Topic2: Gas-Vapor Mixtures and Air-Conditioning

The University of Manchester

Further Reading

Chapter 14
Gas-Vapor Mixtures and Air-Conditioning

Content

- 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.

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 1

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.

$$h_{\text{dry air}} = c_P T = (1.005\ \text{kJ/kg.°C})T \quad (\text{kJ/kg})$$

$$\Delta h_{\text{dry air}} = c_P \Delta T = (1.005\ \text{kJ/kg.°C})\Delta T \quad (\text{kJ/kg})$$

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 2

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.

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

Taking 0°C as the reference temperature, the enthalpy and enthalpy change of dry air can be determined from

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

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

14–1 Dry And Atmospheric Air [3]

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:

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

P_a Partial pressure of dry air

P_v Partial pressure of vapor (vapor pressure)

vapor pressure: It is the pressure water vapor would exert if it existed alone at the temperature and volume of atmospheric air

14–1 Dry And Atmospheric Air [4]

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

Therefore, the *enthalpy of water vapor in air can be taken to be equal to the enthalpy of saturated vapor at the same temperature*. So:

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

For water

h_g = 2500.9 kJ/kg (at 0°C)

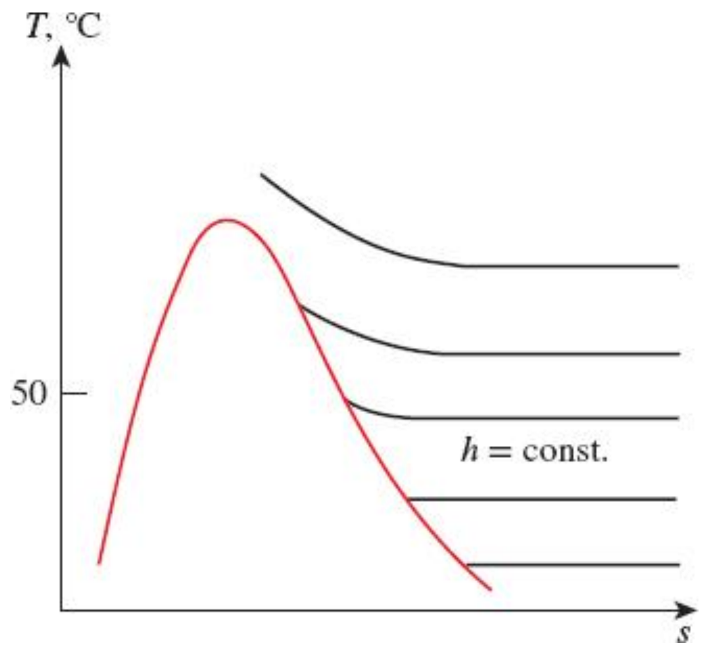
$c_{p,avg}$ = 1.82 kJ/kg·°C (at –10 to 50°C range)

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

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

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.



14–1 Dry And Atmospheric Air [5]

Figure 14–3

In the temperature range –10 to 50°C, the h_g of water can be determined from (Eq. 14-4):

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

with negligible error.

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

14–2 Specific and Relative Humidity of Air [1]

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.

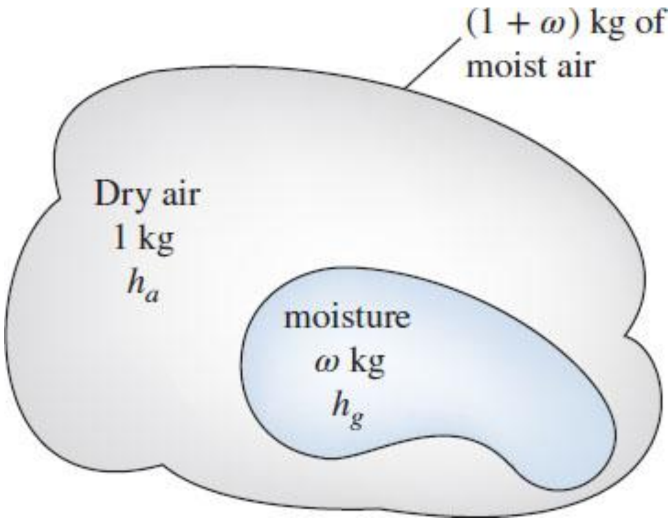
Absolute or **specific humidity** (*humidity ratio*): The mass of water vapor present in a unit mass of dry air.

$$\omega = \frac{m_v}{m_a} \quad (\text{kg water vapor / kg dry air})$$

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

$$\boxed{\omega = \frac{0.622 P_v}{P - P_v}} \quad (\text{kg water vapor / kg dry air})$$

where P is the total pressure.



14–2 Specific and Relative Humidity of Air [2]

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).

$$\boxed{\phi = \frac{m_v}{m_g} = \frac{P_vV/R_vT}{P_gV/R_vT} = \frac{P_v}{P_g}}$$

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

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

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

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.

14–2 Specific and Relative Humidity of Air 3

Figure 14–4

For saturated air, the vapor pressure is equal to the saturation pressure of water.

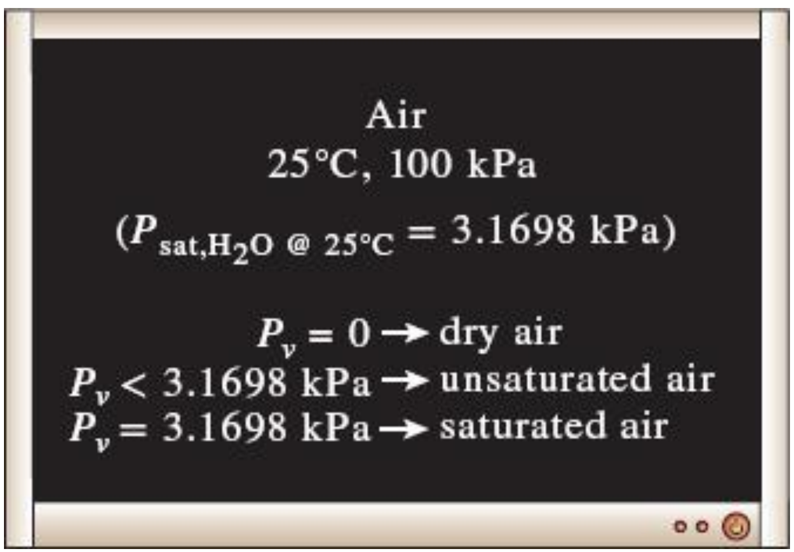
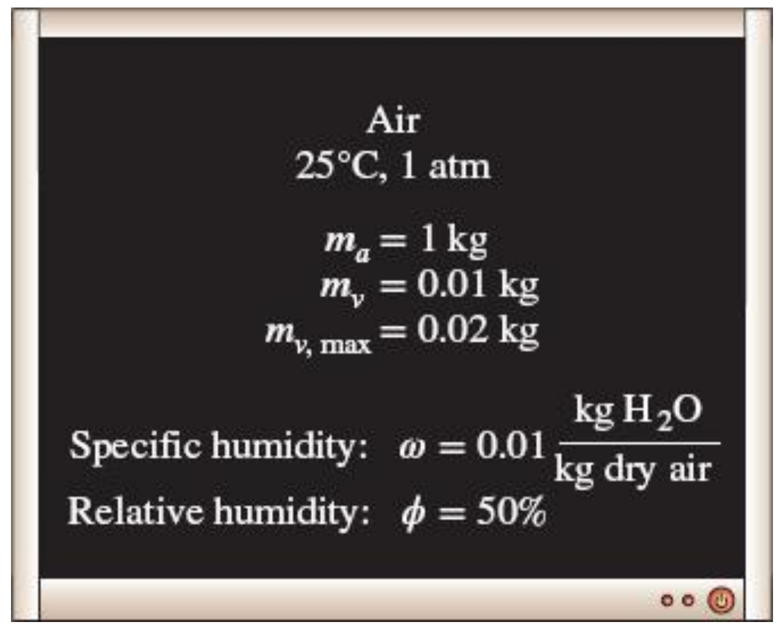


Figure 14–5

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 4

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 specific enthalpy of atmospheric air is expressed *per unit mass of dry air.*

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

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

$$h_v \cong h_g$$

$$\boxed{h = h_a + \omega h_g} \quad (\mathrm{kJ/kg\,dry\,air})$$

(Eq. 14-12)

Dry-bulb temperature: The ordinary temperature of atmospheric air (to differentiate it from other forms of temperatures to be discussed later).

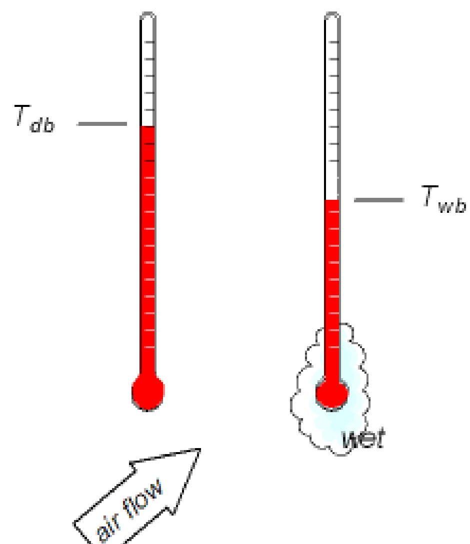
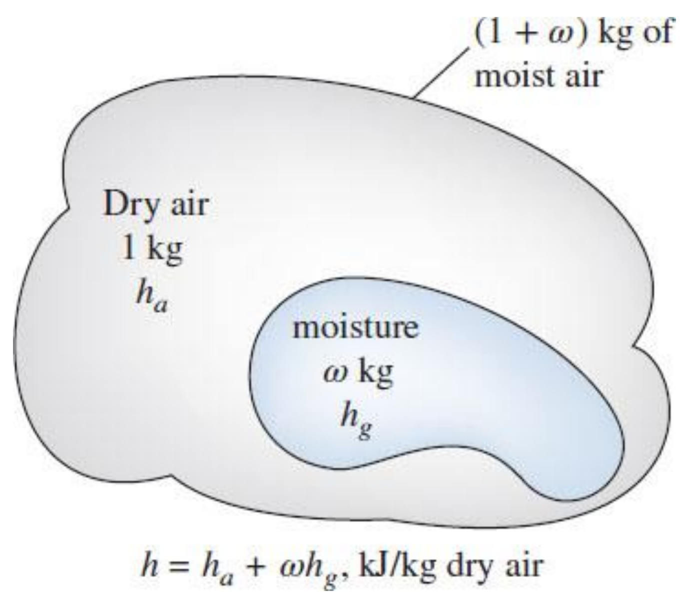


Figure 14–6

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



14–2 Specific and Relative Humidity of Air

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 $P_{sat} = 3.1698$ 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

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

Thus,

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

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

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

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

$$h = h_a + \omega h_v \cong c_p T + \omega h_g$$
$$= (1.005\ \text{kJ/kg·°C})(25°\text{C}) + (0.0152)(2546.5\ \text{kJ/kg})$$
$$= \mathbf{63.8\ kJ/kg\ dry\ air}$$

The enthalpy of water vapor (2546.5 kJ/kg) could also be determined from the approximation given by Eq. 14–4: $h_g = (T) \cong 2500.9 + 1.82T \quad (\text{kJ/kg}) \quad T$ in °C

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

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:

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

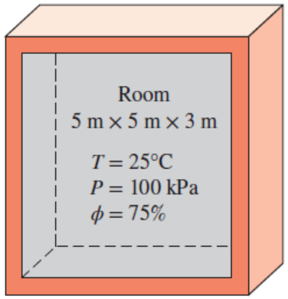
The masses of the dry air and the water vapor are determined from the ideal-gas relation applied to each gas separately: (R_a&R_v are given in Table A-1)

$$m_a = \frac{P_a V_a}{R_a T} = \frac{(97.62\ \text{kPa})(75\ \text{m}^3)}{(0.287\ \text{kPa·m}^3/\text{kg·K})(298\ \text{K})} = \mathbf{85.61\ kg}$$

$$m_v = \frac{P_v V_v}{R_v T} = \frac{(2.38\ \text{kPa})(75\ \text{m}^3)}{(0.4615\ \text{kPa·m}^3/\text{kg·K})(298\ \text{K})} = \mathbf{1.30\ kg}$$

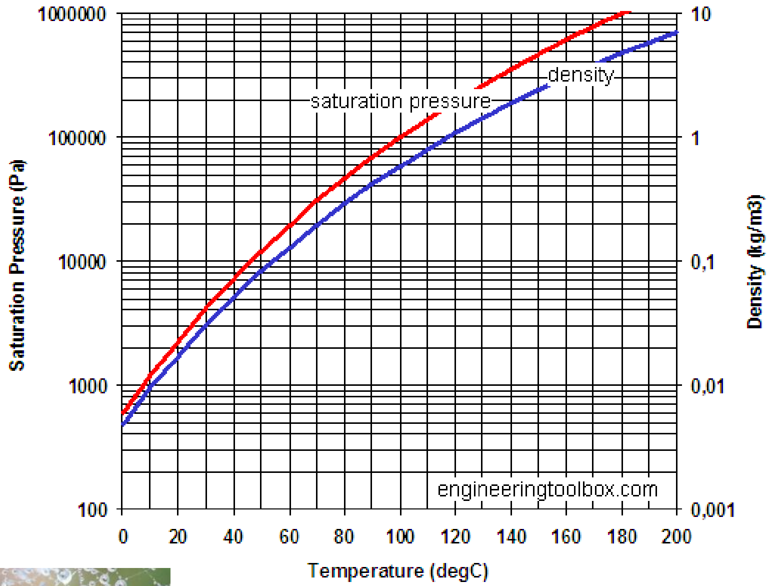
The mass of the water vapor in the air could also be determined from Eq. 14–6:

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



14–3 Dew-Point Temperature [1]

The excess moisture in the air can be condensed on the cool surfaces, forming what we call *dew* (e.g. wet grass in summer mornings). In summer, a considerable amount of water vaporizes during the day. As the temperature falls during the night, so does the “*moisture capacity*” of air.



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.

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

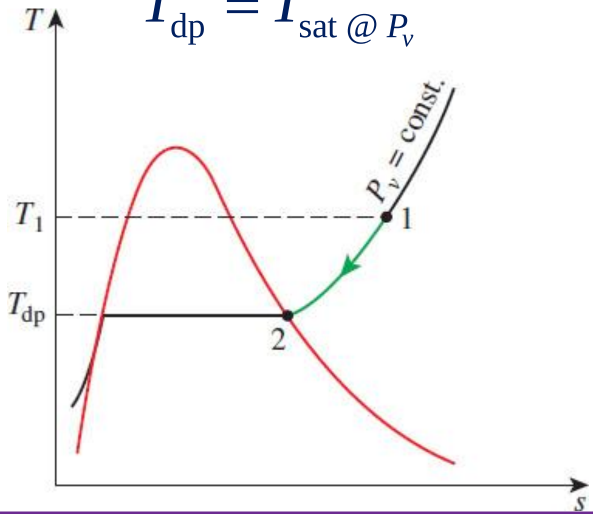


Figure 14–8: Constant-pressure cooling of moist air and the dew-point temperature on the *T*-*s* diagram of water.

14–3 Dew-Point Temperature [2]

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

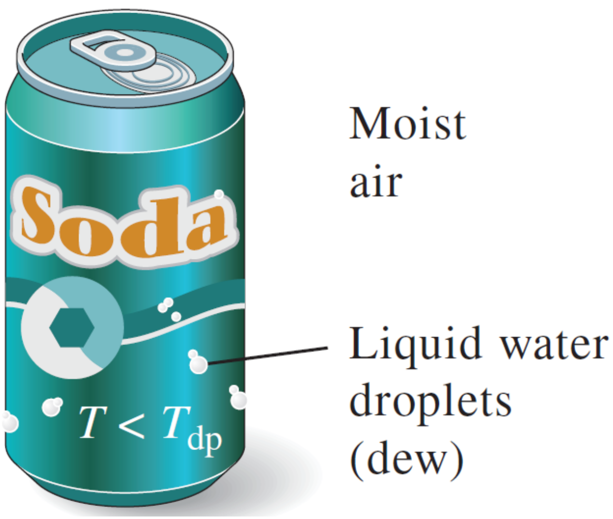


Figure 14–9

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

Condensation on the internal surface of the window/glazing in winter, resulting in black mould, with serious health implications

14–3 Dew-Point Temperature 3

EXAMPLE 14–2 **Fogging of the Windows in a House**

In cold weather, condensation often 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

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

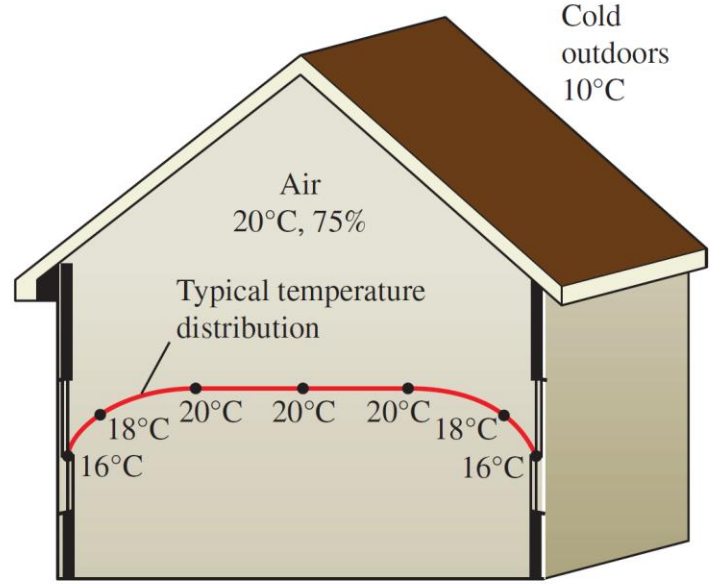
where

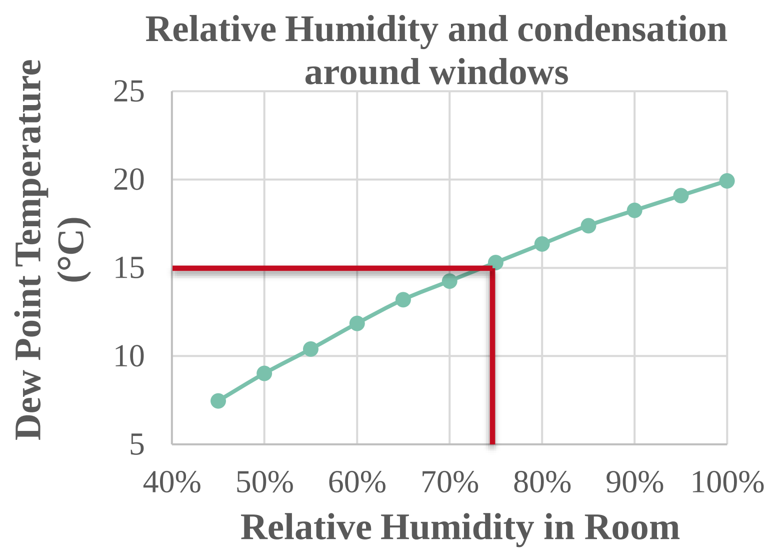
$$P_v = \phi P_{g\ @\ 20°C} = (0.75)(2.3392\ \text{kPa}) = 1.754\ \text{kPa}$$

Thus,

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

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 [1]

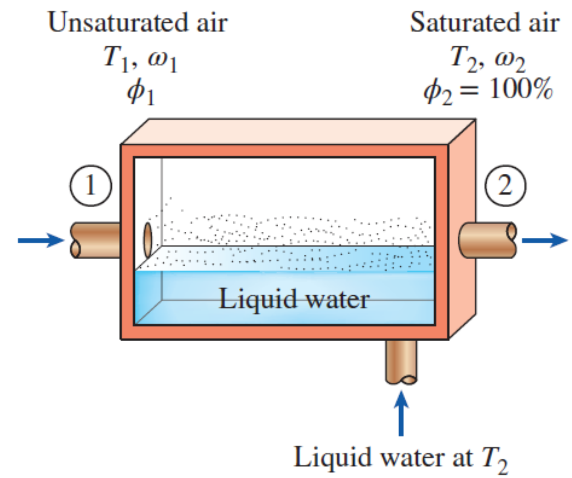
Relative/specific humidity are frequently used in engineering and atmospheric sciences, and it is desirable to relate them to easily measurable quantities such as <u>temperature and pressure</u>.

One way of determining the relative humidity is to determine the dew-point temperature of air, as discussed earlier. 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.

$$T_{\mathrm{dp}} = T_{\mathrm{sat}\,@\,P_v} \qquad \phi = \frac{m_v}{m_g} = \frac{P_v V / R_v T}{P_g V / R_v T} = \frac{P_v}{P_g}$$

Another way of determining the absolute/relative humidity is related to an *adiabatic saturation process*:

- 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%) at temperature T_2, which is called the adiabatic saturation temperature.



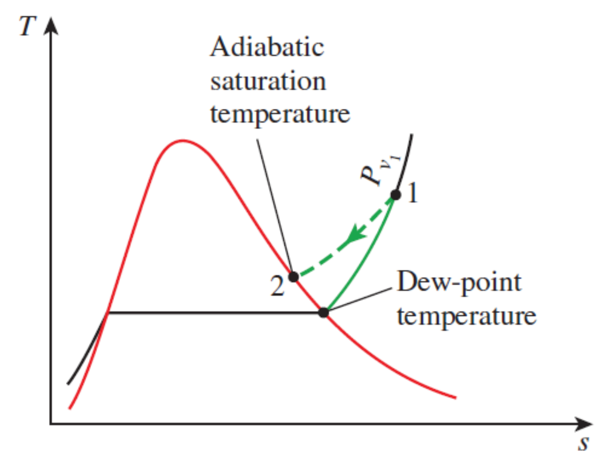


Figure 14–11

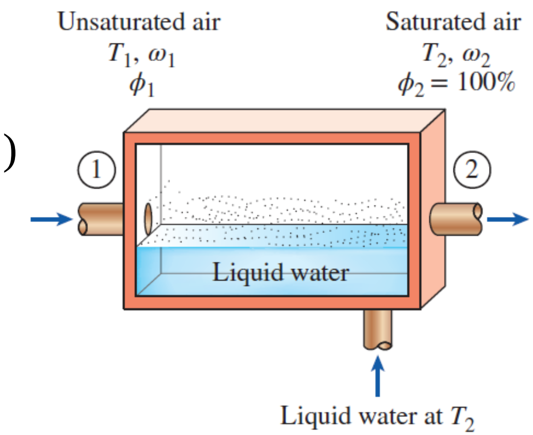
The adiabatic saturation process and its representation on a *T*-*s* diagram of water.

14–4 Adiabatic Saturation and Wet-bulb Temperatures 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.

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



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

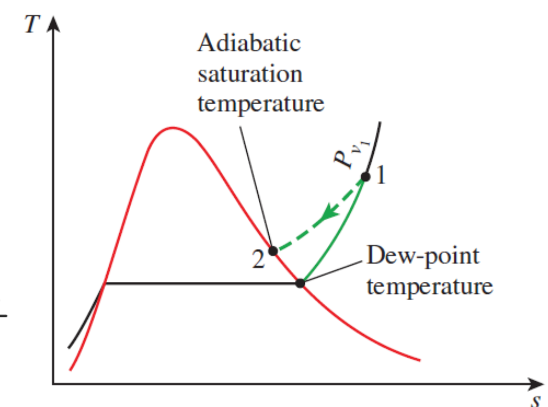
Energy Balance:

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

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

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

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



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

$$\boxed{\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg_2}}{h_{g_1} - h_{f_2}}}$$ where $$\boxed{\omega_2 = \frac{0.622 P_{g_2}}{P_2 - P_{g_2}}}$$ Since $$\omega = \frac{0.622 P_v}{P - P_v}, \quad \phi_2 = 100\,\%$$

14–4 Adiabatic Saturation and Wet-bulb Temperatures 3

- Thus we conclude that the specific humidity (and relative humidity) of air can be determined 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 $\omega_1 = \omega_2$. <u>In general, the adiabatic saturation temperature is between the inlet and dew-point temperatures.</u>

- The adiabatic saturation process 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.

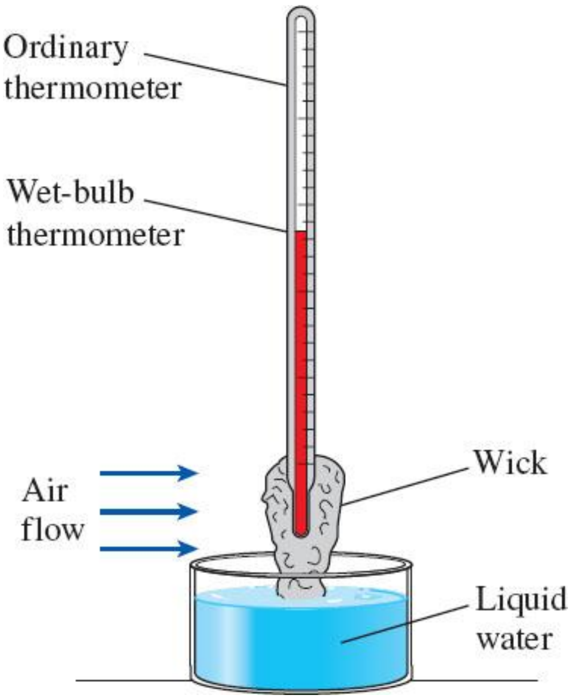
- To determine the absolute and relative humidity of air, a more practical approach is to use a thermometer whose bulb is covered with a cotton wick saturated with water and to blow air over the wick.

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

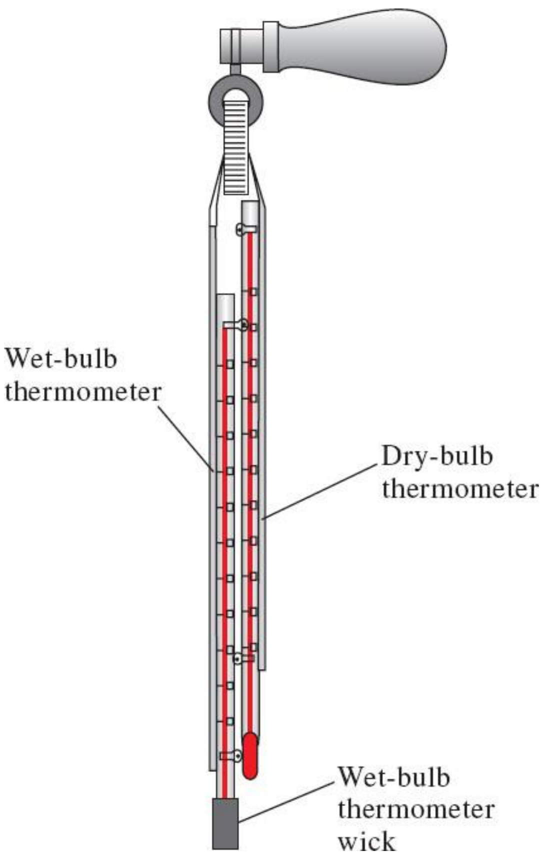


14–4 Adiabatic Saturation and Wet-bulb Temperatures [4]

- 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 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.
- Nowadays, handheld 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.

Figure 14–13

Sling psychrometer.



14–4 Adiabatic Saturation and Wet-bulb Temperatures 5

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–2*a*).

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,

$$\omega_1 = \frac{(1.005\text{ kJ/kg·°C})[(15-25)\text{°C}] + (0.01065)(2465.4\text{ kJ/kg})}{(2546.5 - 62.982)\text{ kJ/kg}}$$
$$= \mathbf{0.00653\ kg\ H_2O/kg\ dry\ air}$$

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

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

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

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

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

TABLE A–4

Saturated water—Temperature table

			Specific volume, m³/kg		*Internal energy,* kJ/kg			*Enthalpy,* kJ/kg		
	Temp., T °C	Sat. Press., P_{sat} kPa	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g
	0.01	0.6117	0.001000	206.00	0.000	2374.9	2374.9	0.001	2500.9	2500.9
	5	0.8725	0.001000	147.03	21.019	2360.8	2381.8	21.020	2489.1	2510.1
	10	1.2281	0.001000	106.32	42.020	2346.6	2388.7	42.022	2477.2	2519.2
2	15	1.7057	0.001001	77.885	62.980	2332.5	2395.5	62.982	2465.4	2528.3
	20	2.3392	0.001002	57.762	83.913	2318.4	2402.3	83.915	2453.5	2537.4
1	25	3.1698	0.001003	43.340	104.83	2304.3	2409.1	104.83	2441.7	2546.5

14–5 The Psychrometric Chart

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.

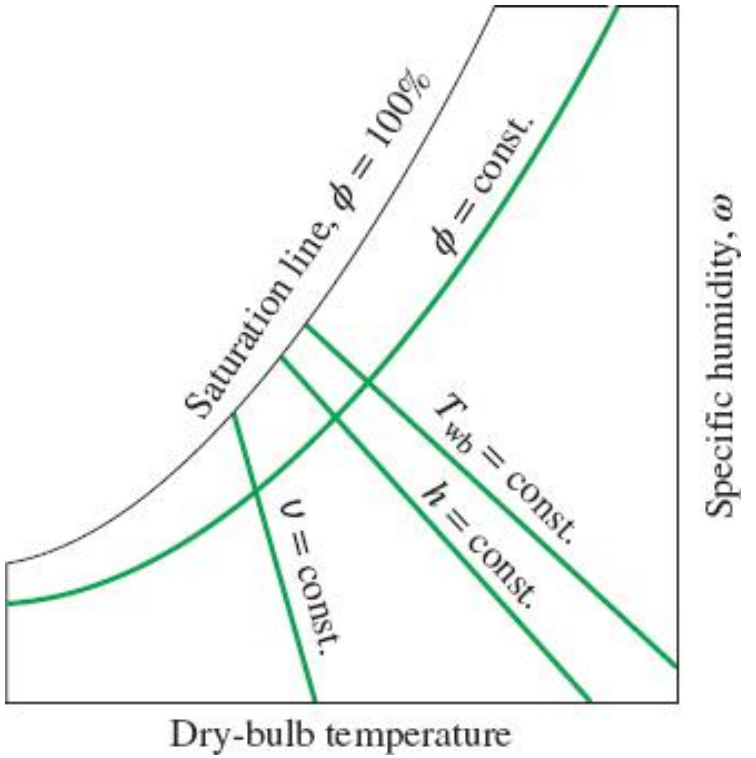


Figure 14–15

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

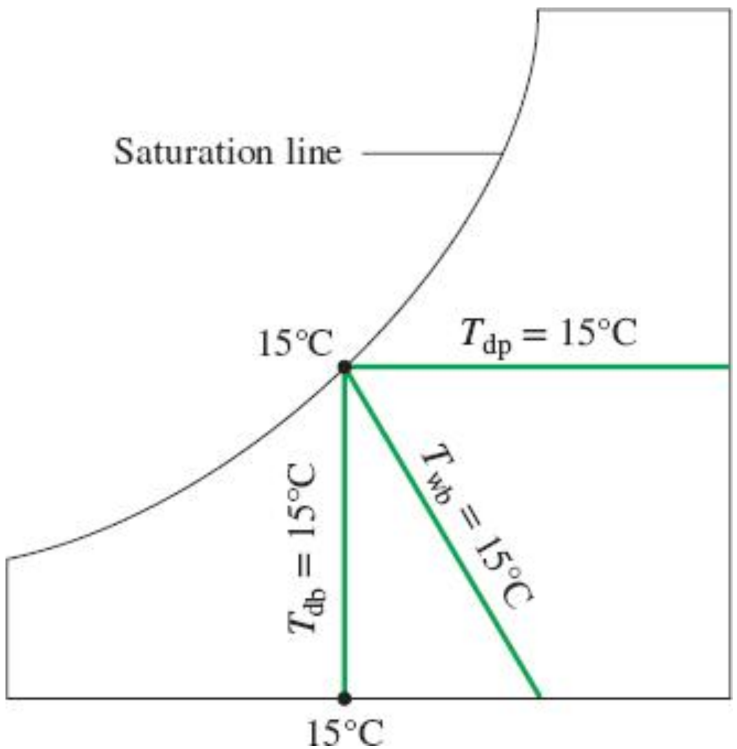
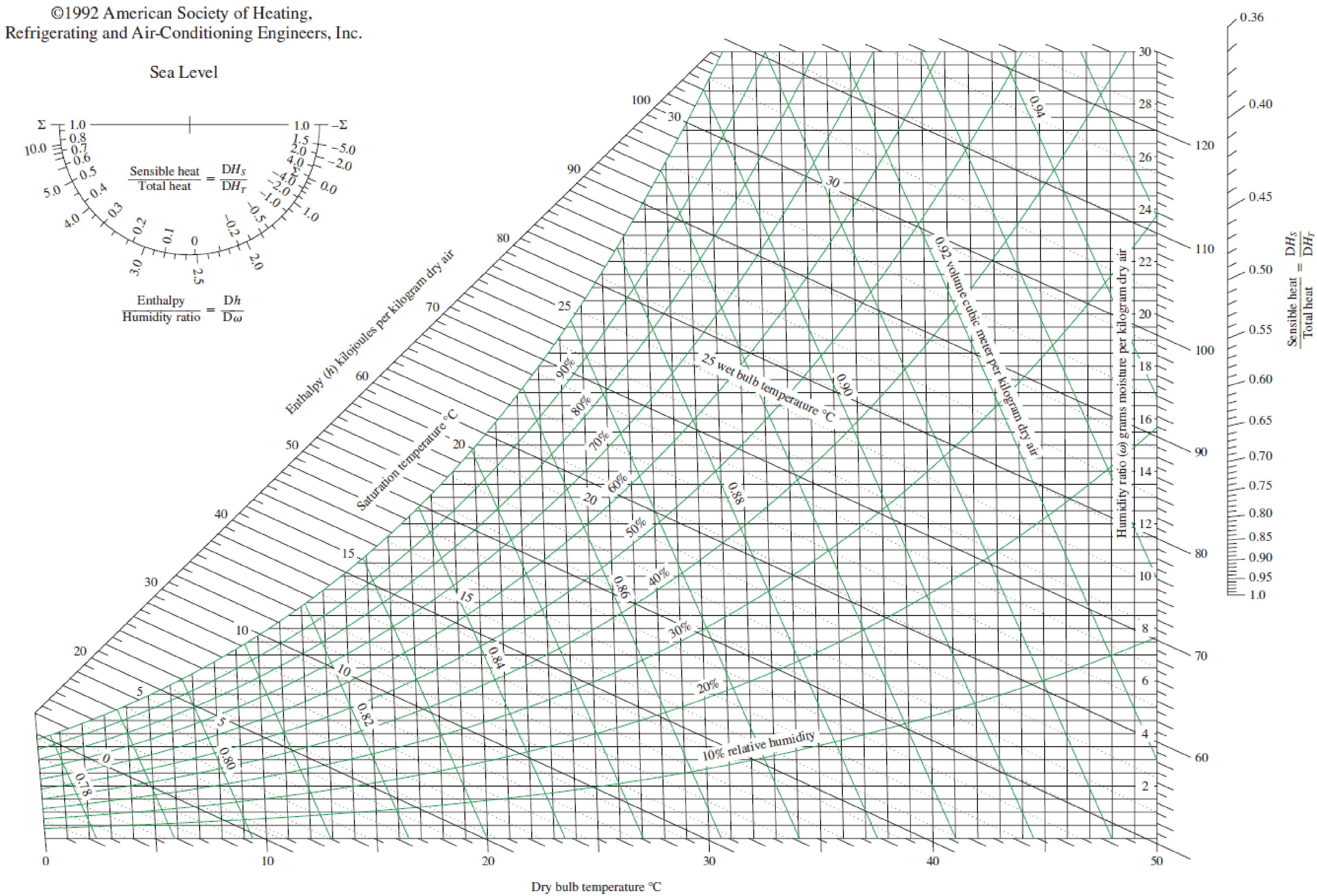


Fig A-31 (Appendix)

MANCHESTER 1824



14–5 The Psychrometric Chart

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 υ = constant lines on both sides of the point. The specific volume is determined by visual interpolation to be

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

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

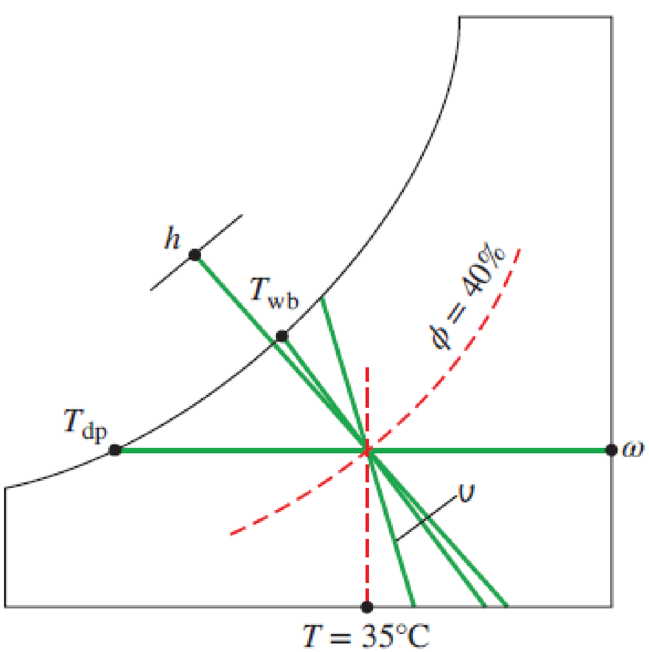
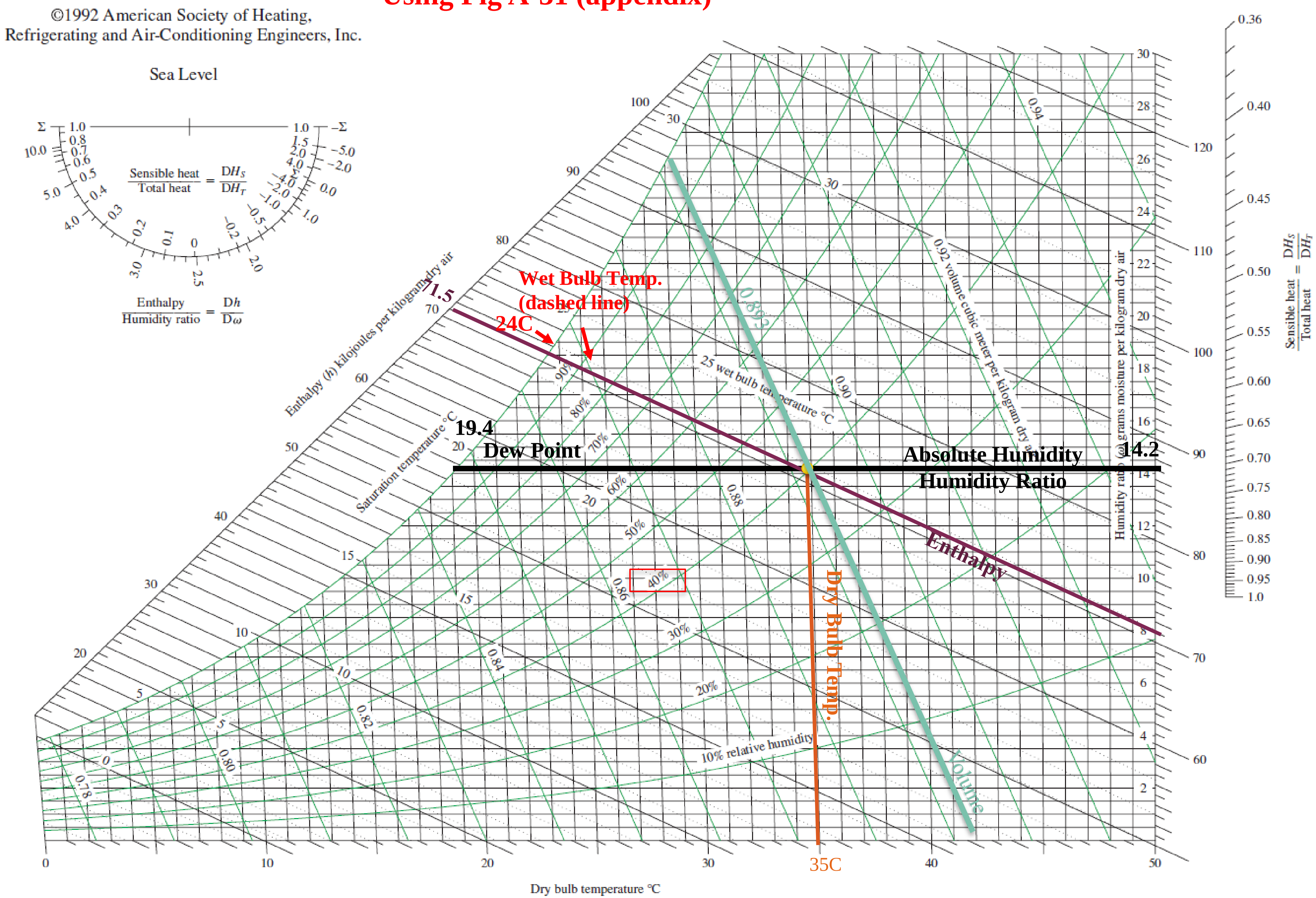


FIGURE 14–16
Schematic for Example 14–4.

Example 14-4
Using Fig A-31 (appendix)



14–6 Human Comfort and Air-conditioning [1]

- 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

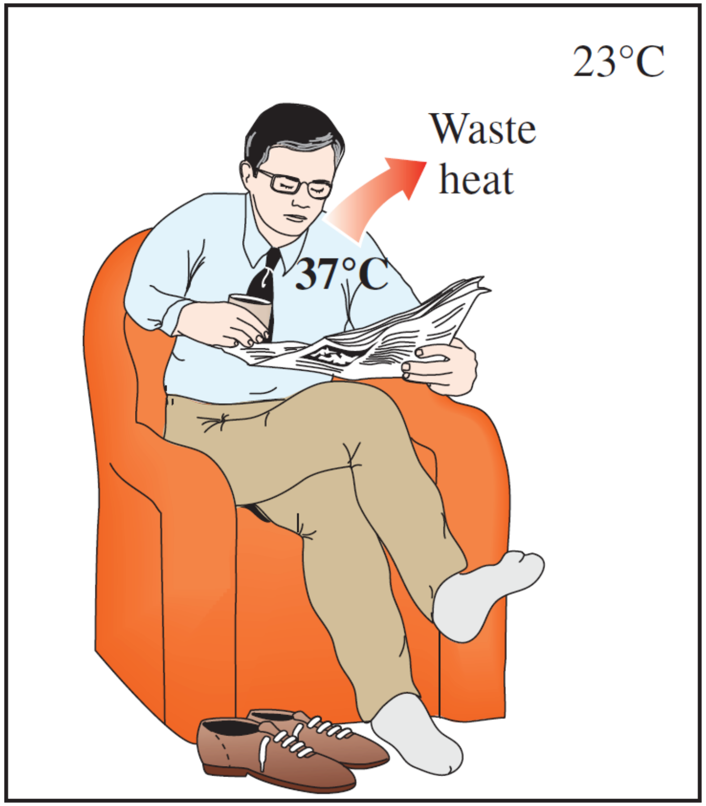
We cannot change the weather, but we can change the climate in a confined space by air-conditioning.

14–6 Human Comfort and Air-conditioning

- 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.

Figure 14–18

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

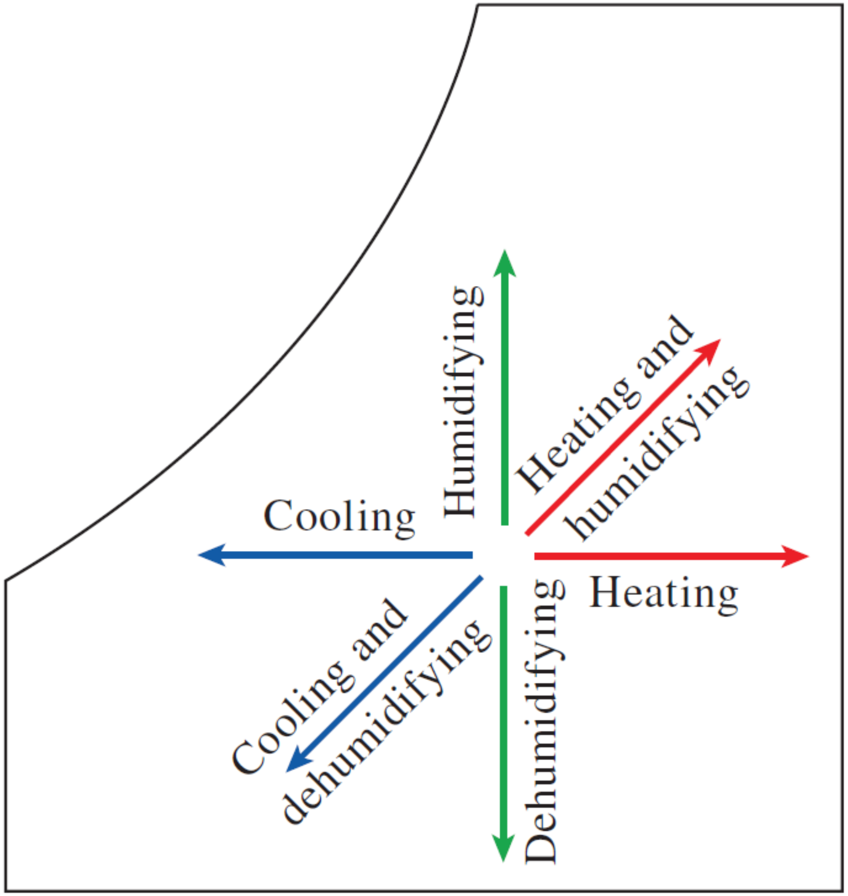


14–7 Air-conditioning Processes [1]

- 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&humidified in winter and cooled&dehumidified in summer.

Figure 14–19

Various air-conditioning processes.



14–7 Air-conditioning Processes [2]

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: $\sum_{\text{in}} \dot{m}_a = \sum_{\text{out}} \dot{m}_a \quad (\text{Kg/s})$

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

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

$$\dot{Q}_{\text{in}} + \cancelto{0}{\dot{W}_{\text{in}}} + \sum_{\text{in}} \dot{m}h = \dot{Q}_{\text{out}} + \cancelto{0}{\dot{W}_{\text{out}}} + \sum_{\text{out}} \dot{m}h$$

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 3

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.

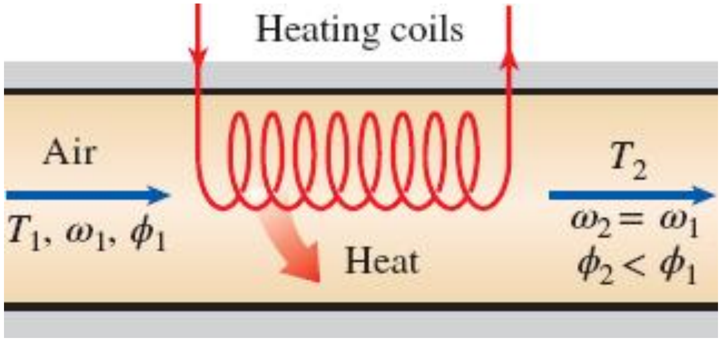
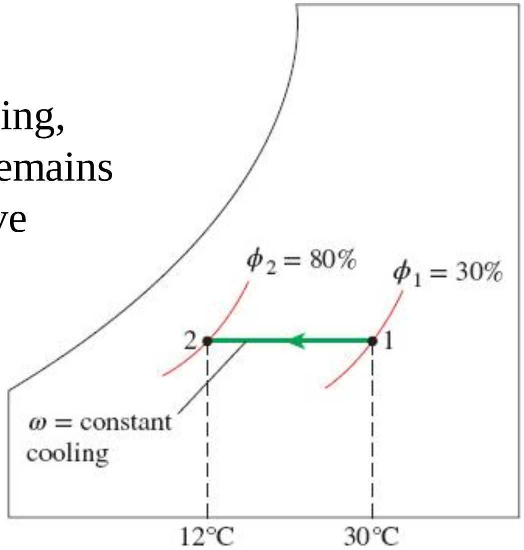


Figure 14–20

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.



14–7 Air-conditioning Processes [4]

The conservation of mass equations for a heating or cooling process that involves no humidification or dehumidification reduce to

$\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$ 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

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

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.

14–7 Air-conditioning Processes

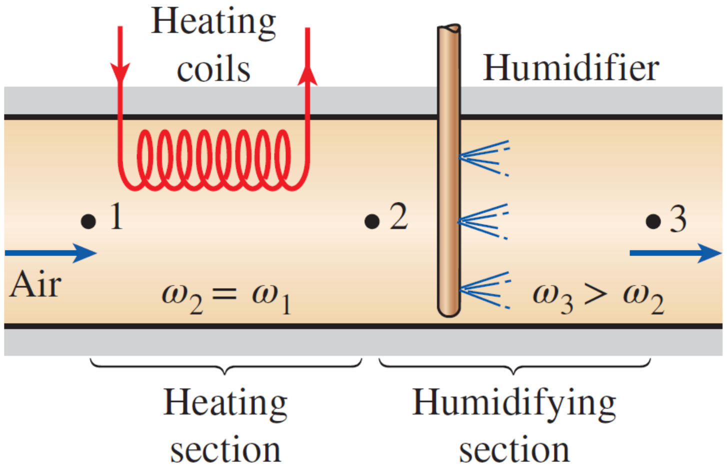
Heating with Humidification

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. 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.



14–7 Air-conditioning Processes 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: $$\dot{Q}_{\text{in}} + \dot{m}_a h_1 = \dot{m}_a h_2 \rightarrow \dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1)$$

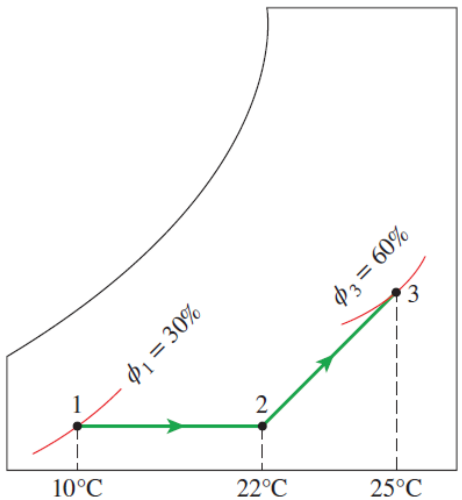
The psychrometric chart offers great convenience in determining the properties of moist air. However, its use is limited to a specified pressure only, which is 1 atm (101.325 kPa) for the one given in the appendix. At pressures other than 1 atm, either other charts for that pressure or the relations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

TABLE A–4

Saturated water—Temperature table

Temp., T °C	Sat. Press., P_{sat} kPa	Specific volume, m^3/kg — Sat. liquid, v_f	Sat. vapor, v_g	Internal energy, kJ/kg — Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Enthalpy, kJ/kg — Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g
0.01	0.6117	0.001000	206.00	0.000	2374.9	2374.9	0.001	2500.9	2500.9
5	0.8725	0.001000	147.03	21.019	2360.8	2381.8	21.020	2489.1	2510.1
10	1.2281	0.001000	106.32	42.020	2346.6	2388.7	42.022	2477.2	2519.2
15	1.7057	0.001001	77.885	62.980	2332.5	2395.5	62.982	2465.4	2528.3
20	2.3392	0.001002	57.762	83.913	2318.4	2402.3	83.915	2453.5	2537.4
25	3.1698	0.001003	43.340	104.83	2304.3	2409.1	104.83	2441.7	2546.5

Interpolation at T=22C required



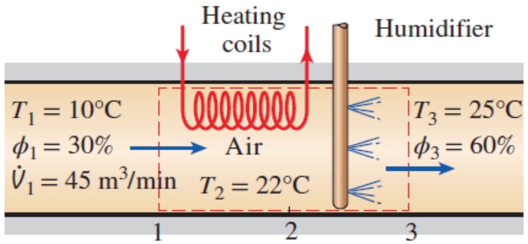


FIGURE 14–24
Schematic and psychrometric chart for Example 14–6.

14–7 Air-conditioning Processes

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

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

$$\upsilon_1 = \frac{R_a T_1}{P_a} = \frac{(0.287\ \text{kPa·m}^3/\text{kg·K})\boxed{(283\ \text{K})}}{99.632\ \text{kPa}} = 0.815\ \text{m}^3/\text{kg dry air}$$

$$\dot{m}_a = \frac{\dot{V}_1}{\upsilon_1} = \frac{45\ \text{m}^3/\text{min}}{0.815\ \text{m}^3/\text{kg}} = 55.2\ \text{kg/min}$$

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

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

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

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,

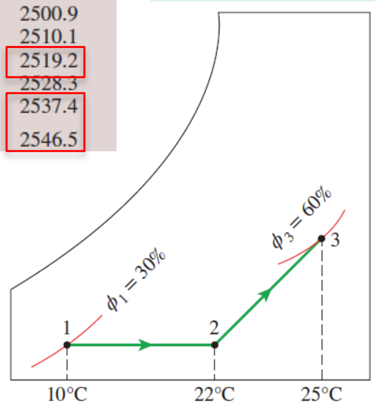
$$\dot{m}_w = (55.2\ \text{kg/min})(0.01206 - 0.0023)$$
$$= \mathbf{0.539\ kg/min}$$

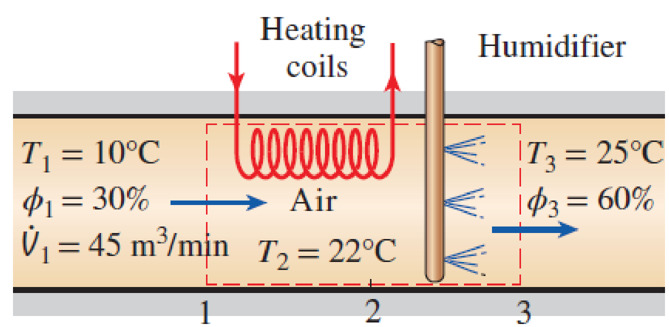
Discussion The result 0.539 kg/min corresponds to a water requirement of close to one ton a day, which is significant.

TABLE A–4

Saturated water—Temperature table

		Specific volume, m³/kg		Internal energy, kJ/kg			Enthalpy, kJ/kg		
Temp., T °C	Sat. Press., P_{sat} kPa	Sat. liquid, υ_f	Sat. vapor, υ_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g
0.01	0.6117	0.001000	206.00	0.000	2374.9	2374.9	0.001	2500.9	2500.9
5	0.8725	0.001000	147.03	21.019	2360.8	2381.8	21.020	2489.1	2510.1
1 10	1.2281	0.001000	106.32	42.020	2346.6	2388.7	42.022	2477.2	2519.2
15	1.7057	0.001001	77.885	62.980	2332.5	2395.5	62.982	2465.4	2528.3
20	2.3392	0.001002	57.762	83.913	2318.4	2402.3	83.915	2453.5	2537.4
2 25	3.1698	0.001003	43.340	104.83	2304.3	2409.1	104.83	2441.7	2546.5



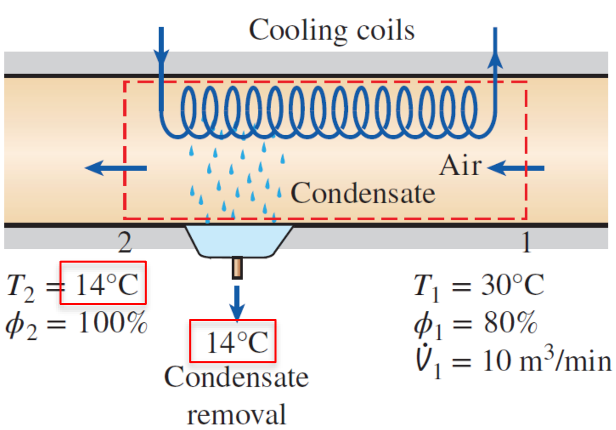


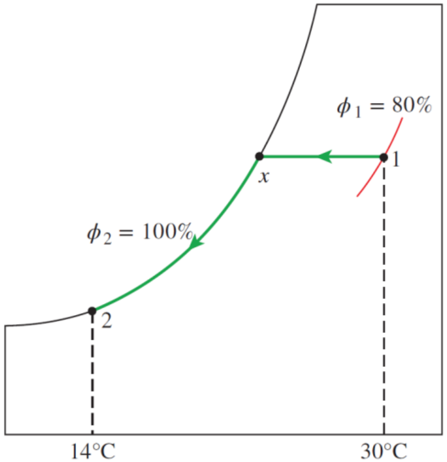
14–7 Air-conditioning Processes 7

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.

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.





14–7 Air-conditioning Processes 7

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

$$h_1 = 85.4 \text{ kJ/kg dry air} \qquad h_2 = 39.3 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0216 \text{ kg } H_2O\text{/kg dry air} \qquad \omega_2 = 0.0100 \text{ kg } H_2O\text{/kg dry air}$$
$$\upsilon_1 = 0.889 \text{ m}^3\text{/kg dry air}$$

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: $\dot m_{a_1}\omega_1 = \dot m_{a_2}\omega_2 + \dot m_w \rightarrow \boxed{\dot m_w = \dot m_a(\omega_1 - \omega_2)}$

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,

$$\boxed{\dot m_a = \frac{\dot V_1}{\upsilon_1}} = \frac{10 \text{ m}^3\text{/min}}{0.889 \text{ m}^3\text{/kg dry air}} = 11.25 \text{ kg/min}$$

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

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

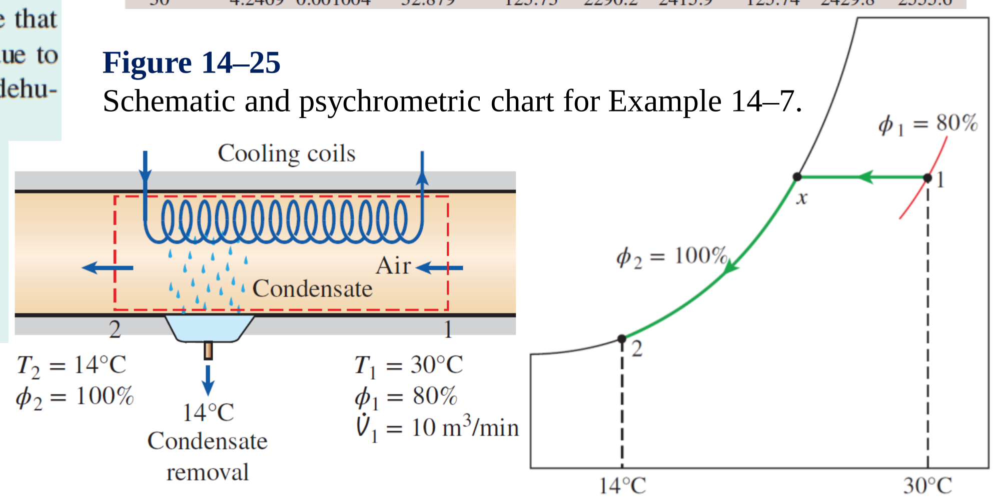
Therefore, this air-conditioning unit removes moisture and heat from the air at rates of 0.131 kg/min and 511 kJ/min, respectively.

TABLE A–4

Saturated water—Temperature table

Temp., T °C	Sat. Press., P_{sat} kPa	Specific volume, m³/kg: Sat. liquid, υ_f	Sat. vapor, υ_g	Internal energy, kJ/kg: Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Enthalpy, kJ/kg: Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g
0.01	0.6117	0.001000	206.00	0.000	2374.9	2374.9	0.001	2500.9	2500.9
5	0.8725	0.001000	147.03	21.019	2360.8	2381.8	21.020	2489.1	2510.1
10	1.2281	0.001000	106.32	42.020	2346.6	2388.7	42.022	2477.2	2519.2
15	1.7057	0.001001	77.885	62.980	2332.5	2395.5	62.982	2465.4	2528.3
20	2.3392	0.001002	57.762	83.913	2318.4	2402.3	83.915	2453.5	2537.4
25	3.1698	0.001003	43.340	104.83	2304.3	2409.1	104.83	2441.7	2546.5
30	4.2469	0.001004	32.879	125.73	2290.2	2415.9	125.74	2429.8	2555.6

Figure 14–25
Schematic and psychrometric chart for Example 14–7.



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Evaporative Cooling

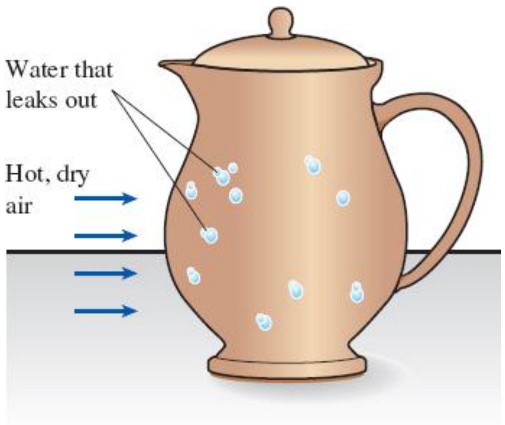
Conventional cooling systems operate on a refrigeration cycle, and they can be used in any part of the world. But they have a high initial and operating cost.

In desert (*hot and dry*) climates, we can avoid the high cost of 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 14–26

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



Evaporative hybrid coolers – Received from Jaeggi.

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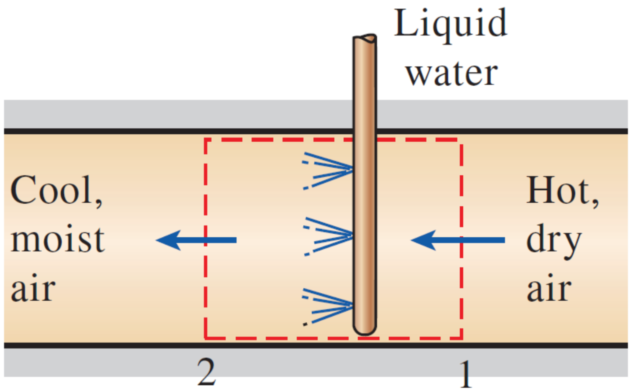
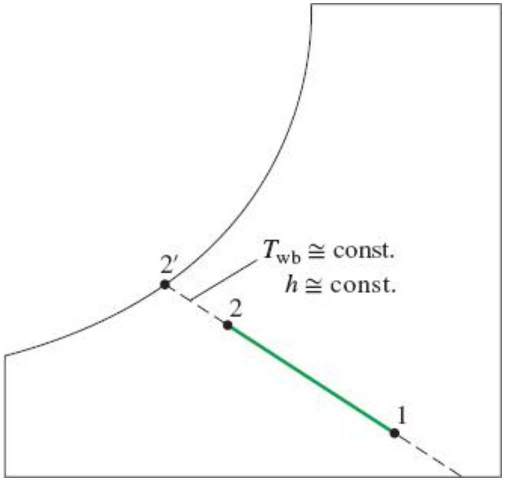
Evaporative cooling process is essentially identical to adiabatic saturation process. Since the constant-wet bulb-temperature (T_{wb}) lines almost coincide with the constant-enthalpy lines, the enthalpy of the airstream can also be assumed to remain constant. That is



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

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

Figure 14–27
Evaporative cooling.



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 follows a line of constant wet-bulb temperature on the psychrometric chart.

14–7 Air-conditioning Processes [10]

EXAMPLE 14–8 **Evaporative Cooling with Soaked Head Cover**

Desert dwellers often wrap their heads with a water-soaked porous cloth (Fig. 14–28). On a desert where the pressure is 1 atm, temperature is 120°F, and relative humidity is 10 percent, what is the temperature of this cloth?

SOLUTION Desert dwellers often wrap their heads with a water-soaked porous cloth. The temperature of this cloth on a desert with a specified temperature and relative humidity is to be determined.

Assumptions Air leaves the head covering as saturated.

Analysis Since the cloth behaves like the wick on a wet-bulb thermometer, the temperature of the cloth will become the wet-bulb temperature. 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 1 atm, 120°F, and 10 percent relative humidity is determined from the psychrometric chart to be

$$T_2 = T_{\text{wb}} = \mathbf{73.7°F}$$

Discussion Note that 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. Also, note that the temperature of air drops by as much as 46°F in this case by evaporative cooling.

Figure 14–28

Head wrap discussed in Example 14-8.

14–7 Air-conditioning Processes [11]

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.

The heat transfer with the surroundings is usually small, and thus the mixing processes can be assumed to be adiabatic

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

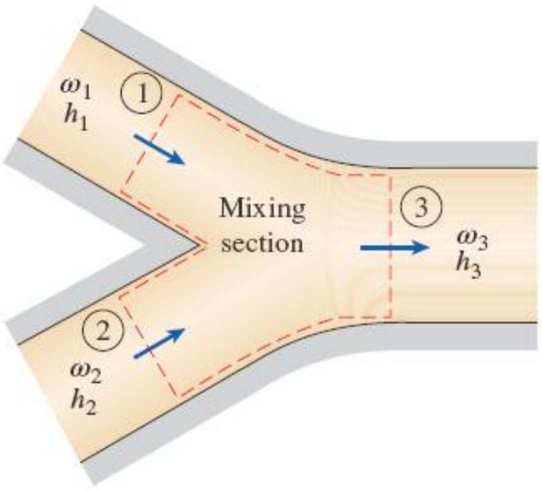
Mass of water vapour: $\omega_1 \dot{m}_{a_1} + \omega_2 \dot{m}_{a_2} = \omega_3 \dot{m}_{a_3}$

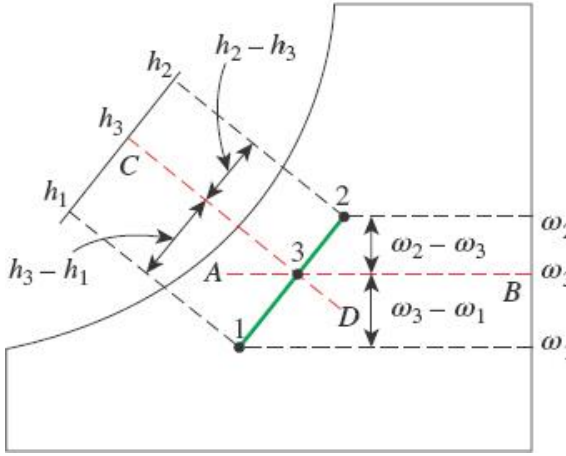
Energy $\dot{m}_{a_1} h_1 + \dot{m}_{a_2} h_2 = \dot{m}_{a_3} h_3$

$$\frac{\dot{m}_{a_1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

By eliminating $\dot{m}_{a_3}$

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.





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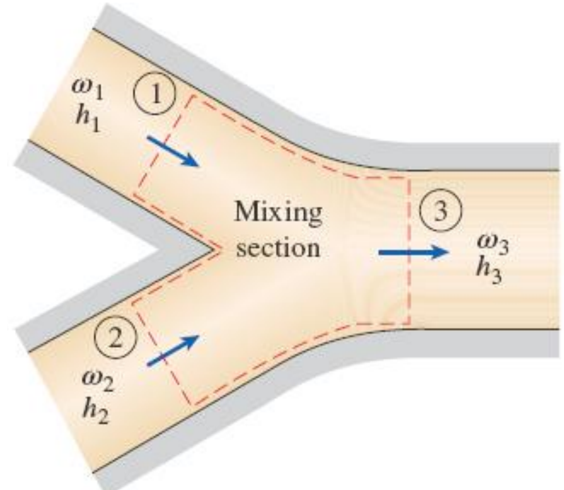
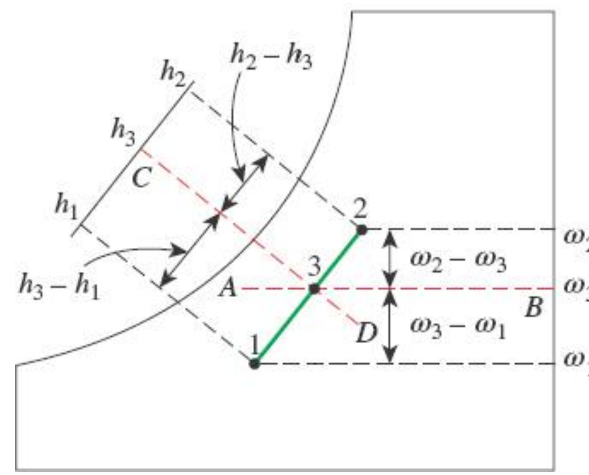
$$\frac{\dot m_{a_1}}{\dot m_{a2}} = \frac{\omega_2-\omega_3}{\omega_3-\omega_1} = \frac{h_2-h_3}{h_3-h_1}$$



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.



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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.

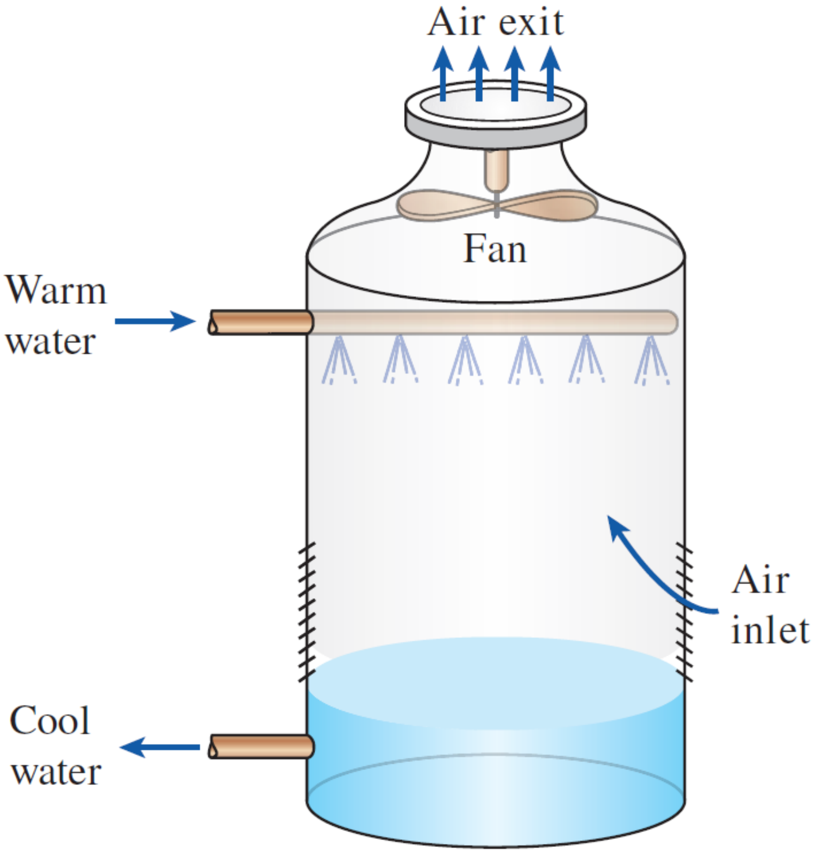
In such cases, the waste heat must be rejected to the atmosphere, with cooling water recirculating and serving as a transport medium for heat transfer between the source and the sink (the atmosphere).

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.



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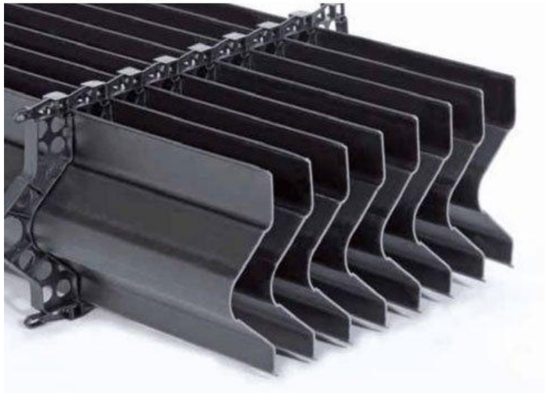
An induced-draft counterflow wet cooling tower is shown schematically in Fig. 14–31. Air is drawn into the tower from the bottom and leaves through the top. Warm water from the condenser is pumped to the top of the tower and is sprayed into this airstream. The purpose of spraying is to expose a large 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.

A typical drift eliminator



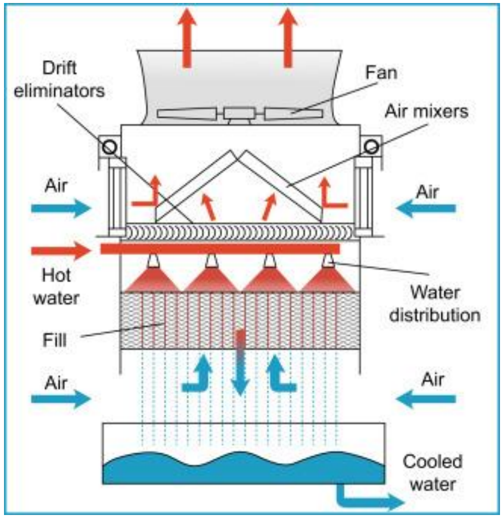


Figure 14–31

An induced-draft counterflow cooling tower also known as forced-draft cooling tower.

MANCHESTER 1824
The University of Manchester

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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.

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 hyperbolic profile is for greater structural strength, not for any thermodynamic reason.

Spray pond: The warm water is sprayed into the air and is cooled by the air as it falls into the pond. This is what started the idea of a cooling tower.

Figure 14–32

Two natural draft cooling towers on a roadside.

Figure 14–33

A spray pond.

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Cooling pond: Dumping the waste heat into a still pond, which is basically a large artificial lake open to the atmosphere. However, we would need about 20 times the area of a spray pond in this case to achieve the same cooling.

Figure 14–34

A cooling pond.

14–7 Air-conditioning Processes

EXAMPLE 14–10 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,

$$h_1 = 42.2 \text{ kJ/kg dry air} \qquad h_2 = 100.0 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0087 \text{ kg H}_2\text{O/kg dry air} \qquad \omega_2 = 0.0273 \text{ kg H}_2\text{O/kg dry air}$$
$$\upsilon_1 = 0.842 \text{ m}^3\text{/kg dry air}$$

Analysis We take the entire *cooling tower* to be the system, which is shown schematically in Fig. 14–35. 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

$$\dot m_3 - \dot m_4 = \dot m_a(\omega_2 - \omega_1) = \dot m_{\text{makeup}}$$

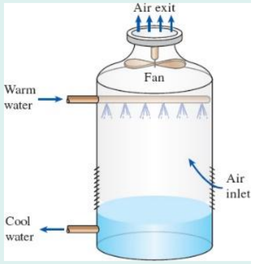
Energy balance: $\displaystyle\sum_{\text{in}} \dot m h = \sum_{\text{out}} \dot m h \rightarrow \dot m_{a_1}h_1 + \dot m_3 h_3 = \dot m_{a_2}h_2 + \dot m_4 h_4$

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

$$\dot V_1 = \dot m_a \upsilon_1 = (96.9 \text{ kg/s})(0.842 \text{ m}^3\text{/kg}) = \mathbf{81.6\ m^3/s}$$

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

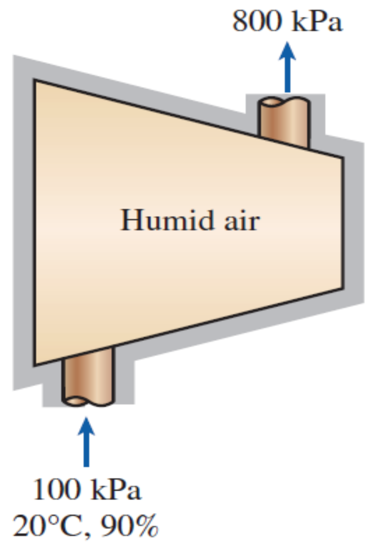
$$\dot m_{\text{makeup}} = \dot m_a(\omega_2 - \omega_1) = (96.9 \text{ kg/s})(0.0273 - 0.0087) = \mathbf{1.80\ kg/s}$$

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

Tutorial Questions

Tutorial Questions

1) Humid air at 100 kPa, 20°C, and 90 percent relative humidity is compressed in a steady-flow, isentropic compressor to 800 kPa. What is the relative humidity of the air at the compressor outlet?

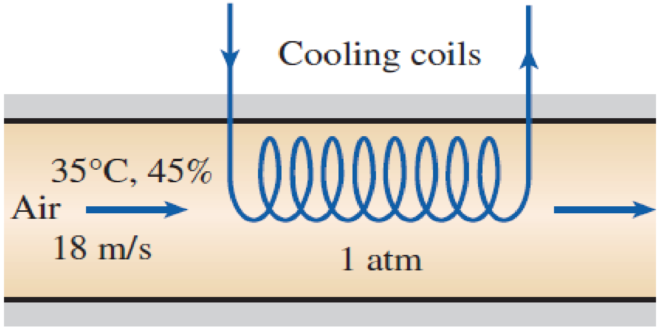


2) Atmospheric pressure is 98 kPa. Determine the relative humidity and specific humidity of the air.

Tutorial Questions

3) 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 35 percent relative humidity. The building is maintained at 24°C and 55 percent relative humidity at all times.

4) Air enters a 30-cm-diameter cooling section at 1 atm, 35°C, and 45 percent relative humidity at 18 m/s. Heat is removed from the air at a rate of 750 kJ/min. Determine (*a*) the exit temperature, (*b*) the exit relative humidity of the air, and (*c*) the exit velocity.



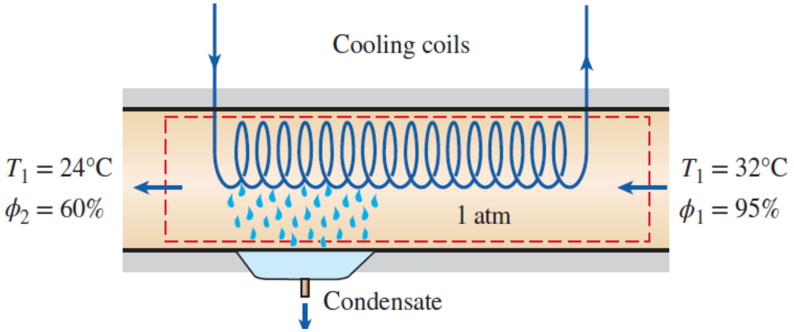
***Answers:* (*a*) 26.5°C, (*b*) 73.1 percent, (*c*) 17.5 m/s**

Tutorial Questions

5) 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**

6) On a summer day in New Orleans, Louisiana, the pressure is 1 atm: the temperature is 32°C; and the relative humidity is 95 percent. This air is to be conditioned to 24°C and 60 percent relative humidity. Determine the amount of cooling, in kJ, required and water removed, in kg, per 1000 m^3 of dry air processed at the entrance to the system.



Tutorial Questions

7) Air at 1 atm, 20°C, and 70 percent relative humidity is first heated to 35°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.

8) A stream of warm air with a dry-bulb temperature of 36°C and a wet-bulb temperature of 30°C is mixed adiabatically with a stream of saturated cool air at 12°C. The dry air mass flow rates of the warm and cool airstreams are 8 and 10 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.

End