SECTION THIRTEEN
HEATING, VENTILATION, AND AIR CONDITIONING

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The necessity of heating, ventilation, and air-conditioning (HVAC) control of environmental conditions within buildings has been well established over the years as being highly desirable for various types of occupancy and comfort conditions as well as for many industrial manufacturing processes. In fact, without HVAC systems, many manufactured products produced by industry that are literally taken for granted would not be available today.

13.1 DEFINITIONS OF TERMS OF HEATING, VENTILATION, AND AIR CONDITIONING (HVAC)

Adiabatic Process. A thermodynamic process that takes place without any heat being added or subtracted and at constant total heat.

Air, Makeup. New, or fresh, air brought into a building to replace losses due to exfiltration and exhausts, such as those from ventilation and chemical hoods.

Air, Return (Recirculated). Air that leaves a conditioned spaced and is returned to the air conditioning equipment for treatment.

Air, Saturated. Air that is fully saturated with water vapor (100% humidity), with the air and water vapor at the same temperature.

Air, Standard. Air at 70°F (21°C) and standard atmospheric pressure [29.92 in (101.3 kPa) of mercury] and weighing about 0.075 lb/ft^3 (1.20 kg/m^3).

Air Change. The complete replacement of room air volume with new supply air.

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Air Conditioning. The process of altering air supply to control simultaneously its humidity, temperature, cleanliness, and distribution to meet specific criteria for a space. Air conditioning may either increase or decrease the space temperature.

Air Conditioning, Comfort. Use of air conditioning solely for human comfort, as compared with conditioning for industrial processes or manufacturing.

Air Conditioning, Industrial. Use of air conditioning in industrial plants where the prime objective is enhancement of a manufacturing process rather than human comfort.

Baseboard Radiation. A heat-surface device, such as a finned tube with a decorative cover.

Blow. Horizontal distance from a supply-air discharge register to a point at which the supply-air velocity reduces to 50 ft/min.

Boiler. A cast-iron or steel container fired with solid, liquid, or gaseous fuels to generate hot water or steam for use in heating a building through an appropriate distribution system.

Boiler-Burner Unit. A boiler with a matching burner whose heat-release capacity equals the boiler heating capacity less certain losses.

Boiler Heating Surface. The interior heating surface of a boiler subject to heat on one side and transmitting heat to air or hot water on the other side.

Boiler Horsepower. The energy required to evaporate 34.5 lb/hr of water at 212°F, equivalent to 33,475 Btu/hr.

Booster Water Pump. In hot-water heating systems, the circulating pump used to move the heating medium through the piping system.

British Thermal Unit (Btu). Quantity of heat required to raise the temperature of 1 lb of water 1°F at or near 39.2°F, which is its temperature of maximum density.

Central Heating or Cooling Plant. One large heating or cooling unit used to heat or cool many rooms, spaces, or zones or several buildings, as compared to individual room, zone, or building units.

Coefficient of Performance. For machinery and heat pumps, the ratio of the effect produced to the total power of electrical input consumed.

Comfort Zone. An area plotted on a psychometric chart to indicate a combination of temperatures and humidities at which, in controlled tests, more than 50% of the persons were comfortable.

Condensate. Liquid formed by the condensation of steam or water vapor.

Condensers. Special equipment used in air conditioning to liquefy a gas.

Condensing Unit. A complete refrigerating system in one assembly, including the refrigerant compressor, motor, condenser, receiver, and other necessary accessories.

Conductance, Thermal *C*. Rate of heat flow across a unit area (usually 1 ft^2) from one surface to the opposite surface under steady-state conditions with a unit temperature difference between the two surfaces.

Conduction, Thermal. A process in which heat energy is transferred through matter by transmission of kinetic energy from particle to particle, the heat flowing from hot points to cooler ones.

Conductivity, Thermal. Quantity of heat energy, usually in Btu, that is transmitted through a substance per unit of time (usually 1 hr) from a unit area (usually 1 ft^2) of surface to an opposite unit surface per unit of thickness (usually 1 in) under a unit temperature difference (usually 1°F) between the surfaces.

Convection. A means of transferring heat in air by natural movement, usually a rotary or circulatory motion caused by warm air rising and cooler air falling.

Cooling. A heat-removal process usually accomplished with air-conditioning equipment.

Cooling, Evaporative. Cooling effect produced by evaporation of water, the required heat for the process being taken from the air. (This method is widely used in dry climates with low wet-bulb temperatures.)

Cooling, Sensible. Cooling of a unit volume of air by a reduction in temperature only.

Cooling Effect, Total. The difference in total heat in an airstream entering and leaving a refrigerant evaporator or cooling coil.

Cooling Tower. A mechanical device used to cool water by evaporation in the outside air. Towers may be atmospheric or induced- or powered-draft type.

Cooling Unit, Self-Contained. A complete air-conditioning assembly consisting of a compressor, evaporator, condenser, fan motor, and air filter ready for plug-in to an electric power supply.

Damper. A plate-type device used to regulate flow of air or gas in a pipe or duct.

Defrosting. A process used for removing ice from a refrigerant coil.

Degree Day. The product of 1 day (24 hr) and the number of degrees Fahrenheit the daily mean temperature is below 65°F. It is frequently used to determine heating-load efficiency and fuel consumption.

Dehumidification. In air conditioning, the removal of water vapor from supply air by condensation of water vapor on the cold surface of a cooling coil.

Diffuser (Register). Outlet for supply air into a space or zone. See also Grille below.

Direct Digital Control (DDC). An electronic control system that uses a computer to analyze HVAC parameters to operate control devices and to start, stop, and optimize mechanical equipment.

Direct Expansion. A means of air conditioning that uses the concept of refrigerant expansion (through a thermostatic expansion valve) in a refrigerant coil to produce a cooling effect.

Ductwork. An arrangement of sheet-metal ducts to distribute supply air, return air, and exhaust air.

Efficiency. Ratio of power output to power input. It does not include considerations of load factor or coefficient of performance.

Emissivity. Ratio of radiant energy that is emitted by a body to that emitted by a perfect black body. An emissivity of 1 is assigned to a perfect black body. A perfect reflector is assigned an emissivity of 0.

Enthalpy. A measure of the total heat (sensible and latent) in a substance and which is equal to its internal energy and its capacity to do work.

Entropy. The ratio of the heat added to a material or substance to the absolute temperature at which the heat is added.

Evaporator. A cooling coil in a refrigeration system in which the refrigerant is evaporated and absorbs heat from the surrounding fluid (airstream).

Exfiltration. Unintentional loss of conditioned supply air by leakage from ductwork, rooms, spaces, etc., that is to be considered a load on the air-conditioning system.

Film Coefficient (Surface Coefficient). Heat transferred from a surface to air or other fluid by convection per unit area of surface per degree temperature difference between the surface and the fluid.

Furnace, Warm-Air. Heating system that uses a direct- or indirect-fired boiler to produce warm air for heating.

Grille. A metal covering, usually decorative, with openings through which supply or return air passes.

Head. Pressure expressed in inches or feet of water. A head of 12 in, or 1 ft, of water is the pressure equivalent to a column of water 12 in, or 1 ft, high. See also Inch of Water below.

Heat, Latent. Heat associated with the change of state (phase) of a substance, for example, from a solid to a liquid (ice to water) or from a liquid to a gas (water to steam vapor).

Heat, Sensible. Heat associated with a change in temperature of a substance.

Heat, Specific. Ratio of the thermal capacity of a substance to the thermal capacity of water.

Heat, Total. Sum of the sensible and latent heat in a substance above an arbitrary datum, usually 32°F or 0°C.

Heat Capacity. Heat energy required to change the temperature of a specific quantity of material 1°.

Heat Pump. A refrigerant system used for heating and cooling purposes.

Heat Transmission Coefficient. Quantity of heat (usually Btu in the United States) transmitted from one substance to another per unit of time (usually 1 hr) through one unit of surface (usually 1 ft^2) of building material per unit of temperature difference (usually 1°F).

Heater, Direct-Fired. A heater that utilizes a flame within a combustion chamber to heat the walls of the chamber and transfers the heat from the walls to air for space heating, as in a warm-air heater.

Heater, Unit. A steam or hot-water heating coil, with a blower or fan and motor, used for space heating.

Heating. The process of transferring heat from a heat source to a space in a building.

Heating, District. A large, central heating facility that provides heat from steam or hot water to a large number of buildings often under different ownership.

Heating, Radiant. Heating by ceiling or wall panels, or both, with surface temperatures higher than that of the human body in such a manner that the heat loss from occupants of the space by radiation is controlled.

Heating, Warm-Air. A heating system that uses warm air, rather than steam or hot water, as the heating medium.

Heating Surface. Actual surface used for transferring heat in a boiler, furnace, or heat exchanger.

Heating System, Automatic. A complete heating system with automatic controls to permit operation without manual controls or human attention.

Heating System, Hot-Water. A heating system that utilizes water at temperatures of about 200°F.

Humidity. Water vapor mixed with dry air.

Humidity, Absolute. Weight of water vapor per unit volume of a vapor-air mixture. It is usually expressed in grains/ft^3 or lb/ft^3.

Humidity, Percent. Ratio of humidity in a volume of air to the maximum amount of water vapor that the air can hold at a given temperature, expressed as a percentage.

Humidity, Relative (RH). Ratio of the vapor pressure in a mixture of air and water vapor to the vapor pressure of the air when saturated at the same temperature.

Humidity, Specific (Humidity Ratio). Ratio of the weight of water vapor, grains, or pounds, per pound of dry air, at a specific temperature.

Hygrometer. A mechanical device used to measure the moisture content of air.

Hygroscopic. Denoting any material that readily absorbs moisture and retains it.

Hygrostat. A mechanical device that is sensitive to changes in humidity and used to actuate other mechanical devices when predetermined limits of humidity are reached.

Inch of Water. A unit of pressure intensity applied to low-pressure systems, such as air-conditioning ducts. It is equivalent to 0.036136 psi.

Infiltration. Leakage into a(n air)-conditioned area of outside air (usually unwanted), which becomes a load on the (air-)conditioning system.

Insulation, Thermal. Any material that slows down the rate of heat transfer (offers thermal resistance) and effects a reduction of heat loss.

Louvers. An arrangement of blades to provide air slots that will permit passage of air and exclude rain or snow.

MBH. 1000 Btu/hr (Btu/h).

Micron. 0.001 mm. It is frequently used to designate particle sizes of dust and the efficiency of filtration by air-conditioning filters.

Modulating. Process of making incremental adjustments, usually by an automatic device operating a valve or damper motor.

Pressure, Absolute. Pressure above an absolute vacuum. Absolute pressure equals the sum of gage and barometric pressures.

Pressure, Atmospheric. Air pressure indicated by a barometer. The standard atmospheric pressure is 29.92 in of mercury, or 14.696 psi (101.3 kPa).

Pressure, Head. Condensing pressure, often considered as the refrigerant compressor-discharge pressure.

Pressure, Saturation. The pressure that corresponds to a specific temperature that will permit simultaneous condensation and evaporation.

Pressure, Suction. The pressure in the suction line of a refrigeration system.

Pressure, Head. See Head above.

Psychrometer. A mechanical device utilizing a wet-bulb and dry-bulb thermometer and used to determine the humidity in an air-water vapor mixture, such as room air.

Psychrometric Chart. A chart used in air-conditioning design and analysis that indicates various properties of an air-water vapor mixture along with various relevant mathematical values.

Psychrometry. A branch of physics that concerns itself with the measurement and determination of atmospheric conditions, with particular emphasis on moisture mixed in the air.

Radiation. Transfer of energy in wave form, from a hot body to a colder body, independent of any matter between the two bodies.

Radiation, Equivalent Direct. Rate of steam condensation at 240 Btu/(hr)(ft^2) of radiator surface.

Refrigerant. A substance that will accept large quantities of heat, that will cause boiling and vaporization at certain temperatures, and that can be utilized in air-conditioning systems.

Register. See Diffuser.

Resistance, Thermal. The thermal quality of a material that resists passage of heat. Also, the opposite of conductance.

Resistivity, Thermal. The reciprocal of conductivity.

Split System. A separation of air-conditioning components, such as location of an air-blower-evaporator coil far from the compressor-condenser unit.

Steam. Water in gas or vapor form.

Steam Trap. A mechanical device that allows water and air to pass but prevents passage of steam.

Subcooling. Cooling at constant pressure of a refrigerant liquid to below its condensing temperature.

Suction Line. The low-temperature, low-pressure refrigerant pipe from an evaporator to a refrigerant compressor.

Sun Effect. Heat from the sun that tends to increase the internal temperature of a space or building.

Temperature, Absolute. Temperature measured on a scale for which zero is set at −273.16°C, or −459.69°F (presumably the temperature at which all molecular motion stops in a gas under constant pressure). The scale is called Kelvin, and 1°K = 1°C = 9/5°F.

Temperature, Design. An arbitrary design criterion used to determine equipment size to produce air conditioning, heating, or cooling capable of maintaining the designated temperature.

Temperature, Dew Point. Temperature of air at which its wet-bulb temperature and dry-bulb temperature are identical and the air is fully saturated with moisture. Condensation of water vapor begins at this temperature and will continue if the temperature is reduced further.

Temperature, Dry-Bulb. Temperature measured by a conventional thermometer. It is used to determine the sensible heat in air.

Temperature, Effective. A single or arbitrary index that combines into a single value the effects of temperature, humidity, and air motion on the sensation of comfort. This value is that of the temperature of still, saturated air that will induce an identical feeling of comfort.

Temperature, Wet-Bulb. Air temperature as indicated by a thermometer with a wet bulb. This temperature is less than the dry-bulb temperature, except when the air is fully saturated with water vapor, or at 100% relative humidity, when wet-bulb and dry-bulb temperatures will be equal.

Ton, Refrigeration. Refrigeration effect equivalent to 200 Btu/min, or 12,000 Btu/hr.

Vapor. The gaseous state of water and other liquid substances.

Vapor Barrier. An impervious material used to prevent the passage of water vapor and to prevent condensation.

Velocity Pressure. The pressure caused by a moving airstream, composed of both velocity pressure and static pressure.

Ventilation. The process of supplying air to any space within a building without noticeable odors and without objectionable levels of contaminants, such as dusts and harmful gases, and of removing stale, polluted air from the space. Outside air is generally used as an acceptable source of ventilation air.

Ventilator, Unit. A type of unit heater with various modes of operation and degrees or percentages of outside air (frequently used for heating classrooms).

Volume, Specific. Volume, ft^3/lb, occupied by a unit weight of air.

Water, Makeup. Generally the water supplied to a cooling tower to replace the cooling water lost by evaporation or bleedoff.

Water Vapor. A psychrometric term used to denote the water in air (actually low-pressure, superheated steam) that has been evaporated into the air at a temperature corresponding to the boiling temperature of water at that very low pressure.

13.2 HEAT AND HUMIDITY

People have always struggled with the problem of being comfortable in their environment. First attempts were to use fire directly to provide heat through cold winters. It was only in recent times that interest and technology permitted development of greater understanding of heat and heating, and substantial improvements in comfort were made. Comfort heating now is a highly developed science and, in conjunction with air conditioning, provides comfort conditions in all seasons in all parts of the world.

As more was learned about humidity and the capacity of the air to contain various amounts of water vapor, greater achievements in environmental control were made. Control of humidity in buildings now is a very important part of heating, ventilation, and air conditioning, and in many cases is extremely important in meeting manufacturing requirements. Today, it is possible to alter the atmosphere or environment in buildings in any manner, to suit any particular need, with great precision and control.

13.2.1 Thermometers and Scales

Energy in the form of heat is transferred from one material or substance to another because of a temperature difference that exists between them. When heat is applied to a material or substance, there will be an increase in average velocity of its molecules or electrons, with an increase in their kinetic energy. Likewise, as heat is removed, there will be a decrease in the average molecular velocity and, therefore, also the electron or molecular kinetic energy.

A thermometer is used to measure the degree of heat in a substance or material. The thermometer includes an appropriate graduated scale to indicate the change in temperature of the substance. The change in temperature as read on a thermometer is a measure of heat transferred to or from the substance. A unit of temperature is called a degree and is equivalent to one graduation on the scale.

By convention, the scale is an interval scale. The Celsius thermometer is a metric system of measuring temperature; 0°C is assigned to the temperature at which water freezes and 100°C to the temperature at which water boils at normal atmospheric conditions. Hence, on a Celsius thermometer, there are 100 intervals or graduations, called degrees, between the freezing and boiling temperatures. Each interval or degree is called 1 Celsius degree.

In the Fahrenheit system, 32°F is used to designate the freezing temperature of water and 212°F the boiling temperature at normal atmospheric pressure. Hence, on the Fahrenheit scale, a degree is equal to $\frac{1}{180}$ of the distance on the scale between the freezing and boiling temperatures. Conversion formulas used for each scale are as follows:

$$°F = 1.8 \times °C + 32 \tag{13.1}$$

$$°C = \frac{5}{9}(°F - 32) \tag{13.2}$$

13.2.2 Thermal Capacity and Specific Heat

The thermal capacity of a substance is indicated by the quantity of heat required to raise the temperature of 1 lb of the substance 1°F. In HVAC calculations, thermal capacity is usually expressed by the British thermal unit (Btu).

One Btu is the amount of heat that is required to increase the temperature of 1 lb of water 1°F at or near 39.2°F, which is the temperature at which water has its maximum density. Conversely, if 1 Btu is removed from 1 lb of water, its temperature will be reduced by 1°F.

Various quantities of heat will produce changes of 1°F per pound of substances other than water. Thus, thermal capacity is entirely dependent on the specific heat of the substances.

The specific heat of a substance is the ratio of the heat content or thermal capacity of a substance to that of water. And by definition, the specific heat of water is unity.

It is customary in HVAC calculations to use specific heat in lieu of thermal capacity, because of the convenience of using the Btu as a unit of heat quantity without conversions. Specific heats of air and some common building materials are shown in Table 13.1. Data for other substances may be obtained from tables in the "ASHRAE Handbook—Fundamentals," American Society of Heating, Refrigerating and Air-Conditioning Engineers. An examination of Table 13.1 indicates that the specific heat of these materials is less than unity and that, of all common substances, water possesses the largest specific heat and the largest thermal capacity.

TABLE 13.1 Specific Heats—Common Materials

Substance	Specific heat, Btu/(lb)(°F)
Air at 80°F	0.24
Water vapor	0.49
Water	1.00
Aluminum	0.23
Brick	0.20
Brass	0.09
Bronze	0.10
Gypsum	0.26
Ice	0.48
Limestone	0.22
Marble	0.21
Sand	0.19
Steel	0.12
Wood	0.45–0.65

13.2.3 Sensible Heat

When heat energy is added to or taken away from a substance, the resulting changes in temperature can be detected by the sense of touch, or sensibly. Therefore, this type of heat is called sensible heat. Since sensible heat is associated with a change in temperature, the quantity of sensible heat energy transferred in a heat exchange is usually calculated from

$$Q = Mc_p(t_2 - t_1) \tag{13.3}$$

where Q = sensible heat, Btu, absorbed or removed
M = mass, lb, of the substance undergoing the temperature change
c_p = specific heat of the substance
$(t_2 - t_1)$ = temperature difference of the substance, where t_2 is the final temperature after the heat exchange and t_1 is the temperature of the material before the heat exchange

13.2.4 Laws of Thermodynamics

The application of the laws of thermodynamics to HVAC calculations is usually limited to two well-known laws. These laws can be expressed differently, but in equivalent ways. A simplification of these laws as follows will permit an easier understanding.

The first law of thermodynamics states that when work performed produces heat, the quantity of the heat produced is proportional to the work performed. And conversely, when heat energy performs work, the quantity of the heat dissipated is proportional to the work performed. Work, ft-lb, is equal to the product of the force, lb, acting on the body for a distance, ft, that the body moves in the direction of the applied force.

Hence, this first law of thermodynamics can be expressed mathematically by the following equation:

$$W = JQ \tag{13.4}$$

where W = work, ft-lb
J = Joule's constant = mechanical equivalent of heat
Q = heat, Btu, generated by the work

Experiments have shown that the mechanical equivalent of heat, known as Joule's constant, is equivalent to 778 ft-lb/Btu. The first law is also known as the law of conservation of energy.

The second law of thermodynamics states that it is impossible for any machine to transfer heat from a substance to another substance at a higher temperature (if the machine is unaided by an external agency). This law can be interpreted to imply that the available supply of energy for doing work in our universe is constantly decreasing. It also implies that any effort to devise a machine to convert a specific quantity of heat into an equivalent amount of work is futile.

Entropy is the ratio of the heat added to a substance to the absolute temperature at which the heat is added.

$$S = \frac{dQ}{T_a} \tag{13.5}$$

where S = entropy
dQ = differential of heat (very small change)
T_a = absolute temperature

The second law of thermodynamics can be expressed mathematically with the use of the entropy concept.

Suppose an engine, which will convert heat into useful mechanical work, receives heat Q_1 from a heat source at temperature T_1 and delivers heat Q_2 at a temperature T_2 to a heat sink after performing work. By the first law of thermodynamics, the law of conservation of energy, Q_2 is less than Q_1 by the amount of work performed. And by the second law of thermodynamics, T_2 is less than T_1. The universe at the start of the process loses entropy $\Delta S_1 = Q_1/T_1$ and at the end of the process gains entropy $\Delta S_2 = Q_2/T_2$. Hence, the net change in the entropy of the universe because of this process will be $\Delta S_2 - \Delta S_1$.

Furthermore, this law requires that this net change must always be greater than zero and that the entropy increase is and must always be an irreversible thermodynamic process.

$$\Delta S_2 - \Delta S_1 > 0 \tag{13.6}$$

Because of the irreversibility of the process, the energy that has become available for performing work is

$$Q_u = T_2(\Delta S_2 - \Delta S_1) \tag{13.7}$$

13.2.5 Absolute Temperature

The definition of entropy given above involves the concept of absolute temperature measured on a ratio scale. The unit of absolute temperature is measured in degrees

Kelvin (°K) in the Celsius system and in degrees Rankine (°R) in the Fahrenheit system. Absolute zero or zero degrees in either system is determined by considering the theoretical behavior of an ideal gas, and for such a gas,

$$P_aV = mRT_a \qquad \text{or} \qquad P_av = RT_a \tag{13.8}$$

where P_a = absolute pressure on the gas, psf
V = volume of the gas, ft^3
v = specific volume of the gas, ft^3/lb = reciprocal of the gas density
m = mass of the gas, lb
T_a = absolute temperature
R = universal gas constant

For a gas under constant pressure, the absolute temperature theoretically will be zero when the volume is zero and all molecular motion ceases. Under these conditions, the absolute zero temperature has been determined to be nearly −273°C and −460°F. Therefore,

$$\text{Kelvin temperature °K} = \text{Celsius temperature} + 273° \tag{13.9}$$

$$\text{Rankine temperature °R} = \text{Fahrenheit temperature} + 460° \tag{13.10}$$

In the Rankine system, the universal gas constant R equals 1545.3 divided by the molecular weight of the gas. For air, $R = 53.4$, and for water vapor, $R = 85.8$.

13.2.6 Latent Heat

The sensible heat of a substance is associated with a sensible change in temperature. In contrast, the latent heat of a substance is always involved with a change in state of a substance, such as from ice to water and from water to steam or water vapor. Latent heat is very important in HVAC calculations and design, because the total heat content of air almost always contains some water in the form of vapor. The concept of latent heat may be clarified by consideration of the changes of state of water.

When heat is added to ice, the temperature rises until the ice reaches its melting point. Then, the ice continues to absorb heat *without a change in temperature* until a required amount of heat is absorbed per pound of ice, at which point it begins melting to form liquid water. The reverse is also true: if the liquid is cooled to the freezing point, this same quantity of heat must be removed to cause the liquid water to change to the solid (ice) state. This heat is called the **latent heat of fusion** for water. It is equal to 144 Btu and will convert 1 lb of ice at 32°F to 1 lb of water at 32°F. Thus,

$$\text{Latent heat of fusion for water} = 144 \text{ Btu/lb} \tag{13.11}$$

If the pound of water is heated further, say to 212°F, then an additional 180 Btu of heat must be added to effect the 180°F sensible change in temperature. At this temperature, any further addition of heat will not increase the temperature of the water beyond 212°F. With the continued application of heat, the water experiences violent agitation, called boiling. The boiling temperature of water is 212°F at atmospheric pressure.

With continued heating, the boiling water absorbs 970 Btu for each pound of water *without a change in temperature* and completely changes its state from liquid at 212°F to water vapor, or steam, at 212°F. Therefore, at 212°F,

$$\text{Latent heat of vaporization of water} = 970 \text{ Btu/lb} \tag{13.12}$$

Conversely, when steam at 212°F is cooled or condensed to a liquid at 212°F, 970 Btu per pound of steam (water) must be removed. This heat removal and change of state is called **condensation.**

When a body of water is permitted to evaporate into the air at normal atmospheric pressure, 29.92 in of mercury, a small portion of the body of water evaporates from the water surface at temperatures below the boiling point. The latent heat of vaporization is supplied by the body of water and the air, and hence both become cooler. The amount of vapor formed and that absorbed by the air above the water surface depends on the capacity of the air to retain water at the existing temperature and the amount of water vapor already in the air.

Table 13.2 lists the latent heat of vaporization of water for various air temperatures and normal atmospheric pressure. More extensive tables of thermodynamic properties of air, water, and steam are given in the "ASHRAE Handbook—Fundamentals," American Society of Heating, Refrigerating and Air-Conditioning Engineers.

TABLE 13.2 Thermal Properties—Dry and Saturated Air at Atmospheric Pressure

Air temperature, °F	Specific volume, ft³/lb		Pounds of water in saturated air per pound of dry air (humidity ratio)	Latent heat of vaporization of water, Btu/lb	Specific enthalpy of dry air h_a,* Btu/lb	Specific enthalpy of saturated air h_s,† Btu/lb	Specific enthalpy of saturation vapor h_g, Btu/lb
	Dry air	Saturated air					
0	11.58	11.59	0.0008		0	0.84	
32	12.39	12.46	0.0038	1075.2	7.69	11.76	1075.2
35	12.46	12.55	0.0043	1073.5	8.41	13.01	1076.5
40	12.59	12.70	0.0052	1070.6	9.61	15.23	1078.7
45	12.72	12.85	0.0063	1067.8	10.81	17.65	1080.9
50	12.84	13.00	0.0077	1065.0	12.01	20.30	1083.1
55	12.97	13.16	0.0092	1062.2	13.21	23.22	1085.2
60	13.10	13.33	0.0111	1059.3	14.41	26.46	1087.4
65	13.22	13.50	0.0133	1056.5	15.61	30.06	1089.6
70	13.35	13.69	0.0158	1053.7	16.82	34.09	1091.8
75	13.47	13.88	0.0188	1050.9	18.02	38.61	1094.0
80	13.60	14.09	0.0223	1048.1	19.22	43.69	1096.1
85	13.73	14.31	0.0264	1045.2	20.42	49.43	1098.3
90	13.85	14.55	0.0312	1042.4	21.62	55.93	1100.4
95	13.98	14.80	0.0367	1039.6	22.83	63.32	1102.6
100	14.11	15.08	0.0432	1036.7	24.03	71.73	1104.7
150	15.37	20.58	0.2125	1007.8	36.1	275.3	1125.8
200	16.63	77.14	2.295	977.7	48.1	2677	1145.8
212				970.2			1150.4

* Enthalpy of dry air is taken as zero for dry air at 0°F.
† Enthalpy of water vapor in saturated air = $h_s - h_a$, including sensible heat above 32°F.

13.2.7 Enthalpy

Enthalpy is a measure of the total heat (sensible and latent) in a substance and is equivalent to the sum of its internal energy plus its ability or capacity to perform work, or PV/J, where P is the pressure of the substance, V its volume, and J its mechanical equivalent of heat. Specific enthalpy is the heat per unit of weight, Btu/lb, and is the property used on psychrometric charts and in HVAC calculations.

The specific enthalpy of dry air h_a is taken as zero at 0°F. At higher temperatures, h_a is equal to the product of the specific heat, about 0.24, multiplied by the temperature, °F. (See Table 13.2.)

The specific enthalpy of saturated air h_s, which includes the latent heat of vaporization of the water vapor, is indicated in Table 13.2. The specific enthalpy of the water vapor or moisture at the air temperature may also be obtained from Table 13.2 by subtracting h_a from h_s.

Table 13.2 also lists the humidity ratio of the air at saturation for various temperatures (weight, lb, of water vapor in saturated air per pound of dry air). In addition, the specific enthalpy of saturated water vapor h_g, Btu/lb, is given in Table 13.2 and represents the sum of the latent heat of vaporization and the specific enthalpy of water at various temperatures.

The specific enthalpy of unsaturated air is equal to the sensible heat of dry air at the existing temperature, with the sensible heat at 0°F taken as zero, plus the product of the humidity ratio of the unsaturated air and h_g for the existing temperature.

13.2.8 Cooling by Evaporation

Evaporation of water requires a supply of heat. If there is no external source of heat, and evaporation occurs, then the water itself must provide the necessary heat of vaporization. In other words, a portion of the sensible heat in the liquid will be converted into the latent heat of vaporization. As a result, the temperature of the liquid remaining will drop. Since no external heat is added or removed by this process of evaporation, it is called **adiabatic cooling.**

Human beings are also cooled adiabatically by evaporation of perspiration from skin surfaces. Similarly, in hot climates with relatively dry air, air conditioning is provided by the vaporation of water into air. And refrigeration is also accomplished by the evaporation of a refrigerant.

13.2.9 Heating by Condensation

Many thermal processes occur without addition or subtraction of heat from the process. Under these conditions, the process is called **adiabatic.**

When a volume of moist air is cooled, a point will be reached at which further cooling cannot occur without reaching a fully saturated condition, that is, 100% saturation or 100% relative humidity. With continued cooling, some of the moisture condenses and appears as a liquid. The temperature at which condensation occurs is called the **dew point temperature.** If no heat is removed by the condensation, then the latent heat of vaporization of the water vapor will be converted to sensible heat in the air, with a resultant rise in temperature.

Thus, an increase in temperature is often accomplished by the formation of fog, and when rain or snow begins to fall, there will usually be an increase in temperature of the air.

13.2.10 Psychrometry

The measurement and determination of atmospheric conditions, particularly relating to the water vapor or moisture content in dry air, is an important branch of physics known as psychrometry. (Some psychrometric terms and conditions have already been presented in this article. Many others remain to be considered.)

An ideal gas follows certain established laws of physics. The mixture of water vapor and dry air behaves at normal atmospheric temperatures and pressures almost as an ideal gas. As an example, air temperatures, volumes, and pressures may be calculated by use of Eq. (13.8), $Pv = RT$.

Dalton's law also applies. It states:

When two or more gases occupy a common space or container, each gas will fill the volume just as if the other gas or gases were not present. Dalton's law also requires:

1. That each gas in a mixture occupy the same volume or space and also be at the same temperature as each other gas in the mixture.
2. That the total weight of the gases in the mixture equal the sum of the individual weights of the gases.
3. That the pressure of a mixture of several gases equal the sum of the pressures that each gas would exert if it existed alone in the volume enclosing the mixture.
4. That the total enthalpy of the mixture of gases equals the sum of the enthalpies of each gas.

An excellent example of the application of Dalton's law of partial pressures is the use of a liquid barometer to indicate atmospheric pressure. The barometer level indicates the sum of the partial pressure of water vapor and the partial pressure of the air.

Partial pressures of air and water vapor are of great importance in psychrometry and are used to calculate the degree of saturation of the air or relative humidity at a specific dry-bulb temperature.

13.2.11 Relative Humidity and Specific Humidity

Relative humidity is sometimes defined by the use of mole fractions, a difficult definition for psychrometric use. Hence, a more usable definition is desired. For this purpose, relative humidity may be closely determined by the ratio of the partial pressure of the water vapor in the air to the saturation pressure of water vapor at the same temperature, usually expressed as a percentage.

Thus, dry air is indicated as 0% relative humidity and fully saturated air is termed 100% relative humidity.

Computation of relative humidity by use of humidity ratios is also often done, but with somewhat less accuracy. Humidity ratio, or specific humidity W_a, at a specific temperature is the weight, lb, of water vapor in air per pound of dry air. If W_s represents the humidity ratio of saturated air at the same temperature (Table 13.2), then relative humidity can be calculated approximately from the equation

$$RH = \frac{W_a}{W_s} \times 100 \tag{13.13}$$

Dalton's law of partial pressures and Eq. (13.6) may also be used to calculate humidity ratios:

$$W_a = 0.622 \frac{p_w}{P - p_w} \tag{13.14}$$

where P = barometric pressure, atmospheric, psi
p_w = partial pressure of water vapor, psi

It is difficult to use this equation, however, because of the difficulty in measuring the partial pressures with special scientific equipment that is required and rarely available outside of research laboratories. Therefore, it is common practice to utilize simpler types of equipment in the field. These will provide direct readings that can be converted into humidity ratios or relative humidity.

A simple and commonly used device is the wet- and dry-bulb thermometer. This device is a packaged assembly consisting of both thermometers and a sock with scales. It is called a **sling psychrometer.** Both thermometers are identical, except that the wet-bulb thermometer is fitted with a wick-type sock over the bulb. The sock is wet with water, and the device is rapidly spun or rotated in the air. As the water in the sock evaporates, a drop in temperature occurs in the remaining water in the sock, and also in the wet-bulb thermometer. When there is no further temperature reduction and the temperature remains constant, the reading is called the wet-bulb temperature. The other thermometer will simultaneously read the dry-bulb temperature.

A difference between the two thermometer readings always exists when the air is less than saturated, at or less than 100% relative humidity. Inspection of a psychrometric chart will indicate that the wet-bulb and dry-bulb temperatures are identical only at fully saturated conditions, that is, at 100% relative humidity. Commercial psychrometers usually include appropriate charts or tables that indicate the relative humidity for a wide range of specific wet- and dry-bulb temperature readings. These tables are also found in books on psychrometry and HVAC books and publications.

13.2.12 Dew-Point Temperatures

Dew is the condensation of water vapor. It is most easily recognized by the presence of droplets in warm weather on grass, trees, automobiles, and many other outdoor surfaces in the early morning. Dew is formed during the night as the air temperature drops, and the air reaches a temperature at which it is saturated with moisture. This is the dew-point temperature. It is also equal to both the wet-bulb temperature and dry-bulb temperature. At the dew-point temperature, the air is fully saturated, that is, at 100% relative humidity. With any further cooling or drop in temperature, condensation begins and continues with any further reduction in temperature. The amount of moisture condensed is the excess moisture that the air cannot hold at saturation at the lowered temperature. The condensation forms drops of water, frequently referred to as *dew*.

Dew-point temperature, thus, is the temperature at which condensation of water vapor begins for any specific condition of humidity and pressure as the air temperature is reduced.

The dew-point temperature can be calculated, when the relative humidity is known, by use of Eq. (13.13) and Table 13.2. For the temperature of the unsaturated air, the humidity ratio at saturation is determined from Table 13.2. The product of the humidity ratio and the relative humidity equals the humidity ratio for the dew-point temperature, which also can be determined from Table 13.2. As an example, to determine the dew-point temperature of air at 90°F and 50% relative humidity, reference to Table 13.2 indicates a humidity ratio at saturation of 0.0312 at 90°F. Multiplication by 0.50 yields a humidity ratio of 0.0156. By interpolation in Table 13.2 between humidity ratios at saturation temperatures of 65 and 70°F, the dew-point temperature is found to be 69.6°F.

A simpler way to determine the dew-point temperature and many other properties of air-vapor mixtures is to use a psychrometric chart. This chart graphically relates dry-bulb, wet-bulb, and dew-point temperatures to relative humidity, humidity ratio, and specific volume of air. Psychrometric charts are often provided in books on psychrometrics and HVAC handbooks.

13.2.13 Refrigeration Ton

A ton of refrigeration is a common term used in air conditioning to designate the cooling rate of air-conditioning equipment. A ton of refrigeration indicates the ability of an evaporator to remove 200 Btu/min or 12,000 Btu/hr. The concept is a carry-over from the days of icemaking and was based on the concept that 200 Btu/min had to be removed from 32°F water to produce 1 ton of ice at 32°F in 24 hr. Hence,

$$
\begin{aligned}
\text{1 ton refrigeration} &= \frac{200\,\dfrac{\text{lb}}{\text{day}} \times 144\,\dfrac{\text{Btu}}{\text{lb}}}{24\,\dfrac{\text{hr}}{\text{day}}} \\
&= 288{,}000 \text{ Btu/day} \qquad (13.15) \\
&= 12{,}000 \text{ Btu/hr} \\
&= 200 \text{ Btu/min}
\end{aligned}
$$

("ASHRAE Handbook—Fundamentals," American Society of Heating, Refrigerating and Air-Conditioning Engineers, 1791 Tully Circle, N. E., Atlanta, GA 30329.)

13.3 MAJOR FACTORS IN HVAC DESIGN

This article presents the necessary concepts for management of heat energy and aims at development of a better understanding of its effects on human comfort. The concepts must be well understood if they are to be applied successfully to modification of the environment in building interiors, computer facilities, and manufacturing processes.

13.3.1 Significance of Design Criteria

Achievement of the desired performance of any HVAC system, whether designed for human comfort or industrial production or industrial process requirements, is significantly related to the development of appropriate and accurate design criteria.

Some of the more common items that are generally considered are as follows:

1. Outside design temperatures:
   Winter and summer
   Dry bulb (DB), wet bulb (WB)
2. Inside design temperatures:
   Winter: heating °F DB and relative humidity
   Summer: cooling °F DB and relative humidity
3. Filtration efficiency of supply air
4. Ventilation requirements
5. Exhaust requirements
6. Humidification
7. Dehumidification
8. Air-change rates
9. Positive-pressure areas
10. Negative-pressure areas
11. Balanced-pressure areas
12. Contaminated exhausts
13. Chemical exhausts and fume hoods
14. Energy conservation devices
15. Economizer system
16. Enthalpy control system
17. Infiltration
18. Exfiltration
19. Controls

13.3.2 Design Criteria Accuracy

Some engineers apply much effort to determination of design conditions with great accuracy. This is usually not necessary, because of the great number of variables involved in the design process. Strict design criteria will increase the cost of the necessary machinery for such optimum conditions and may be unnecessary. It is generally recognized that it is impossible to provide a specific indoor condition that will satisfy every occupant at all times. Hence, HVAC engineers tend to be practical in their designs and accept the fact that the occupants will adapt to minor variations from ideal conditions. Engineers also know that human comfort depends on the type and quantity of clothing worn by the occupants, the types of activities performed, environmental conditions, duration of occupancy, ventilation air, and closeness of and number of people within the conditioned space and recognize that these conditions are usually unpredictable.

13.3.3 Outline of Design Procedure

Design of an HVAC system is not a simple task. The procedure varies considerably from one application or project to another, and important considerations for one project may have little impact on another. But for all projects, to some extent, the following major steps have to be taken:

1. Determine all applicable design conditions, such as inside and outside temperature and humidity conditions for winter and summer conditions, including prevailing winds and speeds.
2. Determine all particular and peculiar interior space conditions that will be maintained.
3. Estimate, for every space, heating or cooling loads from adjacent unheated or uncooled spaces.
4. Carefully check architectural drawings for all building materials used for walls, roofs, floors, ceilings, doors, etc., and determine the necessary thermal coefficients for each.
5. Establish values for air infiltration and exfiltration quantities, for use in determining heat losses and heat gains.
6. Determine ventilation quantities and corresponding loads for heat losses and heat gains.
7. Determine heat or cooling loads due to internal machinery, equipment, lights, motors, etc.
8. Include allowance for effects of solar load.
9. Total the heat losses requiring heating of spaces and heat gains requiring cooling of spaces, to determine equipment capacities.
10. Determine system type and control method to be applied.

13.3.4 Temperatures Determined by Heat Balances

In cold weather, comfortable indoor temperatures may have to be maintained by a heating device. It should provide heat to the space at the same rate as the space is losing heat. Similarly, when cooling is required, heat should be removed from the space at the same rate that it is gaining heat. In each case, there must be a heat-balance between heat in and heat out when heating and the reverse in cooling. Comfortable inside conditions can only be maintained if this heat balance can be controlled or maintained.

The rate at which heat is gained or lost is a function of the difference between the inside air temperature to be maintained and the outside air temperature. Such temperatures must be established for design purposes in order to properly size and select HVAC equipment that will maintain the desired design conditions. Many other conditions that also affect the flow of heat in and out of buildings, however, should also be considered in selection of equipment.

13.3.5 Methods of Heat Transfer

Heat always flows from a hot to a cold object, in strict compliance with the second law of thermodynamics (Art. 13.2). This direction of heat flow occurs by conduction, convection, or radiation and in any combination of these forms.

Thermal conduction is a process in which heat energy is transferred through matter by the transmission of kinetic energy from molecule to molecule or atom to atom.

Thermal convection is a means of transferring heat in air by natural or forced movements of air or a gas. Natural convection is usually a rotary or circular motion caused by warm air rising and cooler air falling. Convection can be mechanically produced (forced convection), usually by use of a fan or blower.

Thermal radiation transfers energy in wave form from a hot body to a relatively cold body. The transfer occurs independently of any material between the two bodies. Radiation energy is converted energy from one source to a very long wave form of electromagnetic energy. Interception of this long wave by solid matter will convert the radiant energy back to heat.

13.3.6 Thermal Conduction and Conductivity

Thermal conduction is the rate of heat flow across a unit area (usually 1 ft^2) from one surface to the opposite surface for a unit temperature difference between the two surfaces and under steady-state conditions. Thus, the heat-flow rate through a plate with unit thickness may be calculated from

$$Q = kA(t_2 - t_1) \tag{13.16}$$

where Q = heat flow rate, Btu/hr
k = coefficient of thermal conductivity for a unit thickness of material, usually 1 in
A = surface area, normal to heat flow, ft^2
t_2 = temperature, °F, on the warm side of the plate
t_1 = temperature, °F, on the cooler side of the plate

The coefficient k depends on the characteristics of the plate. The numerical value of k also depends on the units used for the other variables in Eq. (13.16). When values of k are taken from published tables, units given should be adjusted to agree with the units of the other variables.

In practice, the thickness of building materials often differs from unit thickness. Consequently, use of a coefficient of conductivity for the entire thickness is advantageous. This coefficient, called **thermal conductance,** is derived by dividing the conductivity k by the thickness L, the thickness being the length or path of heat flow.

***Thermal Conductance* C *and Resistanced* R.** Thermal conductance C is the same as conductivity, except that it is based on a specific thickness, instead of 1 in as for conductivity. Conductance is usually used for assemblies of different materials, such as cast-in-place concrete and concrete block with an airspace between. The flow of heat through such an assembly is very complex and is determined under ideal test conditions. In such tests, conductance is taken as the average heat flow from a unit area of surface (usually 1 ft^2) for the total thickness of the assembly. In the case of 9-in-thick concrete, for example, the conductance, as taken from appropriate tables, would be 0.90 Btu/(hr)(ft^2)(°F). (It should be understood, however, that conversion of the conductance C to conductivity k by dividing C by the thickness will produce significant errors.)

Conductance C is calculated from

$$Q = CA\,(t_2 - t_1) \tag{13.17}$$

where $t_2 - t_1$ is the temperature difference causing the heat flow Q, and A is the cross-sectional area normal to the heat flow.

Values of k and C for many building materials are given in tables in "ASHRAE Handbook—Fundamentals" and other publications on air conditioning.

Thermal resistance, the resistance to flow of heat through a material or an assembly of materials, equals the reciprocal of the conductance:

$$R = \frac{1}{C} \tag{13.18}$$

Thermal resistance R is used in HVAC calculations for determining the rate of heat flow per unit area through a nonhomogeneous material or a group of materials.

Air Films. In addition to its dependence on the thermal conductivity or conductance of a given wall section, roof, or other enclosure, the flow of heat is also dependent on the surface air films on each side of the constructions. These air films are very thin and cling to the exposed surface on each side of the enclosures. Each of the air films possesses thermal conductance, which should always be considered in HVAC calculations.

The indoor air film is denoted by f_i and the outdoor film by f_o. Values are given in Table 13.3 for these air films and for interior or enclosed air spaces of assemblies. In this table, the effects of air films along both enclosure surfaces have been taken into account in developing the air-film coefficients. Additional data may be obtained from the "ASHRAE Handbook—Fundamentals."

Air-to-Air Heat Transfer. In the study of heat flow through an assembly of building materials, it is always assumed that the rate of heat flow is constant and continues without change. In other words, a steady-state condition exists. For such a condition, the rate of heat flow in Btu per hour per unit area can be calculated from

$$Q = UA(t_2 - t_1) \tag{13.19}$$

where U = coefficient of thermal transmittance.

TABLE 13.3 Thermal Conductance of Air, Btu/(hr)(ft^2)(°F)

f_i	for indoor air film (still air)	
	Vertical surface, horizontal heat flow	1.5
	Horizontal surface	
	Upward heat flow	1.6
	Downward heat flow	1.1
f_o	for outdoor air film, 15-mi/hr wind (winter)	6.0
f_o	for outdoor air film, 7.5-mi/hr wind (summer)	4.0
C	for vertical air gap, ¾ in or more wide	1.1
C	for horizontal air gap, ¾ in or more wide	
	Upward heat flow	1.2
	Downward heat flow	1.0

***Coefficient of Thermal Transmittance* U.** The coefficient of thermal transmittance U, also known as the overall coefficient of heat transfer, is the rate of heat flow under steady-state conditions from a unit area from the air on one side to the air on the other side of a material or an assembly when a steady temperature difference exists between the air on both sides.

In calculation of the heat flow through a series of different materials, their individual resistances should be determined and totaled to obtain the total resistance R_t. The coefficient of thermal transmittance is then given by the reciprocal of the total resistance:

$$U = \frac{1}{R_t} \tag{13.20}$$

Tables of U values for various constructions are available in the "ASHRAE Handbook—Fundamentals," catalogs of insulation manufacturers, and other publications.

***Computation of* R *and* U.** An assembly that is constructed with several different building materials with different thermal resistances $R_1, R_2, R_3, \ldots, R_n$ provides a total thermal resistance

$$R_t = \frac{1}{f_i} + R_1 + R_2 + R_3 + \cdots + R_n + \frac{1}{f_o} \tag{13.21}$$

including the indoor and outdoor air film resistances f_i and f_o. The U value, coefficient of thermal transmittance, is then determined by use of Eq. (13.20). This coefficient may be substituted in Eq. (13.19) for calculation of the steady-state heat flow through the assembly.

As a typical example, consider an exterior wall section that is constructed of 4-in face brick, 4-in cinder block, ¾-in airspace, and lightweight ¾-in lath and plaster. The wall is 8 ft 6 in high and 12 ft long. The inside air temperature is to be maintained at 68°F, with an outdoor air temperature of +10°F and a 15-mi/hr prevailing wind. What will be the total heat loss through this wall?

From Table 13.3, the indoor air film conductance is 1.5. Its resistance is equal to $1/1.5 = 0.67$. The outdoor air-film conductance for a 15-mi/hr wind is 6.0. Its resistance is equal to $1/6.0 = 0.17$. Conductivity of the 4-in face brick is 5.0. Conductance of the 4-in cinder block is 0.90; of the ¾-in airspace, 1.1, and of the ¾-in lath and lightweight plaster, 7.70. The total resistance of the wall is then:

$$\begin{aligned} R_t &= \frac{1}{6.0} + 4 \times \frac{1}{5.0} + \frac{1}{0.90} + \frac{1}{1.1} + \frac{1}{7.70} + \frac{1}{1.5} \\ &= 0.17 + 0.8 + 1.11 + 0.91 + 0.13 + 0.67 \\ &= 3.79 \end{aligned}$$

and the coefficient of thermal transmittance is

$$U = \frac{1}{3.79} = 0.264$$

The heat flow rate through the entire wall will be, from Eq. (13.19),

$$Q = 0.264 \times (8.5 \times 12.0)(68 - 10) = 1562 \text{ Btu/hr}$$

13.3.7 Thermal Insulation

A substantial reduction in heating and cooling loads can be made by the judicious use of thermal insulation in wall and roof construction. Addition of insulation results in an increase in thermal resistance *R*, or a reduction in the coefficient of heat transfer *U* of the walls and roof.

Any material with high resistance to flow of heat is called insulation. Many kinds of insulation materials are used in building construction. See Art. 12.3.

Note that the maximum overall conductance *U* encountered in building construction is 1.5 Btu/(hr)(ft^2)(°F). This would occur with a sheet-metal wall. The metal has, for practical purposes, no resistance to heat flow. The *U* value of 1.5 is due entirely to the resistance of the inside and outside air films. Most types of construction have *U* factors considerably less than 1.5.

The minimum *U* factor generally found in standard construction with 2 in of insulation is about 0.10.

Since the *U* factor for single glass is 1.13, it can be seen that windows are a large source of heat gain, or heat loss, compared with the rest of the structure. For double glass, the *U* factor is 0.45. For further comparison, the conductivity *k* of most commercial insulations varies from about 0.24 to about 0.34.

13.3.8 Convection

Heating by natural convection is very common, because air very easily transfers heat in this manner. As air becomes warmer, it becomes less dense and rises. As it leaves the proximity of the heating surface, other cooler air moves in to replace the rising volume of heated air. As the warm air rises, it comes in contact with cooler materials, such as walls, glass, and ceilings. It becomes cooler and heavier and, under the influence of gravity, begins to fall. Hence, a circulatory motion of air is established, and heat transfer occurs.

When a heating device called a convector operates in a cool space, heat from the convector is transmitted to the cooler walls and ceiling by convection. The convection process will continue as long as the walls or ceiling are colder and the temperature difference is maintained.

Heating of building interiors is usually accomplished with convectors with hot water or steam as the heating medium. The heating element usually consists of a steel or copper pipe with closely spaced steel or aluminum fins. The convector is mounted at floor level against an exterior wall. The fins are used to greatly increase the area of the heating surface. As cool room air near the floor comes in contact with the hot surfaces of the convector, the air quickly becomes very warm and rises rapidly along the cold wall surface above the convector. Additional cold air at floor level then moves into the convector to replace the heated air. In this manner, the entire room will become heated. This process is called heating by natural convection.

13.3.9 Radiation

The most common form of heat transfer is by radiation. All materials and substances radiate energy and absorb radiation energy.

The sun is a huge radiator and the earth is heated by this immense source of radiated energy, which is often called solar energy (sunshine). Solar-collector devices are used to collect this energy and transfer it indoors to heat the interior of a building.

When radiation from the sun is intercepted by walls, roofs, or glass windows, this heat is transmitted through them and heats the interior of the building and its occupants. The reverse is also true; that is, when the walls are cold, the people in the space radiate their body heat to the cold wall and glass surfaces. If the rate of radiation is high, the occupants will be uncomfortable.

Not all materials radiate or absorb radiation equally. Black- or dark-body materials radiate and absorb energy better than light-colored or shiny materials. Materials with smooth surfaces and light colors are poor absorbers of radiant energy and also poor radiators.

Much of the radiation that strikes the surface of window glass is transmitted to the interior of the building as short-wave radiation. This radiation will strike other objects in the interior and radiate some of this energy back to the exterior, except through glass, as a longer wavelength of radiation energy. The glass does not efficiently transmit the longer wavelengths to the outside. Instead, it acts as a check valve, limiting solar radiation to one-way flow. This one-way flow is desirable in winter for heating. In summer, however, it is not desirable, because the longer-wavelength energy eventually becomes an additional load on the air-conditioning system.

The rate of radiation from an object may be determined by use of the **Stefan-Boltzmann law** of radiation. This law states that the amount of energy radiated from a perfect radiator, or a blackbody, is proportional to the fourth power of the absolute temperature of the body. Because most materials are not perfect radiators or absorbers, a proportionality constant called the hemispherical emittance factor is used with this law. Methods for calculating and estimating radiation transfer rates can be found in the "ASHRAE Handbook—Fundamentals."

The quantity of energy transferred by radiation depends on the individual temperatures of the radiating bodies. These temperatures are usually combined into a **mean radiant temperature** for use in heating and cooling calculations. The mean radiant temperature is the uniform temperature of a block enclosure with which a solid body (or occupant) would exchange the same amount of radiant heat as in the actual nonuniform environment.

13.3.10 Thermal Criteria for Building Interiors

There are three very important conditions to be controlled in buildings for human comfort. These important criteria are dry-bulb temperature, relative humidity, and velocity or rate of air movement in the space.

Measurements of these conditions should be made where average conditions exist in the building, room, or zone and at the breathing line, 3 to 5 ft above the floor. The measurements should be taken where they would not be affected by unusually high heat sources or heat losses. Minor variations or limits from the design conditions, however, are usually acceptable.

The occupied zone of a conditioned space does not encompass the total room volume. Rather, this occupied zone is generally taken as that volume bounded by levels 3 in above the floor and 6 ft above the floor and by vertical planes 2 ft from walls.

Indoor design temperatures are calculated from test data compiled for men and women with various amounts of clothing and for various degrees of physical exertion. For lightly clothed people doing light, active work in relatively still room air, the design dry-bulb temperature can be determined from

$$t = 180 - 1.4t_r \tag{13.22}$$

where t = dry-bulb temperature, °F DB
t_r = mean radiant temperature of the space or room (between 70 and 80°F)

When temperatures of walls, materials, equipment, furniture, etc., in a room are all equal, $t = t_r = 75$°F. With low outside temperature, the building exterior becomes cold, in which case the room temperature should be maintained above 75°F to provide the necessary heat that is being lost to the cold exterior. In accordance with Eq. (13.22), the design dry-bulb temperature should be increased 1.4°F for each 1°F of mean radiant temperature below 75°F in the room. In very warm weather, the design temperature should be decreased correspondingly.

Humidity is often controlled for human comfort. Except in rare cases, relative humidity (RH) usually should not exceed 60%, because the moisture in the air may destroy wood finishes and support mildew. Below 20% RH, the air is so dry that human nostrils become dry and wood furniture often cracks from drying out.

In summer, a relative humidity of 45 to 55% is generally acceptable. In winter, a range of 30 to 35% RH is more desirable, to prevent condensation on windows and in walls and roofs. When design temperatures in the range of 75°F are maintained in a space, the comfort of occupants who are inactive is not noticeably affected by the relative humidity.

Variations from the design criteria are generally permitted for operational facilities. These variations are usually established as a number of degrees above or below the design point, such as 75°F DB ± 2°F. For relative humidity, the permitted variation is usually given as a percent, for example, 55% RH ± 5%.

Design conditions vary widely for many commercial and industrial uses. Indoor design criteria for various requirements are given in the ''Applications'' volume of the ASHRAE Handbook.

13.3.11 Outdoor Design Conditions

The outdoor design conditions at a proposed building site are very important in design of heating and cooling systems. Of major importance are the dry-bulb temperature, humidity conditions, and prevailing winds.

Outside conditions assumed for design purposes affect the heating and cooling plant's physical size, capacity, electrical requirements, and of considerable importance, the estimated cost of the HVAC installation. The reason for this is that in many cases, the differences between indoor and outdoor conditions have a great influence on calculated heating and cooling loads, which determine the required heating and cooling equipment capacities. Since in most cases the design outdoor air temperatures are assumed, the size of equipment will be greatly affected by assumed values.

Extreme outside air conditions are rarely used to determine the size of heating and cooling equipment, since these extreme conditions may occur, in summer or winter, only once in 10 to 50 years. If these extreme conditions were used for equipment selection, the results would be greatly oversized heating and cooling plants and a much greater installed cost than necessary. Furthermore, such oversized equipment will operate most of the year at part load and with frequent cycling of the machines. This results in inefficient operation and, generally, consumption of additional power, because most machines operate at maximum efficiency at full load.

On the other hand, when heating or cooling equipment is properly sized for more frequently occurring outdoor conditions, the plants will operate with less cycling and greater efficiency. During the few hours per year when outside conditions exceed those used for design, the equipment will run continuously in an attempt to maintain the intended interior design conditions. If such conditions persist for a long time, there will probably be a change in interior conditions from design conditions that may or may not be of a minor extent and that may produce uncomfortable conditions for the occupants.

Equipment should be selected with a total capacity that includes a safety factor to cover other types of operation than under steady-state conditions. In the midwest, for instance, the outdoor air temperature may fall as much as 45°F in 2 hr. The heating capacity of a boiler in this case would have to be substantially larger than that required for the calculated heat loss alone. As another example, many heating and cooling systems are controlled automatically by temperature control systems that, at a predetermined time, automatically reset the building temperature downward to maintain, say, 60°F at night for heating. At a predetermined time, for example, 7:30 a.m., before arrival of occupants, the control system instructs the boiler to bring the building up to its design temperature for occupancy. Under these conditions, the boiler must have the additional capacity to comply in a reasonable period of time before the arrival of the occupants.

TABLE 13.4 Recommended Design Outdoor Summer Temperatures

State	City	Dry-bulb temp, °F	Wet-bulb temp, °F	State	City	Dry-bulb temp, °F	Wet-bulb temp, °F
Ala.	Birmingham	95	78	Miss.	Vicksburg	95	78
Ariz.	Flagstaff	90	65	Mo.	St. Louis	95	78
Ariz.	Phoenix	105	75	Mont.	Helena	95	67
Ark.	Little Rock	95	78	Nebr.	Lincoln	95	78
Calif.	Los Angeles	90	70	Nev.	Reno	95	65
Calif.	San Francisco	85	65	N.H.	Concord	90	73
Colo.	Denver	95	65	N.J.	Trenton	95	78
Conn.	Hartford	95	75	N. Mex.	Albuquerque	95	70
D.C.	Washington	95	78	N.Y.	New York	95	75
Fla.	Jacksonville	95	78	N.C.	Greensboro	95	78
Fla.	Miami	95	79	N. Dak.	Bismarck	95	73
Ga.	Atlanta	95	76	Ohio	Cincinnati	95	75
Idaho	Boise	95	65	Okla.	Tulsa	100	77
Ill.	Chicago	95	75	Ore.	Portland	90	68
Ind.	Indianapolis	95	75	Pa.	Philadelphia	95	78
Iowa	Des Moines	95	78	R.I.	Providence	95	75
Kans.	Topeka	100	78	S.C.	Charleston	95	78
Ky.	Louisville	95	78	S. Dak.	Rapid City	95	70
La.	New Orleans	95	80	Tenn.	Nashville	95	78
Maine	Portland	90	73	Tex.	Dallas	100	78
Md.	Baltimore	95	78	Tex.	Houston	95	78
Mass.	Boston	95	75	Utah	Salt Lake City	95	65
Mich.	Detroit	95	75	Vt.	Burlington	90	73
Minn.	Minneapolis	95	75	Va.	Richmond	95	78

TABLE 13.5 Recommended Design Outdoor Winter Temperatures

State	City	Temp, °F	State	City	Temp, °F
Ala.	Birmingham	10	Miss.	Vicksburg	10
Ariz.	Flagstaff	−10	Mo.	St. Louis	0
Ariz.	Phoenix	25	Mont.	Helena	−20
Ark.	Little Rock	5	Nebr.	Lincoln	−10
Calif.	Los Angeles	35	Nev.	Reno	−5
Calif.	San Francisco	35	N.H.	Concord	−15
Colo.	Denver	−10	N.J.	Trenton	0
Conn.	Hartford	0	N. Mex.	Albuquerque	0
D.C.	Washington	0	N.Y.	New York	0
Fla.	Jacksonville	25	N.C.	Greensboro	10
Fla.	Miami	35	N. Dak.	Bismarck	−30
Ga.	Atlanta	10	Ohio	Cincinnati	0
Idaho	Boise	−10	Okla.	Tulsa	0
Ill.	Chicago	−10	Ore.	Portland	10
Ind.	Indianapolis	−10	Pa.	Philadelphia	0
Iowa	Des Moines	−15	R.I.	Providence	0
Kans.	Topeka	−10	S.C.	Charleston	15
Ky.	Louisville	0	S. Dak.	Rapid City	−20
La.	New Orleans	20	Tenn.	Nashville	0
Maine	Portland	−5	Tex.	Dallas	0
Md.	Baltimore	0	Tex.	Houston	20
Mass.	Boston	0	Utah	Salt Lake City	−10
Mich.	Detroit	−10	Vt.	Burlington	−10
Minn.	Minneapolis	−20	Va.	Richmond	15

Accordingly, design outdoor conditions should be selected in accordance with the manner in which the building will be used and, just as important, to obtain reasonable initial cost and low operational costs. Outdoor design conditions for a few cities are shown in Tables 13.4 and 13.5. Much more detailed data are presented in the ''ASHRAE Handbook—Fundamentals.''

The use of outdoor design conditions does not yield accurate estimates of fuel requirements or operating costs, because of the considerable variations of outdoor air temperature seasonally, monthly, daily, and even hourly. These wide variations must be taken into account in attempts to forecast the operating costs of a heating or cooling system.

Since most equipment capacities are selected for calculated loads based on steady-state conditions, usually these conditions will not provide acceptable estimates of annual operating costs. (Wide fluctuations in outside temperatures, however, may not always cause a rapid change in inside conditions as outdoor temperatures rise and fall. For example, in buildings with massive walls and roofs and small windows, indoor temperatures respond slowly to outdoor changes.) Hence, forecasts of fuel requirements or operating costs should be based on the average temperature difference between inside and outside air temperatures on an hourly basis for the entire year. Such calculations are extremely laborious and are almost always performed by a computer that utilizes an appropriate program and local weather tapes for the city involved. Many such programs are currently available from various sources.

13.4 VENTILATION

Ventilation is utilized for many different purposes, the most common being control of humidity and condensation. Other well-known uses include exhaust hoods in restaurants, heat removal in industrial plants, fresh air in buildings, odor removal, and chemical and fume hood exhausts. In commercial buildings, ventilation air is used for replacement of stale, vitiated air, odor control, and smoke removal. Ventilation air contributes greatly to the comfort of the building's occupants. It is considered to be of such importance that many building codes contain specific requirements for minimum quantities of fresh, or outside, air that must be supplied to occupied areas.

Ventilation is also the prime method for reducing employee exposure to excessive airborne contaminants that result from industrial operations. Ventilation is used to dilute contaminants to safe levels or to capture them at their point of origin before they pollute the employees' working environment. The Occupational Safety and Health Act (OSHA) standards set the legal limits for employee exposures to many types of toxic substances.

13.4.1 Methods of Ventilation

Ventilation is generally accomplished by two methods: natural and mechanical. In either case, ventilation air must be air taken from the outdoors. It is brought into the building through screened and louvered or other types of openings, with or without ductwork. In many mechanical ventilation systems, the outside air is brought in through ductwork to an appropriate air-moving device, such as a centrifugal fan. With a network of ductwork, the supply air is distributed to areas where it is needed. Also, mechanical ventilation systems are usually designed to exhaust air from the building with exhaust fans or gravity-type ventilators in the roof or a combination-type system.

Many mechanical ventilation systems are installed for fire protection in buildings to remove smoke, heat, and fire. The design must be capable of satisfying the provisions of the National Fire Protection Association "Standard for Installation of Air-Conditioning systems," NFPA 90-A. The standard also covers installation provisions of air intakes and outlets.

Natural ventilation in buildings is caused by the temperature difference between the air in the building and the outside air and by openings in the outside walls or by a combination of both. With natural ventilation, there should be some means for removing the ventilation air from the building, such as roof-mounted gravity vents or exhaust fans.

13.4.2 Minimum Ventilation Requirements

There are many codes and rules governing minimum standards of ventilation. All gravity or natural-ventilation requirements involving window areas in a room as a given percentage of the floor area or volume are at best approximations. The amount of air movement or replacement by gravity depends on prevailing winds, temperature difference between interior and exterior, height of structure, window-crack area, etc. For controlled ventilation, a mechanical method of air change is recommended.

Where people are working, the amount of ventilation air required will vary from one air change per hour where no heat or offensive odors are generated to about 60 air changes per hour.

At best, a ventilation system is a dilution process, by which the rate of odor or heat removal is equal to that generated in the premises. Occupied areas below grade or in windowless structures require mechanical ventilation to give occupants a feeling of outdoor freshness. Without outside air, a stale or musty odor may result. The amount of fresh air to be brought in depends on the number of persons occupying the premises, type of activity volume of the premises, and amount of heat, moisture, and odor generation. ASHRAE Standard 62 gives the recommended minimum amount of ventilation air required for various activities and ranges from 5 to 50 cfm per person.

The amount of air to be handled, obtained from the estimate of the per person method, should be checked against the volume of the premises and the number of air changes per hour given in Eq. (13.23).

$$\text{Number of air changes per hour} = \frac{60Q}{V} \tag{13.23}$$

where Q = air supplied, ft^3/min
V = volume of ventilated space, ft^3

When the number of changes per hour is too low (below one air change per hour), the ventilation system will take too long to create a noticeable effect when first put into operation. Five changes per hour are generally considered a practical minimum. Air changes above 60 per hour usually will create some discomfort because of air velocities that are too high.

Toilet ventilation and locker-room ventilation are usually covered by local codes—50 ft^3/min per water closet and urinal is the usual minimum for toilets and six changes per hour minimum for both toilets and locker rooms.

13.4.3 Heat, Odor, and Moisture Removal

Removal of concentrated heat, odor, or objectionable vapors by ventilation is best carried out by locating the exhaust outlets as close as possible to the heat source.

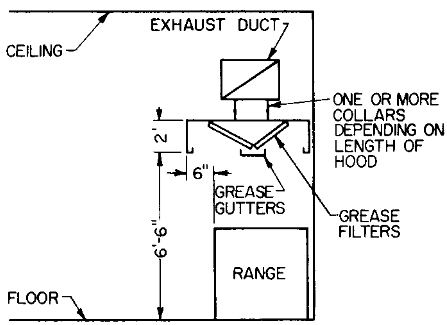

FIGURE 13.1 Canopy hood for exhausting heat from a kitchen range.

When concentrated sources of heat are present, canopy hoods will remove the heat more efficiently.

Figure 13.1 shows a canopy-hood installation over a kitchen range. Grease filters reduce the frequency of required cleaning. When no grease is vaporized, they may be eliminated.

Greasy ducts are serious fire hazards and should be cleaned periodically. There are on the market a number of automatic fire-control systems for greasy ducts. These systems usually consist of fusible-link fire dampers and a means of flame smothering—CO_2, steam, foam, etc.

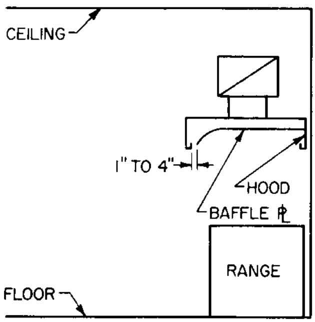

FIGURE 13.2 Double hood for exhausting heat.

Figure 13.2 shows a double hood. This type collects heat more efficiently; i.e., less exhaust air is required to collect a given amount of heat. The crack area is arranged to yield a velocity of about 1000 ft/min.

A curtain of high-velocity air around the periphery of the hood catches the hot air issuing from the range or heat source. Canopy hoods are designed to handle about 50 to 125 ft^3/min of exhaust air per square foot of hood. The total amount of ventilation air should not yield more than 60 changes per hour in the space.

Where hoods are not practical to install and heat will be discharged into the room, the amount of ventilation air may be determined by the following method:

Determine the total amount of sensible heat generated in the premises—lights, people, electrical equipment, etc. This heat will cause a temperature rise and an increase in heat loss through walls, windows, etc. To maintain desired temperature conditions, ventilation air will have to be used to remove heat not lost by transmission through enclosures.

$$q_v = 1.08Q(T_i - T_o) \tag{13.24}$$

where q_v = heat, Btu/hr, carried away by ventilation air
Q = flow of ventilation air, ft^3/min
T_i = indoor temperature to be maintained
T_o = outdoor temperature

With Eq. (13.24), we can calculate the amount of ventilation air required by assuming a difference between room and outdoor temperatures or we can calculate this temperature gradient for a given amount of ventilation air.

The same method may be used to calculate the air quantity required to remove any objectionable chemical generated. For example, assume that after study of a process we determine that a chemical will be evolved in vapor or gas form at the rate of X lb/min. If Y is the allowable concentration in pounds per cubic foot, then $Q = X/Y$, where Q is the ventilation air needed in cubic feet per minute.

Where moisture is the objectionable vapor, the same equation holds, but with X as the pounds per minute of moisture vaporized, Y the allowable concentration of moisture in pounds per cubic foot above outdoor moisture concentration.

Once, the amount of ventilation air is determined, a duct system may be designed to handle it, if necessary.

Ventilation air may be provided by installing either an exhaust system, a supply system, or both.

In occupied areas where no unusual amounts of heat or odors are generated, such as offices and shipping rooms, a supply-air system may be provided, with grilles or ceiling outlets located for good distribution. When the building is tight, a relief system of grilles or ducts to the outside should be provided. But when the relief system is too extensive, an exhaust fan should be installed for a combination supply and exhaust system.

All air exhausted from a space must be replaced by outside air either by infiltration through doors and windows or by a fresh-air makeup system. Makeup air systems that have to operate during the winter season are often equipped with heating coils to temper the cold outside air.

13.4.4 Natural Ventilation

Natural ventilation in buildings is accomplished by use of windows, louvers, skylights, roof ventilators, roof monitors, jalousies, intake hoods, etc. They should be located to admit fresh air only and not near sources of smoke, dust, odors, or polluted air from adjacent sources. Discharge vents should also be provided to eliminate vitiated air from the building. The outlet locations must not discharge toward other fresh-air intakes of the building or its neighbors. In multifloor buildings, vertical vent shafts, or risers, are used to supply ventilation air throughout the building.

13.4.5 Mechanical Ventilation

Mechanical ventilation is almost always preferred over natural ventilation because of reliability and the ability to maintain specific design requirements, such as air changes per hour and face velocities for exhaust hoods. Natural ventilation permits wide variations in ventilation-air quantities and uncertain durations of ventilation. (In critical areas, such as in carcinogenic research laboratories, natural ventilation is never relied upon.) For this reason, mechanical ventilation systems are almost always used where ventilation requirements are critical and must be highly reliable.

Mechanical ventilation is often required by various building codes for various applications as follows:

1. Control of contaminants in the work area for health protection and compliance with OSHA standards for achieving the legal limits set on employee exposure to specific toxic and hazardous substances
2. Fire and explosion prevention for flammable vapors
3. Environmental protection
4. Reuse of valuable industrial materials
5. Human comfort—removal of heat, odors, and tobacco smoke
6. Humidity control
7. Corrosive fumes and noxious gases

Mechanical ventilation may be a single system without heating, cooling, filtration, humidification, dehumidification, etc., or it may include various combinations of these functions. In other words, the systems can be heating-ventilating units or heating–ventilating–air-conditioning (HVAC) units.

In many complex and specialized buildings, certain functional areas will be required to have various degrees of positive pressurization, negative pressurization, or balanced atmospheric conditions. The ventilation air as a part of the system supply air is used to provide the positive and balanced pressures. An exhaust system is utilized to maintain the negative-pressure areas. In many designs, air lost by pressurization is exfiltrated from the system and does not become part of the return-air stream.

Recirculation of ventilation air is prohibited from certain areas, such as toilets, bathrooms, biology labs, chemistry labs, hospital operating rooms, mortuary rooms, isolation rooms, and rooms with flammable vapors, odors, dust, and noxious gases.

In all ventilation systems, a quantity of air equal to the ventilation air should leave the building. If this is not accomplished, then the building will become pressurized, and the ventilation air will exfiltrate through available doors, windows, cracks, crevices, relief vents, etc. Since in many cases this is undesirable and unreliable, exhaust systems are usually employed. The exhaust, in many cases, may be part of a complete HVAC system.

13.5 MOVEMENT OF AIR WITH FANS

Inasmuch as most ventilation systems are designed as mechanical ventilation systems that utilize various kinds of fans, a knowledge of the types of fans in use will be of value in selection of ventilation fans. Fans are used to create a pressure differential that causes air to flow in a system. They generally incorporate one of several types of impellers mounted in an appropriate housing or enclosure. An electric motor usually drives the impeller to move the air.

Two types of fans are commonly used in air-handling and air-moving systems: axial and centrifugal. They differ in the direction of airflow through the impeller.

Centrifugal fans are enclosed in a scroll-shaped housing, which is designed for efficient airstream energy transfer. This type of fan has the most versatility and low first cost and is the workhorse of the industry. Impeller blades may be radial, forward-curved, backward-inclined, or airfoil. When large volumes of air are moved, airfoil or backward-inclined blades are preferable because of higher efficiencies. For smaller volumes of air, forward, curved blades are used with satisfactory results. Centrifugal fans are manufactured with capacities of up to 500,000 ft^3/min and can operate against pressures up to 30 in water gage.

Axial-flow fans are versatile and sometimes less costly than centrifugal fans. The use of axial fans is steadily increasing, because of the availability of controllable-pitch units, with increased emphasis on energy savings. Substantial energy savings can be realized by varying the blade pitch to meet specific duty loads. Axial fans develop static pressure by changing the velocity of the air through the impeller and converting it into static pressure. Axial fans are quite noisy and are generally used by industry where the noise level can be tolerated. When used for HVAC installations, sound attenuators are almost always used in series with the fan for noise abatement. Tubeaxial and vaneaxial are modifications of the axial-flow fan.

Propeller-type fans are also axial fans and are produced in many sizes and shapes. Small units are used for small jobs, such as kitchen exhausts, toilet exhausts, and air-cooled condensers. Larger units are used by industry for ventilation and heat removal in large industrial buildings. Such units have capacities of up to 200,000 ft^3/min of air. Propeller-type fans are limited to operating pressures of about ½ in water gage maximum, and are usually much noisier than centrifugal fans of equal capacities.

Vaneaxial fans are available with capacities up to 175,000 ft^3/min and can operate at pressures up to 12 in water gage. Tubeaxial fans can operate against pressures of only 1 in water gage with only slightly lower capacities.

In addition to the axial and centrifugal fan classifications, a third class for special designs exists. This classification covers tubular centrifugal fans and axial-centrifugal, power roof ventilators. The tubular centrifugal type is often used as a return-air fan in low-pressure HVAC systems. Air is discharged from the impeller in the same way as in standard centrifugal fans and then changed 90° in direction through straightening vanes. Tubular centrifugal fans are manufactured with capacities of more than 250,000 ft^3/min of air and may operate at pressures up to 12 in water gage.

Power roof ventilators are usually roof mounted and utilize either centrifugal or axial blade fans. Both types are generally used in low-pressure exhaust systems for factories, warehouses, etc. They are available in capacities up to about 30,000 ft^3/min. They are, however, limited to operation at a maximum pressure of about ½ in water gage. Powered roof ventilators are also low in first cost and low in operating costs. They can provide positive exhaust ventilation in a space, which is a definite advantage over gravity-type exhaust units. The centrifugal unit is somewhat quieter than the axial-flow type.

Fans vary widely in shapes and sizes, motor arrangements and space requirements. Fan performance characteristics (variation of static pressure and brake horsepower) with changes in the airflow rate (ft^3/min) are available from fan manufacturers and are presented in tabular form or as fan curves.

Dampers. Dampers are mechanical devices that are installed in a moving airstream in a duct to reduce the flow of the stream. They, in effect, purposely produce a pressure drop (when installed) in a duct by substantially reducing the free area of the duct.

Two types of dampers are commonly used by HVAC designers, parallel blade and opposed blade. In both types, the blades are linked together so that a rotation force applied to one shaft simultaneously rotates all blades. The rotation of the blades opens or closes the duct's free area from 0 to 100% and determines the flow rate.

Dampers are used often as opening and closing devices. For this purpose, parallel dampers are preferred.

When dampers are installed in ducts and are adjusted in a certain position to produce a desired flow rate downstream, opposed-blade dampers are preferred. When dampers are used for this purpose, the operation is called **balancing.**

Once the system is balanced and the airflows in all branch ducts are design airflows, the damper positions are not changed until some future change in the system occurs. However, in automatic temperature-control systems, both opening-and-closing and balancing dampers are commonly used. In complex systems, dampers may be modulated to compensate for increased pressure drop by filter loading and to maintain constant supply-air quantity in the system.

Filters. All air-handling units should be provided with filter boxes. Removal of dust from the conditioned air not only lowers building maintenance costs and creates a healthier atmosphere but prevents the cooling and heating coils from becoming blocked up.

Air filters come in a number of standard sizes and thicknesses. The filter area should be such that the air velocity across the filters does not exceed 350 ft/min for low-velocity filters or 550 ft/min for high-velocity filters. Thus, the minimum filter area in square feet to be provided equals the airflow, ft^3/min, divided by the maximum air velocity across the filters, ft/min.

Most air filters are of either the throwaway or cleanable type. Both these types will fit a standard filter rack.

Electrostatic filters are usually employed in industrial installations, where a higher percentage of dust removal must be obtained. Check with manufacturers' ratings for particle-size removal, capacity, and static-pressure loss; also check electric service required. These units generally are used in combination with regular throwaway or cleanable air filters, which take out the large particles, while the charged electrostatic plates remove the smaller ones. See also Art. 13.6.

13.6 DUCT DESIGN

After air discharge grilles and the air handler, which consists of a heat exchanger and blower, have been located, it is advisable to make a single-line drawing showing the duct layout and the air quantities each branch and line must be able to carry.

Of the methods of duct design in use, the equal-friction method is the most practical. It is considered good practice not to exceed a pressure loss of 0.15 in of water per 100 ft of ductwork by friction. Higher friction will result in large power consumption for air circulation. It is also considered good practice to stay below a starting velocity in main ducts of 900 ft/min in residences; 1300 ft/min in schools, theaters, and public buildings; and 1800 ft/min in industrial buildings. Velocity in branch ducts should be about two-thirds of these and in branch risers about one-half.

Too high a velocity will result in noisy and panting ductwork. Too low a velocity will require uneconomical, bulky ducts.

TABLE 13.6 Diameters of Circular Ducts in Inches Equivalent to Rectangular Ducts

Side	4	8	12	18	24	30	36	42	48	60	72	84
3	3.8	5.2	6.2									
4	4.4	6.1	7.3									
5	4.9	6.9	8.3									
6	5.4	7.6	9.2									
7	5.7	8.2	9.9									
12		10.7	13.1									
18		12.9	16.0	19.7								
24		14.6	18.3	22.6	26.2							
30		16.1	20.2	25.2	29.3	32.8						
36		17.4	21.9	27.4	32.0	35.8	39.4					
42		18.5	23.4	29.4	34.4	38.6	42.4	45.9				
48		19.6	24.8	31.2	36.6	41.2	45.2	48.9	52.6			
60		21.4	27.3	34.5	40.4	45.8	50.4	54.6	58.5	65.7		
72		23.1	29.5	37.2	43.8	49.7	54.9	59.6	63.9	71.7	78.8	
84				39.9	46.9	53.2	58.9	64.1	68.8	77.2	84.8	91.9
96					49.5	56.3	62.4	68.2	73.2	82.6	90.5	97.9

TABLE 13.7 Size of Round Ducts for Airflow*

Friction, in H_2O per 100 ft	0.05		0.10		0.15		0.20		0.25		0.30	
Airflow, cfm	Diam, in	Velocity, ft/min	Diam, in	Velocity, ft/min	Diam, in	Velocity, ft/min	Diam, in	Velocity, ft/min	Diam, in	Velocity, ft/min	Diam, in	Velocity, ft/min
50	5.3	350	4.6	450	4.2	530	3.9	600	3.8	660	3.7	710
100	6.8	420	5.8	550	5.4	640	5.1	720	4.8	780	4.7	850
200	8.7	480	7.6	650	6.9	760	6.6	860	6.3	940	6.1	1020
300	10.2	540	8.8	730	8.2	850	7.7	960	7.3	1050	7.1	1120
400	11.5	580	9.8	770	9.0	920	8.5	1040	8.2	1130	7.8	1200
500	12.4	620	11.8	820	9.8	970	9.3	1080	8.8	1160	8.6	1270
1,000	15.8	730	13.7	970	12.8	1140	12.0	1280	11.5	1400	11.2	1500
2,000	20.8	870	18.0	1150	16.6	1370	15.7	1520	15.0	1660	14.5	1780
3,000	24.0	960	21.0	1280	19.7	1500	18.3	1680	17.5	1850		
4,000	26.8	1050	23.4	1360	21.6	1600	20.2	1800				
5,000	29.2	1100	25.5	1460	23.7	1700	22.2	1900				
10,000	37.8	1310	33.2	1770	30.3	2000						

*Based on data in "ASHRAE Handbook—Fundamentals," American Society of Heating, Refrigerating and Air-Conditioning Engineers.

The shape of ducts usually installed as rectangular, because dimensions can easily be changed to maintain the required area. However, as ducts are flattened, the increase in perimeter offers additional resistance to airflow. Thus a flat duct requires an increase in cross section to be equivalent in air-carrying capacity to one more nearly square.

A 12 × 12-in duct, for example, will have an area 1 ft^2 and a perimeter of 4 ft, whereas a 24 × 6-in duct will have the same cross-sectional area but a 5-ft perimeter and thus greater friction. Therefore, a 24 × 7-in duct is more nearly equivalent to the 12 × 12. Equivalent sizes can be determined from tables, such as those in the ''ASHRAE Handbook—Fundamentals,'' where rectangular ducts are rated in terms of equivalent round ducts (equal friction and capacity). Table 13.6 is a shortened version.

Charts also are available in the ASHRAE handbook giving the relationship between duct diameter in inches, air velocity in feet per minute, air quantity in cubic feet per minute, and friction in inches of water pressure drop per 100 ft of duct. Table 13.7 is based on data in the ASHRAE handbook.

In the equal-friction method, the equivalent round duct is determined for the required air flow at the predetermined friction factor. For an example illustrating the method of calculating duct sizes, see Art. 13.11.

(H. E. Bovay, Jr., ''Handbook of Mechanical and Electrical Systems for Buildings,'' F. E. Beaty, Jr., ''Sourcebook of HVAC Details,'' and ''Sourcebook of HVAC Specifications,'' N. R. Grim and R. C. Rosaler, ''Handbook of HVAC Design,'' D. L. Grumman, ''Air-Handling Systems Ready Reference Manual,'' McGraw-Hill Publishing Company, New York; B. Stein et al., ''Mechanical and Electrical Equipment for Buildings,'' 7th ed., John Wiley & Sons, Inc., New York.)

13.7 HEAT LOSSES

Methods and principles for calculation of heat losses are presented in Art 13.3. These methods provide a rational procedure for determination of the size and capacity of a heating plant.

Heat loads for buildings consist of heat losses and gains. Heat losses include those from air infiltration, ventilation air, and conduction through the building exterior caused by low temperatures of outside air. Heat gains include those due to people, hot outside air, solar radiation, electrical lighting and motor loads, and heat from miscellaneous interior equipment. These loads are used to determine the proper equipment size for the lowest initial cost and for operation with maximum efficiency.

Walls and Roofs. Heat loss through the walls and roofs of a building constitutes most of the total heat loss in cold weather. These losses are calculated with Eq. (13.19), $Q = UA\ (t_2 - t_1)$, with the appropriate temperature differential between inside and outside design temperatures.

Architectural drawings should be carefully examined to establish the materials of construction that will be used in the walls and roofs. With this information, the overall coefficient of heat transmittance, or U factor, can be determined as described in Art. 13.3. Also, from the drawings, the height and width of each wall section should be determined to establish the total area for each wall or roof section for use in Eq. (13.19).

Heat Loss through Basement Floors and Walls. Although heat-transmission coefficients through basement floors and walls are available, it is generally not practicable to use them because ground temperatures are difficult to determine because of the many variables involved. Instead, the rate of heat flow can be estimated, for all practical purposes, from Table 13.8. This table is based on groundwater temperatures, which range from about 40 to 60°F in the northern sections of the United States and 60 to 75°F in the southern sections. (For specific areas, see the ''ASHRAE Handbook—Fundamentals.'')

Heat Loss from Floors on Grade. Attempts have been made to simplify the variables that enter into determination of heat loss through floors set directly on the ground. The most practical method breaks it down to a heat flow in Btu per hour per linear foot of edge exposed to the outside. With 2 in of edge insulation, the rate of heat loss is about 50 in the cold northern sections of the United States, 45 in the temperate zones, 40 in the warm south. Corresponding rates for 1-in insulation are 60, 55, and 50. With no edge insulation the rates are 75, 65, and 60 Btu/(hr)(ft).

Heat Loss from Unheated Attics. Top stories with unheated attics above require special treatment. To determine the heat loss through the ceiling, we must calculate the equilibrium attic temperature under design inside and outside temperature conditions. This is done by equating the heat gain to the attic via the ceiling to the heat loss through the roof:

$$U_c A_c (T_i - T_a) = U_r A_r (T_a - T_o) \tag{13.25}$$

where U_c = heat-transmission coefficient for ceiling
U_r = heat-transmission coefficient for roof
A_c = ceiling area
A_r = roof area
T_i = design roof temperature
T_o = design outdoor temperature
T_a = attic temperature

Thus

$$T_a = \frac{U_c A_c T_i + U_r A_r T_o}{U_c A_c + U_r A_r} \tag{13.26}$$

The same procedure should be used to obtain the temperature of other unheated spaces, such as cellars and attached garages.

TABLE 13.8 Below-Grade Heat Losses

Ground water temp, °F	Basement floor loss,* Btu per hr per sq ft	Below-grade wall loss, Btu per hr per sq ft
40	3.0	6.0
50	2.0	4.0
60	1.0	2.0

*Based on basement temperature of 70°F.

Air Infiltration. When the heating load of a building is calculated, it is advisable to figure each room separately, to ascertain the amount of heat to be supplied to each room. Then, compute the load for a complete floor or building and check it against the sum of the loads for the individual rooms.

Once we compute the heat flow through all exposed surfaces of a room, we have the heat load if the room is perfectly airtight and the doors never opened. However, this generally is not the case. In fact, windows and doors, even if weather-stripped, will allow outside air to infiltrate and inside air to exfiltrate. The amount of cold air entering a room depends on crack area, wind velocity, and number of exposures, among other things.

Attempts at calculating window- and door-crack area to determine air leakage usually yield a poor estimate. Faster and more dependable is the air-change method, which is based on the assumption that cold outside air is heated and pumped into the premises to create a static pressure large enough to prevent cold air from infiltrating.

The amount of air required to create this static pressure will depend on the volume of the room.

If the number of air changes taking place per hour N are known, the infiltration Q in cubic feet per minute can be computed from

$$Q = \frac{VN}{60} \tag{13.27}$$

where V = volume of room, ft^3. The amount of heat q in Btu per hour required to warm up this cold air is given by

$$q = 1.08QT \tag{13.28}$$

where Q = ft^3/min of air to be warmed
T = temperature rise

13.8 HEAT GAINS

These differ from heat losses only by the direction of the heat flow. Thus, the methods discussed in Art. 13.7 for heat losses can also be used for determining heat gains. In both cases, the proper inside and outside design conditions and wet-bulb temperatures should be established as described in Art. 13.3.

Heat gains may occur at any time throughout the year. Examples are heat from electric lighting, motor and equipment loads, solar radiation, people, and ventilation requirements. When heat gains occur in cold weather, they should be deducted from the heat loss for the space.

Ventilation and infiltration air in warm weather produce large heat gains and should be added to other calculated heat gains to arrive at the total heat gains for cooling-equipment sizing purposes.

To determine the size of cooling plant required in a building or part of a building, we determine the heat transmitted to the conditioned space through the walls, glass, ceiling, floor, etc., and add all the heat generated in the space. This is the cooling load. The unwanted heat must be removed by supplying cool air. The total cooling load is divided into two parts—sensible and latent.

Sensible and Latent Heat. The part of the cooling load that shows up in the form of a dry-bulb temperature rise is called sensible heat. It includes heat transmitted through walls, windows, roof, floor, etc.; radiation from the sun; and heat from lights, people, electrical and gas appliances, and outside air brought into the air-conditioned space.

Cooling required to remove unwanted moisture from the air-conditioned space is called latent load, and the heat extracted is called latent heat. Usually, the moisture is condensed out on the cooling coils in the cooling unit.

For every pound of moisture condensed from the air, the air-conditioning equipment must remove about 1050 Btu. Instead of rating items that give off moisture in pounds or grains per hour, common practice rates them in Btu per hour of latent load. These items include gas appliances, which give off moisture in products of combustion; steam baths, food, beverages, etc., which evaporate moisture; people; and humid outside air brought into the air-conditioned space.

Design Temperatures for Cooling. Before we can calculate the cooling load, we must first determine a design outside condition and the conditions we want to maintain inside.

For comfort cooling, indoor air at 80°F dry-bulb and 50% relative humidity is usually acceptable.

Table 13.4, p. 13.26, gives recommended design outdoor summer temperatures for various cities. Note that these temperatures are not the highest ever attained; for example, in New York City, the highest dry-bulb temperature recorded exceeds 105°F, whereas the design outdoor dry-bulb temperature is 95°F. Similarly, the wet-bulb temperature is sometimes above the 75°F design wet-bulb for that area.

Heat Gain through Enclosures. To obtain the heat gain through walls, windows, ceilings, floors, etc., when it is warmer outside than in, the heat-transfer coefficient is multiplied by the surface area and the temperature gradient.

Radiation from the sun through glass is another source of heat. It can amount to about 200 Btu/(hr)(ft^2) for a single sheet of unshaded common window glass facing east and west, about three-fourths as much for windows facing northeast and northwest, and one-half as much for windows facing south. For most practical applications, however, the sun effect on walls can be neglected, since the time lag is considerable and the peak load is no longer present by the time the radiant heat starts to work through to the inside surface. Also, if the wall exposed to the sun contains windows, the peak radiation through the glass also will be gone by the time the radiant heat on the walls gets through.

Radiation from the sun through roofs may be considerable. For most roofs, total equivalent temperature differences for calculating solar heat gain through roofs is about 50°F.

Roof Sprays. Many buildings have been equipped with roof sprays to reduce the sun load on the roof. Usually the life of a roof is increased by the spray system, because it prevents swelling, blistering, and vaporization of the volatile components of the roofing material. It also prevents the thermal shock of thunderstorms during hot spells. Equivalent temperature differential for computing heat gain on sprayed roofs is about 18°F.

Water pools 2 to 6 in deep on roofs have been used, but they create structural difficulties. Furthermore, holdover heat into the late evening after the sun has set creates a breeding ground for mosquitoes and requires algae-growth control. Equivalent temperature differential to be used for computing heat gain for water-covered roofs is about 22°F.

Spray control is effected by the use of a water solenoid valve actuated by a temperature controller whose bulb is embedded in the roofing. Tests have been carried out with controller settings of 95, 100, and 105°F. The last was found to be the most practical setting.

The spray nozzles must not be too fine, or too much water is lost by drift. For ridge roofs, a pipe with holes or slots is satisfactory. When the ridge runs north and south, two pipes with two controllers would be practical, for the east pipe would be in operation in the morning and the west pipe in the afternoon.

Heat Gains from Interior Sources. Electric lights and most other electrical appliances convert their energy into heat.

$$q = 3.42W \tag{13.29}$$

where q = Btu/hr developed
W = watts of electricity used

For lighting, if W is taken as the total light wattage, it may be reduced by the ratio of the wattage expected to be consumed at any time to the total installed wattage. This ratio may be unity for commercial applications, such as stores. Where fluorescent lighting is used, add 25% of the lamp rating for the heat generated in the ballast. Where electricity is used to heat coffee, etc., some of the energy is used to vaporize water. Tables in the "ASHRAE Handbook—Fundamentals" give an estimate of the Btu per hour given up as sensible heat and that given up as latent heat by appliances.

Heat gain from people can be obtained from Table 13.9.

Heat Gain from Outside Air. The sensible heat from outside air brought into a conditioned space can be obtained from

$$q_s = 1.08Q(T_o - T_i) \tag{13.30}$$

where q_s = sensible load due to outside air, Btu/hr
Q = ft^3/min of outside air brought into conditioned space
T_o = design dry-bulb temperature of outside air
T_i = design dry-bulb temperature of conditioned space

The latent load due to outside air in Btu per hour is given by

$$q_l = 0.67Q(G_o - G_i) \tag{13.31}$$

where Q = ft^3/min of outside air brought into conditioned space
G_o = moisture content of outside air, grains per pound of air
G_i = moisture content of inside air, grains per pound of air

The moisture content of air at various conditions may be obtained from a psychrometric chart.

Miscellaneous Sources of Heat Gain. In an air-conditioning unit, the fan used to circulate the air requires a certain amount of brake horsepower depending on the air quantity and the total resistance in the ductwork, coils, filters, etc. This horse-

TABLE 13.9 Rates of Heat Gain from Occupants of Conditioned Spaces*

Degree of activity	Typical application	Total heat adults, male, Btu/hr	Total heat adjusted, Btu/hr†	Sensible heat, Btu/hr	Latent heat, Btu/hr
Seated at theater	Theater—matinee	390	330	225	105
Seated at theater	Theater—evening	390	350	245	105
Seated, very light work	Offices, hotels, apartments	450	400	245	155
Moderately active office work	Offices, hotels, apartments	475	450	250	200
Standing, light work; walking	Department store, retail store	550	450	250	200
Walking; standing	Drugstore, bank	550	500	250	250
Sedentary work	Restaurant‡	490	550	275	275
Light bench work	Factory	800	750	275	475
Moderate dancing	Dance hall	900	850	305	545
Walking 3 mph; light machine work	Factory	1000	1000	375	625
Bowling§	Bowling alley	1500	1450	580	870
Heavy work	Factory	1500	1450	580	870
Heavy machine work; lifting	Factory	1600	1600	635	965
Athletics	Gymnasium	2000	1800	710	1090

*Tabulated values are based on 78°F room dry-bulb temperature. For 80°F room dry bulb, the total heat remains the same, but the sensible-heat value should be decreased by approximately 20% and the latent heat values increased accordingly. All values are rounded to the nearest 5 Btu/hr.

†Adjusted total heat gain is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85% of that for an adult male, and that the gain from a child is 75% of that for an adult male.

‡Adjusted total heat value for eating in a restaurant includes 60 Btu/hr for food per individual (30 Btu sensible and 30 Btu latent).

§For bowling, figure one person per alley actually bowling, and all others as sitting (400 Btu/hr) or standing and walking slowly (550 Btu/hr).

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power will dissipate itself in the conditioned air and will show up as a temperature rise. Therefore, we must include the fan brake horsepower in the air-conditioning load. For most low-pressure air-distribution duct systems, the heat from this source varies from 5% of the sensible load for smaller systems to 3½% of the sensible load in the larger systems.

Where the air-conditioning ducts pass through nonconditioned spaces, the ducts must be insulated. The amount of the heat transmitted to the conditioned air through the insulation may be calculated from the duct area and the insulation heat-transfer coefficient.

METHODS OF HEATING BUILDINGS

The preceding articles present the basic knowledge necessary for accurate determination of the heat losses of a building. Such procedures make possible sizing and selection of heating equipment that will provide reliable and satisfactory service.

13.9 GENERAL PROCEDURE FOR SIZING A HEATING PLANT

The basic procedure used in sizing a heating plant is as follows: We isolate the part of the structure to be heated. To estimate the amount of heat to be supplied to that part, we must first decide on the design indoor and outdoor temperatures. For if we maintain a temperature of, say, 70°F inside the structure and the outside temperature is, say, 0°F, then heat will be conducted and radiated to the outside at a rate that can be computed from this 70° temperature difference. If we are to maintain the design inside temperature, we must add heat to the interior by some means at the same rate that it is lost to the exterior.

Recommended design inside temperatures are given in Table 13.10. Recommended design outdoor temperatures for a few cities are given in Table 13.5, p. 13.27. (More extensive data are given in the ‘‘ASHRAE Handbook—Fundamentals,’’ American Society of Heating, Refrigerating and Air-Conditioning Engineers.)

Note that the recommended design outdoor winter temperatures are not the lowest temperatures ever attained in each region. For example, the lowest temperature on record in New York City is −14°F, whereas the design temperature is 0°F. If the design indoor temperature is 70°F, we would be designing for $(70 - 0)/(70 + 14)$, or 83.3%, of the capacity we would need for the short period that a record cold of −14°F would last.

Once we have established for design purposes a temperature gradient (indoor design temperature minus outdoor design temperature) across the building exterior, we obtain the heat-transmission coefficients of the various building materials in the exterior construction for computation of the heat flow per square foot. These coefficients may be obtained from the manufacturers of the materials or from tables, such as those in the ‘‘ASHRAE Handbook—Fundamentals.’’ Next, we have to take off from the plans the areas of exposed walls, windows, roof, etc., to determine the total heat flow, which is obtained by adding the sum of the products of the area, temperature gradient, and heat-transmission coefficient for each item (see Art. 13.3).

TABLE 13.10 Recommended Design Indoor Winter Temperatures

Type of building	Temp, °F
Schools:	
Classrooms	72
Assembly rooms, dining rooms	72
Playrooms, gymnasiums	65
Swimming pool	75
Locker rooms	70
Hospitals:	
Private rooms	72
Operating rooms	75
Wards	70
Toilets	70
Bathrooms	75
Kitchens and laundries	66
Theaters	72
Hotels:	
Bedrooms	70
Ballrooms	68
Residences	72
Stores	68
Offices	72
Factories	65

Choosing Heating-Plant Capacity. Total heat load equals the heat loss through conduction, radiation, and infiltration.

If we provide a heating plant with a capacity equal to this calculated heat load, we shall be able to maintain design room temperature when the design outside temperature prevails, if the interior is already at design room temperature. However, in most buildings the temperature is allowed to drop to as low as 55°F during the night. Thus, theoretically, it will require an infinite time to approach design room temperature. It, therefore, is considered good practice to add 20% to the heating-plant capacity for morning pickup.

The final figure obtained is the minimum heating-plant size required. Consult manufacturers' ratings and pick a unit with a capacity no lower than that calculated by the above method.

On the other hand, it is not advisable to choose a unit too large, because then operating efficiency suffers, increasing fuel consumption.

With a plant of 20% greater capacity than required for calculated heat load, theoretically after the morning pickup, it will run only 100/120, or 83⅓%, of the time. Furthermore, since the design outdoor temperature occurs only during a small percentage of the heating season, during the rest of the heating season the plant would operate intermittently, less than 83⅓% of the time. Thus it is considered good practice to choose a heating unit no smaller than required but not much larger.

If the heating plant will be used to produce hot water for the premises, determine the added capacity required.

13.10 HEATING-LOAD-CALCULATION EXAMPLE

As an example of the method described for sizing a heating plant, let us take the building shown in Fig. 13.3.

A design outdoor temperature of 0°F and an indoor temperature of 70°F are assumed. The wall is to be constructed of 4-in brick with 8-in cinder-block backup. Interior finish is metal lath and plaster (wall heat-transmission coefficient $U = 0.25$).

The method of determining the heat load is shown in Table 13.11.

Losses from the cellar include 4 Btu/(hr)(ft^2) through the walls [column (4)] and 2 Btu/(hr)(ft^2) through the floors. Multiplied by the corresponding areas, they yield the total heat loss in column (6). In addition, some heat is lost because of infiltration of cold air. One-half an air change per hour is assumed, or 71.2 ft^3/min [column (3)]. This causes a heat loss, according to Eq. (13.28), of

$$1.08 \times 7.12 \times 70 = 5400 \text{ Btu/hr}$$

To the total for the cellar, 20% is added to obtain the heat load in column (7).

Similarly, heat losses are obtained for the various areas on the first and second floors. Heat-transmission coefficients [column (4)] were obtained from the "ASHRAE Handbook—Fundamentals," American Society of Heating, Refrigerating and Air-Conditioning Engineers. These were multiplied by the temperature gradient (70 − 0) to obtain the heat losses in column (6).

The total for the building, plus 20%, amounts to 144,475 Btu/hr. A heating plant with approximately this capacity should be selected.

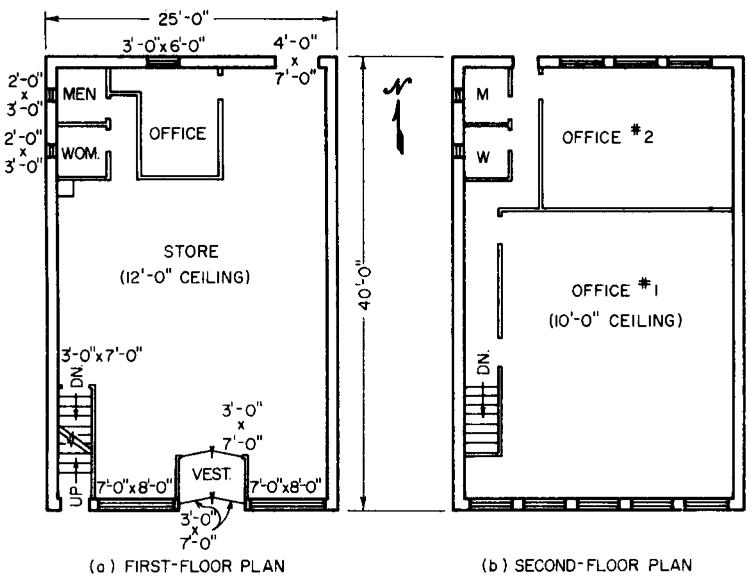

FIGURE 13.3 First-floor and second-floor plans of a two-story building.

TABLE 13.11 Heat-Load Determined for Two-Story Building

Space (1)	Heat-loss source (2)	Net area or cfm infiltration (3)	U or coefficient (4)	Temp gradient, °F (5)	Heat loss, Btu per hr (6)	Total plus 20% (7)
Cellar	Walls	1,170	4		4,680	
	Floor	1,000	2		2,000	
	½ air change	71.2	1.08	70	5,400	14,500
First-floor store	Walls	851	0.25	70	14,900	
	Glass	135	1.13	70	10,680	
	Doors	128	0.69	70	1,350	
	1 air change	170	1.08	70	12,850	47,600
Vestibule	Glass	48	1.13	70	3,800	
	2 air change	12	1.08	70	905	5,650
Office	Walls	99	0.25	70	1,730	
	Glass	21	1.13	70	1,660	
	½ air change	9	1.08	70	680	4,900
Men's room	Walls	118	0.25	70	2,060	
	Glass	6	1.13	70	48	
	½ air change	3	1.08	70	226	2,800
Ladies' room	Walls	60	0.25	70	1,050	
	Glass	6	1.13	70	48	
	½ air change	3	1.08	70	226	1,590
Second-floor office No. 1	Walls	366	0.25	70	6,400	
	Glass	120	1.13	70	9,500	
	Roof	606	0.19	70	8,050	
	½ air change	50	1.08	70	3,780	33,300
Office No. 2	Walls	207	0.25	70	3,620	
	Glass	72	1.13	70	5,700	
	Roof	234	0.19	70	3,120	
	½ air change	19	1.08	70	1,435	16,650
Men's room	Walls	79	0.25	70	1,380	
	Glass	6	1.13	70	475	
	Roof	20	0.19	70	266	
	½ air change	3	1.08	70	226	2,820
Ladies' room	Walls	44	0.25	70	770	
	Glass	6	1.13	70	475	
	Roof	20	0.19	70	266	
	½ air change	3	1.08	70	226	2,080
Hall	Walls	270	0.25	70	4,720	
	Door	21	0.69	70	1,015	
	Roof	75	0.19	70	1,000	
	2 air change	40	1.08	70	3,020	12,585
						144,475

13.11 WARM-AIR HEATING

A warm-air heating system supplies heat to a room by bringing in a quantity of air above room temperature, the amount of heat added by the air being at least equal to that required to counteract heat losses.

A gravity system (without a blower) is rarely installed because it depends on the difference in density of the warm-air supply and the colder room air for the working pressure. Airflow resistance must be kept at a minimum with large ducts and very few elbows. The result usually is an unsightly duct arrangement.

A forced-warm-air system can maintain higher velocities, thus requires smaller ducts, and provides much more sensitive control. For this type of system,

$$q = 1.08Q(T_h - T_i) \tag{13.32}$$

where T_h = temperature of air leaving grille
T_i = room temperature
q = heat added by air, Btu/hr
Q = ft^3/min of air supplied to room

Equation (13.32) indicates that the higher the temperature of the discharge air T_h, the less air need be handled. In cheaper installations, the discharge air may be as high as 170°F and ducts are small. In better systems, more air is handled with a discharge temperature as low as 135 to 140°F. With a room temperature of 70°F, we shall need $(170 - 70)/(135 - 70) = 1.54$ times as much air with the 135°F system as with a 170°F system.

It is not advisable to go much below 135°F with discharge air, because drafts will result. With body temperature at 98°F, air at about 100°F will hardly seem warm. If we stand a few feet away from the supply diffuser (register), 70°F room air will be entrained with the warm supply air and the mixture will be less than 98°F when it reaches us. We probably would complain about the draft.

Supply-air diffusers should be arranged so that they blow a curtain of warm air across the cold, or exposed, walls and windows. (See Fig. 13.4 for a suggested arrangement.) These grilles should be placed near the floor, since the lower-density warm air will rise and accumulate at the ceiling.

Return-air grilles should be arranged in the interior near unexposed walls—in foyers, closets, etc.—and preferably at the ceiling. This is done for two reasons:

1. The warm air in all heating systems tends to rise to the ceiling. This creates a large temperature gradient between floor and ceiling, sometimes as high as 10°F. Taking the return air from the ceiling reduces this gradient.
2. Returning the warmer air to the heating plant is more economical in operation than using cold air from the floor.

Duct sizes can be determined as described in Duct Design, Art. 13.6. To illustrate the procedure, duct sizes will be computed for the structure with floor plans shown in Figs. 13.3 and 13.4. The latter shows the locations chosen for the discharge grilles and indicates that the boiler room, which contains the heating unit, is in the basement.

Table 13.11 showed that a heating unit with 144,475 Btu/hr capacity is required. After checking manufacturers' ratings of forced-warm-air heaters, we choose a unit rated at 160,000 Btu/hr and 2010 ft^3/min.

If we utilized the full fan capacity and supply for the rated 160,000 Btu/hr and applied Eq. (13.32), the temperature rise through the heater would be

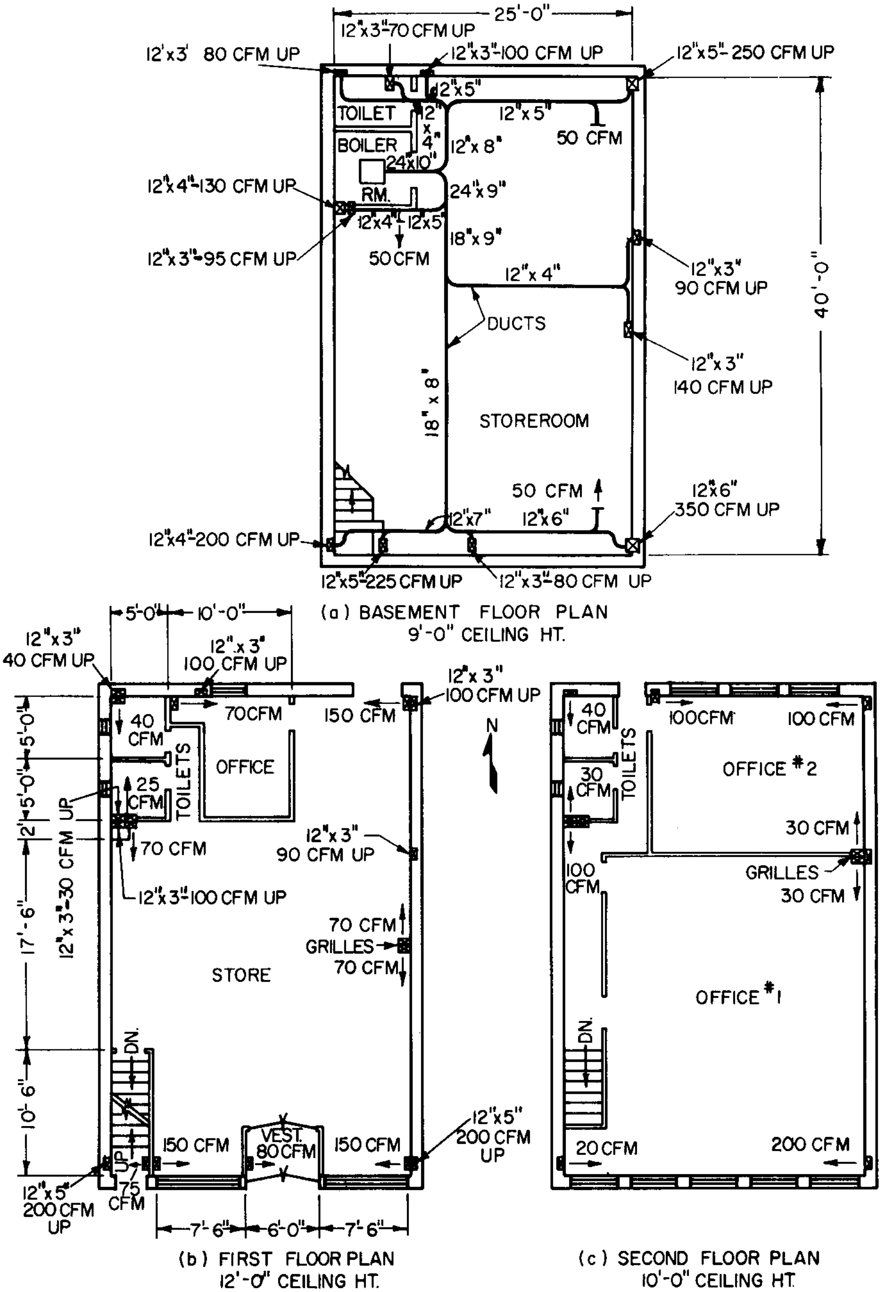

FIGURE 13.4 Layout of a duct system for warm-air heating of the basement, first floor, and second floor of the building shown in Fig. 13.3. The boiler room is in the basement.

TABLE 13.12 Air-Supply Distribution in Accordance with Heat Load

Space	Load, Btu per hr	% of load	Cfm (% × 2,010)
Cellar	14,500	10.05	200
First-floor store	47,600	33.00	660
Vestibule	5,650	3.91	80
Office	4,900	3.39	70
Men's room	2,800	1.94	40
Ladies' room	1,590	1.10	25
Second-floor office No. 1	33,300	23.01	460
Office No. 2	16,650	11.51	230
Men's room	2,820	1.95	40
Ladies' room	2,080	1.44	30
Hall	12,585	8.70	175
	144,475	100.00	2,010

$$\Delta T = \frac{q}{1.08Q} = \frac{160{,}000}{1.08 \times 2010} = 73.6°\text{F}$$

If we adjust the flame (oil or gas) so that the output was the capacity theoretically required, the temperature rise would be

$$\Delta T = \frac{q}{1.08Q} = \frac{144{,}475}{1.08 \times 2010} = 66.5°\text{F}$$

In actual practice, we do not tamper with the flame adjustment in order to maintain the manufacturer's design balance. Instead, the amount of air supplied to each room is in proportion to its load. (See Table 13.12 which is a continuation of Table 13.11 in the design of a forced-air heating system.) Duct sizes can then be determined for the flow indicated in the table (see Fig. 13.5). In this example, minimum duct size for practical purposes is 12 × 3 in. Other duct sizes were obtained from Table 13.7, for a friction loss per 100 ft of 0.15, and equivalent rectangular sizes were obtained from Table 13.6 and shown in Fig. 13.5.

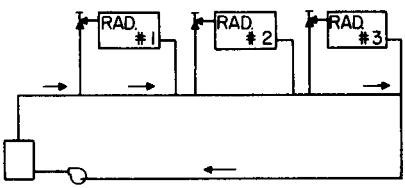

FIGURE 13.5 One-pipe hot-water heating system.

Humidification in Warm-Air Heating. A warm-air heating system lends itself readily to humidification. Most warm-air furnace manufacturers provide a humidifier that can be placed in the discharge bonnet of the heater.

Theoretically, a building needs more moisture when the outside temperature drops. During these colder periods, the heater runs more often, thus vaporizing more water. During warmer periods less moisture is required and less moisture is added because the heater runs less frequently.

Some manufacturers provide a woven asbestos-cloth frame placed in the pan to materially increase the contact surface between air and water. These humidifiers have no control.

Where some humidity control is desired, a spray nozzle connected to the hot-water system with an electric solenoid valve in the line may be actuated by a humidistat.

With humidification, well-fitted storm or double pane windows must be used, for with indoor conditions of 70°F and 30% relative humidity and an outdoor temperature of 0°F, condensation will occur on the windows. Storm windows will cut down the loss of room moisture.

Control of Warm-Air Heating. The sequence of operation of a warm-air heating system is usually as follows:

When the thermostat calls for heat, the heat source is started. When the air chamber in the warm-air heater reaches about 120°F, the fan is started by a sensitive element. This is done so as not to allow cold air to issue from the supply grilles and create drafts.

If the flame size and air quantity are theoretically balanced, the discharge air will climb to the design value of, say, 150°F and remain there during the operation of the heater. However, manual shutoff of grilles by residents, dirty filters, etc., will cause a reduction of airflow and a rise in air temperature above design. A sensitive safety element in the air chamber will shut off the heat source when the discharge temperature reaches a value higher than about 180°F. The heat source will again be turned on when the air temperature drops a given amount.

When the indoor temperature reaches the value for which the thermostat is set, the heat source only is shut off. The fan, controlled by the sensitive element in the air chamber, will be shut off after the air cools to below 120°F. Thus, most of the usable heat is transmitted into the living quarters instead of escaping up the chimney.

With commercial installations, the fan usually should be operated constantly, to maintain proper air circulation in windowless areas during periods when the thermostat is satisfied. If residential duct systems have been poorly designed, often some spaces may be too cool, while others may be too warm. Constant fan operation in such cases will tend to equalize temperatures when the thermostat is satisfied.

Duct systems should be sized for the design air quantity of the heater. Insufficient air will cause the heat source to cycle on and off too often. Too much air may cool the flue gases so low as to cause condensation of the water in the products of combustion. This may lead to corrosion, because of dissolved flue gases.

Warm-Air Perimeter Heating. This type of heating is often used in basementless structures, where the concrete slab is laid directly on the ground. The general arrangement is as follows: The heater discharges warm air to two or more underfloor radial ducts feeding a perimeter duct. Floor grilles or baseboard grilles are located as in a conventional warm-air heating system, with collars connected to the perimeter duct.

To prevent excessive heat loss to the outside, it is advisable to provide a rigid waterproof insulation between the perimeter duct and the outside wall.

Air Supply and Exhaust for Heaters. Special types of packaged, or preassembled, units are available for heating that include a direct oil- or gas-fired heat exchanger complete with operating controls. These units must have a flue to convey the products of combustion to the outdoors. The flue pipe must be incombustible and capable of withstanding high temperatures without losing strength from corrosion. Corrosion is usually caused by sulfuric and sulfurous acids, which are formed during combustion and caused by the presence of sulfur in the fuel.

Gas-fired heaters and boilers are usually provided with a draft hood approved by building officials. This should be installed in accordance with the manufacturer's recommendations. Oil-fired heaters and boilers should be provided with an approved draft stabilizer in the vent pipe. The hoods and stabilizers are used to prevent snuffing out of the flame in extreme cases and pulling of excessive air through the combustion chamber when the chimney draft is above normal, as in extremely cold weather.

Flues for the products of combustion are usually connected to a masonry type of chimney. A chimney may have more than one vertical flue. Where flue-gas temperatures do not exceed 600°F, the chimney should extend vertically 3 ft above the high point of the roof or roof ridge when within 10 ft of the chimney. When chimneys will be used for higher-temperature flue gases, many codes require that the chimney terminate not less than 10 ft higher than any portion of the building within 25 ft.

Many codes call for masonry construction of chimneys for both low- and high-temperature flue gases for low- and high-heat appliances. These codes also often call for fire-clay flue linings that will resist corrosion, softening, or cracking from flue gases at temperatures up to 1800°F.

Flue-pipe construction must be of heat-resistant materials. The cross-sectional area should be not less than that of the outlet on the heating unit. The flue or vent pipe should be as short as possible and have a slope upward of not less than ¼ in/ft. If the flue pipe extends a long distance to the chimney, it should be insulated to prevent heat loss and the formation of corrosive acids by condensation of the combustion products.

All combustion-type heating units require air for combustion, and it must be provided in adequate amounts. Combustion air is usually furnished directly from the outside. This air may be forced through ductwork by a fan or by gravity through an outdoor-air louver or special fresh-air intakes. If outside air is not provided for the heating unit, unsatisfactory results can be expected. The opening should have at least twice the cross-sectional area of the vent pipe leaving the boiler.

(H. E. Bovay, Jr., "Handbook of Mechanical and Electrical Systems for Buildings," and D. L. Grumman, "Air-Handling Systems Ready Reference Manual," McGraw-Hill Publishing Company, New York.)

13.12 HOT-WATER HEATING SYSTEMS

A hot-water heating system consists of a heater or furnace, radiators, piping systems, and circulator.

A gravity system without circulating pumps is rarely installed. It depends on a difference in density of the hot supply water and the colder return water for working head. Piping resistance must be kept to a minimum, and the circulating piping system must be of large size. A forced circulation system can maintain higher water velocities, thus requires much smaller pipes and provides much more sensitive control.

Three types of piping systems are in general use for forced hot-water circulation systems:

One-pipe system (Fig. 13.5). This type has many disadvantages and is not usually recommended. It may be seen in Fig. 13.5 that radiator No. 1 takes hot water from the supply main and dumps the colder water back in the supply main. This causes the supply-water temperature for radiator No. 2 to be lower, requiring

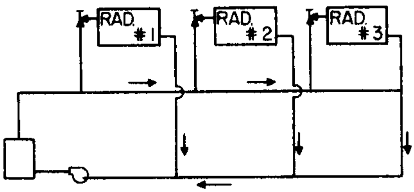

FIGURE 13.6 Two-pipe direct-return hot-water heating system.

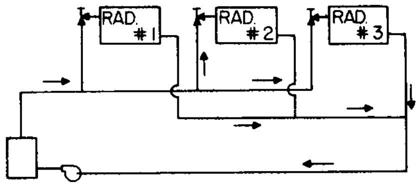

FIGURE 13.7 Two-pipe reversed-return hot-water heating system.

a corresponding increase in radiator size. (Special flow and return fittings are available to induce flow through the radiators.) The design of such a system is very difficult, and any future adjustment or balancing of the system throws the remainder of the temperatures out.

Two-pipe direct-return system (Fig. 13.6). Here are radiators get the same supply-water temperature, but the last radiator has more pipe resistance than the first. This can be balanced out by sizing the pump for the longest run and installing orifices in the other radiators to add an equivalent resistance for balancing.

Two-pipe reversed-return system (Fig. 13.7). The total pipe resistance is about the same for all radiators. Radiator No. 1 has the shortest supply pipe and the longest return pipe, while radiator No. 3 has the longest supply pipe and the shortest return pipe.

Supply design temperatures usually are 180°F, with a 20°F drop assumed through the radiators; thus the temperature of the return riser would be 160°F.

When a hot-water heating system is designed, it is best to locate the radiators, then calculate the water flow in gallons per minute required by each radiator. For a 20°F rise

$$q = 10{,}000Q \tag{13.33}$$

where q = amount of heat required, Btu/hr
Q = flow of water, gal/min

A one-line diagram showing the pipe runs should next be drawn, with gallons per minute to be carried by each pipe noted. The piping may be sized, using friction-flow charts and tables showing equivalent pipe lengths for fittings, with water velocity limited to a maximum of 4 ft/s. (See "ASHRAE Handbook—Fundamentals.") Too high a water velocity will cause noisy flow; too low a velocity will create a sluggish system and costlier piping.

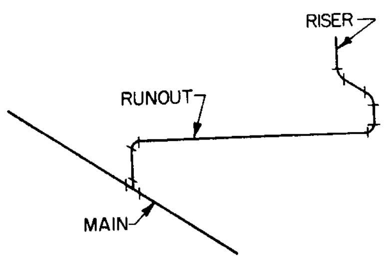

FIGURE 13.8 Swing joint permits expansion and contraction of pipes.

The friction should be between 250 and 600 milinches/ft (1 milinch = 0.001 in). It should be checked against available pump head, or a pump should be picked for the design gallons per minute at the require head.

It is very important that piping systems be made flexible enough to allow for expansion and contraction. Expansion joints are very satisfactory but expensive. Swing joints as shown in Fig. 13.8 should be used where necessary. In this type of branch takeoff, the runout pipe, on expansion or contraction, will cause the threads in the elbows to screw in or out slightly, instead of creating strains in the piping.

(W. J. McGuinness et al., ''Mechanical and Electrical Equipment for Buildings,'' John Wiley & Sons, Inc., New York.)

Hot-Water Radiators. Radiators, whether of the old cast-iron type, finned pipe, or other, should be picked for the size required, in accordance with the manufacturer's ratings. These ratings depend on the average water temperature. For 170°F average water temperature, 1 ft^2 of radiation surface is equal to 150 Btu/hr.

Expansion Tanks for Hot-Water Systems. All hot-water heating systems must be provided with an expansion tank of either the open or closed type.

Figure 13.9 shows an open-type expansion tank. This tank should be located at least 3 ft above the highest radiator and in a location where it cannot freeze.

The size of tank depends on the amount of expansion of the water. From a low near 32°F to a high near boiling, water expands 4% of its volume. Therefore, an expansion tank should be sized for 6% of the total volume of water in radiators, heater, and all piping. That is, the volume of the tank, up to the level of its overflow pipe, should not be less than 6% of the total volume of water in the system.

Figure 13.10 is a diagram of the hookup of a closed-type expansion tank. This tank is only partly filled with water, creating an air cushion to allow for expansion and contraction. The pressure-reducing valve and relief valve are often supplied as a single combination unit.

The downstream side of the reducing valve is set at a pressure below city watermain pressure but slightly higher than required to maintain a static head in the highest radiator. The minimum pressure setting in pounds per square inch is equal to the *height in feet* divided by 2.31.

The relief valve is set above maximum possible pressure. Thus, the system will automatically fill and relieve as required.

Precautions in Hot-Water Piping Layout. One of the most important precautions in a hot-water heating system is to avoid air pockets or loops. The pipe should be pitched so that vented air will collect at points that can be readily vented either automatically or manually. Vents should be located at all radiators.

Pipe traps should contain drains for complete drainage in case of shutdown.

Zones should be valved so that the complete system need not be shut down for repair of a zone. Multiple circulators may be used to supply the various zones at different times, for different temperature settings and different exposures.

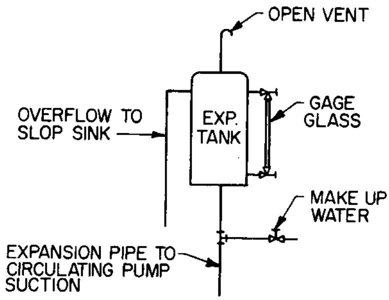

FIGURE 13.9 Open-type expansion tank.

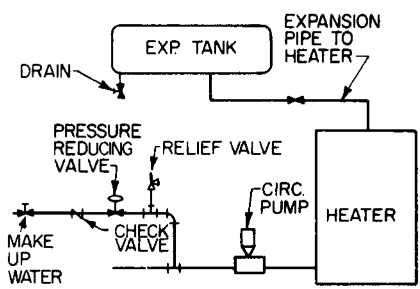

FIGURE 13.10 Closed-type expansion tank.

Allow for expansion and contraction of pipe without causing undue stresses.

All supply and return piping should be insulated.

In very high buildings, the static pressure on the boiler may be too great. To prevent this, heat exchangers may be installed as indicated in Fig. 13.11. The boiler temperature and lowest zone would be designed for 200°F supply water and 180°F return. The second lowest zone and heat exchanger can be designed for 170°F supply water and 150°F return, etc.

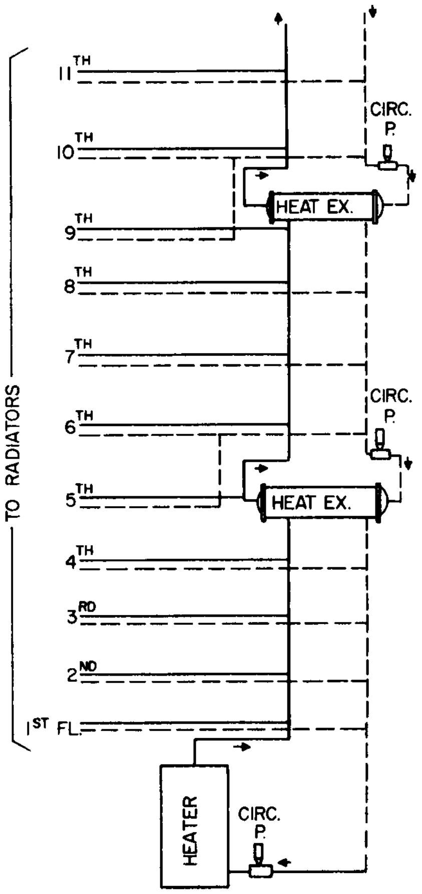

FIGURE 13.11 Piping layout for tall building, including heat exchangers.

Control of Hot-Water Systems. The control system is usually arranged as follows: An immersion thermostat in the heater controls the heat source, such as an oil burner or gas solenoid valve. The thermostat is set to maintain design heater water temperature (usually about 180°F). When the room thermostat calls for heat, it starts the circulator. Thus, an immediate supply of hot water is available for the radiators. A low-limit immersion stat, usually placed in the boiler and wired in series with the room stat and the pump, is arranged to shut off the circulator in the event that the water temperature drops below 70°F. This is an economy measure; if there is a flame failure, water will not be circulated unless it is warm enough to do some good. If the boiler is used to supply domestic hot water via an instantaneous coil or storage tank, hot water will always be available for that purpose. It should be kept in mind that the boiler must be sized for the heating load plus the probable domestic hot-water demand.

High-Temperature, High-Pressure Hot-Water Systems. Some commercial and industrial building complexes have installed hot-water heating systems in which the water temperature is maintained well above 212°F. This is made possible by subjecting the system to a pressure well above the saturation pressure of the water at the design temperature. Such high-temperature hot-water systems present some inherent hazards. Most important is the danger of a leak, because then the water will flash into steam. Another serious condition occurs when a pump's suction strainer becomes partly clogged, creating a pressure drop. This may cause steam to flash in the circulating pump casing and vapor bind the pump.

These systems are not generally used for heating with radiators. They are mostly used in conjunction with air-conditioning installations in which the air-handling units contain a heating coil for winter heating. The advantage of high-temperature hot-water systems is higher rate of heat transfer from the heating medium to the air. This permits smaller circulating piping, pumps, and other equipment.

(H. E. Bovay, Jr., "Handbook of Mechanical and Electrical Systems for Buildings," McGraw-Hill Publishing Company, New York; B. Stein et al., "Mechanical and Electrical Equipment for Buildings," 7th ed., John Wiley & Sons, Inc., New York.)

13.13 STEAM-HEATING SYSTEMS

A steam-heating system consists of a boiler or steam generator and a piping system connecting to individual radiators or convectors.

A **one-pipe heating system** (Fig. 13.12) is the simplest arrangement. The steam-supply pipe to the radiators is also used as a condensate return to the boiler. On startup, as the steam is generated, the air must be pushed out of the pipe and radiators by the steam. This is done with the aid of thermostatic air valves in the radiators. When the system is cold, a small vent hole in the valve is open. After the air is pushed out and steam comes in contact with the thermostatic element, the vent hole automatically closes to prevent escape of steam.

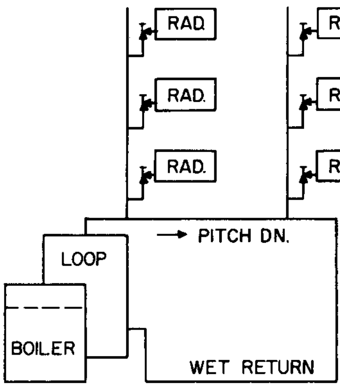

FIGURE 13.12 One-pipe steam-heating system. Condensate returns through supply pipe.

Where pipe runouts are extensive, it is necessary to install large-orifice air vents to eliminate the air quickly from the piping system; otherwise the radiators near the end of the runout may get steam much later than the radiators closest to the boiler.

Air vents are obtainable with adjustable orifice size for balancing a heating system. The orifices of radiators near the boiler are adjusted smaller, while radiators far from the boiler will have orifices adjusted for quick venting. This helps balance the system.

The pipe must be generously sized so as to prevent gravity flow of condensate from interfering with supply steam flow. Pipe capacities for supply risers, runouts, and radiator connections are given in the ASHRAE handbook (American Society of Heating, Refrigeration and Air Conditioning Engineers). Capacities are expressed in square feet of **equivalent direct radiation (EDR):**

$$1 \text{ ft}^2 \text{ EDR} = 240 \text{ Btu/hr} \tag{13.34}$$

where capacities are in pounds per hour, 1 lb/hr = 970 Btu/hr.

Valves on radiators in a one-pipe system must be fully open or closed. If a valve is throttled, the condensate in the radiator will have to slug against a head of steam in the pipe to find its way back to the boiler by gravity. This will cause water hammer.

A **two-pipe system** is shown in Fig. 13.13. The steam supply is used to deliver steam to the supply end of all radiators. The condensate end of each radiator is connected to the return line via a thermostatic drip trap (Fig. 13.14). The trap is

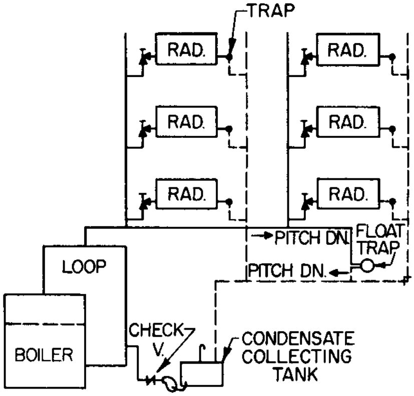

FIGURE 13.13 Two-pipe steam-heating system. Different pipes are used for supply and return processes.

adjusted to open below 180°F and close above 180°F. Thus when there is enough condensate in the radiator to cool the element, it will open and allow the condensate to return to the collecting tank. A float switch in the tank starts the pump to return this condensate to the boiler against boiler steam pressure.

There are many variations in combining the two-pipe system and one-pipe system to create satisfactory systems.

In the **two-pipe condensate pump return system,** the pressure drop available for pipe and radiator loss is equal to boiler pressure minus atmospheric pressure, the difference being the steam gage pressure.

A **wet-return system,** shown in Fig. 13.15, will usually have a smaller head available for pipe loss. It is a self-adjusting system depending on the load. When steam is condensing at a given rate, the condensate will pile up in the return main above boiler level, creating a hydraulic head that forces the condensate into the boiler. The pressure above the water level on the return main will be less than the steam boiler pressure. This pressure difference—boiler pressure minus the pressure that exists above the return main water level—is available for pipe friction and radiator pressure drop.

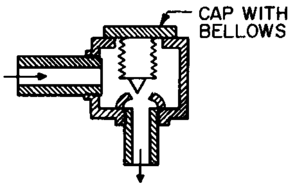

FIGURE 13.14 Drip trap for condensate return of a steam radiator.

On morning start-up when the air around the radiators is colder and the boiler is fired harder than during normal operation in an effort to pick up heat faster, the steam side of the radiators will be higher in temperature than normal. The air-side temperature of the radiators will be lower. This increase in temperature differential will create an increase in heat transfer causing a faster rate of condensation and a greater piling up of condensate to return the water to the boiler.

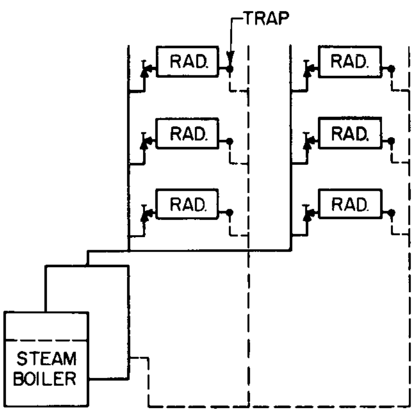

FIGURE 13.15 Wet-return two-pipe steam-heating system.

In laying out a system, the steam-supply runouts must be pitched to remove the condensate from the pipe. They may be pitched back so as to cause the condensate to flow against the steam or pitched front to cause the condensate to flow with the steam.

Since the condensate will pile up in the return pipe to a height above boiler-water level to create the required hydraulic head for condensate flow, a check of boiler-room ceiling height, steam-supply header, height of lowest radiator, height of dry return pipe, etc., is necessary to determine the height the water may rise in the return pipe without flooding these components.

The pipe may have to be oversized where condensate flows against the steam (see "ASHRAE Handbook—Fundamentals"). If the pitch is not steep enough, the steam may carry the condensate along in the wrong direction, causing noise and water hammer.

A **vacuum heating system** is similar to a steam pressure system with a condensate return pump. The main difference is that the steam pressure system can eliminate air from the piping and radiators by opening thermostatic vents to the atmosphere. The vacuum system is usually operated at a boiler pressure below atmospheric. The vacuum pump must, therefore, pull the noncondensables from the piping and radiators for discharge to atmosphere. Figure 13.16 shows diagrammatically the operation of a vacuum pump.

This unit collects the condensate in a tank. A pump circulates water through an eductor (1), pulling out the noncondensable gas to create the required vacuum. The discharge side of the eductor nozzle is above atmospheric pressure. Thus, an automatic control system allows the noncondensable gas to escape to atmosphere as it accumulates, and the excess condensate is returned to the boiler as the tank reaches a given level.

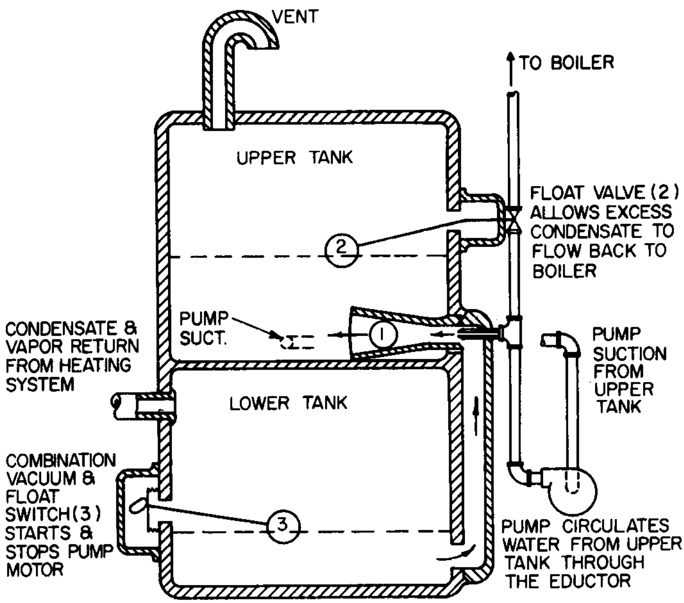

FIGURE 13.16 Diagram of a vacuum pump for a vacuum steam-heating system. The eductor (1) maintains the desired vacuum in the lower tank.

Vacuum systems are usually sized for a total pressure drop varying from ¼ to ½ psi. Long equivalent-run systems (about 200 ft) will use ½ psi total pressure drop to save pipe size.

Some systems operate without a vacuum pump by eliminating the air during morning pickup by hard firing and while the piping system is above atmospheric pressure. During the remainder of the day, when the rate of firing is reduced, a tight system will operate under vacuum.

(H. E. Bovay, Jr., "Handbook of Mechanical and Electrical Systems for Buildings," McGraw-Hill Publishing Company, New York.)

13.14 UNIT HEATERS

Large open areas, such as garages, showrooms, stores, and workshops, are usually best heated by unit heaters placed at the ceiling.

Figure 13.17 shows the usual connections to a steam unit-heater installation. The thermostat is arranged to start the fan when heat is required. The surface thermostat strapped on the return pipe prevents the running of the fan when insufficient steam is available. Where hot water is used for heating, the same arrangement is used, except that the float and the thermostatic trap are eliminated and an air-vent valve is installed. Check manufacturers' ratings for capacities in choosing equipment.

Where steam or hot water is not available, direct gas-fired unit heaters may be installed. However, an outside flue is required to dispose of the products of combustion properly (Fig. 13.18). Check with local ordinances for required flues from direct gas-fired equipment. (Draft diverters are usually included with all gas-fired heaters. Check with the manufacturer when a draft diverter is not included.) For automatic control, the thermostat is arranged to start the fan and open the gas solenoid valve when heat is required. The usual safety pilot light is included by the manufacturer.

Gas-fired unit heaters are often installed in kitchens and other premises where large quantities of air may be exhausted. When a makeup air system is not provided, and the relief air must infiltrate through doors, windows, etc., a negative pressure

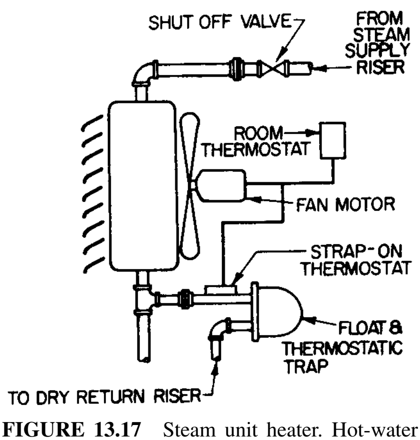

FIGURE 13.17 Steam unit heater. Hot-water heater has air vent, no float and trap.

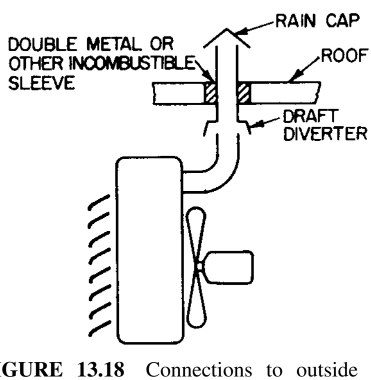

FIGURE 13.18 Connections to outside for gas-fired unit heater.

must result in the premises. This negative pressure will cause a steady downdraft through the flue pipe from the outdoors into the space and prevent proper removal of the products of combustion. In such installations, it is advisable to place the unit heater in an adjoining space from which air is not exhausted in large quantities and deliver the warm air through ducts. Since propeller fans on most unit heaters cannot take much external duct resistance, centrifugal blower unit heaters may give better performance where ductwork is used. Sizes of gas piping and burning rates for gas can be obtained from charts and tables in the "ASHRAE Handbook—Fundamentals" for various capacities and efficiencies.

The efficiency of most gas-fired heating equipment is between 70 and 80%.

13.15 RADIANT HEATING

Radiant heating, or panel heating as it is sometimes referred to, consists of a warm pipe coil embedded in the floor, ceiling, or walls. The most common arrangement is to circulate warm water through the pipe under the floor. Some installations with warm-air ducts, steam pipes, or electric-heating elements have been installed.

Warm-air ducts for radiant heating are not very common. A modified system normally called the perimeter warm-air heating system circulates the warm air around the perimeter of the structure before discharging the air into the premises via grilles.

Pipe coils embedded in concrete floor slabs or plaster ceilings and walls should not be threaded. Ferrous pipe should be welded, while joints in nonferrous metal pipe should be soldered. Return bends should be made with a pipe bender instead of fittings to avoid joints. All piping should be subjected to a hydrostatic test of at least 3 times the working pressure, with a minimum of 150 psig. Inasmuch as repairs are costly after construction is completed, it is advisable to adhere to the above recommendations.

Construction details for ceiling-embedded coils are shown in Figs. 13.19 to 13.22. Floor-embedded coil construction is shown in Fig. 13.23. Wall-panel coils may be installed as in ceiling panels.

Electrically heated panels are usually prefabricated and should be installed in accordance with manufacturer's recommendations and local electrical codes.

The piping and circuiting of a hot-water radiant heating system are similar to hot-water heating systems with radiators or convectors, except that cooler water is used. However, a 20°F water-temperature drop is usually assumed. Therefore, charts used for the design of hot-water piping systems may be used for radiant heating, too. (See "ASHRAE Handbook—Fundamentals.") In radiant heating,

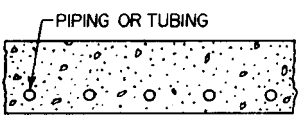

FIGURE 13.19 Pipe embedded in a concrete slab for radiant heating.

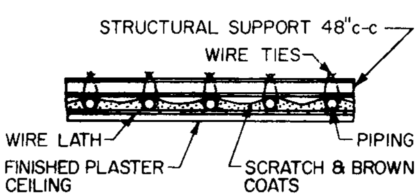

FIGURE 13.20 Pipe embedded in a plaster ceiling for radiant heating.

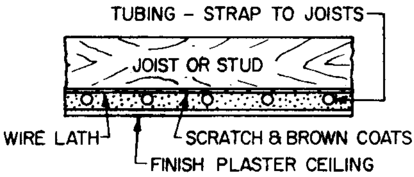

FIGURE 13.21 Pipe coil attached to joists or studs and embedded in plaster for radiant heating.

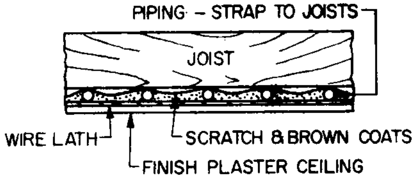

FIGURE 13.22 Pipe coil embedded above lath and plaster ceiling for radiant heating.

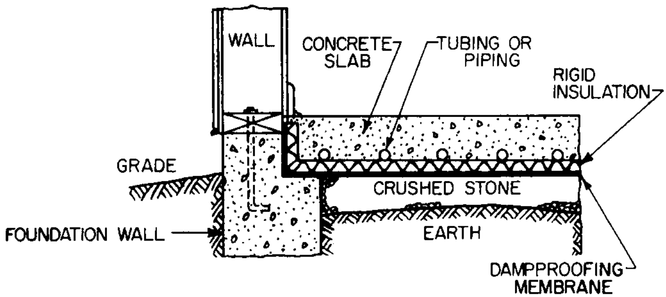

FIGURE 13.23 Pipe coil for radiant heating embedded in a floor slab on grade.

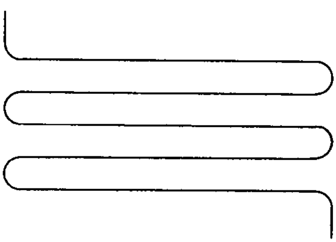

FIGURE 13.24 Continuous pipe coil for radiant heating.

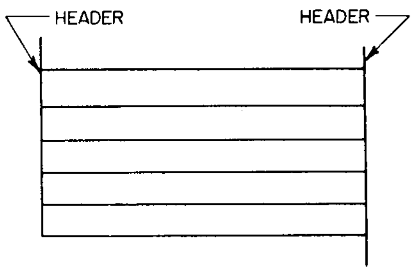

FIGURE 13.25 Piping arranged in a grid for radiant heating.

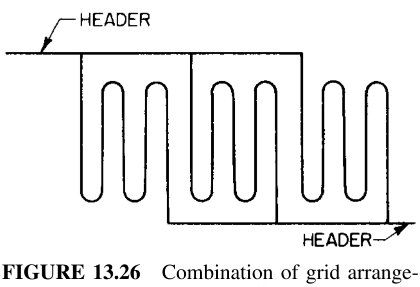

FIGURE 13.26 Combination of grid arrangement and continuous pipe coil.

the radiant coils replace radiators. Balancing valves should be installed in each coil, as in radiators. One wise precaution is to arrange coils in large and extensive areas so as not to have too much resistance in certain circuits. This can be done by using high-resistance continuous coils (Fig. 13.24) or low-resistance grid (Fig. 13.25) or a combination of both (Fig. 13.26).

One of the advantages of this system is its flexibility; coils can be concenthe warmer supply water may be routed to the perimeter or exposed walls and the cooler return water brought to the interior zones.

Panel and Room Temperatures. Heat from the embedded pipes is transmitted to the panel, which in turn supplies heat to the room by two methods: (1) convection and (2) radiation. The amount of heat supplied by convection depends on the temperature difference between the panel and the air. The amount of heat supplied by radiation depends on the difference between the fourth powers of the absolute temperatures of panel and occupants. Thus, as panel temperature is increased, persons in the room receive a greater percentage of heat by radiation than by convection. Inasmuch as high panel temperatures are uncomfortable, it is advisable to keep floor panel temperatures about 85°F or lower and ceiling panel temperatures 100°F or lower. The percentage of radiant heat supplied by a panel at 85°F is about 56% and by one at 100°F about 70%.

Most advocates of panel heating claim that a lower than the usual design inside temperature may be maintained because of the large radiant surface comforting the individual; i.e., a dwelling normally maintained at 70°F may be kept at an air temperature of about 65°F. The low air temperature makes possible a reduction in heat losses through walls, glass, etc., and thus cuts down the heating load. However, during periods when the heating controller is satisfied and the water circulation stops, the radiant-heat source diminishes, creating an uncomfortable condition due to the below-normal room air temperature. It is thus considered good practice to design the system for standard room temperatures (Table 13.10) and the heating plant for the total capacity required.

Design of Panel Heating. Panel output, Btu per hour per square foot, should be estimated to determine panel-heating area required. Panel capacity is determined by pipe spacing, water temperature, area of exposed walls and windows, infiltration air, insulation value of structural and architectural material between coil and occupied space, and insulation value of structural material preventing heat loss from the reverse side and edge of the panel. It is best to leave design of panel heating to a specialist.

(''ASHRAE Handbook—Fundamentals,'' American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1791 Tully Circle, N.E., Atlanta, GA 30329.)

13.16 SNOW MELTING

Design of a snow-melting system for sidewalks, roads, parking areas, etc., involves the determination of a design amount of snowfall, sizing and layout of piping, and selection of heat exchanger and circulating medium. The pipe is placed under the wearing surface, with enough cover to protect it against damage from traffic loads, and a heated fluid is circulated through it.

If friction is too high in extensive runs, use parallel loops (Art. 13.15). All precautions for drainage, fabrication, etc., hold for snow-melting panels as well as interior heating panels.

Table 13.13 gives a design rate of snowfall in inches of water equivalent per hour per square foot for various cities.

TABLE 13.13 Water Equivalent of Snowfall

City	In. per hr per sq ft	City	In. per hr per sq ft
Albany, N.Y.	0.16	Evansville, Ind.	0.08
Asheville, N.C.	0.08	Hartford, Conn.	0.25
Billings, Mont.	0.08	Kansas City, Mo.	0.16
Bismarck, N. Dak.	0.08	Madison, Wis.	0.08
Boise, Idaho	0.08	Minneapolis, Minn.	0.08
Boston, Mass.	0.16	New York, N.Y.	0.16
Buffalo, N.Y.	0.16	Okalhoma City, Okla.	0.16
Burlington, Vt.	0.08	Omaha, Nebr.	0.16
Caribou, Maine	0.16	Philadelphia, Pa.	0.16
Chicago, Ill.	0.06	Pittsburgh, Pa.	0.08
Cincinnati, Ohio	0.08	Portland, Maine	0.16
Cleveland, Ohio	0.08	St. Louis, Mo.	0.08
Columbus, Ohio	0.08	Salt Lake City, Utah	0.08
Denver, Colo.	0.08	Spokane, Wash.	0.16
Detroit, Mich.	0.08	Washington, D.C.	0.16

Table 13.14 gives the required slab output in Btu per hour per square foot at a given circulating-fluid temperature. This temperature may be obtained once we determine the rate of snowfall and assume a design outside air temperature and wind velocity. The table assumes a snow-melting panel as shown in Fig. 13.27.

Once the Btu per hour per square foot required is obtained from Table 13.14 and we know the area over which snow is to be melted, the total Btu per hour needed for snow melting can be computed as the product of the two. It is usual practice to add 40% for loss from bottom of slab.

The circulating-fluid temperature given in Table 13.14 is an average. For a 20°F rise, the fluid temperature entering the panel will be 10°F above that found in the table, and the leaving fluid temperature will be 10°F below the average. The freezing point of the fluid should be a few degrees below the minimum temperature ever obtained in the locality.

Check manufacturer's ratings for antifreeze solution properties to obtain the gallons per minute required and the friction loss to find the pumping head.

When ordering a heat exchanger for a given job, specify to the manufacturer the steam pressure available, fluid temperature to and from the heat exchanger, gallons per minute circulated, and physical properties of the antifreeze solution.

13.17 RADIATORS AND CONVECTORS

In hot-water and steam-heating systems, heat is released to the spaces to be warmed by radiation and convection. The percentage transmitted by either method depends on the type of heat-dispersal unit used.

A common type of unit is the tubular radiator. It is composed of a series of interconnected sections, each of which consists of vertical tubes looped together. Steam-radiator sections are attached by nipples only at the base, whereas hot-water sections are connected at both top and bottom. Steam radiators should not be used for hot-water heating because of the difficulty of venting.

TABLE 13.14. Heat Output and Circulating-Fluid Temperatures for Snow-Melting Systems

		Air temp, 0°F			Air temp, 10°F			Air temp, 20°F			Air temp, 30°F		
		Wind velocity, mi/hr											
		5	10	15	5	10	15	5	10	15	5	10	15
0.08	Slab output, Btu/(hr)(ft^2)	151	205	260	127	168	209	102	128	154	75	84	94
	Fluid temp, °F	108	135	162	97	117	138	85	97	110	70	75	79
0.16	Slab output, Btu/(hr)(ft^2)	218	273	327	193	233	274	165	191	217	135	144	154
	Fluid temp, °F	142	169	198	120	149	170	117	129	142	100	105	109
0.25	Slab output, Btu/(hr)(ft^2)	292	347	401	265	305	346	235	261	287	203	212	221
	Fluid temp, °F	179	206	234	165	186	206	151	163	176	134	139	144

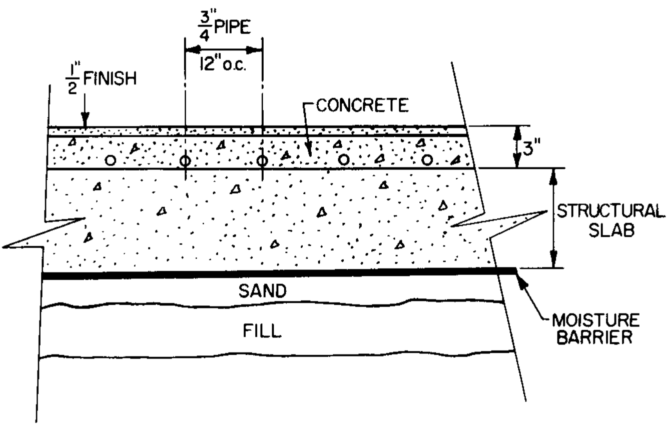

FIGURE 13.27 Pipe coil embedded in a concrete slab for outdoor snow melting.

Pipe-coil radiators are sometimes used in industrial plants. The coils are usually placed on a wall under and between windows and are connected at the ends by branch trees or manifolds. Sometimes, finned-pipe coils are used instead of ordinary pipe. The fins increase the area of heat-transmitting surface.

Since radiators emit heat by convection as well as by radiation, any enclosure should permit air to enter at the bottom and leave at the top.

Convectors, as the name implies, transmit heat mostly by convection. They usually consist of finned heating elements placed close to the floor in an enclosure that has openings at bottom and top for air circulation.

Baseboard units consist of continuous heating pipe in a thin enclosure along the base of exposed walls. They may transmit heat mostly by radiation or by convection. Convector-type units generally have finned-pipe heating elements. Chief advantage is the small temperature difference between floor and ceiling.

13.18 HEAT PUMPS

A heat-pump cycle is a sequence of operations in which the heat of condensation of a refrigerant is used for heating. The heat required to vaporize the refrigerant is taken from ambient air at the stage where the normal refrigeration cycle (Art. 13.22) usually rejects the heat. In the summer cycle, for cooling, the liquid refrigerant is arranged to flow to the cooling coil through an expansion valve, and the hot gas from the compressor is condensed as in the standard refrigeration cycle (Fig. 13.29). During the heating season, the refrigerant gas is directed to the indoor (heating) coil by the use of multiport electric valves. The condensed liquid refrigerant is then directed to the ‘‘condenser’’ via an expansion valve and is evaporated. This method of heating is competitive with fuel-burning systems in warmer climates where the cooling plant can provide enough heat capacity during the winter season and where electric rates are low. In colder latitudes, the cooling plant, when used as a heat pump, is not large enough to maintain design indoor temperatures and is therefore not competitive with fuel-burning plants.

Other heat sources for heat pumps are well water or underground grid coils. These installations usually call for a valve system permitting warm condenser water to be piped to the air-handling unit for winter heating and the cold water from the chiller to be pumped to the air-handling unit for summer cooling. In winter, the well water or the water in the ground coil is pumped through the chiller to evaporate the liquid refrigerant, while in the summer this water is pumped through the condenser.

Packaged units complete with controls are available for both air-to-air and air-to-water use. Units are made as roof-top or grade-mounted units. Special built-up systems with large capacity have also been constructed.

13.19 SOLAR HEATING

Solar radiation may be used to provide space heating, cooling, and domestic hot water, but the economics of the application should be carefully investigated. Returns on the initial investment from savings on fuel costs may permit a payback for solar domestic hot-water systems in about 6 to 10 years. Heating and cooling systems will take much longer. The advantage of solar heat is that it is renewable, nonpolluting, and free. Therefore, use of solar heat will overcome the continuing high cost of energy from other sources and conserve those fuels that are in limited supply. However, the materials used to collect and transfer solar energy (copper, glass, and aluminum) are energy intensive to manufacture and will continue to escalate with rising energy cost.

There are various disadvantages in using solar radiation. For one thing, it is available only when the sun is shining and there are hourly variations in intensity, daily and with the weather. Also, the energy received per square foot of radiated surface is small, generally under 400 Btu/(hr)(ft^2). This is a very low energy flux. It necessitates large areas of solar collectors to obtain sufficient energy for practical applications and also provide a reasonable payback on the investment.

Solar heating or cooling is advantageous only when the cost of the solar energy produced is less than the cost of energy produced by the more conventional methods. In general, the cost of solar systems may be reduced by obtaining minimum initial costs, favorable amortization rates for the equipment required, and governmental investment tax credits. Also, continuous heat loads and an efficient heating-system design will keep costs low. Bear in mind that the efficiency of solar-system designs for heating or cooling depends on the efficiency of solar collectors, the efficiency of the conversion of the solar radiation to a more useful form of energy, and the efficiency of storage of that energy from the time of conversion until the time of use.

The simplest method of collecting solar energy is by use of a flat-plate-type collector (Fig. 13.28*b*). The collector is mounted in a manner that allows its flat surface to be held normal, or nearly so, to the sun's rays, thereby, in effect, trapping the solar radiation within the collector. Flat-plate-type collectors are used only in low-temperature systems (70 to 180°F). Evacuated-tube focusing/concentrating collectors generate much higher temperatures by minimizing heat losses and concentrating sunlight on a reduced absorber surface. Evacuated-tube type collectors operate in a range of 185 to 250°F, while the concentrating-type collectors operate in an even higher range of 250 to 500°F, or more.

Precautions should be taken in design and installation of solar collectors on low-sloped roofs to prevent excessive roof deflections or overloads. Also, care should be taken to avoid damaging the roofing or creating conditions that would cause premature roofing failures or lead to higher roofing repair, maintenance, or replacement costs. In particular:

1. Solar collectors should not be installed where ponding of rainwater may occur or in a manner that will obstruct drainage of the roof.
2. Solar collector supports and the roof should be designed for snow loads as well as for wind loads on the collectors, including uplift.
3. The installation should not decrease the fire rating of the roof.
4. The roofing should be protected by boards or other means during erection of the collector. If the roofing membrane must be penetrated for the supports or piping, a roofer should install pipe sleeves, flashing, and other materials necessary to keep out water. If bitumen is used, it should not be permitted to splatter on the collector cover plates.
5. At least 24-in clearance should be provided between the bottom of the collector frames and the roof, to permit inspection, maintenance, repair, and replacement of the roof. Similarly, clearance should be provided at the ends of the frames.
6. At least 14-in clearance should be provided between thermal pipes and the roof surface for the preceding reason. The pipes should be supported on the collector frames, to the extent possible, rather than on the roof.

A typical solar heating and cooling system consists of several major system components: collectors, heat storage, supplemental heat source, and auxiliary equipment.

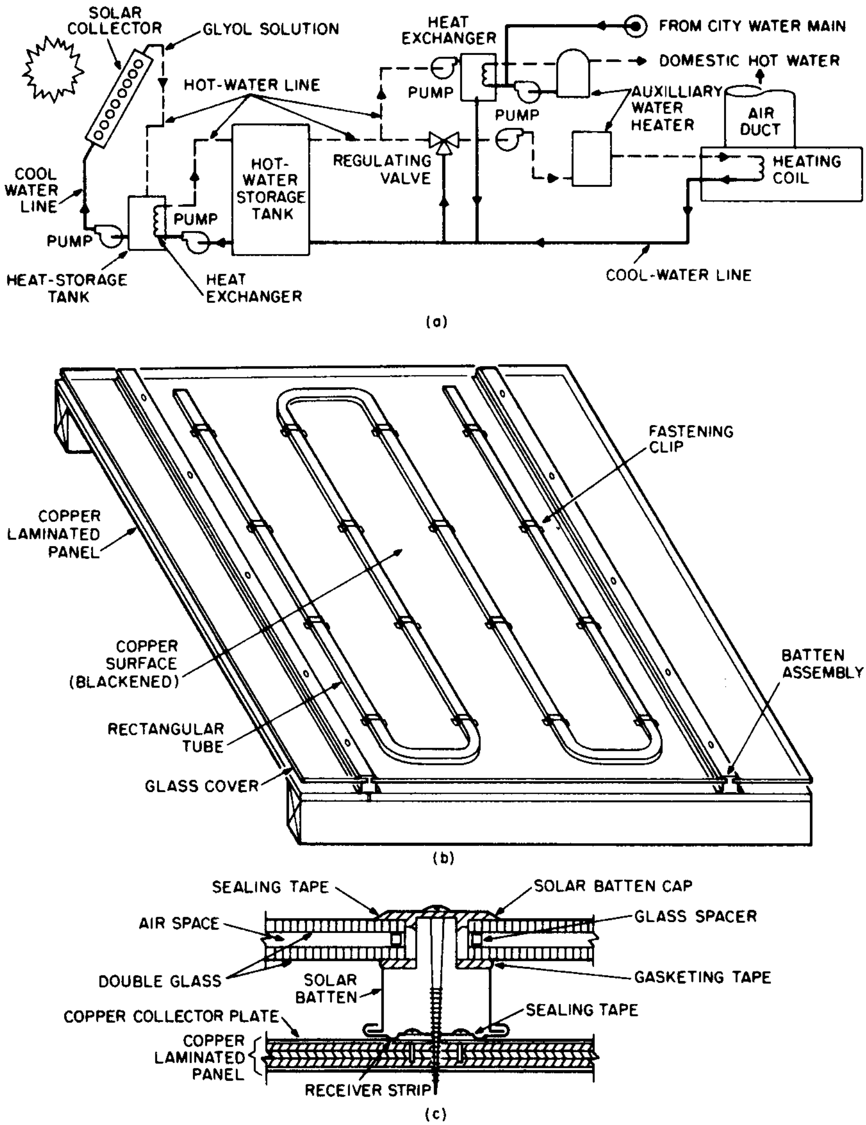

FIGURE 13.28 Example of a solar-heating system for a building: (*a*) heat absorbed from solar radiation by a glycol solution is transferred to water for heating a building and producing domestic hot water; (*b*) solar collector for absorbing solar radiation; (*c*) detail of the batten assembly of the solar collector. (*Reprinted with permission from F. S. Merritt and J. Ambrose, "Building Engineering and Systems Design," 2d ed., Van Nostrand Reinhold Company, New York.*)

Many liquid solutions are used for heat transmission in solar systems. A typical solution may be a mixture of water and 10% propylene-glycol antifreeze solution. It is pumped through the collector tubes in a closed-loop circuit to a heat exchanger, where the solution transfers its energy to a water-heating circuit (Fig. 13.28*a*). The cold glycol solution is then pumped back to the collectors.

The heated water travels in its own closed circuit to a hot-water storage tank and back to the exchanger. Hot water from the storage tank is withdrawn, as required, to satisfy building heating requirements.

The hot water may be pumped from the storage tank to another heat exchanger to produce hot water for domestic use. Hot water from the storage tank may also be pumped through another closed-loop circuit to a heating coil mounted in a central heating blower unit for space heating. In both cases, the cooled water is returned to the storage tank or to the heat exchanger served by the glycol solution.

For use when insufficient solar heat is available, an auxiliary water heater should be added to the system. Providing hot water for heating and domestic purposes, the auxiliary heater may be electrically operated or gas, oil, or coal fired.

When solar heat is to be used for cooling, the hot-water loop must be modified. A three-way valve is placed in the hot-water line to direct the hot water to an absorption-type water chiller. The chilled water discharge is then pumped to a cooling coil in the main air-handling unit.

Passive solar systems and attention to solar exposure have had a significant impact on HVAC design and architecture of buildings. Proper exposures, shading, and daylighting techniques have gained widespread use.

METHODS OF COOLING AND AIR CONDITIONING

The methods for establishing accurate heat losses for heating are also applicable for heat gains and air conditioning. It is mandatory that such procedures be used in order that the necessary cooling equipment can be sized and selected with the lowest first cost that will provide reliable and satisfactory service. (See Arts. 13.7 and 13.8.)

13.20 SIZING AN AIR-CONDITIONING PLANT

Consider the building shown in Fig. 13.3 (p. 13.43). Assume only the first and second floors will be air conditioned. The design outdoor condition is assumed to be 95°F DB (dry-bulb) and 75°F WB (wet-bulb). The design indoor condition is 80°F DB and 50% relative humidity.

The temperature gradient across an exposed wall will be 15°F (95°F − 80°F). The temperature gradient between a conditioned and an interior nonconditioned space, such as the cellar ceiling, is assumed to be 10°F.

Exterior walls are constructed of 4-in brick with 8-in cinder backup; interior finish is plaster on metal lath. Partitions consist of 2 × 4 studs, wire lath, and plaster. First floor has double flooring on top of joists; cellar ceiling is plaster on metal lath.

Lights may be assumed to average 4 W/ft^2. Assume 50 persons will be in the store, 2 in the first-floor office, 10 in the second-floor office No. 1, and 5 in second-floor office No. 2.

Cooling Measured in Tons. Once we obtain the cooling load in Btu per hour, we convert the load to tons of refrigeration by Eq. (13.15).

Cooling Requirements for First-Floor Store (Table 13.15*a*).

$$\text{Load in tons} = \frac{86{,}951}{12{,}000} = 7.25 \text{ tons}$$

If supply air is provided at 18°F differential, with fresh air entering the unit through an outside duct, the dehumidified air flow required for sensible heat is [from Eq. (13.30)]:

TABLE 13.15*a* First-Floor Store Load

	Area, ft^2		$U(T_o - T_i)$, °F	Heat load, Btu/hr
Heat gain through enclosures:				
North wall	92	×	0.25 × 15	344
North door	28	×	0.46 × 15	193
South wall	412	×	0.25 × 15	1,540
South glass	154	×	1.04 × 15	2,405
East wall	480	×	0.25 × 15	1,800
West wall	228	×	0.25 × 15	854
Partition	263	×	0.39 × 10	1,025
Partition door	49	×	0.46 × 10	225
Floor	841	×	0.25 × 10	2,100
Ceiling	112	×	0.25 × 10	280
Load due to conduction				10,766
Sun load on south glass, 154 ft^2 × 98				15,100
Occupants (sensible heat) from Table 13.9, 50 × 315				15,750
Lights [Eq. (13.29)] 841 ft^2 × 4 W/ft^2 × 3.42				11,500
				53,116
Fan hp (4% of total)				2,125
Total internal sensible heat load				55,241
Fresh air (sensible heat) at 10 ft^3/min per person [Eq. (13.30),* 500 ft^3/min × 1.08 × 15]				8,100
Total sensible load				63,341
Occupants (latent heat) from Table 13.9, 50 × 325				16,250
Fresh air (latent heat) Eq. (13.31), 500 × 0.67(99 − 77)				7,360
Total latent load				23,610
Total load				86,951

*Current criteria typically requires higher levels of ventilation air.

TABLE 13.15*b* First-Floor Office Load

	Btu/hr	
Heat gain through enclosures:		
North wall, 102 × 0.25 × 15	382	
North glass, 18 × 1.04 × 15	281	
Partition, 30 × 0.39 × 10	117	
Floor, 79 × 0.25 × 10	198	
	978	
2 occupants (sensible load) at 255	510	
Lights, 79 × 4 × 3.42	1080	
	2568	
Fan hp 4%	103	
Total internal sensible load		2671
Fresh air (sensible load) at 2 changes per hour, 32 × 1.08 × 15		518
Total sensible load		3189
Occupants (latent load), 2 at 255	510	
Fresh air (latent load), 32 × 0.67(99 − 77)	473	
Total latent load		983
Total load		4172

$$Q = \frac{55{,}241}{1.08 \times 18} = 2842 \text{ ft}^3/\text{min}$$

If ducted fresh air is not provided and all the infiltration air enters directly into the premises, the flow required is computed as follows:

$$\text{Store volume} = 841 \times 12 = 10{,}100 \text{ ft}^3$$

$$\text{Infiltration} = \frac{10{,}100}{60} \times 1.5 = 252 \text{ ft}^3/\text{min}$$

$$\text{Sensible load from the infiltration air [Eq. (13.30)]} = 252 \times 1.08 \times 15 = 4090 \text{ Btu/hr}$$

$$\text{Internal sensible load plus infiltration} = 55{,}241 + 4090 = 59{,}331 \text{ Btu/hr}$$

$$\text{Flow without ducted fresh air} = \frac{59{,}331}{1.08 \times 18} = 3052 \text{ ft}^3/\text{min}$$

Cooling Requirements for First-Floor Office (Table 13.15*b*).

$$\text{Load in tons} = \frac{4172}{12{,}000} = 0.35 \text{ ton}$$

$$\text{Supply air required with ducted fresh air} = \frac{2671}{1.08 \times 18} = 137 \text{ ft}^3/\text{min}$$

$$\text{Supply air required without ducted fresh air} = \frac{3189}{1.08 \times 18} = 164 \text{ ft}^3/\text{min}$$

Cooling Requirements for Second-Floor Office No. 1 (Table 13.15*c*).

$$\text{Load in tons} = \frac{41{,}888}{12{,}000} = 3.5 \text{ tons}$$

$$\text{Supply air required with ducted fresh air} = \frac{33{,}768}{1.08 \times 18} = 1737 \text{ ft}^3/\text{min}$$

$$\text{Supply air required without ducted fresh air} = \frac{36{,}678}{1.08 \times 18} = 1887 \text{ ft}^3/\text{min}$$

Cooling Requirements for Second-Floor Office No. 2 (Table 13.15*d*).

$$\text{Load in tons} = \frac{13{,}362}{12{,}000} = 1.13 \text{ tons}$$

$$\text{Supply air required with ducted fresh air} = \frac{9697}{1.08 \times 18} = 499 \text{ ft}^3/\text{min}$$

$$\text{Supply air required without ducted fresh air} = \frac{10{,}947}{1.08 \times 18} = 563 \text{ ft}^3/\text{min}$$

TABLE 13.15*c* Second-Floor Office No. 1

	Btu/hr	
Heat gain through enclosures:		
South wall, 130 × 0.25 × 15	388	
South glass, 120 × 1.04 × 15	1,870	
East wall, 260 × 0.25 × 15	975	
West wall, 40 × 0.25 × 15	150	
Partition, 265 × 0.39 × 10	1,035	
Door in partition, 35 × 0.46 × 10	161	
Roof, 568 × 0.19 × 54	5,820	
	10,399	
Sun load on south glass, 120 × 98	11,750	
Occupants (sensible load), 10 × 255	2,550	
Lights, 568 × 4 × 3.42	7,770	
	32,469	
Fan hp 4 %	1,299	
Total internal sensible load		33,768
Fresh air (sensible load), 180 × 1.08 × 15		2,910
Total sensible load		36,678
Occupants (latent load), 10 × 255	2,550	
Fresh air (latent load), 180 × 0.67(99 − 77)	2,660	
Total latent load		5,210
Total load		41,888

TABLE 13.15*d* Second-Floor Office No. 2

	Btu/hr	
Heat gain through enclosures:		
North wall, 103 × 0.25 × 15	386	
North glass, 72 × 1.04 × 15	1,110	
East wall, 132 × 0.25 × 15	495	
Partition, 114 × 0.39 × 10	445	
Door in partition, 18 × 0.46 × 10	83	
Roof, 231 × 0.19 × 54	2,370	
	4,889	
Occupants (sensible load), 5 × 255	1,275	
Lights, 231 × 4 × 3.42	3,160	
	9,324	
Fan hp 4%	373	
Total internal sensible load		9,697
Fresh air (sensible load), 77 × 1.08 × 15		1,250
Total sensible load		10,947
Occupants (latent load), 5 × 255	1,275	
Fresh air (latent load), 77 × 0.67(99 − 77)	1,140	
Total latent load		2,415
Total load		13,362

Cooling and supply-air requirements for the building are summarized in Table 13.16.

13.21 REFRIGERATION CYCLES

Figure 13.29 shows the basic air-conditioning cycle of the **direct-expansion type.** The compressor takes refrigerant gas at a relatively low pressure and compresses it to a higher pressure. The hot gas is passed to a condenser where heat is removed and the refrigerant liquefied. The liquid is then piped to the cooling coil of the air-handling unit and allowed to expand to a lower pressure (suction pressure). The liquid vaporizes or is boiled off by the relatively warm air passing over the coil.

TABLE 13.16 Cooling-Load Analysis for Building in Fig. 13.3 (p. 13.43)

Space	Tons	Flow with ducted fresh air, ft^3/min	Flow without ducted fresh air, ft^3/min
First-floor store	7.25	2842	3052
First-floor office	0.35	137	164
Second-floor office No. 1	3.50	1737	1887
Second-floor office No. 2	1.13	499	563
	12.23	5215	5666

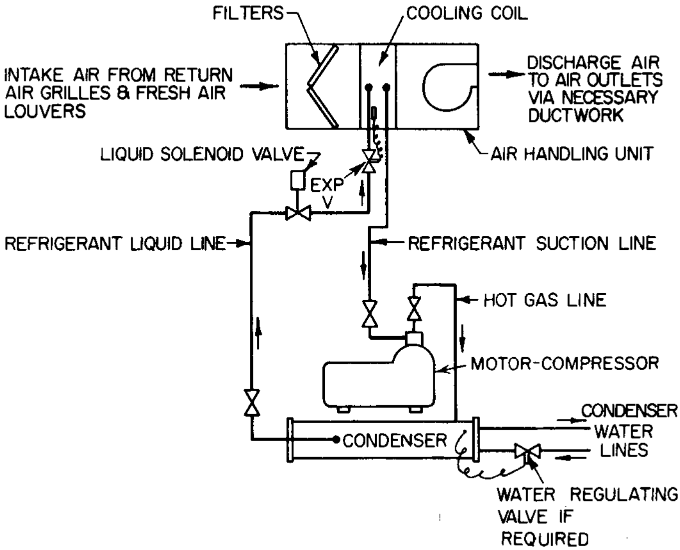

FIGURE 13.29 Direct-expansion air-conditioning cycle.

The compressor pulls away the vaporized refrigerant to maintain the required low coil pressure with its accompanying low temperature.

Chilled-Water Refrigeration Cycle. In some systems, water is chilled by the refrigerant and circulated to units in or near spaces to be cooled (Fig. 13.30), where air is cooled by the water.

In water-cooled and belt-driven air-conditioning compressors formerly used, motor winding heat was usually dissipated into the atmosphere outside the conditioned space (usually into the compressor rooms). For average air-conditioning service 1 hp could produce about 1 ton of cooling. When sealed compressor-motor units came into use, the motor windings were arranged to give off their heat to the refrigerant suction gas. This heat was therefore added to the cooling load of the compressor. The result was that 1 hp could produce only about 0.85 ton of cooling.

Because of water shortages, many communities restrict the direct use of city water for condensing purposes. As an alternative, water can be cooled with cooling towers and recirculated. But for smaller systems, cooling towers have some inherent disadvantages. As a result, air-cooled condensers have become common for small- and medium-sized air-conditioning systems. Because of the higher head pressures resulting from air-cooled condensers, each horsepower of compressor-motor will produce only about 0.65 ton of cooling. Because of these developments, air-conditioning equipment manufacturers rate their equipment in Btu per hour output and kilowatts required for the rated output, at conditions standardized by the industry, instead of in horsepower.

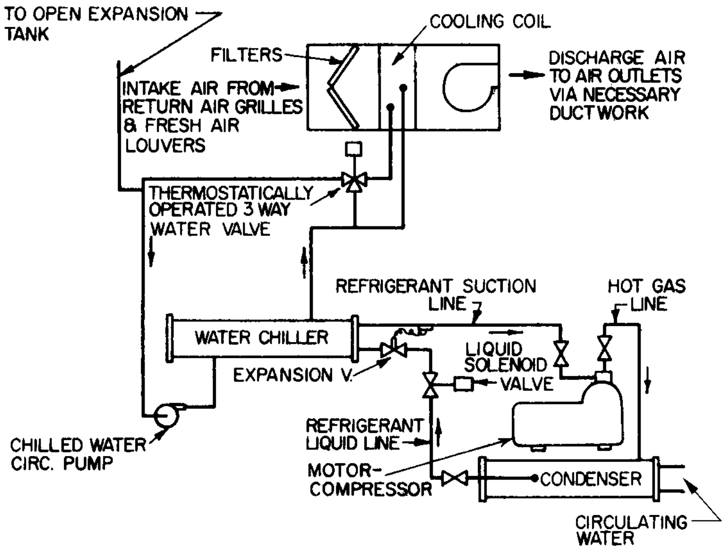

FIGURE 13.30 Chilled-water air-conditioning cycle.

13.22 AIR-DISTRIBUTION TEMPERATURE FOR COOLING

The air-distribution system is the critical part of an air-conditioning system. If insufficient air is circulated, proper cooling cannot be done. On the other hand, handling large quantities of air is expensive in both initial cost and operation. The amount of cool air required increases rapidly the closer its temperature is brought to the desired room temperature.

If, for example, we wish to maintain 80°F DB (dry-bulb) in a room and we introduce air at 60°F, the colder air when warming up to 80°F will absorb an amount of sensible heat equal to q_s. According to Eq. (13.30), $q_s = 1.08Q_1(80 - 60) = 21.6Q_1$, where Q_1 is the required airflow in cubic feet per minute. From Eq. (13.30), it can be seen also that, if we introduce air at 70°F, with a temperature rise of 10°F instead of 20°F, $q_s = 10.8Q_2$, and we shall have to handle twice as much air to do the same amount of sensible cooling.

From a psychrometric chart, the dew point of a room at 80°F DB and 50% relative humidity is found to be 59°F. If the air leaving the air-conditioning unit is 59°F or less, the duct will sweat and will require insulation. Even if we spend the money to insulate the supply duct, the supply grilles may sweat and drip. Therefore, theoretically, to be safe, the air leaving the air-conditioning unit should be 60°F or higher.

Because there are many days when outside temperatures are less than design conditions, the temperature of the air supplied to the coil will fluctuate, and the temperature of the air leaving the coil may drop a few degrees. This will result in sweating ducts. It is good practice, therefore, to design the discharge-air temperature about 3°F higher than the room dew point.

Thus, for 80°F DB, 50% RH, and dew point at 59°F, the minimum discharge air temperature would be 62°F as insurance against sweating. The amount of air to be handled may be obtained from Eq. (13.30), with a temperature difference of 18°F.

13.23 CONDENSERS

If a water-cooled condenser is employed to remove heat from the refrigerant, a water tower (Fig. 13.31) may be used to cool the condenser discharge water, which can then be recirculated back to the condenser.

Where practical, the water condenser and tower can be replaced by an evaporative condenser as in Fig. 13.32.

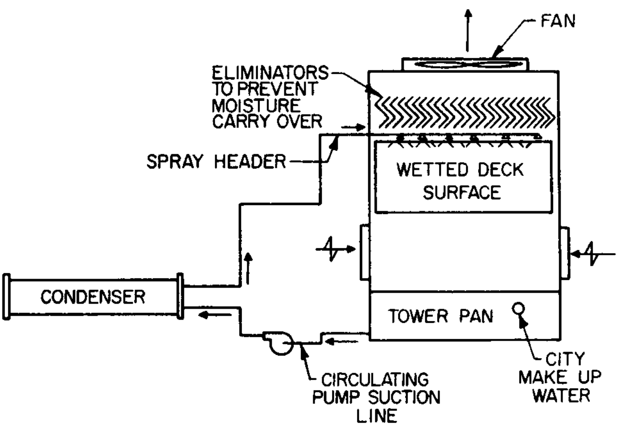

FIGURE 13.31 Water tower connection to a condenser of an air-conditioning system.

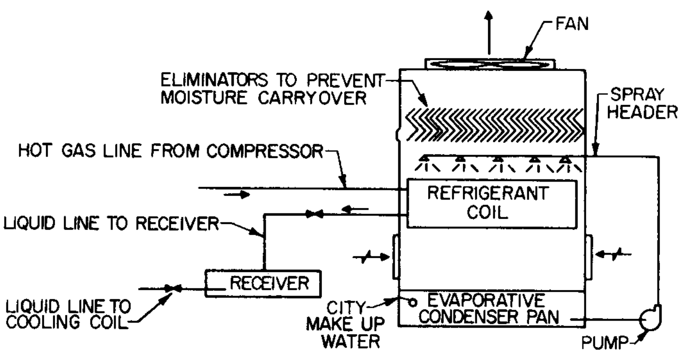

FIGURE 13.32 Evaporative condenser for an air-conditioning system.

The capacity of heat rejection equipment, such as towers or evaporative condensers, depends on the wet-bulb temperature. The capacity of these units decreases as the wet-bulb temperature increases.

Such equipment should be sized for a wet-bulb temperature a few degrees above that used for sizing air-conditioning equipment.

As an example, consider an area where the design wet-bulb temperature is 75°F. If we size the air-conditioning equipment for this condition, we shall be able to maintain design inside conditions when the outside conditions happen to be 75°F WB. There will be a few days a year, however, when the outside air may register 79 or 80°F WB. During the higher wet-bulb days, with the air-conditioning equipment in operation, we shall balance out at a relative humidity above design. For example, if the design relative humidity is 50%, we may balance out at 55% or higher.

However, if the water towers and evaporative condensers are designed for 75°F WB, then at a wet-bulb temperature above 75, the equipment capacity would be decreased and the compressor head pressure would build up too high, overload the motor, and kick out on the overload relays. So we use 78°F WB for the design of heat rejection equipment. Now, on a 78°F WB day, the compressor head pressure would be at design pressure, the equipment will be in operation, although maintaining room conditions a little less comfortable than desired. At 80°F WB, the compressor head pressure will build up above design pressure and the motor will be drawing more than the design current but usually less than the overload rating of the safety contact heaters.

Water condensers are sized for 104°F refrigerant temperature and 95°F water leaving.

The amount of water in gallons per minute required for condensers is subject to the type of heat rejection equipment used and manufacturer's data should be used. A rule of thumb is 3 gpm/ton per 10°F water-temperature rise.

When choosing a condenser for water-tower use, we must determine the water temperature available from the cooling tower with 95°F water to the tower. Check the tower manufacturer for the capacity required at the design WB, and 95°F water to the sprays, and the appropriate wet-bulb approach. (Wet-bulb approach is equal to the number of degrees the temperature of the water leaving the tower is above the wet-bulb temperature. This should be for an economic arrangement about 7°F.) Thus, for 78°F, the water leaving the tower would be 85°F, and the condenser would be designed for a 10°F water-temperature rise, i.e., for water at 95°F to the tower and 85°F leaving the tower.

Evaporative condensers should be picked for the required capacity at a design wet-bulb temperature for the area in which they are to be installed. Manufacturers' ratings should be checked before the equipment is ordered.

For small- and medium-size cooling systems, air-cooled condensers are available for outdoor installation with propeller fans or for indoor installation usually with centrifugal blowers for forcing outdoor air through ducts to and from the condensers.

13.24 COMPRESSOR-MOTOR UNITS

Compressor capacity decreases with increase in head pressure (or temperature) and fall in suction pressure (or temperature). Therefore, in choosing a compressor-motor unit, one must first determine design conditions for which this unit is to operate; for example:

Suction temperature—whether at 35°F, 40°F, 45°F, etc.

Head pressure—whether serviced by a water-cooled or air-cooled condenser.

Manufacturers of compressor-motor units rate their equipment for Btu per hour output and kilowatt and amperage draw at various conditions of suction and discharge pressure. These data are available from manufacturers, and should be checked before a compressor-motor unit is ordered.

Although capacity of a compressor decreases with the suction pressure (or back pressure), it is usual practice to rate compressors in tons or Btu per hour at various suction temperatures.

For installations in which the latent load is high, such as restaurants, bars, and dance halls, where a large number of persons congregate, the coil temperature will have to be brought low enough to condense out large amounts of moisture. Thus, we find that a suction temperature of about 40°F will be required. In offices, homes, etc., where the latent load is low, 45°F suction will be satisfactory. Therefore, choose a compressor with capacity not less than the total cooling load and rated at 104°F head temperature, and a suction temperature between 40 and 45°F, depending on the nature of the load.

Obtain the brake horsepower of the compressor at these conditions, and make sure that a motor is provided with horsepower not less than that required by the compressor.

A standard NEMA motor can be loaded about 15 to 20% above normal. Do not depend on this safety factor, for it will come in handy during initial pull-down periods, excess occupancy, periods of low-voltage conditions, etc.

Compressor manufacturers have all the above data available for the asking so that one need not guess.

Sealed compressors used in most air-conditioning systems consist of a compressor and a motor coupled together in a single housing. These units are assembled, dehydrated, and sealed at the factory. Purchasers cannot change the capacity of the compressor by changing its speed or connect it to motors with different horsepower. It therefore is necessary to specify required tonnage, suction temperature, and discharge temperature so that the manufacturer can supply the correct, balanced, sealed compressor-motor unit.

13.25 COOLING EQUIPMENT—CENTRAL PLANT PACKAGED UNITS

For an economical installation, the air-handling units should be chosen to handle the least amount of air without danger of sweating ducts and grilles. In most comfort cooling, an 18°F discharge temperature below room temperature will be satisfactory.

If a fresh-air duct is used, the required fresh air is mixed with the return air, and the mix is sent through the cooling coil; i.e., the fresh-air load is taken care of in the coil, and discharge air must take care of the internal load only. Then, from Eq. (13.30), the amount of air to be handled is

$$Q = \frac{q_s}{1.08\ \Delta T} \tag{13.35}$$

where q_s = internal sensible load
ΔT = temperature difference between room and air leaving coil (usually 18°F)

If a fresh-air duct is not installed, and the outside air is allowed to infiltrate into the premises, we use Eq. (13.30) with the sensible part of the load of the infiltration air added to q_s, because the outside air infiltrating becomes part of the internal load.

Once the amount of air to be handled is determined, choose a coil of face area such that the coil-face velocity V would be not much more than 500 ft/min. Some coil manufacturers recommend cooling-coil-face velocities as high as 700 ft/min. But there is danger of moisture from the cooling coil being carried along the air stream at such high velocities.

$$V = \frac{Q}{A_c} \tag{13.36}$$

where Q = air flow, ft^3/min
A_c = coil face area, ft^2
V = air velocity, ft/min

The number of rows of coils can be determined by getting the manufacturer's capacity ratings of the coils for three, four, five, six, etc., rows deep and choosing a coil that can handle not less than the sensible and latent load at the working suction temperature.

Multizone air-handling units (Fig. 13.33) are used to control the temperature of more than one space without use of a separate air-handling unit for each zone. When a zone thermostat calls for cooling, the damper motor for that zone opens the cold deck dampers and closes or throttles the warm deck dampers. Thus, the same unit can provide cooling for one zone while it can provide heating for another zone at the same time.

This style of unit is not typically used in new installations, but may be encountered in existing facilities. Various controllers are available to raise these units closer to efficiencies expected in current designs.

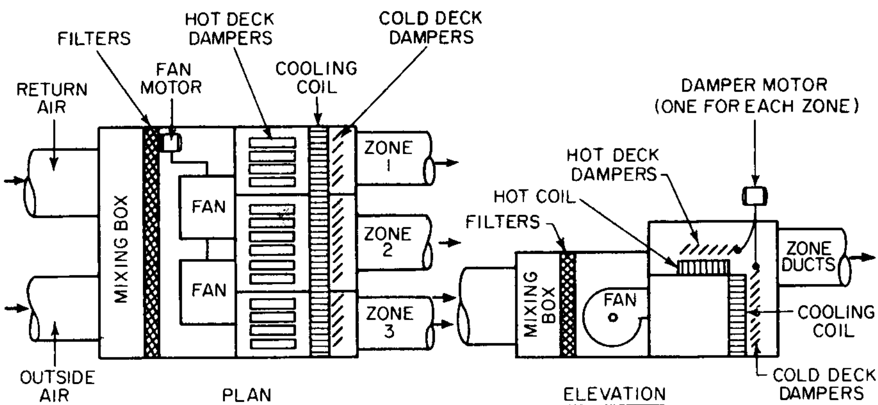

FIGURE 13.33 Multizone air-handling unit.

13.26 ZONING

In large and complex buildings, there will be many spaces that have different inside design conditions. The reason for this is that it is very difficult, in many cases, to provide an air-handling unit that discharges supply air at a given set of conditions to satisfy the various spaces with different design conditions. Common practice is to combine areas with similar design requirements into a zone. Each zone is then served by a separate air-conditioning unit independently of the other zones.

In some cases, a zone may be satisfied by use of reheat coils to satisfy the zone requirements. However, reheating the low-temperature supply air consumes energy. Hence, this practice is in disfavor and in many cases prohibited by building codes and governmental policies. Other formerly well-established systems, such as multizone and dual-duct systems, are also considered energy consumers and are also in disfavor. Exceptions that permit these systems by codes and statutes are for specific types of manufacturing or processing systems, or for areas where the cooled space must be maintained at very specific temperatures and humidities, such as computer rooms, libraries, operating rooms, paper and printing operations, etc. The most common system in use for zone control with central air-handling equipment is the variable-air-volume system. (See Art. 13.31.)

(H. E. Bovay, Jr., "Handbook of Mechanical and Electrical Systems for Buildings," McGraw-Hill Publishing Company, New York.)

13.27 PACKAGED AIR-CONDITIONING UNITS

To meet the great demand for cheaper air-conditioning installations, manufacturers produce packaged, or preassembled, units. These vary from 4000 to 30,000 Btu/hr for window units and 9000 Btu/hr and up for commercial units. Less field labor is required to install them than for custom-designed installations.

Packaged window units operate on the complete cycle shown in Fig. 13.30, but the equipment is very compactly arranged. The condenser is air cooled and projects outside the window. The cooling coil extends inside. Both the cooling-coil fan and condenser fan usually are run by the same motor.

These packaged units require no piping, just an electric receptacle of adequate capacity. Moisture that condenses on the cooling coil runs via gutters to a small sump near the condenser fan. Many manufacturers incorporate a disk slinger to spray this water on the hot condenser coil, which vaporizes it and exhausts it to the outside. This arrangement serves a double purpose:

1. Gets rid of humidity from the room without piping.
2. Helps keep the head pressure down with some evaporative cooling.

Floor-type and ceiling-type packaged units also contain the full air-conditioning cycle (Fig. 13.30). The air-cooled packaged units are usually placed near a window to reduce the duct runs required for the air-cooled condenser.

Roof-type packaged units are available for a variety of applications. These units contain a complete cooling cycle (usually with an air-cooled condenser) and a furnace. All controls are factory prewired, and the refrigeration cycle is completely installed at the factory. Necessary ductwork, wiring, and gas piping are supplied in the field.

A combination packaged unit including an evaporative condenser is also available.

Most packaged units are standardized to handle about 400 ft^3/min of air per ton of refrigeration. This is a good average air quantity for most installations. For restaurants, bars, etc., where high latent loads are encountered, the unit handles more than enough air; also, the sensible capacity is greater than required. However, the latent capacity may be lower than desired. This will result in a somewhat higher relative humidity in the premises.

For high sensible-load jobs, such as homes, offices, etc., where occupancy is relatively low, air quantities are usually too low for the installed tonnage. Latent capacity, on the other hand, may be higher than desired. Thus, if we have a total load of 5 tons—4½ tons sensible and ½ ton latent—and we have available a 5-ton unit with a capacity of 4 tons sensible and 1 ton latent, it is obvious that during extreme weather the unit will not be able to hold the dry-bulb temperature down, but the relative humidity will be well below design value. Under such conditions, these units may be satisfactory. In the cases in which the latent capacity is too low and we have more than enough sensible capacity, we can set the thermostat below design indoor dry-bulb temperature and maintain a lower dry-bulb temperature and higher relative humidity to obtain satisfactory comfort conditions. Where we have insufficient sensible capacity, we have to leave the thermostat at the design setting. This will automatically yield a higher dry-bulb temperature and lower relative humidity—and the premises will be comfortable if the total installed capacity is not less than the total load.

As an example of the considerations involved in selecting packaged air-conditioning equipment, let us consider the structure in Fig. 13.4, p. 13.46, and the load analysis in Table 13.12, p. 13.47.

If this were an old building, it would be necessary to do the following:

First-floor store (load 7.25 tons). Inasmuch as a 5-ton unit is too small, we must choose the next larger size—a 7½-ton packaged unit. This unit has a greater capacity than needed to maintain design conditions. However, many people would like a somewhat lower temperature than 80°F dry bulb, and this unit will be capable of maintaining such conditions. Also, if there are periods when more than 50 occupants will be in the store, extra capacity will be available.

First-floor office (load 0.35 tons). Ordinarily a 4000-Btu/hr window-unit would be required to do the job. But because the store unit has spare capacity, it would be advisable to arrange a duct from the store 7½-ton unit to cool the office.

Second-floor offices (No. 1, load 3.50 tons; No. 2, load 1.13 tons). A 5-ton unit is required for the two offices. But it will be necessary to provide a fresh-air connection to eliminate the fresh-air load from the internal load, to reduce air requirements to the rated flow of the unit. Then, the 2000 ft^3/min rating of the 5-ton packaged unit will be close enough to the 2236 ft^3/min required for the two offices.

Inasmuch as the total tonnage is slightly above that required, we will balance out at a slightly lower relative humidity than 50%, if the particular packaged unit selected is rated at a sensible capacity equal to the total sensible load.

A remote air-conditioning system would be more efficient, because it could be designed to meet the needs of the building more closely. A single air-handling unit for both floors can be arranged with the proper ductwork. However, local ordinances should be checked, since some cities have laws preventing direct-expansion systems servicing more than one floor. A chilled water system (Fig. 13.31) could be used instead.

Consideration for part load performance is very important. Remember that equipment is selected for maximum design conditions that occur only 2–5% of the time. Knowledge of criticality of temperature and relative humidity must be determined. For packaged equipment, slightly undersizing the system may be more desirable to maintain radiant heat control at part load by reducing cycling. Heat wheels and desiccant dehumidification systems are also in widespread use to better maintain conditions and save energy, given code-required increases in ventilation air.

Split Systems. Split systems differ from other packaged air-conditioning equipment in that the noisy components of the system, notably the refrigerant compressor and air-cooled condenser or cooling tower fans, are located outdoors, away from the air-handling unit. This arrangement is preferred for apartment buildings, residences, hospitals, libraries, churches, and other buildings for which quiet operation is required. In these cases, the air-handler is the only component located within the occupied area. It is usually selected for low revolutions per minute to minimize fan noise. For such installations, chilled-water piping, or direct-expansion refrigerating piping must be extended from the cooling coil to the remote compressor and condensing unit, including operating controls.

Packaged Chillers. These units consist of a compressor, water chiller, condenser, and all automatic controls. The contractor has to provide the necessary power and condensing water, from either a cooling tower, city water, or some other source such as well or river. The controls are arranged to cycle the refrigeration compressor to maintain a given chilled-water temperature. The contractor need provide only insulated piping to various chilled-water air-handling units.

Packaged chillers also are available with air-cooled condensers for complete outdoor installation of the unit, or with a condenserless unit for indoor installation connected to an outdoor air-cooled condenser.

The larger packaged chillers, 50 to 1000 tons, are generally powered by centrifugal compressors. When the units are so large that they have to be shipped knocked down, they are usually assembled by the manufacturer's representative on the job site, sealed, pressure-tested, evacuated, dehydrated, and charged with the proper amount of refrigerant.

13.28 ABSORPTION UNITS FOR COOLING

Absorption systems for commercial use are usually arranged as central station chillers. In place of electric power, these units use a source of heat to regenerate the refrigerant. Gas, oil, or low-pressure steam is used.

In the absorption system used for air conditioning, the refrigerant is water. The compressor of the basic refrigeration cycle is replaced by an absorber, pump, and generator. A weak solution of lithium bromide is heated to evaporate the refrigerant (water). The resulting strong solution is cooled and pumped to a chamber where it absorbs the cold refrigerant vapor (water vapor) and becomes dilute.

Condensing water required with absorption equipment is more than that required for electric-driven compressors. Consult the manufacturer in each case for the water quantities and temperature required for the proper operation of this equipment.

The automatic-control system with these packaged absorption units is usually provided by the manufacturer and generally consists of control of the amount of steam, oil, or gas used for regeneration of the refrigerant to maintain the chilled water temperature properly. In some cases, condenser-water flow is varied to control this temperature.

When low-cost steam is available, low-pressure, single-stage absorption systems may be more economical to operate than systems with compressors. In general, steam consumption is about 20 lb/hr per ton of refrigeration. Tower cooling water required is about 3.7 gal/min per ton with 85°F water entering the absorption unit and 102°F water leaving it.

As an alternative, high-pressure steam, two-stage absorption units are available that operate much more efficiently than the low-pressure units. The high-pressure units use about 12 lb of steam per ton of refrigeration at full load. This is a 40% savings in energy.

13.29 DUCTS FOR AIR CONDITIONING

In designing a duct system for air conditioning, we must first determine air-outlet locations. If wall grilles are used, they should be spaced about 10 ft apart to avoid dead spots. Round ceiling outlets should be placed in the center of a zone. Rectangular ceiling outlets are available that blow in either one, two, three, or four directions.

Manufacturers' catalog ratings should be checked for sizing grilles and outlets. These catalogs give the recommended maximum amount of air to be handled by an outlet for the various ceiling heights. They also give grille sizes for various lengths of blows. It is obvious that the farther the blow, the higher must be the velocity of the air leaving the grille. Also the higher the velocity, the higher must be the pressure behind the grille.

When grilles are placed back to back in a duct as in Fig. 13.34, be sure that grille A and grille B have the same throw; for if the pressure in the duct is large enough for the longer blow, the short-blow grille will bounce the air off the opposite wall, causing serious drafts. But if the pressure in the duct is just enough for the short blow, the long-blow grille will never reach the opposite wall. Figure 13.35 is recommended for unequal blows because it allows adjustment of air and buildup of a higher static pressure for the longer blow.

In some modern buildings perforated ceiling panels are used to supply conditioned air to the premises. Supply ductwork is provided in the plenum above the suspended ceiling as with standard ceiling outlets. However, with perforated panels, less acoustical fill is used to match the remainder of the hung ceiling.

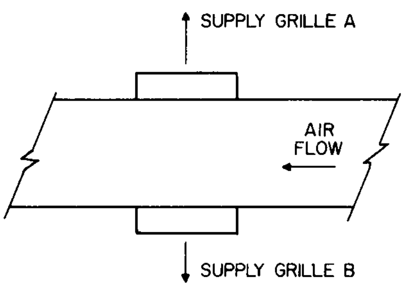

FIGURE 13.34 Grilles placed back to back on an air-conditioning duct with no provision for adjustment of discharge.

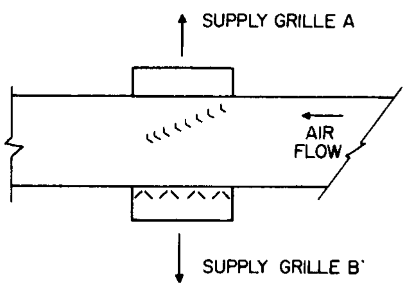

FIGURE 13.35 Vanes placed in a duct to adjust air flow to back-to-back grilles, useful when air throws are unequal.

After all discharge grilles and the air-handling unit are located, it is advisable to make a single-line drawing of the duct run. The air quantities each line and branch must carry should be noted. Of the few methods of duct-system design in use, the equal-friction method is most practical. For most comfort cooling work, it is considered good practice not to exceed 0.15-in friction per 100 ft of ductwork. It is also well to keep the air below 1500 ft/min starting velocity.

If a fresh-air duct is installed the return-air duct should be sized for a quantity of air equal to the supply air minus the fresh air.

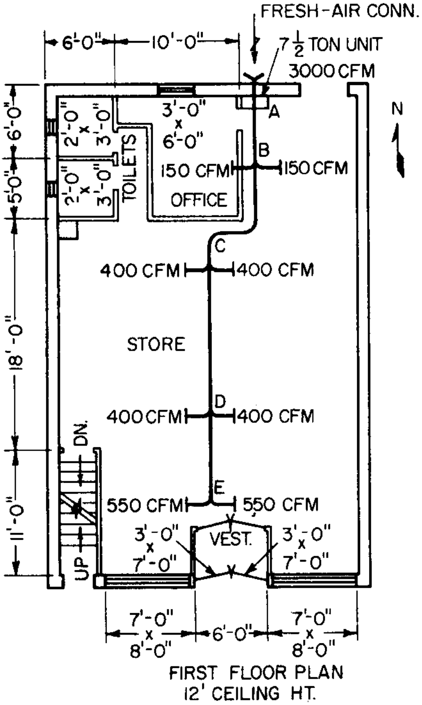

FIGURE 13.36 Ductwork for cooling a store.

It is advisable, where physically possible, to size the fresh-air duct for the full capacity of the air-handling unit. For example, a 10-ton system handling 4000 ft^3/min of supply air—1000 ft^3/min fresh air and 3000 ft^3/min return air—should have the fresh-air duct sized for 4000 ft^3/min of air. A damper in the fresh-air duct will throttle the air to 1000 ft^3/min during the cooling season; however, during an intermediate season, when the outside air is mild enough, cooling may be obtained by operating only the supply-air fan and opening the damper, thus saving the operation of the 10-ton compressor-motor unit.

As an example of the method for sizing an air-conditioning duct system, let us determine the ductwork for the first floor of the building in Fig. 13.36. Although a load analysis shows that the air requirement is 2979 ft^3/min, we must design the ducts to handle the full capacity of air of the packaged unit we supply. Handling less air will unbalance the unit, causing a drop in suction temperature, and may cause freezing up of the coil. If a 7½-ton packaged unit is used, for example, the ducts should have a capacity at least equal to the 3000 ft^3/min at which this unit is rated.

Table 13.17 shows the steps in sizing the ducts. The 3000 ft^3/min is apportioned to the various zones in the store in proportion to the load from each, and the flow for each segment of duct is indicated in the second column of the table. Next, the size of an equivalent round duct to handle each airflow is determined from Table 13.8 with friction equal to 0.15 in per 100 ft. The size of rectangular duct to be used is obtained from Table 13.6.

The preceding example of ductwork design falls into the category of low-pressure duct systems. This type of design is used for most air-distribution systems that are not too extensive, such as one- or two-floor systems, offices, and residences. In general, the starting air velocity is below 2000 ft/min, and the fan static pressure is below 3 in of water.

For large multistory buildings, high-velocity air-distribution duct systems often are used. These systems operate at duct velocities well above 3000 ft/min and

TABLE 13.17 Duct Calculation for 7½-ton Packaged Unit for Store (Fig. 13.37)

Duct	Cfm	Equivalent round duct (friction = 0.15 in per 100 ft, max velocity = 1500 fpm), diam, in	Rectangular duct, in
A-B	3000	19.7	28 × 12
B-C	2700	18.4	24 × 12
C-D	1900	16.1	18 × 12
D-E	1100	13.2	12 × 12

above 3-in static pressure. Obvious advantages include smaller ducts and lower building cost, since smaller plenums are needed above hung ceilings. Disadvantages are high power consumption for fans and need for an air-pressure-reducing valve and sound attenuation box for each air outlet, resulting in higher power consumption for operation of the system.

Some of the more elaborate heating and air-conditioning installations consist of a high-pressure warm-air duct system and a high-pressure cold-air duct system. Each air outlet is mounted in a sound attenuation box with pressure-reducing valves and branches from the warm- and cold-air systems (Fig. 13.37). Room temperature is controlled by a thermostat actuating two motorized volume dampers. When cooling is required, the thermostat activates the motor to shift the warm-air damper to the closed or throttled position and the cold-air damper to the open position.

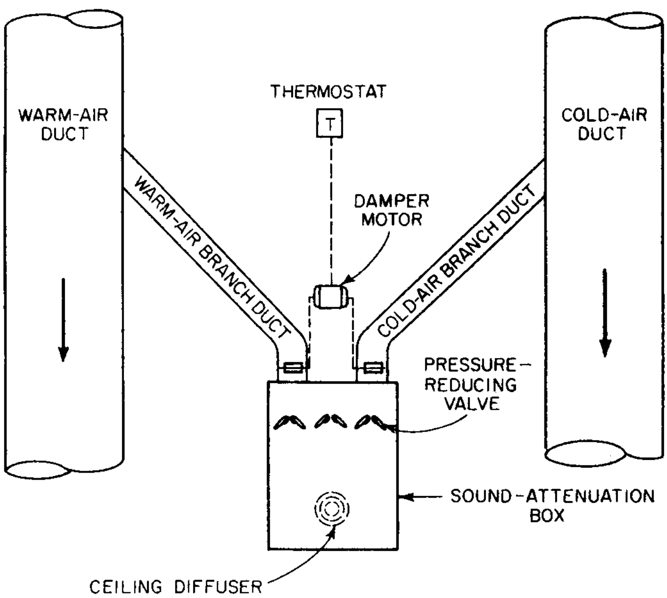

FIGURE 13.37 Double-duct air-distribution system.

13.30 BUILT-UP AIR-CONDITIONING UNITS

Built-up air-conditioning units differ from packaged units in that built-up units are assembled at the site, whereas components of packaged units are preassembled in a factory. Built-up units are usually limited to the larger-size units, with capacity of 50 tons and over. They provide cooling air in summer and heated air in winter. In intermediate seasons, the units may provide 0 to 100% outside air for ventilation, with economy control or enthalpy control. The units come complete with prefilters, final filters, return-air fan, dampers, and controls, as required. The units may be installed outdoors at grade or on rooftop, or indoors as a central-plant unit.

A built-up unit usually is enclosed with sandwich-type, insulated panels, which incorporate thermal insulation between steel or cement-asbestos sheets. Special details have been developed for connections of enclosure panels to provide rigidity and to make them weatherproof or airtight. Access doors should be provided between major in-line components.

Built-up units can be made as a complete system with return-air fan, refrigerant compressors and air-cooled condensers for outside installations. Other types without these components are available for split systems that utilize remote refrigerant compressors and condensing units. In chilled-water systems, chilled water is pumped to the cooling coil from the remote chilled-water unit. Heating water is also piped to the heating coil from a remote heat exchanger but with its own separate pumping system.

13.31 VARIABLE-AIR-VOLUME (VAV) SYSTEMS

The VAV concept as applied to air conditioning is based on the idea of varying the supply-air volume, with constant temperature, to meet changing load conditions of interior spaces or zones (Fig. 13.38). Many installations utilize packaged roof units, although large or small built-up units may also be used.

Rooftop units may be used for the following VAV applications:

1. Interior VAV cooling and perimeter radiation
2. Interior VAV cooling and constant-volume perimeter heating

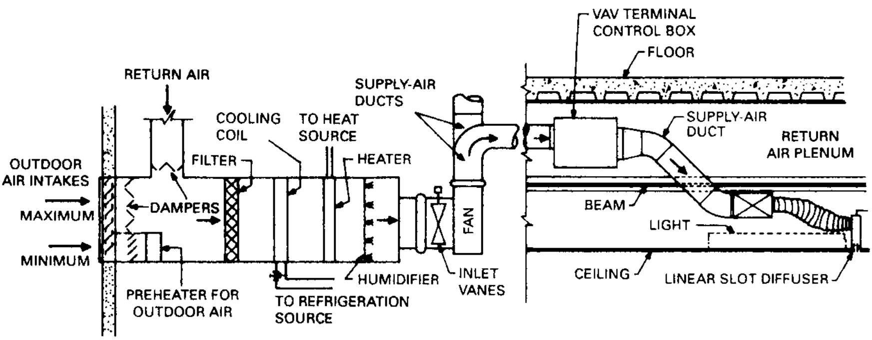

FIGURE 13.38 VAV system varies the supply-air volume to meet changing load conditions.

3. Interior VAV cooling with constant-volume, perimeter-heating, fan-coil units
4. Interior VAV cooling with fan-powered perimeter heating or cooling
5. Interior VAV cooling with perimeter VAV reheat
6. Interior VAV cooling, double-duct, with VAV perimeter cooling and heating

These systems vary widely in advantages and disadvantages and should be analyzed very carefully in the preliminary stages of design for each project.

13.31.1 Zoning of Building Interiors

Zoning is very important in the design stage and will contribute greatly to the success or failure of the HVAC system. An insufficient number of zones will lead to loss of temperature control in areas that do not have the controlling thermostat. The more zones utilized, the better will be the comfort conditions attained, but with some increase in initial costs.

13.31.2 Air Distribution

All types of diffusers (registers) may be used with VAV control units. Standard diffusers with reduced air-flow rates, however, often cause ‘‘dumping’’ of discharge air directly downward with wide variations in space temperature. Many authorities recommend linear-slot-type diffusers to negate dumping. Such diffusers discharge air into the room with a horizontal flow that hugs the ceiling. This is called the *coanda characteristic effect* and is very effective in distributing supply air throughout a room.

13.31.3 VAV Terminal Control Boxes

For producing variable volume other than by modulation of the supply fan, terminal control units of many types may be used:

Shut-off control diffusers are used for interior cooling. They are available with many features, such as shut-off operation, multiple slots, integral slot diffuser, electric-pneumatic or system-powered controls. They are available in 200- to 800-cfm capacities, with choice of aspirating, unit-mounted pneumatic, system powered or electric thermostats. Savings are attributable to modulations of supply-air fans, which also provide complete control flexibility with reduced control costs. These diffusers are easy to install and are compatible with most ceiling systems.

Fan-powered control units are generally used for perimeter and special-use areas. They feature VAV operation in the cooling mode. They are available with pressure-compensating controls with factory-installed hot-water or multistage electric coils that have pneumatic or electric controls. A built-in, side-mounted fan recirculates plenum air. Fan-powered units permit maximum modulation of the central fan as the control unit modulates to full shut-off VAV position. A terminal fan in the control unit is operated to maintain minimum flow in perimeter zones only. Heat generated by lighting fixtures is also used to assist in heating of the perimeter zones. Additional or supplemental heat is provided by unit-mounted hot-water or electric coils. No reheating of conditioned air is utilized.

Dual-duct VAV control units are specifically designed for dual-duct systems that require VAV for perimeter areas. These units feature pressure-compensated shut-off operation with pneumatic, electric, or system-powered controls, with a variable-volume cold deck and constant-volume or variable-volume hot deck. They are available with multiple outlets and factory-calibrated mixing points. Dual-duct control units maximize energy savings with complete control flexibility when fans modulate both heating and cooling air flows. Heating and cooling air can overlap to maintain minimum air flows with superior room-temperature control.

13.31.4 Dual-Duct VAV Control Diffusers

These assemblies consist of the dual-duct VAV control unit and linear-slot diffusers (Fig. 13.39). They are used for VAV dual-duct perimeters and special-use areas. They provide all the features and benefits described previously.

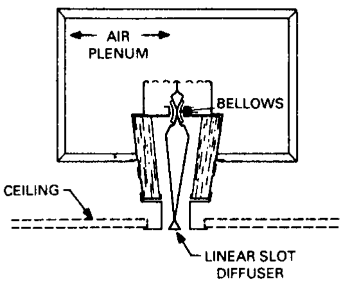

FIGURE 13.39 VAV terminal with linear-slot diffuser.

Linear-slot diffusers are available for all VAV systems. These diffusers maintain the desired coanda effect down to ±5% of air flow. They are available with one- or two-way throws, with up to four slots at 1000 cfm per diffuser. They may operate with aspirating electric, pneumatic or system-powered, unit-mounted thermostats. Factory-installed fire dampers are also available. Benefits include elimination of dumping at reduced flows, ease of relocation, and high capacity at low first cost, as well as reduced control costs with unit-mounted thermostats.

13.31.5 Noise Considerations

Rooftop packaged units used for VAV air conditioning may transmit high levels of noise to occupied spaces. Such noise is generally caused by compressor, condenser fans, or supply-duct exhaust fans. Computerized duct-design programs are available that indicate the sound generation that can be expected at any point in the duct distribution system. Examination of the predicted levels gives designers the opportunity to make design alterations to avoid adverse acoustical conditions.

Location of rooftop units is also important. They should not be installed over offices, conference rooms, or other critical areas. Units preferably should be located over storage rooms, corridors, or utility areas, where higher sound levels would be acceptable.

(H. E. Bovay, Jr., "Handbook of Mechanical and Electrical Systems for Buildings," N. R. Grimm and R. C. Rosaler, "Handbook of HVAC Design," McGraw-Hill Publishing Company, New York; B. Stein et al., "Mechanical and Electrical Equipment for Buildings," John Wiley & Sons, Inc., New York.)

13.32 AIR-WATER SYSTEMS

An alternative to the air-to-air systems described in Art. 13.31, air-water systems furnish chilled water from a remote chiller or central plant to the room terminal devices. These contain a cooling coil or a heating coil, or both. Room temperature is maintained by varying the flow of chilled water or heating fluid through the coils with valves that respond to the thermostat. Ventilation air is provided from a separate central plant directly to the room or the terminal device.

Two-pipe or four-pipe systems are used for distribution of chilled water and hot water to the room terminals from a central plant. In a two-pipe system, the supply pipe may carry either chilled water or hot water and the second pipe is used as a return. The four-pipe system provides two pipes for chilled-water supply and return and two pipes for hot-water supply and return. The installed cost of the two-pipe system is less than that of the four-pipe system, but the versatility is less. The major disadvantage of the two-pipe system is its inability to provide both heating and cooling with a common supply pipe on days for which both heating and cooling are desired. The four-pipe system has a major drawback in loss of temperature control whenever a changeover from cooling to heating is desired. To overcome this, thermostats are used that permit selection of either cooling or heating by a manual changeover at the thermostats.

Terminal devices for air-water systems are usually of the fan-coil or induction types.

Fan-Coil Terminal Units. A fan-coil terminal device consists of a fan or blower section, chilled-water coil, hot-water heating coil or electric-resistance heating elements, filter, return-air connection, and a housing for these components with an opening for ventilation air. The electric-resistance heating coil is often used with two-pipe systems to provide the performance of a four-pipe system without the cost of the two extra pipes for hot water, insulation, pumps, etc. Fan-coil units may be floor mounted, ceiling mounted-exposed or ceiling mounted-recessed, or ceiling mounted-recessed with supply- and return-air ductwork. When furnished with heating coils, the units are usually mounted on the outside wall or under a window, to neutralize the effects of perimeter heat losses.

Built-in centrifugal fans recirculate room air through the cooling coil. Chilled water circulating through the coil absorbs the room heat load. Ventilation air that is conditioned by another remote central plant is ducted throughout the building and supplied directly to the room or room terminal devices, such as a fan-coil unit. A room thermostat varies the amount of cooling water passing through the cooling coil, thus varying the discharge temperature from the terminal unit and satisfying the room thermostat.

Induction Terminal Units. These units are frequently used in large office buildings. The units are served by a remote air-handling unit that provides high-pressure conditioned air, which may be heated or cooled and is referred to as primary air. It is distributed to individual induction units that are located on the outside walls of each room or zone. At the terminal induction unit, a flow of high-pressure primarily air through several nozzles induces a flow of room air through the heating or cooling coil in the unit. Room air mixes with the primary air to provide a mixed-air temperature that satisfies the thermal requirements of the space. In most systems, the ratio of induced air to primary air is about 4 to 1.

The induction system is a large energy consumer because of the extra power required to maintain the high pressure necessary to deliver the primary air to the room induction nozzles and induce room air to flow through the unit coils. Also, the induction terminal unit operates simultaneously with heating and cooling, wasting energy as in a terminal-reheat type of operation.

Air-water systems generally have substantially lower installed and operating costs than all-air systems. They do not, however, provide as good control over room temperature, humidity, air quality, air movement, and noise. The best control of an air-water system is achieved with a fan-coil unit with supplemental ventilation air from a central, primary-air system that provides ventilation air.

(H. E. Bovay, Jr., "Handbook of Mechanical and Electrical Systems for Buildings," and N. R. Grimm and R. C. Rosaler, "Handbook of HVAC Design," McGraw-Hill Publishing Company, New York.)

13.33 CONTROL SYSTEMS FOR AIR CONDITIONING

One commonly used system has a thermostat wired in series with the compressor holding-coil circuit (Fig. 13.40). Thus the compressor will stop and start as called for by room conditions. The high-low pressure switch in series with the thermostat is a safety device that stops the compressor when the head pressure is too high and when the suction pressure approaches the freezing temperature of the coil. A liquid solenoid will shut off the flow of refrigerant when the compressor stops, to prevent flooding the coil back to the compressor during the off cycle. This valve may be eliminated when the air-handling unit and compressor are close together.

A second type of control is the pump-down system (Fig. 13.41). The thermostat shuts off the flow of the refrigerant, but the compressor will keep running. With the refrigerant supply cut off, the back pressure drops after all the liquid in the coil vaporizes. Then, the high-low pressure switch cuts off the compressor.

Either of these two systems is satisfactory. However, the remainder of this discussion will be restricted to the pump-down system.

Additional safety controls are provided on packaged units to reduce compressor burnouts and increase the average life of these units. The controls include crankcase

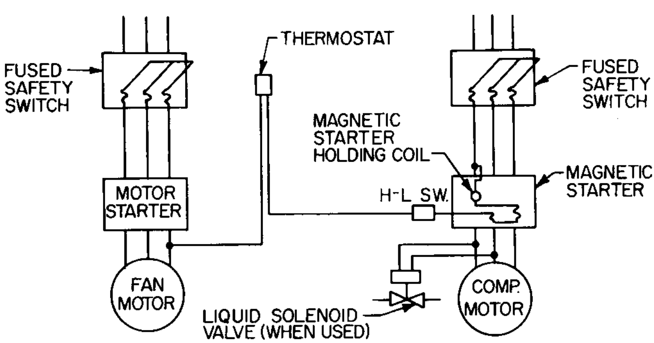

FIGURE 13.40 Control circuit for an air-conditioning system in which the thermostat controls compressor operation.

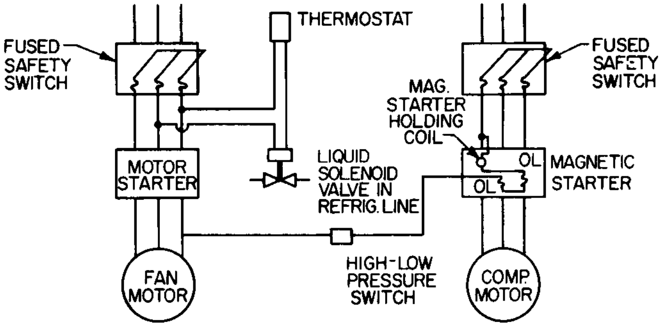

FIGURE 13.41 Pump-down system for control of air conditioning.

heaters, motor-winding thermostats, and nonrecycling timers. These controls are usually prewired in the factory. The manufacturer supplies installation and wiring instructions for interconnecting the various components of the air-conditioning system.

Where city water is used for condensing purposes, an automatic water-regulator valve is supplied (usually by the manufacturer). Figure 13.42 shows a cross section of such a valve. The power element is attached to the hot-gas discharge of the compressor. As the head pressure builds up, the valve is opened more, allowing a greater flow of water to the condenser, thus condensing the refrigerant at a greater rate. When the compressor is shut off, the head pressure drops, the flow of water being cut off by the action of a power spring working in opposition of the power element. As the water temperature varies, the valve responds to the resulting head pressure and adjusts the flow automatically to maintain design head pressure.

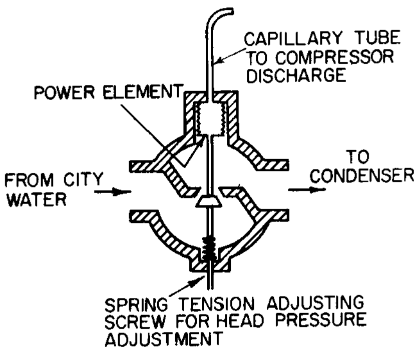

FIGURE 13.42 Water-regulating valve for condensing in an air-conditioning system.

When a water tower is used, the automatic water-regulating valve should be removed from the circuit, because it offers too much resistance to the flow of water. It is not usually necessary to regulate the flow of water in a cooling-tower system. For when the outside wet-bulb temperature is so low that the tower yields too low a water temperature, then air conditioning generally is not needed.

Figure 13.43 shows a simplified wiring diagram for a tower system. Because of the interconnecting wiring between the magnetic starters, the tower fan cannot run unless the air-conditioning fan is in operation. Also, the circulating pump cannot run unless the tower fan is in operation, and the compressor cannot run unless the circulating pump is in operation.

When the system is started, the air-conditioning fan should be operated first, then the tower via switch No. 1, the pump via switch No. 2, and the compressor via switch No. 3. When shutting down, switch No. 3 should be snapped off, then switch No. 2, then switch No. 1, and then the air-conditioning fan.

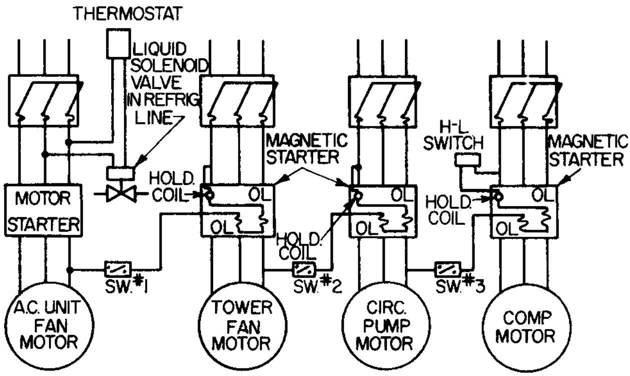

FIGURE 13.43 Control circuit for a cooling-tower-type HVAC system.

In buildings where one tower and pump are used to provide water for many units, pressure switches may be used in series with the holding coil of each compressor motor starter. No interconnecting wiring is necessary, for the compressor will not be able to run unless the pump is in operation and provides the necessary pressure to close the contacts on the pressure switch.

Some city ordinances require that indirect-expansion, or chilled-water, systems be installed in public buildings, such as theaters, night clubs, hotels, and depots. This type of system is illustrated in Fig. 13.30. The refrigerant is used to cool water and the water is circulated through the cooling coil to cool the air. The water temperature should be between 40 and 45°F, depending on whether the occupancy is a high latent load or a low latent load. Because the cost of equipment is increased and the capacity decreased as water temperature is lowered, most designers use 45°F water as a basis for design and use a much deeper cooling coil for high-latent-load installations.

The amount of chilled water in gallons per minute to be circulated may be obtained from

$$Q = \frac{24 \times \text{tons of refrigeration}}{\Delta T} \tag{13.37}$$

where ΔT is the temperature rise of water on passing through the cooling coil, usually 8 or 10°F. The smaller the temperature change, the more water to be circulated and the greater the pumping cost, but the better will be the heat transfer through the water chiller and cooling coil.

Coil manufacturers' catalogs usually give the procedure for picking the number of rows of coil necessary to do the required cooling with the water temperatures available.

The water chiller should be sized to cool the required flow of water from the temperature leaving the cooling coil to that required by the coil at a given suction temperature.

Assuming 45°F water supplied to the cooling coil and a temperature rise of 8°F, the water leaving the coil would be at 53°F. The chiller would then be picked to cool the required flow from 53 to 45°F at 37°F suction. The lower the suction

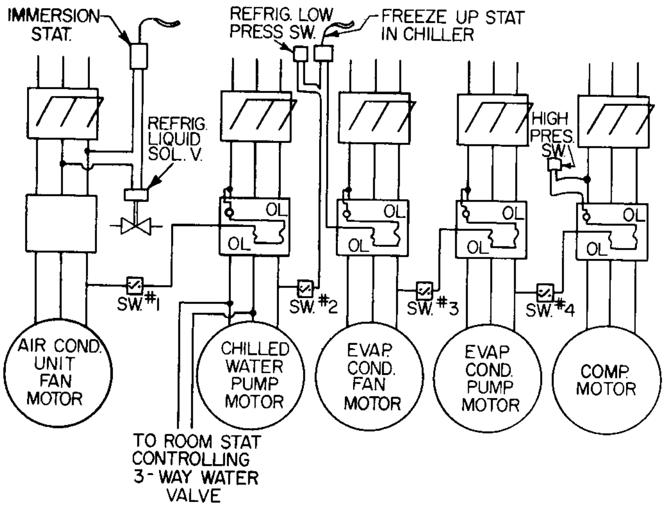

FIGURE 13.44 Control circuit for a chilled-water system with evaporative condenser.

temperature, the smaller will be the amount of heat-transfer surface required in the chiller. Also, the lower will be the compressor capacity. In no case should the suction temperature be less than 35°F, since the freezing point of water—32°F—is too close. A frozen and cracked chiller is expensive to replace.

A simplified control system for a chilled-water system with an evaporative condenser is shown in Fig. 13.44.

13.34 HEATING AND AIR CONDITIONING

Most manufacturers of air-conditioning equipment allow space in the air-handling compartment for the installation of a humidification unit and a heating coil for hot water or steam. These make it possible to humidify and heat the air in cold weather. Before a decision is reached to heat through the air-conditioning duct system, it is important to consider the many pitfalls present:

When a unit provides air conditioning in a single room, a heating coil may do the job readily, provided the Btu-per-hour rating of the coil is equal to or greater than the heating load. The fact that the supply grilles are high is usually no disadvantage; since winter heating is designed for the same amount of air as summer cooling, a small temperature rise results, and the large air-change capacity of the air-handling unit creates enough mixing to prevent serious stratification. Difficulties usually are encountered, however, in a structure with both exposed and interior zones. In the winter, the exposed zones need heating, while the interior zones are warm. If the heating thermostat is located in the exposed zone, the interior zones will become overheated. If the thermostat is located in the interior zone, the exposed zones will be too cold. Where some heat is generated in the interior zones, the system may require cooling of the interior and heating of the exterior zones at the same time. Thus, it is impossible to do a heating and cooling job with a single system in such structures.

Where heating and cooling are to be done with the same duct system, the air-handling equipment should be arranged to service individual zones—one or more units for the exposed zones, and one or more units for the interior zones.

When a heating system is already present and an air-conditioning system is added, a heating coil may be used to temper the outside air to room temperature. A duct-type thermostat may be placed in the discharge of the unit to control the steam or hot-water valve. When a room thermostat is used, the spare coil capacity may be used for quick morning pickup. Later in the day, the system may be used for cooling the premises with outside air if the building-heating system or other internal heat sources overheat the premises. In buildings with large window areas, it is advisable to place under the windows for use in cold weather some radiation to supply heat in addition to that from the heating coil, to counteract the down draft from the cold glass surfaces.

The heating-coil size should be such that its face area is about equal to the cooling-coil face area. The number of rows deep should be checked with manufacturers' ratings. When the unit is to be used for tempering only, the coil need be sized for the fresh-air load only [Eq. (13.30)].

When large quantities of outside air are used, it is usual practice to install a preheat coil in the fresh-air duct and a reheat coil in the air-conditioning unit. Install the necessary filters before the preheat coil to prevent clogging.

13.35 CONTROL OF COMPUTERIZED HVAC SYSTEMS

Control of HVAC systems ranges from simple thermostat "on-off" control to control by sophisticated electronic and computerized systems. Many variations of electronic and computerized systems are available for system control and energy management.

Programmable Thermostats. These are available for control of the total environment. They provide energy savings and improved comfort levels and are widely utilized to control many types of commercial unitary products. Different programmable types are available for heat pumps and for single- and two-stage heating and cooling equipment.

Programmable thermostats conserve energy by automatically raising or lowering the space temperature to preprogrammed settings several times each day. Building temperature adjustment is performed during occupied or unoccupied hours, again conserving energy.

Other types of thermostats are available that produce energy savings:

- Reduction of overshoot on recovery from the setback temperature, limiting the rate of temperature rise
- Selective recovery from the setback temperature by utilizing lower stages (single-stage and multistage only)
- Selective recovery from the setback temperature using a heat pump in lieu of an auxiliary heat source (heat-pump thermostat)

- Prevention of room temperature from deviating from a setpoint under varying load conditions

Microprocessor-Based Systems. Computerized energy-management systems are available to control complex HVAC systems. These microprocessor-based systems are capable of saving enough energy that they can pay for themselves in 2 years or less. Energy management systems are available in many variations and with many features. The most common include the following capabilities: equipment scheduling, duty cycling of equipment, temperature compensated control, demand limiting, optimum start/stop, and night setback.

Equipment Scheduling. Running HVAC equipment on a continuous basis, including periods when operation is not required, results in excessive and nonproductive energy consumption. These periods include lunch hours, evenings, weekends, and holidays. By using a scheduling program, the control panel promptly shuts down HVAC units and other units at preset times. A typical scheduling program may provide four start/stop times for each load on an 8-day cycle. It will also allow automatically for holidays, leap year, and daylight savings time.

Duty Cycling. Duty cycling applied to HVAC equipment utilizes an on-off control to reduce operating time and increase energy savings. The duty-cycling program will permit up to eight cycle patterns for each load. On-off protection timers are provided for each piece of equipment to assure equipment safety.

Temperature-Compensated Control. Maximum occupant comfort is ensured by use of a computerized control that provides equipment cycling with automatic temperature override. Cycling will be reduced or stopped, depending on whether room temperatures fall outside present comfort levels. Hence, the guesswork of selecting proper cycling strategies is eliminated.

Demand Limiting. As much as 50% of bills for use of electrical energy may be attributed to kW demand charges. A demand-limiting program may save a substantial amount of utility energy costs over a short period of time. A computerized control, in particular, can limit costly demand charges by use of a predictive demand program that anticipates electrical demand peaks. When the control panel senses that a peak is about to occur, it simply turns off selected loads on a priority basis until the peak subsides, thereby reducing the energy bill.

Optimum Start/Stop. Energy can be wasted by HVAC equipment that is scheduled to start and stop on worst-case conditions. This scheduling results in operations that begin too soon and run too long. Computerized control monitors indoor and outside temperatures and determines the most efficient start and stop times without sacrificing individual comfort. Optimum start/stop also allows independent control of individual comfort zones.

Night Setback. Utilization of night setback temperatures saves energy by automatically converting the specific comfort zone to a predetermined nighttime-temperature setpoint. The setpoint overrides the daytime setting for every load or zone. HVAC equipment will operate to maintain the nighttime setting until morning warm-up or cool-down is required. Night setback also allows independent control of each zone.

13.36 DIRECT DIGITAL CONTROL

Heating, ventilation, and air conditioning has usually been controlled by three basic systems: pneumatic, electric, electronic, but another technology, known as direct digital control (DDC), can lower costs and improve comfort in a vast range of complex applications. DDC is an electronic system that uses a digital microprocessor (computer) to analyze various parameters in HVAC, such as temperatures, pressures, air flows, time of day, and many other types of data. The information obtained is used to operate control devices, such as dampers and valves, and to start/stop mechanical equipment such as air-handling units and heating and cooling pumps, in accordance with instructions programmed into the microprocessors' memory.

DDC also offers features in addition to the basic functions, such as monitoring and alarm, data gathering, and energy monitoring and management. DDC systems also accommodate lighting-control and security-system functions and management. They can delegate all of the control logic, including determination of necessary output signals, to be transmitted to the control devices and to the microprocessor. Development of lower-cost chips (computers on a chip) may reduce limitations on the number of devices a single microprocessor can handle.

DDC technology tends toward distributed control, which permits multiple controllers to be used, each with its own microprocessor. This allows each controller to operate independently; after it has been programmed, it can operate independently. A twisted-wire pair is usually used as a data link for connecting the controllers together. This permits exchange of information with each other or with a host computer and, in many cases, with desktop personal computers (PCs).

The host computer is connected to the distributed controllers and is considered as the operator's link. It permits the user to exchange information on the status of any portion of the HVAC system and to revise control schedules, values, or setpoints. When the host computer is not being used for these tasks, it can be used with PCs for usual or conventional tasks, while the controllers continue to operate with their most current set of instructions.

Capacity of a controller is usually measured by its point capacity. Input or output points may be either analog or digital classifications. Analog inputs are those with values that change with time, such as temperatures and pressure. Digital outputs are those that describe a condition that may be of two types only such as on and off.

The advantages of DDC compared with pneumatic control systems are numerous. The greatest advantage is the ability to achieve impressive energy conservation without sacrificing comfort. DDC permits a system to be controlled in an optimum manner while maintaining space temperatures within desired limits. Also, DDC is an extremely flexible control system that permits an operator to control the parameters and control sequence to maintain maximum efficiency. On large installations with many buildings, air-handling units may be reprogrammed for readjustment of parameters for improved comfort conditions or optimal energy conservation, or both. These requirements are simpler to meet with a DDC system than with a pneumatic system.

An analog output signal is used for modulating dampers or varying flow quantities through a liquid-control valve. To start and stop electrical equipment, a digital output is used and is usually a dry type of contact.

As an alternative, a "mix" of DDC and pneumatic actuators may be used on the controlled devices, but electronically operated. In some cases, use of electronic actuation may be economical, particularly for larger valves and dampers. When a pneumatic output signal is required from the controller, a simple digital-to-pneumatic relay converts the electronic signal to a variable pressure signal. Pneumatic actuators provide the additional advantages of excellent reliability and low maintenance cost.

13.37 INDUSTRIAL AIR CONDITIONING

Certain manufacturing processes call for close control of temperature and relative humidity. A typical control system is shown in Fig. 13.45. The humidistat may be of the single-pole, double-throw type.

On a rise in room humidity, the refrigeration compressor cuts in for the purpose of dehumidification. Since the sensible capacity of the cooling coil usually is much higher than the sensible load, whereas the latent capacity does not differ too much from the latent load, undercooling results. The room thermostat will then send enough steam or hot water to the reheat coil to maintain proper room temperature.

A small drop in room relative humidity will cut off the compressor. A further drop in room relative humidity will energize the water or steam solenoid valve to add moisture to the air. The thermostat is also arranged to cut off all steam from the reheat coil when the room temperature reaches the thermostat setting. However, there may be periods when the humidistat will not call for the operation of the refrigeration compressor and the room temperature will climb above the thermostat setting. An arrangement is provided for the thermostat to cut in the compressor to counteract the increase in room temperature.

When calculating the load for an industrial system, designers should allow for the thermal capacity and moisture content of the manufactured product entering and leaving the room. Often, this part of the load is a considerable percentage of the total.

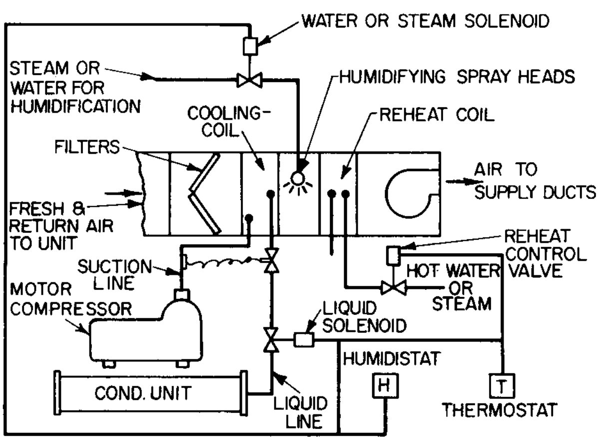

FIGURE 13.45 Circuit for close control of temperature and humidity.

13.38 CHEMICAL COOLING

When the dew point of the conditions to be maintained is 45°F or less, the method of controlling temperature and humidity described in Art. 13.37 cannot be used because the coil suction temperature must be below freezing and the coil will freeze up, preventing flow of air. For room conditions with dew point below 45°F, chemical methods of moisture removal, such as silica gel or lithium chloride, may be used. This equipment is arranged to have the air pass through part of the chemical and thus give up its moisture, while another part of the system regenerates the chemical by driving off the moisture previously absorbed. The psychrometric process involved in this type of moisture removal is adiabatic or, for practical purposes, may be considered as being at constant wet-bulb temperature. For example, if we take room air at 70°F and 30% relative humidity and pass the air through a chemical moisture absorber, the air leaving the absorber will be at a wet-bulb temperature of 53°F; i.e., the conditions leaving may be 75°F and 19% RH or 80°F and 11% RH, etc.

It may be noticed that the latent load here is converted to sensible load, and a refrigeration system will be necessary to do sensible cooling. See Fig. 13.46 for the control of such a system.

13.39 YEAR-ROUND AIR CONDITIONING

Electronic-computer rooms and certain manufacturing processes require temperature and humidity to be closely controlled throughout the year. Air-conditioning systems for this purpose often need additional controls for the condensing equipment.

When city water is used for condensing and the condensers are equipped with a water regulating valve, automatic throttling of the water by the valve automatically compensates for low water temperatures in winter. When cooling towers or evap-

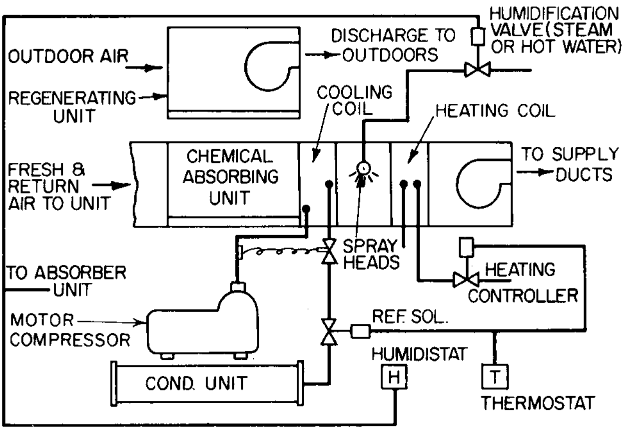

FIGURE 13.46 Controls for cooling with a chemical moisture absorber.

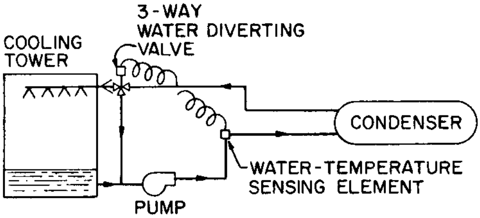

FIGURE 13.47 Piping layout for control of temperature of water from a cooling tower.

orative condensers are used, low wet-bulb temperatures of outside air in winter will result in too low a head pressure for proper unit operation. A controller may be used to recirculate some of the humid discharge air back to the evaporative-condenser suction to maintain proper head pressure. With cooling towers, the water temperature is automatically controlled by varying the amount of air across the towers, or by recirculating some of the return condenser water to the supply. Figure 13.47 shows the piping arrangement.

When air-cooled condensers are used in cold weather, the head pressure may be controlled in one of the following ways: for a single-fan condenser, by varying the fan speed; for a multifan condenser, by cycling the fans or by a combination of cycling and speed variation of fans for closer head-pressure control.

Some head-pressure control systems are arranged to flood refrigerant back to the condenser coil when the head pressure drops. Thus, the amount of heat-transfer surface is reduced, and the head pressure is prevented from dropping too low.