GAS MIXTURES AND PSYCHROMETRICS

U p to this point, we have limited our consideration to thermodynamic systems that involve a single pure substance such as water, refrigerant-134a, or nitrogen. Many important thermodynamic applications, however, involve *mixtures* of several pure substances rather than a single pure substance. Therefore, it is important to develop an understanding of mixtures and learn how to handle them.

In this chapter, we deal with nonreacting gas mixtures. A nonreacting gas mixture can be treated as a pure substance since it is usually a homogeneous mixture of different gases. The properties of a gas mixture obviously will depend on the properties of the individual gases (called *components* or *constituents*) as well as on the amount of each gas in the mixture. Therefore, it is possible to prepare tables of properties for mixtures. This has been done for common mixtures such as air. It is not practical to prepare property tables for every conceivable mixture composition, however, since the number of possible compositions is endless. Therefore, we need to develop rules for determining mixture properties from a knowledge of mixture composition and the properties of the individual components. We do this first for ideal-gas mixtures and then for real-gas mixtures. The basic principles involved are also applicable to liquid or solid mixtures, called *solutions*.

At temperatures below the critical temperature, the gas phase of a substance is frequently referred to as a vapor. The term vapor implies a gaseous state that is close to the saturation region of the substance, raising the possibility of condensation during a process.

When we deal with a gas–vapor mixture, the vapor may condense out of the mixture during a process, forming a two-phase mixture. This may complicate the analysis considerably. Therefore, a gas–vapor mixture needs to be treated differently from an ordinary gas mixture.

Several gas–vapor mixtures are encountered in engineering. In this chapter, we consider the *air–water-vapor mixture,* which is the most commonly encountered gas–vapor mixture in practice. We also discuss *air-conditioning,* which is the primary application area of air–water-vapor mixtures.

CONTENTS

9–1 ▪ COMPOSITION OF A GAS MIXTURE: MASS AND MOLE FRACTIONS

To determine the properties of a mixture, we need to know the *composition* of the mixture as well as the properties of the individual components. There are two ways to describe the composition of a mixture: either by specifying the number of moles of each component, called **molar analysis,** or by specifying the mass of each component, called **gravimetric analysis.**

Consider a gas mixture composed of k components. The mass of the mixture m_m is the sum of the masses of the individual components, and the mole number of the mixture N_m is the sum of the mole numbers of the individual components* (Figs. 9–1 and 9–2). That is,

$$m_m = \sum_{i=1}^{k} m_i \quad \text{and} \quad N_m = \sum_{i=1}^{k} N_i \tag{9–1a, b}$$

The ratio of the mass of a component to the mass of the mixture is called the **mass fraction** mf, and the ratio of the mole number of a component to the mole number of the mixture is called the **mole fraction** y:

$$\text{mf}_i = \frac{m_i}{m_m} \quad \text{and} \quad y_i = \frac{N_i}{N_m} \tag{9–2a, b}$$

Dividing Eq. 9–1a by m_m or Eq. 9–1b by N_m, we can easily show that the sum of the mass fractions or mole fractions for a mixture is equal to 1 (Fig. 9–3):

$$\sum_{i=1}^{k} \text{mf}_i = 1 \quad \text{and} \quad \sum_{i=1}^{k} y_i = 1$$

The mass of a substance can be expressed in terms of the mole number N and molar mass M of the substance as $m = NM$. Then the **apparent** (or **average**) **molar mass** and the **gas constant** of a mixture can be expressed as

$$M_m = \frac{m_m}{N_m} = \frac{\sum m_i}{N_m} = \frac{\sum N_i M_i}{N_m} = \sum_{i=1}^{k} y_i M_i \quad \text{and} \quad R_m = \frac{R_u}{M_m} \tag{9–3a, b}$$

The molar mass of a mixture can also be expressed as

$$M_m = \frac{m_m}{N_m} = \frac{m_m}{\sum m_i/M_i} = \frac{1}{\sum m_i/(m_m M_i)} = \frac{1}{\sum_{i=1}^{k} \dfrac{\text{mf}_i}{M_i}} \tag{9–4}$$

Mass and mole fractions of a mixture are related by

$$\text{mf}_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = y_i \frac{M_i}{M_m} \tag{9–5}$$
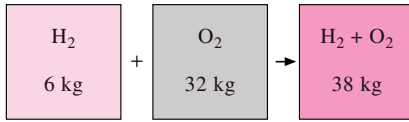

FIGURE 9–1
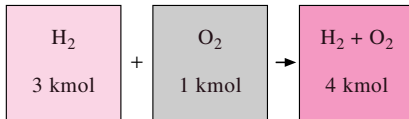The mass of a mixture is equal to the sum of the masses of its components.

FIGURE 9–2
The number of moles of a nonreacting mixture is equal to the sum of the number of moles of its components.
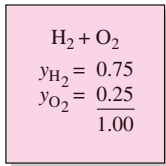

FIGURE 9–3
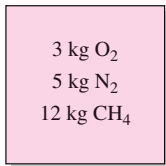The sum of the mole fractions of a mixture is equal to 1.

FIGURE 9–4
Schematic for Example 9–1.

EXAMPLE 9–1   **Mass and Mole Fractions of a Gas Mixture**

Consider a gas mixture that consists of 3 kg of O_2, 5 kg of N_2, and 12 kg of CH_4, as shown in Fig. 9–4. Determine (*a*) the mass fraction of each component, (*b*) the mole fraction of each component, and (*c*) the average molar mass and gas constant of the mixture.

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*Throughout this chapter, the subscript m will denote the gas mixture and the subscript i will denote any single component of the mixture.

SOLUTION   The schematic of the gas mixture is given in Fig. 9–4. We note that this is a gas mixture that consists of three gases of known masses.

Analysis   (*a*) The total mass of the mixture is

$$m_m + m_{O_2} + m_{N_2} + m_{CH_4} = 3 + 5 + 12 = 20 \text{ kg}$$

Then the mass fraction of each component becomes

$$mf_{O_2} = \frac{m_{O_2}}{m_m} = \frac{3 \text{ kg}}{20 \text{ kg}} = \textbf{0.15}$$

$$mf_{N_2} = \frac{m_{N_2}}{m_m} = \frac{5 \text{ kg}}{20 \text{ kg}} = \textbf{0.25}$$

$$mf_{CH_4} = \frac{m_{CH_4}}{m_m} = \frac{12 \text{ kg}}{20 \text{ kg}} = \textbf{0.60}$$

(*b*) To find the mole fractions, we need to determine the mole numbers of each component first:

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{3 \text{ kg}}{32 \text{ kg/kmol}} = 0.094 \text{ kmol}$$

$$N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{5 \text{ kg}}{28 \text{ kg/kmol}} = 0.179 \text{ kmol}$$

$$N_{CH_4} = \frac{m_{CH_4}}{M_{CH_4}} = \frac{12 \text{ kg}}{16 \text{ kg/kmol}} = 0.750 \text{ kmol}$$

Thus,

$$N_m = N_{O_2} + N_{N_2} + N_{CH_4} = 0.094 + 0.179 + 0.750 = 1.023 \text{ kmol}$$

and

$$y_{O_2} = \frac{N_{O_2}}{N_m} = \frac{0.094 \text{ kmol}}{1.023 \text{ kmol}} = \textbf{0.092}$$

$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{0.179 \text{ kmol}}{1.023 \text{ kmol}} = \textbf{0.175}$$

$$y_{CH_4} = \frac{N_{CH_4}}{N_m} = \frac{0.750 \text{ kmol}}{1.023 \text{ kmol}} = \textbf{0.733}$$

(*c*) The average molar mass and gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{20 \text{ kg}}{1.023 \text{ kmol}} = \textbf{19.6 kg/kmol}$$

or

$$M_m = \sum y_i M_i = y_{O_2} M_{O_2} + y_{N_2} M_{N_2} + y_{CH_4} M_{CH_4}$$
$$= (0.092)(32) + (0.175)(28) + (0.733)(16)$$
$$= 19.6 \text{ kg/kmol}$$

Also,

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/(kmol} \cdot \text{K)}}{19.6 \text{ kg/kmol}} = \textbf{0.424 kJ/(kg} \cdot \textbf{K)}$$

Discussion   When mass fractions are available, the molar mass and mass fractions could also be determined directly from Eqs. 9–4 and 9–5.

9–2 ▪ *P-v-T* BEHAVIOR OF GAS MIXTURES: IDEAL AND REAL GASES

An ideal gas is defined as a gas whose molecules are spaced far apart so that the behavior of a molecule is not influenced by the presence of other molecules—a situation encountered at low densities. We also mentioned that real gases approximate this behavior closely when they are at a low pressure or high temperature relative to their critical-point values. The *P-v-T* behavior of an ideal gas is expressed by the simple relation $Pv = RT$, which is called the *ideal-gas equation of state.* The *P-v-T* behavior of real gases is expressed by more complex equations of state or by $Pv = ZRT$, where Z is the compressibility factor.

When two or more ideal gases are mixed, the behavior of a molecule normally is not influenced by the presence of other similar or dissimilar molecules, and therefore a nonreacting mixture of ideal gases also behaves as an ideal gas. Air, for example, is conveniently treated as an ideal gas in the range where nitrogen and oxygen behave as ideal gases. When a gas mixture consists of real (nonideal) gases, however, the prediction of the *P-v-T* behavior of the mixture becomes rather involved.

The prediction of the *P-v-T* behavior of gas mixtures is usually based on two models: *Dalton's law of additive pressures* and *Amagat's law of additive volumes.* Both models are described and discussed below.
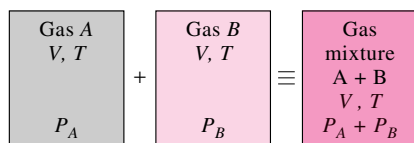

FIGURE 9–5
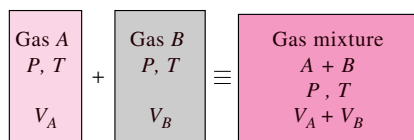Dalton's law of additive pressures for a mixture of two ideal gases.

Dalton's law of additive pressures: *The pressure of a gas mixture is equal to the sum of the pressures each gas would exert if it existed alone at the mixture temperature and volume (Fig. 9–5).*

Amagat's law of additive volumes: *The volume of a gas mixture is equal to the sum of the volumes each gas would occupy if it existed alone at the mixture temperature and pressure (Fig. 9–6).*

FIGURE 9–6
Amagat's law of additive volumes for a mixture of two ideal gases.

Dalton's and Amagat's laws hold exactly for ideal-gas mixtures, but only approximately for real-gas mixtures. This is due to intermolecular forces that may be significant for real gases at high densities. For ideal gases, these two laws are identical and give identical results.

Dalton's and Amagat's laws can be expressed as follows:

Dalton's law:
$$P_m = \sum_{i=1}^{k} P_i(T_m, V_m) \quad \left. \begin{array}{l} \text{exact for ideal gases,} \\ \text{approximate} \\ \text{for real gases} \end{array} \right\} \tag{9–6}$$

Amagat's law:
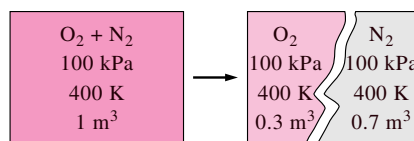$$V_m = \sum_{i=1}^{k} V_i(T_m, P_m) \tag{9–7}$$

FIGURE 9–7
The volume a component would occupy if it existed alone at the mixture *T* and *P* is called the *component volume* (for ideal gases, it is equal to the partial volume $y_i V_m$).

In these relations, P_i is called the **component pressure** and V_i is called the **component volume** (Fig. 9–7). Note that V_i is the volume a component *would* occupy if it existed alone at T_m and P_m, not the actual volume occupied by the component in the mixture. (In a vessel that holds a gas mixture, each component fills the entire volume of the vessel. Therefore, the volume of each component is equal to the volume of the vessel.) Also, the ratio P_i/P_m is called the **pressure fraction** and the ratio V_i/V_m is called the **volume fraction** of component *i*.

Ideal-Gas Mixtures

For ideal gases, P_i and V_i can be related to y_i by using the ideal-gas relation for both the components and the gas mixture:

$$\frac{P_i(T_m, V_m)}{P_m} = \frac{N_i R_u T_m / V_m}{N_m R_u T_m / V_m} = \frac{N_i}{N_m} = y_i$$

$$\frac{V_i(T_m, P_m)}{V_m} = \frac{N_i R_u T_m / P_m}{N_m R_u T_m / P_m} = \frac{N_i}{N_m} = y_i$$

Therefore,

$$\frac{P_i}{P_m} = \frac{V_i}{V_m} = \frac{N_i}{N_m} = y_i \tag{9–8}$$

Equation 9–8 is strictly valid for ideal-gas mixtures since it is derived by assuming ideal-gas behavior for the gas mixture and each of its components. The quantity $y_i P_m$ is called the **partial pressure** (identical to the *component pressure* for ideal gases), and the quantity $y_i V_m$ is called the **partial volume** (identical to the *component volume* for ideal gases). *Note that for an ideal-gas mixture, the mole fraction, the pressure fraction, and the volume fraction of a component are identical.*

The composition of an ideal-gas mixture (such as the exhaust gases leaving a combustion chamber) is frequently determined by a volumetric analysis (called the Orsat Analysis) and Eq. 9–8. A sample gas at a known volume, pressure, and temperature is passed into a vessel containing reagents that absorb one of the gases. The volume of the remaining gas is then measured at the original pressure and temperature. The ratio of the reduction in volume to the original volume (volume fraction) represents the mole fraction of that particular gas.

Real-Gas Mixtures

Dalton's law of additive pressures and Amagat's law of additive volumes can also be used for real gases, often with reasonable accuracy. This time, however, the component pressures or component volumes should be evaluated from relations that take into account the deviation of each component from ideal-gas behavior. One way of doing that is to use more exact equations of state (van der Waals, Beattie–Bridgeman, Benedict–Webb–Rubin, etc.) instead of the ideal-gas equation of state. Another way is to use the compressibility factor (Fig. 9–8) as
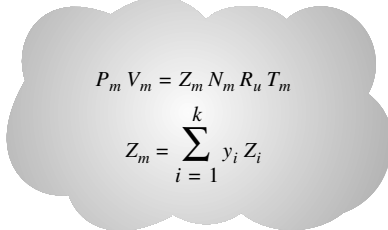
$$PV = ZNR_u T \tag{9–9}$$

The compressibility factor of the mixture Z_m can be expressed in terms of the compressibility factors of the individual gases Z_i by applying Eq. 9–9 to both sides of Dalton's law or Amagat's law expression and simplifying. We obtain

$$Z_m = \sum_{i=1}^{k} y_i Z_i \tag{9–10}$$

where Z_i is determined either at T_m and V_m (Dalton's law) or at T_m and P_m (Amagat's law) for each individual gas. It may seem that using either law will give the same result, but it does not.

The compressibility-factor approach, in general, gives more accurate results when the Z_i's in Eq. 9–10 are evaluated by using Amagat's law instead of

$$P_m V_m = Z_m N_m R_u T_m$$

$$Z_m = \sum_{i=1}^{k} y_i Z_i$$

FIGURE 9–8
One way of predicting the *P-v-T* behavior of a real-gas mixture is to use compressibility factors.

Dalton's law. This is because Amagat's law involves the use of mixture pressure P_m, which accounts for the influence of intermolecular forces between the molecules of different gases. Dalton's law disregards the influence of dissimilar molecules in a mixture on each other. As a result, it tends to underpredict the pressure of a gas mixture for a given V_m and T_m. Therefore, Dalton's law is more appropriate for gas mixtures at low pressures. Amagat's law is more appropriate at high pressures.

Note that there is a significant difference between using the compressibility factor for a single gas and for a mixture of gases. The compressibility factor predicts the P-υ-T behavior of single gases rather accurately, as discussed in Chap. 3, but not for mixtures of gases. When we use compressibility factors for the components of a gas mixture, we account for the influence of like molecules on each other; the influence of dissimilar molecules remains largely unaccounted for. Consequently, a property value predicted by this approach may be considerably different from the experimentally determined value.

9–3 ▪ PROPERTIES OF GAS MIXTURES: IDEAL AND REAL GASES

Consider a gas mixture that consists of 2 kg of N_2 and 3 kg of CO_2. The total mass (an *extensive property*) of this mixture is 5 kg. How did we do it? Well, we simply added the mass of each component. This example suggests a simple way of evaluating the **extensive properties** of a nonreacting ideal- or real-gas mixture: *Just add the contributions of each component of the mixture* (Fig. 9–9). Then the total internal energy, enthalpy, and entropy of a gas mixture can be expressed, respectively, as
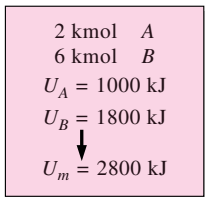
$$U_m = \sum_{i=1}^{k} U_i = \sum_{i=1}^{k} m_i u_i = \sum_{i=1}^{k} N_i \bar{u}_i \qquad \text{(kJ)} \qquad \text{(9–11)}$$

$$H_m = \sum_{i=1}^{k} H_i = \sum_{i=1}^{k} m_i h_i = \sum_{i=1}^{k} N_i \bar{h}_i \qquad \text{(kJ)} \qquad \text{(9–12)}$$

$$S_m = \sum_{i=1}^{k} S_i = \sum_{i=1}^{k} m_i s_i = \sum_{i=1}^{k} N_i \bar{s}_i \qquad \text{(kJ/K)} \qquad \text{(9–13)}$$

By following a similar logic, the changes in internal energy, enthalpy, and entropy of a gas mixture during a process can be expressed, respectively, as

$$\Delta U_m = \sum_{i=1}^{k} \Delta U_i = \sum_{i=1}^{k} m_i \Delta u_i = \sum_{i=1}^{k} N_i \Delta \bar{u}_i \qquad \text{(kJ)} \qquad \text{(9–14)}$$

$$\Delta H_m = \sum_{i=1}^{k} \Delta H_i = \sum_{i=1}^{k} m_i \Delta h_i = \sum_{i=1}^{k} N_i \Delta \bar{h}_i \qquad \text{(kJ)} \qquad \text{(9–15)}$$

$$\Delta S_m = \sum_{i=1}^{k} \Delta S_i = \sum_{i=1}^{k} m_i \Delta s_i = \sum_{i=1}^{k} N_i \Delta \bar{s}_i \qquad \text{(kJ/K)} \qquad \text{(9–16)}$$

Now reconsider the same mixture, and assume that both N_2 and CO_2 are at 25°C. The temperature (an *intensive* property) of the mixture is, as you would expect, also 25°C. Notice that we did not add the component temperatures to

| 2 kmol | A |
| 6 kmol | B |

$U_A = 1000$ kJ
$U_B = 1800$ kJ
↓
$U_m = 2800$ kJ

FIGURE 9–9
The extensive properties of a mixture are determined by simply adding the properties of the components.

determine the mixture temperature. Instead, we used some kind of averaging scheme, a characteristic approach for determining the **intensive properties** of a gas mixture. The internal energy, enthalpy, and entropy of a gas mixture *per unit mass* or *per unit mole* of the mixture can be determined by dividing the equations above by the mass or the mole number of the mixture (m_m or N_m). We obtain (Fig. 9–10)

$$u_m = \sum_{i=1}^{k} \mathrm{mf}_i u_i \quad \text{and} \quad \bar{u}_m = \sum_{i=1}^{k} y_i \bar{u}_i \qquad \text{(kJ/kg or kJ/kmol)} \qquad \textbf{(9–17)}$$

$$h_m = \sum_{i=1}^{k} \mathrm{mf}_i h_i \quad \text{and} \quad \bar{h}_m = \sum_{i=1}^{k} y_i \bar{h}_i \qquad \text{(kJ/kg or kJ/kmol)} \qquad \textbf{(9–18)}$$

$$s_m = \sum_{i=1}^{k} \mathrm{mf}_i s_i \quad \text{and} \quad \bar{s}_m = \sum_{i=1}^{k} y_i \bar{s}_i \qquad \text{(kJ/kg · K or kJ/kmol · K)} \qquad \textbf{(9–19)}$$

Similarly, the specific heats of a gas mixture can be expressed as

$$C_{v,m} = \sum_{i=1}^{k} \mathrm{mf}_i C_{v,i} \quad \text{and} \quad \overline{C}_{v,m} = \sum_{i=1}^{k} y_i \overline{C}_{v,i} \qquad \text{(kJ/kg · °C or kJ/kmol · °C)}$$
$$\textbf{(9–20)}$$

$$C_{p,m} = \sum_{i=1}^{k} \mathrm{mf}_i C_{p,i} \quad \text{and} \quad \overline{C}_{p,m} = \sum_{i=1}^{k} y_i \overline{C}_{p,i} \qquad \text{(kJ/kg · °C or kJ/kmol · °C)} \quad \textbf{(9–21)}$$

Notice that *properties per unit mass involve mass fractions* (mf_i) *and properties per unit mole involve mole fractions* (y_i).

The relations just given are exact for ideal-gas mixtures, and approximate for real-gas mixtures. (In fact, they are also applicable to nonreacting liquid and solid solutions especially when they form an "ideal solution.") The only major difficulty associated with these relations is the determination of properties for each individual gas in the mixture. The analysis can be simplified greatly, however, by treating the individual gases as an ideal gas, if doing so does not introduce a significant error.

FIGURE 9–10
The intensive properties of a mixture are determined by weighted averaging.

EXAMPLE 9–2   Mixing Two Ideal Gases in a Tank

An insulated rigid tank is divided into two compartments by a partition. One compartment contains 7 kg of oxygen gas at 40°C and 100 kPa, and the other compartment contains 4 kg of nitrogen gas at 20°C and 150 kPa. Now the partition is removed, and the two gases are allowed to mix. Determine (*a*) the mixture temperature and (*b*) the mixture pressure after equilibrium has been established.

SOLUTION   We take the entire contents of the tank (both compartments) as the system (Fig. 9–11). This is a *closed system* since no mass crosses the boundary during the process. We note that the volume of a rigid tank is constant and thus $v_2 = v_1$ and there is no boundary work done.
Assumptions   **1** We assume both gases to be ideal gases, and their mixture to be an ideal-gas mixture. This assumption is reasonable since both the oxygen and nitrogen are well above their critical temperatures and well below their critical pressures. **2** The tank is insulated and thus there is no heat transfer. **3** There are no other forms of work involved.
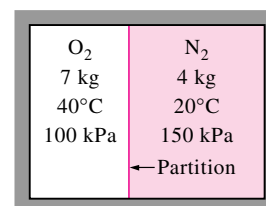

FIGURE 9–11
Schematic for Example 9–2.

Analysis (*a*) Noting that there is no energy transfer to or from the tank, the energy balance for the system can be expressed as

$$E_{in} - E_{out} = \Delta E_{system}$$

$$0 = \Delta U = \Delta U_{N_2} + \Delta U_{O_2}$$

$$[mC_v(T_m - T_1)]_{N_2} + [mC_v(T_m - T_1)]_{O_2} = 0$$

By using C_v values at room temperature (from Table A–2*a*), the final temperature of the mixture is determined to be

$$(4 \text{ kg})(0.743 \text{ kJ/kg} \cdot °C)(T_m - 20°C) + (7 \text{ kg})(0.658 \text{ kJ/kg} \cdot °C)(T_m - 40°C) = 0$$

$$T_m = \textbf{32.2°C}$$

(*b*) The final pressure of the mixture is determined from the ideal-gas relation

$$P_m V_m = N_m R_u T_m$$

where

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{7 \text{ kg}}{32 \text{ kg/kmol}} = 0.219 \text{ kmol}$$

$$N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{4 \text{ kg}}{28 \text{ kg/kmol}} = 0.143 \text{ kmol}$$

$$N_m = N_{O_2} + N_{N_2} = 0.219 + 0.143 = 0.362 \text{ kmol}$$

and

$$V_{O_2} = \left(\frac{NR_u T_1}{P_1}\right)_{O_2} = \frac{(0.219 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(313 \text{ K})}{100 \text{ kPa}} = 5.70 \text{ m}^3$$

$$V_{N_2} = \left(\frac{NR_u T_1}{P_1}\right)_{N_2} = \frac{(0.143 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(293 \text{ K})}{150 \text{ kPa}} = 2.32 \text{ m}^3$$

$$V_m = V_{O_2} + V_{N_2} = 5.70 + 2.32 = 8.02 \text{ m}^3$$

Thus,

$$P_m = \frac{N_m R_u T_m}{V_m} = \frac{(0.362 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(305.2 \text{ K})}{8.02 \text{ m}^3} = \textbf{114.5 kPa}$$

Discussion We could also determine the mixture pressure by using $P_m V_m = m_m R_m T_m$, where R_m is the apparent gas constant of the mixture. This would require a knowledge of mixture composition in terms of mass or mole fractions.

9–4 ▪ DRY AND ATMOSPHERIC AIR

Air is a mixture of nitrogen, oxygen, and small amounts of some other gases. Air in the atmosphere normally contains some water vapor (or *moisture*) and is referred to as **atmospheric air.** By contrast, air that contains no water vapor is called **dry air.** It is often convenient to treat air as a mixture of water vapor and dry air since the composition of dry air remains relatively constant, but the amount of water vapor changes as a result of condensation and evaporation from oceans, lakes, rivers, showers, and even the human body. Although the amount of water vapor in the air is small, it plays a major role in human comfort. Therefore, it is an important consideration in air-conditioning applications.

The temperature of air in air-conditioning applications ranges from about -10 to about 50°C. In this range, dry air can be treated as an ideal gas with a constant C_p value of 1.005 kJ/(kg · K) [0.240 Btu/(lbm · R)] with negligible error (under 0.2 percent), as illustrated in Fig. 9–12. Taking 0°C as the reference temperature, the enthalpy and enthalpy change of dry air can be determined from

$$h_{\text{dry air}} = C_p T = (1.005 \text{ kJ/kg} \cdot °C)T \qquad (\text{kJ/kg}) \qquad \textbf{(9–22a)}$$

and

$$\Delta h_{\text{dry air}} = C_p \, \Delta T = (1.005 \text{ kJ/kg} \cdot °C) \, \Delta T \qquad (\text{kJ/kg}) \qquad \textbf{(9–22b)}$$

where T is the air temperature in °C and ΔT is the change in temperature. In air-conditioning processes we are concerned with the *changes* in enthalpy Δh, which is independent of the reference point selected.

It certainly would be very convenient to also treat the water vapor in the air as an ideal gas and you would probably be willing to sacrifice some accuracy for such convenience. Well, it turns out that we can have the convenience without much sacrifice. At 50°C, the saturation pressure of water is 12.3 kPa. At pressures below this value, water vapor can be treated as an ideal gas with negligible error (under 0.2 percent), even when it is a saturated vapor. Therefore, water vapor in air behaves as if it existed alone and obeys the ideal-gas relation $Pv = RT$. Then the atmospheric air can be treated as an ideal-gas mixture whose pressure is the sum of the partial pressure of dry air* P_a and that of water vapor P_v:

$$P = P_a + P_v \qquad (\text{kPa}) \qquad \textbf{(9–23)}$$

The partial pressure of water vapor is usually referred to as the **vapor pressure.** It is the pressure water vapor would exert if it existed alone at the temperature and volume of atmospheric air.

Since water vapor is an ideal gas, the enthalpy of water vapor is a function of temperature only, that is, $h = h(T)$. This can also be observed from the *T-s* diagram of water given in Fig. A–9 and Fig. 9–13 where the constant-enthalpy lines coincide with constant-temperature lines at temperatures below 50°C. Therefore, *the enthalpy of water vapor in air can be taken to be equal to the enthalpy of saturated vapor at the same temperature.* That is,

$$h_v(T, \text{low } P) \cong h_g(T) \qquad \textbf{(9–24)}$$

The enthalpy of water vapor at 0°C is 2501.3 kJ/kg. The average C_p value of water vapor in the temperature range -10 to 50°C can be taken to be 1.82 kJ/kg · °C. Then the enthalpy of water vapor can be determined approximately from

$$h_g(T) \cong 2501.3 + 1.82T \qquad (\text{kJ/kg}) \qquad T \text{ in } °C \qquad \textbf{(9–25)}$$

or

$$h_g(T) \cong 1061.5 + 0.435T \qquad (\text{Btu/lbm}) \qquad T \text{ in } °F \qquad \textbf{(9–26)}$$

*Throughout this chapter, the subscript *a* will denote dry air and the subscript *v* will denote water vapor.

DRY AIR	
T, °C	C_p, kJ/kg · °C
−10	1.0038
0	1.0041
10	1.0045
20	1.0049
30	1.0054
40	1.0059
50	1.0065

FIGURE 9–12
The C_p of air can be assumed to be constant at 1.005 kJ/kg · °C in the temperature range -10 to 50°C with an error under 0.2 percent.

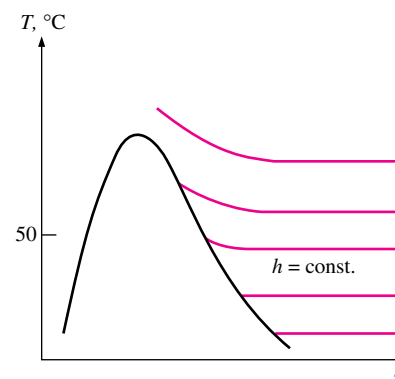**FIGURE 9–13**
At temperatures below 50°C, the $h =$ constant lines coincide with the $T =$ constant lines in the superheated vapor region of water.

WATER VAPOR		
	h_g, kJ/kg	Difference,
T, °C	Table A-4	Eq. 9–25	kJ/kg
−10	2482.9	2483.1	−0.2
0	2501.3	2501.3	0.0
10	2519.8	2519.5	0.3
20	2538.1	2537.7	0.4
30	2556.3	2555.9	0.4
40	2574.3	2574.1	0.2
50	2592.1	2592.3	−0.2

FIGURE 9–14
In the temperature range −10 to 50°C, the h_g of water can be determined from Eq. 9–25 with negligible error.

AIR
25°C, 100 kPa

$(P_{sat, H_2O @ 25°C} = 3.169 \text{ kPa})$

$P_v = 0 \rightarrow$ dry air
$P_v < 3.169$ kPa $\rightarrow$ unsaturated air
$P_v = 3.169$ kPa $\rightarrow$ saturated air

FIGURE 9–15
For saturated air, the vapor pressure is equal to the saturation pressure of water.

AIR
25°C, 1atm

$m_a = 1$ kg
$m_v = 0.01$ kg
$m_{v,max} = 0.02$ kg

Specific humidity: $\omega = 0.01 \dfrac{\text{kg } H_2O}{\text{kg dry air}}$

Relative humidity: $\phi = 50\%$

FIGURE 9–16
Specific humidity is the actual amount of water vapor in 1 kg of dry air, whereas relative humidity is the ratio of the actual amount of moisture in the air to the maximum amount of moisture air can hold at that temperature.

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in the temperature range −10 to 50°C (or 15 to 120°F), with negligible error, as shown in Fig. 9–14.

9–5 ▪ SPECIFIC AND RELATIVE HUMIDITY OF AIR

The amount of water vapor in the air can be specified in various ways. Probably the most logical way is to specify directly the mass of water vapor present in a unit mass of dry air. This is called **absolute** or **specific humidity** (also called *humidity ratio*) and is denoted by ω:

$$\omega = \frac{m_v}{m_a} \qquad \text{(kg water vapor/kg dry air)} \qquad \textbf{(9–27)}$$

The specific humidity can also be expressed as

$$\omega = \frac{m_v}{m_a} = \frac{P_v V/(R_v T)}{P_a V/(R_a T)} = \frac{P_v/R_v}{P_a/R_a} = 0.622 \frac{P_v}{P_a} \qquad \textbf{(9–28)}$$

or

$$\omega = \frac{0.622 P_v}{P - P_v} \qquad \text{(kg water vapor/kg dry air)} \qquad \textbf{(9–29)}$$

where P is the total pressure.

Consider 1 kg of dry air. By definition, dry air contains no water vapor, and thus its specific humidity is zero. Now let us add some water vapor to this dry air. The specific humidity will increase. As more vapor or moisture is added, the specific humidity will keep increasing until the air can hold no more moisture. At this point, the air is said to be saturated with moisture, and it is called **saturated air.** Any moisture introduced into saturated air will condense. The amount of water vapor in saturated air at a specified temperature and pressure can be determined from Eq. 9–29 by replacing P_v by P_g, the saturation pressure of water at that temperature (Fig. 9–15).

The amount of moisture in the air has a definite effect on how comfortable we feel in an environment. However, the comfort level depends more on the amount of moisture the air holds (m_v) relative to the maximum amount of moisture the air can hold at the same temperature (m_g). The ratio of these two quantities is called the **relative humidity** ϕ (Fig. 9–16)

$$\phi = \frac{m_v}{m_g} = \frac{P_v V/(R_v T)}{P_g V/(R_v T)} = \frac{P_v}{P_g} \qquad \textbf{(9–30)}$$

where

$$P_g = P_{sat @ T} \qquad \textbf{(9–31)}$$

Combining Eqs. 9–29 and 9–30, we can also express the relative humidity as

$$\phi = \frac{\omega P}{(0.622 + \omega)P_g} \qquad \text{and} \qquad \omega = \frac{0.622\phi P_g}{P - \phi P_g} \qquad \textbf{(9–32a, b)}$$

The relative humidity ranges from 0 for dry air to 1 for saturated air. Note that the amount of moisture air can hold depends on its temperature. Therefore, the relative humidity of air changes with temperature even when its specific humidity remains constant.

Atmospheric air is a mixture of dry air and water vapor, and thus the enthalpy of air is expressed in terms of the enthalpies of the dry air and the water vapor. In most practical applications, the amount of dry air in the air–water-vapor mixture remains constant, but the amount of water vapor changes. Therefore, the enthalpy of atmospheric air is expressed *per unit mass of dry air* instead of per unit mass of the air–water-vapor mixture.

The total enthalpy (an extensive property) of atmospheric air is the sum of the enthalpies of dry air and the water vapor:

$$H = H_a + H_v = m_a h_a + m_v h_v$$

Dividing by m_a gives

$$h = \frac{H}{m_a} = h_a + \frac{m_v}{m_a} h_v = h_a + \omega h_v$$

or

$$h = h_a + \omega h_g \qquad \text{(kJ/kg dry air)} \qquad \textbf{(9–33)}$$

since $h_v \cong h_g$ (Fig. 9–17).

Also note that the ordinary temperature of atmospheric air is frequently referred to as the **dry-bulb temperature** to differentiate it from other forms of temperatures that shall be discussed.

EXAMPLE 9–3   The Amount of Water Vapor in Room Air

A 5-m × 5-m × 3-m room shown in Fig. 9–18 contains air at 25°C and 100 kPa at a relative humidity of 75 percent. Determine (*a*) the partial pressure of dry air, (*b*) the specific humidity, (*c*) the enthalpy per unit mass of the dry air, and (*d*) the masses of the dry air and water vapor in the room.

SOLUTION   A sketch of the room is given in Fig. 9–18. Both the air and the vapor fill the entire room, and thus the volume of each gas is equal to the volume of the room.

Assumptions   The dry air and the water vapor in the room are ideal gases.

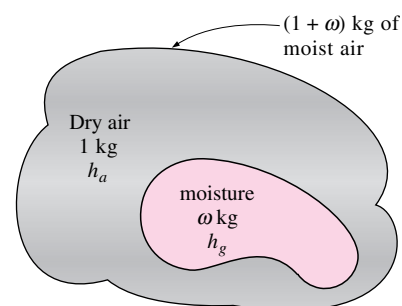***Analysis***   (*a*) The partial pressure of dry air can be determined from

$$P_a = P - P_{v+}$$

where

$$P_v = \phi P_g = \phi P_{\text{sat @ 25°C}} = (0.75)(3.169 \text{ kPa}) = 2.38 \text{ kPa}$$

Thus,

$$P_a = (100 - 2.38) \text{ kPa} = \textbf{97.62 kPa}$$
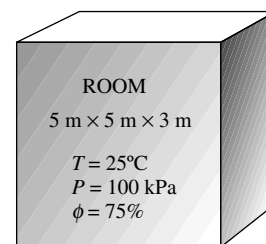
(*b*) The specific humidity of air is

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(2.38 \text{ kPa})}{(100 - 2.38) \text{ kPa}} = \textbf{0.0152 kg H}_2\textbf{O/kg dry air}$$

$h = h_a + \omega h_g$, kJ/kg dry air

FIGURE 9–17
The enthalpy of moist (atmospheric) air is expressed per unit mass of dry air, not per unit mass of moist air.

ROOM
5 m × 5 m × 3 m

$T = 25$°C
$P = 100$ kPa
$\phi = 75\%$

FIGURE 9–18
Schematic for Example 9–3.

(c) The enthalpy of air per unit mass of dry air is determined from Eq. 9–33, where h_g is taken from Table A–4:

$$h = h_a + \omega h_v \cong C_p T + \omega h_g$$
$$= (1.005 \text{ kJ/kg} \cdot °\text{C})(25°\text{C}) + (0.0152)(2547.2 \text{ kJ/kg})$$
$$= \textbf{63.8 kJ/kg dry air}$$

The enthalpy of water vapor (2547.2 kJ/kg) could also be determined from the approximation given by Eq. 9–25:

$$h_{g \text{ @ } 25°C} \cong 2501.3 + 1.82(25) = 2546.8 \text{ kJ/kg}$$

which is very close to the value obtained from Table A–4.

(d) Both the dry air and the water vapor fill the entire room completely. Therefore, the volume of each gas is equal to the volume of the room:

$$V_a = V_v = V_{\text{room}} = (5)(5)(3) = 75 \text{ m}^3$$

The masses of the dry air and the water vapor are determined from the ideal-gas relation applied to each gas separately:

$$m_a = \frac{P_a V_a}{R_a T} = \frac{(97.62 \text{ kPa})(75 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = \textbf{85.61 kg}$$
$$m_v = \frac{P_v V_v}{R_v T} = \frac{(2.38 \text{ kPa})(75 \text{ m}^3)}{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = \textbf{1.3 kg}$$

The mass of the water vapor in the air could also be determined from Eq. 9–27:

$$m_v = \omega m_a = (0.0152)(85.61 \text{ kg}) = 1.3 \text{ kg}$$

9–6 ▪ DEW-POINT TEMPERATURE

If you live in a humid area, you are probably used to waking up most summer mornings and finding the grass wet. You know it did not rain the night before. So what happened? Well, the excess moisture in the air simply condensed on the cool surfaces, forming what we call *dew*. In summer, a considerable amount of water vaporizes during the day. As the temperature falls during the night, so does the "moisture capacity" of air, which is the maximum amount of moisture air can hold. (What happens to the relative humidity during this process?) After a while, the moisture capacity of air equals its moisture content. At this point, air is saturated, and its relative humidity is 100 percent. Any further drop in temperature results in the condensation of some of the moisture, and this is the beginning of dew formation.

The **dew-point temperature** T_{dp} is defined as *the temperature at which condensation begins when the air is cooled at constant pressure.* In other words, T_{dp} is the saturation temperature of water corresponding to the vapor pressure:

$$T_{\text{dp}} = T_{\text{sat @ } P_v} \tag{9–34}$$

This is also illustrated in Fig. 9–19. As the air cools at constant pressure, the vapor pressure P_v remains constant. Therefore, the vapor in the air (state 1)
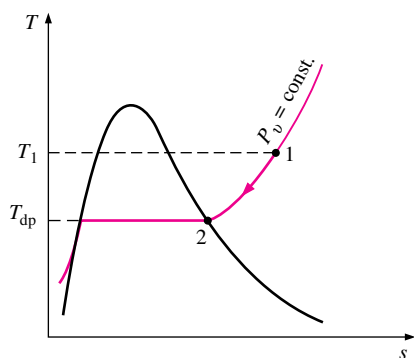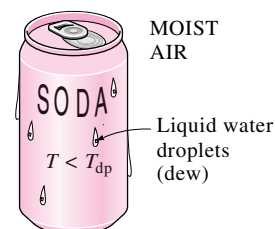

FIGURE 9–19
Constant-pressure cooling of moist air and the dew-point temperature on the *T-s* diagram of water.

undergoes a constant-pressure cooling process until it strikes the saturated vapor line (state 2). The temperature at this point is T_{dp}, and if the temperature drops any further, some vapor condenses out. As a result, the amount of vapor in the air decreases, which results in a decrease in P_v. The air remains saturated during the condensation process and thus follows a path of 100 percent relative humidity (the saturated vapor line). The ordinary temperature and the dew-point temperature of saturated air are identical.

You have probably noticed that when you buy a cold canned drink from a vending machine on a hot and humid day, dew forms on the can. The formation of dew on the can indicates that the temperature of the drink is below the dew-point temperature of the surrounding air (Fig. 9–20).

The dew-point temperature of room air can be determined easily by cooling some water in a metal cup by adding small amounts of ice and stirring. The temperature of the outer surface of the cup when dew starts to form on the surface is the dew-point temperature of the air.

FIGURE 9–20
When the temperature of a cold drink is below the dew-point temperature of the surrounding air, it "sweats."

EXAMPLE 9–4   Fogging of the Windows in a House

In cold weather, condensation frequently occurs on the inner surfaces of the windows due to the lower air temperatures near the window surface. Consider a house, shown in Fig. 9–21, that contains air at 20°C and 75 percent relative humidity. At what window temperature will the moisture in the air start condensing on the inner surfaces of the windows?

SOLUTION   The interior of a house is maintained at a specified temperature and humidity. The window temperature at which fogging will start is to be determined.
Analysis   The temperature distribution in a house, in general, is not uniform. When the outdoor temperature drops in winter, so does the indoor temperature near the walls and the windows. Therefore, the air near the walls and the windows remains at a lower temperature than at the inner parts of a house even though the total pressure and the vapor pressure remain constant throughout the house. As a result, the air near the walls and the windows will undergo a P_v = constant cooling process until the moisture in the air starts condensing. This will happen when the air reaches its dew-point temperature T_{dp}. The dew point is determined from Eq. 9–34 to be
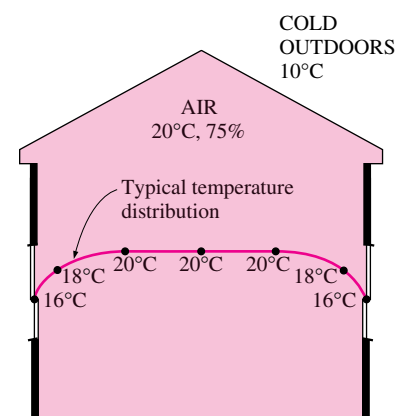
$$T_{dp} = T_{\text{sat } @ P_v}$$

where

$$P_v = \phi P_{g \, @ \, 20°C} = (0.75)(2.339 \text{ kPa}) = 1.754 \text{ kPa}$$

Thus,

$$T_{dp} = T_{\text{sat } @ \, 1.754 \text{ kPa}} = \mathbf{15.4°C}$$

Therefore, the inner surface of the window should be maintained above 15.4°C if condensation on the window surfaces is to be avoided.
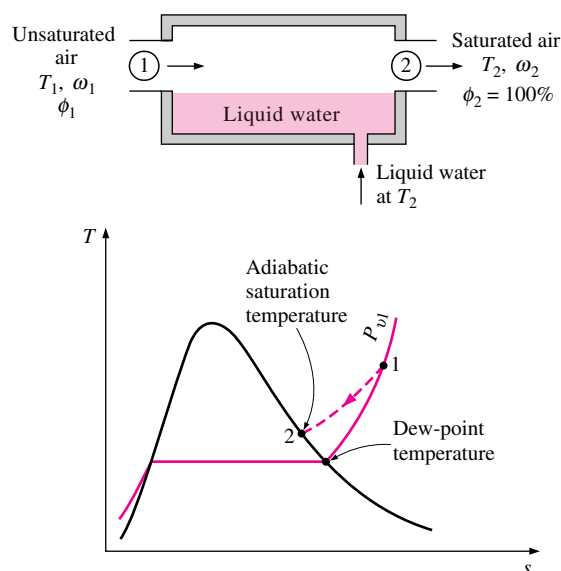

FIGURE 9–21
Schematic for Example 9–4.

9–7 ▪ ADIABATIC SATURATION AND WET-BULB TEMPERATURES

Relative humidity and specific humidity are frequently used in engineering and atmospheric sciences, and it is desirable to relate them to easily measurable quantities such as temperature and pressure. One way of determining the

FIGURE 9–22
The adiabatic saturation process and
its representation on a *T-s* diagram
of water.

relative humidity is to determine the dew-point temperature of air, as discussed in the last section. Knowing the dew-point temperature, we can determine the vapor pressure P_v and thus the relative humidity. This approach is simple, but not quite practical.

Another way of determining the absolute or relative humidity is related to an *adiabatic saturation process,* shown schematically and on a *T-s* diagram in Fig. 9–22. The system consists of a long insulated channel that contains a pool of water. A steady stream of unsaturated air that has a specific humidity of ω_1 (unknown) and a temperature of T_1 is passed through this channel. As the air flows over the water, some water will evaporate and mix with the airstream. The moisture content of air will increase during this process, and its temperature will decrease, since part of the latent heat of vaporization of the water that evaporates will come from the air. If the channel is long enough, the airstream will exit as saturated air ($\phi = 100$ percent) at temperature T_2, which is called the **adiabatic saturation temperature.**

If makeup water is supplied to the channel at the rate of evaporation at temperature T_2, the adiabatic saturation process described above can be analyzed as a steady-flow process. The process involves no heat or work interactions, and the kinetic and potential energy changes can be neglected. Then the conservation of mass and conservation of energy relations for this two-inlet, one-exit steady-flow system reduces to the following:

Mass balance:

$$\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a \qquad \text{(The mass flow rate of dry air remains constant)}$$

$$\dot{m}_{w_1} + \dot{m}_f = \dot{m}_{w_2} \qquad \text{(The mass flow rate of vapor in the air increases by an amount equal}$$
or
$$\text{to the rate of evaporation } \dot{m}_f)$$

$$\dot{m}_a\omega_1 + \dot{m}_f = \dot{m}_a\omega_2$$

Thus,

$$\dot{m}_f = \dot{m}_a(\omega_2 - \omega_1)$$

Energy balance:

$$\dot{E}_{in} = \dot{E}_{out} \qquad (\text{since } \dot{Q} = 0 \text{ and } \dot{W} = 0)$$

$$\dot{m}_a h_1 + \dot{m}_f h_{f_2} = \dot{m}_a h_2$$

or

$$\dot{m}_a h_1 + \dot{m}_a(\omega_2 - \omega_1)h_{f_2} = \dot{m}_a h_2$$

Dividing by $\dot{m}_a$ gives

$$h_1 + (\omega_2 - \omega_1)h_{f_2} = h_2$$

or

$$(C_p T_1 + \omega_1 h_{g_1}) + (\omega_2 - \omega_1)h_{f_2} = (C_p T_2 + \omega_2 h_{g_2})$$

which yields

$$\omega_1 = \frac{C_p(T_2 - T_1) + \omega_2 h_{fg_2}}{h_{g_1} - h_{f_2}} \qquad \text{(9–35)}$$

where, from Eq. 9–32*b*,

$$\omega_2 = \frac{0.622 P_{g_2}}{P_2 - P_{g_2}} \qquad \text{(9–36)}$$

since $\phi_2 = 100$ percent. Thus we conclude that the specific humidity (and relative humidity) of air can be determined from Eqs. 9–35 and 9–36 by measuring the pressure and temperature of air at the inlet and the exit of an adiabatic saturator.

If the air entering the channel is already saturated, then the adiabatic saturation temperature T_2 will be identical to the inlet temperature T_1, in which case Eq. 9–35 yields $\omega_1 = \omega_2$. In general, the adiabatic saturation temperature will be between the inlet and dew-point temperatures.

The adiabatic saturation process discussed above provides a means of determining the absolute or relative humidity of air, but it requires a long channel or a spray mechanism to achieve saturation conditions at the exit. A more practical approach is to use a thermometer whose bulb is covered with a cotton wick saturated with water and to blow air over the wick, as shown in Fig. 9–23. The temperature measured in this matter is called the **wet-bulb temperature** T_{wb}, and it is commonly used in air-conditioning applications.

The basic principle involved is similar to that in adiabatic saturation. When unsaturated air passes over the wet wick, some of the water in the wick evaporates. As a result, the temperature of the water drops, creating a temperature difference (which is the driving force for heat transfer) between the air and the water. After a while, the heat loss from the water by evaporation equals the heat gain from the air, and the water temperature stabilizes. The thermometer reading at this point is the wet-bulb temperature. The wet-bulb temperature
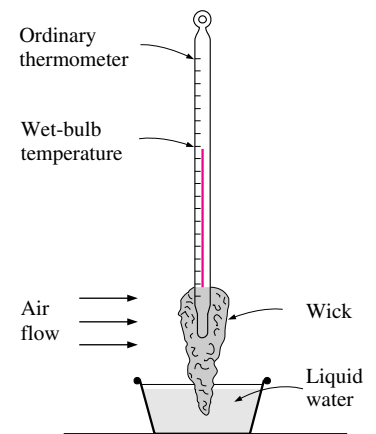
FIGURE 9–23
A simple arrangement to measure the wet-bulb temperature.

Wet-bulb thermometer

Dry-bulb thermometer

Wet-bulb thermometer wick
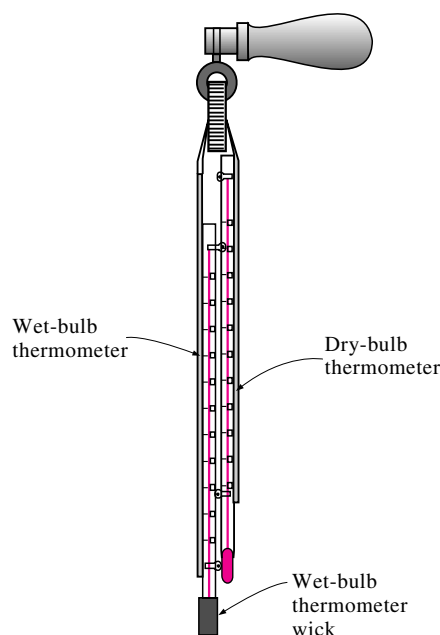
FIGURE 9–24
Sling psychrometer.

can also be measured by placing the wet-wicked thermometer in a holder attached to a handle and rotating the holder rapidly, that is, by moving the thermometer instead of the air. A device that works on this principle is called a *sling psychrometer* and is shown in Fig. 9–24. Usually a dry-bulb thermometer is also mounted on the frame of this device so that both the wet- and dry-bulb temperatures can be read simultaneously.

Advances in electronics made it possible to measure humidity directly in a fast and reliable way. It appears that sling psychrometers and wet-wicked thermometers are about to become things of the past. Today, hand-held electronic humidity measurement devices based on the capacitance change in a thin polymer film as it absorbs water vapor are capable of sensing and digitally displaying the relative humidity within 1 percent accuracy in a matter of seconds.

In general, the adiabatic saturation temperature and the wet-bulb temperature are not the same. However, for air–water-vapor mixtures at atmospheric pressure, the wet-bulb temperature happens to be approximately equal to the adiabatic saturation temperature. Therefore, the wet-bulb temperature T_{wb} can be used in Eq. 9–14 in place of T_2 to determine the specific humidity of air.

EXAMPLE 9–5   **The Specific and Relative Humidity of Air**

The dry- and the wet-bulb temperatures of atmospheric air at 1 atm (101.325 kPa) pressure are measured with a sling psychrometer and determined to be 25 and 15°C, respectively. Determine (*a*) the specific humidity, (*b*) the relative humidity, and (*c*) the enthalpy of the air.

SOLUTION   Dry- and wet-bulb temperatures are given. The specific humidity, relative humidity, and enthalpy are to be determined.

Analysis   (*a*) The specific humidity ω_1 is determined from Eq. 9–35,

$$\omega_1 = \frac{C_p(T_2 - T_1) + \omega_2 h_{fg_2}}{h_{g_1} - h_{f_2}}$$

where T_2 is the wet-bulb temperature and ω_2 is

$$\omega_2 = \frac{0.622 P_{g_2}}{P_2 - P_{g_2}} = \frac{(0.622)(1.705 \text{ kPa})}{(101.325 - 1.705)\text{ kPa}}$$
$$= 0.01065 \text{ kg } H_2O/\text{kg dry air}$$

Thus,

$$\omega_1 = \frac{(1.005 \text{ kJ/kg} \cdot °C)[(15 - 25)°C] + (0.01065)(2465.9 \text{ kJ/kg})}{(2547.2 - 62.99)\text{ kJ/kg}}$$

$$= \textbf{0.00653 kg } \mathbf{H_2O}\textbf{/kg dry air}$$

(*b*) The relative humidity ϕ_1 is determined from Eq. 9–32*a* to be

$$\phi_1 = \frac{\omega_1 P_2}{(0.622 + \omega_1)P_{g_1}} = \frac{(0.00653)(101.325 \text{ kPa})}{(0.622 + 0.00653)(3.169 \text{ kPa})} = \textbf{0.332 or 33.2\%}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from Eq. 9–33:

$$h_1 = h_{a_1} + \omega_1 h_{v_1} \cong C_p T_1 + \omega_1 h_{g_1}$$
$$h_1 = [1.005 \text{ kJ/(kg} \cdot °C)](25°C) + (0.00653)(2547.2 \text{ kJ/kg})$$
$$= \textbf{41.8 kJ/kg dry air}$$

9–8 ▪ THE PSYCHROMETRIC CHART

The state of the atmospheric air at a specified pressure is completely speci-fied by two independent intensive properties. The rest of the properties can be calculated easily from the relations already given. The sizing of a typical air-conditioning system involves numerous such calculations, which may eventually get on the nerves of even the most patient engineers. Therefore, there is clear motivation to computerize calculations or to do these calcula-tions once and to present the data in the form of easily readable charts. Such charts are called **psychrometric charts,** and they are used extensively in air-conditioning applications. A psychrometric chart for a pressure of 1 atm (101.325 kPa or 14.696 psia) is given in Fig. A–33 in SI units and in Fig. A–33E in English units. Psychrometric charts at other pressures (for use at considerably higher elevations than sea level) are also available.

The basic features of the psychrometric chart are illustrated in Fig. 9–25. The dry-bulb temperatures are shown on the horizontal axis, and the specific humidity is shown on the vertical axis. (Some charts also show the vapor pres-sure on the vertical axis since at a fixed total pressure P there is a one-to-one correspondence between the specific humidity ω and the vapor pressure P_v, as can be seen from Eq. 9–29.) On the left end of the chart, there is a curve (called the *saturation line*) instead of a straight line. All the saturated air states are lo-cated on this curve. Therefore, it is also the curve of 100 percent relative hu-midity. Other constant relative-humidity curves have the same general shape.

Lines of constant wet-bulb temperature have a downhill appearance to the right. Lines of constant specific volume (in m^3/kg dry air) look similar, except they are steeper. Lines of constant enthalpy (in kJ/kg dry air) lie very nearly parallel to the lines of constant wet-bulb temperature. Therefore, the constant-wet-bulb-temperature lines are used as constant-enthalpy lines in some charts.

For saturated air, the dry-bulb, wet-bulb, and dew-point temperatures are identical (Fig. 9–26). Therefore, the dew-point temperature of atmospheric air at any point on the chart can be determined by drawing a horizontal line (a line of ω = constant or P_v = constant) from the point to the saturated curve. The temperature value at the intersection point is the dew-point temperature.

The psychrometric chart also serves as a valuable aid in visualizing the air-conditioning processes. An ordinary heating or cooling process, for example, will appear as a horizontal line on this chart if no humidification or dehumidi-fication is involved (that is, ω = constant). Any deviation from a horizontal line indicates that moisture is added or removed from the air during the process.
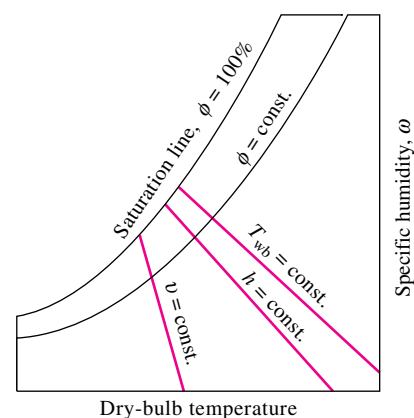

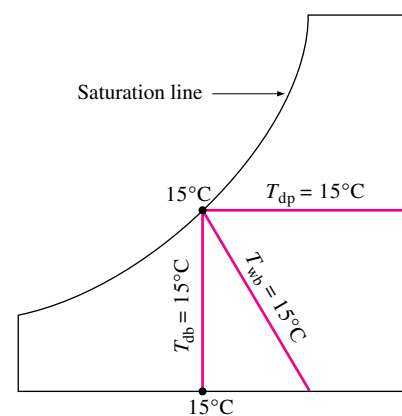**FIGURE 9–25**
Schematic for a psychrometric chart.

FIGURE 9–26
For saturated air, the dry-bulb, wet-bulb, and dew-point temperatures are identical.

EXAMPLE 9–6   The Use of the Psychrometric Chart

Consider a room that contains air at 1 atm, 35°C, and 40 percent relative hu-midity. Using the psychrometric chart, determine (a) the specific humidity, (b) the enthalpy, (c) the wet-bulb temperature, (d) the dew-point temperature, and (e) the specific volume of the air.

SOLUTION   At a given total pressure, the state of atmospheric air is completely specified by two independent properties such as the dry-bulb temperature and the relative humidity. Other properties are determined by directly reading their values at the specified state.
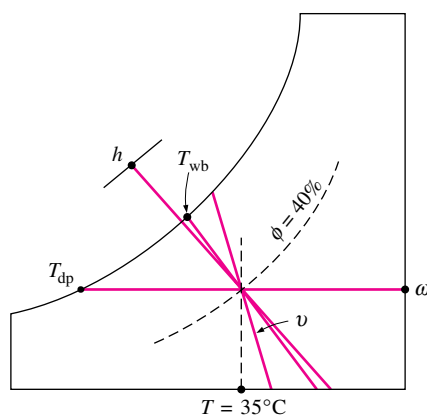
FIGURE 9–27
Schematic for Example 9–6.

Analysis (*a*) The specific humidity is determined by drawing a horizontal line from the specified state to the right until it intersects with the ω axis, as shown in Fig. 9–27. At the intersection point we read

$$\omega = \textbf{0.0142 kg H}_2\textbf{O/kg dry air}$$

(*b*) The enthalpy of air per unit mass of dry air is determined by drawing a line parallel to the h = constant lines from the specific state until it intersects the enthalpy scale. At the intersection point we read

$$h = \textbf{71.5 kJ/kg dry air}$$

(*c*) The wet-bulb temperature is determined by drawing a line parallel to the T_{wb} = constant lines from the specified state until it intersects the saturation line. At the intersection point we read

$$T_{wb} = \textbf{24°C}$$

(*d*) The dew-point temperature is determined by drawing a horizontal line from the specified state to the left until it intersects the saturation line. At the intersection point we read

$$T_{dp} = \textbf{19.4°C}$$

(*e*) The specific volume per unit mass of dry air is determined by noting the distances between the specified state and the v = constant lines on both sides of the point. The specific volume is determined by visual interpolation to be

$$v = \textbf{0.893 m}^3\textbf{/kg dry air}$$

9–9 ▪ HUMAN COMFORT AND AIR-CONDITIONING

Human beings have an inherent weakness—they want to feel comfortable. They want to live in an environment that is neither hot nor cold, neither humid nor dry. However, comfort does not come easily since the desires of the human body and the weather usually are not quite compatible. Achieving comfort requires a constant struggle against the factors that cause discomfort, such as high or low temperatures and high or low humidity. As engineers, it is our duty to help people feel comfortable. (Besides, it keeps us employed.)

It did not take long for people to realize that they could not change the weather in an area. All they can do is change it in a confined space such as a house or a workplace (Fig. 9–28). In the past, this was partially accomplished by fire and simple indoor heating systems. Today, modern air-conditioning systems can heat, cool, humidify, dehumidify, clean, and even deodorize the air—in other words, *condition* the air to peoples' desires. Air-conditioning systems are designed to *satisfy* the needs of the human body; therefore, it is essential that we understand the thermodynamic aspects of the body.

The human body can be viewed as a heat engine whose energy input is food. As with any other heat engine, the human body generates waste heat that must be rejected to the environment if the body is to continue operating. The rate of heat generation depends on the level of the activity. For an average
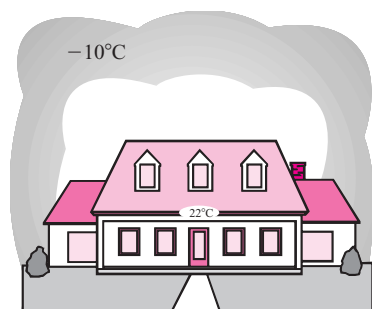

FIGURE 9–28
We cannot change the weather, but we can change the climate in a confined space by air-conditioning.

adult male, it is about 87 W when sleeping, 115 W when resting or doing office work, 230 W when bowling, and 440 W when doing heavy physical work. The corresponding numbers for an adult female are about 15 percent less. (This difference is due to the body size, not the body temperature. The deep-body temperature of a healthy person is maintained constant at 37°C.) A body will feel comfortable in environments in which it can dissipate this waste heat comfortably (Fig. 9–29).

Heat transfer is proportional to the temperature difference. Therefore in cold environments, a body will lose more heat than it normally generates, which results in a feeling of discomfort. The body tries to minimize the energy deficit by cutting down the blood circulation near the skin (causing a pale look). This lowers the skin temperature, which is about 34°C for an average person, and thus the heat transfer rate. A low skin temperature causes discomfort. The hands, for example, feel painfully cold when the skin temperature reaches 10°C (50°F). We can also reduce the heat loss from the body either by putting barriers (additional clothes, blankets, etc.) in the path of heat or by increasing the rate of heat generation within the body by exercising. For example, the comfort level of a resting person dressed in warm winter clothing in a room at 10°C (50°F) is roughly equal to the comfort level of an identical person doing moderate work in a room at about −23°C (–10°F). Or we can just cuddle up and put our hands between our legs to reduce the surface area through which heat flows.

In hot environments, we have the opposite problem—we do not seem to be dissipating enough heat from our bodies, and we feel as if we are going to burst. We dress lightly to make it easier for heat to get away from our bodies, and we reduce the level of activity to minimize the rate of waste heat generation in the body. We also turn on the fan to continuously replace the warmer air layer that forms around our bodies as a result of body heat by the cooler air in other parts of the room. When doing light work or walking slowly, about half of the rejected body heat is dissipated through perspiration as *latent heat* while the other half is dissipated through convection and radiation as *sensible heat.* When resting or doing office work, most of the heat (about 70 percent) is dissipated in the form of sensible heat whereas when doing heavy physical work, most of the heat (about 60 percent) is dissipated in the form of latent heat. The body helps out by perspiring or sweating more. As this sweat evaporates, it absorbs latent heat from the body and cools it. Perspiration is not much help, however, if the relative humidity of the environment is close to 100 percent. Prolonged sweating without any fluid intake will cause dehydration and reduced sweating, which may lead to a rise in body temperature and a heat stroke.

Another important factor that affects human comfort is heat transfer by radiation between the body and the surrounding surfaces such as walls and windows. The sun's rays travel through space by radiation. You warm up in front of a fire even if the air between you and the fire is quite cold. Likewise, in a warm room you will feel chilly if the ceiling or the wall surfaces are at a considerably lower temperature. This is due to direct heat transfer between your body and the surrounding surfaces by radiation. Radiant heaters are commonly used for heating hard-to-heat places such as car repair shops.

The comfort of the human body depends primarily on three factors: the (dry-bulb) temperature, relative humidity, and air motion (Fig. 9–30). The temperature of the environment is the single most important index of comfort.
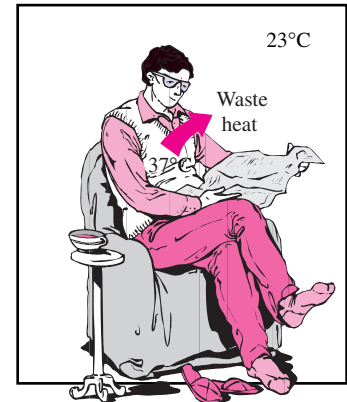
FIGURE 9–29
A body feels comfortable when it can freely dissipate its waste heat, and no more.
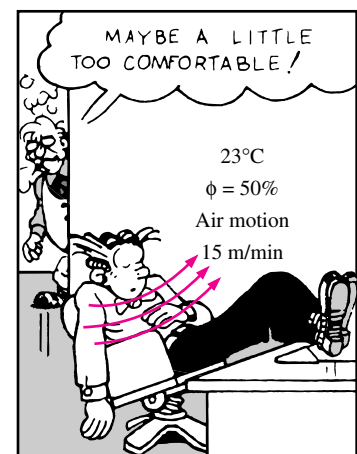
FIGURE 9–30
A comfortable environment.
(Reprinted with special permission of King Features Syndicate.)

Most people feel comfortable when the environment temperature is between 22 and 27°C (72 and 80°F). The relative humidity also has a considerable effect on comfort since it affects the amount of heat a body can dissipate through evaporation. Relative humidity is a measure of air's ability to absorb more moisture. High relative humidity slows down heat rejection by evapo-ration, and low relative humidity speeds it up. Most people prefer a relative humidity of 40 to 60 percent.

Air motion also plays an important role in human comfort. It removes the warm, moist air that builds up around the body and replaces it with fresh air. Therefore, air motion improves heat rejection by both convection and evapo-ration. Air motion should be strong enough to remove heat and moisture from the vicinity of the body, but gentle enough to be unnoticed. Most people feel comfortable at an airspeed of about 15 m/min. Very-high-speed air motion causes discomfort instead of comfort. For example, an environment at 10°C (50°F) with 48 km/h winds feels as cold as an environment at −7°C (20°F) with 3 km/h winds as a result of the body-chilling effect of the air motion (the *windchill factor*). Other factors that affect comfort are air cleanliness, odor, noise, and radiation effect.

9–10 ▪ AIR-CONDITIONING PROCESSES

Maintaining a living space or an industrial facility at the desired temperature and humidity requires some processes called air-conditioning processes. These processes include *simple heating* (raising the temperature), *simple cool-ing* (lowering the temperature), *humidifying* (adding moisture), and *dehumid-ifying* (removing moisture). Sometimes two or more of these processes are needed to bring the air to a desired temperature and humidity level.

Various air-conditioning processes are illustrated on the psychrometric chart in Fig. 9–31. Notice that simple heating and cooling processes appear as hori-zontal lines on this chart since the moisture content of the air remains constant (ω = constant) during these processes. Air is commonly heated and humidified in winter and cooled and dehumidified in summer. Notice how these processes appear on the psychrometric chart.

Most air-conditioning processes can be modeled as steady-flow processes, and thus the *mass balance* relation $\dot{m}_{in} = \dot{m}_{out}$ can be expressed for *dry air* and *water* as
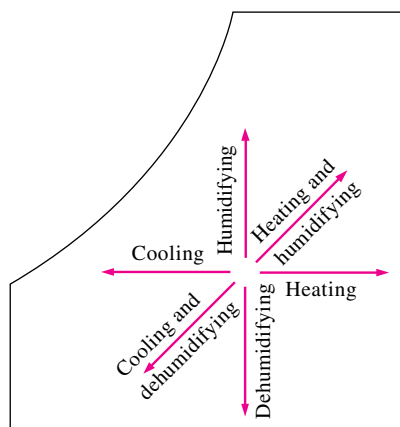

FIGURE 9–31
Various air-conditioning processes.

Mass balance for dry air: $\quad \sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \quad$ (kg/s) **(9–37)**

Mass balance for water: $\quad \sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \quad$ or $\quad \sum \dot{m}_{a,i}\omega_i = \sum \dot{m}_{a,e}\omega_e$ **(9–38)**

where the subscripts *i* and *e* denote the inlet and the exit states, respectively. Disregarding the kinetic and potential energy changes, the *steady-flow energy balance* relation $\dot{E}_{in} = \dot{E}_{out}$ can be expressed in this case as

$$\dot{Q}_{in} + \dot{W}_{in} + \sum \dot{m}_i h_i = \dot{Q}_{out} + \dot{W}_{out} + \sum \dot{m}_e h_e \qquad \text{(9–39)}$$

The work term usually consists of the *fan work input,* which is small relative to the other terms in the energy balance relation. Next we examine some com-monly encountered processes in air-conditioning.

Simple Heating and Cooling (ω = constant)

Many residential heating systems consist of a stove, a heat pump, or an electric resistance heater. The air in these systems is heated by circulating it through a duct that contains the tubing for the hot gases or the electric resistance wires, as shown in Fig. 9–32. The amount of moisture in the air remains constant during this process since no moisture is added to or removed from the air. That is, the specific humidity of the air remains constant (ω = constant) during a heating (or cooling) process with no humidification or dehumidification. Such a heating process will proceed in the direction of increasing dry-bulb temperature following a line of constant specific humidity on the psychrometric chart, which appears as a horizontal line.

Notice that the relative humidity of air decreases during a heating process even if the specific humidity ω remains constant. This is because the relative humidity is the ratio of the moisture content to the moisture capacity of air at the same temperature, and moisture capacity increases with temperature. Therefore, the relative humidity of heated air may be well below comfortable levels, causing dry skin, respiratory difficulties, and an increase in static electricity.

A cooling process at constant specific humidity is similar to the heating process discussed above, except the dry-bulb temperature decreases and the relative humidity increases during such a process, as shown in Fig. 9–33. Cooling can be accomplished by passing the air over some coils through which a refrigerant or chilled water flows.

The conservation of mass equations for a heating or cooling process that involves no humidification or dehumidification reduce to $\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a$ for dry air and $\omega_1 = \omega_2$ for water. Neglecting any fan work that may be present, the conservation of energy equation in this case reduces to

$$\dot{Q} = \dot{m}_a(h_2 - h_1) \quad \text{or} \quad q = h_2 - h_1$$

where h_1 and h_2 are enthalpies per unit mass of dry air at the inlet and the exit of the heating or cooling section, respectively.

Heating with Humidification

Problems associated with the low relative humidity resulting from simple heating can be eliminated by humidifying the heated air. This is accomplished by passing the air first through a heating section (process 1-2) and then through a humidifying section (process 2-3), as shown in Fig. 9–34.

The location of state 3 depends on how the humidification is accomplished. If steam is introduced in the humidification section, this will result in humidification with additional heating ($T_3 > T_2$). If humidification is accomplished by spraying water into the airstream instead, part of the latent heat of vaporization will come from the air, which will result in the cooling of the heated airstream ($T_3 < T_2$). Air should be heated to a higher temperature in the heating section in this case to make up for the cooling effect during the humidification process.

FIGURE 9–32
During simple heating, specific humidity remains constant, but relative humidity decreases.

FIGURE 9–33
During simple cooling, specific humidity remains constant, but relative humidity increases.

FIGURE 9–34
Heating with humidification.

EXAMPLE 9–7   Heating and Humidification of Air

An air-conditioning system is to take in outdoor air at 10°C and 30 percent relative humidity at a steady rate of 45 m³/min and to condition it to 25°C and
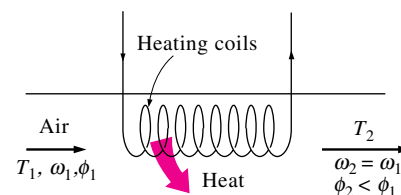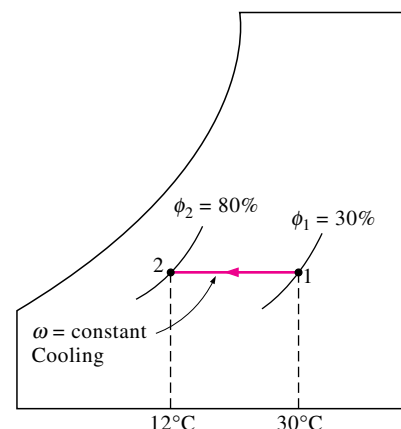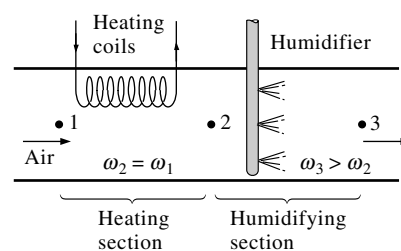
60 percent relative humidity. The outdoor air is first heated to 22°C in the heat-
ing section and then humidified by the injection of hot steam in the humidify-
ing section. Assuming the entire process takes place at a pressure of 100 kPa,
determine (*a*) the rate of heat supply in the heating section and (*b*) the mass
flow rate of the steam required in the humidifying section.

SOLUTION We will take the system to be the *heating* or the *humidifying sec-
tion,* as appropriate. The schematic of the system and the psychrometric chart
of the process are shown in Fig. 9–35. We note that the amount of water vapor
in the air remains constant in the heating section ($\omega_1 = \omega_2$) but increases in
the humidifying section ($\omega_3 > \omega_2$).

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry
air remains constant during the entire process. **2** Dry air and water vapor are
ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) Applying the mass and energy balances on the heating section
gives

Dry air mass balance: $\quad\quad\quad\quad\quad \dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a$

Water mass balance: $\quad\quad\quad\quad \dot{m}_{a_1}\omega_1 = \dot{m}_{a_2}\omega_2 \;\rightarrow\; \omega_1 = \omega_2$

Energy: $\quad\quad\quad\quad\quad\quad \dot{Q}_{in} + \dot{m}_a h_1 = \dot{m}_a h_2 \;\rightarrow\; \dot{Q}_{in} = \dot{m}_a(h_2 - h_1)$

The psychrometric chart offers great convenience in determining the properties
of moist air. However, its use is limited to a specified pressure only, which is
1 atm (101.325 kPa) for the one given in the appendix. At pressures other than
1 atm, either other charts for that pressure or the relations developed earlier
should be used. In our case, the choice is clear:
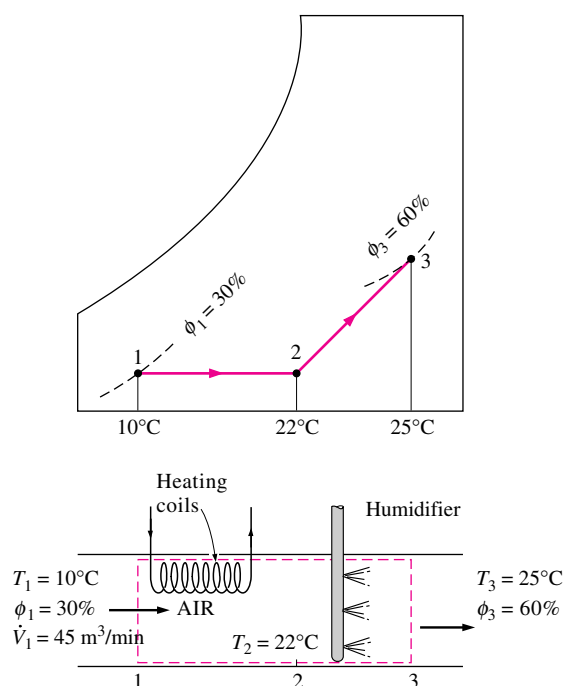
FIGURE 9–35
Schematic and psychrometric chart
for Example 9–7.

$$P_{v_1} = \phi_1 P_{g_1} = \phi_1 P_{\text{sat @ }10°C} = (0.3)(1.2276 \text{ kPa}) = 0.368 \text{ kPa}$$

$$P_{a_1} = P_1 - P_{v_1} = (100 - 0.368) \text{ kPa} = 99.632 \text{ kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a_1}} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(283 \text{ K})}{99.632 \text{ kPa}} = 0.815 \text{ m}^3/\text{kg dry air}$$

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{45 \text{ m}^3/\text{min}}{0.815 \text{ m}^3/\text{kg}} = 55.2 \text{ kg/min}$$

$$\omega_1 = \frac{0.622 P_{v_1}}{P_1 - P_{v_1}} = \frac{0.622(0.368 \text{ kPa})}{(100 - 0.368) \text{ kPa}} = 0.0023 \text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = C_p T_1 + \omega_1 h_{g_1} = (1.005 \text{ kJ/kg} \cdot °C)(10°C) + (0.0023)(2519.8 \text{ kJ/kg})$$
$$= 15.8 \text{ kJ/kg dry air}$$

$$h_2 = C_p T_2 + \omega_2 h_{g_2} = (1.005 \text{ kJ/kg} \cdot °C)(22°C) + (0.0023)(2541.7 \text{ kJ/kg})$$
$$= 28.0 \text{ kJ/kg dry air}$$

since $\omega_2 = \omega_1$. Then the rate of heat transfer to air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (55.2 \text{ kg/min})[(28.0 - 15.8) \text{ kJ/kg}]$$
$$= \textbf{673 kJ/min}$$

(b) The mass balance for water in the humidifying section can be expressed as

$$\dot{m}_{a_2} \omega_2 + \dot{m}_w = \dot{m}_{a_3} \omega_3$$

or

$$\dot{m}_w = \dot{m}_a (\omega_3 - \omega_2)$$

where

$$\omega_3 = \frac{0.622 \phi_3 P_{g_3}}{P_3 - \phi_3 P_{g_3}} = \frac{0.622(0.60)(3.169 \text{ kPa})}{[100 - (0.60)(3.169)] \text{ kPa}}$$
$$= 0.01206 \text{ kg H}_2\text{O/kg dry air}$$

Thus,

$$\dot{m}_w = (55.2 \text{ kg/min})(0.01206 - 0.0023)$$
$$= \textbf{0.539 kg/min}$$

Cooling with Dehumidification

The specific humidity of air remains constant during a simple cooling process, but its relative humidity increases. If the relative humidity reaches undesirably high levels, it may be necessary to remove some moisture from the air, that is, to dehumidify it. This requires cooling the air below its dew-point temperature.

The cooling process with dehumidifying is illustrated schematically and on the psychrometric chart in Fig. 9–36 in conjunction with Example 9–8. Hot, moist air enters the cooling section at state 1. As it passes through the cooling coils, its temperature decreases and its relative humidity increases at
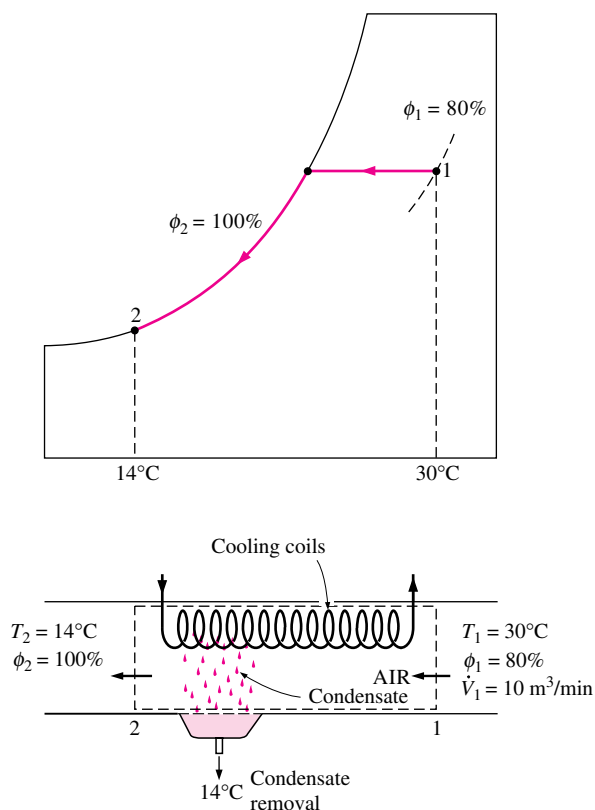
FIGURE 9–36
Schematic and psychrometric chart for Example 9–8.

constant specific humidity. If the cooling section is sufficiently long, air will reach its dew point (state 2, saturated air). Further cooling of air results in the condensation of part of the moisture in the air. Air remains saturated during the entire condensation process, which follows a line of 100 percent relative humidity until the final state (state 3) is reached. The water vapor that condenses out of the air during this process is removed from the cooling section through a separate channel. The condensate is usually assumed to leave the cooling section at T_3.

The cool, saturated air at state 3 is usually routed directly to the room, where it mixes with the room air. In some cases, however, the air at state 3 may be at the right specific humidity but at a very low temperature. In such cases, air is passed through a heating section where its temperature is raised to a more comfortable level before it is routed to the room.

EXAMPLE 9–8   **Cooling and Dehumidification of Air**

Air enters a window air conditioner at 1 atm, 30°C, and 80 percent relative humidity at a rate of 10 m³/min, and it leaves as saturated air at 14°C. Part of the moisture in the air that condenses during the process is also removed at 14°C. Determine the rates of heat and moisture removal from the air.

SOLUTION   We take the *cooling section* to be the system. The schematic of the system and the psychrometric chart of the process are shown in Fig. 9–36.

We note that the amount of water vapor in the air decreases during the process ($\omega_2 < \omega_1$) due to dehumidification.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and the water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis Applying the mass and energy balances on the cooling and dehumidification section gives

Dry air mass balance: $\quad\quad\quad \dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a$

Water mass balance: $\quad\quad\quad \dot{m}_{a_1}\omega_1 = \dot{m}_{a_2}\omega_2 + \dot{m}_w \quad \rightarrow \quad \dot{m}_w = \dot{m}_a(\omega_1 - \omega_2)$

Energy balance: $\quad\quad\quad\quad \sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e \quad \rightarrow$

$$\dot{Q}_{out} = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. Therefore, we can determine the properties of the air at both states from the psychrometric chart to be

$\quad\quad h_1 = 85.4$ kJ/kg dry air
$\quad\quad \omega_1 = 0.0216$ kg H_2O/kg dry air
$\quad\quad v_1 = 0.889$ m³/kg dry air

and

$\quad\quad h_2 = 39.3$ kJ/kg dry air
$\quad\quad \omega_2 = 0.0100$ kg H_2O/kg dry air

Also,

$\quad\quad h_w = h_{f\,@\,14°C} = 58.8$ kJ/kg $\quad\quad$ (Table A–4)

Then

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{10 \text{ m}^3/\text{min}}{0.889 \text{ m}^3/\text{kg dry air}} = 11.25 \text{ kg/min}$$

$$\dot{m}_w = (11.25 \text{ kg/min})(0.0216 - 0.0100) = \textbf{0.131 kg/min}$$

$$\dot{Q}_{out} = (11.25 \text{ kg/min})[(85.4 - 39.3) \text{ kJ/kg}] - (0.131 \text{ kg/min})(58.8 \text{ kJ/kg})$$
$$= \textbf{511 kJ/min}$$

Therefore, this air-conditioning unit removes moisture and heat from the air at rates of 0.131 kg/min and 511 kJ/min, respectively.
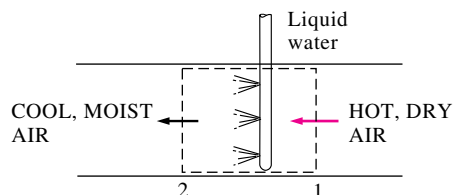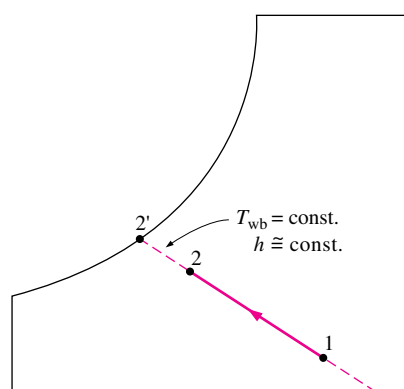
Evaporative Cooling

Conventional cooling systems operate on a refrigeration cycle, and they can be used in any part of the world. But they have a high initial and operating cost. In desert (hot and dry) climates, we can avoid the high cost of cooling by using *evaporative coolers,* also known as *swamp coolers.*

Evaporative cooling is based on a simple principle: As water evaporates, the latent heat of vaporization is absorbed from the water body and the surrounding air. As a result, both the water and the air are cooled during the process.
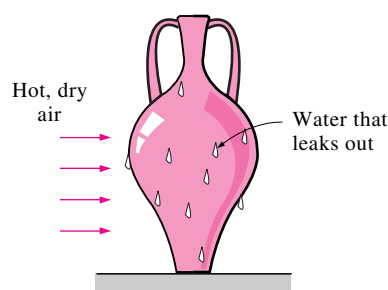
FIGURE 9–37
Water in a porous jug left in an open,
breezy area cools as a result
of evaporative cooling.

FIGURE 9–38
Evaporative cooling.

This approach has been used for thousands of years to cool water. A porous jug or pitcher filled with water is left in an open, shaded area. A small amount of water leaks out through the porous holes, and the pitcher "sweats." In a dry environment, this water evaporates and cools the remaining water in the pitcher (Fig. 9–37).

You have probably noticed that on a hot, dry day the air feels a lot cooler when the yard is watered. This is because water absorbs heat from the air as it evaporates. An evaporative cooler works on the same principle. The evaporative cooling process is shown schematically and on a psychrometric chart in Fig. 9–38. Hot, dry air at state 1 enters the evaporative cooler, where it is sprayed with liquid water. Part of the water evaporates during this process by absorbing heat from the airstream. As a result, the temperature of the airstream decreases and its humidity increases (state 2). In the limiting case, the air will leave the evaporative cooler saturated at state 2′. This is the lowest temperature that can be achieved by this process.

The evaporative cooling process is essentially identical to the adiabatic saturation process since the heat transfer between the airstream and the surroundings is usually negligible. Therefore, the evaporative cooling process follows a line of constant wet-bulb temperature on the psychrometric chart. (Note that this will not exactly be the case if the liquid water is supplied at a temperature different from the exit temperature of the airstream.) Since the constant-wet-bulb-temperature lines almost coincide with the constant-enthalpy lines, the enthalpy of the airstream can also be assumed to remain constant. That is,

$$T_{wb} \cong \text{constant} \tag{9–40}$$

and

$$h \cong \text{constant} \tag{9–41}$$

during an evaporative cooling process. This is a reasonably accurate approximation, and it is commonly used in air-conditioning calculations.

EXAMPLE 9–9   **Evaporative Cooling of Air by a Swamp Cooler**

Air enters an evaporative (or swamp) cooler at 14.7 psi, 95°F, and 20 percent relative humidity, and it exits at 80 percent relative humidity. Determine (*a*) the exit temperature of the air and (*b*) the lowest temperature to which the air can be cooled by this evaporative cooler.

SOLUTION   The schematic of the evaporative cooler and the psychrometric chart of the process are shown in Fig. 9–39.
Analysis   (*a*) If we assume the liquid water is supplied at a temperature not much different from the exit temperature of the airstream, the evaporative cooling process follows a line of constant wet-bulb temperature on the psychrometric chart. That is,

$$T_{wb} \cong \text{constant}$$

The wet-bulb temperature at 95°F and 20 percent relative humidity is determined from the psychrometric chart to be 66.0°F. The intersection point of the

$T_{wb} = 66.0°F$ and the $\phi = 80$ percent lines is the exit state of the air. The temperature at this point is the exit temperature of the air, and it is determined from the psychrometric chart to be

$$T_2 = \mathbf{70.4°F}$$

(*b*) In the limiting case, air will leave the evaporative cooler saturated ($\phi = 100$ percent), and the exit state of the air in this case will be the state where the $T_{wb} = 66.0°F$ line intersects the saturation line. For saturated air, the dry- and the wet-bulb temperatures are identical. Therefore, the lowest temperature to which air can be cooled is the wet-bulb temperature, which is

$$T_{min} = T_{2'} = \mathbf{66.0°F}$$
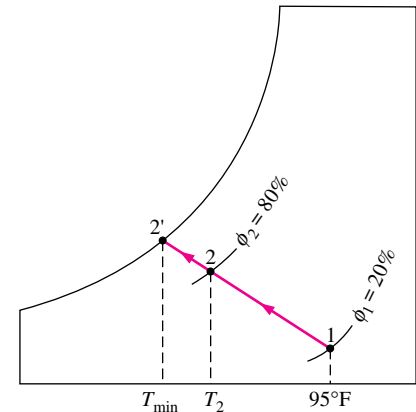
Adiabatic Mixing of Airstreams

Many air-conditioning applications require the mixing of two airstreams. This is particularly true for large buildings, most production and process plants, and hospitals, which require that the conditioned air be mixed with a certain fraction of fresh outside air before it is routed into the living space. The mixing is accomplished by simply merging the two airstreams, as shown in Fig. 9–40.

The heat transfer with the surroundings is usually small, and thus the mixing processes can be assumed to be adiabatic. Mixing processes normally involve no work interactions, and the changes in kinetic and potential energies, if any, are negligible. Then the mass and energy balances for the adiabatic mixing of two airstreams reduce to

Mass of dry air: $$\dot{m}_{a_1} + \dot{m}_{a_2} = \dot{m}_{a_3} \tag{9–42}$$

Mass of water vapor: $$\omega_1 \dot{m}_{a_1} + \omega_2 \dot{m}_{a_2} = \omega_3 \dot{m}_{a_3} \tag{9–43}$$

Energy: $$\dot{m}_{a_1} h_1 + \dot{m}_{a_2} h_2 = \dot{m}_{a_3} h_3 \tag{9–44}$$

Eliminating $\dot{m}_{a_3}$ from these relations, we obtain

$$\frac{\dot{m}_{a_1}}{\dot{m}_{a_2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1} \tag{9–45}$$

This equation has an instructive geometric interpretation on the psychrometric chart. It shows that the ratio of $\omega_2 - \omega_3$ to $\omega_3 - \omega_1$ is equal to the ratio of $\dot{m}_{a_1}$ to $\dot{m}_{a_2}$. The states that satisfy this condition are indicated by the dashed line *AB*. The ratio of $h_2 - h_3$ to $h_3 - h_1$ is also equal to the ratio of $\dot{m}_{a_1}$ to $\dot{m}_{a_2}$, and the states that satisfy this condition are indicated by the dashed line *CD*. The only state that satisfies both conditions is the intersection point of these two dashed lines, which is located on the straight line connecting states 1 and 2. Thus we conclude that *when two airstreams at two different states* (*states 1 and 2*) *are mixed adiabatically, the state of the mixture* (*state 3*) *will lie on the straight line connecting states 1 and 2 on the psychrometric chart, and the ratio of the distances 2-3 and 3-1 is equal to the ratio of mass flow rates* $\dot{m}_{a_1}$ *and* $\dot{m}_{a_2}$.

The concave nature of the saturation curve and the conclusion above lead to an interesting possibility. When states 1 and 2 are located close to the saturation curve, the straight line connecting the two states will cross the saturation
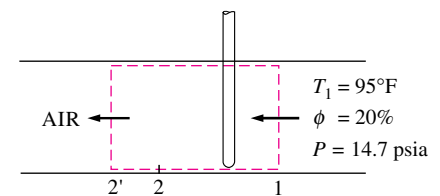

FIGURE 9–39
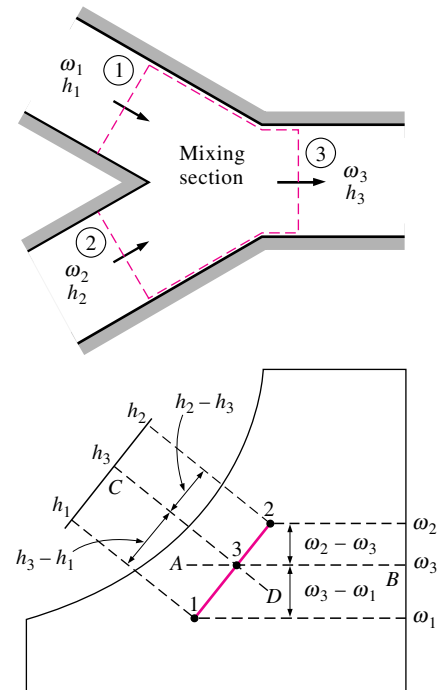Schematic and psychrometric chart for Example 9–9.

FIGURE 9–40
When two airstreams at states 1 and 2 are mixed adiabatically, the state of the mixture lies on the straight line connecting the two states.

curve, and state 3 may lie to the left of the saturation curve. In this case, some water will inevitably condense during the mixing process.
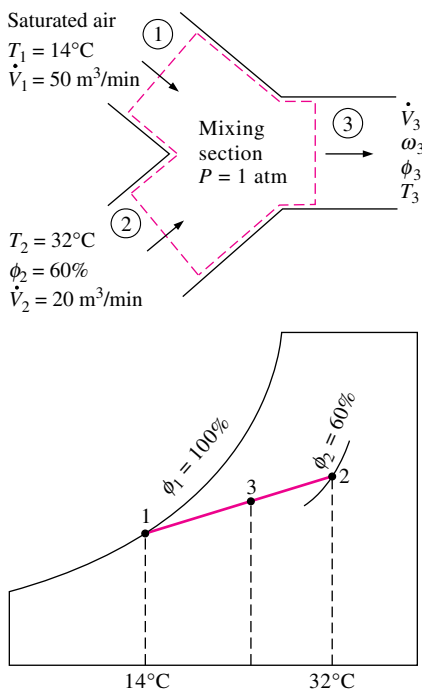

EXAMPLE 9–10   **Mixing of Conditioned Air with Outdoor Air**

Saturated air leaving the cooling section of an air-conditioning system at 14°C at a rate of 50 m³/min is mixed adiabatically with the outside air at 32°C and 60 percent relative humidity at a rate of 20 m³/min. Assuming that the mixing process occurs at a pressure of 1 atm, determine the specific humidity, the relative humidity, the dry-bulb temperature, and the volume flow rate of the mixture.

SOLUTION   We take the *mixing section* of the streams as the system. The schematic of the system and the psychrometric chart of the process are shown in Fig. 9–41. We note that this is a steady-flow mixing process.

Assumptions   **1** Steady operating conditions exist. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Analysis   The properties of each inlet stream are determined from the psychrometric chart to be

$$h_1 = 39.4 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.010 \text{ kg } H_2O/\text{kg dry air}$$
$$v_1 = 0.826 \text{ m}^3/\text{kg dry air}$$

and

$$h_2 = 79.0 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0182 \text{ kg } H_2O/\text{kg dry air}$$
$$v_2 = 0.889 \text{ m}^3/\text{kg dry air}$$

Then the mass flow rates of dry air in each stream are

$$\dot{m}_{a_1} = \frac{\dot{V}_1}{v_1} = \frac{50 \text{ m}^3/\text{min}}{0.826 \text{ m}^3/\text{kg dry air}} = 60.5 \text{ kg/min}$$

$$\dot{m}_{a_2} = \frac{\dot{V}_2}{v_2} = \frac{20 \text{ m}^3/\text{min}}{0.889 \text{ m}^3/\text{kg dry air}} = 22.5 \text{ kg/min}$$

From the mass balance of dry air,

$$\dot{m}_{a_3} = \dot{m}_{a_1} + \dot{m}_{a_2} = (60.5 + 22.5) \text{ kg/min} = 83 \text{ kg/min}$$

The specific humidity and the enthalpy of the mixture can be determined from Eq. 9–45,

$$\frac{\dot{m}_{a_1}}{\dot{m}_{a_2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$
$$\frac{60.5}{22.5} = \frac{0.0182 - \omega_3}{\omega_3 - 0.010} = \frac{79.0 - h_3}{h_3 - 39.4}$$

which yield

$$\omega_3 = \textbf{0.0122 kg } H_2O/\textbf{kg dry air}$$
$$h_3 = 50.1 \text{ kJ/kg dry air}$$

Saturated air
$T_1 = 14°C$
$\dot{V}_1 = 50 \text{ m}^3/\text{min}$

Mixing section
$P = 1 \text{ atm}$

$\dot{V}_3$
ω_3
ϕ_3
T_3

$T_2 = 32°C$
$\phi_2 = 60\%$
$\dot{V}_2 = 20 \text{ m}^3/\text{min}$

$\phi_1 = 100\%$
$\phi_2 = 60\%$

14°C   32°C

FIGURE 9–41
Schematic and psychrometric chart for Example 9–10.

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

$$T_3 = \mathbf{19.0°C}$$
$$\phi_3 = \mathbf{89\%}$$
$$v_3 = 0.844 \text{ m}^3/\text{kg dry air}$$

Finally, the volume flow rate of the mixture is determined from

$$\dot{V}_3 = \dot{m}_{a_3} v_3 = (83 \text{ kg/min})(0.844 \text{ m}^3/\text{kg}) = \mathbf{70.1 \text{ m}^3/\text{min}}$$

Discussion   Notice that the volume flow rate of the mixture is approximately equal to the sum of the volume flow rates of the two incoming streams. This is typical in air-conditioning applications.

Wet Cooling Towers

Power plants, large air-conditioning systems, and some industries generate large quantities of waste heat that is often rejected to cooling water from nearby lakes or rivers. In some cases, however, the water supply is limited or thermal pollution is a serious concern. In such cases, the waste heat must be rejected to the atmosphere, with cooling water recirculating and serving as a transport medium for heat transfer between the source and the sink (the atmosphere). One way of achieving this is through the use of wet cooling towers.

A **wet cooling tower** is essentially a semienclosed evaporative cooler. An induced-draft counterflow wet cooling tower is shown schematically in Fig. 9–42. Air is drawn into the tower from the bottom and leaves through the top. Warm water from the condenser is pumped to the top of the tower and is sprayed into this airstream. The purpose of spraying is to expose a large surface area of water to the air. As the water droplets fall under the influence of gravity, a small fraction of water (usually a few percent) evaporates and cools the remaining water. The temperature and the moisture content of the air increase during this process. The cooled water collects at the bottom of the tower and is pumped back to the condenser to pick up additional waste heat. Makeup water must be added to the cycle to replace the water lost by evaporation and air draft. To minimize water carried away by the air, drift eliminators are installed in the wet cooling towers above the spray section.

The air circulation in the cooling tower just described is provided by a fan, and therefore it is classified as a forced-draft cooling tower. Another popular type of cooling tower is the **natural-draft cooling tower,** which looks like a large chimney and works like an ordinary chimney. The air in the tower has a high water-vapor content, and thus it is lighter than the outside air. Consequently, the light air in the tower rises, and the heavier outside air fills the vacant space, creating an airflow from the bottom of the tower to the top. The flow rate of air is controlled by the conditions of the atmospheric air. Natural-draft cooling towers do not require any external power to induce the air, but they cost a lot more to build than forced-draft cooling towers. The natural-draft cooling towers are hyperbolic in profile, as shown in Fig. 9–43, and some are over 100 m high. The hyperbolic profile is for greater structural strength, not for any thermodynamic reason.
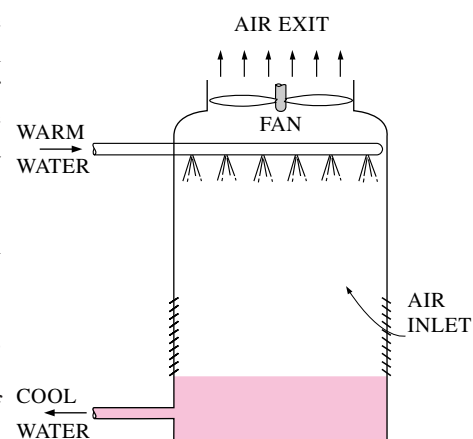
FIGURE 9–42
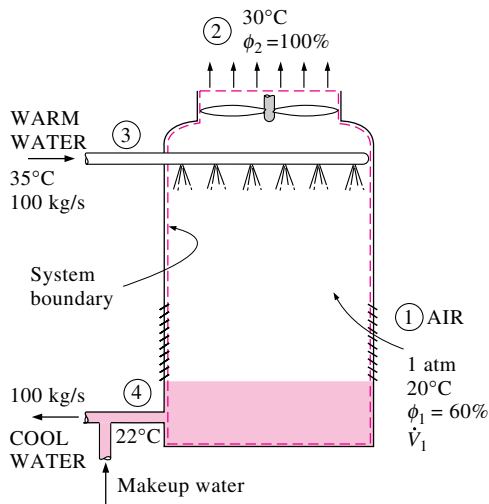An induced-draft counterflow cooling tower.
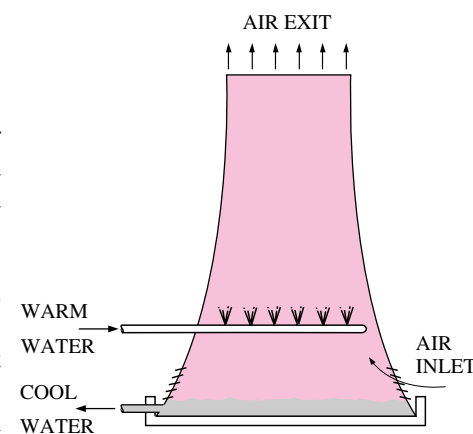
FIGURE 9–43
A natural-draft cooling tower.

The idea of a cooling tower started with the **spray pond,** where the warm water is sprayed into the air and is cooled by the air as it falls into the pond, as shown in Fig. 9–44. Some spray ponds are still in use today. However, they require 25 to 50 times the area of a cooling tower, water loss due to air drift is high, and they are unprotected against dust and dirt.

We could also dump the waste heat into a still **cooling pond,** which is basically a large artificial lake open to the atmosphere. Heat transfer from the pond surface to the atmosphere is very slow, however, and we would need about 20 times the area of a spray pond in this case to achieve the same cooling.

FIGURE 9–44
A spray pond (photo by Yunus Çengel).

FIGURE 9–45
Schematic for Example 9–11.

EXAMPLE 9–11   Cooling of a Power Plant by a Cooling Tower

Cooling water leaves the condenser of a power plant and enters a wet cooling tower at 35°C at a rate of 100 kg/s. The water is cooled to 22°C in the cooling tower by air that enters the tower at 1 atm, 20°C, and 60 percent relative humidity and leaves saturated at 30°C. Neglecting the power input to the fan, determine (*a*) the volume flow rate of air into the cooling tower and (*b*) the mass flow rate of the required makeup water.

SOLUTION   We take the entire *cooling tower* to be the system, which is shown schematically in Fig. 9–45. We note that the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation.

Assumptions   **1** Steady operating conditions exist and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and the water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

Analysis   Applying the mass and energy balances on the cooling tower gives

Dry air mass balance: $\qquad\qquad\qquad\qquad \dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a$

Water mass balance: $\qquad\qquad\qquad \dot{m}_3 + \dot{m}_{a_1}\omega_1 = \dot{m}_4 + \dot{m}_{a_2}\omega_2$

or $\qquad\qquad\qquad \dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$

Energy balance: $\qquad \sum \dot{m}_i h_i = \sum \dot{m}_e h_e \;\; \rightarrow \;\; \dot{m}_{a_1} h_1 + \dot{m}_3 h_3 = \dot{m}_{a_2} h_2 + \dot{m}_4 h_4$

or

$$\dot{m}_3 h_3 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4$$

Solving for $\dot{m}_a$ gives

$$\dot{m}_a = \frac{\dot{m}_3(h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1)h_4}$$

From the psychrometric chart,

$$h_1 = 42.2 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0087 \text{ kg H}_2\text{O/kg dry air}$$
$$\upsilon_1 = 0.842 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 100.0 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0273 \text{ kg H}_2\text{O/kg dry air}$$

From Table A–4, $h_3 \cong h_{f @ 35°C} = 146.68 \text{ kJ/kg H}_2\text{O}$
$$h_4 \cong h_{f @ 22°C} = 92.33 \text{ kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(100 \text{ kg/s})[(146.68 - 92.33) \text{ kJ/kg}]}{[(100.0 - 42.2) \text{ kJ/kg}] - [(0.0273 - 0.0087)(92.33) \text{ kJ/kg}]} = 96.9 \text{ kg/s}$$

Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a \upsilon_1 = (96.9 \text{ kg/s})(0.842 \text{ m}^3/\text{kg}) = \mathbf{81.6 \text{ m}^3/\text{s}}$$

(*b*) The mass flow rate of the required makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (96.9 \text{ kg/s})(0.0273 - 0.0087) = \mathbf{1.80 \text{ kg/s}}$$

Discussion   Note that over 98 percent of the cooling water is saved and recirculated in this case.

SUMMARY

A mixture of two or more gases of fixed chemical composition is called a *nonreacting gas mixture*. The composition of a gas mixture is described by specifying either the *mole fraction* or the *mass fraction* of each component, defined as

$$\text{mf}_i = \frac{m_i}{m_m} \quad \text{and} \quad y_i = \frac{N_i}{N_m}$$

where

$$m_m = \sum_{i=1}^{k} m_i \quad \text{and} \quad N_m = \sum_{i=1}^{k} N_i$$

The *apparent* (or average) *molar mass* and *gas constant* of a mixture are expressed as

$$M_m = \frac{m_m}{N_m} = \sum_{i=1}^{k} y_i M_i \quad \text{and} \quad R_m = \frac{R_u}{M_m}$$

Also,

$$\text{mf}_i = y_i \frac{M_i}{M_m} \quad \text{and} \quad M_m = \frac{1}{\sum_{i=1}^{k} \dfrac{\text{mf}_i}{M_i}}$$

Dalton's law of additive pressures states that the pressure of a gas mixture is equal to the sum of the pressures each gas would exert if it existed alone at the mixture temperature and volume. *Amagat's law of additive volumes* states that the

volume of a gas mixture is equal to the sum of the volumes each gas would occupy if it existed alone at the mixture temperature and pressure. Dalton's and Amagat's laws hold exactly for ideal-gas mixtures, but only approximately for real-gas mixtures. They can be expressed as

Dalton's law: $$P_m = \sum_{i=1}^{k} P_i(T_m, V_m)$$

Amagat's law: $$V_m = \sum_{i=1}^{k} V_i(T_m, P_m)$$

Here P_i is called the *component pressure* and V_i is called the *component volume*. Also, the ratio P_i/P_m is called the *pressure fraction* and the ratio V_i/V_m is called the *volume fraction* of component *i*. For *ideal gases,* P_i and V_i can be related to y_i by

$$\frac{P_i}{P_m} = \frac{V_i}{V_m} = \frac{N_i}{N_m} = y_i$$

The quantity $y_i P_m$ is called the *partial pressure* and the quantity $y_i V_m$ is called the *partial volume*. The *P-υ-T* behavior of real-gas mixtures can be predicted by using generalized compressibility charts. The compressibility factor of the mixture can be expressed in terms of the compressibility factors of the individual gases as

$$Z_m = \sum_{i=1}^{k} y_i Z_i$$

where Z_i is determined either at T_m and V_m (Dalton's law) or at T_m and P_m (Amagat's law) for each individual gas.

The *extensive properties* of a gas mixture, in general, can be determined by summing the contributions of each component of the mixture. The evaluation of *intensive properties* of a gas mixture, however, involves averaging in terms of mass or mole fractions:

$$U_m = \sum_{i=1}^{k} U_i = \sum_{i=1}^{k} m_i u_i = \sum_{i=1}^{k} N_i \bar{u}_i \qquad \text{(kJ)}$$

$$H_m = \sum_{i=1}^{k} H_i = \sum_{i=1}^{k} m_i h_i = \sum_{i=1}^{k} N_i \bar{h}_i \qquad \text{(kJ)}$$

$$S_m = \sum_{i=1}^{k} S_i = \sum_{i=1}^{k} m_i s_i = \sum_{i=1}^{k} N_i \bar{s}_i \qquad \text{(kJ/K)}$$

and

$$u_m = \sum_{i=1}^{k} \text{mf}_i u_i \quad \text{and} \quad \bar{u}_m = \sum_{i=1}^{k} y_i \bar{u}_i \qquad \begin{array}{l}\text{(kJ/kg or}\\ \text{kJ/kmol)}\end{array}$$

$$h_m = \sum_{i=1}^{k} \text{mf}_i h_i \quad \text{and} \quad \bar{h}_m = \sum_{i=1}^{k} y_i \bar{h}_i \qquad \begin{array}{l}\text{(kJ/kg or}\\ \text{kJ/kmol)}\end{array}$$

$$s_m = \sum_{i=1}^{k} \text{mf}_i s_i \quad \text{and} \quad \bar{s}_m = \sum_{i=1}^{k} y_i \bar{s}_i$$

$$[\text{kJ/(kg} \cdot \text{K) or kJ(kmol} \cdot \text{K)}]$$

$$C_{v,m} = \sum_{i=1}^{k} \text{mf}_i C_{v,i} \quad \text{and} \quad \bar{C}_{v,m} = \sum_{i=1}^{k} y_i \bar{C}_{v,i}$$

$$C_{p,m} = \sum_{i=1}^{k} \text{mf}_i C_{p,i} \quad \text{and} \quad \bar{C}_{p,m} = \sum_{i=1}^{k} y_i \bar{C}_{p,i}$$

These relations are exact for ideal-gas mixtures, and approximate for real-gas mixtures. The properties or property changes of individual components can be determined by using ideal-gas or real-gas relations developed in earlier chapters.

The air in the atmosphere normally contains some water vapor, and it is referred to as *atmospheric air*. By contrast, air that contains no water vapor is called *dry air*. In the temperature range encountered in air-conditioning applications, both the dry air and the water vapor can be treated as ideal gases. The enthalpy change of dry air during a process can be determined from

$$\Delta h_{\text{dry air}} = C_p \, \Delta T = [1.005 \text{ kJ/(kg} \cdot \text{°C)}] \, \Delta T \qquad \text{(kJ/kg)}$$

The atmospheric air can be treated as an ideal-gas mixture whose pressure is the sum of the partial pressure of dry air P_a and that of the water vapor P_v,

$$P = P_a + P_v \qquad \text{(kPa)}$$

The enthalpy of water vapor in the air can be taken to be equal to the enthalpy of the saturated vapor at the same temperature:

$$h_v(T, \text{low } P) \cong h_g(T) \cong 2501.3 + 1.82T \quad \text{(kJ/kg)} \quad T \text{ in °C}$$
$$\cong 1061.5 + 0.435T \quad \text{(Btu/lbm)} \quad T \text{ in °F}$$

in the temperature range -10 to 50°C (15 to 120°F).

The mass of water vapor present in 1 unit mass of dry air is called the *specific* or *absolute humidity* ω,

$$\omega = \frac{m_v}{m_a} = \frac{0.622 P_v}{P - P_v} \qquad \text{(kg H}_2\text{O/kg dry air)}$$

where P is the total pressure of air and P_v is the vapor pressure. There is a limit on the amount of vapor the air can hold at a given temperature. Air that is holding as much moisture as it can at a given temperature is called *saturated air*. The ratio of the amount of moisture air holds (m_v) to the maximum amount of moisture air can hold at the same temperature (m_g) is called the *relative humidity* ϕ,

$$\phi = \frac{m_v}{m_g} = \frac{P_v V/(R_v T)}{P_g V/(R_v T)} = \frac{P_v}{P_g}$$

where $P_g = P_{\text{sat} @ T}$. The relative and specific humidities can also be expressed as

$$\phi = \frac{\omega P}{(0.622 + \omega)P_g} \quad \text{and} \quad \omega = \frac{0.622 \phi P_g}{P - \phi P_g}$$

Relative humidity ranges from 0 for dry air to 1 for saturated air.

The enthalpy of atmospheric air is expressed *per unit mass of dry air*, instead of per unit mass of the air–water-vapor mixture, as

$$h = h_a + \omega h_g \qquad \text{(kJ/kg dry air)}$$

The ordinary temperature of atmospheric air is referred to as the *dry-bulb temperature* to differentiate it from other forms of temperatures. The temperature at which condensation begins if the air is cooled at constant pressure is called the *dew-point temperature* T_{dp}:

$$T_{\text{dp}} = T_{\text{sat} @ P_v}$$

Relative humidity and specific humidity of air can be determined by measuring the *adiabatic saturation temperature* of air, which is the temperature air attains after flowing over water in a long adiabatic channel until it is saturated,

$$\omega_1 = \frac{C_p(T_2 - T_1) + \omega_2 h_{fg_2}}{h_{g_1} - h_{f_2}}$$

where

$$\omega_2 = \frac{0.622 P_{g_2}}{P_2 - P_{g_2}}$$

and T_2 is the adiabatic saturation temperature. A more practical approach in air-conditioning applications is to use a thermometer whose bulb is covered with a cotton wick saturated with water and to blow air over the wick. The temperature measured in this manner is called the *wet-bulb temperature* T_{wb}, and it is used in place of the adiabatic saturation temperature. The properties of atmospheric air at a specified total pressure are presented in the form of easily readable charts, called *psychrometric charts.* The lines of constant enthalpy and the lines of constant wet-bulb temperature are very nearly parallel on these charts.

Most air-conditioning processes can be modeled as steady-flow processes, and therefore they can be analyzed by applying the steady-flow mass (for both dry air and water) and energy balances,

Dry air mass:
$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e}$$

Water mass:
$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e}$$

or
$$\sum \dot{m}_{a,i}\omega_i = \sum \dot{m}_{a,e}\omega_e$$

Energy:
$$\dot{Q}_{in} + \dot{W}_{in} + \sum \dot{m}_i h_i = \dot{Q}_{out} + \dot{W}_{out} + \sum \dot{m}_e h_e$$

where subscripts i and e denote inlet and exit states, respectively. The changes in kinetic and potential energies are assumed to be negligible.

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PROBLEMS*

Composition of Gas Mixtures

9–1C   What is the *apparent gas constant* for a gas mixture? Can it be larger than the largest gas constant in the mixture?

9–2C   Consider a mixture of two gases. Can the apparent molar mass of this mixture be determined by simply taking the arithmetic average of the molar masses of the individual gases? When will this be the case?

9–3C   What is the *apparent molar mass* for a gas mixture? Does the mass of every molecule in the mixture equal the apparent molar mass?

9–4C   Consider a mixture of several gases of identical masses. Will all the mass fractions be identical? How about the mole fractions?

*Problems designated by a "C" are concept questions, and students are encouraged to answer them all. Problems designated by an "E" are in English units, and the SI users can ignore them. Problems with a CD-EES icon ⊛ are solved using EES, and complete solutions together with parametric studies are included on the enclosed CD. Problems with a computer-EES icon 🖳 are comprehensive in nature, and are intended to be solved with a computer, preferably using the EES software that accompanies this text.

9–5C   The sum of the mole fractions for an ideal-gas mixture is equal to 1. Is this also true for a real-gas mixture?

9–6C   What are mass and mole fractions?

9–7C   Using the definitions of mass and mole fractions, derive a relation between them.

9–8C   Somebody claims that the mass and mole fractions for a mixture of CO_2 and N_2O gases are identical. Is this true? Why?

9–9C   Consider a mixture of two gases A and B. Show that when the mass fractions mf_A and mf_B are known, the mole fractions can be determined from

$$y_A = \frac{M_B}{M_A(1/mf_A - 1) + M_B} \quad \text{and} \quad y_B = 1 - y_A$$

where M_A and M_B are the molar masses of A and B.

9–10   The composition of moist air is given on a molar basis to be 78 percent N_2, 20 percent O_2, and 2 percent water vapor. Determine the mass fractions of the constituents of air.

9–11   A gas mixture has the following composition on a mole basis: 60 percent N_2 and 40 percent CO_2. Determine the gravimetric analysis of the mixture, its molar mass, and gas constant.

9–12 Repeat Prob. 9–11 by replacing N_2 by O_2.

9–13 A gas mixture consists of 5 kg of O_2, 8 kg of N_2, and 10 kg of CO_2. Determine (*a*) the mass fraction of each component, (*b*) the mole fraction of each component, and (*c*) the average molar mass and gas constant of the mixture.

9–14 Determine the mole fractions of a gas mixture that consists of 60 percent CH_4 and 40 percent CO_2 by mass. Also, determine the gas constant of the mixture.

9–15 A gas mixture consists of 8 kmol of H_2 and 2 kmol of N_2. Determine the mass of each gas and the apparent gas constant of the mixture. *Answers:* 16 kg, 56 kg, 1.155 kJ/kg · K

9–16E A gas mixture consists of 5 lbmol of H_2 and 3 lbmol of N_2. Determine the mass of each gas and the apparent gas constant of the mixture.

P-v-T Behavior of Gas Mixtures

9–17C Is a mixture of ideal gases also an ideal gas? Give an example.

9–18C Express Dalton's law of additive pressures. Does this law hold exactly for ideal-gas mixtures? How about nonideal-gas mixtures?

9–19C Express Amagat's law of additive volumes. Does this law hold exactly for ideal-gas mixtures? How about nonideal-gas mixtures?

9–20C How is the *P-v-T* behavior of a component in an ideal-gas mixture expressed? How is the *P-v-T* behavior of a component in a real-gas mixture expressed?

9–21C What is the difference between the *component pressure* and the *partial pressure*? When are these two equivalent?

9–22C What is the difference between the *component volume* and the *partial volume*? When are these two equivalent?

9–23C In a gas mixture, which component will have the higher partial pressure—the one with the higher mole number or the one with the larger molar mass?

9–24C Consider a rigid tank that contains a mixture of two ideal gases. A valve is opened and some gas escapes. As a result, the pressure in the tank drops. Will the partial pressure of each component change? How about the pressure fraction of each component?

9–25C Consider a rigid tank that contains a mixture of two ideal gases. The gas mixture is heated, and the pressure and temperature in the tank rise. Will the partial pressure of each component change? How about the pressure fraction of each component?

9–26C Is this statement correct? *The volume of an ideal-gas mixture is equal to the sum of the volumes of each individual gas in the mixture.* If not, how would you correct it?

9–27C Is this statement correct? *The temperature of an ideal-gas mixture is equal to the sum of the temperatures of each individual gas in the mixture.* If not, how would you correct it?

9–28C Is this statement correct? *The pressure of an ideal-gas mixture is equal to the sum of the partial pressures of each individual gas in the mixture.* If not, how would you correct it?

9–29C Explain how a real-gas mixture can be treated as a pseudopure substance using Kay's rule.

9–30 A rigid tank contains 8 kmol of O_2 and 10 kmol of CO_2 gases at 290 K and 150 kPa. Estimate the volume of the tank. *Answer:* 289 m³

9–31 Repeat Prob. 9–30 for a temperature of 350 K.

9–32 A rigid tank contains 0.5 kmol of Ar and 2 kmol of N_2 at 250 kPa and 280 K. The mixture is now heated to 400 K. Determine the volume of the tank and the final pressure of the mixture.

9–33 A gas mixture at 300 K and 200 kPa consists of 1 kg of CO_2 and 3 kg of CH_4. Determine the partial pressure of each gas and the apparent molar mass of the gas mixture.

9–34E A gas mixture at 600 R and 20 psia consists of 1 lbm of CO_2 and 3 lbm of CH_4. Determine the partial pressure of each gas and the apparent molar mass of the gas mixture.

9–35 A 0.3-m³ rigid tank contains 0.6 kg of N_2, and 0.4 kg of O_2 at 300 K. Determine the partial pressure of each gas and the total pressure of the mixture.
Answers: 178.1 kPa, 103.9 kPa, 282.0 kPa

9–36 A gas mixture at 350 K and 300 kPa has the following volumetric analysis: 65 percent N_2, 20 percent O_2, and 15 percent CO_2. Determine the mass fraction and partial pressure of each gas.

9–37 A rigid tank that contains 2 kg of N_2 at 25°C and 200 kPa is connected to another rigid tank that contains 3 kg of O_2 at 25°C and 500 kPa. The valve connecting the two tanks is opened, and the two gases are allowed to mix. If the final mixture temperature is 25°C, determine the volume of each tank and the final mixture pressure.
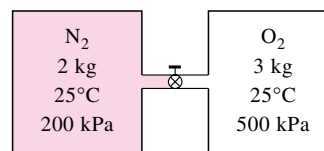*Answers:* 0.884 m³, 0.465 m³, 303.4 kPa

FIGURE P9–37

Properties of Gas Mixtures

9–38C Is the total internal energy of an ideal-gas mixture equal to the sum of the internal energies of each individual gas in the mixture? Answer the same question for a real-gas mixture.

9–39C Is the specific internal energy of a gas mixture equal to the sum of the specific internal energy of each individual gas in the mixture?

9–40C Answer Probs. 9–38C and 9–39C for entropy.

9–41C Is the total internal energy change of an ideal-gas mixture equal to the sum of the internal energy changes of each individual gas in the mixture? Answer the same question for a real-gas mixture.

9–42C When evaluating the entropy change of the components of an ideal-gas mixture, do we have to use the partial pressure of each component or the total pressure of the mixture?

9–43C Suppose we want to determine the enthalpy change of a real-gas mixture undergoing a process. The enthalpy change of each individual gas is determined by using the generalized enthalpy chart, and the enthalpy change of the mixture is determined by summing them. Is this an exact approach? Explain.

9–44 An insulated rigid tank is divided into two compartments by a partition. One compartment contains 0.5 kmol of CO_2 at 27°C and 200 kPa, and the other compartment contains 7.5 kmol of H_2 gas at 40°C and 400 kPa. Now the partition is removed, and the two gases are allowed to mix. Determine (*a*) the mixture temperature and (*b*) the mixture pressure after equilibrium has been established. Assume constant specific heats at room temperature for both gases.
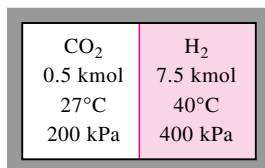
CO_2	H_2
0.5 kmol	7.5 kmol
27°C	40°C
200 kPa	400 kPa

FIGURE P9–44

9–45 A 0.9-m^3 rigid tank is divided into two equal compartments by a partition. One compartment contains Ne at 20°C and 100 kPa, and the other compartment contains Ar at 50°C and 200 kPa. Now the partition is removed, and the two gases are allowed to mix. Heat is lost to the surrounding air during this process in the amount of 15 kJ. Determine (*a*) the final mixture temperature and (*b*) the final mixture pressure.
Answers: (*a*) 16.2°C, (*b*) 138.9 kPa

9–46 Repeat Prob. 9–45 for a heat loss of 8 kJ.

9–47 An equimolar mixture of helium and argon gases is to be used as the working fluid in a closed-loop gas-turbine cycle. The mixture enters the turbine at 1.2 MPa and 1300 K and expands isentropically to a pressure of 200 kPa. Determine the work output of the turbine per unit mass of the mixture.
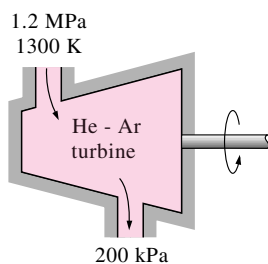
1.2 MPa
1300 K

He - Ar
turbine

200 kPa
FIGURE P9–47

9–48E A mixture of 80 percent N_2 and 20 percent CO_2 gases (on a mass basis) enters the nozzle of a turbojet engine at 90 psia and 1800 R with a low velocity, and it expands to a pressure of 12 psia. If the isentropic efficiency of the nozzle is 92 percent, determine (*a*) the exit temperature and (*b*) the exit velocity of the mixture. Assume constant specific heats at room temperature.

9–49E Reconsider Prob. 9–48E. Using EES (or other) software, first solve the stated problem and then, for all other conditions being the same, resolve the problem to determine the composition of the nitrogen and carbon dioxide that is required to have an exit velocity of 2600 ft/s at the nozzle exit.

9–50 A piston-cylinder device contains a mixture of 0.2 kg of H_2 and 1.6 kg of N_2 at 100 kPa and 300 K. Heat is now transferred to the mixture at constant pressure until the volume is doubled. Assuming constant specific heats at the average temperature, determine (*a*) the heat transfer and (*b*) the entropy change of the mixture.

9–51 An insulated tank that contains 1 kg of O_2 at 15°C and 300 kPa is connected to a 2-m^3 uninsulated tank that contains N_2 at 50°C and 500 kPa. The valve connecting the two tanks is opened, and the two gases form a homogeneous mixture at 25°C. Determine (*a*) the final pressure in the tank, and (*b*) the heat transfer.
Answers: (*a*) 444.6 kPa, (*b*) 187.2 kJ loss

9–52 Reconsider Prob. 9–51. Using EES (or other) software, compare the results obtained assuming ideal-gas behavior with constant specific heats at the average temperature, and using real-gas data obtained from EES by assuming variable specific heats over the temperature range.

Dry and Atmospheric Air: Specific and Relative Humidity

9–53C Is it possible to obtain saturated air from unsaturated air without adding any moisture? Explain.

9–54C Is the relative humidity of saturated air necessarily 100 percent?

9–55C Moist air is passed through a cooling section where it is cooled and dehumidified. How do (*a*) the specific humidity and (*b*) the relative humidity of air change during this process?

9–56C What is the difference between dry air and atmospheric air?

9–57C Can the water vapor in air be treated as an ideal gas? Explain.

9–58C What is vapor pressure?

9–59C How would you compare the enthalpy of water vapor at 20°C and 2 kPa with the enthalpy of water vapor at 20°C and 0.5 kPa?

9–60C What is the difference between the specific humidity and the relative humidity?

9–61C How will (*a*) the specific humidity and (*b*) the relative humidity of the air contained in a well-sealed room change as it is heated?

9–62C How will (*a*) the specific humidity and (*b*) the relative humidity of the air contained in a well-sealed room change as it is cooled?

9–63C Consider a tank that contains moist air at 3 atm and whose walls are permeable to water vapor. The surrounding air at 1 atm pressure also contains some moisture. Is it possible for the water vapor to flow into the tank from surroundings? Explain.

9–64C Why are the chilled water lines always wrapped with vapor barrier jackets?

9–65C Explain how vapor pressure of the ambient air is determined when the temperature, total pressure, and the relative humidity of air are given.

9–66 A tank contains 21 kg of dry air and 0.3 kg of water vapor at 30°C and 100 kPa total pressure. Determine (*a*) the specific humidity, (*b*) the relative humidity, and (*c*) the volume of the tank.

9–67 Repeat Prob. 9–66 for a temperature of 35°C.

9–68 A room contains air at 20°C and 98 kPa at a relative humidity of 85 percent. Determine (*a*) the partial pressure of dry air, (*b*) the specific humidity of the air, and (*c*) the enthalpy per unit mass of dry air.

9–69 Repeat Prob. 9–68 for a pressure of 85 kPa.

9–70E A room contains air at 70°F and 14.6 psia at a relative humidity of 85 percent. Determine (*a*) the partial pressure of dry air, (*b*) the specific humidity, and (*c*) the enthalpy per unit mass of dry air.
Answers: (*a*) 14.291 psia, (*b*) 0.0134 lbm H_2O/lbm dry air, (*c*) 31.43 Btu/lbm dry air

9–71 Determine the masses of dry air and the water vapor contained in a 240-m³ room at 98 kPa, 23°C, and 50 percent relative humidity. *Answers:* 273 kg, 2.5 kg

Dew-Point, Adiabatic Saturation, and Wet-Bulb Temperatures

9–72C What is the dew-point temperature?

9–73C Andy and Wendy both wear glasses. On a cold winter day, Andy comes from the cold outside and enters the warm house while Wendy leaves the house and goes outside. Whose glasses are more likely to be fogged? Explain.

9–74C In summer, the outer surface of a glass filled with iced water frequently "sweats." How can you explain this sweating?

9–75C In some climates, cleaning the ice off the windshield of a car is a common chore on winter mornings. Explain how ice forms on the windshield during some nights even when there is no rain or snow.

9–76C When are the dry-bulb and dew-point temperatures identical?

9–77C When are the adiabatic saturation and wet-bulb temperatures equivalent for atmospheric air?

9–78 A house contains air at 25°C and 65 percent relative humidity. Will any moisture condense on the inner surfaces of the windows when the temperature of the window drops to 10°C?

9–79 After a long walk in the 8°C outdoors, a person wearing glasses enters a room at 25°C and 40 percent relative humidity. Determine whether the glasses will become fogged.

9–80 Repeat Prob. 9–79 for a relative humidity of 70 percent.

9–81E A thirsty woman opens the refrigerator and picks up a cool canned drink at 40°F. Do you think the can will "sweat" as she enjoys the drink in a room at 80°F and 50 percent relative humidity?

9–82 The dry- and wet-bulb temperatures of atmospheric air at 95 kPa are 25 and 20°C, respectively. Determine (*a*) the specific humidity, (*b*) the relative humidity, and (*c*) the enthalpy of the air, in kJ/kg dry air.

9–83 The air in a room has a dry-bulb temperature of 22°C and a wet-bulb temperature of 16°C. Assuming a pressure of 100 kPa, determine (*a*) the specific humidity, (*b*) the relative humidity, and (*c*) the dew-point temperature.
Answers: (*a*) 0.0091 kg H_2O/kg dry air, (*b*) 54.0 percent, (*c*) 12.3°C

9–84 Reconsider Prob. 9–83. Determine the required properties using EES (or other) software. What would the property values be at a pressure of 300 kPa?

9–85E The air in a room has a dry-bulb temperature of 70°F and a wet-bulb temperature of 60°F. Assuming a pressure of 14.7 psia, determine (*a*) the specific humidity, (*b*) the relative humidity, and (*c*) the dew-point temperature.
Answers: (*a*) 0.0087 lbm H_2O/lbm dry air, (*b*) 55.8 percent, (*c*) 53.2°F

Psychrometric Chart

9–86C How do constant-enthalpy and constant-wet-bulb-temperature lines compare on the psychrometric chart?

9–87C At what states on the psychrometric chart are the dry-bulb, wet-bulb, and dew-point temperatures identical?

9–88C How is the dew-point temperature at a specified state determined on the psychrometric chart?

9–89C Can the enthalpy values determined from a psychrometric chart at sea level be used at higher elevations?

9–90 The air in a room is at 1 atm, 32°C, and 60 percent relative humidity. Determine (*a*) the specific humidity, (*b*) the enthalpy (in kJ/kg dry air), (*c*) the wet-bulb temperature, (*d*) the dew-point temperature, and (*e*) the specific volume of the air (in m³/kg dry air). Use the psychrometric chart or available software.

9–91 Reconsider Prob. 9–90. Determine the required properties using EES (or other) software instead of the pychrometric chart. What would the property values be at a location at 1500 m altitude?

9–92 A room contains air at 1 atm, 26°C, and 70 percent relative humidity. Using the psychrometric chart, determine (*a*) the specific humidity, (*b*) the enthalpy (in kJ/kg dry air), (*c*) the wet-bulb temperature, (*d*) the dew-point temperature, and (*e*) the specific volume of the air (in m³/kg dry air).

9–93 Reconsider Prob. 9–92. Determine the required properties using EES (or other) software instead of the pychrometric chart. What would the property values be at a location at 2000 m altitude?

9–94E A room contains air at 1 atm, 82°F, and 70 percent relative humidity. Using the psychrometric chart, determine (*a*) the specific humidity, (*b*) the enthalpy (in Btu/lbm dry air), (*c*) the wet-bulb temperature, (*d*) the dew-point temperature, and (*e*) the specific volume of the air (in ft³/lbm dry air).

9–95E Reconsider Prob. 9–94E. Determine the required properties using EES (or other) software instead of the pychrometric chart. What would the property values be at a location at 5000 ft altitude?

9–96 The air in a room has a pressure of 1 atm, a dry-bulb temperature of 24°C, and a wet-bulb temperature of 17°C. Using the psychrometric chart, determine (*a*) the specific humidity, (*b*) the enthalpy (in kJ/kg dry air), (*c*) the relative humidity, (*d*) the dew-point temperature, and (*e*) the specific volume of the air (in m³/kg dry air).

9–97 Reconsider Prob. 9–96. Determine the required properties using EES (or other) software instead of the pychrometric chart. What would the property values be at a location at 3000 m altitude?

Human Comfort and Air-Conditioning

9–98C What does a modern air-conditioning system do besides heating or cooling the air?

9–99C How does the human body respond to (*a*) hot weather, (*b*) cold weather, and (*c*) hot and humid weather?

9–100C What is the radiation effect? How does it affect human comfort?

9–101C How does the air motion in the vicinity of the human body affect human comfort?

9–102C Consider a tennis match in cold weather where both players and spectators wear the same clothes. Which group of people will feel colder? Why?

9–103C Why do you think little babies are more susceptible to cold?

9–104C How does humidity affect human comfort?

9–105C What are humidification and dehumidification?

9–106C What is metabolism? What is the range of metabolic rate for an average man? Why are we interested in the metabolic rate of the occupants of a building when we deal with heating and air-conditioning?

9–107C Why is the metabolic rate of women, in general, lower than that of men? What is the effect of clothing on the environmental temperature that feels comfortable?

9–108C What is sensible heat? How is the sensible heat loss from a human body affected by the (*a*) skin temperature, (*b*) environment temperature, and (*c*) air motion?

9–109C What is latent heat? How is the latent heat loss from the human body affected by the (*a*) skin wettedness and (*b*) relative humidity of the environment? How is the rate of evaporation from the body related to the rate of latent heat loss?

9–110 An average person produces 0.25 kg of moisture while taking a shower and 0.05 kg while bathing in a tub. Consider a family of four who each shower once a day in a bathroom that is not ventilated. Taking the heat of vaporization of water to be 2450 kJ/kg, determine the contribution of showers to the latent heat load of the air conditioner per day in summer.

9–111 An average (1.82 kg or 4.0 lbm) chicken has a basal metabolic rate of 5.47 W and an average metabolic rate of 10.2 W (3.78 W sensible and 6.42 W latent) during normal activity. If there are 100 chickens in a breeding room, determine the rate of total heat generation and the rate of moisture production in the room. Take the heat of vaporization of water to be 2430 kJ/kg.

9–112 A department store expects to have 120 customers and 15 employees at peak times in summer. Determine the contribution of people to the total cooling load of the store.

9–113E In a movie theater in winter, 500 people, each generating heat at a rate of 100 W, are watching a movie. The heat losses through the walls, windows, and the roof are estimated to be 120,000 Btu/h. Determine if the theater needs to be heated or cooled.

9–114 For an infiltration rate of 1.2 air changes per hour (ACH), determine sensible, latent, and total infiltration heat load of a building at sea level, in kW, that is 20 m long, 13 m wide, and 3 m high when the outdoor air is at 32°C and 50 percent relative humidity. The building is maintained at 24°C and 50 percent relative humidity at all times.

9–115 Repeat Prob. 9–114 for an infiltration rate of 1.8 ACH.

Simple Heating and Cooling

9–116C How do relative and specific humidities change during a simple heating process? Answer the same question for a simple cooling process.

9–117C Why does a simple heating or cooling process appear as a horizontal line on the psychrometric chart?

9–118 Air enters a heating section at 95 kPa, 15°C, and 30 percent relative humidity at a rate of 4 m³/min, and it leaves

at 25°C. Determine (*a*) the rate of heat transfer in the heating section and (*b*) the relative humidity of the air at the exit.
Answers: (*a*) 46.2 kJ/min, (*b*) 16.1 percent

9–119E A heating section consists of a 15-in-diameter duct that houses a 4-kW electric resistance heater. Air enters the heating section at 14.7 psia, 50°F, and 40 percent relative humidity at a velocity of 25 ft/s. Determine (*a*) the exit temperature, (*b*) the exit relative humidity of the air, and (*c*) the exit velocity. *Answers:* (*a*) 56.8°F, (*b*) 30.8 percent, (*c*) 25.4 ft/s

9–120 Air enters a 40-cm-diameter cooling section at 1 atm, 32°C, and 30 percent relative humidity at 18 m/s. Heat is removed from the air at a rate of 1200 kJ/min. Determine (*a*) the exit temperature, (*b*) the exit relative humidity of the air, and (*c*) the exit velocity.
Answers: (*a*) 24.4°C, (*b*) 46.6 percent, (*c*) 17.6 m/s
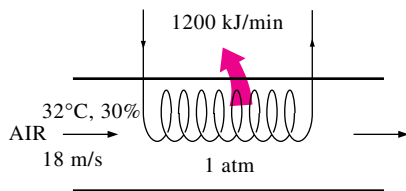
FIGURE P9–120

9–121 Repeat Prob. 9–120 for a heat removal rate of 800 kJ/min.

Heating with Humidification

9–122C Why is heated air sometimes humidified?

9–123 Air at 1 atm, 15°C, and 60 percent relative humidity is first heated to 20°C in a heating section and then humidified by introducing water vapor. The air leaves the humidifying section at 25°C and 65 percent relative humidity. Determine (*a*) the amount of steam added to the air, and (*b*) the amount of heat transfer to the air in the heating section.
Answers: (*a*) 0.0065 kg H_2O/kg dry air, (*b*) 5.1 kJ/kg dry air

9–124E Air at 14.7 psia, 55°F, and 60 percent relative humidity is first heated to 72°F in a heating section and then humidified by introducing water vapor. The air leaves the humidifying section at 75°F and 65 percent relative humidity. Determine (*a*) the amount of steam added to the air, in lbm H_2O/lbm dry air, and (*b*) the amount of heat transfer to the air in the heating section, in Btu/lbm dry air.

9–125 An air-conditioning system operates at a total pressure of 1 atm and consists of a heating section and a humidifier that supplies wet steam (saturated water vapor) at 100°C. Air enters the heating section at 10°C and 70 percent relative humidity at a rate of 35 m³/min, and it leaves the humidifying section at 20°C and 60 percent relative humidity. Determine (*a*) the temperature and relative humidity of air when it leaves the heating section, (*b*) the rate of heat transfer in the heating section, and

(*c*) the rate at which water is added to the air in the humidifying section.
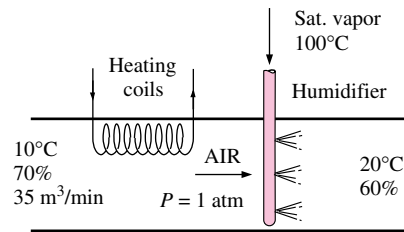

FIGURE P9–125

9–126 Repeat Prob. 9–125 for a total pressure of 95 kPa for the airstream.
Answers: (*a*) 19.5°C, 37.7 percent, (*b*) 391 kJ/min, (*c*) 0.147 kg/min

Cooling with Dehumidification

9–127C Why is cooled air sometimes reheated in summer before it is discharged to a room?

9–128 Air enters a window air conditioner at 1 atm, 32°C, and 70 percent relative humidity at a rate of 3 m³/min, and it leaves as saturated air at 12°C. Part of the moisture in the air that condenses during the process is also removed at 12°C. Determine the rates of heat and moisture removal from the air.
Answers: 173.6 kJ/min, 0.042 kg/min

9–129 An air-conditioning system is to take in air at 1 atm, 34°C, and 70 percent relative humidity and deliver it at 22°C and 50 percent relative humidity. The air flows first over the cooling coils, where it is cooled and dehumidified, and then over the resistance heating wires, where it is heated to the desired temperature. Assuming that the condensate is removed from the cooling section at 10°C, determine (*a*) the temperature of air before it enters the heating section, (*b*) the amount of heat removed in the cooling section, and (*c*) the amount of heat transferred in the heating section, both in kJ/kg dry air.

9–130 Air enters a 30-cm-diameter cooling section at 1 atm, 35°C, and 60 percent relative humidity at 120 m/min. The air is cooled by passing it over a cooling coil through which cold water flows. The water experiences a temperature rise of 8°C. The air leaves the cooling section
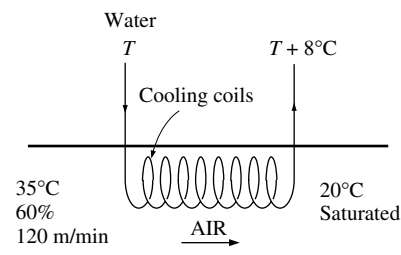

FIGURE P9–130

saturated at 20°C. Determine (*a*) the rate of heat transfer, (*b*) the mass flow rate of the water, and (*c*) the exit velocity of the airstream.

9–131 [EES] Reconsider Prob. 9–130. Using EES (or other) software, develop a general solution of the problem in which the input variables may be supplied and parametric studies performed. For each set of input variables for which the pressure is atmospheric, show the process on the psychrometric chart.

9–132 Repeat Prob. 9–130 for a total pressure of 95 kPa for air.
Answers: (*a*) −293.3 kJ/min, (*b*) 8.77 kg/min, (*c*) 113 m/min

9–133E Air enters a 1-ft-diameter cooling section at 14.7 psia, 90°F, and 60 percent relative humidity at 600 ft/min. The air is cooled by passing it over a cooling coil through which cold water flows. The water experiences a temperature rise of 14°F. The air leaves the cooling section saturated at 70°F. Determine (*a*) the rate of heat transfer, (*b*) the mass flow rate of the water, and (*c*) the exit velocity of the airstream.

9–134E [EES] Reconsider Prob. 9–133E. Using EES (or other) software, study the effect of the total pressure of the air over the range 14.3 to 15.2 psia on the required results. Plot the required results as functions of air total pressure.

9–135E Repeat Prob. 9–133E for a total pressure of 14.4 psia for air.

Evaporative Cooling

9–136C Does an evaporation process have to involve heat transfer? Describe a process that involves both heat and mass transfer.

9–137C During evaporation from a water body to air, under what conditions will the latent heat of vaporization be equal to the heat transfer from the air?

9–138C What is evaporative cooling? Will it work in humid climates?

9–139 Air enters an evaporative cooler at 1 atm, 36°C, and 20 percent relative humidity at a rate of 4 m³/min, and it leaves with a relative humidity of 90 percent. Determine (*a*) the exit temperature of the air and (*b*) the required rate of water supply to the evaporative cooler.
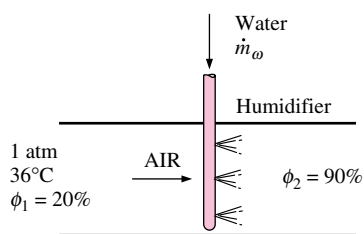

FIGURE P9–139

9–140E Air enters an evaporative cooler at 14.7 psia, 90°F, and 20 percent relative humidity at a rate of 150 ft³/min, and it leaves with a relative humidity of 90 percent. Determine (*a*) the exit temperature of air and (*b*) the required rate of water supply to the evaporative cooler.
Answers: (*a*) 64°F, (*b*) 0.06 lbm/min

9–141 Air enters an evaporative cooler at 95 kPa, 35°C, and 30 percent relative humidity and exits saturated. Determine the exit temperature of air.   *Answer:* 21.1°C

9–142E Air enters an evaporative cooler at 14.5 psia, 93°F, and 30 percent relative humidity and exits saturated. Determine the exit temperature of air.

9–143 Air enters an evaporative cooler at 1 atm, 32°C, and 30 percent relative humidity at a rate of 2 m³/min and leaves at 22°C. Determine (*a*) the final relative humidity and (*b*) the amount of water added to air.

9–144 What is the lowest temperature that air can attain in an evaporative cooler if it enters at 1 atm, 29°C, and 40 percent relative humidity?   *Answer:* 19.3°C

9–145 Air at 1 atm, 15°C, and 60 percent relative humidity is first heated to 30°C in a heating section and then passed through an evaporative cooler where its temperature drops to 25°C. Determine (*a*) the exit relative humidity and (*b*) the amount of water added to air, in kg H_2O/kg dry air.

Adiabatic Mixing of Airstreams

9–146C Two unsaturated airstreams are mixed adiabatically. It is observed that some moisture condenses during the mixing process. Under what conditions will this be the case?

9–147C Consider the adiabatic mixing of two airstreams. Does the state of the mixture on the psychrometric chart have to be on the straight line connecting the two states?

9–148 Two airstreams are mixed steadily and adiabatically. The first stream enters at 32°C and 40 percent relative humidity at a rate of 20 m³/min, while the second stream enters at 12°C and 90 percent relative humidity at a rate of 25 m³/min. Assuming that the mixing process occurs at a pressure of 1 atm, determine the specific humidity, the relative humidity, the dry-bulb temperature, and the volume flow rate of the mixture.
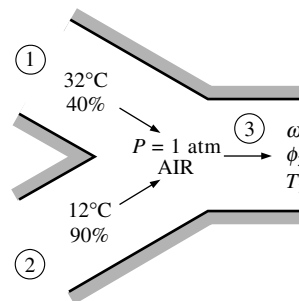*Answers:* 0.0096 kg H_2O/kg dry air, 63.4 percent, 20.6°C, 45.0 m³/min

FIGURE P9–148

9–149 Repeat Prob. 9–148 for a total mixing-chamber pressure of 95 kPa.

9–150E During an air-conditioning process, 900 ft³/min of conditioned air at 65°F and 30 percent relative humidity is mixed adiabatically with 300 ft³/min of outside air at 80°F and 90 percent relative humidity at a pressure of 1 atm. Determine (a) the temperature, (b) the specific humidity, and (c) the relative humidity of the mixture.
Answers: (a) 69.5°F, *(b)* 0.0078 lbm H₂O/lbm dry air, *(c)* 49.0 percent

9–151E Reconsider Prob. 9–150E. Using EES (or other) software, develop a general solution of the problem in which the input variables may be supplied and parametric studies performed. For each set of input variables for which the pressure is atmospheric, show the process on the psychrometric chart.

9–152 A stream of warm air with a dry-bulb temperature of 40°C and a wet-bulb temperature of 32°C is mixed adiabatically with a stream of saturated cool air at 18°C. The dry air mass flow rates of the warm and cool airstreams are 8 and 6 kg/s, respectively. Assuming a total pressure of 1 atm, determine (a) the temperature, (b) the specific humidity, and (c) the relative humidity of the mixture.

9–153 Reconsider Prob. 9–152. Using EES (or other) software, determine the effect of the mass flow rate of saturated cool airstream on the mixture temperature, specific humidity, and relative humidity. Vary the mass flow rate of saturated cool air from 0 to 16 kg/s while maintaining the mass flow rate of warm air constant at 8 kg/s. Plot the mixture temperature, specific humidity, and relative humidity as functions of the mass flow rate of cool air, and discuss the results.

Wet Cooling Towers

9–154C How does a natural-draft wet cooling tower work?

9–155C What is a spray pond? How does its performance compare to the performance of a wet cooling tower?

9–156 The cooling water from the condenser of a power plant enters a wet cooling tower at 40°C at a rate of 90 kg/s. The water is cooled to 25°C in the cooling tower by air that enters the tower at 1 atm, 23°C, and 60 percent relative humidity and leaves saturated at 32°C. Neglecting the power input to the fan, determine (a) the volume flow rate of air into the cooling tower and (b) the mass flow rate of the required makeup water.

9–157E The cooling water from the condenser of a power plant enters a wet cooling tower at 110°F at a rate of 100 lbm/s. Water is cooled to 80°F in the cooling tower by air that enters the tower at 1 atm, 76°F, and 60 percent relative humidity and leaves saturated at 95°F. Neglecting the power input to the fan, determine (a) the volume flow rate of air into the cooling tower and (b) the mass flow rate of the required makeup water.
Answers: (a) 1325 ft³/s, *(b)* 2.42 lbm/s

9–158 A wet cooling tower is to cool 60 kg/s of water from 40 to 26°C. Atmospheric air enters the tower at 1 atm with dry- and wet-bulb temperatures of 22 and 16°C, respectively, and leaves at 34°C with a relative humidity of 90 percent. Using the psychrometric chart, determine (a) the volume flow rate of air into the cooling tower and (b) the mass flow rate of the required makeup water. *Answers: (a)* 44.9 m³/s, *(b)* 1.16 kg/s
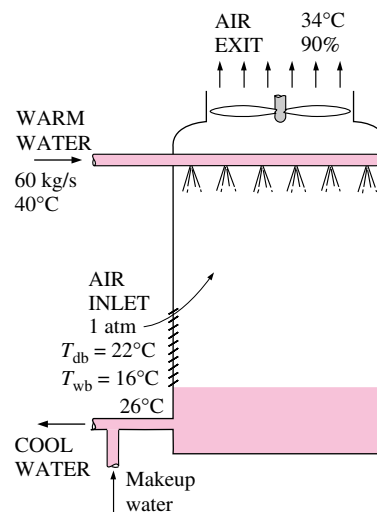

FIGURE P9–158

9–159 A wet cooling tower is to cool 50 kg/s of cooling water from 40 to 25°C at a location where the atmospheric pressure is 96 kPa. Atmospheric air enters the tower at 20°C and 70 percent relative humidity and leaves saturated at 35°C. Neglecting the power input to the fan, determine (a) the volume flow rate of air into the cooling tower and (b) the mass flow rate of the required makeup water.
Answers: (a) 33.2 m³/s, *(b)* 1.04 kg/s

Review Problems

9–160 Air has the following composition on a mole basis: 21 percent O₂, 78 percent N₂, and 1 percent Ar. Determine the gravimetric analysis of air and its molar mass. *Answers:* 23.2 percent O₂, 75.4 percent N₂, 1.4 percent Ar; 28.96 kg/kmol

9–161 Using EES (or other) software, write a program to determine the mole fractions of the components of a mixture of three gases with known molar masses when the mass fractions are given, and to determine the mass fractions of the components when the mole fractions are given. Run the program for a sample case, and give the results.

9–162 Using EES (or other) software, write a program to determine the apparent gas constant, constant-volume specific heat, and internal energy of a mixture of three ideal gases when the mass fractions and other properties of the constituent gases are given. Run the program for a sample case, and give the results.

9–163 The condensation of the water vapor in compressed-air lines is a major concern in industrial facilities, and the compressed air is often dehumidified to avoid the problems associated with condensation. Consider a compressor that compresses ambient air from the local atmospheric pressure of 92 kPa to a pressure of 800 kPa (absolute). The compressed air is then cooled to the ambient temperature as it flows through the compressed-air lines. Disregarding any pressure losses, determine if there will be any condensation in the compressed-air lines on a day when the ambient air is at 25°C and 40 percent relative humidity.

9–164E The relative humidity of air at 80°F and 14.7 psia is increased from 30 percent to 90 percent during a humidification process at constant temperature and pressure. Determine the percent error involved in assuming the density of air to have remained constant.

9–165 Dry air whose molar analysis is 78.1 percent N_2, 20.9 percent O_2, and 1 percent Ar flows over a water body until it is saturated. If the pressure and temperature of air remain constant at 1 atm and 25°C during the process, determine (*a*) the molar analysis of the saturated air and (*b*) the density of air before and after the process. What do you conclude from your results?

9–166E Determine the mole fraction of the water vapor at the surface of a lake whose surface temperature is 60°F, and compare it to the mole fraction of water in the lake, which is very nearly 1.0. The air at the lake surface is saturated, and the atmospheric pressure at lake level can be taken to be 13.8 psia.

9–167 Determine the mole fraction of dry air at the surface of a lake whose temperature is 12°C. The air at the lake surface is saturated, and the atmospheric pressure at lake level can be taken to be 100 kPa.

9–168E Consider a room that is cooled adequately by an air conditioner whose cooling capacity is 7500 Btu/h. If the room is to be cooled by an evaporative cooler that removes heat at the same rate by evaporation, determine how much water needs to be supplied to the cooler per hour at design conditions.

9–169E The capacity of evaporative coolers is usually expressed in terms of the flow rate of air in ft^3/min (or cfm), and a practical way of determining the required size of an evaporative cooler for an 8-ft-high house is to multiply the floor area of the house by 4 (by 3 in dry climates and by 5 in humid climates). For example, the capacity of an evaporative cooler for a 30-ft-long, 40-ft-wide house is $1200 \times 4 = 4800$ cfm. Develop an equivalent rule of thumb for the selection of an evaporative cooler in SI units for 2.4-m-high houses whose floor areas are given in m^2.

9–170 A cooling tower with a cooling capacity of 100 tons (440 kW) is claimed to evaporate 15,800 kg of water per day. Is this a reasonable claim?

9–171 The U.S. Department of Energy estimates that 190,000 barrels of oil would be saved per day if every household in the United States raised the thermostat setting in summer by 6°F (3.3°C). Assuming the average cooling season to be 120 days and the cost of oil to be $20/barrel, determine how much money would be saved per year.

9–172E The thermostat setting of a house can be lowered by 2°F by wearing a light long-sleeved sweater, or by 4°F by wearing a heavy long-sleeved sweater for the same level of comfort. If each °F reduction in thermostat setting reduces the heating cost of a house by 4 percent at a particular location, determine how much the heating costs of a house can be reduced by wearing heavy sweaters if the annual heating cost of the house is $600.

9–173 The air-conditioning costs of a house can be reduced by up to 10 percent by installing the outdoor unit (the condenser) of the air conditioner at a location shaded by trees and shrubs. If the air-conditioning costs of a house are $500 a year, determine how much the trees will save the home owner in the 20-year life of the system.

9–174 A 5-m^3 tank contains saturated air at 25°C and 97 kPa. Determine (*a*) the mass of the dry air, (*b*) the specific humidity, and (*c*) the enthalpy of the air per unit mass of the dry air.
Answers: (*a*) 5.49 kg, (*b*) 0.0210 kg H_2O/kg dry air, (*c*) 78.62 kJ/kg dry air

9–175 Reconsider Prob. 9–174. Using EES (or other) software, determine the properties of the air at the initial state. Study the effects of heating the air at constant volume until the pressure is 110 kPa. Plot the required heat transfer, in kJ, as a function of pressure.

9–176E Air at 15 psia, 60°F, and 50 percent relative humidity flows in an 8-in-diameter duct at a velocity of 50 ft/s. Determine (*a*) the dew-point temperature, (*b*) the volume flow rate of air, and (*c*) the mass flow rate of dry air.

9–177 Air enters a cooling section at 97 kPa, 35°C, and 20 percent relative humidity at a rate of 9 m^3/min, where it is cooled until the moisture in the air starts condensing. Determine (*a*) the temperature of the air at the exit and (*b*) the rate of heat transfer in the cooling section.

9–178 Outdoor air enters an air-conditioning system at 10°C and 40 percent relative humidity at a steady rate of 22 m^3/min, and it leaves at 25°C and 55 percent relative humidity. The outdoor air is first heated to 22°C in the heating section and then humidified by the injection of hot steam in the humidifying section. Assuming the entire process takes place at a pressure of 1 atm, determine (*a*) the rate of heat supply in the heating section and (*b*) the mass flow rate of steam required in the humidifying section.

9–179 Air enters an air-conditioning system that uses refrigerant-134a at 30°C and 70 percent relative humidity at a rate of 4 m^3/min. The refrigerant enters the cooling section at

700 kPa with a quality of 20 percent and leaves as saturated vapor. The air is cooled to 20°C at a pressure of 1 atm. Determine (*a*) the rate of dehumidification, (*b*) the rate of heat transfer, and (*c*) the mass flow rate of the refrigerant.

9–180 Repeat Prob. 9–179 for a total pressure of 95 kPa for air.

9–181 An air-conditioning system operates at a total pressure of 1 atm and consists of a heating section and an evaporative cooler. Air enters the heating section at 10°C and 70 percent relative humidity at a rate of 30 m^3/min, and it leaves the evaporative cooler at 20°C and 60 percent relative humidity. Determine (*a*) the temperature and relative humidity of the air when it leaves the heating section, (*b*) the rate of heat transfer in the heating section, and (*c*) the rate of water added to air in the evaporative cooler.
Answers: (*a*) 28.3°C, 23.0 percent, (*b*) 696 kJ/min, (*c*) 0.13 kg/min

9–182 Reconsider Prob. 9–181. Using EES (or other) software, study the effect of total pressure in the range 94 to 100 kPa on the results required in the problem. Plot the results as functions of total pressure.

9–183 Repeat Prob. 9–181 for a total pressure of 96 kPa.

9–184 Conditioned air at 13°C and 90 percent relative humidity is to be mixed with outside air at 34°C and 40 percent relative humidity at 1 atm. If it is desired that the mixture have a relative humidity of 60 percent, determine (*a*) the ratio of the dry air mass flow rates of the conditioned air to the outside air and (*b*) the temperature of the mixture.

9–185 Reconsider Prob. 9–184. Determine the desired quantities using EES (or other) software instead of the psychrometric chart. What would the answers be at a location at an atmospheric pressure of 80 kPa?

9–186 A natural-draft cooling tower is to remove 50 MW of waste heat from the cooling water that enters the tower at 42°C and leaves at 27°C. Atmospheric air enters the tower at 1 atm with dry- and wet-bulb temperatures of 23 and 18°C, respectively, and leaves saturated at 37°C. Determine (*a*) the mass flow rate of the cooling water, (*b*) the volume flow rate of air into the cooling tower, and (*c*) the mass flow rate of the required makeup water.

9–187 Reconsider Prob. 9–186. Using EES (or other) software, investigate the effect of air inlet wet-bulb temperature on the required air volume flow rate and the makeup water flow rate when the other input data are the stated values. Plot the results as functions of wet-bulb temperature.

Design and Essay Problems

9–188 Prolonged exposure to mercury even at relatively low but toxic concentrations in the air is known to cause permanent mental disorders, insomnia, and pain and numbness in the hands and the feet, among other things. Therefore, the maximum allowable concentration of mercury vapor in the air at workplaces is regulated by federal agencies. These regulations require that the average level of mercury concentration in the air does not exceed 0.1 mg/m^3.

Consider a mercury spill that occurs in an airtight storage room at 20°C in San Francisco during an earthquake. Calculate the highest level of mercury concentration in the air that can occur in the storage room, in mg/m^3, and determine if it is within the safe level. The vapor pressure of mercury at 20°C is 0.173 Pa. Propose some guidelines to safeguard against the formation of toxic concentrations of mercury vapor in air in storage rooms and laboratories.

9–189 The condensation and even freezing of moisture in building walls without effective vapor retarders are of real concern in cold climates as they undermine the effectiveness of the insulation. Investigate how the builders in your area are coping with this problem, whether they are using vapor retarders or vapor barriers in the walls, and where they are located in the walls. Prepare a report on your findings, and explain the reasoning for the current practice.

9–190 The air-conditioning needs of a large building can be met by a single central system or by several individual window units. Considering that both approaches are commonly used in practice, the right choice depends on the situation on hand. Identify the important factors that need to be considered in decision making, and discuss the conditions under which an air-conditioning system that consists of several window units is preferable over a large single central system, and vice versa.

9–191 Identify the major sources of heat gain in your house in summer, and propose ways of minimizing them and thus reducing the cooling load.

9–192 Write an essay on different humidity measurement devices, including electronic ones, and discuss the advantages and disadvantages of each device.

9–193 Design an inexpensive evaporative cooling system suitable for use in your house. Show how you would obtain a water spray, how you would provide airflow, and how you would prevent water droplets from drifting into the living space.