CHAPTER 14

GAS-VAPOR MIXTURES & AIR-CONDITIONING

2022 Fall semester

Objectives

- 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

Atmospheric air: Air in the atmosphere containing some water vapor (or *moisture*).

Dry air: Air that contains no water vapor.

Water vapor in the air plays a major role in human comfort. Therefore, it is an important consideration in air-conditioning applications.

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

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

In air-conditioning processes we are concerned with the changes in enthalpy Δh, which is independent of the reference point selected.

14-1 DRY AND ATMOSPHERIC AIR

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.

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:

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

P_a Partial pressure of dry air

P_v Partial pressure of vapor (vapor pressure)

$h = h(T)$ since water vapor is an ideal gas

$$h_v(T, \text{low } P) \cong h_g(T)$$

For water

h_g = 2500.9 kJ/kg (at 0°C)

$c_{p,avg}$ = 1.82 kJ/kg·°C

(at –10 to 50°C range)

$$\boxed{h_g(T) \cong 2500.9 + 1.82T \quad (\text{kJ/kg}) \quad T \text{ in °C}} \quad (14\text{-}4)$$

$$h_g(T) \cong 1060.9 + 0.435T \quad (\text{Btu/lbm}) \quad T \text{ in °F}$$

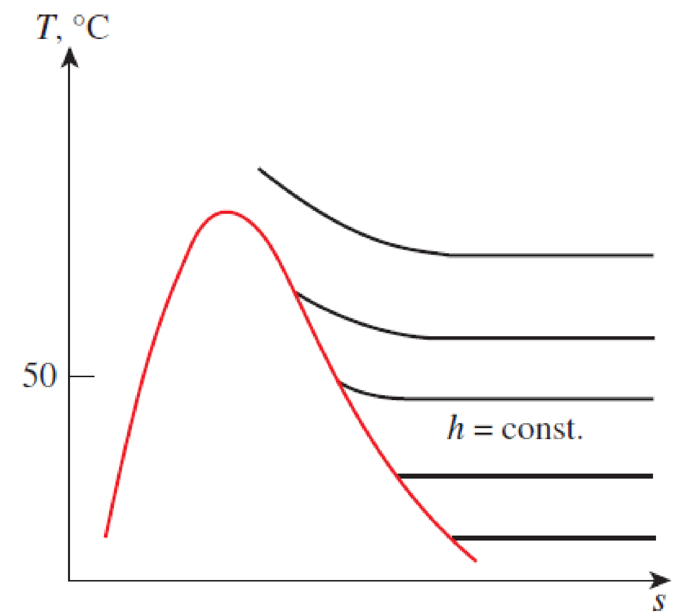


FIGURE 14–2
At temperatures below 50°C, the h = constant lines coincide with the T = constant lines in the superheated vapor region of water.

T, °C	Water vapor h_g, kJ/kg: Table A-4	Water vapor h_g, kJ/kg: 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

Absolute or **specific humidity** (*humidity ratio*): **ω**

The mass of water vapor present in a unit mass of dry air.

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

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

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

14.2 SPECIFIC AND RELATIVE HUMIDITY OF AIR

Saturated air: The air saturated with moisture.

Relative humidity: The ratio of the amount of moisture the air holds (m_v) to the maximum amount of moisture the air can hold at the same temperature (m_g).

$$\boxed{\phi = \frac{m_v}{m_g} = \frac{P_v V / R_v T}{P_g V / R_v T} = \frac{P_v}{P_g}}$$ ★(14-9)

$$P_g = P_{\text{sat @ } T}$$

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

What is the relative humidity of dry air and saturated air?

14.2 SPECIFIC AND RELATIVE HUMIDITY OF AIR

Air
25°C, 100 kPa

$(P_{sat,H_2O\ @\ 25°C} = 3.1698\ \text{kPa})$

$P_v = 0 \rightarrow$ dry air
$P_v < 3.1698\ \text{kPa} \rightarrow$ unsaturated air
$P_v = 3.1698\ \text{kPa} \rightarrow$ saturated air

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

Air
25°C, 1 atm

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

Specific humidity: $\omega = 0.01\ \dfrac{\text{kg } H_2O}{\text{kg dry air}}$
Relative humidity: $\phi = 50\%$

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.

14.2 SPECIFIC AND RELATIVE HUMIDITY OF AIR

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.*

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

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

$$h_v \cong h_g$$

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

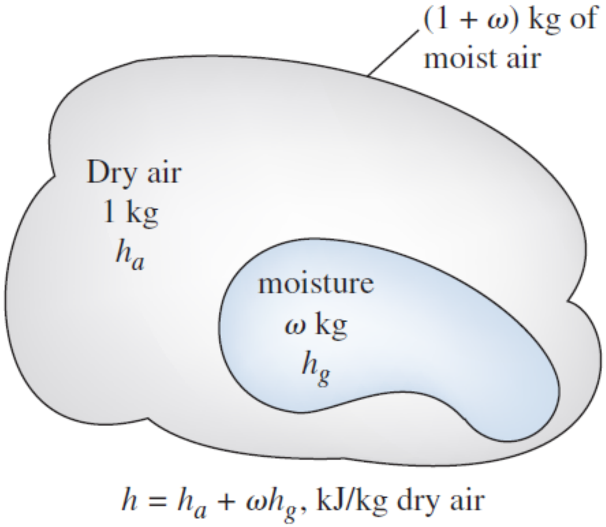


FIGURE 14–6
The enthalpy of moist (atmospheric) air is expressed per unit mass of dry air, not per unit mass of moist air.

Dry-bulb temperature:
The ordinary temperature of atmospheric air.

Dew-point temperature T_{dp}: The temperature at which condensation begins when the air is cooled at constant pressure.

T_{dp} is the saturation temperature of water corresponding to the vapor pressure.

$$\boxed{T_{dp} = T_{sat\,@\,P_v}} \qquad (14\text{-}13)$$

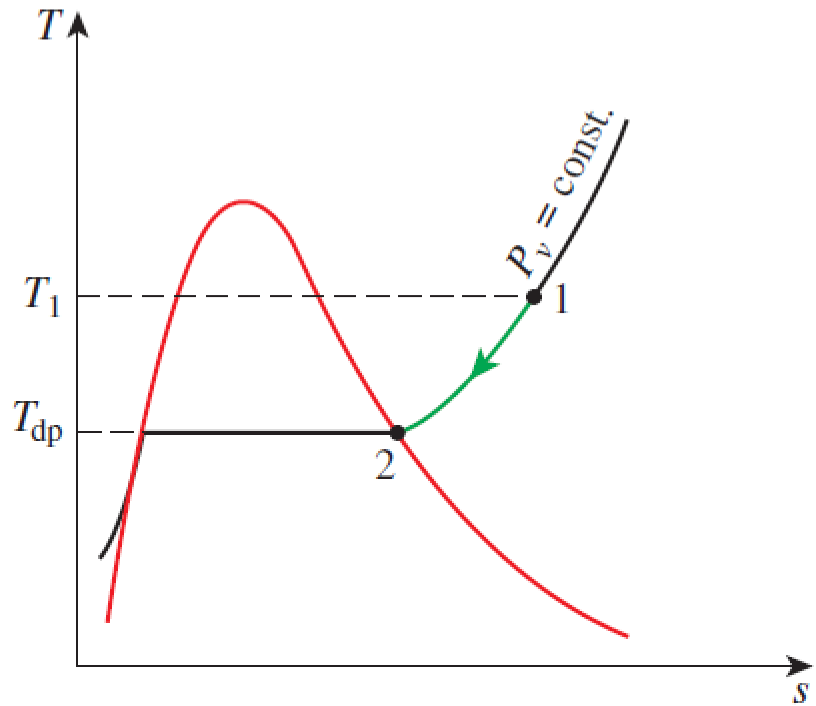


FIGURE 14–8
Constant-presssure cooling of moist air and the dew-point temperature on the *T*-*s* diagram of water.

14.3 DEW-POINT TEMPERATURE

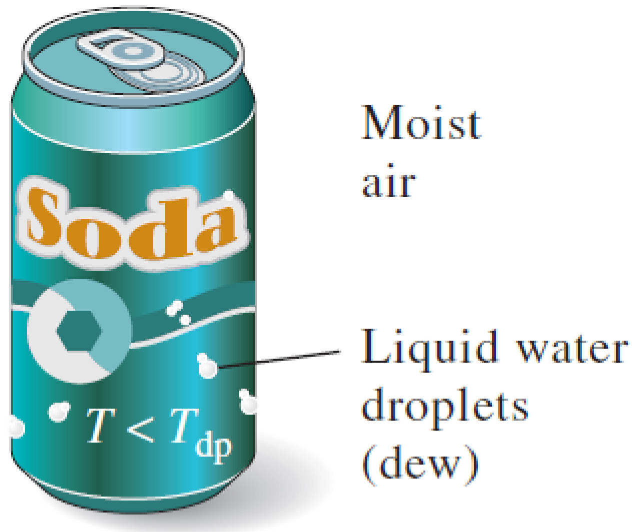
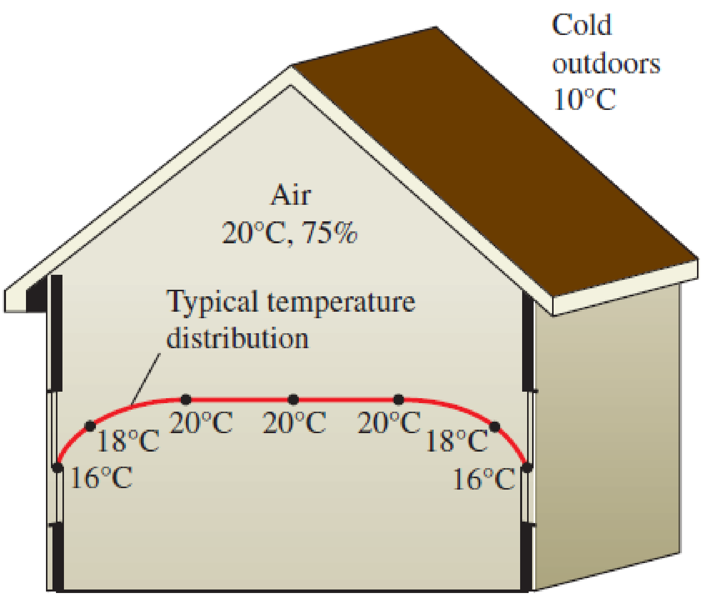


FIGURE 14–9
When the temperature of a cold drink is below the dew-point temperature of the surrounding air, it "sweats."

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?



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

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

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

14.4 ADIABATIC SATURATION AND WET-BULB TEMP.

$\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a$ (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 $\dot{m}_f$)

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

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

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

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

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

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg_2}}{h_{g_1} - h_{f_2}} \qquad \omega_2 = \frac{0.622 P_{g_2}}{P_2 - P_{g_2}}$$

The specific humidity (and relative humidity) of air can be determined from these equations by measuring the pressure and temperature of air at the inlet and the exit of an adiabatic saturator.

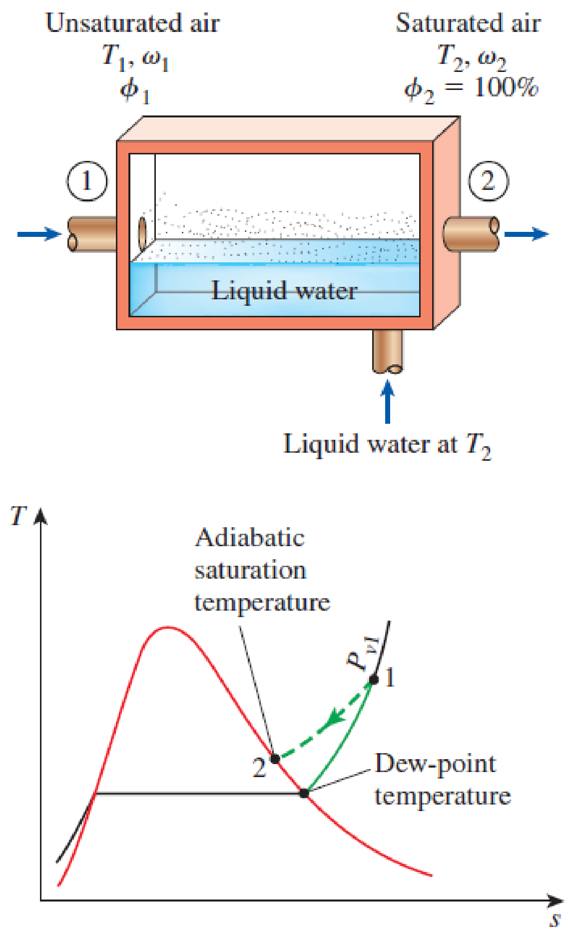


FIGURE 14–11
The adiabatic saturation process and its representation on a *T-s* diagram of water.

The adiabatic saturation process is not practical. To determine the absolute and relative humidity of air, 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.

The temperature measured is the **wet-bulb temperature T_{wb}** and it is commonly used in A-C applications.

For air–water vapor mixtures at atmospheric pressure, wet-bulb temperature T_{wb} is approximately equal to the adiabatic saturation temperature.

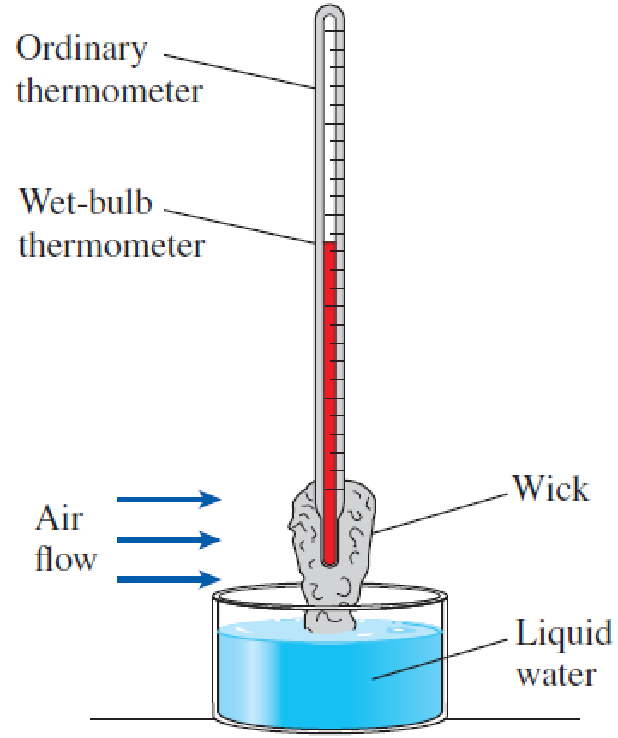


FIGURE 14–12
A simple arrangement to measure the wet-bulb temperature.

14.4 ADIABATIC SATURATION AND WET-BULB TEMP.

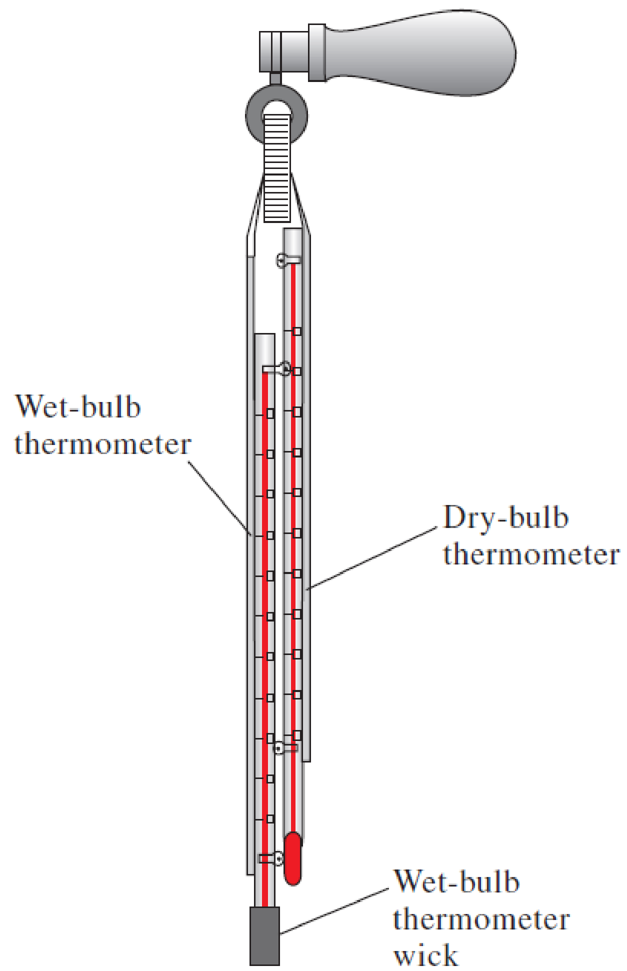


FIGURE 14–13
Sling psychrometer.

Psychrometric charts: Present moist air properties in a convenient form. They are used extensively in A-C applications. The psychrometric chart serves as a valuable aid in visualizing the A-C processes such as heating, cooling, and humidification.

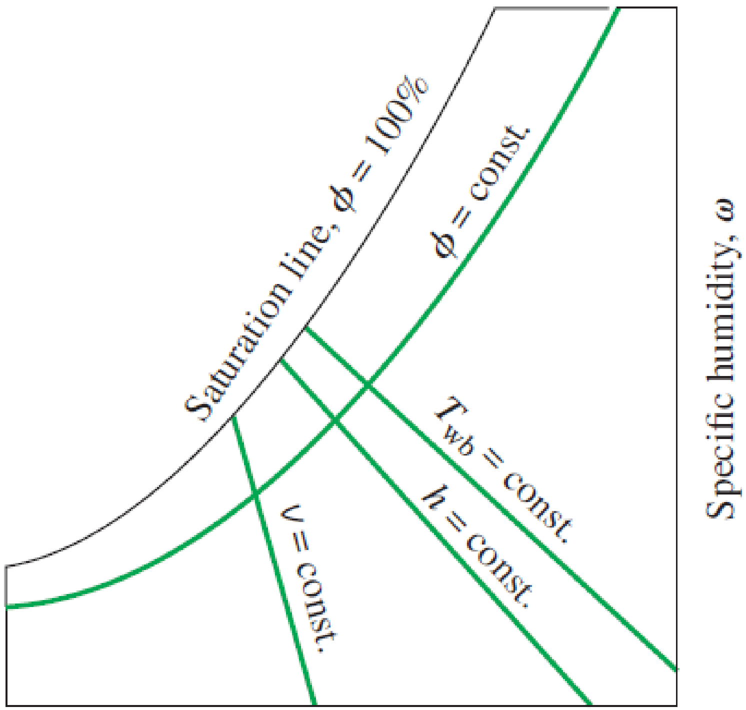


FIGURE 14–14
Schematic for a psychrometric chart.

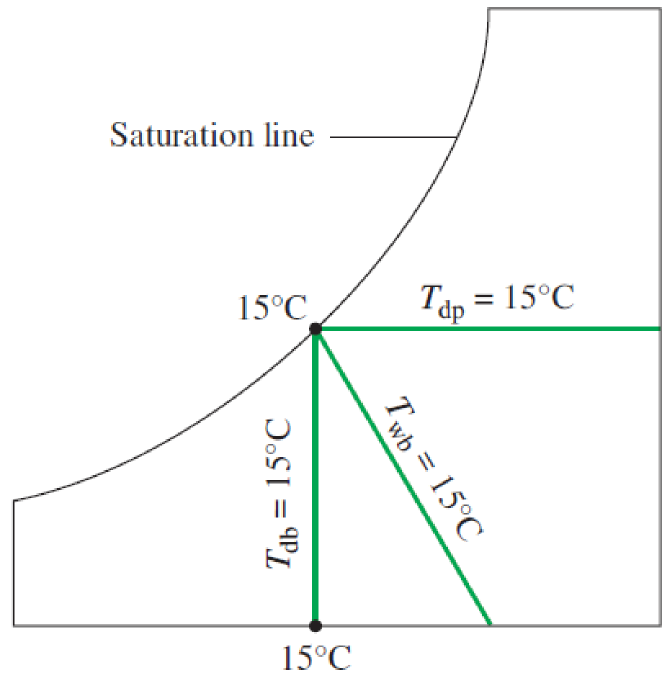


FIGURE 14–15
For saturated air, the dry-bulb, wet-bulb, and dew-point temperatures are identical.

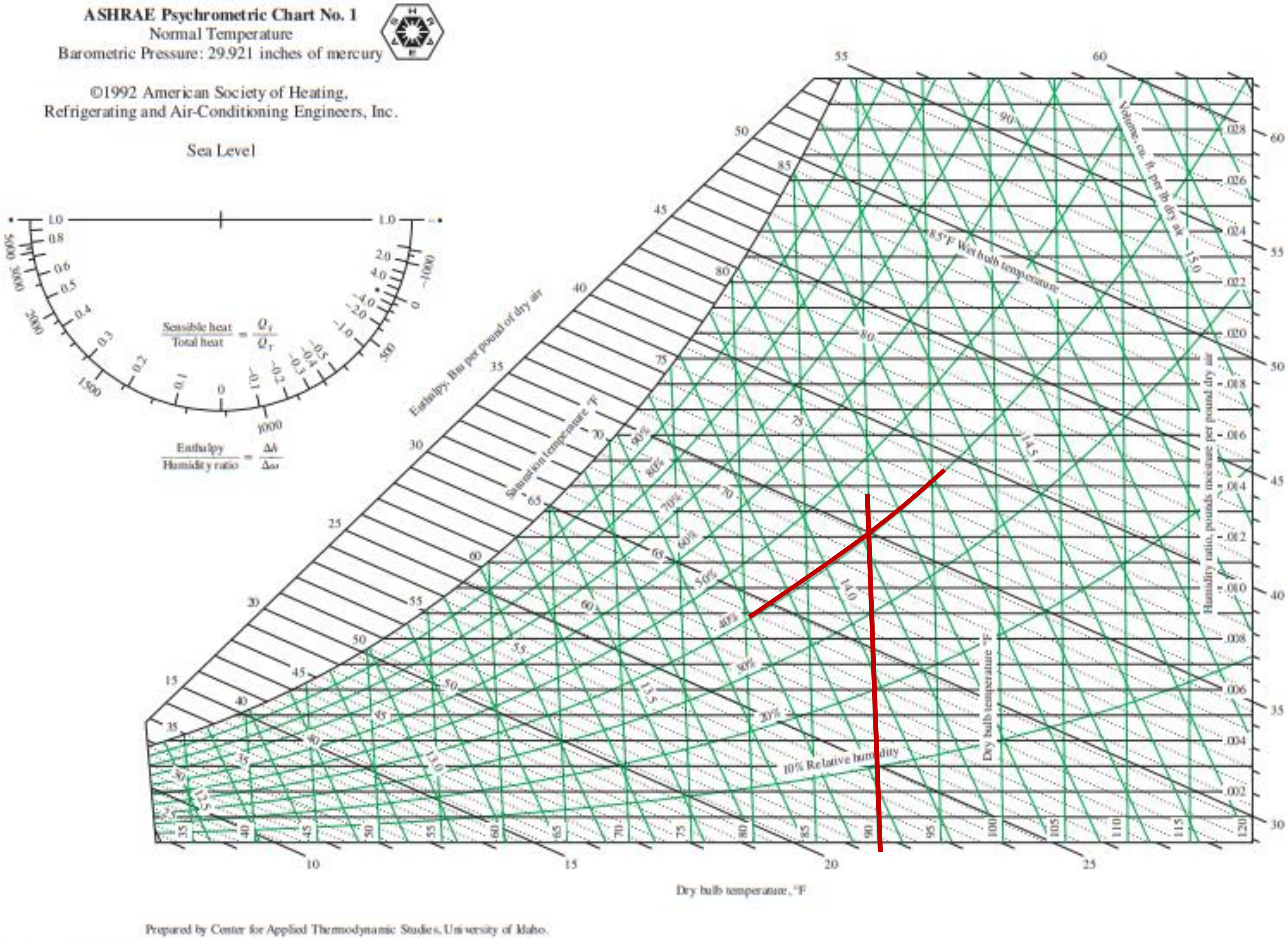


FIGURE A–31E

Psychrometric chart at 1 atm total pressure.

From the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA.

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.

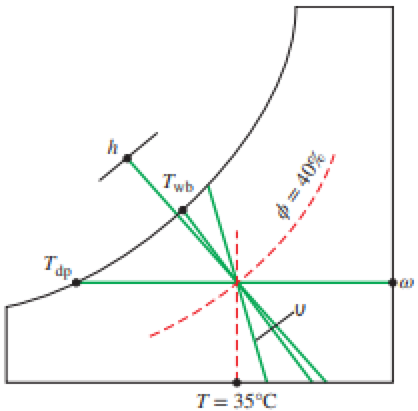


FIGURE 14–16
Schematic for Example 14–4.

14.6 HUMAN COMFORT AND AIR-CONDITIONING

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.

The rate of heat generation by human body 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, and 440 W when doing heavy physical work.

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*.

In an environment at 10°C with 48 km/h winds feels as cold as an environment at -7°C with 3 km/h winds as a result of the body-chilling effect of the air motion (the *wind-chill factor*).

FIGURE 14–17
We cannot change the weather, but we can change the climate in a confined space by air-conditioning.

14.6 HUMAN COMFORT AND AIR-CONDITIONING

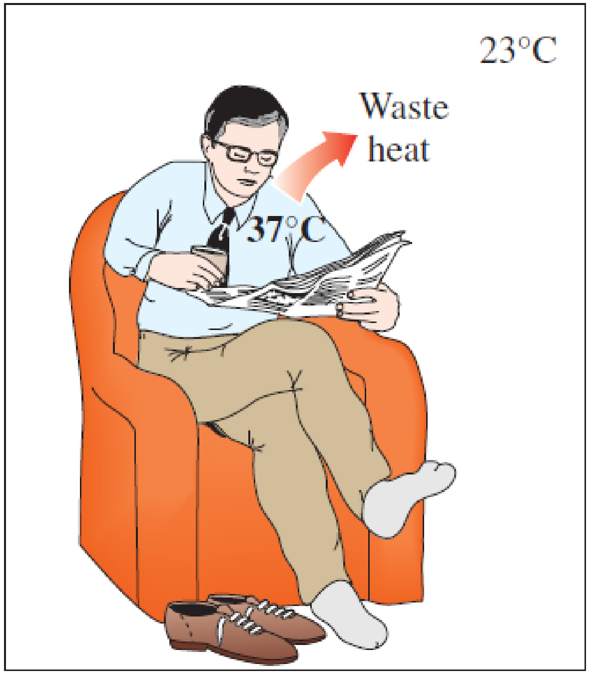


FIGURE 14–18
A body feels comfortable when it can freely dissipate its waste heat, and no more.

The comfort of the human body depends primarily on three factors: the **(dry-bulb) temperature, relative humidity,** and **air motion**.

The relative humidity affects the amount of heat a body can dissipate through evaporation. Most people prefer a relative humidity of 40 to 60%.

Air motion removes the warm, moist air that builds up around the body and replaces it with fresh air. Air motion should be strong enough to remove heat and moisture from the vicinity of the body, but gentle enough to be unnoticed.

An important factor that affects human comfort is heat transfer by radiation between the body and the surrounding surfaces such as walls and windows.

Other factors that affect comfort are air cleanliness, odor, and noise.

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.

Air is commonly heated and humidified in winter and cooled and dehumidified in summer.

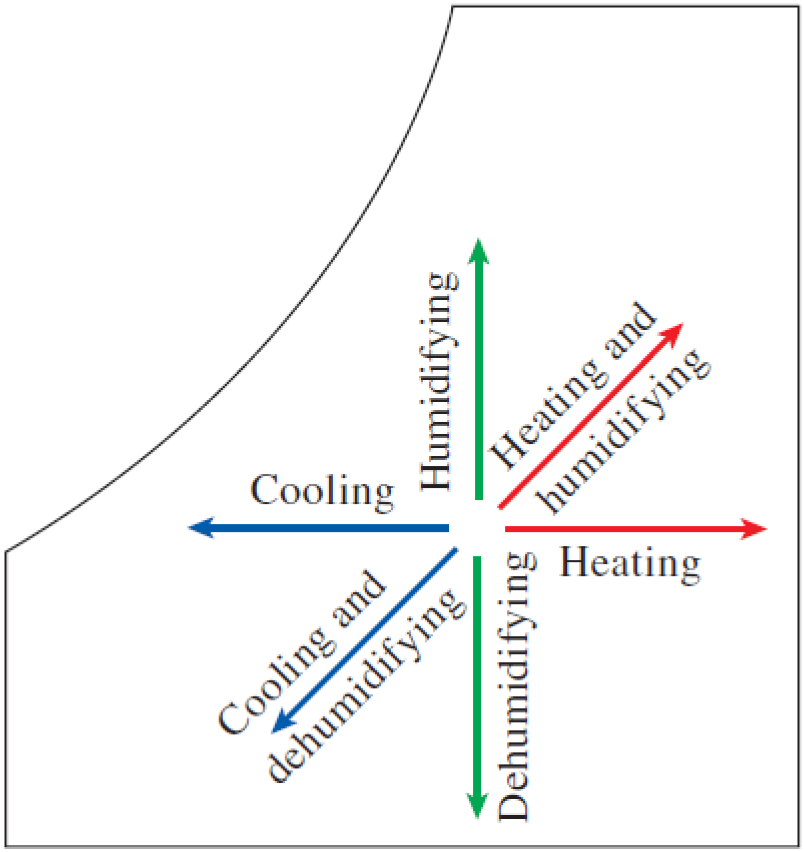


FIGURE 14–19
Various air-conditioning processes.

14.7 AIR-CONDITIONING PROCESSES

Most air-conditioning processes can be modeled as steady-flow processes with the following general mass and energy balances:

- **Mass balance :** $\dot{m}_{\text{in}} = \dot{m}_{\text{out}}$

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

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

- **Energy balance :** $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$

$$\rightarrow \boxed{\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} + \sum_{\text{in}} \dot{m}h = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \sum_{\text{out}} \dot{m}h} \qquad (14\text{-}18)$$

The work term usually consists of the *fan work input*, which is small relative to the other terms in the energy balance relation.

14.7 AIR-CONDITIONING PROCESSES

1) Simple Heating and Cooling (ω = constant)

Many residential heating systems consist of a stove, a heat pump, or an electric resistance heater. The air in these systems is heated by circulating it through a duct that contains the tubing for the hot gases or the electric resistance wires.

Cooling can be accomplished by passing the air over some coils through which a refrigerant or chilled water flows.

Heating and cooling appear as a horizontal line since no moisture is added to or removed from the air.

1) Simple Heating and Cooling (ω = constant)

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

Water mass balance : $\omega_1 = \omega_2$

Energy balance :

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

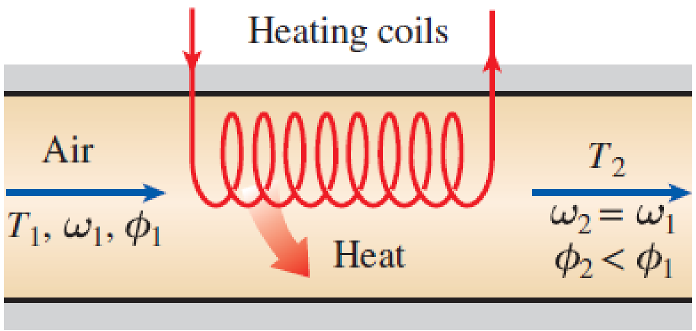


FIGURE 14–20
During simple heating, specific humidity remains constant, but relative humidity decreases.

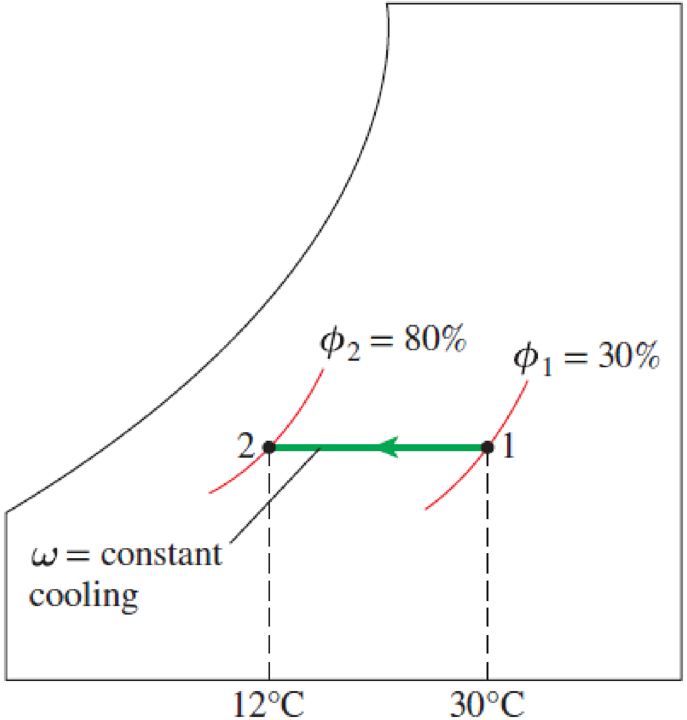


FIGURE 14–21
During simple cooling, specific humidity remains constant, but relative humidity increases.

EXAMPLE 14–5 **Cooling of Air**

Humid air at 1 atm, 100°F, and 70 percent relative humidity is cooled at constant pressure to the dew-point temperature (Fig. 14–22). Determine the cooling, in Btu/lbm dry air, required for this process.

SOLUTION Humid air at a specified state is cooled at constant pressure to the dew-point temperature. The cooling required for this process is 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 ($\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$). **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the cooling section since the process involves no humidification or dehumidification. The inlet and exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Fig. A–31E) to be

$$h_1 = 56.7 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0296 \text{ lbm H}_2\text{O/lbm dry air } (= \omega_2)$$
$$T_{\text{dp},1} = 88.4°\text{F}$$

The exit state enthalpy is

$$\left.\begin{aligned} P &= 1 \text{ atm} \\ T_2 &= T_{\text{dp},1} = 88.4°\text{F} \\ \phi_2 &= 1 \end{aligned}\right\} h_2 = 53.8 \text{ Btu/lbm dry air}$$

From the energy balance on air in the cooling section,

$$q_{\text{out}} = h_1 - h_2 = 56.7 - 53.8 = \mathbf{2.9 \text{ Btu/lbm dry air}}$$

Discussion Air is cooled by 11.6°C during this process. The specific humidity remains constant during a simple cooling process and is represented by a horizontal line in the psychrometric chart.

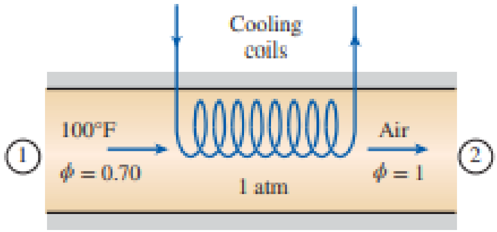


FIGURE 14–22
Schematic for Example 14–5.

2) Heating with Humidification

Problems 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 and then through a humidifying section.

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

$\dot{m}_{a_2}\omega_2 + \dot{m}_w = \dot{m}_{a_3}\omega_3 \rightarrow \dot{m}_w = \dot{m}_a(\omega_3 - \omega_2)$

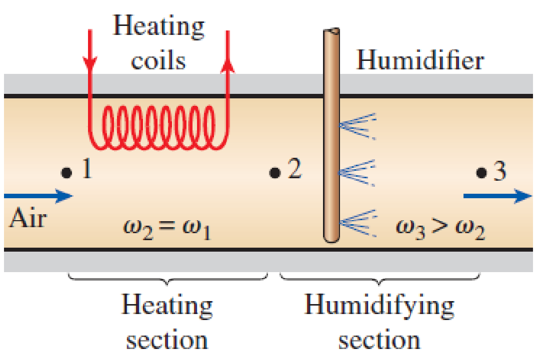
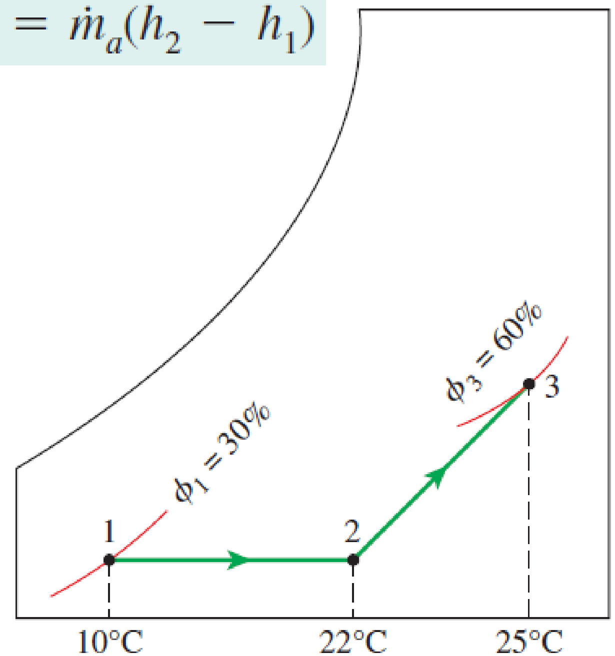
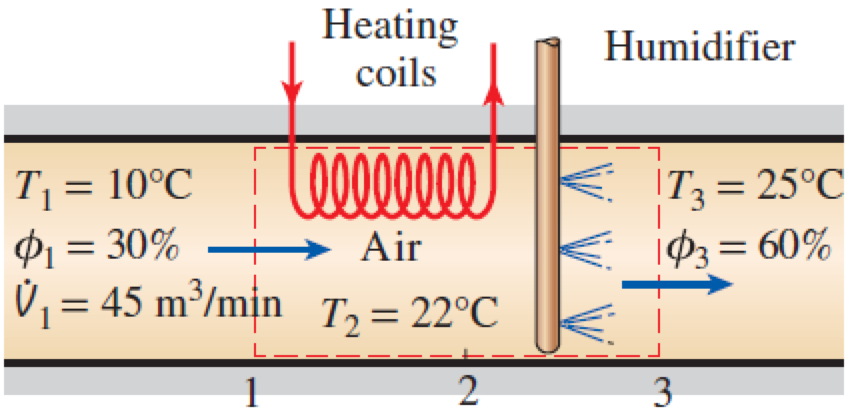


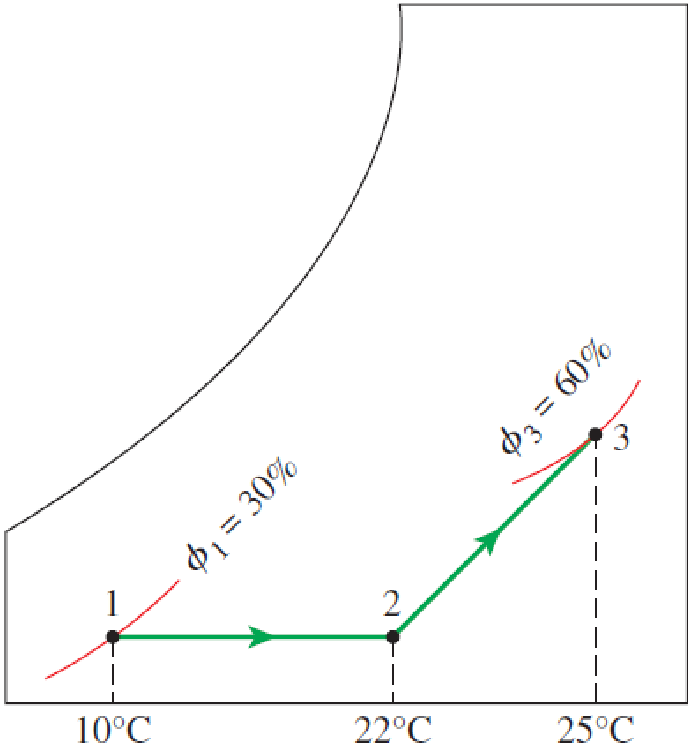
FIGURE 14–22
Heating with humidification.



2) Heating with Humidification

Heating and Humidification of Air





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

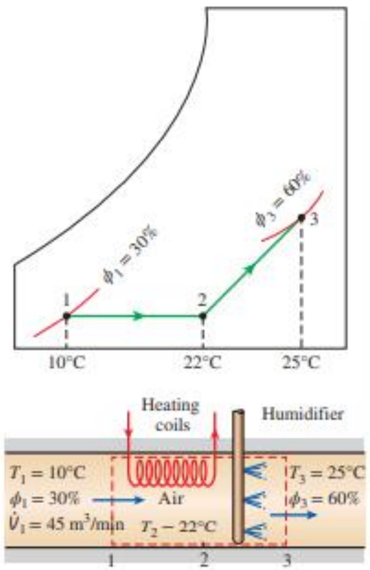


FIGURE 14–24
Schematic and psychrometric chart for Example 14–6.

EXAMPLE 14–6 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: $\dot{Q}_{in} + \dot{m}_a h_1 = \dot{m}_a h_2 \rightarrow \dot{Q}_{in} = \dot{m}_a(h_2 - h_1)$

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

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

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

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

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

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

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

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

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

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

(*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.

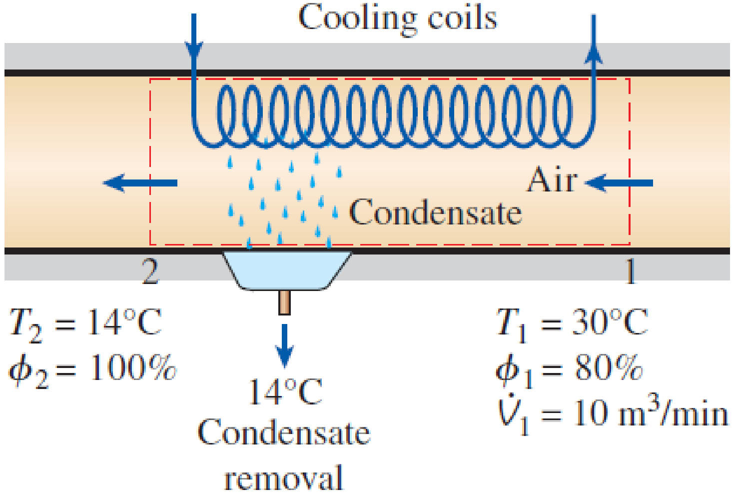
3) Cooling with Dehumidification

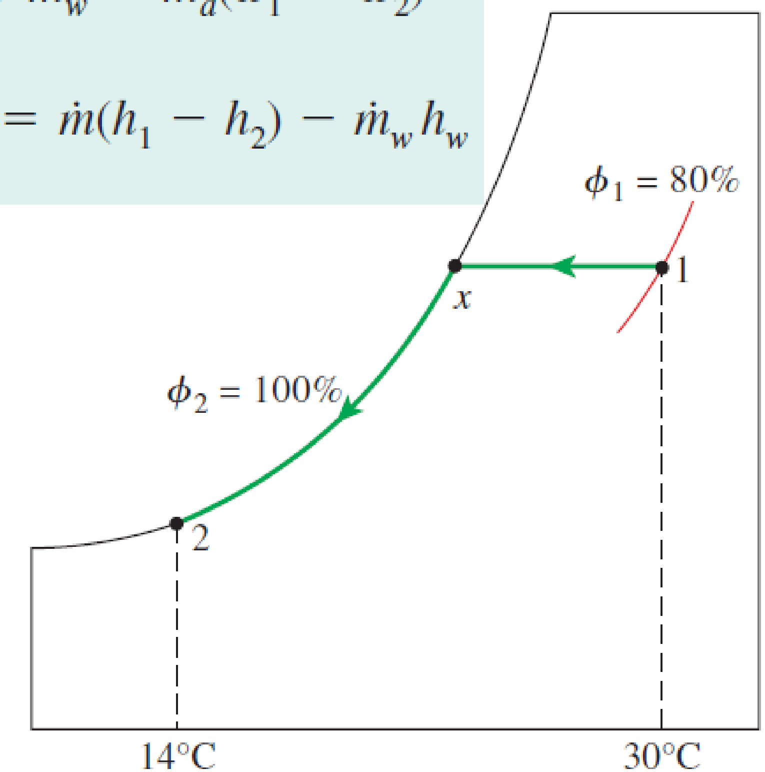
The specific humidity of air remains constant during a simple cooling process, but its relative humidity increases. If the relative humidity reaches undesirably high levels, it may be necessary to remove some moisture from the air, that is, to dehumidify it. This requires cooling the air below its dew-point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

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

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

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





EXAMPLE 14–7 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

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

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

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

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

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

Then,

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

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

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

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

4) Evaporative Cooling

In desert (*hot and dry*) climates, we can avoid the high cost of conventional cooling by using *evaporative coolers*, also known as *swamp coolers*.

As water evaporates, the latent heat of vaporization is absorbed from the water body and the surrounding air. As a result, both the water and the air are cooled during the process.

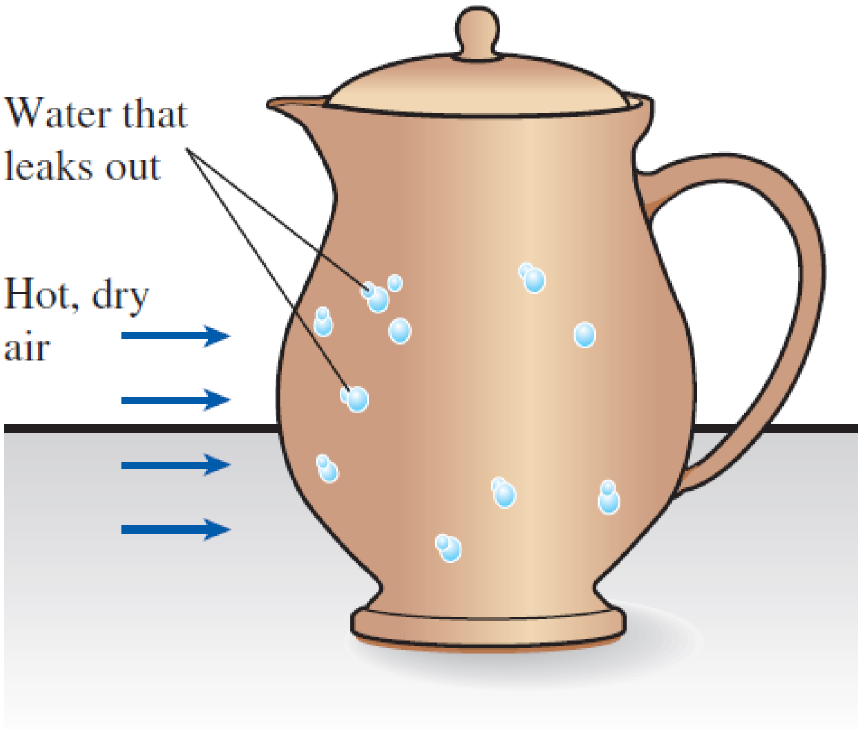
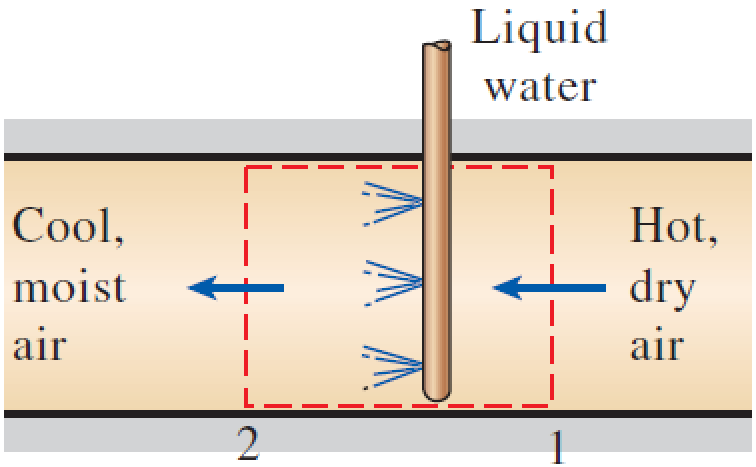
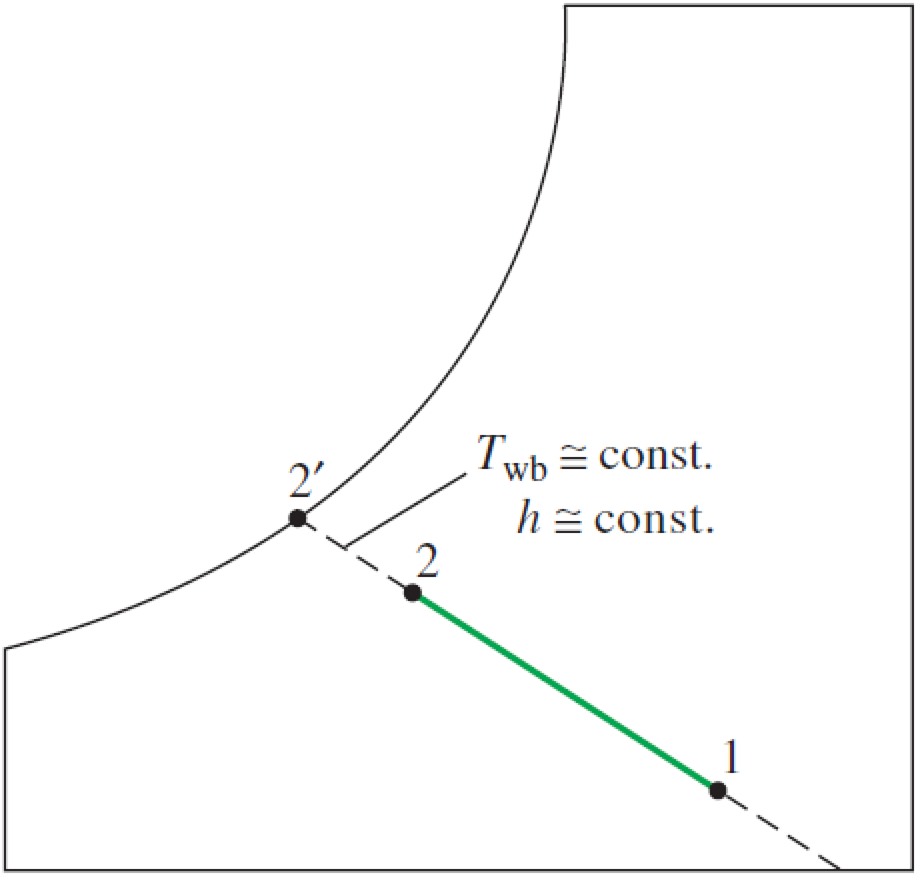


FIGURE 14–25
Water in a porous jug left in an open, breezy area cools as a result of evaporative cooling.

14.7 AIR-CONDITIONING PROCESSES

Evaporative cooling process is essentially identical to adiabatic saturation process.





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

$$h \cong \text{constant}$$

EXAMPLE 14–7 Evaporative Cooling with Soaked Head Cover

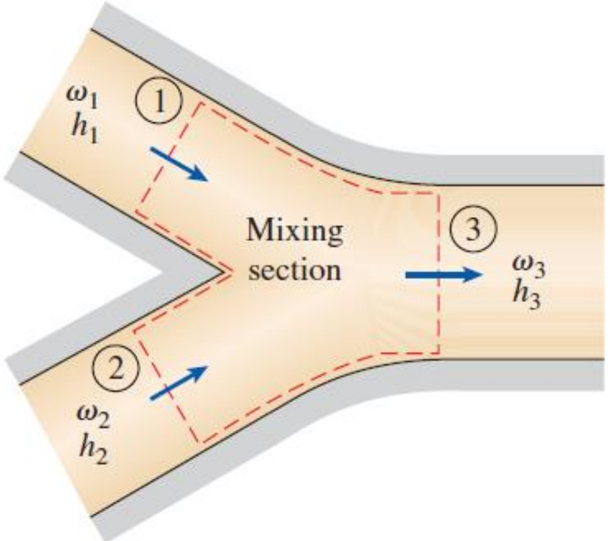
Desert dwellers often wrap their heads with a water-soaked porous cloth (Fig. 14-27). On a desert where the pressure is 1 atm, temperature is 50°C and relative humidity is 10 percent, what is the temperature of this cloth?

$T_{\mathrm{wb}} \approx \text{constant}$

$T_2 = T_{wb} =$ **23.8°C**

5) Adiabatic Mixing of Airstreams

Many A-C 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.



Mass of dry air:	$\dot{m}_{a_1} + \dot{m}_{a_2} = \dot{m}_{a_3}$
Mass of water vapor:	$\omega_1\dot{m}_{a_1} + \omega_2\dot{m}_{a_2} = \omega_3\dot{m}_{a_3}$
Energy:	$\dot{m}_{a_1}h_1 + \dot{m}_{a_2}h_2 = \dot{m}_{a_3}h_3$

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

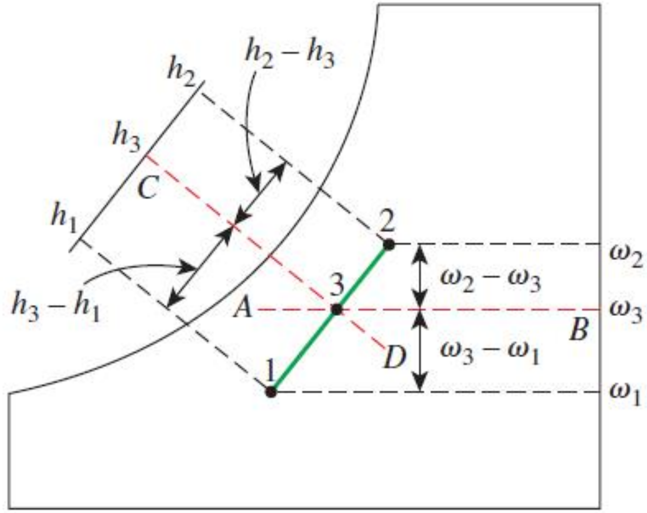


FIGURE 14–28
... two airstreams at states 1 and 2 are mixed adiabatically, the state of the mixture lies on the straight line connecting the two states.

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EXAMPLE 14–9 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, the 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

$$h_1 = 39.4 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.010 \text{ kg } H_2O\text{/kg dry air}$$
$$\upsilon_1 = 0.826 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 79.0 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0182 \text{ kg } H_2O\text{/kg dry air}$$
$$\upsilon_2 = 0.889 \text{ m}^3\text{/kg dry air}$$

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

$$\dot{m}_{a_1} = \frac{\dot{V}_1}{\upsilon_1} = \frac{50 \text{ m}^3\text{/min}}{0.826 \text{ m}^3\text{/kg dry air}} = 60.5 \text{ kg/min}$$

$$\dot{m}_{a_2} = \frac{\dot{V}_2}{\upsilon_2} = \frac{20 \text{ m}^3\text{/min}}{0.889 \text{ m}^3\text{/kg dry air}} = 22.5 \text{ kg/min}$$

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:

$$T_3 = \mathbf{19.0°C}$$
$$\phi_3 = \mathbf{89\%}$$
$$\upsilon_3 = 0.844 \text{ m}^3\text{/kg dry air}$$

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

$$\dot{V}_3 = \dot{m}_{a_3}\upsilon_3 = (83 \text{ kg/min})(0.844 \text{ m}^3\text{/kg}) = \mathbf{70.1 \text{ m}^3\text{/min}}$$

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.

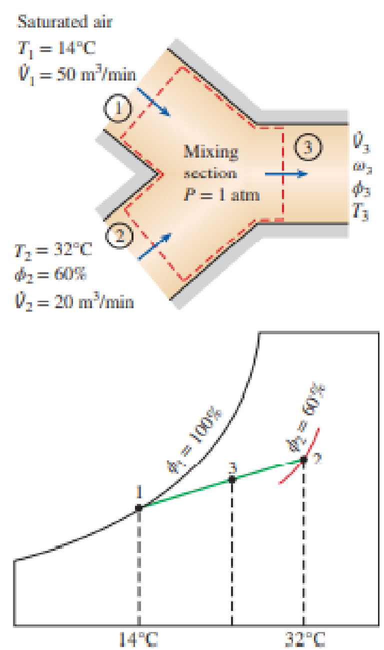


FIGURE 14–30
Schematic and psychrometric chart for Example 14–9.

6) 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 semi-enclosed evaporative cooler.

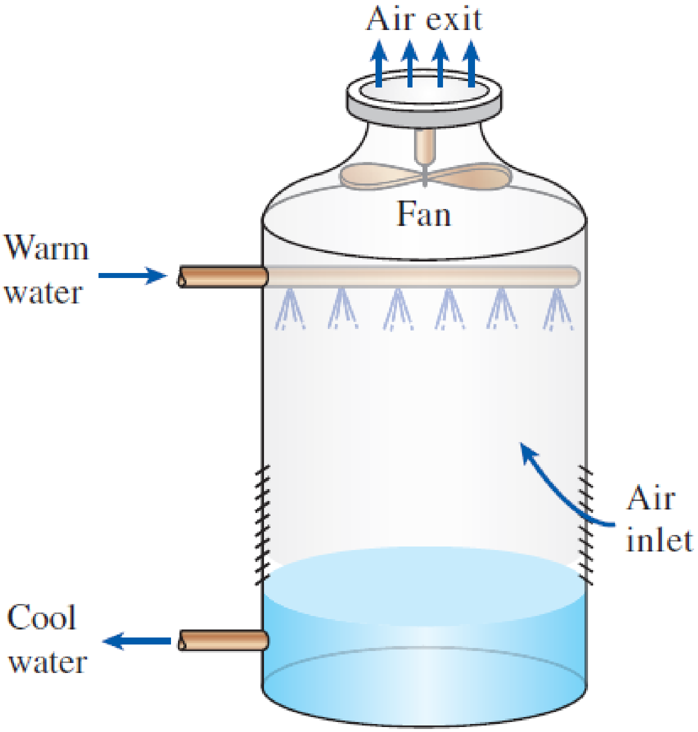


FIGURE 14–30
An induced-draft counterflow cooling tower.

14.7 AIR-CONDITIONING PROCESSES

Natural-draft cooling tower: It 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.

Spray pond: The warm water is sprayed into the air and is cooled by the air as it falls into the pond,

Cooling pond: Dumping the waste heat into a still pond, which is basically a large artificial lake open to the atmosphere.

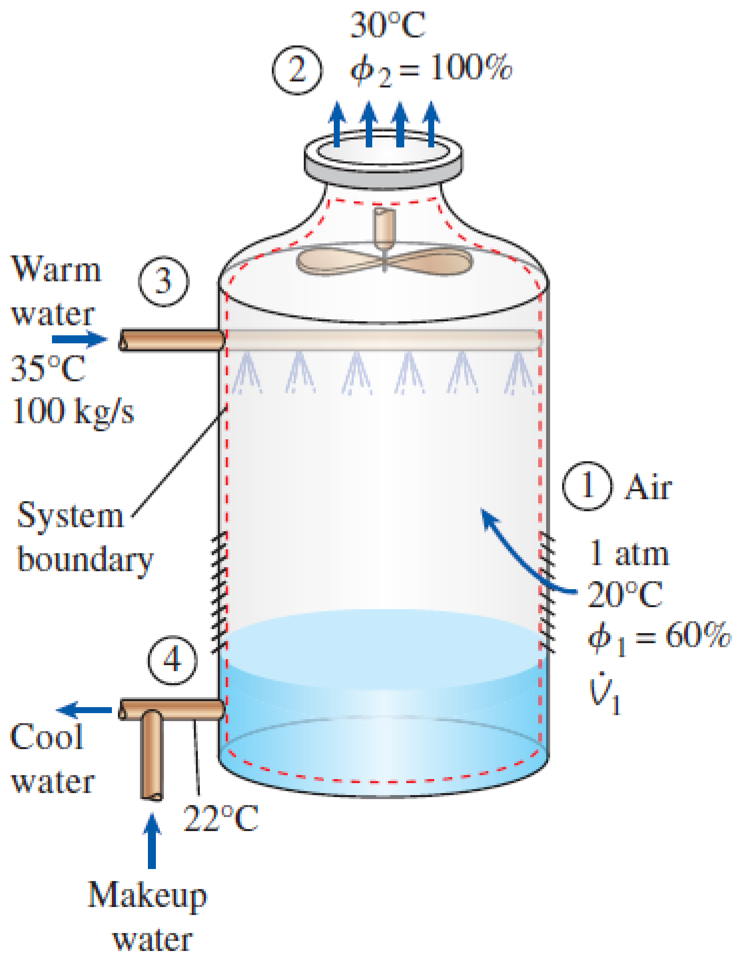
FIGURE 14–31
Two natural draft cooling towers on a roadside.

FIGURE 14–32
A spray pond.

14.7 AIR-CONDITIONING PROCESSES

FIGURE 14–34
Cooling pond.

Cooling of a Power Plant by a Cooling Tower



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

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

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

Energy balance:

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

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

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

Summary

- Dry and atmospheric air
- Specific and relative humidity of air
- Dew-point temperature
- Adiabatic saturation and wet-bulb temperatures
- The psychrometric chart
- Human comfort and air-conditioning
- Air-conditioning processes
  - ✓ Simple heating and cooling
  - ✓ Heating with humidification
  - ✓ Cooling with dehumidification
  - ✓ Evaporative cooling
  - ✓ Adiabatic mixing of airstreams
  - ✓ Wet cooling towers