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.

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.

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

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

and

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

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

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

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.

The enthalpy of water vapor in air can be taken to be equal to the enthalpy of saturated vapor at the same temperature. That is,

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

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

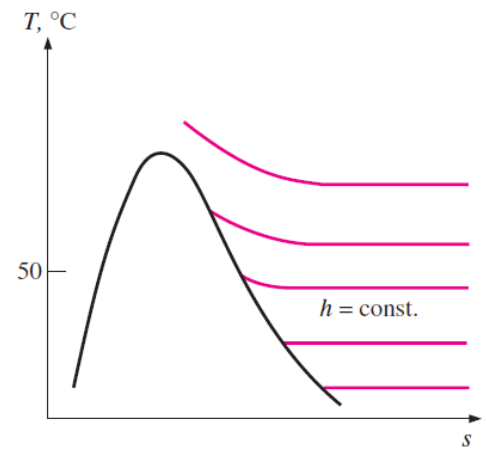


FIGURE 14–2

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

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

14–2 SPECIFIC AND RELATIVE HUMIDITY OF AIR

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

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

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

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

where P is the total pressure.

Vapor pressure of air remains constant ($P_{v2} = P_{v1}$) during a simple heating process.

The amount of water vapor in saturated air at a specified temperature and pressure can be determined from Eq. 14–8 by replacing $\boldsymbol{P_v}$ by $\boldsymbol{P_g}$, the saturation pressure of water at that temperature.

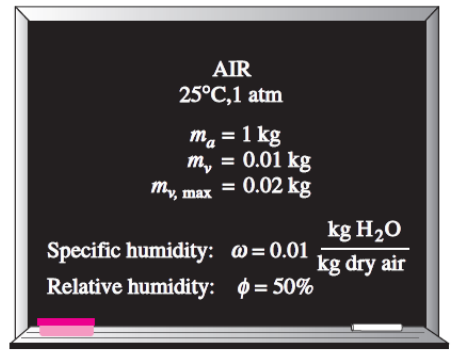


FIGURE 14–5

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.

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.

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

$$P_g = P_{\text{sat @ } T} \qquad \textbf{(14–10)}$$

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

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:

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

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

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.

14–3 DEW-POINT TEMPERATURE

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:

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

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.

14–4 ADIABATIC SATURATION AND WET-BULB TEMPERATURE

One 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 v1 (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.**

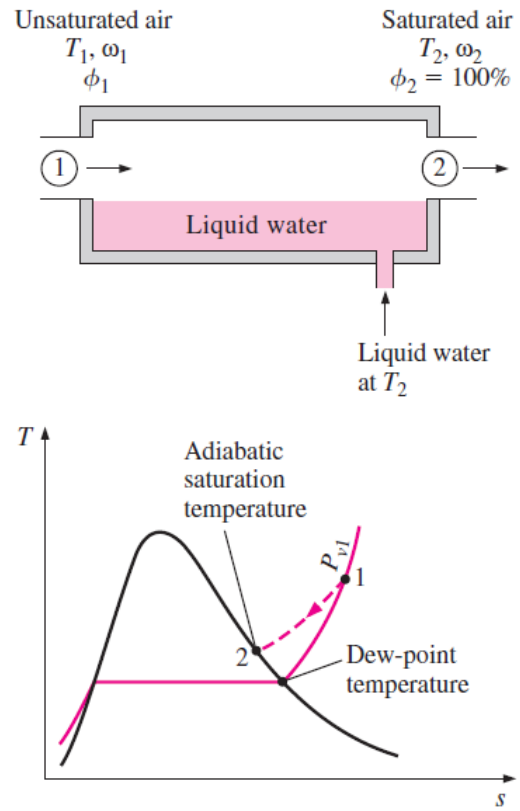


FIGURE 14–11
The adiabatic saturation process and its representation on a *T-s* diagram of water.

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

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

Since Φ_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.

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.

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.

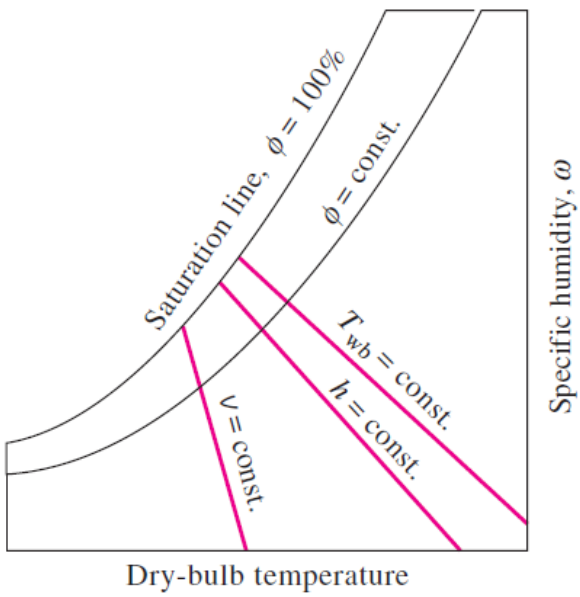


FIGURE 14–14

Schematic for a psychrometric chart.

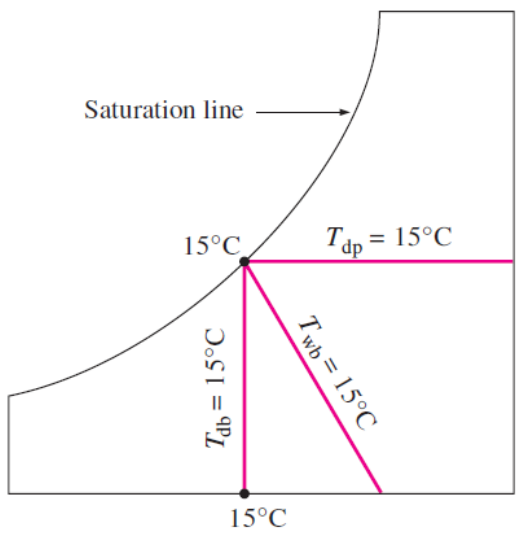


FIGURE 14–15

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

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

14–6 HUMAN COMFORT AND AIR-CONDITIONING

As engineers, it is our duty to help people feel comfortable. (Besides, it keeps us employed.)

14–7 AIR-CONDITIONING PROCESSES

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

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

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

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

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

Simple Heating and Cooling (ω = constant)

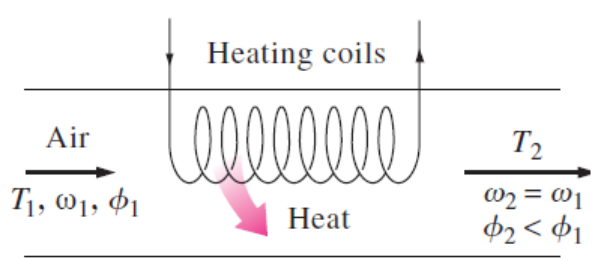


FIGURE 14–21

During simple heating, specific humidity remains constant, but relative humidity decreases.

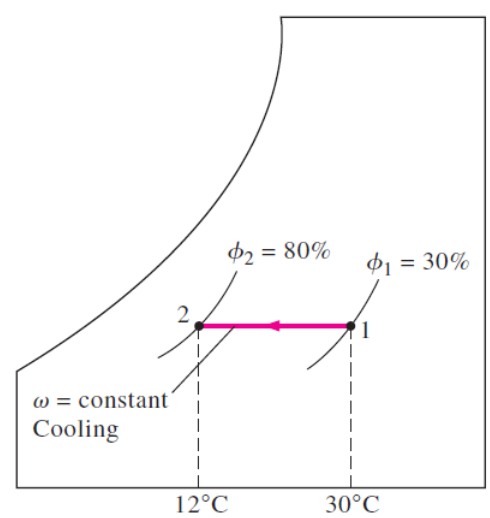


FIGURE 14–22

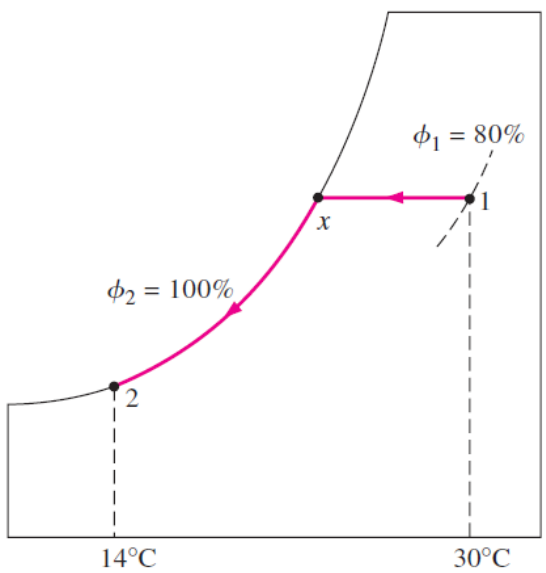
During simple cooling, specific humidity remains constant, but relative humidity increases.

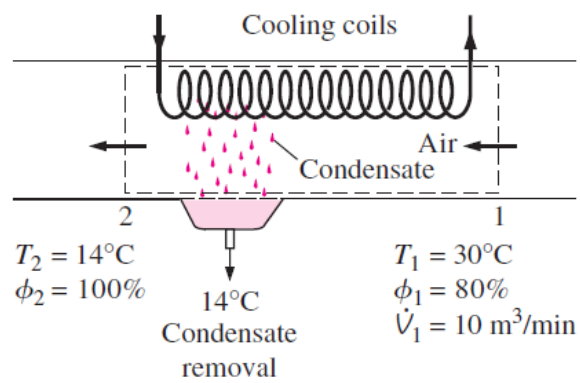
Heating with Humidification

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.

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.





Evaporative Cooling

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.

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.

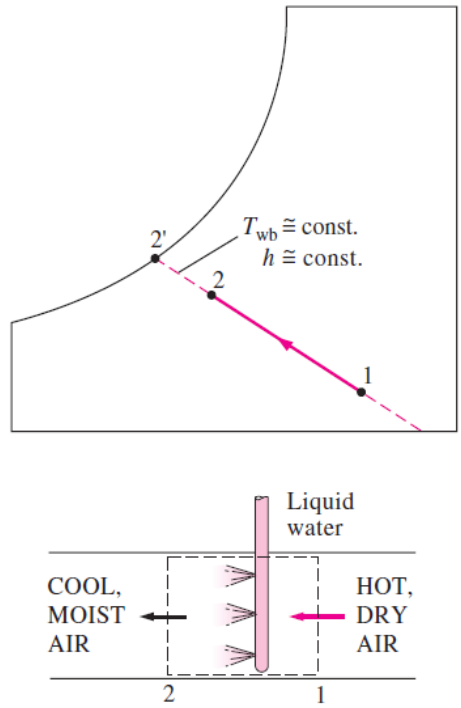


FIGURE 14–27
Evaporative cooling.

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

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

and

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

during an evaporative cooling process.

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.

Mass of dry air: $$\dot{m}_{a_1} + \dot{m}_{a_2} = \dot{m}_{a_3} \quad (14\text{–}21)$$

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

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

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

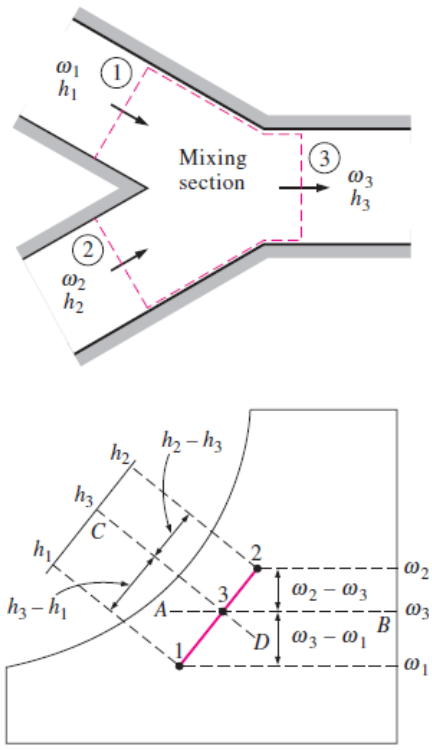
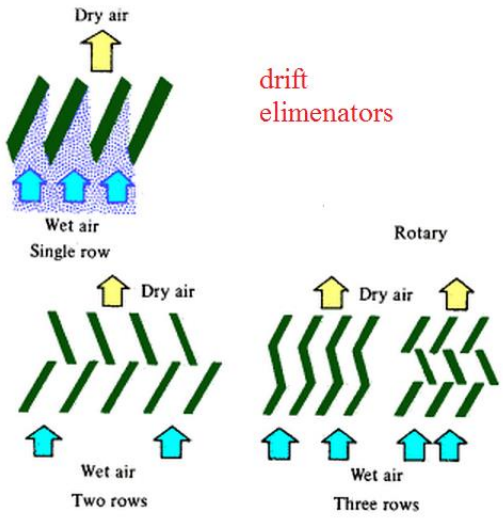


FIGURE 14–29
When two airstreams at states 1 and 2 are mixed adiabatically, the state of the mixture lies on the straight line connecting the two states.

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.

Wet Cooling Towers

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.