Chapter 14

GAS–VAPOR MIXTURES AND AIR-CONDITIONING

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.

In Chap. 13, we discussed mixtures of gases that are usually above their critical temperatures. Therefore, we were not concerned about any of the gases condensing during a process. Not having to deal with two phases greatly simplified the analysis. When we are dealing with a gas–vapor mixture, however, 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.

Objectives

The objectives of Chapter 14 are to:

- Differentiate between *dry air* and *atmospheric air*.
- Define and calculate the specific and relative humidity of atmospheric air.
- Calculate the dew-point temperature of atmospheric air.
- Relate the adiabatic saturation temperature and wet-bulb temperatures of atmospheric air.
- Use the psychrometric chart as a tool to determine the properties of atmospheric air.
- Apply the principles of the conservation of mass and energy to various air-conditioning processes.

14–1 ▪ 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. 14–1. Taking 0°C as the reference temperature, the enthalpy and enthalpy change of dry air can be determined from

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 14–1

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.

$$h_{\text{dry air}} = c_p T = (1.005 \text{ kJ/kg} \cdot {}^\circ\text{C})T \qquad (\text{kJ/kg}) \tag{14–1a}$$

and

$$\Delta h_{\text{dry air}} = c_p \Delta T = (1.005 \text{ kJ/kg} \cdot {}^\circ\text{C})\, \Delta T \qquad (\text{kJ/kg}) \tag{14–1b}$$

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:

$$P = P_a + P_v \qquad (\text{kPa}) \tag{14–2}$$

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. 14–2 where the constant-enthalpy lines coincide with constant-temperature lines at temperatures

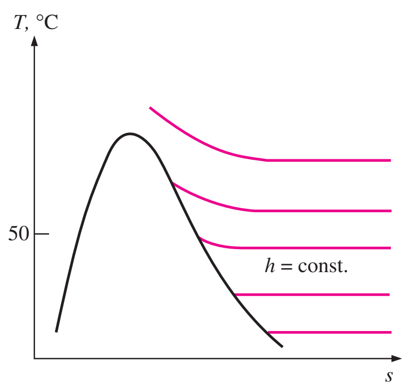


FIGURE 14–2

At temperatures below 50°C, the h = constant lines coincide with the T = constant lines in the superheated vapor region of water.

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

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,

$$h_v(T, \text{low } P) \cong h_g(T) \tag{14–3}$$

The enthalpy of water vapor at 0°C is 2500.9 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

$$h_g(T) \cong 2500.9 + 1.82T \qquad (\text{kJ/kg}) \qquad T \text{ in °C} \tag{14–4}$$

or

$$h_g(T) \cong 1060.9 + 0.435T \qquad (\text{Btu/lbm}) \qquad T \text{ in °F} \tag{14–5}$$

in the temperature range −10 to 50°C (or 15 to 120°F), with negligible error, as shown in Fig. 14–3.

WATER VAPOR

	h_g, kJ/kg		
T, °C	Table A-4	Eq. 14-4	Difference, kJ/kg
−10	2482.1	2482.7	−0.6
0	2500.9	2500.9	0.0
10	2519.2	2519.1	0.1
20	2537.4	2537.3	0.1
30	2555.6	2555.5	0.1
40	2573.5	2573.7	−0.2
50	2591.3	2591.9	−0.6

FIGURE 14–3
In the temperature range −10 to 50°C, the h_g of water can be determined from Eq. 14–4 with negligible error.

14–2 ▪ 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:

$$\omega = \frac{m_v}{m_a} \qquad (\text{kg water vapor/kg dry air}) \tag{14–6}$$

The specific humidity can also be expressed as

$$\omega = \frac{m_v}{m_a} = \frac{P_v V/R_v T}{P_a V/R_a T} = \frac{P_v/R_v}{P_a/R_a} = 0.622\frac{P_v}{P_a} \tag{14–7}$$

or

$$\omega = \frac{0.622P_v}{P - P_v} \qquad (\text{kg water vapor/kg dry air}) \tag{14–8}$$

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. 14–8 by replacing P_v by P_g, the saturation pressure of water at that temperature (Fig. 14–4).

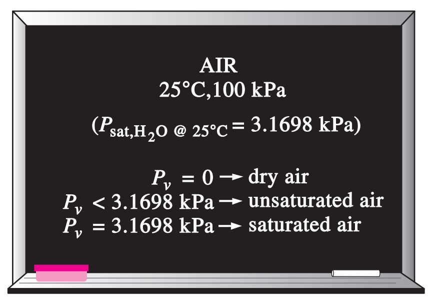


FIGURE 14–4
For saturated air, the vapor pressure is equal to the saturation pressure of water.

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. 14–5)

$$\phi = \frac{m_v}{m_g} = \frac{P_v V/R_v T}{P_g V/R_v T} = \frac{P_v}{P_g} \tag{14–9}$$

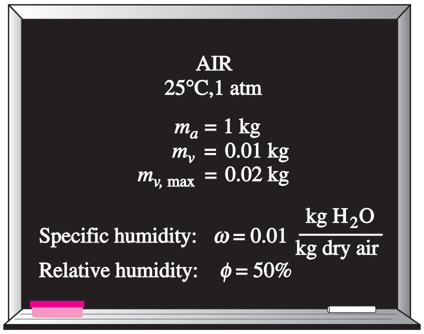


FIGURE 14–5

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 at a given temperature to the maximum amount of moisture air can hold at the same temperature.

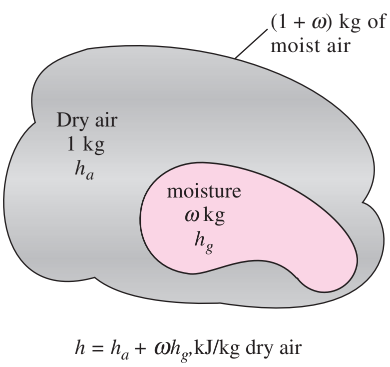


FIGURE 14–6

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

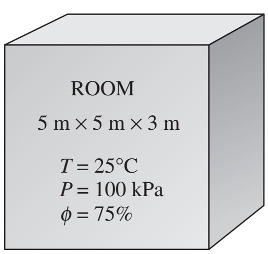


FIGURE 14–7

Schematic for Example 14–1.

where

$$P_g = P_{\text{sat @ }T} \tag{14–10}$$

Combining Eqs. 14–8 and 14–9, we can also express the relative humidity as

$$\phi = \frac{\omega P}{(0.622 + \omega)P_g} \quad \text{and} \quad \omega = \frac{0.622\phi P_g}{P - \phi P_g} \tag{14–11a, b}$$

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

$$h = h_a + \omega h_g \qquad (\text{kJ/kg dry air}) \tag{14–12}$$

since $h_v \cong h_g$ (Fig. 14–6).

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 14–1 The Amount of Water Vapor in Room Air

A 5-m × 5-m × 3-m room shown in Fig. 14–7 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 The relative humidity of air in a room is given. The dry air pressure, specific humidity, enthalpy, and the masses of dry air and water vapor in the room are to be determined.

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

Properties The constant-pressure specific heat of air at room temperature is c_p=1.005 kJ/kg · K (Table A–2*a*). For water at 25°C, we have T_{sat} = 3.1698 kPa and h_g = 2546.5 kJ/kg (Table A–4).

Analysis (*a*) The partial pressure of dry air can be determined from Eq. 14–2:

$$P_a = P - P_v$$

where

$$P_v = \phi P_g = \phi P_{\text{sat @ 25°C}} = (0.75)(3.1698 \text{ kPa}) = 2.38 \text{ kPa}$$

Thus,

$$P_a = (100 - 2.38) \text{ kPa} = \mathbf{97.62 \text{ kPa}}$$

(*b*) The specific humidity of air is determined from Eq. 14–8:

$$\omega = \frac{0.622P_v}{P - P_v} = \frac{(0.622)(2.38 \text{ kPa})}{(100 - 2.38) \text{ kPa}} = \mathbf{0.0152 \text{ kg } H_2O/\text{kg dry air}}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from Eq. 14–12:

$$\begin{aligned} h &= h_a + \omega h_v \cong c_pT + \omega h_g \\ &= (1.005 \text{ kJ/kg}\cdot°\text{C})(25°\text{C}) + (0.0152)(2546.5 \text{ kJ/kg}) \\ &= \mathbf{63.8 \text{ kJ/kg dry air}} \end{aligned}$$

The enthalpy of water vapor (2546.5 kJ/kg) could also be determined from the approximation given by Eq. 14–4:

$$h_{g \text{ @ 25°C}} \cong 2500.9 + 1.82(25) = 2546.4 \text{ kJ/kg}$$

which is almost identical 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:

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

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

$$m_a = \frac{P_aV_a}{R_aT} = \frac{(97.62 \text{ kPa})(75 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})} = \mathbf{85.61 \text{ kg}}$$

$$m_v = \frac{P_vV_v}{R_vT} = \frac{(2.38 \text{ kPa})(75 \text{ m}^3)}{(0.4615 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})} = \mathbf{1.30 \text{ kg}}$$

The mass of the water vapor in the air could also be determined from Eq. 14–6:

$$m_v = \omega m_a = (0.0152)(85.61 \text{ kg}) = 1.30 \text{ kg}$$

14–3 ▪ 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

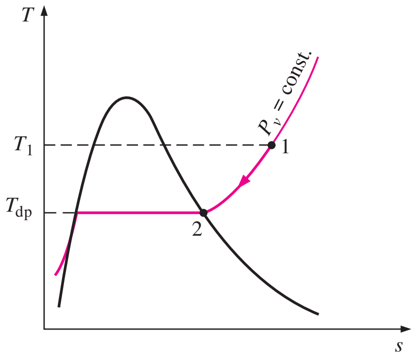


FIGURE 14–8

Constant-presssure cooling of moist air and the dew-point temperature on the *T-s* diagram of water.

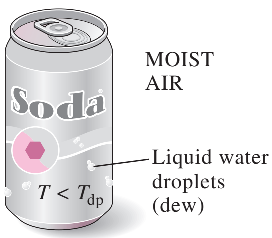


FIGURE 14–9

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

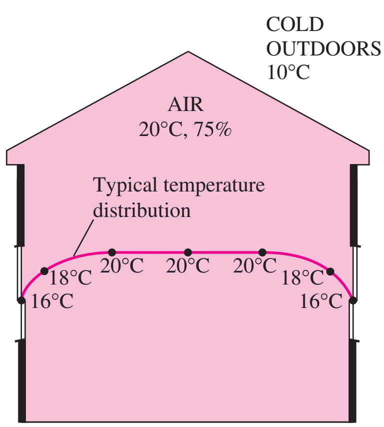


FIGURE 14–10

Schematic for Example 14–2.

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:

$$T_{dp} = T_{sat @ P_v} \tag{14–13}$$

This is also illustrated in Fig. 14–8. As the air cools at constant pressure, the vapor pressure P_v remains constant. Therefore, the vapor in the air (state 1) 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. 14–9).

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.

EXAMPLE 14–2 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. 14–10, 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 starts is to be determined.

Properties The saturation pressure of water at 20°C is P_{sat} = 2.3392 kPa (Table A–4).

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 undergoes a P_v = *constant* cooling process until the moisture in the air starts condensing. This happens when the air reaches its dew-point temperature T_{dp}, which is determined from Eq. 14–13 to be

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

where

$$P_v = \phi P_{g\ @\ 20°C} = (0.75)(2.3392\ kPa) = 1.754\ kPa$$

Thus,

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

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

14–4 ▪ 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 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. 14–11. 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 evaporates and mixes with the airstream. The moisture content of air increases during this process, and its temperature decreases, since part of the latent heat of vaporization of the water that evaporates comes from the air. If the channel is long enough, the airstream exits as saturated air (ϕ = 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:

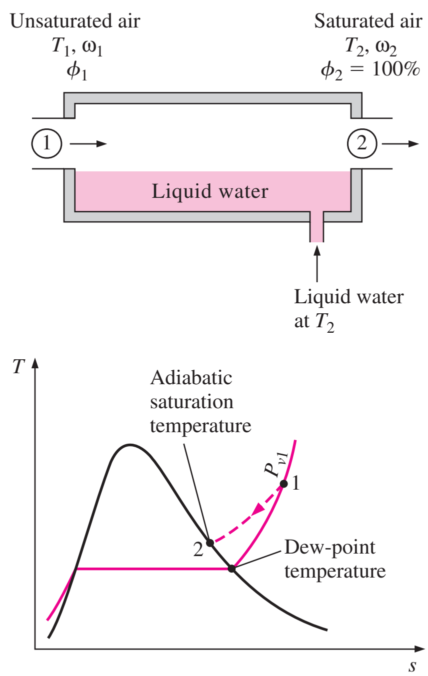


FIGURE 14–11

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

Mass balance:

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

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

or

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

Thus,

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

Energy balance:

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

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

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

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg_2}}{h_{g_1} - h_{f_2}} \qquad \textbf{(14–14)}$$

where, from Eq. 14–11*b*,

$$\omega_2 = \frac{0.622P_{g_2}}{P_2 - P_{g_2}} \qquad \textbf{(14–15)}$$

since $\phi_2 = 100$ percent. Thus we conclude that the specific humidity (and relative humidity) of air can be determined from Eqs. 14–14 and 14–15 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. 14–14 yields $\omega_1 = \omega_2$. In general, the adiabatic saturation temperature is 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. 14–12. The temperature measured in this manner 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 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

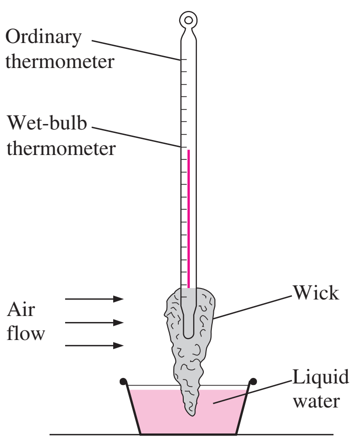


FIGURE 14–12

A simple arrangement to measure the wet-bulb temperature.

on this principle is called a *sling psychrometer* and is shown in Fig. 14–13. 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. 14–14 in place of T_2 to determine the specific humidity of air.

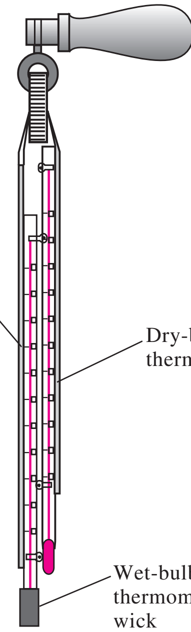


FIGURE 14–13

Sling psychrometer.

EXAMPLE 14–3 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.

Properties The saturation pressure of water is 1.7057 kPa at 15°C, and 3.1698 kPa at 25°C (Table A–4). The constant-pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg · K (Table A–2a).

Analysis (*a*) The specific humidity ω_1 is determined from Eq. 14–14,

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

where T_2 is the wet-bulb temperature and ω_2 is

$$\omega_2 = \frac{0.622P_{g_2}}{P_2 - P_{g_2}} = \frac{(0.622)(1.7057\text{ kPa})}{(101.325 - 1.7057)\text{ kPa}}$$

$$= 0.01065\text{ kg H}_2\text{O/kg dry air}$$

Thus,

$$\omega_1 = \frac{(1.005\text{ kJ/kg}\cdot{}^\circ\text{C})[(15 - 25)^\circ\text{C}] + (0.01065)(2465.4\text{ kJ/kg})}{(2546.5 - 62.982)\text{ kJ/kg}}$$

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

(*b*) The relative humidity ϕ_1 is determined from Eq. 14–11*a* to be

$$\phi_1 = \frac{\omega_1 P_2}{(0.622 + \omega_1)P_{g_1}} = \frac{(0.00653)(101.325\text{ kPa})}{(0.622 + 0.00653)(3.1698\text{ kPa})} = \mathbf{0.332\ or\ 33.2\%}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from Eq. 14–12:

$$h_1 = h_{a_1} + \omega_1 h_{v_1} \cong c_p T_1 + \omega_1 h_{g_1}$$
$$= (1.005 \text{ kJ/kg}\cdot°\text{C})(25°\text{C}) + (0.00653)(2546.5 \text{ kJ/kg})$$
$$= \mathbf{41.8 \text{ kJ/kg dry air}}$$

Discussion The previous property calculations can be performed easily using EES or other programs with built-in psychrometric functions.

14–5 ▪ THE PSYCHROMETRIC CHART

The state of the atmospheric air at a specified pressure is completely specified by two independent intensive properties. The rest of the properties can be calculated easily from the previous relations. 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 calculations 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–31 in SI units and in Fig. A–31E in English units. Psychrometric charts at other pressures (for use at considerably higher elevations than sea level) are also available.

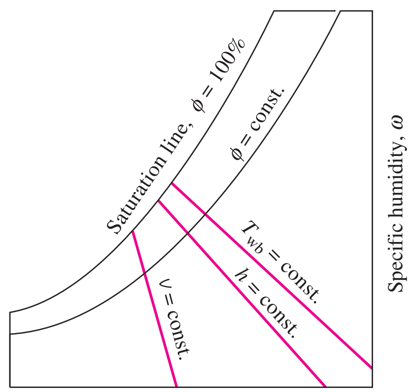


FIGURE 14–14

Schematic for a psychrometric chart.

The basic features of the psychrometric chart are illustrated in Fig. 14–14. 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 pressure 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. 14–8.) 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 located on this curve. Therefore, it is also the curve of 100 percent relative humidity. 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.

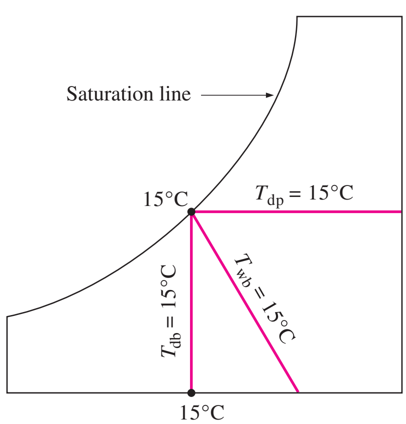


FIGURE 14–15

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

For saturated air, the dry-bulb, wet-bulb, and dew-point temperatures are identical (Fig. 14–15). 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, appears as a horizontal line on this chart if no humidification or dehumidification 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.

EXAMPLE 14–4 The Use of the Psychrometric Chart

Consider a room that contains air at 1 atm, 35°C, and 40 percent relative humidity. 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 The relative humidity of air in a room is given. The specific humidity, enthalpy, wet-bulb temperature, dew-point temperature, and specific volume of the air are to be determined using the psychrometric chart.

Analysis 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.

(*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. 14–16. At the intersection point we read

$$\omega = \mathbf{0.0142\ kg\ H_2O/kg\ dry\ air}$$

(*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, giving

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

(*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, giving

$$T_{wb} = \mathbf{24°C}$$

(*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, giving

$$T_{dp} = \mathbf{19.4°C}$$

(*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

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

Discussion Values read from the psychrometric chart inevitably involve reading errors, and thus are of limited accuracy.

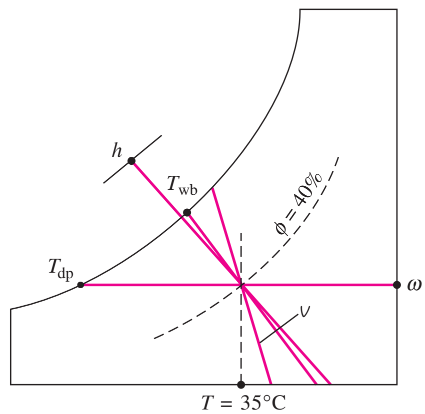


FIGURE 14–16
Schematic for Example 14–4.

14–6 ▪ 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.)

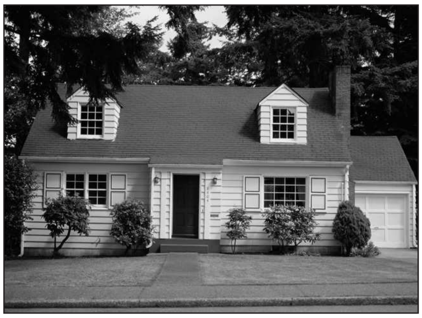


FIGURE 14–17

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

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. 14–17). 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 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 about 37°C.) A body will feel comfortable in environments in which it can dissipate this waste heat comfortably (Fig. 14–18).

FIGURE 14–18

A body feels comfortable when it can freely dissipate its waste heat, and no more.

Heat transfer is proportional to the temperature difference. Therefore in cold environments, a body loses 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 causes 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 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. 14–19). The temperature of the environment is the single most important index of comfort. 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 evaporation, 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 evaporation. 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 *wind-chill factor*). Other factors that affect comfort are air cleanliness, odor, noise, and radiation effect.

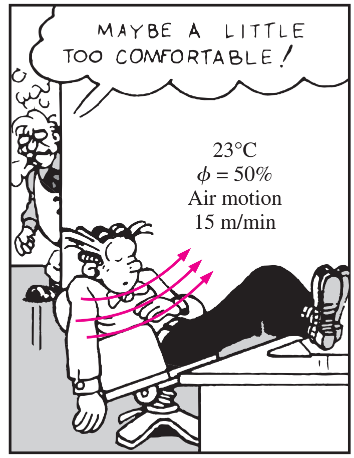


FIGURE 14–19

A comfortable environment.

14–7 ▪ 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 cooling* (lowering the temperature), *humidifying* (adding moisture), and *dehumidifying* (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. 14–20. Notice that simple heating and cooling processes appear as horizontal 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.

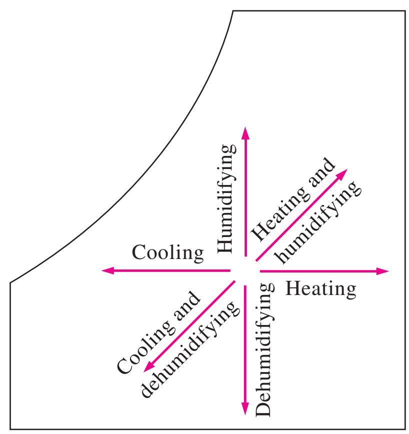


FIGURE 14–20

Various air-conditioning processes.

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

Mass balance for dry air: $$\sum_{in} \dot{m}_a = \sum_{out} \dot{m}_a \qquad \text{(kg/s)} \tag{14–16}$$

Mass balance for water: $$\sum_{in} \dot{m}_w = \sum_{out} \dot{m}_w \quad \text{or} \quad \sum_{in} \dot{m}_a \omega = \sum_{out} \dot{m}_a \omega \tag{14–17}$$

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

$$\dot{Q}_{in} + \dot{W}_{in} + \sum_{in} \dot{m}h = \dot{Q}_{out} + \dot{W}_{out} + \sum_{out} \dot{m}h \tag{14–18}$$

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 commonly encountered processes in air-conditioning.

Simple Heating and Cooling (ω = constant)

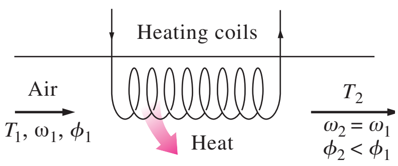


FIGURE 14–21
During simple heating, specific humidity remains constant, but relative humidity decreases.

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. 14–21. 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 proceeds 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.

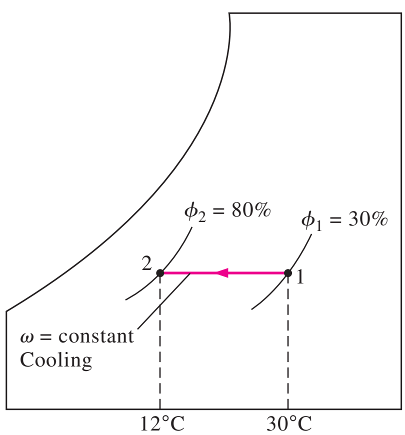


FIGURE 14–22
During simple cooling, specific humidity remains constant, but relative humidity increases.

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. 14–22. 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. 14–23.

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 comes from the air, which results 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.

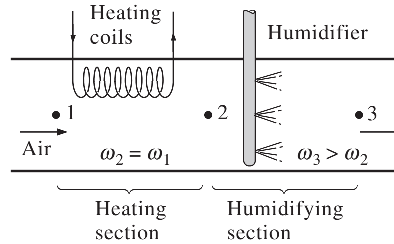


FIGURE 14–23

Heating with humidification.

EXAMPLE 14–5 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 m^3/min and to condition it to 25°C and 60 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 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 Outdoor air is first heated and then humidified by steam injection. The rate of heat transfer and the mass flow rate of steam are to be determined.

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.

Properties The constant-pressure specific heat of air at room temperature is $c_p = 1.005$ kJ/kg · K, and its gas constant is $R_a = 0.287$ kJ/kg · K (Table A–2*a*). The saturation pressure of water is 1.2281 kPa at 10°C, and 3.1698 kPa at 25°C. The enthalpy of saturated water vapor is 2519.2 kJ/kg at 10°C, and 2541.0 kJ/kg at 22°C (Table A–4).

Analysis We take the system to be the *heating* or the *humidifying section,* as appropriate. The schematic of the system and the psychrometric chart of the process are shown in Fig. 14–24. 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 ($\omega_3 > \omega_2$).

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

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

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

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

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

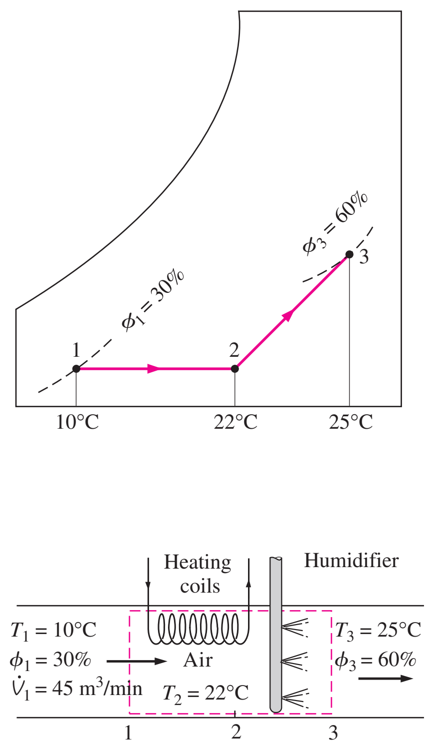


FIGURE 14–24

Schematic and psychrometric chart for Example 14–5.

1 atm, either other charts for that pressure or the relations developed earlier should be used. In our case, the choice is clear:

$$P_{v_1} = \phi_1 P_{g_1} = \phi P_{\text{sat @ 10°C}} = (0.3)(1.2281 \text{ kPa}) = 0.368 \text{ kPa}$$

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

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

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

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

$$h_1 = c_p T_1 + \omega_1 h_{g_1} = (1.005 \text{ kJ/kg}\cdot°\text{C})(10°\text{C}) + (0.0023)(2519.2 \text{ kJ/kg})$$
$$= 15.8 \text{ kJ/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g_2} = (1.005 \text{ kJ/kg}\cdot°\text{C})(22°\text{C}) + (0.0023)(2541.0 \text{ kJ/kg})$$
$$= 28.0 \text{ kJ/kg dry air}$$

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

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

(*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

$$\omega_3 = \frac{0.622\phi_3 P_{g_3}}{P_3 - \phi_3 P_{g_3}} = \frac{0.622(0.60)(3.1698 \text{ kPa})}{[100 - (0.60)(3.1698)] \text{ kPa}}$$
$$= 0.01206 \text{ kg H}_2\text{O/kg dry air}$$

Thus,

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

Discussion The result 0.539 kg/min corresponds to a water requirement of close to one ton a day, which is significant.

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. 14–25 in conjunction with Example 14–6. 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 constant specific humidity. If the cooling section is sufficiently long, air reaches its dew point (state *x*, 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 2) 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_2.

The cool, saturated air at state 2 is usually routed directly to the room, where it mixes with the room air. In some cases, however, the air at state 2 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.

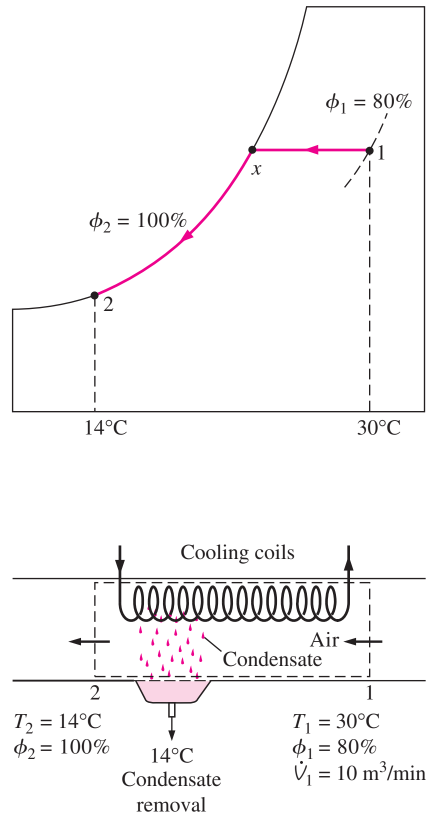


FIGURE 14–25

Schematic and psychrometric chart for Example 14–6.

EXAMPLE 14–6 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 m^3/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 Air is cooled and dehumidified by a window air conditioner. The rates of heat and moisture removal are to be determined.

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.

Properties The enthalpy of saturated liquid water at 14°C is 58.8 kJ/kg (Table A–4). Also, 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

$$h_1 = 85.4 \text{ kJ/kg dry air} \qquad h_2 = 39.3 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.0216 \text{ kg H}_2\text{O/kg dry air} \quad \text{and} \quad \omega_2 = 0.0100 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.889 \text{ m}^3\text{/kg dry air}$$

Analysis 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. 14–25. We note that the amount of water vapor in the air decreases during the process ($\omega_2 < \omega_1$) due to dehumidification. Applying the mass and energy balances on the cooling and dehumidification section gives

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

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

Energy balance: $$\sum_{\text{in}} \dot{m}h = \dot{Q}_{\text{out}} + \sum_{\text{out}} \dot{m}h \quad \rightarrow \quad \dot{Q}_{\text{out}} = \dot{m}(h_1 - h_2) - \dot{m}_w h_w$$

Then,

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

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

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

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

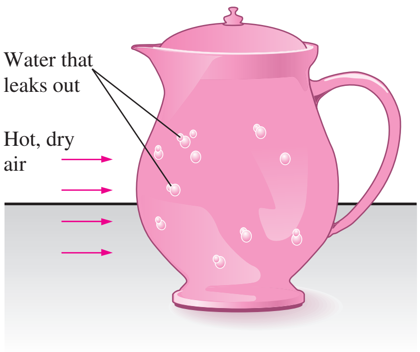


FIGURE 14–26

Water in a porous jug left in an open, breezy area cools as a result of evaporative cooling.

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. 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. 14–26).

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. 14–27. 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 leaves the evaporative cooler saturated at state 2′. This is the lowest temperature that can be achieved by this process.

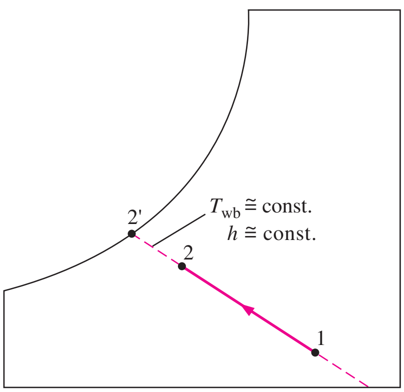
The evaporative cooling process is essentially identical to the adiabatic saturation process since the heat transfer between the airstream and the surroundings is usually negligible. Therefore, the evaporative cooling process follows a line of constant wet-bulb temperature on the psychrometric chart. (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,

$$T_{\text{wb}} \cong \text{constant} \tag{14–19}$$

and

$$h \cong \text{constant} \tag{14–20}$$

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



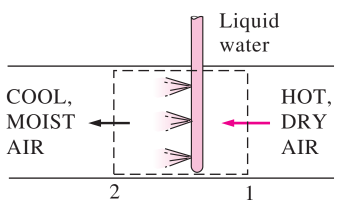


FIGURE 14–27

Evaporative cooling.

EXAMPLE 14–7 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 Air is cooled steadily by an evaporative cooler. The temperature of discharged air and the lowest temperature to which the air can be cooled are to be determined.

Analysis The schematic of the evaporative cooler and the psychrometric chart of the process are shown in Fig. 14–28.

(*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,

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

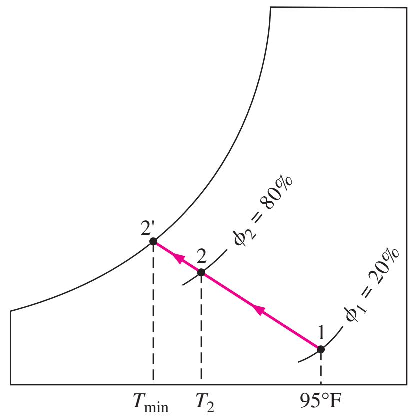
The wet-bulb temperature at 95°F and 20 percent relative humidity is determined from the psychrometric chart to be 66.0°F. The intersection point of the $T_{\text{wb}} = 66.0$°F and the $\phi = 80$ percent lines is the exit state of the air. The temperature at this point is the exit temperature of the air, and it is determined from the psychrometric chart to be

$$T_2 = \mathbf{70.4°F}$$

(*b*) In the limiting case, air leaves the evaporative cooler saturated ($\phi = 100$ percent), and the exit state of the air in this case is the state where the $T_{\text{wb}} = 66.0$°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

$$T_{\min} = T_{2'} = \mathbf{66.0°F}$$

Discussion Note that the temperature of air drops by as much as 30°F in this case by evaporative cooling.



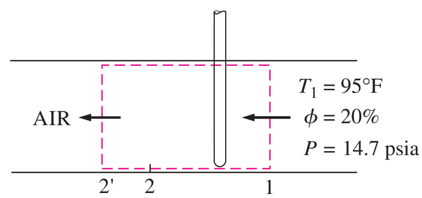


FIGURE 14–28

Schematic and psychrometric chart for Example 14–7.

Adiabatic Mixing of Airstreams

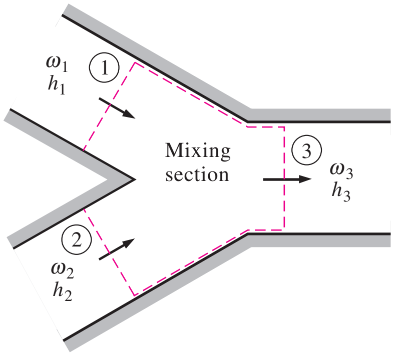
Many air-conditioning applications require the mixing of two airstreams. This is particularly true for large buildings, most production and process plants, and hospitals, which require that the conditioned air be mixed with a certain fraction of fresh outside air before it is routed into the living space. The mixing is accomplished by simply merging the two airstreams, as shown in Fig. 14–29.

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: $$\dot{m}_{a_1} + \dot{m}_{a_2} = \dot{m}_{a_3} \quad \textbf{(14–21)}$$

Mass of water vapor: $$\omega_1 \dot{m}_{a_1} + \omega_2 \dot{m}_{a_2} = \omega_3 \dot{m}_{a_3} \quad \textbf{(14–22)}$$

Energy: $$\dot{m}_{a_1} h_1 + \dot{m}_{a_2} h_2 = \dot{m}_{a_3} h_3 \quad \textbf{(14–23)}$$



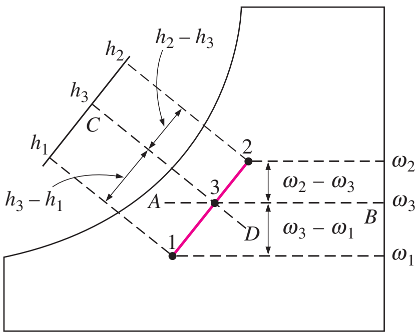


FIGURE 14–29

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.

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

$$\frac{\dot{m}_{a_1}}{\dot{m}_{a_2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1} \qquad \textbf{(14–24)}$$

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) lies 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 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 14–8 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 m^3/min is mixed adiabatically with the outside air at 32°C and 60 percent relative humidity at a rate of 20 m^3/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 Conditioned air is mixed with outside air at specified rates. The specific and relative humidities, dry-bulb temperature, and the flow rate of the mixture are to be determined.

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.

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

$$h_1 = 39.4 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.010 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.826 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 79.0 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0182 \text{ kg H}_2\text{O/kg dry air}$$
$$v_2 = 0.889 \text{ m}^3\text{/kg dry air}$$

Analysis 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. 14–30. We note that this is a steady-flow mixing process.

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. 14–24,

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

$$\frac{60.5}{22.5} = \frac{0.0182 - \omega_3}{\omega_3 - 0.010} = \frac{79.0 - h_3}{h_3 - 39.4}$$

which yield

$$\omega_3 = \mathbf{0.0122 \text{ kg } H_2O/\text{kg dry air}}$$

$$h_3 = 50.1 \text{ kJ/kg dry air}$$

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.

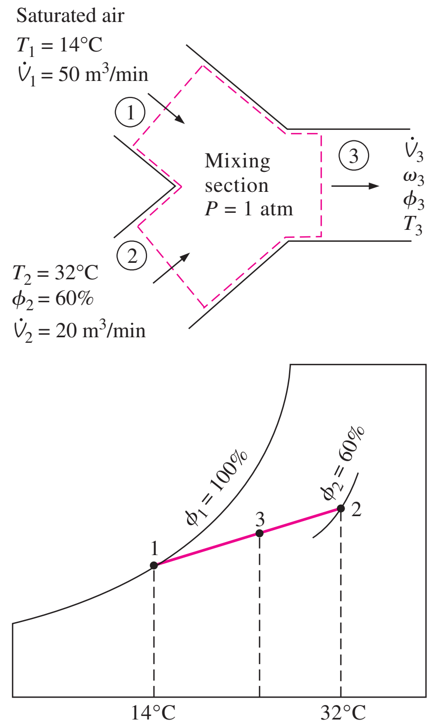


FIGURE 14–30

Schematic and psychrometric chart for Example 14–8.

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 cooling 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

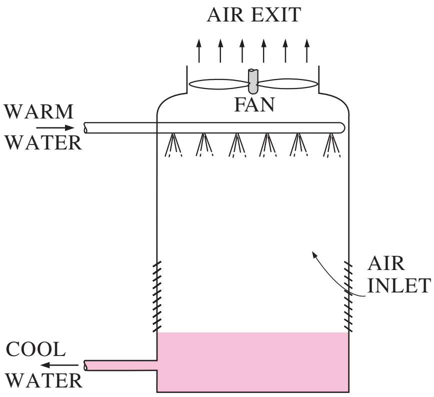


FIGURE 14–31

An induced-draft counterflow cooling tower.

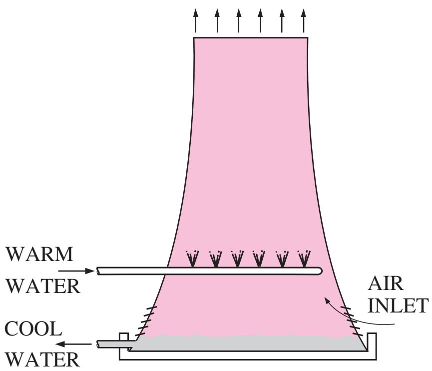


FIGURE 14–32

A natural-draft cooling tower.

FIGURE 14–33

A spray pond.

Photo by Yunus Çengel.

Fig. 14–31. 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 absorb 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 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. 14–32, and some are over 100 m high. The hyperbolic profile is for greater structural strength, not for any thermodynamic reason.

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. 14–33. 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.

EXAMPLE 14–9 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. 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 Warm cooling water from a power plant is cooled in a wet cooling tower. The flow rates of makeup water and air are to be determined.

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.

Properties The enthalpy of saturated liquid water is 92.28 kJ/kg at 22°C and 146.64 kJ/kg at 35°C (Table A–4). From the psychrometric chart,

$$h_1 = 42.2 \text{ kJ/kg dry air} \qquad h_2 = 100.0 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.0087 \text{ kg H}_2\text{O/kg dry air} \qquad \omega_2 = 0.0273 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.842 \text{ m}^3\text{/kg dry air}$$

Analysis We take the entire *cooling tower* to be the system, which is shown schematically in Fig. 14–34. 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.

(*a*) Applying the mass and energy balances on the cooling tower gives

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

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

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

Energy balance: $$\sum_{\text{in}} \dot{m}h = \sum_{\text{out}} \dot{m}h \rightarrow \dot{m}_{a_1}h_1 + \dot{m}_3 h_3 = \dot{m}_{a_2}h_2 + \dot{m}_4 h_4$$

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

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

Substituting,

$$\dot{m}_a = \frac{(100 \text{ kg/s})[(146.64 - 92.28) \text{ kJ/kg}]}{[(100.0 - 42.2) \text{ kJ/kg}] - [(0.0273 - 0.0087)(92.28) \text{ kJ/kg}]} = 96.9 \text{ kg/s}$$

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

$$\dot{V}_1 = \dot{m}_a v_1 = (96.9 \text{ kg/s})(0.842 \text{ m}^3\text{/kg}) = \mathbf{81.6 \text{ m}^3\text{/s}}$$

(*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.

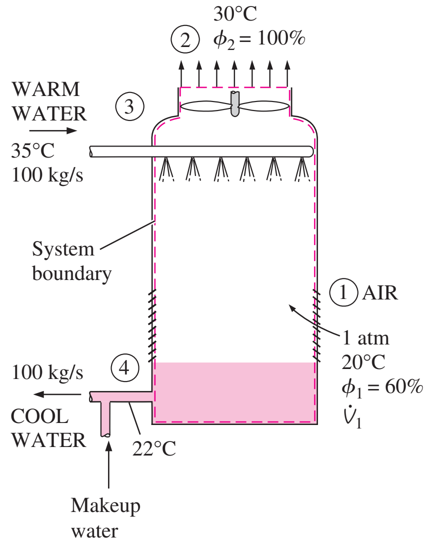


FIGURE 14–34

Schematic for Example 14–9.

SUMMARY

In this chapter we discussed the air–water-vapor mixture, which is the most commonly encountered gas–vapor mixture in practice. 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

$$\Delta h_{\text{dry air}} = c_p \Delta T = (1.005 \text{ kJ/kg} \cdot \text{°C}) \Delta T$$

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,

$$P = P_a + P_v$$

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:

$$h_v(T, \text{low } P) \cong h_g(T) \cong 2500.9 + 1.82T \quad (\text{kJ/kg}) \quad T \text{ in °C}$$
$$\cong 1060.9 + 0.435T \quad (\text{Btu/lbm}) \quad T \text{ in °F}$$

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

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

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

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,

$$\phi = \frac{m_v}{m_g} = \frac{P_v}{P_g}$$

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

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

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.

The needs of the human body and the conditions of the environment are not quite compatible. Therefore, it often becomes necessary to change the conditions of a living space to make it more comfortable. Maintaining a living space or an industrial facility at the desired temperature and humidity may require simple heating (raising the temperature), simple cooling (lowering the temperature), humidifying (adding moisture), or dehumidifying (removing moisture). Sometimes two or more of these processes are needed to bring the air to the desired temperature and humidity level.

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: $$\sum_{\text{in}} \dot{m}_a = \sum_{\text{out}} \dot{m}_a$$

Water mass: $$\sum_{\text{in}} \dot{m}_w = \sum_{\text{out}} \dot{m}_w \quad \text{or} \quad \sum_{\text{in}} \dot{m}_a\omega = \sum_{\text{out}} \dot{m}_a\omega$$

Energy: $$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} + \sum_{\text{in}} \dot{m}h = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \sum_{\text{out}} \dot{m}h$$

The changes in kinetic and potential energies are assumed to be negligible.

During a simple heating or cooling process, the specific humidity remains constant, but the temperature and the relative humidity change. Sometimes air is humidified after it is heated, and some cooling processes include dehumidification. In dry climates, air can be cooled via evaporative cooling by passing it through a section where it is sprayed with water. In locations with limited cooling water supply, large amounts of waste heat can be rejected to the atmosphere with minimum water loss through the use of cooling towers.

REFERENCES AND SUGGESTED READINGS

1. ASHRAE. *1981 Handbook of Fundamentals*. Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1981.
2. S. M. Elonka. "Cooling Towers." *Power*, March 1963.
3. W. F. Stoecker and J. W. Jones. *Refrigeration and Air Conditioning*. 2nd ed. New York: McGraw-Hill, 1982.
4. K. Wark and D. E. Richards. *Thermodynamics*. 6th ed. New York: McGraw-Hill, 1999.
5. L. D. Winiarski and B. A. Tichenor. "Model of Natural Draft Cooling Tower Performance." *Journal of the Sanitary Engineering Division, Proceedings of the American Society of Civil Engineers*, August 1970.

PROBLEMS*

Dry and Atmospheric Air: Specific and Relative Humidity

14–1C Is it possible to obtain saturated air from unsaturated air without adding any moisture? Explain.

14–2C Is the relative humidity of saturated air necessarily 100 percent?

14–3C 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?

14–4C What is the difference between dry air and atmospheric air?

14–5C Can the water vapor in air be treated as an ideal gas? Explain.

14–6C What is vapor pressure?

14–7C 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?

14–8C What is the difference between the specific humidity and the relative humidity?

14–9C 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?

14–10C 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?

14–11C 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.

14–12C Why are the chilled water lines always wrapped with vapor barrier jackets?

14–13C Explain how vapor pressure of the ambient air is determined when the temperature, total pressure, and the relative humidity of air are given.

14–14 An 8 m^3-tank contains saturated air at 30°C, 105 kPa. Determine (*a*) the mass of dry air, (*b*) the specific humidity, and (*c*) the enthalpy of the air per unit mass of the dry air.

14–15 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.

14–16 Repeat Prob. 14–15 for a temperature of 24°C.

14–17 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.

14–18 Repeat Prob. 14–17 for a pressure of 85 kPa.

14–19E 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

14–20 Determine the masses of dry air and the water vapor contained in a 240-m^3 room at 98 kPa, 23°C, and 50 percent relative humidity. *Answers:* 273 kg, 2.5 kg

*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 DVD. 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.

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

14–21C What is the dew-point temperature?

14–22C 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.

14–23C In summer, the outer surface of a glass filled with iced water frequently "sweats." How can you explain this sweating?

14–24C 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.

14–25C When are the dry-bulb and dew-point temperatures identical?

14–26C When are the adiabatic saturation and wet-bulb temperatures equivalent for atmospheric air?

14–27 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?

14–28 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.

14–29 Repeat Prob. 14–28 for a relative humidity of 30 percent.

14–30E 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?

14–31 The dry- and wet-bulb temperatures of atmospheric air at 95 kPa are 25 and 17°C, respectively. Determine (*a*) the specific humidity, (*b*) the relative humidity, and (*c*) the enthalpy of the air, in kJ/kg dry air.

14–32 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*) 0.0090 kg H_2O/kg dry air, (*b*) 54.1 percent, (*c*) 12.3°C

14–33 Reconsider Prob. 14–32. Determine the required properties using EES (or other) software. What would the property values be at a pressure of 300 kPa?

14–34E The air in a room has a dry-bulb temperature of 80°F and a wet-bulb temperature of 65°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*) 0.0097 lbm H_2O/lbm dry air, (*b*) 44.7 percent, (*c*) 56.6°F

14–35 Atmospheric air at 35°C flows steadily into an adiabatic saturation device and leaves as a saturated mixture at 25°C. Makeup water is supplied to the device at 25°C. Atmospheric pressure is 98 kPa. Determine the relative humidity and specific humidity of the air.

Psychrometric Chart

14–36C How do constant-enthalpy and constant-wet-bulb-temperature lines compare on the psychrometric chart?

14–37C At what states on the psychrometric chart are the dry-bulb, wet-bulb, and dew-point temperatures identical?

14–38C How is the dew-point temperature at a specified state determined on the psychrometric chart?

14–39C Can the enthalpy values determined from a psychrometric chart at sea level be used at higher elevations?

14–40 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 m^3/kg dry air). Use the psychrometric chart or available software.

14–41 Reconsider Prob. 14–40. Determine the required properties using EES (or other) software instead of the psychrometric chart. What would the property values be at a location at 1500 m altitude?

14–42 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 m^3/kg dry air).

14–43 Reconsider Prob. 14–42. Determine the required properties using EES (or other) software instead of the psychrometric chart. What would the property values be at a location at 2000 m altitude?

14–44E 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 ft^3/lbm dry air).

14–45E Reconsider Prob. 14–44E. Determine the required properties using EES (or other) software instead of the psychrometric chart. What would the property values be at a location at 5000 ft altitude?

14–46 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 m^3/kg dry air).

14–47 Reconsider Prob. 14–46. Determine the required properties using EES (or other) software instead of the psychrometric chart. What would the property values be at a location at 3000 m altitude?

Human Comfort and Air-Conditioning

14–48C What does a modern air-conditioning system do besides heating or cooling the air?

14–49C How does the human body respond to (*a*) hot weather, (*b*) cold weather, and (*c*) hot and humid weather?

14–50C What is the radiation effect? How does it affect human comfort?

14–51C How does the air motion in the vicinity of the human body affect human comfort?

14–52C Consider a tennis match in cold weather where both players and spectators wear the same clothes. Which group of people will feel colder? Why?

14–53C Why do you think little babies are more susceptible to cold?

14–54C How does humidity affect human comfort?

14–55C What are humidification and dehumidification?

14–56C 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?

14–57C 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?

14–58C 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?

14–59C 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?

14–60 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.

14–61 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.

14–62 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.

14–63E In a movie theater in winter, 500 people, each generating sensible heat at a rate of 70 W, are watching a movie. The heat losses through the walls, windows, and the roof are estimated to be 130,000 Btu/h. Determine if the theater needs to be heated or cooled.

14–64 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.

14–65 Repeat Prob. 14–64 for an infiltration rate of 1.8 ACH.

Simple Heating and Cooling

14–66C How do relative and specific humidities change during a simple heating process? Answer the same question for a simple cooling process.

14–67C Why does a simple heating or cooling process appear as a horizontal line on the psychrometric chart?

14–68 Air enters a heating section at 95 kPa, 12°C, and 30 percent relative humidity at a rate of 6 m^3/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*) 91.1 kJ/min, (*b*) 13.3 percent

14–69E 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*) 56.6°F, (*b*) 31.4 percent, (*c*) 25.4 ft/s

14–70 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

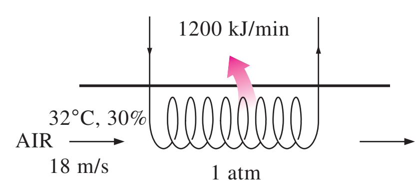


FIGURE P14–70

14–71 Repeat Prob. 14–70 for a heat removal rate of 800 kJ/min.

Heating with Humidification

14–72C Why is heated air sometimes humidified?

14–73 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

14–74E Air at 14.7 psia, 50°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 55 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.

14–75 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 m^3/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.

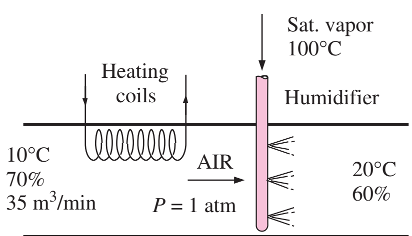


FIGURE P14–75

14–76 Repeat Prob. 14–75 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

14–77C Why is cooled air sometimes reheated in summer before it is discharged to a room?

14–78 Air enters a window air conditioner at 1 atm, 32°C, and 70 percent relative humidity at a rate of 2 m^3/min, and it leaves as saturated air at 15°C. Part of the moisture in the air that condenses during the process is also removed at 15°C. Determine the rates of heat and moisture removal from the air. *Answers:* 97.7 kJ/min, 0.023 kg/min

14–79 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.

14–80 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 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.

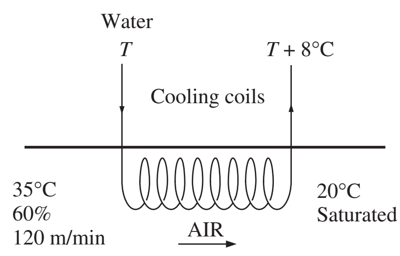


FIGURE P14–80

14–81 Reconsider Prob. 14–80. 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.

14–82 Repeat Prob. 14–80 for a total pressure of 95 kPa for air. *Answers:* (*a*) 293.2 kJ/min, (*b*) 8.77 kg/min, (*c*) 113 m/min

14–83E 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.

14–84E Reconsider Prob. 14–83E. 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.

14–85E Repeat Prob. 14–83E for a total pressure of 14.4 psia for air.

14–86 Atmospheric air from the inside of an automobile enters the evaporator section of the air conditioner at 1 atm, 27°C and 50 percent relative humidity. The air returns to the automobile at 10°C and 90 percent relative humidity. The passenger compartment has a volume of 2 m^3 and 5 air changes per minute are required to maintain the inside of the automobile at the desired comfort level. Sketch the psychrometric diagram for the atmospheric air flowing through the air conditioning process. Determine the dew point and wet bulb temperatures at the inlet to the evaporator section, in °C. Determine the required heat transfer rate from the atmospheric air to the evaporator fluid, in kW. Determine the rate of condensation of water vapor in the evaporator section, in kg/min.

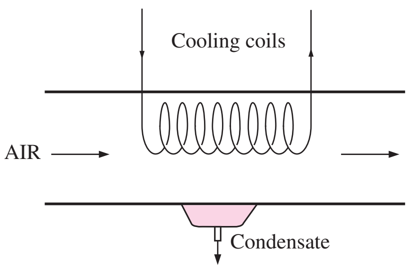


FIGURE P14–86

14–87 Two thousand cubic meters per hour of atmospheric air at 28°C with a dew point temperature of 25°C flows into an air conditioner that uses chilled water as the cooling fluid. The atmospheric air is to be cooled to 18°C. Sketch the system hardware and the psychrometric diagram for the process. Determine the mass flow rate of the condensate water, if any, leaving the air conditioner, in kg/h. If the cooling water has a 10°C temperature rise while flowing through the air conditioner, determine the volume flow rate of chilled water supplied to the air conditioner heat exchanger, in m^3/min. The air conditioning process takes place at 100 kPa.

14–88 An automobile air conditioner uses refrigerant-134a as the cooling fluid. The evaporator operates at 275 kPa gauge and the condenser operates at 1.7 MPa gage. The compressor requires a power input of 6 kW and has an isentropic efficiency of 85 percent. Atmospheric air at 22°C and 50 percent relative humidity enters the evaporator and leaves at 8°C and 90 percent relative humidity. Determine the volume flow rate of the atmospheric air entering the evaporator of the air conditioner, in m^3/min.

14–89 Air from a workspace enters an air conditioner unit at 30°C dry bulb and 25°C wet bulb. The air leaves the air conditioner and returns to the space at 25°C dry-bulb and 6.5°C dew-point temperature. If there is any, the condensate leaves the air conditioner at the temperature of the air leaving the cooling coils. The volume flow rate of the air returned to the workspace is 1000 m^3/min. Atmospheric pressure is 98 kPa. Determine the heat transfer rate from the air, in kW, and the mass flow rate of condensate water, if any, in kg/h.

Evaporative Cooling

14–90C Does an evaporation process have to involve heat transfer? Describe a process that involves both heat and mass transfer.

14–91C 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?

14–92C What is evaporative cooling? Will it work in humid climates?

14–93 Air enters an evaporative cooler at 1 atm, 36°C, and 20 percent relative humidity at a rate of 4 m^3/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.

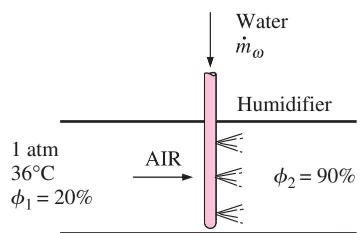


FIGURE P14–93

14–94E Air enters an evaporative cooler at 14.7 psia, 90°F, and 20 percent relative humidity at a rate of 150 ft^3/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*) 65°F, (*b*) 0.06 lbm/min

14–95 Air enters an evaporative cooler at 95 kPa, 40°C, and 25 percent relative humidity and exits saturated. Determine the exit temperature of air. *Answer:* 23.1°C

14–96E 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.

14–97 Air enters an evaporative cooler at 1 atm, 32°C, and 30 percent relative humidity at a rate of 5 m^3/min and leaves at 22°C. Determine (*a*) the final relative humidity and (*b*) the amount of water added to air.

14–98 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

14–99 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

14–100C 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?

14–101C 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?

14–102 Two airstreams are mixed steadily and adiabatically. The first stream enters at 32°C and 40 percent relative humidity at a rate of 20 m^3/min, while the second stream enters at 12°C and 90 percent relative humidity at a rate of 25 m^3/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 m^3/min

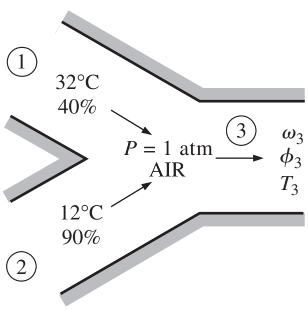


FIGURE P14–102

14–103 Repeat Prob. 14–102 for a total mixing-chamber pressure of 90 kPa.

14–104E During an air-conditioning process, 900 ft^3/min of conditioned air at 65°F and 30 percent relative humidity is mixed adiabatically with 300 ft^3/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*) 68.7°F, (*b*) 0.0078 lbm H_2O/lbm dry air, (*c*) 52.1 percent

14–105E Reconsider Prob. 14–104E. 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.

14–106 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.

14–107 Reconsider Prob. 14–106. Using EES (or other) software, determine the effect of the mass flow rate of saturated cool air stream 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

14–108C How does a natural-draft wet cooling tower work?

14–109C What is a spray pond? How does its performance compare to the performance of a wet cooling tower?

14–110 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.

14–111E 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 ft^3/s, (*b*) 2.42 lbm/s

14–112 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 m^3/s, (*b*) 1.16 kg/s

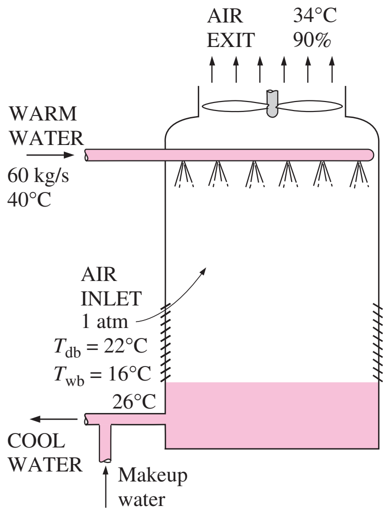


FIGURE P14–112

14–113 A wet cooling tower is to cool 25 kg/s of cooling water from 40 to 30°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*) 11.2 m^3/s, (*b*) 0.35 kg/s

14–114 A natural-draft cooling tower is to remove waste heat from the cooling water flowing through the condenser of a steam power plant. The turbine in the steam power plant receives 42 kg/s of steam from the steam generator. Eighteen percent of the steam entering the turbine is extracted for various feedwater heaters. The condensate of the higher pressure feedwater heaters is trapped to the next lowest pressure feedwater heater. The last feedwater heater operates at 0.2 MPa and all of the steam extracted for the feedwater heaters is throttled from the last feedwater heater exit to the condenser operating at a pressure of 10 kPa. The remainder of the steam produces work in the turbine and leaves the lowest pressure stage of the turbine at 10 kPa with an entropy of 7.962 kJ/kg · K. The cooling tower supplies the cooling water at 26°C to the condenser, and cooling water returns from the condenser to the cooling tower at 40°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.

Review Problems

14–115 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 20°C and 50 percent relative humidity.

14–116E The relative humidity of air at 80°F and 14.7 psia is increased from 25 to 75 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.

14–117 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?

14–118E 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.

14–119 Determine the mole fraction of dry air at the surface of a lake whose temperature is 18°C. The air at the lake surface is saturated, and the atmospheric pressure at lake level can be taken to be 100 kPa.

14–120E 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.

14–121E 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.

14–122 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?

14–123E 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.

14–124E 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.

14–125 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.

14–126 A 3-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*) 3.29 kg, (*b*) 0.0210 kg H_2O/kg dry air, (*c*) 78.6 kJ/kg dry air

14–127 Reconsider Prob. 14–126. Using EES (or other) software, determine the properties of the air at the initial state. Study the effect 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.

14–128E 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.

14–129 Air enters a cooling section at 97 kPa, 35°C, and 30 percent relative humidity at a rate of 6 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.

14–130 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.

14–131 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.

14–132 Repeat Prob. 14–131 for a total pressure of 95 kPa for air.

14–133 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 relatively 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, 22.3 percent, (*b*) 696 kJ/min, (*c*) 0.13 kg/min

14–134 Reconsider Prob. 14–133. 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.

14–135 Repeat Prob. 14–133 for a total pressure of 96 kPa.

14–136 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.

14–137 Reconsider Prob. 14–136. 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?

14–138 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.

14–139 Reconsider Prob. 14–138. 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.

14–140 Atmospheric air enters an air-conditioning system at 30°C and 70 percent relative humidity with a volume flow rate of 4 m^3/min and is cooled to 20°C and 20 percent relative humidity at a pressure of 1 atm. The system uses refrigerant-134a as the cooling fluid that enters the cooling section at

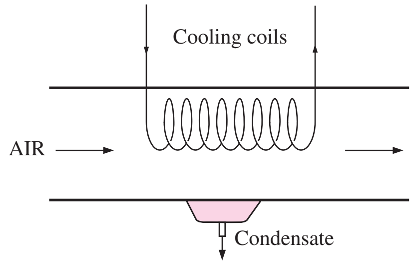


FIGURE P14–140

350 kPa with a quality of 20 percent and leaves as a saturated vapor. Draw a schematic and show the process on the psychrometric chart. What is the heat transfer from the air to the cooling coils, in kW? If any water is condensed from the air, how much water will be condensed from the atmospheric air per min? Determine the mass flow rate of the refrigerant, in kg/min.

14–141 An uninsulated tank having a volume of 0.5 m^3 contains air at 35°C, 130 kPa, and 20 percent relative humidity. The tank is connected to a water supply line in which water flows at 50°C. Water is sprayed into the tank until the relative humidity of the air–vapor mixture is 90 percent. Determine the amount of water supplied to the tank, in kg, the final pressure of the air–vapor mixture in the tank, in kPa, and the heat transfer required during the process to maintain the air– vapor mixture in the tank at 35°C.

14–142 Air flows steadily through an isentropic nozzle. The air enters the nozzle at 35°C, 200 kPa and 50 percent relative humidity. If no condensation is to occur during the expansion process, determine the pressure, temperature, and velocity of the air at the nozzle exit.

Fundamentals of Engineering (FE) Exam Problems

14–143 A room is filled with saturated moist air at 25°C and a total pressure of 100 kPa. If the mass of dry air in the room is 100 kg, the mass of water vapor is

(*a*) 0.52 kg (*b*) 1.97 kg (*c*) 2.96 kg
(*d*) 2.04 kg (*e*) 3.17 kg

14–144 A room contains 50 kg of dry air and 0.6 kg of water vapor at 25°C and 95 kPa total pressure. The relative humidity of air in the room is

(*a*) 1.2% (*b*) 18.4% (*c*) 56.7%
(*d*) 65.2% (*e*) 78.0%

14–145 A 40-m^3 room contains air at 30°C and a total pressure of 90 kPa with a relative humidity of 75 percent. The mass of dry air in the room is

(*a*) 24.7 kg (*b*) 29.9 kg (*c*) 39.9 kg
(*d*) 41.4 kg (*e*) 52.3 kg

14–146 A room contains air at 30°C and a total pressure of 96.0 kPa with a relative humidity of 75 percent. The partial pressure of dry air is

(*a*) 82.0 kPa (*b*) 85.8 kPa (*c*) 92.8 kPa
(*d*) 90.6 kPa (*e*) 72.0 kPa

14–147 The air in a house is at 20°C and 50 percent relative humidity. Now the air is cooled at constant pressure. The temperature at which the moisture in the air will start condensing is

(*a*) 8.7°C (*b*) 11.3°C (*c*) 13.8°C
(*d*) 9.3°C (*e*) 10.0°C

14–148 On the psychrometric chart, a cooling and dehumidification process appears as a line that is
(*a*) horizontal to the left
(*b*) vertical downward
(*c*) diagonal upwards to the right (NE direction)
(*d*) diagonal upwards to the left (NW direction)
(*e*) diagonal downwards to the left (SW direction)

14–149 On the psychrometric chart, a heating and humidification process appears as a line that is
(*a*) horizontal to the right
(*b*) vertical upward
(*c*) diagonal upwards to the right (NE direction)
(*d*) diagonal upwards to the left (NW direction)
(*e*) diagonal downwards to the right (SE direction)

14–150 An air stream at a specified temperature and relative humidity undergoes evaporative cooling by spraying water into it at about the same temperature. The lowest temperature the air stream can be cooled to is
(*a*) the dry bulb temperature at the given state
(*b*) the wet bulb temperature at the given state
(*c*) the dew point temperature at the given state
(*d*) the saturation temperature corresponding to the humidity ratio at the given state
(*e*) the triple point temperature of water

14–151 Air is cooled and dehumidified as it flows over the coils of a refrigeration system at 85 kPa from 30°C and a humidity ratio of 0.023 kg/kg dry air to 15°C and a humidity ratio of 0.015 kg/kg dry air. If the mass flow rate of dry air is 0.7 kg/s, the rate of heat removal from the air is

(*a*) 5 kJ/s (*b*) 10 kJ/s (*c*) 15 kJ/s
(*d*) 20 kJ/s (*e*) 25 kJ/s

14–152 Air at a total pressure of 90 kPa, 15°C, and 75 percent relative humidity is heated and humidified to 25°C and 75 percent relative humidity by introducing water vapor. If the mass flow rate of dry air is 4 kg/s, the rate at which steam is added to the air is

(*a*) 0.032 kg/s (*b*) 0.013 kg/s (*c*) 0.019 kg/s
(*d*) 0.0079 kg/s (*e*) 0 kg/s

Design and Essay Problems

14–153 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.

14–154 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.

14–155 Identify the major sources of heat gain in your house in summer, and propose ways of minimizing them and thus reducing the cooling load.

14–156 Write an essay on different humidity measurement devices, including electronic ones, and discuss the advantages and disadvantages of each device.

14–157 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.