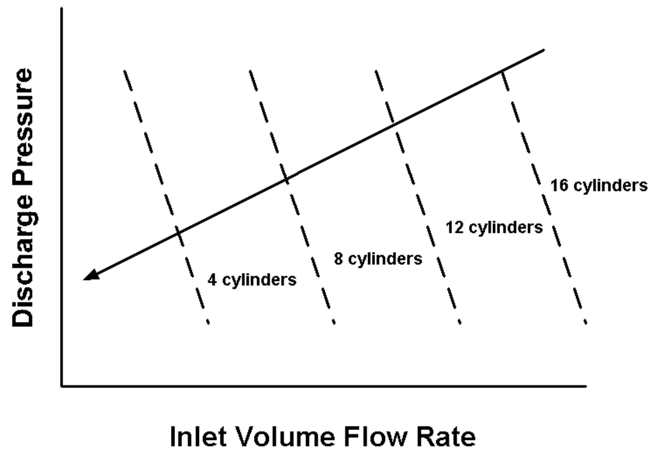

FIGURE 4.2.4 Volume-pressure relationships for a reciprocating compressor.

The four compressors used in vapor compression chillers are each briefly described below. While centrifugal and screw compressors are primarily used in chiller applications, reciprocating and scroll compressors are also used in smaller unitary packaged air conditioners and heat pumps.

Reciprocating Compressors

The reciprocating compressor is a positive displacement compressor. On the intake stroke of the piston, a fixed amount of gas is pulled into the cylinder. On the compression stroke, the gas is compressed until the discharge valve opens. The quantity of gas compressed on each stroke is equal to the displacement of the cylinder. Compressors used in chillers have multiple cylinders, depending on the capacity of the compressor. Reciprocating compressors use refrigerants with low specific volumes and relatively high pressures. Most reciprocating chillers used in building applications currently employ R-22.

Modern high-speed reciprocating compressors are generally limited to a pressure ratio of approximately nine. The reciprocating compressor is basically a constant-volume variable-head machine. It handles various discharge pressures with relatively small changes in inlet-volume flow rate as shown by the heavy line (labeled 16 cylinders) in Figure 4.2.4. Condenser operation in many chillers is related to ambient conditions, for example, through cooling towers, so that on cooler days the condenser pressure can be reduced. When the air conditioning load is lowered, less refrigerant circulation is required. The resulting load characteristic is represented by the solid line that runs from the upper right to lower left of Figure 4.2.4.

The compressor must be capable of matching the pressure and flow requirements imposed by the system. The reciprocating compressor matches the imposed discharge pressure at any level up to its limiting pressure ratio. Varying capacity requirements can be met by providing devices that unload individual or multiple cylinders. This unloading is accomplished by blocking the suction or discharge valves that open either manually or automatically. Capacity can also be controlled through the use of variable speed or multi-speed motors. When capacity control is implemented on a compressor, other factors at part-load conditions need to considered, such as (a) effect on compressor vibration and sound when unloaders are used, (b) the need for good oil return because of lower refrigerant velocities, and (c) proper functioning of expansion devices at the lower capacities.

With most reciprocating compressors, oil is pumped into the refrigeration system from the compressor during normal operation. Systems must be designed carefully to return oil to the compressor crankcase to provide for continuous lubrication and also to avoid contaminating heat-exchanger surfaces.

Reciprocating compressors usually are arranged to start unloaded so that normal torque motors are adequate for starting. When gas engines are used for reciprocating compressor drives, careful matching of the torque requirements of the compressor and engine must be considered.

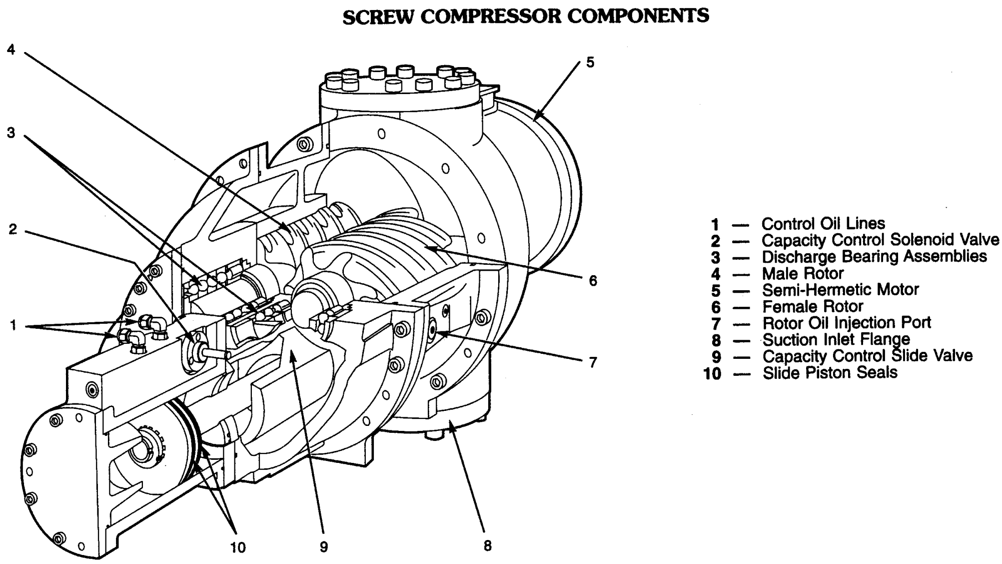

FIGURE 4.2.5 Illustration of a twin-screw compressor design (courtesy of Carrier Corporation).

Screw Compressors

Screw compressors, first introduced in 1958 (Thevenot, 1979), are positive displacement compressors. They are available in the capacity ranges that overlap with reciprocating compressors and small centrifugal compressors. Both twin-screw and single-screw compressors are used in chillers. The twin-screw compressor is also called the helical rotary compressor. Figure 4.2.5 shows a cutaway of a twin-screw compressor design. There are two main rotors (screws). One is designated male (4 in the figure) and the other female (6 in the figure).

The compression process is accomplished by reducing the volume of the refrigerant with the rotary motion of screws. At the low pressure side of the compressor, a void is created when the rotors begin to unmesh. Low pressure gas is drawn into the void between the rotors. As the rotors continue to turn, the gas is progressively compressed as it moves toward the discharge port. Once reaching a predetermined volume ratio, the discharge port is uncovered and the gas is discharged into the high pressure side of the system. At a rotation speed of 3600 rpm, a screw compressor has over 14,000 discharges per minute (ASHRAE, 1996).

Fixed suction and discharge ports are used with screw compressors instead of valves, as used in reciprocating compressors. These set the *built-in volume ratio* — the ratio of the volume of fluid space in the meshing rotors at the beginning of the compression process to the volume in the rotors as the discharge port is first exposed. Associated with the built-in volume ratio is a pressure ratio that depends on the properties of the refrigerant being compressed. Screw compressors have the capability to operate at pressure ratios of above 20:1 (ASHRAE, 1996). Peak efficiency is obtained if the discharge pressure imposed by the system matches the pressure developed by the rotors when the discharge port is exposed. If the interlobe pressure in the screws is greater or less than discharge pressure, energy losses occur but no harm is done to the compressor.

Capacity modulation is accomplished by slide valves that provide a variable suction bypass or delayed suction port closing, reducing the volume of refrigerant compressed. Continuously variable capacity control is most common, but stepped capacity control is offered in some manufacturers' machines. Variable discharge porting is available on some machines to allow control of the built-in volume ratio during operation.

Oil is used in screw compressors to seal the extensive clearance spaces between the rotors, to cool the machines, to provide lubrication, and to serve as hydraulic fluid for the capacity controls. An oil separator is required for the compressor discharge flow to remove the oil from the high-pressure refrigerant so that performance of system heat exchangers will not be penalized and the oil can be returned for reinjection in the compressor.

Screw compressors can be direct driven at two-pole motor speeds (50 or 60 Hz). Their rotary motion makes these machines smooth running and quiet. Reliability is high when the machines are applied properly. Screw compressors are compact so they can be changed out readily for replacement or maintenance. The efficiency of the best screw compressors matches or exceeds that of the best reciprocating compressors at full load. High isentropic and volumetric efficiencies can be achieved with screw compressors because there are no suction or discharge valves and small clearance volumes. Screw compressors for building applications generally use either R-134a or R-22.

Scroll Compressors

The principle of the scroll compressor was first patented in 1905 (Matsubara el al., 1987). However, the first commercial units were not built until the early 1980s and were sold in Japan in residential heat pump systems (Senshu et al., 1985). Of the different electric driven chillers discussed in this section, scroll chillers have the smallest range in capacity. Only one U.S. manufacturer currently offers a scroll chiller, and these are limited to capacities below 200 kW (57 tons). Scroll compressors are built in sizes as small as 3 kW (.05 ton). Scroll compressors are primarily used in direct expansion air conditioners, heat pumps, and some refrigeration applications. Chillers using scroll compressors currently only use R-22. However, direct expansion air conditioners with scroll compressors that use R-410A have recently been introduced into the market, and it would be reasonable to expect a switch to an HFC in chillers in the near future.

Scroll compressors have two spiral-shaped scroll members that are assembled 180° out of phase (Figure 4.2.6). One scroll is fixed while the other “orbits” the first. Vapor is compressed by sealing it off at the edge of the scrolls and reducing the volume of the gas as it moves invward toward the discharge port. Figure 4.2.6a shows the two scrolls at the instance that vapor has entered the compressor and compression begins. The orbiting motion of the second scroll forces the pocket of vapor toward the discharge port while decreasing its volume (Figures 4.2.6b–h). In Figures 4.2.6c and f, the two scrolls open at the ends and allow new pockets of vapor to be admitted for compression. Compression is a nearly continuous process in a scroll compressor.

Scroll compressors offer several advantages over reciprocating compressors. First, relatively large suction and discharge ports can be used to reduce pressure losses. Second, the separation of the suction and discharge processes reduces the heat transfer between those processes. Third, with no valves and re-expansion losses, they have higher volumetric efficiencies. The plots of part load efficiencies in Figure 4.2.3 show that scroll compressors have better efficiencies down to 25% part load than do reciprocating compressors. Capacities of systems with scroll compressors can be varied by using variable speed motors or by using multiple suction ports at different locations within the two spiral members.

Centrifugal Compressors

The reciprocating, screw, and scroll compressors are all positive displacement compressors. They each work by taking a fixed volume of low pressure refrigerant and reducing it to achieve compression. In contrast, the centrifugal compressor uses dynamic compression. The primary operating component of the compressor is the impeller. The center of the impeller has vanes that draw the low pressure refrigerant vapor into radial passages internal to the impeller. The impeller rotates and accelerates the gas and increases its kinetic energy. When the gas leaves the impeller, it flows to a circular diffuser passage, the *volute*, where the gas is decelerated and the pressure is increased (Trane, 1980). Centrifugal chillers have operational speeds from 3600 to 35,000 rpm.

Figure 4.2.7 shows a cutaway of a centrifugal compressor that has three impellers. Refrigerant flows from the bottom left to the right through the compressor. This compressor has three stages of compression. Centrifugal compressors with multiple stages can generate a pressure ratio up to 18:1, but their high discharge temperatures limit the efficiency of the simple cycle at these high pressure ratios. As a result, they operate with evaporator temperatures in the same range as reciprocating compressors. Multistage centrifugal compressors are built for direct connection to high-speed drives.

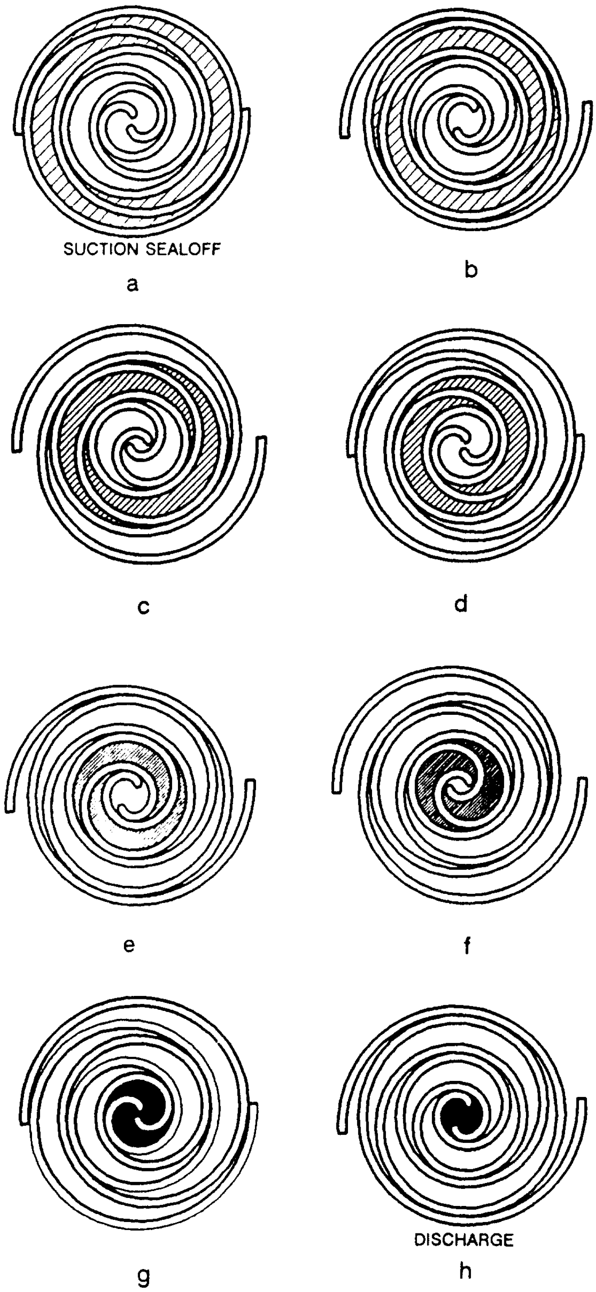

FIGURE 4.2.6 Operation of the scrolls in a scroll compressor. (From ASHRAE, 1996).

Figure 4.2.7 also shows inlet vanes at the entrance to the first impeller. These vanes control the capacity of the compressor by adjusting the angle at which the low pressure refrigerant enters the impeller. Inlet vanes also help to stabilize compressor performance over a wide range of load conditions and to prevent surge (Trane, 1980).

Under certain low load conditions, flow can reverse through the impeller. This phenomena is called surge and is unique to centrifugal compressors. Surge increases inefficiency, static pressure fluctuations, vibration, and noise in the compressor (Trane, 1980).

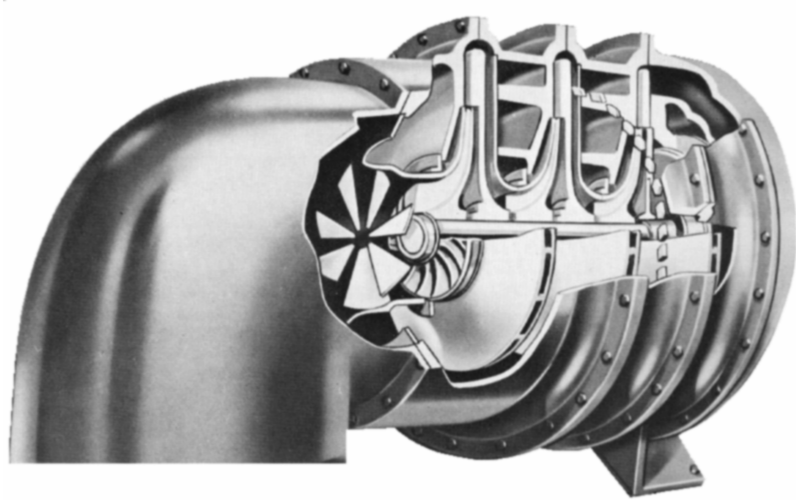

FIGURE 4.2.7 Cutaway of a three stage centrifugal compressor with guide vanes at the inlet (courtesy of the Trane Company).

The centrifugal compressor has a more complex pressure-volume characteristic than reciprocating machines, as shown by the system characteristic curve in Figure 4.2.8a. Changing discharge pressure may cause relatively large changes in inlet volume. Adjustment of variable inlet vanes allows the compressor to operate anywhere below the system line to conditions imposed by the system. A variable-speed controller offers an alternative way to match the compressor's characteristics to the system load, as shown in the lower half of Figure 4.2.8b. The maximum head capability is fixed by the operating speed of the compressor. Both methods have advantages: generally, variable inlet vanes provide a wider range of capacity reduction; variable speed usually is more efficient. Maximum efficiency and control can be obtained by combining both methods of control.

The centrifugal compressor has a surge point — a minimum-volume flow below which stable operation cannot be maintained. The percentage of load at which the surge point occurs depends on the number of impellers, design-pressure ratio, operating speed, and variable inlet-vane setting. The system design and controls must keep the inlet volume above this point.

Provision for minimum load operation is strongly recommended for all installations because there will be fluctuations in plant load. The difference between the operating characteristics of the positive displacement compressor and the centrifugal compressor are important considerations in chiller plant design to achieve satisfactory performance. Unlike positive displacement compressors, the centrifugal compressor will not rebalance abnormally high system heads. The drive arrangement for the centrifugal compressor must be selected with sufficient speed to meet the maximum head anticipated. The relatively flat head characteristics of the centrifugal compressor necessitates different control approaches than for positive displacement machines, particularly when parallel compressors are utilized. These differences, which account for most of the troubles experienced in centrifugal-compressor systems, cannot be overlooked in the design of a chiller system.

Absorption Chillers

The first absorption machine was patented in 1859 by Ferdinand Carr (Thevenot, 1979) and used an ammonia/water solution. The design was produced in Europe and the U.S., and by 1876 over 600 absorption systems had been sold in the U.S. These systems were primarily used for producing ice. During the late 1800s and early 1900s, different combinations of fluids were tested in absorption machines. Lithium bromide and water were not used until 1940 (Thevenot, 1979). Through the 1960s, both absorption and centrifugal chillers competed for large-building air conditioning. However, with the rising prices of oil and gas in the 1970s, absorption chillers became more costly to operate than centrifugal

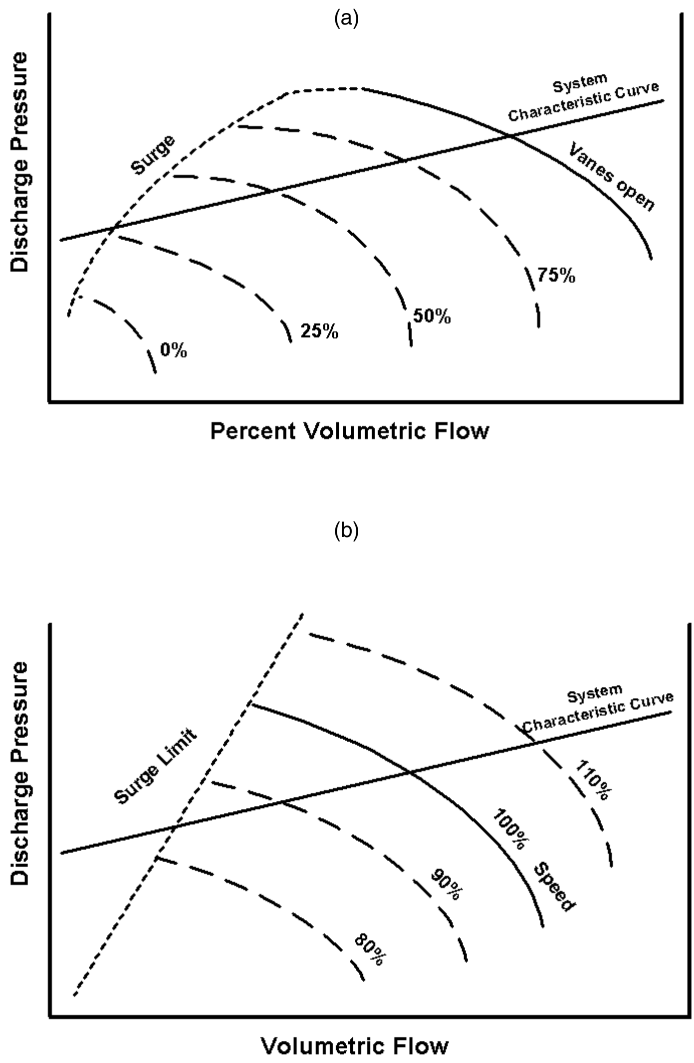

FIGURE 4.2.8 Volume-pressure relationships in a centrifugal compressor. (a) With variable inlet-van control at constant rotational speed. (b) With variable speed control at a constant inlet vane opening.

chillers (Wang, 1993). With the introduction of the more efficient two stage absorption systems in the 1980s and reduction in oil and gas costs by the mid 1980s, absorption systems again became a competitive option for cooling in buildings.

Absorption systems offer at least three advantages over conventional electric vapor compression systems. First, they do not use CFC or HCFC refrigerants. The solutions used in absorption systems are not refrigerants that could someday be eliminated because of ozone depletion or global warming concerns. Second, absorption systems can utilize a variety of heat sources, including natural gas, steam, solar-heated water, and waste heat from a turbine or industrial process. If the source of heat is from waste heat, such as from a co-generation system, absorption systems may provide the lowest cost alternative for providing chilled water for air conditioning. Because sources of energy besides electricity are used, installation of an absorption system can be used to reduce peak electrical demand in situations where electrical demand charges are high. Third, because of the absence of heavy rotating parts, absorption systems produce much less vibration and noise compared to large centrifugal systems (Carrier, 1964).

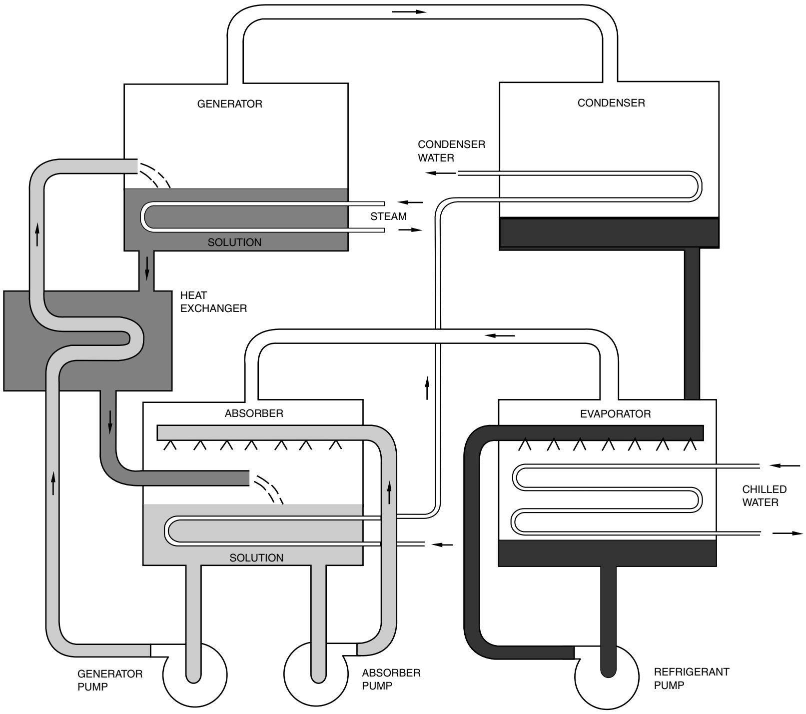

FIGURE 4.2.9 Schematic of the single-effect absorption cycle (courtesy of Carrier Corporation).

Two different absorption systems are currently used for air conditioning applications: (1) a water-lithium bromide system where water is the refrigerant and lithium bromide is the absorbent and (2) an ammonia-water system where ammonia is the refrigerant and water is the absorbent. Absorption systems sold in the U.S. for commercial building applications are almost exclusively water-lithium bromide chillers. One manufacturer sells an ammonia-water system for residential applications in the U.S. Most refrigeration applications of absorption systems are with ammonia and water. Because of the larger applications of water-lithium bromide systems in commercial buildings, the focus of the discussion in this section is on those systems. A description of ammonia-water absorption technology can be found in ASHRAE (1996).

Water-lithium bromide absorption machines can be classified by the method of heat input and whether the cycle is singe or multiple effect (ASHRAE, 1996). Indirect fired chillers use steam or hot liquids as a heat source. Direct fired chillers use the heat energy from the firing of fossil fuels. Heat-recovery chillers use waste gases as the heat source. Single effect and double effect chillers are described below.

The basic components for a single-effect water-lithium bromide absorption system are shown in Figure 4.2.9. The major components of the system are listed below (Carrier 1964):

- **Evaporator** — The evaporator is the section where chilled water is cooled by evaporating the refrigerant (water) over chilled water tubes. Operating pressures in the evaporator must be near a vacuum (less than 1 kPa; 0.1 psi) for evaporation to occur at a low enough temperature to produce chilled water in the tubes for air conditioning purposes.
- **Absorber** — Water vapor from the evaporator is absorbed by the lithium bromide into a liquid solution in this section. Condenser water is circulated through pipes into the liquid solution in the absorber to remove heat energy released during the absorption process. The absorber operates at the same pressure as the evaporator.
- **Generator** — The liquid solution from the absorber is pumped through a heat exchanger to the generator which is a part of the high pressure side of the system. Typical pressures in this section

range from 5 to 7 kPA (0.7 to 1.0 psi), which is considerably below atmospheric pressure. In the generator, the water-rich liquid solution is heated where the water boils off from the solution and is transported to the condenser section. The solution can be heated with steam or other hot fluids through piping or it can be heated by a burner. Once water has evaporated from the water-lithium bromide solution, the dilute water-lithium bromide solution is then returned to the absorber.

- **Condenser** — The relatively high-pressure water vapor from the generator is condensed to liquid in the condenser. This is accomplished by circulating water in piping through the condenser that can absorb the latent heat energy of the water vapor. Once condensed into liquid, the water then flows through an expansion valve and into the low pressure evaporator, which completes the cycle. The condenser water that flows through the pipes is typically sent to a cooling tower where it can be cooled and then recirculated back to the system.
- **Pumps** — At least three pumps are required. The absorber and refrigerant (evaporator) pumps are primarily used to recirculate liquids in their respective sections. The generator pump moves the concentrated water-lithium bromide solution from the absorber to the generator.
- **Heat Exchanger** — The dilute solution of water-lithium bromide is much hotter than the concentrated solution being pumped from the absorber. The heat exchanger reduces energy use by heating the concentrated liquid flowing to the generator as it cools the hot dilute solution flowing from the generator to the absorber. If the dilute solution passing through the heat exchanger does not contain enough refrigerant (water) and is cooled too much, crystallization of the lithium bromide can occur. Leaks or process upsets that cause the generator to over-concentrate the solution are indicated when this occurs. The slushy mixture formed does not harm the machine but does interfere with operation. External heat and added water may be required to redissolve the mixture
- **Purge Unit** — The pressures throughout the system are far below atmospheric pressure. The purge unit is required to remove air and other noncondensable gases that can leak into the system and to maintain the required system pressures.

Absorption systems also include components not shown in Figure 4.2.9 or listed above (ASHRAE, 1994). Palladium cells are used to continuously remove small amounts of hydrogen generated by corrosion. Corrosion inhibitors protect internal parts from the corrosive absorbent solution in the presence of air. Performance additives enhance the heat and mass transfer coefficients of the water-lithium bromide solutions. Flow control from the generator to the absorber is typically achieved with a control valve.

The COP of an absorption system is the cooling achieved in the evaporator divided by the heat input to the generator. The COP of a single-effect water-lithium bromide chiller generally is from 0.65 to 0.70. The heat rejected by the cooling tower from both the condenser and the absorber is the sum of the waste heat supplied plus the cooling effect produced. Thus, absorption systems require larger cooling towers and cooling water flows than do vapor compression systems.

Single-effect water-lithium bromide chillers can still be found operating in older buildings and chiller plants. However, most new systems being sold by major manufacturers are double-effect chillers because of their improved efficiency over single-effect technology (EPRI, 1996). Double-effect absorption systems have a two-stage generator (Figure 4.2.10) with heat input temperatures greater than 150°C (300°F). The basic operation of the double-effect machine is the same as the single-effect machine except that an additional generator, condenser, and heat exchanger are used. Energy from an external heat source is used to boil the dilute lithium bromide (absorbent) solution. The vapor from the primary generator flows in tubes to the second-effect generator. It is hot enough to boil and concentrate absorbent, which creates more refrigerant vapor without any extra energy input. Dual-effect machines typically use steam or hot liquids as input. Coefficients of performance above 1.0 can be obtained with these machines.

Figure 4.2.11 illustrates the part load performance of a dual effect chiller. The steam consumption ratio (SCR) is plotted as a function of capacity at several condenser water temperatures. For the 29.4°C (85°F) curve, at 50% capacity, the system uses only about 5% less steam than at full load. Information such as that shown in Figure 4.2.11 is available from manufacturers. With these data, performance of

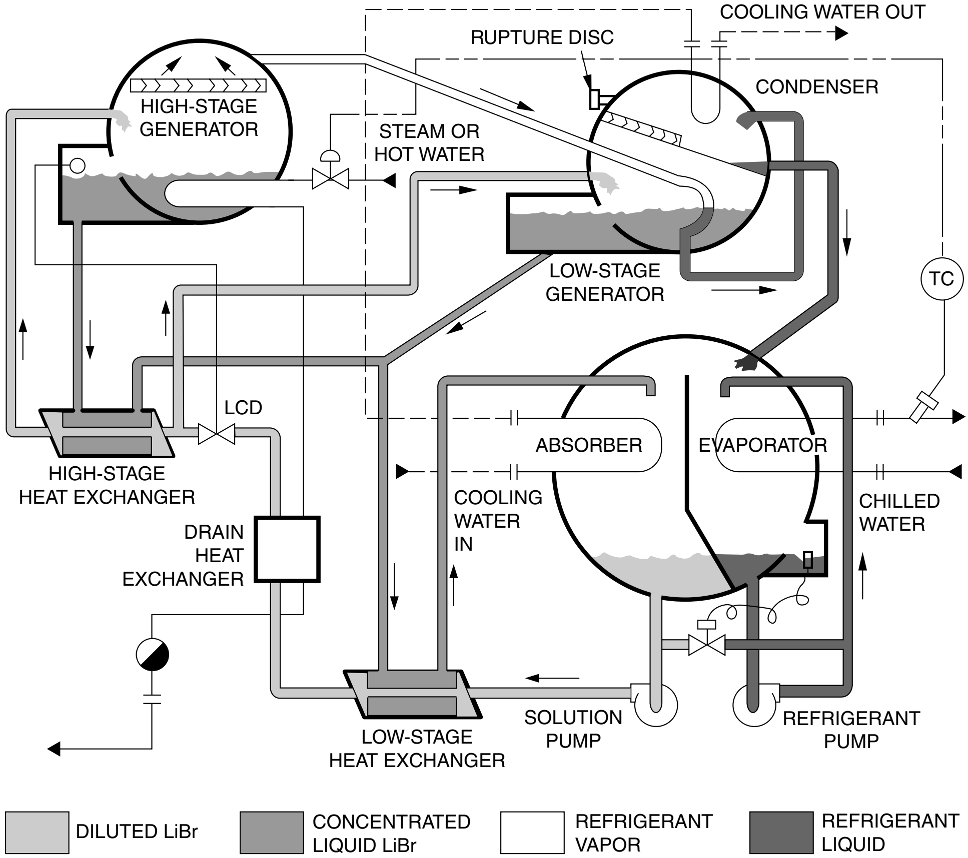

FIGURE 4.2.10 Two-stage water-lithium bromide absorption system. LCD = level control device, TC = temperature controller (capacity control) [Courtesy of Carrier Corporation].

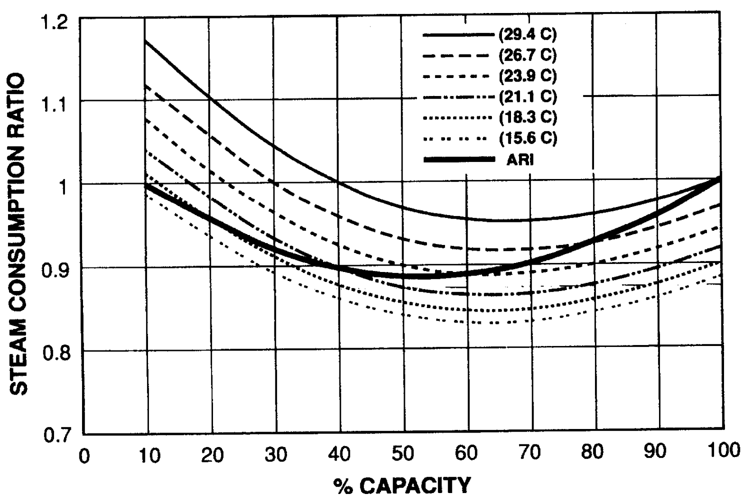

FIGURE 4.2.11 Part load performance curve for a dual-effect absorption chiller.

the system can be estimated over a wide range of conditions. The ARI load line is based on a 1.4°C (2.5°F) reduction in condenser water temperature for every 10% reduction in load. Absorption chillers are tested in accordance with ARI Standard 560 (ARI, 1992).

Chiller Controls

New chillers are equipped with electronic control systems which provide for safe operation and capacity control to meet the cooling load. The controls allow for setpoint temperature of the chilled water leaving the evaporator. This section describes some of the controls found on many chillers. For a specific chiller, the manufacturer's literature should be consulted.

Expansion Valves

The primary purpose of an expansion device is to control the amount of refrigerant entering the evaporator and to ensure that only superheated vapor reaches the compressor. In the process, the refrigerant entering the valve expands from a relatively high-pressure subcooled liquid to a saturated low-pressure liquid/vapor mixture. Other types of flow control devices, such as pressure regulators and float valves, can also be found in some refrigeration systems. Discussion of these can be found in Wang (1993). The most common expansion valves found in modern chillers are thermal expansion valves and electronic expansion valves, discussed below. Three other expansion devices are found in refrigeration systems or smaller air conditioning equipment: constant pressure expansion valves, short tube restrictors, and capillary tubes.

Thermostatic Expansion Valve — The thermostatic expansion valve (TXV) uses the superheat of the gas leaving the evaporator to control the refrigerant flow into the evaporator. Its primary function is to provide superheated vapor to the suction of the compressor. A TXV is mounted near the entrance to the evaporator and has a capillary tube extending from its top that is connected to a small bulb (Figure 4.2.12). The bulb is mounted on the refrigerant tubing near the evaporator outlet. The capillary tube and bulb are filled with the thermostatic charge (ASHRAE, 1998). This charge often consists of a vapor or liquid that is the same substance as the refrigerant used in the system. The response of the TXV and the superheat setting can be adjusted by varying the type of charge in the capillary tube and bulb.

The operation of a TXV is straightforward. Liquid enters the TXV and expands to a mixture of liquid and vapor at pressure P_2 (Figure 4.2.12). The refrigerant evaporates as it travels through the evaporator and reaches the outlet where it is superheated. If the load on the evaporator is increased, the superheat leaving the evaporator will increase. This increase in flow will increase the temperature and pressure (P_1) of the charge within the bulb and capillary tube. Within the top of the TXV is a diaphragm. With an increase in pressure of the thermostatic charge, a greater force is exerted on the diaphragm, which forces the valve port to open and allow more refrigerant into the evaporator. The larger refrigerant flow reduces the evaporator superheat back to the desired level.

The capacity of TXVs is determined on the basis of opening superheat values. TXV capacities are published for a range in evaporator temperatures and valve pressure drops. TXV ratings are based on liquid only entering the valve. The presence of flash (two-phase) gas will reduce the capacity substantially.

Electronic Expansion Valve — The electronic expansion valve (EEV) has become popular in recent years on larger or more expensive systems where its cost can be justified. EEVs can be heat motor activated, magnetically modulated, pulse width modulated, and step motor driven (ASHRAE, 1998). These are an integral part of the refrigeration system in a chiller. The EEV size and type are chosen by the manufacturer. EEVs can be used with digital control systems to provide control of the refrigeration system based on input variables from throughout the system. They offer more precise control of the refrigerant system than do TXVs. Also, some manufacturers make EEVs that are capable of flow in either direction through the EEV. This allows one EEV to replace two TXVs in heat pump applications. The selection of the valve size is similar to that for TXVs. The system refrigerant, evaporator load, liquid temperature, desired capacity, and pressure drop across the valve must be known.

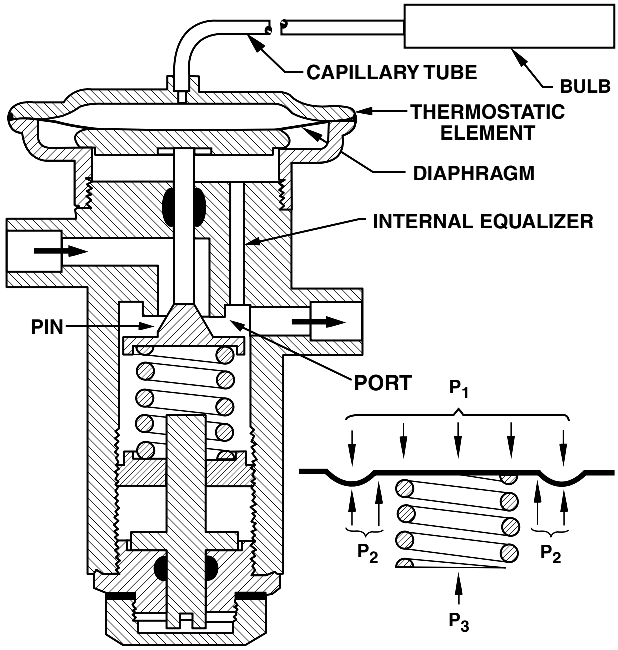

FIGURE 4.2.12 Cross section of a thermal expansion valve. P_1 = thermostatic element pressure, P_2 = evaporator pressure, and P_3 = pressure equivalent of the superheat spring force (reprinted with permission of ASHRAE).

Head Pressure Controls

Air cooled chillers operating during low outdoor temperatures require some means of maintaining an adequate condensing pressure to ensure proper system performance. The two primary reasons for needing adequate condensing pressure include (1) higher condensing pressure helps keep the refrigerant upstream of the expansion valve subcooled to prevent premature flashing in the liquid line, and (2) higher pressures ensure that the pressure differential between the condenser and evaporator are high enough so the expansion valve can provide proper control of the system.

The two most common ways of controlling head (condenser) pressure include a three-way pressure regulating valve and airflow control through the condenser. The three-way pressure regulating valve has two refrigerant inlet ports and one outlet port. One inlet comes from the outlet of the condenser and the other inlet is connected to a refrigerant line that bypasses the condenser. Under normal, high outdoor temperature operation, only refrigerant from the condenser passes through the valve. However, at low ambient temperatures, the valve begins to close off flow through the condenser and increases its pressure. This action forces refrigerant through the bypass around the condenser. This type of valve is usually preset to hold condenser pressure above a specific value.

As the outdoor temperature drops, the air conditioning load typically decreases and the capacity of the condenser increases. To reduce condenser capacity and maintain an acceptable minimum condenser pressure, the air flow through the condenser can be reduced at lower outdoor temperatures. Larger condensers have multiple fans. Individual fans can be sequenced to maintain condenser pressure. Another approach is to use variable speed control on at least one fan.

Head pressure controls are needed to maintain proper operation of the refrigerant side of the system. However, the artificially high condenser pressures created by head pressure controls decrease the efficiency of the chiller and increase energy use at low outdoor temperatures.

Capacity Controls

The type of capacity control used in a chiller depends on the type of chiller. With the growing affordability of variable speed drive technology, many chillers are equipped with variable speed electric drives. With variable speed drives, capacity is controlled directly by the speed of the compressor(s). Variable speed drives offer excellent energy saving opportunities compared to some other capacity control technologies.

Reciprocating chillers with multiple cylinders often use cylinder unloading to reduce capacity as the thermal load on the building drops. Unloading is accomplished by either bypassing gas to the suction chamber, blocking the suction or discharge valve, or closing the suction valve (Wang, 1993).

Capacity modulation for both screw and centrifugal compressors is discussed in earlier sections describing these two compressors.

A capacity control technique found on some older systems is hot-gas bypass control. With this technique, some of the hot discharge gas from the compressor bypasses the condenser and expansion valve and is introduced between the expansion valve and the evaporator. Hot-gas bypass provides a wide range of control of cooling capacity in the evaporator. However, the technique does not provide any energy saving at low loads, is discouraged by current building standards, and is prohibited in federal buildings (Wang, 1993).

Safety Controls

Chiller safety controls are provided to shut the chiller down in case of a malfunction. A short summary of some of the more important safety controls is provided below. A more complete discussion can be found in Wang (1993) and ASHRAE (1998).

Low pressure controls ensure that the compressor operates only if the suction pressure is above a set value. At low suction pressures, the refrigerant flow rate can drop below the rate needed to cool electric motors in hermetic systems. Low pressures can occur if the chilled water flow drops too low or if the chiller has lost refrigerant.

High pressure controls shut the compressor down if the discharge pressure reaches a high enough value to possibly cause damage to the compressor. High discharge temperatures are usually associated with high discharge pressures. The lubricant in the refrigeration system can begin to break down at high discharge temperatures.

Low temperature control in chillers keeps the chilled water from freezing in the evaporator. If the water freezes, the evaporator can be damaged.

Oil pressure failure control protects the compressor. Insufficient lubrication of the compressor can result from low oil pressure. Thus, the compressor would be shut down if this condition is indicated.

Motor overload controls shut down the motor to keep it from overheating caused by overloading. Thermal sensors inside the motor sense temperature in the motor windings. The electric motor current can also be measured to prevent it from exceeding a preset fixed value.

Centrifugal Chiller Controls

Controls unique to centrifugal chillers include *surge protection, air purge*, and *demand limit* controls. Surge occurs in a centrifugal compressor when the refrigerant flow is reversed and refrigerant flows from the discharge to the suction in the compressor. If surge is detected, the condenser water temperature is lowered to reduce the condenser refrigerant pressure and to eliminate the surge.

Gases, such as air and water vapor, can leak into the chiller. Purging is normally done automatically at fixed intervals to eliminate these gases. In older chillers, for every kilogram of air purged from the system, (1.5–9 kg) (3–20 lb) of refrigerant could be exhausted (Carrier, 1999). Purge systems in new chillers reduce refrigerant losses by a factor of 10 to 15 (Carrier, 1999).

Centrifugal chillers are normally applied in large buildings where electrical demand charges may be high during certain parts of the day. The demand limit controls can be used to limit current draw to 40–100% of full load. Limiting the power consumption also limits the capacity of the system.

Absorption Chiller Controls

Absorption chillers require a variety of limit and safety controls (ASHRAE, 1994). Low-temperature chilled water control allows the user to set the exiting chilled water temperature. The low-temperature refrigerant limit control reduces the loading on the chiller as the refrigerant temperature drops. If the refrigerant temperature drops enough, this control will shut off the machine. The absorbent concentration limit

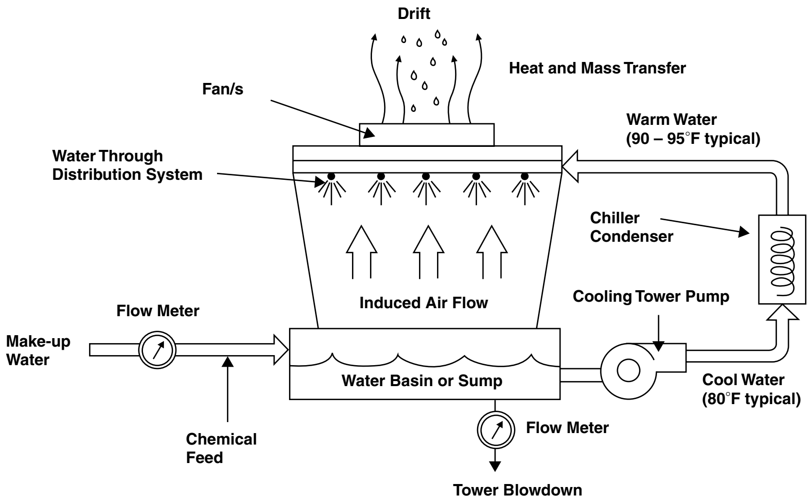

FIGURE 4.2.13 Mechanical draft cooling tower.

control prevents lithium bromide from crystallizing by reducing the loading of the chiller based on temperature and pressure measurements in the water-lithium bromide solutions. High pressure and temperature limit controls limit the pressure of the generator and the maximum operating temperature of the solution near the burners on direct fired machines. Absorption systems also include flow control switches for chilled water, condenser water, and pump motor coolant to shut down the system if flow is stopped in these circuits. Some systems provide either a modulating valve or variable speed pump to control the flow of concentrated solution from the absorber to the generator.

The cooling water entering the absorber tubes is usually limited to between 7 and 43°C (45–109°F) (ASHRAE, 1994). If temperatures drop below 7°C (45°F) or there is a sudden decrease in the cooling water temperature, crystallization of the absorbent solution can occur in the heat exchanger. Most systems have a control that limits the heat input in the generator to the entering cooling water temperature in the absorber.

Cooling Towers

If a chiller is used to provide chilled water for building air conditioning, then the heat energy that is absorbed through that process must be rejected. The two most common ways to reject thermal energy from the vapor compression process are either directly to the air or through a cooling tower. In a cooling tower, water is recirculated and evaporatively cooled through direct contact heat transfer with the ambient air. This cooled water can then be used to absorb and reject the thermal energy from the condenser of the chiller. The most common cooling tower used for HVAC applications is the mechanical draft cooling tower (Figure 4.2.13). The mechanical draft tower uses one or more fans to force air through the tower, a heat transfer media or fill that brings the recirculated water into contact with the air, a water basin (sump) to collect the recirculated water, and a water distribution system to ensure even dispersal of the water into the tower fill.

Figure 4.2.14 shows the relationship between the recirculating water and air as they interact in a counterflow cooling tower. The evaporative cooling process involves simultaneous heat and mass transfer as the water comes into contact with the atmospheric air. Ideally, the water distribution system causes the water to splash or atomize into smaller droplets, increasing the surface area of water available for heat transfer. The approach to the wet-bulb is a commonly used indicator of tower size and performance. It is defined as the temperature difference between the cooling water leaving the tower and the wet-bulb of the air entering the tower. Theoretically, the water being recirculated in a tower could reach the wet-bulb temperature, but this does not occur in actual tower operations.

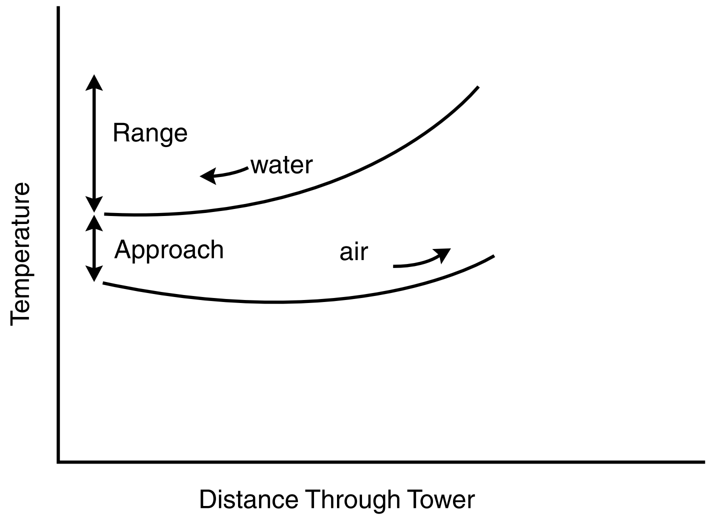

FIGURE 4.2.14 Air/water temperature relationship in a counterflow cooling tower.

The range for a chiller/tower combination is determined by the condenser thermal load and the cooling water flow rate, not by the capacity of the cooling tower. The range is defined as the temperature difference between the water entering the cooling tower and that leaving. The driver of tower performance is the ambient wet-bulb temperature. The lower the average wet-bulb temperature, the "easier" it is for the tower to attain the desired range, typically 6°C (10°F) for HVAC applications. Thus, in a hot, dry climate towers can be sized smaller than those in a hot and humid area for a given heat load.

Cooling towers are widely used because they allow designers to avoid some common problems with rejection of heat from different processes. The primary advantage of the mechanical draft cooling tower is its ability to cool water to within 3–6°C (5–10°F) of the ambient wet-bulb temperature. This means more efficient operation of the connected chilling equipment because of improved (lower) head pressure operation which is a result of the lower condensing water temperatures supplied from the tower.

Cooling Tower Designs

The ASHRAE *Systems and Equipment Handbook* (1996) describes over 10 types of cooling tower designs. Three basic cooling tower designs are used for most common HVAC applications. Based upon air and water flow direction and location of the fans, these towers can be classified as counterflow induced draft, crossflow induced draft, and counterflow forced draft.

One component common to all cooling towers is the heat transfer packing material, or fill, installed below the water distribution system and in the air path. The two most common fills are splash and film. Splash fill tends to maximize the surface area of water available for heat transfer by forcing water to break apart into smaller droplets and remain entrained in the air stream for a longer time. Successive layers of staggered splash bars are arranged through which the water is directed. Film fill achieves this effect by forcing water to flow in thin layers over densely packed fill sheets that are arranged for vertical flow. Towers using film type fill are usually more compact for a given thermal load, an advantage if space for the tower site is limited. Splash fill is not as sensitive to air or water distribution problems and performs better where water quality is so poor that excessive deposits in the fill material are a problem.

Counterflow Induced Draft — Air in a counterflow induced draft cooling tower is drawn through the tower by a fan or fans located at the top of the tower. The air enters the tower at louvers in the base and then comes into contact with water that is distributed from basins at the top of the tower. Thus, the relative directions are counter (down for the water, up for the air) in this configuration. This arrangement

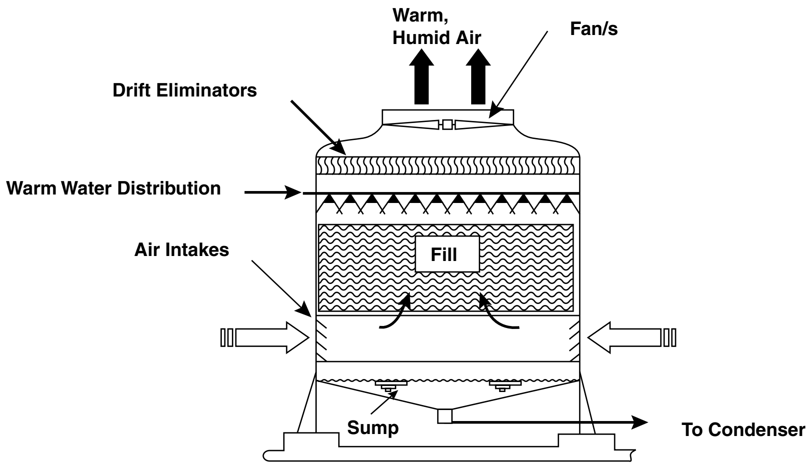

FIGURE 4.2.15 Counterflow induced draft cooling tower.

is shown in Figure 4.2.15. In this configuration, the temperature of the water decreases as it falls down through the counterflowing air, and the air is heated and humidified. Droplets of water that might have been entrained in the air stream are caught at the drift eliminators and returned to the sump. Air and some carryover droplets are ejected through the fans and out the top of the tower. The water that has been cooled collects in the sump and is pumped back to the condenser.

Counterflow towers generally have better performance than crossflow types because of the even air distribution through the tower fill material. These towers also eject air at higher velocities which reduces problems with exhaust air recirculation into the tower. However, these towers are also somewhat taller than crossflow types and thus require more condenser pump head.

Crossflow Induced Draft — As in the counterflow cooling tower, the fan in the crossflow tower is located at the top of the unit (Figure 4.2.16). Air enters the tower at side or end louvers and moves horizontally through the tower fill. Water is distributed from the top of the tower where it is directed into the fill and is cooled by direct contact heat transfer with the air in crossflow (air horizontal and water down). Water collected in the sump is pumped back to the chiller condenser. The increased airflow possible with the crossflow tower allows these towers to have a much lower overall height. This results in lower pump head required on the condenser water pump compared to the counterflow tower. The reduced height also increases the possibility of recirculating the exhaust air from the top of the tower back into the side or end air intakes which can reduce the tower's effectiveness.

Counterflow Forced Draft — Counterflow forced draft cooling towers have the fan mounted at or near the bottom of the unit near the air intakes (Figure 4.2.17). As in the other towers, water is distributed down through the tower and its fill, and through direct contact with atmospheric air it is cooled. Thermal operation of this tower is similar to the counterflow induced draft cooling tower. Fan vibration is not as severe for this arrangement compared to induced draft towers. There is also some additional evaporative cooling benefit because the fan discharges air directly across the sump which further cools the water. There are some disadvantages to this tower. First, the air distribution through the fill is uneven, which reduces tower effectiveness. Second, there is risk of exhaust air recirculation because of the high suction velocity at the fan inlets, which can reduce tower effectiveness. These towers find applications in small- and medium-sized systems.

Materials

Cooling towers operate in a continuously wet condition that requires construction materials to meet challenging criteria. Besides the wet conditions, recirculating water could have a high concentration of mineral salts due to the evaporation process. Cooling tower manufacturers build their units from a

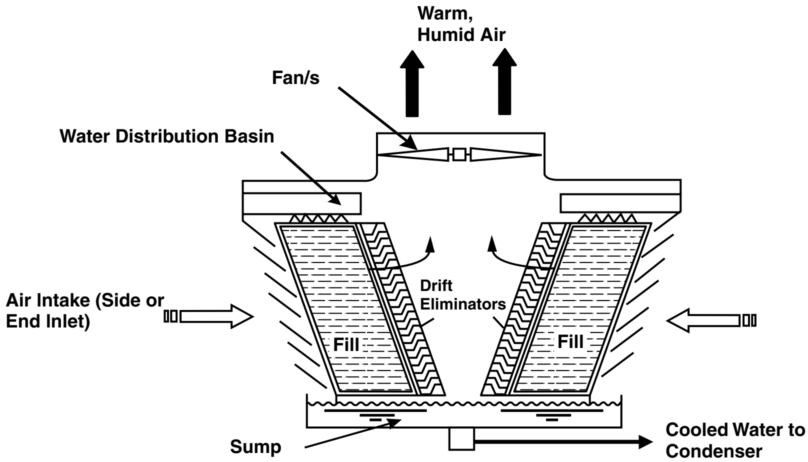

FIGURE 4.2.16 Crossflow induced draft cooling tower.

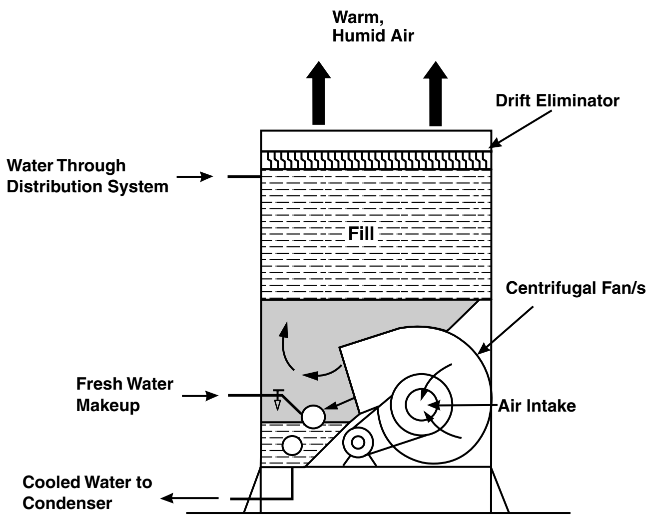

FIGURE 4.2.17 Counterflow forced draft cooling tower.

combination of materials that provide the best combination of corrosion resistance and cost. Wood is a traditional material used in cooling tower construction. Redwood or fir are often used and are usually pressure treated with preservative chemicals. Chemicals such as chromated copper arsenate or acid copper chromate help prevent decay due to fungi or destruction by termites.

Galvanized steel is commonly used for small- to mid-sized cooling tower structures. Hardware is usually made of brass or bronze. Critical components, such as drive shafts, hardware mounting points, etc., may be made from 302 or 304 stainless steel. Cast iron can be found in base castings, motor housings, and fan hubs. Metals coated with plastics are finding application for special components.

Many manufacturers make extensive use of fiberglass-reinforced plastic (FRP) in their structure, pipe, fan blades, casing, inlet louvers, and connection components. Polyvinyl chloride (PVC) is used for fill media, drift eliminators, and louvers. Fill bars and flow orifices are commonly injection molded from polypropylene and acrylonitrile butadiene styrene (ABS).

Concrete is normally used for the water basin or sump of field erected towers. Tiles or masonry are used in specialty towers when aesthetics are important.

TABLE 4.2.6 Cooling Tower Design Parameters

Value	Typical Range or Description
Heat load — kW (Btu/hr)	Determined for the specific application
Condenser water flow rate — L/s (gpm)	0.06 L/s/kW (3 gpm/ton) of rejected load is commonly used to size the cooling tower water recirculation rate
Entering condenser water temperature	32°C (90°F) to 46°C (115°F) is a common range for HVAC and refrigeration applications. A nominal value would be 35°C (95°F).
Leaving condenser water temperature	27°C (80°F) to 32°C (90°F) is a common range for HVAC and refrigeration applications. This value depends on ambient wet-bulb temperature. A nominal value would be about 27°C (80°F).
Outdoor wet-bulb temperature	Depends upon geographical location. The designer should consult local weather archives (use 1 or 2.5% summer value). A typical conservative design value is 25°C (78°F).
Range	Depends on water recirculation rate and load to be rejected. Range can be as high as 8°C (15°F). A typical value is 5.5°C (10°F).
Approach	Varies from 3°C (5°F) to 7°C (12°F) for HVAC applications. Approach less than 3°C (5°F) is not economical (extremely large tower required).

Performance

Rejection of the heat load produced at the chilling equipment is the primary goal of a cooling tower system. This heat rejection can be accomplished with an optimized system that minimizes the total compressor power requirements of the chiller and the tower loads such as the fans and condenser pumps. Several criteria must be determined before the designer can complete a thorough cooling tower analysis, including selection of tower range, water-to-air ratio, approach, fill type and configuration, and water distribution system. Table 4.2.6 lists some of the common design criteria and normally accepted ranges for cooling towers.

Most common HVAC applications requiring a cooling tower will use an "off the shelf" unit from a cooling tower manufacturer. Manufacturer representatives are usually well informed about their products and their proper application. After the project design process has produced the information called for in Table 4.2.6, it is time to contact one or more cooling tower representatives and seek their input on correct tower selection.

Control Scheme with Chillers — Most cooling towers are subject to large changes in load and ambient wet-bulb temperature during normal operations. For a typical cooling tower, the tower fan energy consumption is approximately 10% of the electric power used by the chiller compressor. The condenser pumps are about 2–5% of the compressor power. Controlling the capacity of a tower to supply adequately cooled water to the condenser while minimizing energy use is a desirable operational scheme. Probably the most common control scheme employed for towers serving an HVAC load is to maintain a fixed leaving water temperature, usually 27°C (80°F). Fan cycling is a common method to achieve this cooling tower control strategy and is applicable to multiunit and multicell tower installations. However, this control method does not minimize total energy consumed by the chiller/cooling tower system components.

Lowering the condensing water temperature increases a chiller's efficiency. As long as the evaporator temperature is constant, a reduced condenser temperature will yield a lower pressure difference between the evaporator and condenser and reduce the load on the compressor. However, it is important to recognize that the efficiency improvements initially gained through lower condenser temperatures are limited. Improved chiller efficiency may be offset by increased tower fan and pumping costs. Maintaining a constant approach at some minimum temperature is desirable as long as the condensing temperature does not fall below the chiller manufacturer's recommendations.

Since most modern towers use two- or three-speed fans, a near optimal control scheme can be developed as follows (Braun and Diderrich, 1990):

- Tower fans should be sequenced to maintain a constant approach during part load operation to minimize chiller/cooling tower energy use.
- The product of range and condensing water flow rate, or the heat energy rejected, should be used to determine the sequencing of the tower fans.
- Develop a simple relationship between tower capacity and tower fan sequencing.

De Saulles and Pearson (1997) found that savings for a setpoint control versus the near optimal control for a cooling tower were very similar. Their control scheme called for the tower to produce water at the lowest setpoint possible, but not less than the chiller manufacturer would allow, and to compare that operation to the savings obtained using near optimal control as described above. They found that the level of savings that could be achieved was dependent on the load profile and the method of optimization. Their simulations showed 2.5 to 6.5% energy savings for the single setpoint method while the near optimal control yielded savings of 3 to 8%. Use of variable speed fans would increase the savings only in most tower installations. It is more economical to operate multiple cooling tower fans at the same speed than to operate one at maximum before starting the next fan. Variable speed fans should be used when possible in cooling towers.

The system designer should ensure that any newly installed cooling tower is tested according to ASME Standard PTC 23 (ASME 1986) or CTI Standard ATC-105. These field tests ensure that the tower is performing as designed and can meet the heat rejection requirements for the connected chiller or refrigeration load.

Selection Criteria

The criteria listed in Table 4.2.6 are usually known *a priori* by the designer. If not known explicitly, then commonly accepted values can be used. These criteria are used to determine the tower capacity needed to reject the heat load at design conditions. Other considerations besides the tower's capacity include economics, servicing, environmental considerations, and aesthetics. Many of these factors are interrelated, but, if possible, they should all be evaluated when selecting a particular tower design.

Because economics is an important part of the selection process, two methods are commonly used — life-cycle costing and payback analysis. These procedures compare equipment on the basis of owning, operation, and maintenance costs. Other criteria can also affect final selection of a cooling tower design: building codes, structural considerations, serviceability, availability of qualified service personnel, and operational flexibility for changing loads. In addition, noise from towers can become a sensitive environmental issue. If local building code sound limits are an issue, sound attenuators at the air intakes and the tower fan exit should be considered. Aesthetics can be a problem with modern architectural buildings or on sites with limited land space. Several tower manufacturers can erect custom units that can completely mask the cooling tower and its operation.

Applications

Unlike chillers, pumps, and air handlers, the cooling tower must be installed in an open space with careful consideration of factors that might cause recirculation (recapture of a portion of warm and humid exhaust air by the same tower) or restrict air flow. A poor tower siting situation might lead to recirculation, a problem not restricted to wet cooling towers. Similar recirculation can occur with air-cooled condensing equipment as well. With cooling tower recirculation, performance is adversely affected by the increase in entering wet-bulb temperature. The primary causes of recirculation are poor siting of the tower adjacent to structures, inadequate exhaust air velocity, or insufficient separation between the exhaust and intake of the tower.

Multiple tower installations are susceptible to interference — when the exhaust air from one tower is drawn into a tower located downwind. Symptoms similar to the recirculation phenomenon then plague the downwind tower. For recirculation, interference, or physically blocking air-flow to the tower the result is larger approach and range which contribute to higher condensing pressure at the chiller. Both recirculation and interference can be avoided through careful planning and layout.

Another important consideration when siting a cooling tower installation is the effect of fogging, or plume, and carryover. Fogging occurs during cooler weather when moist warm air ejected from the tower comes into contact with the cold ambient air, condenses, and forms fog. Fog from cooling towers can limit visibility and can be an architectural nuisance. Carryover is when small droplets of entrained water in the air stream are not caught by the drift eliminators and are ejected in the exhaust air stream. These droplets then precipitate out from the exhaust air and fall to the ground like a light mist or rain (in extreme cases). Carryover or drift contains minerals and chemicals from the water treatment in the tower and can cause staining or discoloration of the surfaces it settles upon. To mitigate problems with fog or carryover, as with recirculation, the designer should consider nearby traffic patterns, parking areas, prevailing wind direction, large glass areas, or other architectural considerations.

Operation and Maintenance

***Winter Operation* —** If chillers or refrigeration equipment are being used in cold weather, freeze protection should be considered to avoid formation of ice on or in the cooling tower. Capacity control is one method that can be used to control water temperature in the tower and its components. Electric immersion heaters are usually installed in the tower sump to provide additional freeze protection. Since icing of the air intakes can be especially detrimental to tower performance, the fans can be reversed to de-ice these areas. If the fans are operating in extremely cold weather, ice can accumulate on the leading edges of the fan blades, which can cause serious imbalance in the fan system. Instrumentation to detect out-of-limits vibration or eccentricity in rotational loads should be installed. As with any operational equipment, frequent visual inspections during extreme weather are recommended.

***Water Treatment* —** The water circulating in a cooling tower must be at an adequate quality level to help maintain tower effectiveness and prevent maintenance problems from occurring. Impurities and dissolved solids are concentrated in tower water because of the continuous evaporation process as the water is circulated through the tower. Dirt, dust, and gases can also find their way into the tower water and either become entrained in the circulating water or settle into the tower sump. To reduce the concentration of these contaminants, a percentage of the circulating water is drained or blown-down. In smaller evaporatively cooled systems, this process is called a bleed-off and is continuous. Blow-down is usually 0.8 to 1.2% of the total water circulation rate and helps to maintain reduced impurity concentrations and to control scale formation. If the tower is served with very poor water quality, additional chemical treatments might be needed to inhibit corrosion, control biological growth, and limit the collection of silt. If the tower installation presents continuing water quality problems, a water treatment specialist should be consulted.

***Legionellosis* —** *Legionellosis* has been connected with evaporative condensers, cooling towers, and other building hydronic components. Researchers have found that well-maintained towers with good water quality control were not usually associated with contamination by *Legionella pneumophila* bacteria. In a position paper concerning *Legionellosis*, the Cooling Tower Institute (CTI, 1996) stated that cooling towers are prone to colonization by *Legionella* and have the potential to create and distribute aerosol droplets. Optimum growth of the bacteria was found to be at about 37°C (99°F) which is an easily attained temperature in a cooling tower.

The CTI proposed recommendations regarding cooling tower design and operation to minimize the presence of *Legionella*. They do not recommend frequent or routine testing for *Legionella pneumophila* bacteria because there is difficulty interpreting test results. A clean tower can quickly be reinfected, and a contaminated tower does not mean an outbreak of the disease will occur.

***Maintenance* —** The cooling tower manufacturer usually provides operating and maintenance (O&M) manuals with a new tower installation. These manuals should include a complete list of all parts used and replaceable in the tower and also details on the routine maintenance required for the cooling tower. At a minimum, the following should also be included as part of the maintenance program for a cooling tower installation.

- Periodic inspection of the entire unit to ensure it is in good repair.
- Complete periodic draining and cleaning of all wetted surfaces in the tower. This gives the opportunity to remove accumulations of dirt, slime, scale, and areas where algae or bacteria might develop.
- Periodic water treatment for biological and corrosion control.
- Continuous documentation on operation and maintenance of the tower. This develops the baseline for future O&M decisions and is very important for a proper maintenance policy.

TABLE 4.2.7 Packaged Equipment Advantages and Disadvantages

Disadvantages	Advantages
Limited performance choices because of fixed component sizing.	Individual room control is allowed.
Unitary systems generally not good for close space humidity control.	Cooling and heating are available at any time and are independent of operation in other spaces.
Space temperature control is usually two-position which causes temperature swing.	Individual ventilation, when included with the unit, is available whenever the unit is operated.
Packaged system life is relatively short.	Unit capacities are certified by the manufacturer.
Energy usage will be higher than a central system because of fixed capacity increments and tendency to oversize equipment.	Equipment in unoccupied spaces can easily be turned off which is an easy energy conservation opportunity.
Full use of economizer cycle is usually not possible.	Unitary equipment operation is usually very simple.
Air distribution control is restricted on individual room units.	Packaged equipment requires less floor space than central systems.
Sound levels of equipment can be objectionable.	First costs are low.
Outside air for ventilation is usually limited or set at a fixed quantity.	Equipment can be located such that shorter duct runs or reduced duct space is allowed.
Aesthetics of units can be unappealing.	Installation is relatively simple; no factory-trained personnel are required.
Filtering option for air flow through units can be limited.	
Condensate from units can be a nuisance.	
Maintenance can be an issue because of the number of units, location, or difficult access.	

4.2.4 Packaged Equipment

Central HVAC systems are not always the best application for a particular cooling or heating load. Initial costs for central systems are usually much higher than unitary or packaged systems. There may also be physical constraints on the size of the mechanical components that can be installed in the building. Unitary or packaged systems come factory assembled and provide only cooling or combined heating and cooling. These systems are manufactured in a variety of configurations that allow the designer to meet almost any application. Cabinet or skid-mounted for easy installation, typical units generally consist of an evaporator, blower, compressor, condenser, and, if a combined system, a heating section. The capacities of the units ranges from approximately 5 kW to 460 kW (1.5 to 130 tons). Typical unitary systems are single-packaged units (window units, rooftop units), split-system packaged units, heat pump systems, and water source heat pump systems. Unitary systems do not last as long (only 8 to 15 years) as central HVAC equipment and are often less efficient.

Unitary systems find application in buildings up to eight stories in height, but they are more generally used in one-, two-, or three-story buildings that have smaller cooling loads. They are most often used for retail spaces, small office buildings, and classrooms. Unitary equipment is available only in pre-established capacity increments with set performance characteristics, such as total L/s (cfm) delivered by the unit's air handler. Some designers combine central HVAC systems with packaged equipment used on perimeter building zones. This composite can solve humidity and space temperature requirements better than packaged units alone. This also works well in buildings where it is impractical for packaged units to serve interior spaces.

Table 4.2.7 lists some of the advantages and disadvantages of packaged and unitary HVAC equipment. Table 4.2.8 lists energy efficiency ratings (EERs) for typical electric air- and water-cooled split and single package units with capacity greater than 19 kW (65,000 Btuh).

Typically, commercial buildings use unitary systems with cooling capacities greater than 18 kW (5 tons). In some cases, however, due to space requirements, physical limitations, or small additions, residential-sized unitary systems are used. If a unitary system is 10 years or older, energy savings can be achieved by replacing unitary systems with properly sized, energy-efficient models.

TABLE 4.2.8 Unitary Package System Rating

Product Type[a] and Size	Recommended EER[b]	Best Available EER (1998)
Air source 19–40 kW (65–135 MBtuh)	10.3 or more	13.5
Air source 40–73 kW (135–240 MBtuh)	9.7 or more	11.5
Air source >73 kW (240 MBtuh)	10.0 or more	11.7
Water source 19–40 kW (65–135 MBtuh)	11.5 or more	12.5
Water source >40 kW (135 MBtuh)	11.0 or more	11.0
Product Type	Recommended SEER[c,d]	Best Available SEER (1998)
Residential Air Conditioner[e]	12.0 or more	18.0

[a] Electric air- and water-cooled split system and single package units with capacity over 19 kW (65,000 Btuh) are covered here.
[b] EER, or energy efficiency ratio, is the cooling capacity in kW (Btu/h) of the unit divided by its electrical input (in watts) at standard (ARI) conditions of 35°C (95°F) for air-cooled equipment, and 29°C (85°F) entering water for water-cooled models.
[c] Based on ARI 210/240 test procedure.
[d] SEER (seasonal energy efficiency ratio) is the total cooling output kW (Btu) provided by the unit during its normal annual usage period for cooling divided by the total energy input (in Wh) during the same period.
[e] Split system and single package units with total capacity under 19 kW (65,000 Btuh) are covered here. This analysis excludes window units and packaged terminal units.

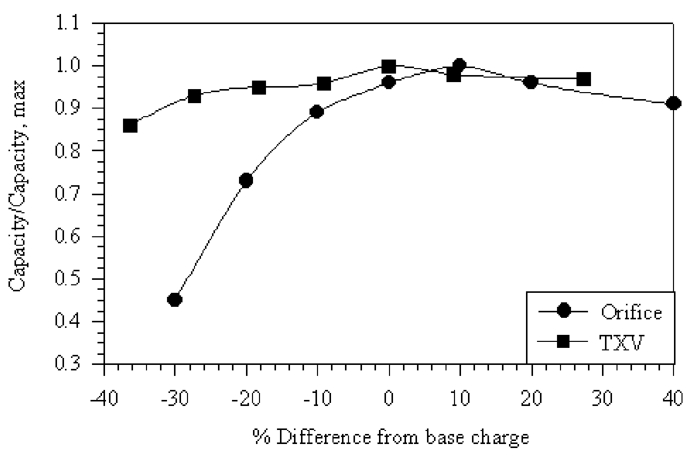

FIGURE 4.2.18 Comparison between TXV and short-tube orifice systems capacity for a range of charging conditions and 95°F (35°C) outdoor temperature. (From Rodriquez et al., 1996).

As with any HVAC equipment, proper maintenance and operation will ensure optimum performance and life for a system. Split-system air conditioners and heat pumps are the most common units applied in residential and small commercial applications. These units are typically shipped to the construction site as separate components; after the condenser (outdoor unit) and the evaporator (indoor unit) are mounted, the refrigerant piping is connected between them. The air conditioning technician must ensure that the unit is properly charged with refrigerant and check for proper operation. If the system is under- or over-charged, performance can be adversely affected. Rodriquez et al. (1996) found that performance of an air conditioning system equipped with a short tube orifice was affected by improper charge (Figure 4.2.18).

The plot in Figure 4.2.18 clearly shows that for a 20% under-charge in refrigerant, a unit with a short tube orifice suffers a 30% decrease in cooling capacity. This same study also investigated the effects of return-air leakage. A common problem with new installations is improper sealing of duct connections

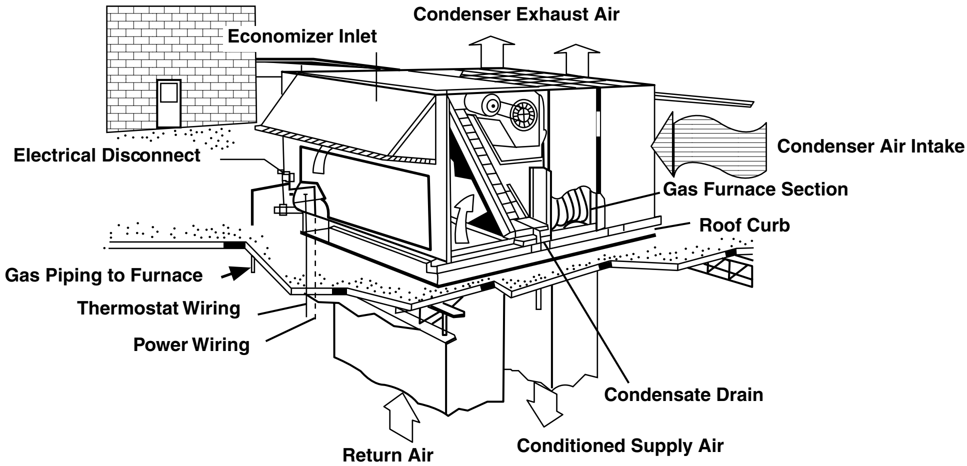

FIGURE 4.2.19 Rooftop packaged heating and air conditioning unit. (Adapted from Carrier Corporation).

at the diffusers and grills as well as around the return-air plenum. Leakage amounts as low as 5% in the return air ducts resulted in capacity and efficiency reductions of almost 20% for high humidity climates. These reductions dropped to about 7% for low humidity climates. The results of the charging and leakage studies suggest the need for the installation contractor, maintenance contractor, and system owner to ensure the proper installation of the air conditioning system.

Packaged Units

Packaged units are complete HVAC units that are usually mounted on the exterior of a structure (roof or wall) freeing up valuable indoor floor space (Figure 4.2.19). They can also be installed on a concrete housekeeping pad at ground level. Because they are self-contained, complete manufactured units, installation costs are usually lower than for a site-built HVAC system.

Single-package units consist of a blower section, filter bank, evaporator coil, at least one compressor (larger units may have more than one), and an air-cooled condensing section. Units may also come equipped with a heating section. Heating is accomplished using either natural gas or electricity. Heat pump systems can be used in situations where electricity is the only source of energy. Unitary heat pumps are restricted in size to no more than 70 kW (20 tons).

As packaged units age and deteriorate, their efficiency often decreases while the need for maintenance increases. Upgrading existing packaged units to high-efficiency models will result in substantial long-term energy savings. In the last 10 to 15 years, manufacturers have made significant improvements in the efficiency of packaged units. The efficiency of energy transfer at both the evaporator and condenser coils has been improved, high-efficiency motors are now standard, and blower and compressor designs have improved in high-efficiency packaged units. Scroll compressors are now commonplace on medium-sized (70 to 210 kW; 20 to 60 ton) rooftop units. Energy efficiencies of newer units have a SEER in the range of 9.50 to 13.0. It is not uncommon to find older units operating at efficiencies as low as 6.0, and most operate at less than 9.0. Gas-fired heating sections typically have an annual fuel utilization efficiency (AFUE) of about 80%. All newer packaged rooftop units are equipped with factory-installed microprocessor controls. These controls make maintaining equipment easier and improve energy efficiency of both the unit and the overall HVAC system. Control features include temperature setback and on/off scheduling. Larger systems can be delivered with variable air volume capability. Also, most units have an optional communication interface for connection to an energy management control system.

Vertical Packaged Units

Vertical packaged units are typically designed for indoor or through-the-wall installation. These units are applied in hotels and apartments. Some designs have a water-cooled condenser, which can be fed

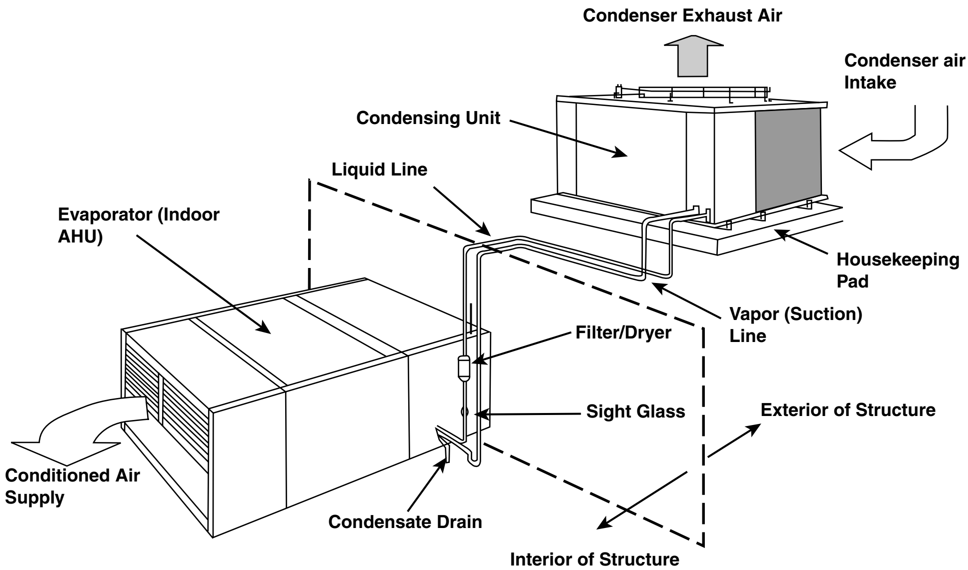

FIGURE 4.2.20 Split system diagram (courtesy of the Trane Co.).

from a cooling tower and/or city water. Many others use standard air-cooled condensers. Both style units have all other components mounted inside the package. Ductwork, if needed, can be connected to the unit to distribute the air.

Split-System Packaged Units

Split-system packaged units can have the condenser mounted on an outdoor housekeeping pad or on a rooftop. Refrigerant piping connects the compressor section to an indoor air handling unit and evaporator coil. Unless they are heat pump type units, they cannot provide heat to the space. Heating coils can be installed in the air handling section, particularly if there is a central source of heat such as hot water or steam from a boiler. Alternatively, the indoor unit can be coupled to a gas-fired furnace section to provide heating.

Air Source Heat Pumps

Air source heat pump (ASHP) systems are typically rooftop units, either packaged complete or as split systems. Split-package heat pumps are designed with an air handling unit located inside the conditioned space, while the condenser and compressor are packaged in units for outdoor installation on a housekeeping pad or on the roof. During cooling mode, the heat pump operates an air conditioner. During heating mode, the system is reversed and extracts energy from the outside air and provides it to the space. Each of these cycles is shown schematically in Figures 4.2.21 and 4.2.22, respectively. The size of unitary heat pump systems ranges from approximately 5 to 70 kW ($1^1/_2$ to 20 tons). In some cases, existing packaged cooling units with electric resistance heat can be upgraded to heat pumps for improved energy efficiency.

Heat pump applications are best suited to mild climates, such as the southeastern portion of the U.S., and to areas where natural gas for heating is less available. Space heating needs may exceed the capacity of the heat pump during extremely cold weather. This is because the units are most often sized to satisfy the cooling load requirements. As the outdoor temperature drops, the coefficient of performance (COP) of the heat pump decreases. A 26 kW ($7^1/_2$ ton) rooftop heat pump unit that has a high temperature (8.3°C) COP of 3.0 can have a low temperature (–8.3°C) COP of 2.0 or less. Because the capacity also drops with outdoor temperature, heat pumps require supplemental electric resistance heat to maintain temperature in the building. Figure 4.2.23 shows typical trends in capacity and COP for an air source heat pump. Chapter 4.2 discusses the characteristics of heat pumps.

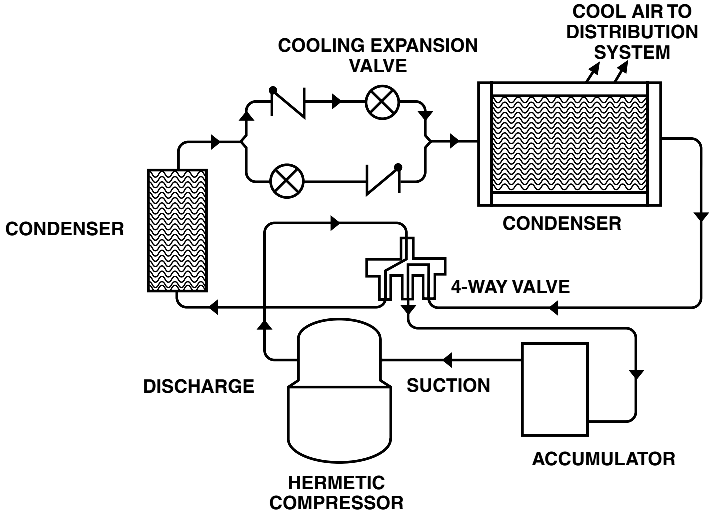

FIGURE 4.2.21 Air or water source heat pump in cooling mode (courtesy of the Trane Co.).

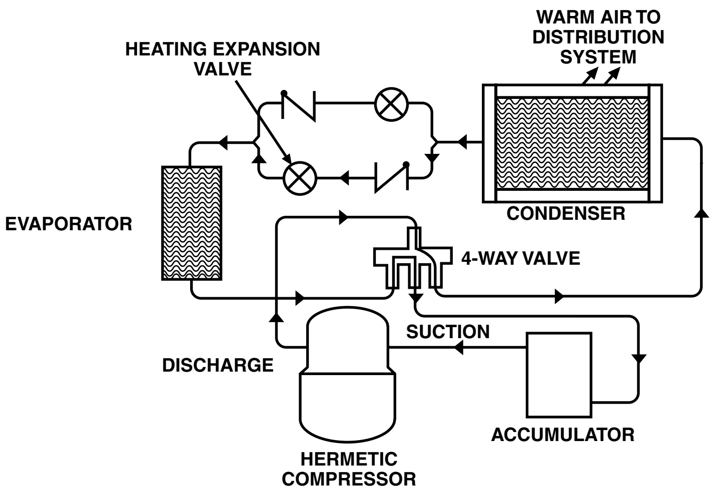

FIGURE 4.2.22 Heat pump schematic showing heating cycle (courtesy of the Trane Co.).

When the ambient air temperature approaches 0°C (32°F), heat pumps operating in the heating mode will begin to build a layer of frost on the outdoor heat exchanger. After a sufficient run-time under these conditions, the unit must go into a defrost cycle. This short (<10 min) cycle melts the frost from the heat exchanger and at the end of the cycle returns the unit to normal heating operation. During the defrost cycle, supplemental heating must be used to supply comfort heating indoors. The electrical energy penalty can become significant under extreme ambient frosting conditions (consistently cold and moist) which coincide with high space heating requirements. Various methods have been used to engage the start of the defrost cycle. A timed cycle can be set to start defrosting at a determined interval, typically about 1.5 hours. The defrost cycle can be terminated either by a control element sensing the coil pressure or a thermostat measuring the temperature of the liquid refrigerant in the outdoor coil. When this temperature reaches about 26°C (80°F) the cycle ends and the unit returns to normal heating operation. Another method utilizes two temperature sensors. One measures outdoor air, and the other responds to refrigerant temperature in the outdoor coil. As frost builds, the temperature difference between these sensors increases, and at a predetermined setpoint, the defrost cycle is started.

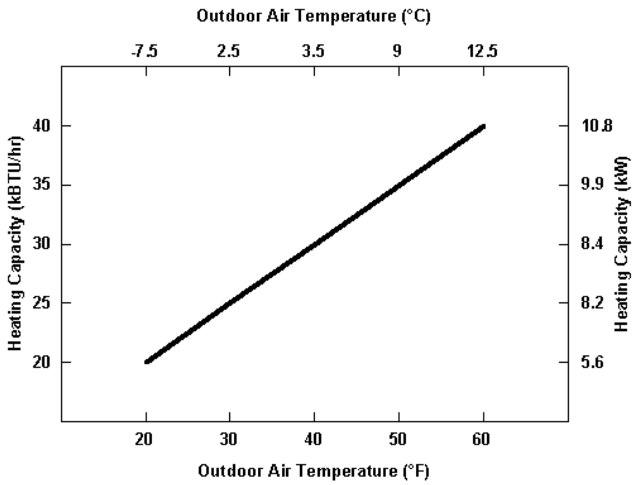

FIGURE 4.2.23 System heating capacity as a function of outdoor air temperature.

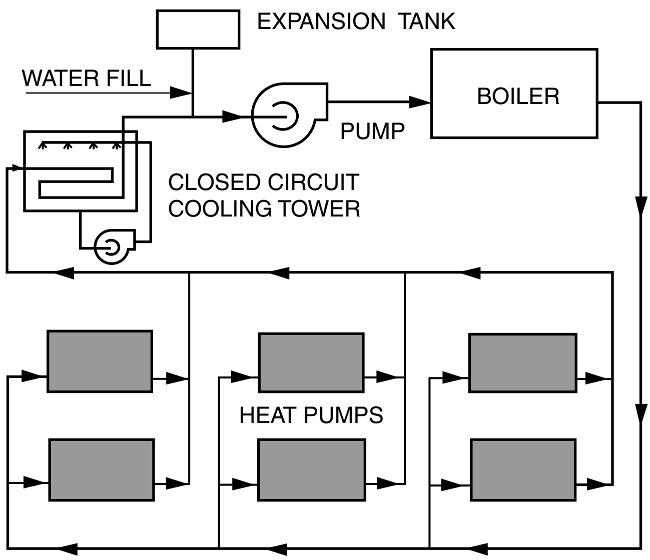

FIGURE 4.2.24 Water loop heat pump system (courtesy of SMACNA).

Water Source Heat Pumps

Water loop, or water source heat pumps use water instead of air to transfer energy between the building and outside. In an air-to-air heat pump system, energy is removed from the indoor air and rejected to the outside air during the cooling cycle. The reverse happens during the heating cycle. However, in a water loop heat pump, water replaces the outdoor air as the source or sink for energy, depending on the cycle in use. In hot weather, a cooling tower removes heat from the water loop; in cooler weather, a central boiler heats the water. As shown in Figure 4.2.24, water loop heat pump systems allow for simultaneous heating and cooling by multiple separate and distinct units and thus increase individual comfort. Recovering heat from cooled areas and recycling it into other areas adds to the system's efficiency. Size of water source heat pumps ranges from approximately 2 to 88 kW (1/2 to 25 tons). Efficiencies of water source units are generally higher than their air-to-air counterparts, with an EER of 11.0 and COP of 3.8 to 4.0 not uncommon. High-efficiency water-source heat pumps have an EER as high as 14.0 to 15.0 and a COP as high as 4.4.

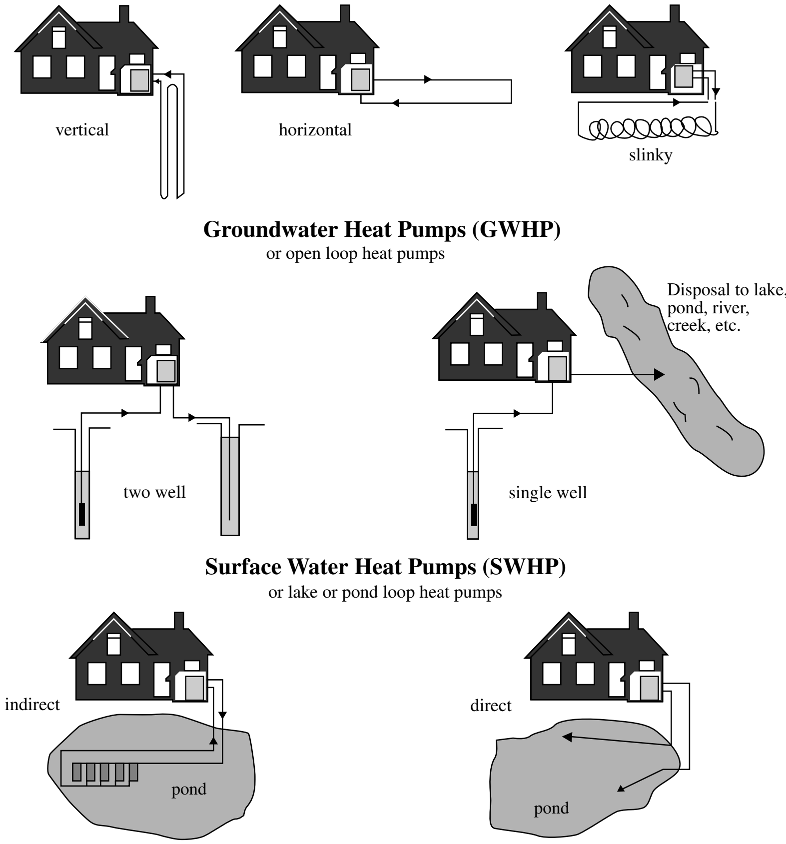

FIGURE 4.2.25 Different ground source heat pump configurations.

Geothermal Heat Pumps

Geothermal heat pumps (Figure 4.2.25) are heat pumps that draw energy from or deposit energy to the ground or groundwater. In the winter, a geothermal heat pump (GHP) transfers thermal energy from the ground or groundwater to provide space heating. In the summer, the energy transfer process is reversed; the ground or groundwater absorbs thermal energy from the conditioned space and cools the air. A GHP benefits from the nearly constant year round ground and groundwater. These temperatures are higher on average than winter air temperatures and lower on average than summer temperatures. The heat pump does not have to work as hard to extract thermal energy from or transfer energy to the ground or groundwater at a moderate temperature as from the cold air in winter or hot air in summer. The energy efficiency of a GHP is thus higher than that of a conventional ASHP. Many GHPs are also more efficient than fossil fuel furnaces in the heating mode.

Each system may also have a desuperheater to supplement the building's water heater, or a full-demand water heater to meet all of the building's hot water needs. The desuperheater transfers excess thermal energy from the GHP's compressor to a hot water tank. In summer, hot water is provided free; in the winter, water heating costs can be reduced by up to 50%. Although residential GHPs are generally more expensive to install than ASHPs, they operate more efficiently than ASHPs. GHPs can also be installed without a backup heat source over a very wide range of climates (EPA, 1993). For commercial buildings, GHPs are very competitive with boilers, chillers, and cooling towers.

The primary difference between an ASHP and a GHP is the investment in a ground loop for energy collection and rejection that is required for the GHP system. Whether a GHP is cost effective relative to a conventional ASHP depends upon generating annual energy cost savings that are high enough to pay for the ground loop in a relatively short time.

Performance Ratings

Most heating and cooling performance ratings are useful for comparing units of the same type (i.e., ASHP to ASHP, or GHP to GHP). The ratings used for different types of equipment (furnaces, ASHP, GHP), however, are not generally comparable. As a result, it is useful to know what the ratings values include. All heat pumps are rated by the ARI. Results are published in the *Directory of Certified Applied Air Conditioning Products* (for GHPs) and the *Directory of Certified Unitary Products* (for ASHPs). For water source heat pumps (the type of heat pump used in all GHP systems), cooling performance is defined by the EER. Electrical input includes compressor, fans, and "pumping" allowance (for the groundwater or ground loop). Heating performance is defined by the the COP. This is the heating effect produced by the unit divided by the energy equivalent of the electrical input resulting in a dimensionless (no units) value.

Both the COP and EER values for groundwater heat pumps are single point (valid only at the specific test conditions used in the rating) steady state values. In contrast, HSPF and SEER values published for air source equipment are seasonal values that depend on both steady state and transient tests. Ratings for GHPs are published under two different headings: ARI Standard 325 (ARI 1998b) and ARI Standard 330 (ARI 1998a). These ratings are intended for specific applications and cannot be used interchangeably. Standard 325 is intended for groundwater heat pump systems. Performance (EER and COP) is published at two water temperatures: 21°C and 10°C (70° and 50°F). The pumping penalty used in Standard 325 (ARI 1998b) is higher than the pumping allowance for Standard 330. Standard 330 is intended for closed loop or ground-coupled GHPs and is based upon entering water temperature of 25°C (77°F) in the cooling mode and 0°C (32°F) in the heating mode. One of the limitations of this rating is that the temperatures used are reflective of a northern climate. Southern installations would see higher temperatures entering the heat pump and, thus, have better winter and poorer summer performance than indicated.

The major difference between ratings for ASHPs and GHPs is that the air source values are seasonal. They are intended to reflect the total heating or cooling output for the season divided by the total electrical input for the season. These ratings (HSPF — heating, SEER — cooling) cannot be directly compared to the GHP EER and COP numbers. ASHPs are rated under Standard 210/240 (ARI 1994). To simplify the process, a number of assumptions are made regarding operation of the heat pump. The rating is based on a moderate U.S. climate and, as a result, is not reflective of either very cold or very warm areas of the country.

4.2.5 Evaporative Cooling

Evaporative air conditioning is an effective method of cooling hot, dry air. Evaporative air conditioning uses no refrigerant gases or mechanical vapor compression in producing the cooling effect. The decrease in electrical consumption and zero use of CFCs possible with evaporative air conditioning equipment means they help reduce greenhouse gas emissions and ozone depletion problems (Foster, 1991). Evaporative air conditioning is the cooling effect provided by the adiabatic evaporation of water in air. Air is drawn through wetted pads or sprays and its sensible heat energy goes towards evaporating some water which reduces the air dry-bulb temperature. In the ideal evaporative process (applies to cooling towers as well) the temperature approaches the ambient air wet-bulb temperature. A typical evaporative air conditioner or "swamp cooler" is shown in Figure 4.2.26.

These coolers contain evaporative media and a water circulating pump to lift the sump water to a distributing system which directs water down through the media and back to the sump. The fan pulls air through the evaporative media where it is cooled by direct contact with the wetted surface area and the water, and then it is delivered to the space to be cooled. Residential-sized units are either side- or down-draft depending on the evaporative media configuration. Evaporative air conditioning units are currently rated by total air delivery, and common sizes range from 5600 to 113,300 l/s (2000 to 40,000 cfm).

Two primary methods of evaporative cooling are used. *Direct cooling*, in which the water evaporates directly into the air-stream, reduces the temperature and humidifies the air. With *indirect cooling*, primary

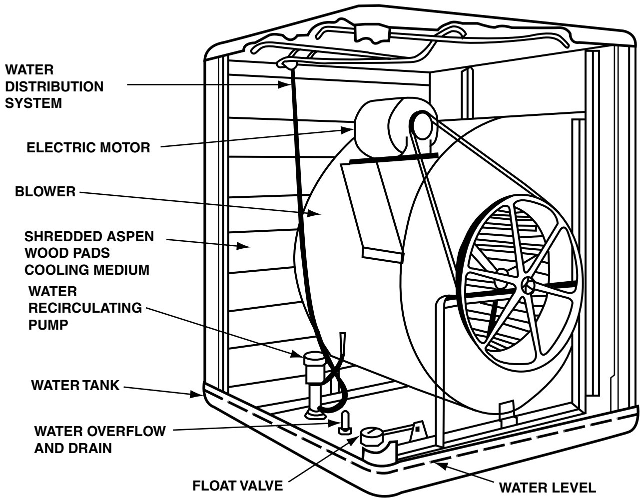

FIGURE 4.2.26 Side-draft evaporative air conditioning unit (courtesy of the Trane Co.).

air is sensibly cooled with a heat exchanger, while the secondary air operates as in the traditional direct cooling mode on the "wet" side of the heat exchanger. Clean, commercial, wetted media evaporative air conditioners typically operate at an evaporation efficiency of approximately 80%. Water use in evaporative coolers depends on air flow, the effectiveness of the wetted media, and the wet-bulb temperature of the incoming air. Fans are usually centrifugal, forward-curved types complete with motor and drive.

Evaporative air conditioning consumes significantly less energy than vapor compression refrigeration equipment of similar cooling capacity. These units operate with a fan and a small water pump. Direct systems in low humidity zones can show energy savings of 60 to 80% over mechanical cooling systems. System selection is usually based on air quantity required to properly cool a space and the system static pressure required for the duct system. To provide comfort cooling in most applications of evaporative air conditioners, 60–120 l/s/m^2 (2–4 cfm/ft^2) is adequate.

Evaporative air conditioning is useful in many commercial and industrial applications, such as schools, commercial greenhouses, laundries, warehouses, factories, kitchens (make-up air), and poultry houses, among others. The largest application area for evaporative air conditioners is in the southwestern U.S. This area experiences warm, dry weather during the cooling season and presents a good opportunity for evaporative air conditioners. They can be employed in all types of buildings that require cooling during times when ambient wet-bulb temperatures are below 18°C (65°F) and where cooling loads cannot be met with outside air only (economizer cycle). Direct evaporative air conditioners are not suitable for areas with strict humidity control requirements. See Kreider et al. (2001) for details.

Regular inspection and maintenance of evaporative air conditioners is required to ensure proper service and effectiveness of the unit. Water treatment (or bleed) is required to help prevent excessive scaling on the interior wetted surfaces of the unit.

REFERENCES

ARI 1992, *Absorption Water Chilling and Water Heating Packages*, ARI Standard 560-92, Air Conditioning and Refrigeration Institute, Arlington, VA.

ARI 1994, *Unitary Air-Conditioning and Air-Source Heat Pump Equipment*, ARI Standard 210/240, Air Conditioning and Refrigeration Institute, Arlington, VA.

ARI 1998a, *Ground Source Closed-Loop Heat Pumps*, ARI Standard 330-98, Air Conditioning and Refrigeration Institute, Arlington, VA.

ARI 1998b, *Ground Water-Source Heat Pumps*, ARI Standard 325-98, Air Conditioning and Refrigeration Institute, Arlington, VA.

ARI 1998, *Standard for Water Chilling Packages Using the Vapor Compression Cycle*, ARI Standard 550, Air Conditioning and Refrigeration Institute, Arlington, VA.

ASHRAE 1992, *Number Designation and Safety Classification of Refrigerants*, ANSI/ASHRAE Standard 34-1992, American Society of Heating, Refrigeration, and Air Conditioning Engineers, Atlanta, GA.

ASHRAE, 1994, *Safety Code for Mechanical Refrigeration*, ANSI/ASHRAE Standard 15-1994. American Society of Heating, Refrigeration, and Air Conditioning Engineers, Atlanta, GA.

ASHRAE 1995, *Methods of Testing Liquid Chilling Packages*, ASHRAE Standard 30, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

ASHRAE, 1996, *ASHRAE Handbook of HVAC Systems and Equipment*, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

ASHRAE, 1997, *ASHRAE Handbook of Fundamentals*, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

ASHRAE, 1998, *ASHRAE Handbook of Refrigeration*, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA.

ASME, 1986, Atmospheric Water Cooling Equipment, *Performance Test Code* PTC 23-86, American Society of Mechanical Engineers, New York, NY.

Bitondo, M.J. and Tozzi, M.J., 1999, *Chiller Plant Control Multiple Chiller Controls*, Carrier Corporation White Paper, Syracuse, NY, August.

Blue, J.L. et al., 1979, *Building Energy Use Data Book*, 2nd Edition, ORNL-5552, Oak Ridge National Laboratory, Oak Ridge, TN, December.

Braun, J.E. and Diderrich, G.T., 1990, Near-optimal Control of Cooling Towers for Chilled Water Systems, *ASHRAE Transactions*, 96(2), pp. 806–813.

Bullock, C.E., 1997, Theoretical Performance of Carbon Dioxide in Subcritical and Transcritical Cycles, *Refrigerants for the 21st Century*, Proc. ASHRAE/NIST Refrigerants Conf., Gaithersburg, MD, pp. 20–26.

Carrier Corporation 1999, The Money Leaking from the Mechanical Room: A Practical Guide to Addressing Chiller Leaks, *HVAC Analysis*, Vol. 2, No. 2, Syracuse, NY.

Carrier Corporation 1964, *System Design Manual, Part 7, Refrigeration Equipment*, Syracuse, NY.

CTI, 1990, Acceptance Test Code for Water-Cooling Towers, *Standard* ATC 105-90, Cooling Tower Institute, Houston, TX.

CTI, 1996, Legionellosis Position Statement, Cooling Tower Institute, Houston, TX.

De Saulles, T. and Pearson, C.C., 1997, Energy Performance Evaluation of Set Point Control and Near Optimal Control for Cooling Towers Used in Water Chilling Systems, http://www.virtual-conference.com/cibse97/conference/papers/56-content.htm.

Didion, D. 1996, Pratical Considerations in the Use of Refrigerant Mixtures, Presented at the ASHRAE Winter Meeting, Atlanta, Georgia.

EPA, 1993, *Space Conditioning: The Next Frontier*, Report 430-R-93-004, Environmental Protection Agency, Washington, D.C., April.

Foster, R.E., 1991, *Evaporative Air-Conditioning Technologies and Contributions to Reducing Greenhouse Gases*, Asia-Pacific Conf. CFC Issues and Greenhouse Effects, Singapore, May.

Holihan, P., 1998, Analysis of Geothermal Heat Pump Manufacturer's Survey Data, *Renewable Energy 1998: Issues and Trends*, Energy Information Administration.

Jackson, J. and Johnson, W., 1998, *Commercial Energy Use: A Disaggregation by Fuel, Building Type, and End Use*, ORNL/CON-14, Oak Ridge National Laboratory, Oak Ridge, TN, February.

King, G., 1986, *Basic Refrigeration*, Business News Publishing Company, Troy, Michigan.

Kramer, D., 1991, Why Not Propane?, *ASHRAE Journal*, 33, (6), pp. 52–55.

Kreider, J.F., Rabl, A., and Curtiss, P., 2001, *Heating and Cooling of Buildings*, McGraw-Hill, New York, NY.

Matsubara, K., Suefuji, K., and Kuno, H., 1987, The Latest Compressor Technologies for Heat Pumps in Japan, in *Heat Pumps*, K. Zimmerman and R.H. Powell, Jr., Eds., Lewis Chelsea, MI.

Molina, M.J. and Rowland, F.S., 1974, Stratospheric Sink for Chlorofluoromethanes: Chlorine Atoms Catalyzed Destruction of Ozone, *Nature*, 249, pp. 810–812.

National Refrigerants, 1992, *Refrigerant Reference Guide*, Philadelphia, PA.
Rodriquez, A.G., O'Neal, D.L., Bain, J.A, and Davis, M.A., 1996*, The Effect of Refrigerant Charge, Duct Leakage, and Evaporator Air Flow on the High Temperature Performance of Air Conditioners and Heat Pumps, Final Report*, EPRI TR-106542, July.
Salas, C. E. and Salas, M., 1992, *Guide to Refrigeration CFCs*, Fairmont Press, Liburn, GA.
Sand, J.R., Fischer, S.K., and Baxter, V.D., 1999, Comparison of TEWI for Fluorocarbon Alternative Refrigerants and Technologies in Residential Heat Pumps and Air-Conditioners, *ASHRAE Transactions*, Vol. 105, Pt. 1, pp. 1209–1218.
Senshu, T. Araik, A., Oguni, K., and Harada, F., 1985, Annual Energy-Saving Effect of Capacity Modulated Air Conditioner Equipped with Inverter-Driven Scroll Compressor, *ASHRAE Transactions*, Vol. 91, Part 2.
Smit, K., Keder, J., and Tidball, R., 1996, *Electric Chiller Handbook*, TR-105951, Electric Power Research Institute, Palo Alto, CA, February.
Stoecker, W.F., 1994, Comparison of Ammonia with Other Refrigerants for District Cooling Plant Chillers, *ASHRAE Transactions*, Vol. 100, Pt. 1, pp. 1126–1135.
Thevenot, R. 1979, *A History of Refrigeration Throughout the World*, International Institute of Refrigeration, Paris, France, pp. 39–46.
Trane, 1980, *Centrifugal Water Chillers*, The Trane Company, La Crosse, WI.
Troth, S. J., 1994, Air Conditioning with Ammonia for District Cooling, *ASHRAE Journal*, Vol. 36, No. 7, pp. 28–36.
United Nations Environmental Program (UNEP), 1987, *Montreal Protocol on Substances that Deplete the Ozone Layer — Final Act.*
U.S. Census Bureau, 1999, *Statistical Abstract of the United States*, Washington, D.C.
U.S. Department of Energy, 1998, *A Look at Commercial Buildings in 1995: Characteristics, Energy Consumption, and Energy Expenditures*, DOE/EIA-0625(95), Washington, D.C., October.
U.S. Environmental Protection Agency, 1993a, Class I Nonessential Products Ban, Section 610 of the Clean Air Act Amendments of 1990, *Federal Register*, 4768, January 15, 1993.
U.S. Environmental Protection Agency, 1993b, The Accelerated Phaseout of Ozone-Depleting Substances, *Federal Register*, December 10, 1993.
Wang, S. K., 1994, *Handbook of Air Conditioning and Refrigeration*, McGraw-Hill, Inc., New York, NY.

4.3 Ventilation and Air Handling Systems

Ellen M. Franconi and James B. Bradford

Secondary systems transfer heating and cooling energy between central plants and building spaces. This chapter introduces all popular air handling systems and discusses common configurations. Descriptions and design considerations are presented for the following air system components: air filters, humidifiers, coils, fans, ducts, terminal units, and diffusers. The chapter concludes with a discussion of air system controls and an overview of system design procedures.

Air-handling systems encompass the components and function of mechanical ventilation systems and provide air conditioning as well. Thus, air handling systems include a central cooling coil in addition to fans, heating, humidification, heat reclamation, and cooling through use of outdoor air.

Most modern, large commercial buildings require air conditioning to maintain occupant comfort. Historically, commercial buildings had shallow floor plans to maximize natural lighting. Their space-conditioning loads were shell-dominant. With the advent of fluorescent fixtures, air conditioning, and larger building designs, commercial buildings have become internal-gain dominant — requiring year-round air conditioning regardless of climate.

All-air systems meet the entire cooling load with cold air supplied to the conditioned space. Heating may also be supplied through the air system, at the zone, or both. Instead of air systems, water systems may be used to meet air conditioning loads. Because air has a much lower heat capacity than water, a

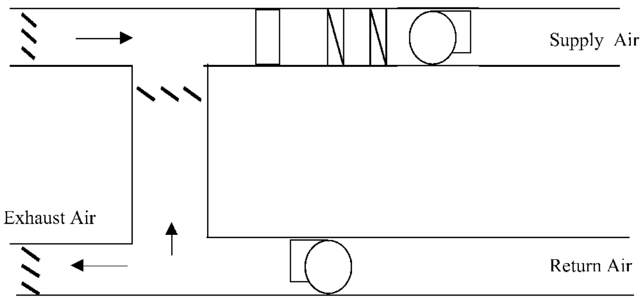

FIGURE 4.3.1 Simple air-handling unit.

much larger volume of air must be moved to meet the same load. Air systems are convenient because they incorporate ventilation within the system. Also, they are well-suited for utilizing an air-side economizer and controlling humidity. Air systems require no piping in the occupied building space. However, air systems do require more building space to accommodate the ductwork. Leaking ducts may not damage building interiors or structure, but leaks are not usually detected and they can decrease system efficiency.

4.3.1 Anatomy of Air Handling Systems

The function of an air handling system is to supply conditioned air to one or several building zones. The term *zone* refers to a thermal space that has comfort conditions controlled by a single thermostat. The air handling system supplies air at a specific flow rate and temperature to the zone in order to meet its heating or cooling load.

A building's HVAC air system may be distributed or centralized. For example, distributed systems can use local direct-expansion, packaged systems that have window, wall, or exterior mounted installations located close to the zones served. Centralized systems are installed in building mechanical rooms and provide heating and cooling to the zones through extensive ductwork. Central air systems distribute cooling and heating provided by the building plant. Generally, plant equipment for centralized systems includes chillers, cooling towers, and boilers.

While this distinction between air system types exists, the basic anatomy of all-air systems is similar. Figure 4.3.1 presents the layout for a typical system. The fundamental equipment components found in most air handling units (AHU) include dampers, air filtration devices, coils, and fans. The system may also include humidifiers and heat recovery devices.

As shown in Figure 4.3.1, during operation of a typical AHU, the outdoor air mixes with return air and passes through an air filter and a variety of air conditioning devices. The cooling coils cool, or the heating coils heat, the mixed air to maintain a supply, return, or zone air setpoint temperature. In systems that control both temperature and humidity, a heating or reheat coil may be present downstream of the cooling coil. The reheat coil raises the temperature of the cooled supply air if the cooling coil over-cools the air in order to remove humidity. Preheat coils may be present upstream of the cooling coil if it is necessary to heat the outside air stream to prevent freezing of the cooling coil during cold outdoor periods. If installed, the humidifier adds moisture to the air stream during winter months.

The supply fan draws and/or blows the air through the AHU equipment components, ductwork, terminal units, and diffusers to supply the required air flow rate to the zone. Return-air fans may be required in central systems to overcome the return system pressure drop and move the air from the building to the AHU or exhaust the air from the building.

Central System Advantages and Disadvantages

There are several advantages that central, all-air system designs have over other types of systems, including water systems and distributed (single zone) air systems. Some advantages include

- The location of equipment is in a centralized, unoccupied location that consolidates and facilitates maintenance.
- Piping is not within the conditioned space, reducing the possibility of damage in occupied areas.
- Air systems make it possible to cool the building with outdoor air.
- Air systems provide flexibility in zoning and comfort control.

Some of the disadvantages associated with all-air central distribution systems are

- Additional space is required for duct work.
- The central distribution fan may frequently need to operate during unoccupied hours in cold climates.
- Proper operation and zone comfort rely on a thorough air-balancing of the system.
- Extensive cooling and reheating of supply air may be required for systems serving zones with diverse loads.

System Configurations

While air handlers share a basic form and set of components, the components can be arranged and controlled in different ways. In general, the configuration categories describe the number of zones, duct air-path, and fan type. Specifically, the system categories include

- single or multiple zone
- single or dual air paths
- constant or variable air volume

Single or Multiple Zone

Many air handling systems provide conditioned air to a single zone. For large multistory buildings, however, it is not practical to use many AHUs that each serve only one zone. Instead, air handling systems designed to serve several zones, each with is own thermostat control, are used. The multiple zone air system design presents challenges to the engineer to accommodate diverse loads while maintaining system efficiency.

The system diagram shown in Figure 4.3.1 is complete in its representation for the simplest of all air systems — a single zone system. In multiple zone systems in large buildings, one AHU may supply space conditioning to all zones on 20 or more floors. Figure 4.3.2 shows a system schematic that represents the layout of a multiple zone, centralized system.

In a single zone system, the supply air flow rate is constant when the fan is on. To satisfy variations in the zone load in off-design conditions, the system fan may cycle on and off, or the supply-air temperature may vary. For multiple zone systems, a means for varying the amount of cooling and heating supplied to each zone must be included as part of the system design. The means for meeting diverse zone loads include varying the temperature of the air introduced to the zone or varying its volumetric flow rate. More details of how different configurations and controls are used in multizone systems are provided in the system category descriptions below.

Single or Dual Duct Systems

The distribution system air path may be either single duct or dual duct. Systems are defined as single duct if they have a single air path for supplying both heating and cooling by the system. All single zone systems are the single duct type, while multiple zones may have a single duct or a dual duct arrangement. In multiple zone, single duct systems, there is a primary air stream that serves each zone and a terminal unit that can reduce air flow and/or add heat to the supply air stream to meet the zone setpoint.

A dual-duct system supplies heating and cooling in separate ducts, each referred to as a *deck*. Typically, the hot deck is maintained at 90–95°F. The cold deck is maintained at 50–55°F. The heated and cooled air streams are blended by thermostatically controlled mixing boxes to provide the proper temperature and flow of air to each zone. In these systems, the warm and cool air streams may be mixed near the

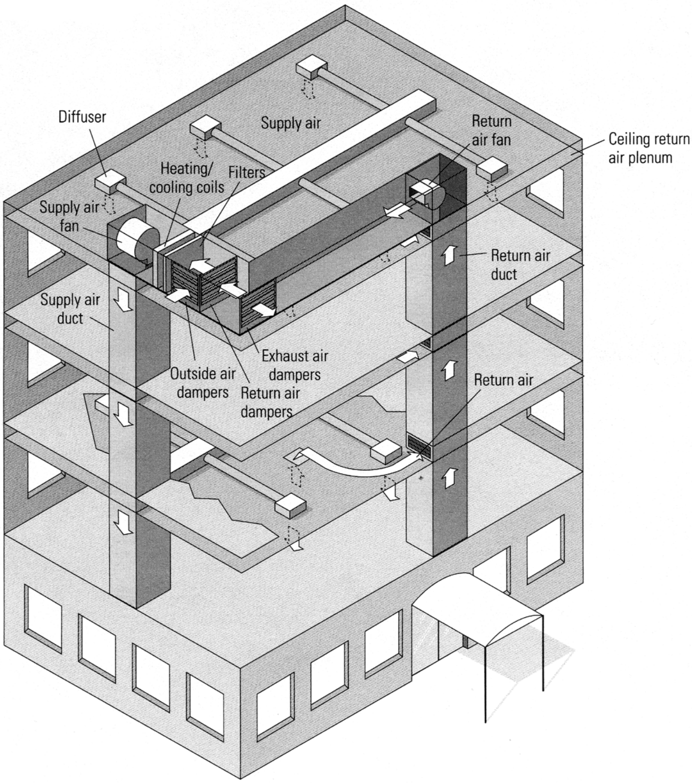

FIGURE 4.3.2 Multiple zone system schematic (courtesy of E-source, Boulder, CO).

central supply fan or near the zone. Systems that mix supply air near the zone are referred to as *dual duct systems* while systems that mix supply air near the central air handler are called *multizone systems*. While the convention is to name such systems as described above, the system names are poorly chosen since both types have dual ducts and serve multizones. Figures 4.3.3 and 4.3.4 present system schematics for the two types of systems, a dual duct and a multizone system, respectively. As shown, the two systems are functionally the same, but the location of the air blending is different. These systems are by nature inefficient since hot and cold energy streams are used simultaneously to meet zone loads. This approach goes against second law of thermodynamics design guidelines. In addition, the multizone system violates low-pressure, fan-power-saving design principles since the proximity of the mixing box to the fan results in high pressure drops and box damper leakage due to high flow velocities.

Variable or Constant Air Volume Systems

As mentioned previously, single duct, single zone systems are constant air volume systems. Single duct multiple zone systems, dual duct systems, and multizone* systems may be either *constant air volume systems* or *variable air volume systems*. Constant air volume (CAV) systems supply a constant flow rate of air to the building whenever the fans are on. CAV systems use the simplest type of AHU. In variable air volume (VAV) systems, flow modulation is achieved through fan dampering or motor speed adjustment.

Both CAV and VAV systems are engineered to meet the same peak building zone loads. However, the means by which the systems meet off-design conditions vary. Frequently in multiple zone CAV systems, supply air is cooled to a constant temperature sufficient for meeting design loads. In off-design conditions,

*The terms "multiple zone" and "multizone" are distinct. "Multiple zone" describes buildings with more than one zone; "multizone" is a type of air handler that creates conditioned air streams for several zones by mixing hot and cold streams responding to thermostat signals from each zone.