CHAPTER 64
INDOOR ENVIRONMENTAL CONTROL

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64.1 MOIST AIR PROPERTIES AND CONDITIONING PROCESSES

64.1.1 Properties of Moist Air

Atmospheric air is a mixture of many gases plus water vapor and countless pollutants. Aside from the pollutants, which may vary considerably from place to place, the composition of the dry air alone is relatively constant, varying slightly with time, location, and altitude. In 1949 a standard composition of dry air was fixed by the International Joint Committee on Psychrometric Data, as shown in Table 64.1.[1]

Table 64.1 Composition of Dry Air[7]

Constituent	Molecular Mass	Volume Fraction
Oxygen	32.000	0.2095
Nitrogen	28.016	0.7809
Argon	39.944	0.0093
Carbon dioxide	44.010	0.0003

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The molecular mass M of dry air is 28.965, and the gas constant R is 53.353 ft · lbf/lbm · R or 287 J/kg · K.

The basic medium in air-conditioning practice is a mixture of dry air and water vapor. The amount of water vapor may vary from zero to a maximum determined by the temperature and pressure of the mixture. The latter case is called saturated air, a state of neutral equilibrium between the moist air and the liquid or solid phases of water.

Moist air up to about 3 atm pressure obeys the perfect gas law with sufficient accuracy for engineering calculations. The Gibb's–Dalton law for a mixture of perfect gases states that the mixture pressure is equal to the sum of the partial pressures of the constituents. Because the various constituents of the dry air may be considered to be one gas, it follows that the total pressure P of moist air is the sum of the partial pressures of the dry air p_a and the water vapor p_v:

$$P = p_a + p_v$$

Humidity ratio W (sometimes called the specific humidity) is the ratio of the mass of the water vapor m_v to the mass of the dry air m_a in the mixture:

$$W = \frac{m_v}{m_a}$$

Relative humidity ϕ is the ratio of the mole fraction of the water vapor x_v in a mixture to the mole fraction x_s of the water vapor in a saturated mixture at the same temperature and pressure:

$$\phi = \left(\frac{x_v}{x_s}\right)_{t,P}$$

For a mixture of perfect gases the mole fraction is equal to the partial pressure ratio of each constituent. The mole fraction of the water vapor is

$$x_v = \frac{p_v}{P}$$

Thus

$$\phi = \frac{p_v/P}{p_s/P} = \frac{p_v}{p_s}$$

Dew point temperature t_d is the temperature of saturated moist air at the same pressure and humidity ratio as the given mixture. It can be shown that

$$\phi = \frac{Wp_a}{0.6219\, p_s}$$

where p_s is the saturation pressure of the water vapor at the mixture temperature.

The enthalpy i of a mixture of perfect gases is equal to the sum of the enthalpies of each constituent and is usually referenced to a unit mass of dry air:

$$i = i_a + Wi_v$$

Each term has the units of energy per unit mass of dry air. With the assumption of perfect-gas behavior the enthalpy is a function of temperature only. If zero Fahrenheit or Celsius is selected as the reference state where the enthalpy of dry air is zero, and if the specific heats c_{pa} and c_{pv} are assumed to be constant, simple relations result:

$$i_a = c_{pa}t$$
$$i_v = i_g + c_{pv}t$$

where the enthalpy of saturated water vapor i_g at 0°F is 1061.2 Btu/lbm and 2501.3 kJ/kg at 0°C.

64.1.2 The Psychrometric Chart

At a given pressure and temperature of an air–water vapor mixture one additional property is required to completely specify the state, except at saturation.

A practical device used to determine the third property is the psychrometer. This apparatus consists of two thermometers, or other temperature-sensing elements, one of which has a wetted cotton wick covering the bulb. The temperatures indicated by the psychrometer are called the wet bulb and the dry bulb temperatures. The wet bulb temperature is the additional property needed to determine the state of moist air.

To facilitate engineering computations, a graphical representation of the properties of moist air has been developed and is known as a psychrometric chart, Fig. 64.1.[2]

In Fig. 64.1 dry bulb temperature is plotted along the horizontal axis in degrees Fahrenheit or Celsius. The dry bulb temperature lines are straight but not exactly parallel and incline slightly to the left. Humidity ratio is plotted along the vertical axis on the right-hand side of the chart in lbm_v/lbm_a or kg_v/kg_a. The scale is uniform with horizontal lines. The saturation curve with values of the wet bulb temperature curves upward from left to right. Dry bulb, wet bulb, and dew point temperatures all coincide on the saturation curve. Relative humidity lines with a shape similar to the saturation curve appear at regular intervals. The enthalpy scale is drawn obliquely on the left of the chart with parallel enthalpy lines inclined downward to the right. Although the wet bulb temperature lines appear to coincide with the enthalpy lines, they diverge gradually in the body of the chart and are not parallel to one another. The spacing of the wet bulb lines is not uniform. Specific volume lines appear inclined from the upper left to the lower right and are not parallel. A protractor with two scales appears at the upper left of the chart. One scale gives the sensible heat ratio and the other the ratio of enthalpy difference to humidity ratio difference. The enthalpy, specific volume, and humidity ratio scales are all based on a unit mass of dry air.

64.1.3 Space Conditioning Processes

When air is heated or cooled without the loss or gain of moisture, the process is a straight horizontal line on the psychrometric chart because the humidity ratio is constant. Such processes can occur when moist air flows through a heat exchanger. In cooling, if the surface temperature is below the dew point temperature of the moist air, dehumidification will occur. This process will be considered later. Figure 64.2 shows a schematic of a device used to heat or cool air. Under steady-flow–steady-state conditions the energy balance becomes

$$\dot{m}_a i_2 + \dot{q} = \dot{m}_a i_1$$

The direction of the heat transfer is implied by the terms heating and cooling, and i_1 and i_2 may be obtained from the psychrometric chart. The convenience of the chart is evident. Figure 64.3 shows heating and cooling processes. The relative humidity decreases when the moist air is heated. The reverse process of cooling results in an increase in relative humidity.

When moist air is cooled to a temperature below its dew point, some of the water vapor will condense and leave the air stream. Figure 64.4 shows a schematic of a cooling and dehumidifying device and Fig. 64.5 shows the process on the psychrometric chart. Although the actual process path will vary considerably depending on the type surface, surface temperature, and flow conditions, the heat and mass transfer can be expressed in terms of the initial and final states. The total amount of heat transfer from the moist air is

$$\dot{q} = \dot{m}_a(i_1 - i_2) - \dot{m}_a(W_1 - W_2)i_w$$

The last term on the right-hand side is usually small compared to the others and is often neglected.

The cooling and dehumidifying process involves both sensible heat transfer, associated with the decrease in dry bulb temperature, and latent heat transfer, associated with the decrease in humidity ratio. We may also express the latent heat transfer as

$$\dot{q}_l = \dot{m}_a(i_1 - i_a)$$

and the sensible heat transfer is given by

$$\dot{q}_s = \dot{m}_a(i_a - i_2)$$

The energy of the condensate has been neglected. Obviously

$$\dot{q} = \dot{q}_s + \dot{q}_l$$

The sensible heat factor (SHF) is defined as $\dot{q}_s/\dot{q}$. This parameter is shown on the semicircular scale of Fig. 64.1.

A device to heat and humidify moist air is shown schematically in Fig. 64.6. An energy balance on the device and a mass balance on the water yields

$$\frac{i_2 - i_1}{W_2 - W_1} = \frac{\dot{q}}{\dot{m}_w} + i_w$$

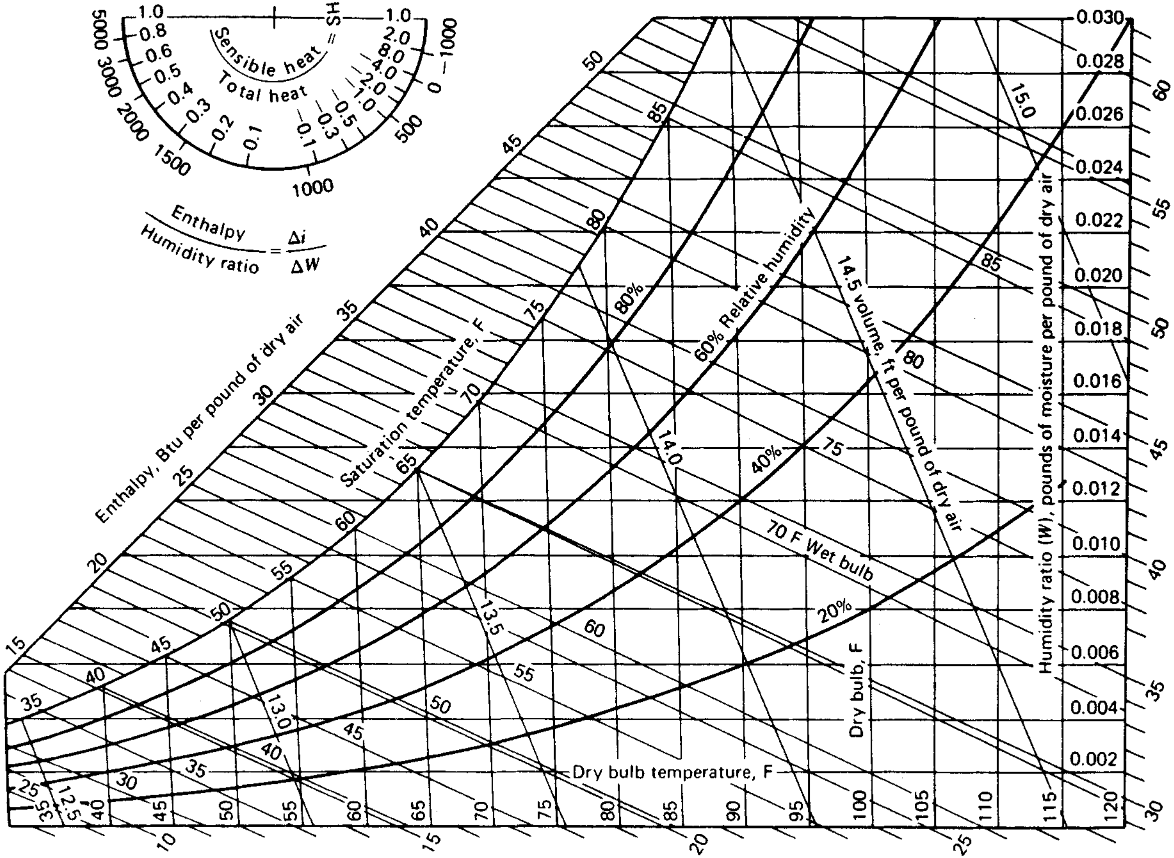

Fig. 64.1 Abridgment of ASHRAE psychrometric chart. (Reprinted by permission from *ASHRAE*.)

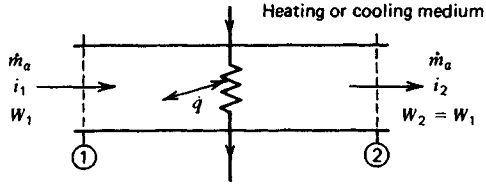

Fig. 64.2 Schematic of a heating or cooling device.[7]

This gives the direction of a straight line that connects the initial and final states on the psychrometric chart. Figure 64.7 shows a typical combined heating and humidifying process.

A graphical procedure makes use of the circular scale in Fig. 64.1 to solve for state 2. The ratio of enthalpy to humidity ratio $\Delta i/\Delta w$ is defined as

$$\frac{\Delta i}{\Delta W} = \frac{i_2 - i_1}{W_2 - W_1} = \frac{\dot{q}}{\dot{m}_w} + i_w$$

Figure 64.7 shows the procedure where a straight line is laid out parallel to the line on the protractor through state point 1. The intersection of this line with the computed value of w_2 determines the final state.

Moisture is frequently added without the addition of heat. In such cases, $q = 0$ and

$$\frac{\Delta i}{\Delta W} = \frac{i_2 - i_1}{W_2 - W_1} = i_w$$

The direction of the process on the psychrometric chart can therefore vary considerably. If the injected water is saturated vapor at the dry bulb temperature, the process will proceed at a constant dry bulb temperature. If the water enthalpy is greater than saturation, the air will be cooled and humidified. Figure 64.8 shows these processes. When liquid water at the wet bulb temperature is injected, the process follows a line of constant wet bulb temperature.

The mixing of air streams is quite common in air-condition systems, usually under adiabatic conditions and with steady flow. Figure 64.9 illustrates the mixing of two air streams. Combined energy and mass balances give

$$\frac{i_2 - i_3}{i_3 - i_1} = \frac{W_2 - W_3}{W_3 - W_1} = \frac{\dot{m}_{a1}}{\dot{m}_{a2}}$$

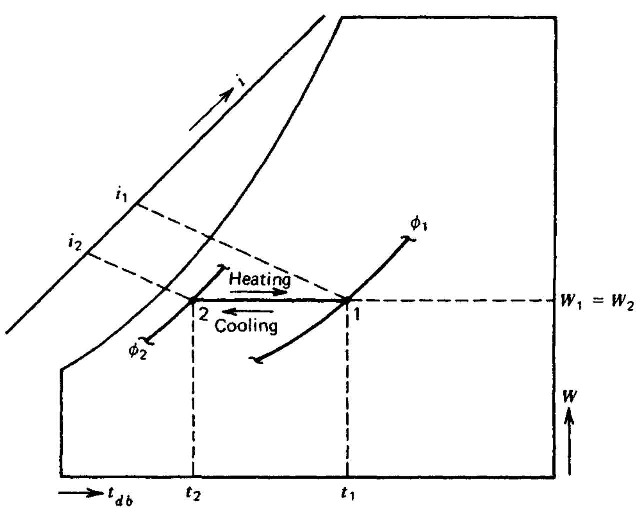

Fig. 64.3 Sensible heating and cooling process.[7]

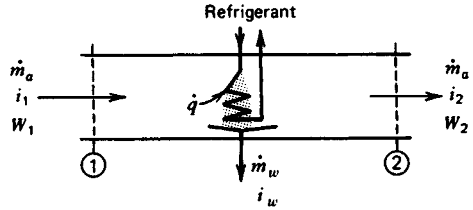

Fig. 64.4 Schematic of a cooling and dehumidifying device.[7]

This shows that the state of the mixed streams must lie on a straight line between states 1 and 2. This is shown in Fig. 64.10. The length of the various line segments are proportional to the masses of dry air mixed. This fact provides a very convenient graphical procedure for solving mixing problems.

The complete air-conditioning system may involve two or more of the processes just considered. In the air conditioning of a space during the summer the air supplied must have a sufficiently low temperature and moisture content to absorb the total heat gain of the space. Therefore, as the air flows through the space, it is heated and humidified. If the system is a closed loop, the air is then returned to the conditioning equipment where it is cooled and dehumidified and supplied to the space again. If fresh air is required in the space, outdoor air may be mixed with the return air before it goes to the cooling and dehumidifying equipment. During the winter months the same general processes occur but in reverse. During the summer months the heating and humidifying elements are inactive, and during the winter the cooling and dehumidifying coil is inactive. With appropriate controls, however, all of the elements may be continuously active to maintain precise conditions in the space.

The previous section treated the common-space air-conditioning problem assuming that the system was operating steadily at the design condition. Actually the space requires only a part of the designed capacity of the conditioning equipment most of the time. A control system functions to match the required cooling or heating of the space to the conditioning equipment by varying one or more system parameters. For example, the quantity of air circulated through the coil and to the space may be varied in proportion to the space load. This approach is known as variable air volume (VAV). Another approach is to circulate a constant amount of air to the space, but some of the return air is diverted around the coil and mixed with air coming off the coil to obtain a supply air temperature that is proportional to the space load. This is known as face and bypass control, because face and bypass dampers are used to divert the flow. Another possibility is to vary the coil surface temperature

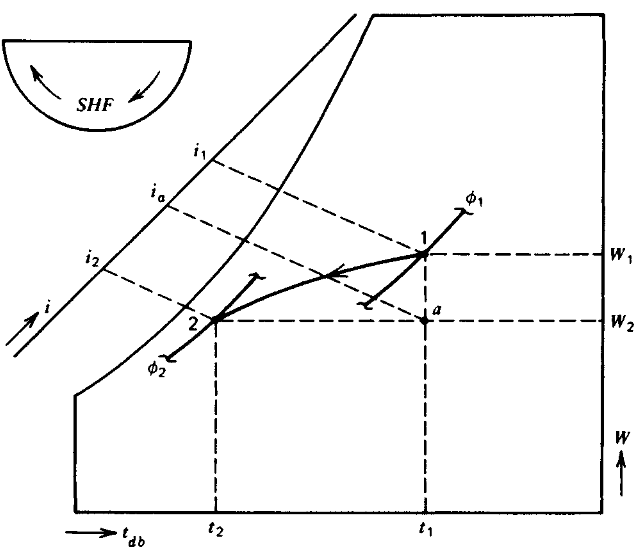

Fig. 64.5 Cooling and dehumidifying process.[7]

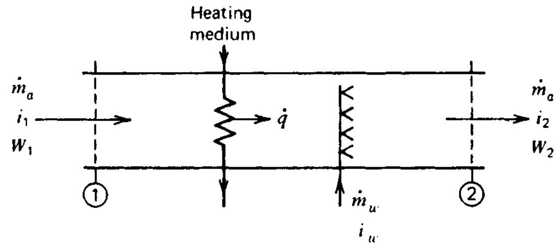

Fig. 64.6 Schematic of a heating and humidifying device.[7]

with respect to the required load by changing the temperature or the amount of heating or cooling fluid entering the coil. This technique is usually used in conjunction with VAV and face and bypass systems. However, control of the coolant temperature or quantity may be the only variable in some systems.

64.1.4 Human Comfort

Air conditioning is the simultaneous control of temperature, humidity, cleanliness, odor, and air circulation as required by the occupants of the space. We are concerned with the conditions that actually provide a comfortable and healthful environment. Not everyone within a given space can be made completely comfortable by one set of conditions, owing to a number of factors, many of which cannot be completely explained. However, clothing, age, sex, and the level of activity of each person are considerations. The factors that influence comfort, in their order of importance, are temperature, radiation, humidity, and air motion, and the quality of the air with regard to odor, dust, and bacteria. With a complete air-conditioning system all of these factors may be controlled simultaneously. In most cases a comfortable environment can be maintained when two or three of these factors are controlled. The *ASHRAE Handbook of Fundamentals* is probably the most up-to-date and complete source of information relating to the physiological aspects of thermal comfort.[3] ASHRAE Comfort Standard 55 defines acceptable thermal comfort as an environment that at least 80% of the occupants will find thermally acceptable.[4]

A complex regulating system in the body acts to maintain the deep body temperature at approximately 98.6°F or 36.9°C. If the environment is maintained at suitable conditions so that the body can easily maintain an energy balance, a feeling of comfort will result.

Two basic mechanisms within the body control the body temperature. The first is a decrease or increase in the internal energy production as the body temperature rises or falls, a process called metabolism. The metabolic rate depends on the level of activity such as rest, work, or exercise. The

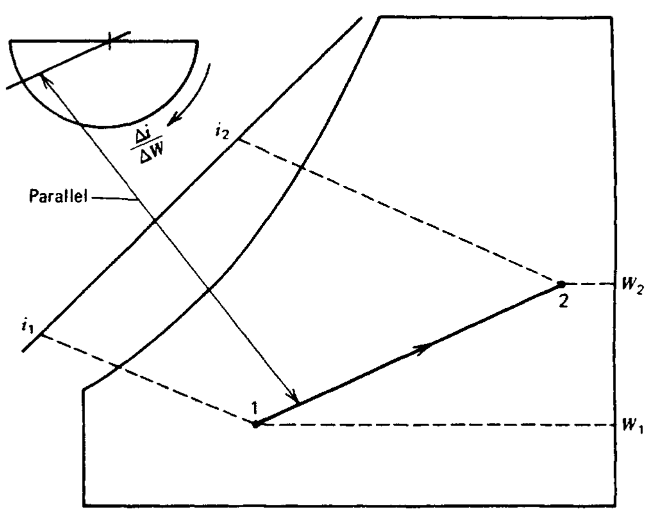

Fig. 64.7 Typical heating and humidifying process.[7]

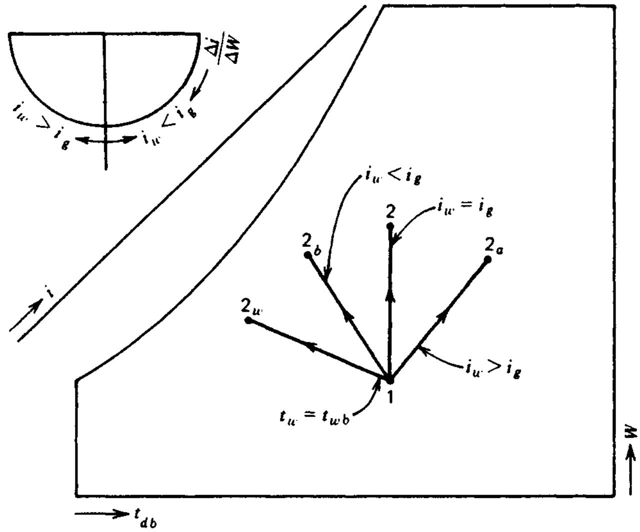

Fig. 64.8 Humidification processes without heat transfer.[7]

second is the control of the rate of heat dissipation by changing the rate of cutaneous blood circulation (the blood circulation near the surface of the skin). In this way heat transfer from the body can be increased or decreased.

Heat transfer to or from the body is principally by convection and conduction and, therefore, the air motion in the immediate vicinity of the body is a very important factor. Radiation exchange between the body and surrounding surfaces, however, can be important if the surfaces surrounding the body are at different temperatures than the air.

Another very important regulatory function of the body is sweating. Under very warm conditions great quantities of moisture can be released by the body to help cool itself.

There are many parameters to describe the environment in term of comfort. The dry bulb temperature is the single most important index of comfort. This is especially true when the relative humidity is between 40% and 60%. The dry bulb temperature is especially important for comfort in the colder regions. When humidity is high, the significance of the dry bulb temperature is less.

The dew point temperature is a good single measure of the humidity of the environment. The usefulness of the dew point temperature in specifying comfort conditions is, however, limited.

The wet bulb temperature is useful in describing comfort conditions in the regions of high temperature and high humidity where dry bulb temperature has less significance. For example, the upper limit for tolerance of the average individual with normal clothing is a wet bulb of about 86°F or 30°C when the air movement is in the neighborhood of 50–75 ft/mm or 0.25–0.38 m/sec.

Relative humidity, although a direct index, has no real meaning in terms of comfort unless the accompanying dry bulb temperature is known. Very high or very low relative humidity is generally associated with discomfort, however.

Air movement is important since the convective heat transfer from the body depends on the velocity of the air moving over it. One is more comfortable in a warm humid environment if the air movement is high. If the temperature is low, one becomes uncomfortable if the air movement is too high. Generally, when air motion is in the neighborhood of 50 ft/min or 0.25 m/sec, the average person will be comfortable.

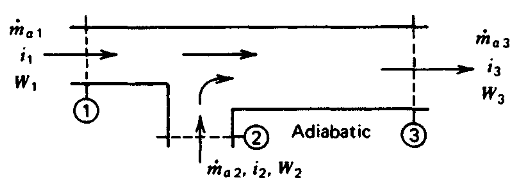

Fig. 64.9 Schematic adiabatic mixing of two air streams.[7]

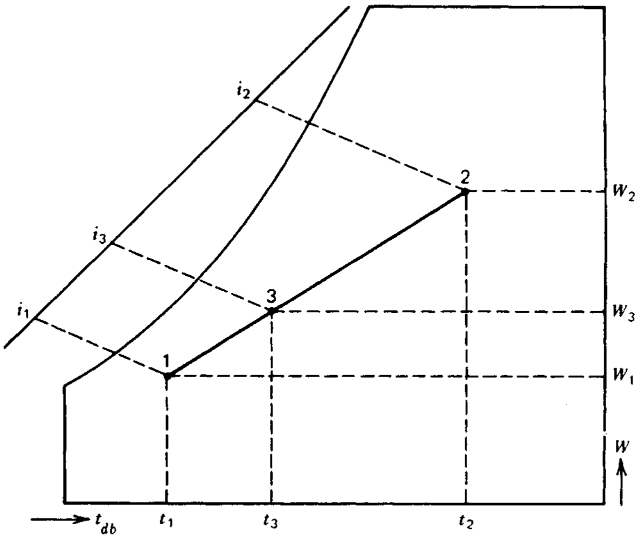

Fig. 64.10 Adiabatic mixing process.[7]

Clothing, through its insulation properties, is an important modifier of body heat loss and comfort. Clothing insulation can be described in terms of its clo value [1 clo = 0.88 ft^2 · hr · °F/Btu = 0.155 m^2 · C/W]. A heavy two-piece business suit and accessories has an insulation value of about 1 clo, whereas a pair of shorts is about 0.05 clo.

Ventilation. The dominating function of outdoor air is to control air quality, and spaces that are more or less continuously occupied require some outdoor air. The required outdoor air is dependent on the rate of contaminant generation and the maximum acceptable contaminant level. In most cases more outdoor air than necessary is supplied. However, some overzealous attempts to save energy through reduction of outdoor air have caused poor-quality indoor air. Table 64.2, from ASHRAE Standard 62-89 (1989), prescribes the requirements for acceptable air quality.[4] Ventilation air is the combination of outdoor air, of acceptable quality, and of recirculated air from the conditioned space which after passing through the air-conditioning unit becomes supply air. The ventilation air may be 100% outdoor air. The term makeup air may be used synonymously with outdoor air, and the terms return and recirculated air are often used interchangeably. A situation could exist where the supply

Table 64.2 National Primary Ambient-Air Quality Standards for Outdoor Air as Set by the U.S. Environmental Protection Agency

	Long Term			Short Term		
	Concentration		Averaging	Concentration		Averaging
Contaminant	μg/m^3	ppm		μg/m^3	ppm	
Sulfur dioxide	80	0.03	1 year	365[a]	0.14[a]	24 hours
Particles (PM 10)	50[b]	—	1 year	150[a]	—	24 hours
Carbon monoxide				40,000[a]	35[a]	1 hour
Carbon monoxide				10,000[a]	9[a]	8 hours
Oxidants (ozone)				235[c]	0.12[c]	1 hour
Nitrogen dioxide	100	0.055	1 year			
Lead	1.5	—	3 months[d]			

[a]Not to be exceeded more than once per year.

[b]Arithmetic mean

[c]Standard is attained when expected number of days per calendar year with maximal average concentrations above 0.12 ppm (235 μg/m^3) is equal to or less than 1.

[d]Three-month period is a calendar quarter.

Source: Reprinted by permission from ANSI/ASHRAE Standard 62-89, 1989 (1).

air required to match the heating or cooling load is greater than the ventilation air. In that case an increased amount of air would be recirculated to meet this condition.

A minimum supply of outdoor air is necessary to dilute the carbon dioxide produced by metabolism and expired from the lungs. This value, 15 cfm or 7.5 liter/sec per person, allows an adequate factor of safety to account for health variations and some increased activity levels. Therefore, outdoor air requirements should never be less than 15 cfm or 7.5 liter/sec per person regardless of the treatment of the recirculated air. Some applications require more than this minimum.[4]

64.2 SPACE HEATING

64.2.1 Heat Transmission in Structures

The design of a heating system is dependent on a good estimate of the heat loss in the space to be conditioned. Precise calculation of heat-transfer rates is difficult, but experience and experimental data make reliable estimates possible. Because most of the calculations require a great deal of repetitive work, tables that list coefficients and other data for typical situations are used. Thermal resistance is a very useful concept and is used extensively.

Generally all three modes of heat transfer—conduction, convection, and radiation—are important in building heat gain and loss.

Thermal conduction is heat transfer between parts of a continuum because of the transfer of energy between particles or groups of particles at the atomic level. The Fourier equation expresses steady-state conduction in one dimension:

$$\dot{q} = -kA\frac{dt}{dx}$$

where q = heat transfer rate, Btu/hr or W
k = thermal conductivity, Btu/hr · ft · °F or W/m · °C
A = area normal to heat flow, ft or m
dt/dx = temperature gradient, °F/ft or °C/m

A negative sign appears because $\dot{q}$ flows in the positive direction of x when dt/dx is negative.

Consider the flat wall of Fig. 64.11*a*, where uniform temperatures t_1 and t_2 are assumed to exist on each surface. If the thermal conductivity, the heat-transfer rate, and the area are constant, integration gives

$$\dot{q} = \frac{-kA(t_2 - t_1)}{(x_2 - x_1)}$$

Another very useful form is

$$\dot{q} = \frac{-(t_2 - t_1)}{R'}$$

where R' is the thermal resistance defined by

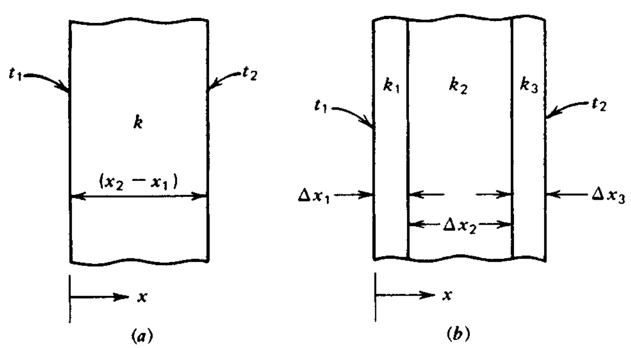

Fig. 64.11 Nomenclature for conduction in plane walls.[7]

$$R' = \frac{x_2 - x_1}{kA} = \frac{\Delta x}{kA}$$

The thermal resistance for a unit area of material is very commonly used. This quantity, sometimes called the R-factor, is referred to as the unit thermal resistance or simply the unit resistance, R. For a plane wall the unit resistance is

$$R = \frac{\Delta x}{k}$$

Note that thermal resistance R' is analogous to electrical resistance and q and $t_2 - t_1$ are analogous to current and potential difference in Ohm's law. This analogy provides a very convenient method of analyzing a wall or slab made up of two or more layers of dissimilar material. Figure 64.11*b* shows a wall constructed of three different materials. The heat transferred by conduction is given by

$$R' = R_1' + R_2' + R_3' = \frac{\Delta x_1}{k_1 A} + \frac{\Delta x_2}{k_2 A} + \frac{\Delta x_3}{k_3 A}$$

Thermal convection is the transport of energy by mixing in addition to conduction. Convection is associated with fluids in motion, generally through a pipe or duct or along a surface. In the very thin layer of fluid next to the surface the transfer of energy is by conduction. In the main body of the fluid mixing is the dominant energy-transfer mechanism. A combination of conduction and mixing exists between these two regions. The transfer mechanism is complex and highly dependent on whether the flow is laminar or turbulent.

The usual, simplified approach in convection is to express the heat-transfer rate as

$$\dot{q} = hA(t - t_w)$$

where $\dot{q}$ = heat transfer rate from fluid to wall, Btu/hr or W
h = film coefficient, Btu/hr · ft² · °F or W/m² sec
t = bulk temperature of the fluid, °F or °C
t_w = wall temperature, °F or °C

The film coefficient h, sometimes called the unit surface conductance or alternatively the convective heat transfer coefficient, may also be expressed in terms of thermal resistance:

$$\dot{q} = \frac{t - t_w}{R'}$$

where

$$R' = \frac{1}{hA} \quad (\text{hr} \cdot {}^\circ\text{F/Btu or } {}^\circ\text{C/W})$$

or

$$R = \frac{1}{h} = \frac{1}{C}$$

where C is the unit thermal conductance. The thermal resistance for convection may be summed with the thermal resistances arising from pure conduction.

The film coefficient h depends on the fluid, the fluid velocity, the flow channel, and the degree of development of the flow field. Many correlations exist for predicting the film coefficient under various conditions. Correlations for forced convection are given in Chapter 3 of the *ASHRAE Handbook.*[2,5]

When the bulk of the fluid is moving relative to the heat-transfer surface, the mechanism is called forced convection, because such motion is usually caused by a blower, fan, or pump, which is forcing the flow. In forced convection buoyancy forces are negligible. In free convection, on the other hand, the motion of the fluid is due entirely to buoyancy forces, usually confined to a layer near the heated or cooled surface. Free convection is often referred to as natural convection.

Natural or free convection is an important part of HVAC applications. Various empirical relations for natural convection film coefficients can be found in the *ASHRAE Handbook of Fundamentals* (1997).[2]

Most building structures have forced convection along outer walls or roofs, and natural convection in inside air spaces and on the inner walls. There is considerable variation in surface conditions, and both the direction and magnitude of the air motion on outdoor surfaces are very unpredictable. The film coefficient for these situations usually ranges from about 1.0 Btu/hr · ft^2 · °F or 6 W/m^2 · °C for free convection up to about 6 Btu/hr · ft^2 · °F or 35 W/m^2 · °C for forced convection with an air velocity of about 15 miles per hour, 20 ft/sec, or 6 m/sec. Because of the low film coefficients the amount of heat transferred by thermal radiation may be equal to or larger than that transferred by free convection.

Thermal radiation, the transfer of thermal energy by electromagnetic waves, can occur in a perfect vacuum and is actually impeded by an intervening medium. The direct net transfer of energy by radiation between two surfaces which see only each other and which are separated by a nonabsorbing medium is given by

$$\dot{q}_{1-2} = \frac{\sigma(T_1^4 - T_2^4)}{\dfrac{1-\epsilon_1}{A_1\epsilon_1} + \dfrac{1}{A_1F_{12}} + \dfrac{1-\epsilon_2}{A_2\epsilon_2}}$$

where σ = Boltzmann constant, 0.1713×10^{-8} Btu/hr · ft^2 · °R^4 or 5.673×10^{-8} W/m · K^4
T = absolute temperature, °R or K
ϵ = emittance
A = surface area, ft^2 or m^2
F = configuration factor, a function of geometry only

It has been assumed that both surfaces are "gray" (where the emittance ϵ equals the absorptance α).[6] Figure 64.12 shows situations where radiation may be a significant factor. For the wall,

$$\dot{q}_i = \dot{q}_w = \dot{q}_r + \dot{q}_o$$

and for the air space,

$$\dot{q}_i = \dot{q}_r + \dot{q}_c = \dot{q}_o$$

The resistances can be combined to obtain an equivalent overall resistance R' with which the heat-transfer rate can be computed using

$$\dot{q} = \frac{-(t_o - t_i)}{R'}$$

The thermal resistance for radiation is not easily computed, however, because of the fourth power temperature relationship.

Tables are available that give conductances and resistances for air spaces as a function of position, direction of heat flow, air temperature, and the effective emittance of the space.[5] The effective emittance E is given by

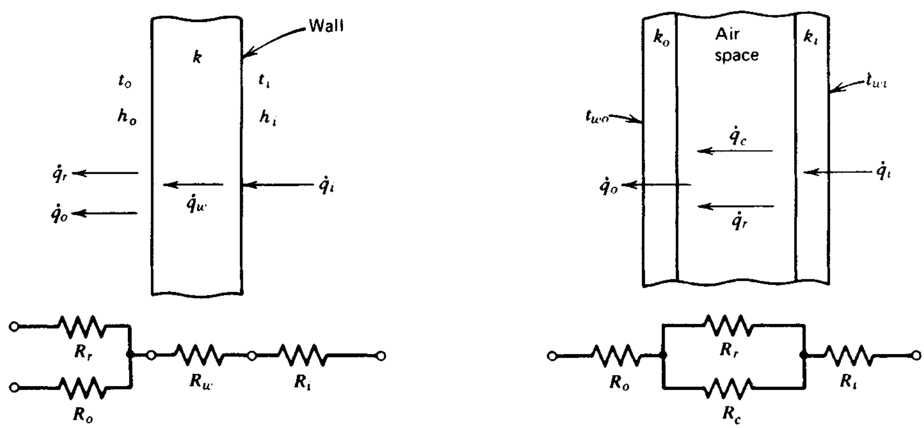

Fig. 64.12 Wall and air space illustrating thermal radiation effects.[7]

$$\frac{1}{E} = \frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1$$

where ϵ_1 and ϵ_2 are for each surface of the air space. Resistors connected in series may be replaced by an equivalent resistor equal to the sum of the series resistors; it will have an equivalent effect on the circuit:

$$R'_e = R'_1 + R'_2 + R'_3 + \cdots + R'_n$$

Figure 64.13 is an example of a wall being heated or cooled by a combination of convection and radiation on each surface and having five different resistances through which the heat must be conducted. The equivalent thermal resistance R'_e for the wall is given by

$$R'_e = R'_i + R'_1 + R'_2 + R'_3 + R'_o$$

Each of the resistances may be expressed in terms of fundamental variables giving

$$R'_e = \frac{1}{h_i A_1} + \frac{\Delta x_1}{k_1 A_1} + \frac{\Delta x_2}{k_2 A_2} + \frac{\Delta x_3}{k_3 A_3} + \frac{1}{h_o A_o}$$

The film coefficients and the thermal conductivities may be obtained from tables. For a plane wall, the areas are all equal and cancel.

The concept of thermal resistance is very useful and convenient in the analysis of complex arrangements of building materials. After the equivalent thermal resistance has been determined for a configuration, however, the overall unit thermal conductance, usually called the overall heat transfer coefficient U, is frequently used:

$$U = \frac{1}{R'A} = \frac{1}{R} \quad (\text{Btu/hr} \cdot \text{ft}^2 \cdot {}^\circ\text{F or W/m}^2 \cdot {}^\circ\text{C})$$

The heat transfer rate is then given by

$$\dot{q} = UA\,\Delta t$$

where UA = conductance, Btu/hr · °F or W/°C
A = surface area, ft^2 or m^2
Δt = overall temperature difference, °F or °C

Tabulated Overall Heat Transfer Coefficients. For convenience of the designer, tables have been constructed that give overall coefficients for many common building sections including walls and floors, doors, windows, and skylights. The tables in the *ASHRAE Handbook* have a great deal of flexibility and are widely used.[2]

64.2.2 Design Conditions

Prior to the design of the heating system an estimate must be made of the maximum probable heat loss of each room or space to be heated. During the coldest months, sustained periods of very cold, cloudy, and stormy weather with relatively small variation in outdoor temperature may occur. In this situation heat loss from the space will be relatively constant and in the absence of internal heat gains will peak during the early morning hours. Therefore, for design purposes the heat loss is usually

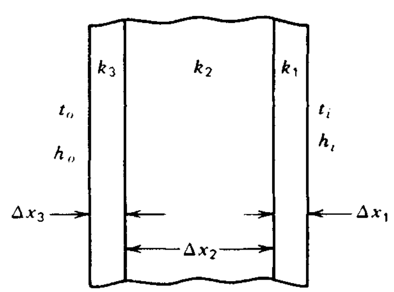

Fig. 64.13 Wall with thermal resistances in series.[7]

estimated for steady-state heat transfer for some reasonable design temperature. Transient analyses are often used to study the actual energy requirements of a structure in simulation studies. In such cases solar effects and internal heat gains are taken into account.

Here is the general procedure for calculation of design heat losses of a structure[7]:

1. Select the outdoor design conditions: temperature, humidity, and wind direction and speed.
2. Select the indoor design conditions to be maintained.
3. Estimate the temperature in any adjacent unheated spaces.
4. Select the transmission coefficients and compute the heat losses for walls, floors, ceilings, windows, doors, and floor slabs.
5. Compute the heat load due to infiltration.
6. Compute the heat load due to outdoor ventilation air. This may be done as part of the air quantity calculation.
7. Sum the losses due to transmission and infiltration.

The ideal heating system would provide enough heat to match the heat loss from the structure. However, weather conditions vary considerably from year to year, and heating systems designed for the worst weather conditions on record would have a great excess of capacity most of the time. The failure of a system to maintain design conditions during brief periods of severe weather is usually not critical. However, close regulation of indoor temperature may be critical for some industrial processes.

The outdoor design temperature should generally be the 97½% value. The 97½% value is the temperature equaled or exceeded 97½% of the total hours (2160) in December, January, and February. During a normal winter there would be about 54 hr at or below the 97½% value. For the Canadian stations the 97½% value pertains only to hours in January. If the structure is of lightweight construction, or poorly insulated, has considerable glass, and space temperature control is critical, however, 99% values should be considered. Should the outdoor temperature fall below the design value for some extended period, the indoor temperature may do likewise. The performance expected by the owner is a very important factor, and the designer should make clear to the owner the various factors considered in the design.

The indoor design temperature should be kept relatively low so that the heating equipment will not be oversized. Even properly sized equipment operates under partial load, at reduced efficiency, most of the time; therefore, any oversizing aggravates this condition and lowers the overall system efficiency. The indoor design value of relative humidity should be compatible with a healthful environment and the thermal and moisture integrity of the building envelope.

64.2.3 Calculation of Heat Losses

The heat transferred through walls, ceiling, roof, window glass, floors, and doors is all sensible heat transfer, referred to as transmission heat loss and computed from

$$\dot{q} = UA\,(t_i - t_o)$$

A separate calculation is made for each different surface in each room of the structure. To ensure a thorough job in estimating the heat losses, a worksheet should be used to provide a convenient and orderly way of recording all the coefficients and areas. Summations are conveniently made by room and for the complete structure.

All structures have some air leakage or infiltration. This means a heat loss because the cold dry outdoor air must be heated to the inside design temperature and moisture must be added to increase the humidity to the design value. The heat required is given by

$$\dot{q}_s = \dot{m}_o c_p (t_i - t_o)$$

where $\dot{m}_o$ = mass flow rate of the infiltrating air, lbm/hr or kg/sec
c_p = specific heat capacity of the moist air, Btu/lbm · °F or J/kg · °C

Infiltration is usually estimated on the basis of volume flow rate at outdoor conditions:

$$\dot{q}_s = \frac{\dot{Q} c_p\,(t_i - t_o)}{v_o}$$

where $\dot{Q}$ = volume flow rate, ft³/hr or m³/sec
v_o = specific volume, ft³/lbm or m³/sec

The latent heat required to humidify the air is given by

$$\dot{q}_l = \dot{m}(W_i - W_o)i_{fg}$$

where $W_i - W_o$ = difference in design humidity ratio, $\text{lbm}_v/\text{lbm}_a$ or kg_v/kg_a
i_{fg} = latent heat of vaporization at indoor conditions, Btu/lbm_v or J/kg_v

In terms of volume flow rate,

$$\dot{q}_l = \frac{Q}{v_o}(W_i - W_o)i_{fg}$$

Infiltration can account for a large portion of the heating load.

Two methods are used in estimating air infiltration in building structures. In one method the estimate is based on the characteristics of the windows and doors and the pressure difference between inside and outside. This is known as the crack method, because of the cracks around window sash and doors. The other approach is the air-change method, which is based on an assumed number of air changes per hour for each room depending on the number of windows and doors. The crack method is generally considered to be the most accurate when the window and pressure characteristics can be properly evaluated. However, the accuracy of predicting air infiltration is restricted by the limited information on the air-leakage characteristics of the many components that make up a structure. The pressure differences are also difficult to predict because of variable wind conditions and stack effect in tall buildings.

64.2.4 Air Requirements

There are many cases, especially in residential and light commercial applications, when the latent heat loss is quite small and may be neglected. The air quantity is then computed from

$$\dot{q} = \dot{m}c_p\,(t_s - t_r)$$

or

$$\dot{q} = \frac{\dot{Q}c_p}{v_s}(t_s - t_r)$$

where v_s = specific volume of supplied air, ft^3/lbm or m^3/kg
t_s = temperature of supplied air, °F or °C
t_r = room temperature, °F or °C

Residential and light commercial equipment operates with a temperature rise of 60–80°F or 33–44°C, whereas commercial applications will allow higher temperatures. The temperature of the air to be supplied must not be high enough to cause discomfort to occupants before it becomes mixed with room air.

In the unit-type equipment typically used for residences and small commercial buildings each size is able to circulate a relatively fixed quantity of air. Therefore, the air quantity is fixed within a narrow range when the heating equipment is selected. A slightly oversized unit is usually selected with the capacity to circulate a larger quantity of air than theoretically needed. Another condition that leads to greater quantities of circulated air for heating than needed is the greater air quantity sometimes required for cooling and dehumidifying. The same fan is used throughout the year and must therefore be large enough for the maximum air quantity required. Some units have different fan speeds for heating and for cooling.

After the total air flow rate required for the complete structure has been determined, the next step is to allocate the correct portion of the air to each room or space. This is necessary for design of the duct system. Obviously, the air quantity for each room should be apportioned according to the heating load for that space; therefore,

$$\dot{Q}_{rn} = \dot{Q}(\dot{q}_{rn}/\dot{q})$$

where Q_{rn} = volume flow rate of air supplied to room n, ft^3/min or m^3/sec
$\dot{q}_{rn}$ = total heat loss rate of room n, Btu/hr or W

The worksheet should have provisions for recording the air quantity for the structure and for each room.

64.2.5 Fuel Requirements

It is often desirable to estimate the quantity of energy necessary to heat the structure under typical weather conditions and with typical imputs from internal heat sources. This is a distinct procedure from design heat load calculations, which are usually made for one set of design conditions neglecting solar effects and internal heat sources. Simulation usually requires a digital computer.

In some cases where computer simulation is not possible or cannot be justified, such as residential buildings, reasonable results can be obtained using hand calculation methods such as the degree-day or bin method.

The degree-day procedure for computing fuel requirements is based on the assumption that, on a long-term basis, solar and internal gains will offset heat loss when the mean daily outdoor temperature is 65°F or 18°C. It is further assumed that fuel consumption will be proportional to the difference between the mean daily temperature and 65°F or 18°C. Degree days are defined by the relationship

$$\mathrm{DD} = \frac{(t - t_a)N}{24}$$

where N is the number of hours for which the average temperature t_a is computed and t is 65°F or 18°C. The general relation for fuel calculations using this procedure is

$$F = \frac{24\ \mathrm{DD}\dot{q}C_D}{\eta(t_i - t_o)H}$$

where F = the quantity of fuel required for the period desired; the units depend on H
DD = the degree days for period desired, °F-day or °C-day
$\dot{q}$ = the total calculated heat loss based on design condition, t_i and t_o, Btu/hr or W
η = an efficiency factor, which includes the effects of rated full load efficiency, part load performance, oversizing, and energy conservation devices
H = the heating value of fuel, Btu or kWhr per unit volume or mass
C_D = the interim correction factor for degree days based on 65°F or 18°C, Fig. 64.14

64.3 SPACE COOLING

64.3.1 Heat Gain, Cooling Load, and Heat Extraction Rate

A larger number of variables are considered in making cooling load calculations than in heating load calculations. In design for cooling, transient analysis must be used if satisfactory results are to be obtained. This is because the instantaneous heat gain into a conditioned space is quite variable with time primarily because of the strong transient effect created by the hourly variation in solar radiation. There may be an appreciable difference between the heat gain of the structure and the heat removed by the cooling equipment at a particular time. This difference is caused by the storage and subsequent transfer of energy from the structure and contents to the circulated air. If this is not taken into account, the cooling and dehumidifying equipment will usually be grossly oversized and estimates of energy requirements meaningless.

Heat gain is the rate at which energy is transferred to or generated within a space. It has two components, sensible heat and latent heat, which must be computed and tabulated separately. Heat gains usually occur in the following forms:

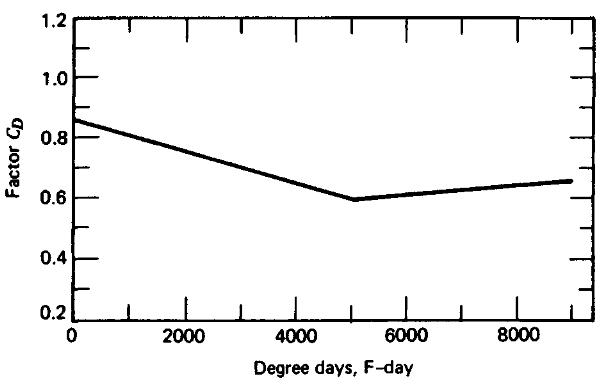

Fig. 64.14 Correction factor. (Reprinted by permission from *ASHRAE*.)

1. Solar radiation through openings.
2. Heat conduction through boundaries with convection and radiation from the inner surface into the space.
3. Sensible heat convection and radiation from internal objects.
4. Ventilation (outside) and infiltration air.
5. Latent heat gains generated within the space.

The cooling load is the rate at which energy must be removed from a space to maintain the temperature and humidity at the design values. The cooling load will generally differ from the heat gain at any instant of time, because the radiation from the inside surface of walls and interior objects as well as the solar radiation coming directly into the space through openings does not heat the air within the space directly. This radiant energy is mostly absorbed by floors, interior walls, and furniture, which are then cooled primarily by convection as they attain temperatures higher than that of the room air. Only when the room air receives the energy by convection does this energy become part of the cooling load. The heat-storage characteristics of the structure and interior objects determine the thermal lag and therefore the relationship between heat gain and cooling load. For this reason the thermal mass (product of mass and specific heat) of the structure and its contents must be considered in such cases. The reduction in peak cooling load because of the thermal lag can be quite important in sizing the cooling equipment.

The heat-extraction rate is the rate at which energy is removed from the space by the cooling and dehumidifying equipment. This rate may be equal to the cooling load. However, this is rarely true and some fluctuation in room temperature occurs. Because the cooling load is below the peak or design value most of the time, intermittent or variable operation of the cooling equipment is required.

64.3.2 Design Conditions

The problem of selecting outdoor design conditions for calculation of heat gain is similar to that for heat loss. Again it is not reasonable to design for the worst conditions on record because a great excess of capacity will result. The heat-storage capacity of the structure also plays an important role in this regard. A massive structure will reduce the effect of overload from short intervals of outdoor temperature above the design value. The *ASHRAE Handbook of Fundamentals* gives extensive outdoor design data.[2] Tabulation of dry bulb and mean coincident wet bulb temperatures that are equaled or exceeded 1%, 2½%, and 5% of the total hours during June through September (2928 hr) are given. The 2½% values are recommended for design purposes by ASHRAE.[2,5] The daily range of temperature is the difference between the average maximum and average minimum for the warmest month. The daily range is usually larger for the higher elevations where temperatures may be quite low late at night and during the early morning hours. The daily range has an effect on the energy stored by the structure. The variation in dry bulb temperature for a typical design day may be computed using the peak outdoor dry bulb temperature and the daily range, assuming a cosine relation with a maximum temperature at 3 PM and a minimum at 5 AM.

The local wind velocity for summer conditions is usually taken to be about one-half the winter design value but not less than about 7½ mph or 3.4 m/sec.

The indoor design conditions for the average job in the United States and Canada is 75°F or 24°C dry bulb and a relative humidity of 50%, when activity and dress of the occupants are light. The designer should be alert for unusual circumstances that may lead to uncomfortable conditions. Certain activities may require occupants to engage in active work or require heavy protective clothing, both of which would require lower design temperatures.

64.3.3 Calculation of Heat Gains

Design cooling loads for one day as well as long-term energy calculations may be done using the transfer-function approach. Details of this method are discussed in the *ASHRAE Handbook of Fundamentals.*[2]

It is not always practical to compute the cooling load using the transfer-function method; therefore, a hand calculation method has been developed from the transfer-function procedure and is referred to as the cooling-load-temperature-difference (CLTD) method. The method involves extensive use of tables and charts and various factors to express the dynamic nature of the problem and predicts cooling loads within about 5% of the values given by the transfer-function method.[5]

The CLTD method makes use of a temperature difference in the case of walls and roofs and cooling load factors (CLF) in the case of solar gain through windows and internal heat sources. The CLTD and CLF vary with time and are a function of environmental conditions and building parameters. They have been derived from computer solutions using the transfer-function procedure. A great deal of care has been taken to sample a wide variety of conditions in order to obtain reasonable accuracy. These factors have been derived for a fixed set of surface and environmental conditions; therefore, correction factors must often be applied. In general, calculations proceed as follows.

For walls and roofs,

$$\dot{q}_\theta = (U)(A)(\text{CLTD})_\theta$$

where U = overall heat transfer coefficient, Btu/hr · ft² · °F or W/m² · °C
A = area, ft² or m²
$(\text{CLTD})_\theta$ = temperature difference which gives the cooling load at time θ, °F or °C

The CLTD accounts for the thermal response (lag) in the heat transfer through the wall or roof, as well as the response (lag) due to radiation of part of the energy from the interior surface of the wall to objects within the space.

For solar gain through glass

$$\dot{q}_\theta = (A)(\text{SC})(\text{SHGF})(\text{CLF})_\theta$$

where A = area, ft² or m²
SC = shading coefficient (internal shade)
SHGF = solar heat gain factor, Btu/hr · ft² or W/m²
$(\text{CLF})_\theta$ = cooling load factor for time θ

The SHGF is the maximum for a particular month, orientation, and latitude. The CLF accounts for the variation of the SHGF with time, the massiveness of the structure, and internal shade. Again the CLF accounts for the thermal response (lag) of the radiant part of the solar input.

For internal heat sources

$$\dot{q}_\theta = (\dot{q}_i)(\text{CLF})_\theta$$

where $\dot{q}_i$ = instantaneous heat gain from lights, people, and equipment, Btu/hr or W
$(\text{CLF})_\theta$ = cooling load factor for time θ

The CLF accounts for the thermal response of the space to the various internal heat gains and is slightly different for each.

The time of day when the peak cooling load will occur must be estimated. In fact, two different types of peaks need to be determined. First, the time of the peak load for each room is needed in order to compute the air quantity for that room. Second, the time of the peak load for a zone served by a central unit is required to size the unit. It is at these peak times that cooling load calculations should be made. The estimated times when the peak load will occur are determined from the tables of CLTD and CLF values together with the orientation and physical characteristics of the room or space. The times of the peak cooling load for walls, roofs, windows, and so on, is obvious in the tables and the most dominant cooling load components will then determine the peak time for the entire room or zone. For example, rooms facing west with no exposed roof will experience a peak load in the late afternoon or early evening. East-facing rooms tend to peak during the morning hours. A zone made up of east and west rooms with no exposed roofs will tend to peak when the west rooms peak. If there is a roof, the zone will tend to peak when the roof peaks. High internal loads may dominate the cooling load in some cases and cause an almost uniform load throughout the day.

The details of computing the various cooling load components are discussed in ASHRAE Cooling and Heating Load Calculation Manual.[5]

It is emphasized that the total space cooling load does not generally equal the load imposed on the central cooling unit or cooling coil. The outdoor ventilation air is usually mixed with return air and conditioned before it is supplied to the space. The air circulating fan may be upstream of the coil, in which case the fan power input is a load on the coil. In the case of vented light fixtures, the heat absorbed by the return air is imposed on the coil and not the room.

The next steps are to determine the air quantities and to select the equipment. These steps may be reversed depending on the type of equipment to be used.

64.3.4 Air Requirements

Computing air quantity for cooling and dehumidification requires the use of psychrometric charts. The cooling and dehumidifying coil is designed to match the sensible and latent heat requirements of a particular job and the fan is sized to handle the required volume of air. The fan, cooling coil, control dampers, and the enclosure for these components, referred to as an air handler, are assembled at the factory in a wide variety of coil and fan models to suit almost any requirement. The design engineer usually specifies the entering and leaving moist air conditions, the volume flow rate of the air, and the total pressure the fan must produce.

Specifically constructed equipment cannot be justified for small commercial and residential applications. Furthermore, these applications generally have a higher sensible heat factor, and dehumidification is not as critical as it is in large commercial buildings. Therefore, the equipment is manufactured to operate at or near one particular set of conditions. For example, typical residential and light commercial cooling equipment operates with a coil SHF of 0.75–0.8 with the air entering the coil at about 80°F or 27°C dry bulb and 67°F or 19°C wet bulb temperature. This equipment usually has a capacity of less than 10 tons or 35 kW. When the peak cooling load and latent heat requirements are appropriate, this less expensive type of equipment is used. In this case the air quantity is determined in a different way. The peak cooling load is first computed as 1.3 times the peak sensible cooling load for the structure to match the coil SHF. The equipment is then selected to match the peak cooling load as closely as possible. The air quantity is specified by the manufacturer for each unit and is about 400 cfm/ton or 0.0537 $m^3/sec \cdot kW$. The total air quantity is then divided among the various rooms according to the cooling load of each room.

64.3.5 Fuel Requirements

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

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

64.4 AIR-CONDITIONING EQUIPMENT

64.4.1 Central Systems

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

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

All-Air Systems

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

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

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

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

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