Chapter 14
GAS-VAPOR MIXTURES AND AIR CONDITIONING

Dry and Atmospheric Air, Specific and Relative Humidity

14-1C Yes; by cooling the air at constant pressure.

14-2C Yes.

14-3C Specific humidity will decrease but relative humidity will increase.

14-4C Dry air does not contain any water vapor, but atmospheric air does.

14-5C Yes, the water vapor in the air can be treated as an ideal gas because of its very low partial pressure.

14-6C The partial pressure of the water vapor in atmospheric air is called vapor pressure.

14-7C The same. This is because water vapor behaves as an ideal gas at low pressures, and the enthalpy of an ideal gas depends on temperature only.

14-8C Specific humidity is the amount of water vapor present in a unit mass of dry air. Relative humidity is the ratio of the actual amount of vapor in the air at a given temperature to the maximum amount of vapor air can hold at that temperature.

14-9C The specific humidity will remain constant, but the relative humidity will decrease as the temperature rises in a well-sealed room.

14-10C The specific humidity will remain constant, but the relative humidity will decrease as the temperature drops in a well-sealed room.

14-11C A tank that contains moist air at 3 atm is located in moist air that is at 1 atm. The driving force for moisture transfer is the vapor pressure difference, and thus it is possible for the water vapor to flow into the tank from surroundings if the vapor pressure in the surroundings is greater than the vapor pressure in the tank.

14-12C Insulations on *chilled water lines* are always wrapped with *vapor barrier jackets* to eliminate the possibility of vapor entering the insulation. This is because moisture that migrates through the insulation to the cold surface will condense and remain there indefinitely with no possibility of vaporizing and moving back to the outside.

14-13C When the temperature, total pressure, and the relative humidity are given, the vapor pressure can be determined from the psychrometric chart or the relation $P_v = \phi P_{sat}$ where P_{sat} is the saturation (or boiling) pressure of water at the specified temperature and ϕ is the relative humidity.

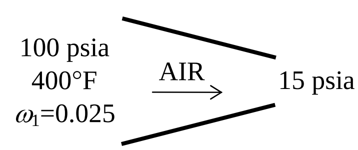
14-14E Humid air is expanded in an isentropic nozzle. The amount of water vapor that has condensed during the process is to be determined.

Assumptions The air and the water vapor are ideal gases.

Properties The specific heat ratio of air at room temperature is k = 1.4 (Table A-2a). The saturation properties of water are to be obtained from water tables.

Analysis Since the mole fraction of the water vapor in this mixture is very small,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (860\text{ R})\left(\frac{15\text{ psia}}{100\text{ psia}}\right)^{0.4/1.4} = 500\text{ R}$$



We will assume that the air leaves the nozzle at a relative humidity of 100% (will be verified later). The vapor pressure and specific humidity at the outlet are then

$$P_{v,2} = \phi_2 P_{g,2} = \phi_2 P_{\text{sat @ 40°F}} = (1.0)(0.12173\text{ psia}) = 0.1217\text{ psia}$$

$$\omega_2 = \frac{0.622 P_{v,2}}{P - P_{v,2}} = \frac{(0.622)(0.1217\text{ psia})}{(15 - 0.1217)\text{ psia}} = 0.00509\text{ lbm H}_2\text{O/lbm dry air}$$

This is less than the inlet specific humidity (0.025 lbm/lbm dry air), the relative humidity at the outlet must be 100% as originally assumed. The amount of liquid formation is then

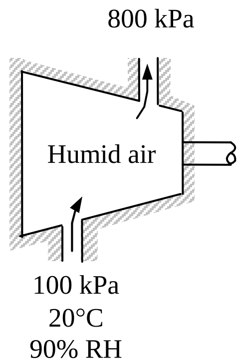
$$\Delta\omega = \omega_1 - \omega_2 = 0.025 - 0.00509 = \mathbf{0.0199\ lbm\ H_2O/lbm\ dry\ air}$$

14-15 Humid air is compressed in an isentropic compressor. The relative humidity of the air at the compressor outlet is to be determined.

Assumptions The air and the water vapor are ideal gases.

Properties The specific heat ratio of air at room temperature is k = 1.4 (Table A-2a). The saturation properties of water are to be obtained from water tables.

Analysis At the inlet,



$$P_{v,1} = \phi_1 P_{g,1} = \phi_1 P_{\text{sat @ 20°C}} = (0.90)(2.3392\text{ kPa}) = 2.105\text{ kPa}$$

$$\omega_2 = \omega_1 = \frac{0.622 P_{v,1}}{P - P_{v,1}} = \frac{(0.622)(2.105\text{ kPa})}{(100 - 2.105)\text{ kPa}} = 0.0134\text{ kg H}_2\text{O/kg dry air}$$

Since the mole fraction of the water vapor in this mixture is very small,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (293\text{ K})\left(\frac{800\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} = 531\text{ K}$$

The saturation pressure at this temperature is

$$P_{g,2} = P_{\text{sat @ 258°C}} = 4542\text{ kPa} \quad \text{(from EES)}$$

The vapor pressure at the exit is

$$P_{v,2} = \frac{\omega_2 P_2}{\omega_2 + 0.622} = \frac{(0.0134)(800)}{0.0134 + 0.622} = 16.87\text{ kPa}$$

The relative humidity at the exit is then

$$\phi_2 = \frac{P_{v,2}}{P_{g,2}} = \frac{16.87}{4542} = \mathbf{0.0037} = \mathbf{0.37\%}$$

14-16 A tank contains dry air and water vapor at specified conditions. The specific humidity, the relative humidity, and the volume of the tank are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The specific humidity can be determined form its definition,

$$\omega = \frac{m_v}{m_a} = \frac{0.3\,\text{kg}}{21\,\text{kg}} = \mathbf{0.0143\,kg\,H_2O/kg\,dry\,air}$$

21 kg dry air
0.3 kg H_2O vapor
30°C
100 kPa

(*b*) The saturation pressure of water at 30°C is

$$P_g = P_{\text{sat @ 30°C}} = 4.2469\,\text{kPa}$$

Then the relative humidity can be determined from

$$\phi = \frac{\omega P}{(0.622+\omega)P_g} = \frac{(0.0143)(100\,\text{kPa})}{(0.622+0.0143)(4.2469\,\text{kPa})} = \mathbf{52.9\%}$$

(*c*) The volume of the tank can be determined from the ideal gas relation for the dry air,

$$P_v = \phi P_g = (0.529)(4.2469\,\text{kPa}) = 2.245\,\text{kPa}$$

$$P_a = P - P_v = 100 - 2.245 = 97.755\,\text{kPa}$$

$$V = \frac{m_a R_a T}{P_a} = \frac{(21\,\text{kg})(0.287\,\text{kJ/kg}\cdot\text{K})(303\,\text{K})}{97.755\,\text{kPa}} = \mathbf{18.7\,m^3}$$

14-17 A tank contains dry air and water vapor at specified conditions. The specific humidity, the relative humidity, and the volume of the tank are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The specific humidity can be determined form its definition,

$$\omega = \frac{m_v}{m_a} = \frac{0.3\,\text{kg}}{21\,\text{kg}} = \mathbf{0.0143\,kg\,H_2O/kg\,dry\,air}$$

21 kg dry air
0.3 kg H_2O vapor
24°C
100 kPa

(*b*) The saturation pressure of water at 24°C is

$$P_g = P_{\text{sat @24°C}} = 2.986\,\text{kPa}$$

Then the relative humidity can be determined from

$$\phi = \frac{\omega P}{(0.622+\omega)P_g} = \frac{(0.0143)(100\,\text{kPa})}{(0.622+0.0143)2.986\,\text{kPa}} = \mathbf{75.2\%}$$

(*c*) The volume of the tank can be determined from the ideal gas relation for the dry air,

$$P_v = \phi P_g = (0.752)(2.986\,\text{kPa}) = 2.245\,\text{kPa}$$

$$P_a = P - P_v = 100 - 2.245 = 97.755\,\text{kPa}$$

$$V = \frac{m_a R_a T}{P_a} = \frac{(21\,\text{kg})(0.287\,\text{kJ/kg}\cdot\text{K})(297\,\text{K})}{97.755\,\text{kPa}} = \mathbf{18.3\,m^3}$$

14-18 A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

Assumptions The air and the water vapor are ideal gases.

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

AIR
20°C
98 kPa
85% RH

$$P_v = \phi P_g = \phi P_{\text{sat @ 20°C}} = (0.85)(2.3392\ \text{kPa}) = 1.988\ \text{kPa}$$
$$P_a = P - P_v = 98 - 1.988 = \mathbf{96.01\ kPa}$$

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(1.988\ \text{kPa})}{(98 - 1.988)\ \text{kPa}} = \mathbf{0.0129\ kg\ H_2O/kg\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$\begin{aligned} h &= h_a + \omega h_v \cong c_p T + \omega h_g \\ &= (1.005\ \text{kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0129)(2537.4\ \text{kJ/kg}) \\ &= \mathbf{52.78\ kJ/kg\ dry\ air} \end{aligned}$$

14-19 A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

Assumptions The air and the water vapor are ideal gases.

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

AIR
20°C
85 kPa
85% RH

$$P_v = \phi P_g = \phi P_{\text{sat @ 20°C}} = (0.85)(2.3392\ \text{kPa}) = 1.988\ \text{kPa}$$
$$P_a = P - P_v = 85 - 1.988 = \mathbf{83.01\ kPa}$$

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(1.988\ \text{kPa})}{(85 - 1.988)\ \text{kPa}} = \mathbf{0.0149\ kg\ H_2O/kg\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$\begin{aligned} h &= h_a + \omega h_v \cong c_p T + \omega h_g \\ &= (1.005\ \text{kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0149)(2537.4\ \text{kJ/kg}) \\ &= \mathbf{57.90\ kJ/kg\ dry\ air} \end{aligned}$$

14-20E A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

Assumptions The air and the water vapor are ideal gases.

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

AIR
70°F
14.6 psia
85% RH

$$P_v = \phi P_g = \phi P_{\text{sat @ 70°F}} = (0.85)(0.36334\text{ psia}) = 0.309\text{ psia}$$

$$P_a = P - P_v = 14.6 - 0.309 = \mathbf{14.291\ psia}$$

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622P_v}{P - P_v} = \frac{(0.622)(0.309\text{ psia})}{(14.6 - 0.309)\text{ psia}} = \mathbf{0.0134\ lbm\ H_2O/lbm\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$\begin{aligned} h &= h_a + \omega h_v \cong c_p T + \omega h_g \\ &= (0.24\text{ Btu/lbm}\cdot\text{°F})(70\text{°F}) + (0.0134)(1091.8\text{ Btu/lbm}) \\ &= \mathbf{31.43\ Btu/lbm\ dry\ air} \end{aligned}$$

14-21 The masses of dry air and the water vapor contained in a room at specified conditions and relative humidity are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis The partial pressure of water vapor and dry air are determined to be

$$P_v = \phi P_g = \phi P_{\text{sat @ 23°C}} = (0.50)(2.811\text{ kPa}) = 1.41\text{ kPa}$$

$$P_a = P - P_v = 98 - 1.41 = 96.59\text{ kPa}$$

ROOM
240 m^3
23°C
98 kPa
50% RH

The masses are determined to be

$$m_a = \frac{P_a \mathbf{V}}{R_a T} = \frac{(96.59\text{ kPa})(240\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(296\text{ K})} = \mathbf{272.9\ kg}$$

$$m_v = \frac{P_v \mathbf{V}}{R_v T} = \frac{(1.41\text{ kPa})(240\text{ m}^3)}{(0.4615\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(296\text{ K})} = \mathbf{2.47\ kg}$$

Dew-point, Adiabatic Saturation, and Wet-bulb Temperatures

14-22C Dew-point temperature is the temperature at which condensation begins when air is cooled at constant pressure.

14-23C Andy's. The temperature of his glasses may be below the dew-point temperature of the room, causing condensation on the surface of the glasses.

14-24C The outer surface temperature of the glass may drop below the dew-point temperature of the surrounding air, causing the moisture in the vicinity of the glass to condense. After a while, the condensate may start dripping down because of gravity.

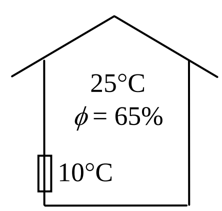
14-25C When the temperature falls below the dew-point temperature, dew forms on the outer surfaces of the car. If the temperature is below 0°C, the dew will freeze. At very low temperatures, the moisture in the air will freeze directly on the car windows.

14-26C When the air is saturated (100% relative humidity).

14-27C These two are approximately equal at atmospheric temperatures and pressure.

14-28 A house contains air at a specified temperature and relative humidity. It is to be determined whether any moisture will condense on the inner surfaces of the windows when the temperature of the window drops to a specified value.

Assumptions The air and the water vapor are ideal gases.



Analysis The vapor pressure P_v is uniform throughout the house, and its value can be determined from

$$P_v = \phi P_{g\,@\,25°\text{C}} = (0.65)(3.1698\ \text{kPa}) = 2.06\ \text{kPa}$$

The dew-point temperature of the air in the house is

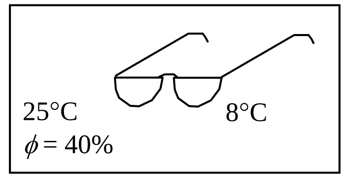
$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,2.06\ \text{kPa}} = \mathbf{18.0°C}$$

That is, the moisture in the house air will start condensing when the temperature drops below 18.0°C. Since the windows are at a lower temperature than the dew-point temperature, some moisture **will condense** on the window surfaces.

14-29 A person wearing glasses enters a warm room at a specified temperature and relative humidity from the cold outdoors. It is to be determined whether the glasses will get fogged.

Assumptions The air and the water vapor are ideal gases.

Analysis The vapor pressure P_v of the air in the house is uniform throughout, and its value can be determined from



$$P_v = \phi P_{g\,@\,25°\text{C}} = (0.40)(3.1698\text{ kPa}) = 1.268\text{ kPa}$$

The dew-point temperature of the air in the house is

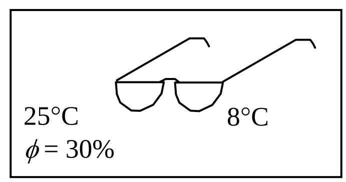
$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,1.268\text{ kPa}} = \mathbf{10.5°C} \quad \text{(from EES)}$$

That is, the moisture in the house air will start condensing when the air temperature drops below 10.5°C. Since the glasses are at a lower temperature than the dew-point temperature, some moisture will condense on the glasses, and thus they **will get fogged**.

14-30 A person wearing glasses enters a warm room at a specified temperature and relative humidity from the cold outdoors. It is to be determined whether the glasses will get fogged.

Assumptions The air and the water vapor are ideal gases.

Analysis The vapor pressure P_v of the air in the house is uniform throughout, and its value can be determined from



$$P_v = \phi P_{g\,@\,25°\text{C}} = (0.30)(3.1698\text{ kPa}) = 0.95\text{ kPa}$$

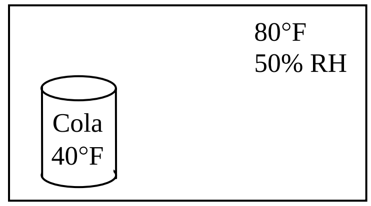
The dew-point temperature of the air in the house is

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,0.95\text{ kPa}} = \mathbf{6.2°C} \quad \text{(from EES)}$$

That is, the moisture in the house air will start condensing when the air temperature drops below 6.2°C. Since the glasses are at a higher temperature than the dew-point temperature, moisture will not condense on the glasses, and thus they **will not get fogged**.

14-31E A woman drinks a cool canned soda in a room at a specified temperature and relative humidity. It is to be determined whether the can will sweat.

Assumptions The air and the water vapor are ideal gases.



Analysis The vapor pressure P_v of the air in the house is uniform throughout, and its value can be determined from

$$P_v = \phi P_{g\,@\,80°\mathrm{F}} = (0.50)(0.50745\ \mathrm{psia}) = 0.254\ \mathrm{psia}$$

The dew-point temperature of the air in the house is

$$T_{\mathrm{dp}} = T_{\mathrm{sat}\,@\,P_v} = T_{\mathrm{sat}\,@\,0.254\ \mathrm{psia}} = \mathbf{59.7°F} \ \ \text{(from EES)}$$

That is, the moisture in the house air will start condensing when the air temperature drops below 59.7°C. Since the canned drink is at a lower temperature than the dew-point temperature, some moisture will condense on the can, and thus it **will sweat.**

14-32 The dry- and wet-bulb temperatures of atmospheric air at a specified pressure are given. The specific humidity, the relative humidity, and the enthalpy of air are to be determined.

Assumptions The air and the water vapor are ideal gases.

95 kPa
25°C
T_{wb} = 17°C

Analysis (*a*) We obtain the properties of water vapor from EES. The specific humidity ω_1 is determined from

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

where T_2 is the wet-bulb temperature, and ω_2 is determined from

$$\omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(1.938\ \mathrm{kPa})}{(95 - 1.938)\ \mathrm{kPa}} = 0.01295\ \mathrm{kg\ H_2O/kg\ dry\ air}$$

Thus,

$$\omega_1 = \frac{(1.005\ \mathrm{kJ/kg\cdot°C})(17-25)°\mathrm{C} + (0.01295)(2460.6\ \mathrm{kJ/kg})}{(2546.5 - 71.36)\ \mathrm{kJ/kg}} = \mathbf{0.00963\ kg\ H_2O/kg\ dry\ air}$$

(*b*) The relative humidity ϕ_1 is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622 + \omega_1) P_{g1}} = \frac{(0.00963)(95\ \mathrm{kPa})}{(0.622 + 0.00963)(3.1698\ \mathrm{kPa})} = 0.457 \ \text{ or } \ \mathbf{45.7\%}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h_1 = h_{a1} + \omega_1 h_{v1} \cong c_p T_1 + \omega_1 h_{g1} = (1.005\ \mathrm{kJ/kg\cdot°C})(25°\mathrm{C}) + (0.00963)(2546.5\ \mathrm{kJ/kg})$$
$$= \mathbf{49.65\ kJ/kg\ dry\ air}$$

14-33 The dry- and wet-bulb temperatures of air in room at a specified pressure are given. The specific humidity, the relative humidity, and the dew-point temperature are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) We obtain the properties of water vapor from EES. The specific humidity ω_1 is determined from

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

100 kPa
22°C
T_{wb} = 16°C

where T_2 is the wet-bulb temperature, and ω_2 is determined from

$$\omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(1.819\text{ kPa})}{(100 - 1.819)\text{ kPa}} = 0.01152\text{ kg H}_2\text{O/kg dry air}$$

Thus,

$$\omega_1 = \frac{(1.005\text{ kJ/kg}\cdot°\text{C})(16-22)°\text{C} + (0.01152)(2463.0\text{ kJ/kg})}{(2541.1 - 67.17)\text{ kJ/kg}} = \mathbf{0.00903\ kg\ H_2O/kg\ dry\ air}$$

(*b*) The relative humidity ϕ_1 is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622 + \omega_1)P_{g1}} = \frac{(0.00903)(100\text{ kPa})}{(0.622 + 0.0091)(2.6452\text{ kPa})} = 0.541 \quad \text{or} \quad \mathbf{54.1\%}$$

(*c*) The vapor pressure at the inlet conditions is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 22°C}} = (0.541)(2.6452\text{ kPa}) = 1.432\text{ kPa}$$

Thus the dew-point temperature of the air is

$$T_{dp} = T_{\text{sat @ }P_v} = T_{\text{sat @ 1.432 kPa}} = \mathbf{12.3°C}$$

14-34 EES Problem 14-33 is reconsidered. The required properties are to be determined using EES at 100 and 300 kPa pressures.

Analysis The problem is solved using EES, and the solution is given below.

```
Tdb=22 [C]
Twb=16 [C]
P1=100 [kPa]
P2=300 [kPa]

h1=enthalpy(AirH2O;T=Tdb;P=P1;B=Twb)
v1=volume(AirH2O;T=Tdb;P=P1;B=Twb)
Tdp1=dewpoint(AirH2O;T=Tdb;P=P1;B=Twb)
w1=humrat(AirH2O;T=Tdb;P=P1;B=Twb)
Rh1=relhum(AirH2O;T=Tdb;P=P1;B=Twb)

h2=enthalpy(AirH2O;T=Tdb;P=P2;B=Twb)
v2=volume(AirH2O;T=Tdb;P=P2;B=Twb)
Tdp2=dewpoint(AirH2O;T=Tdb;P=P2;B=Twb)
w2=humrat(AirH2O;T=Tdb;P=P2;B=Twb)
Rh2=relhum(AirH2O;T=Tdb;P=P2;B=Twb)
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SOLUTION

```
h1=45.09 [kJ/kga]
h2=25.54 [kJ/kga]
P1=100 [kPa]
P2=300 [kPa]
Rh1=0.541
Rh2=0.243
Tdb=22 [C]
Tdp1=12.3 [C]
Tdp2=0.6964 [C]
Twb=16 [C]
v1=0.8595 [m^3/kga]
v2=0.283 [m^3/kga]
w1=0.009029 [kgv/kga]
w2=0.001336 [kgv/kga]
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14-35E The dry- and wet-bulb temperatures of air in room at a specified pressure are given. The specific humidity, the relative humidity, and the dew-point temperature are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The specific humidity ω_1 is determined from

14.7 psia
80°F
T_{wb} = 65°F

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

where T_2 is the wet-bulb temperature, and ω_2 is determined from

$$\omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(0.30578\text{ psia})}{(14.7 - 0.30578)\text{ psia}} = 0.01321\text{ lbm H}_2\text{O/lbm dry air}$$

Thus,

$$\omega_1 = \frac{(0.24\text{ Btu/lbm}\cdot°\text{F})(65 - 80)°\text{F} + (0.01321)(1056.5\text{ Btu/lbm})}{(1096.1 - 33.08)\text{ Btu/lbm}} = \mathbf{0.00974\ lbm\ H_2O/lbm\ dry\ air}$$

(*b*) The relative humidity ϕ_1 is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622 + \omega_1)P_{g1}} = \frac{(0.00974)(14.7\text{ psia})}{(0.622 + 0.00974)(0.50745\text{ psia})} = 0.447 \quad \text{or} \quad \mathbf{44.7\%}$$

(*c*) The vapor pressure at the inlet conditions is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 70°F}} = (0.447)(0.50745\text{ psia}) = 0.2268\text{ psia}$$

Thus the dew-point temperature of the air is

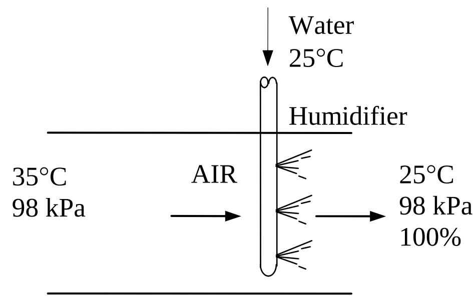
$$T_{dp} = T_{\text{sat @ }P_v} = T_{\text{sat @ 0.2268 psia}} = \mathbf{56.6°F} \qquad \text{(from EES)}$$

14-36 Atmospheric air flows steadily into an adiabatic saturation device and leaves as a saturated vapor. The relative humidity and specific humidity of air 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The exit state of the air is completely specified, and the total pressure is 98 kPa. The properties of the moist air at the exit state may be determined from EES to be

$$h_2 = 78.11\ \text{kJ/kg dry air}$$
$$\omega_2 = 0.02079\ \text{kg H}_2\text{O/kg dry air}$$



The enthalpy of makeup water is

$$h_{w2} = h_{f@\,25°\text{C}} = 104.83\ \text{kJ/kg} \quad \text{(Table A-4)}$$

An energy balance on the control volume gives

$$h_1 + (\omega_2 - \omega_1)h_w = h_2$$
$$h_1 + (0.02079 - \omega_1)(104.83\ \text{kJ/kg}) = 78.11\ \text{kJ/kg}$$

Pressure and temperature are known for inlet air. Other properties may be determined from this equation using EES. A hand solution would require a trial-error approach. The results are

$$h_1 = 77.66\ \text{kJ/kg dry air}$$
$$\omega_1 = \mathbf{0.01654\ kg\ H_2O/kg\ dry\ air}$$
$$\phi_1 = \mathbf{0.4511}$$

Psychrometric Chart

14-37C They are very nearly parallel to each other.

14-38C The saturation states (located on the saturation curve).

14-39C By drawing a horizontal line until it intersects with the saturation curve. The corresponding temperature is the dew-point temperature.

14-40C No, they cannot. The enthalpy of moist air depends on ω, which depends on the total pressure.

14-41 [*Also solved by EES on enclosed CD***]** The pressure, temperature, and relative humidity of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the wet-bulb temperature, the dew-point temperature, and the specific volume of the air are to be determined.

Analysis From the psychrometric chart (Fig. A-31) we read

(*a*) $\omega = 0.0181\,\text{kg H}_2\text{O / kg dry air}$

(*b*) $h = 78.4\ \text{kJ / kg dry air}$

(*c*) $T_{\text{wb}} = 25.5°\text{C}$

(*d*) $T_{\text{dp}} = 23.3°\text{C}$

(*e*) $\boldsymbol{v} = 0.890\,\text{m}^3\text{ / kg dry air}$

14-42 The pressure, temperature, and relative humidity of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the wet-bulb temperature, the dew-point temperature, and the specific volume of the air are to be determined.

Analysis From the psychrometric chart (Fig. A-31) we read

(*a*) $\omega = 0.0148\,\text{kg H}_2\text{O / kg dry air}$

(*b*) $h = 63.9\ \text{kJ / kg dry air}$

(*c*) $T_{\text{wb}} = 21.9°\text{C}$

(*d*) $T_{\text{dp}} = 20.1°\text{C}$

(*e*) $\boldsymbol{v} = 0.868\,\text{m}^3\text{ / kg dry air}$

14-43 EES Problem 14-42 is reconsidered. The required properties are to be determined using EES. Also, the properties are to be obtained at an altitude of 2000 m.

Analysis The problem is solved using EES, and the solution is given below.

```
Tdb=26 [C]
Rh=0.70
P1=101.325 [kPa]
Z = 2000 [m]
P2=101.325*(1-0.02256*Z*convert(m,km))^5.256  "Relation giving P as a function of altitude"

      h1=enthalpy(AirH2O,T=Tdb,P=P1,R=Rh)
      v1=volume(AirH2O,T=Tdb,P=P1,R=Rh)
      Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,R=Rh)
      w1=humrat(AirH2O,T=Tdb,P=P1,R=Rh)
      Twb1=wetbulb(AirH2O,T=Tdb,P=P1,R=Rh)

      h2=enthalpy(AirH2O,T=Tdb,P=P2,R=Rh)
      v2=volume(AirH2O,T=Tdb,P=P2,R=Rh)
      Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,R=Rh)
      w2=humrat(AirH2O,T=Tdb,P=P2,R=Rh)
      Twb2=wetbulb(AirH2O,T=Tdb,P=P2,R=Rh)
```

SOLUTION

```
h1=63.88 [kJ/kg]
h2=74.55 [kJ/kg]
P1=101.3 [kPa]
P2=79.49 [kPa]
Rh=0.7
Tdb=26 [C]
Tdp1=20.11 [C]
Tdp2=20.11 [C]
Twb1=21.87 [C]
Twb2=21.59 [C]
v1=0.8676 [m^3/kg]
v2=1.113 [m^3/kg]
w1=0.0148 [kg/kg]
w2=0.01899 [kg/kg]
Z=2000 [m]
```

14-44 The pressure and the dry- and wet-bulb temperatures of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the relative humidity, the dew-point temperature, and the specific volume of the air are to be determined.

Analysis From the psychrometric chart (Fig. A-31) we read

(*a*) $\omega = 0.0092 \text{ kg H}_2\text{O / kg dry air}$

(*b*) $h = 47.6 \text{ kJ / kg dry air}$

(*c*) $\phi = 49.6\%$

(*d*) $T_{\text{dp}} = 12.8°\text{C}$

(*e*) $\boldsymbol{v} = 0.855 \text{ m}^3 \text{/ kg dry air}$

14-45 EES Problem 14-44 is reconsidered. The required properties are to be determined using EES. Also, the properties are to be obtained at an altitude of 3000 m.

Analysis The problem is solved using EES, and the solution is given below.

```
Tdb=24 [C]
Twb=17 [C]
P1=101.325 [kPa]
Z = 3000 [m]
P2=101.325*(1-0.02256*Z*convert(m,km))^5.256  "Relation giving P as function of altitude"

     h1=enthalpy(AirH2O,T=Tdb,P=P1,B=Twb)
     v1=volume(AirH2O,T=Tdb,P=P1,B=Twb)
     Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,B=Twb)
     w1=humrat(AirH2O,T=Tdb,P=P1,B=Twb)
     Rh1=relhum(AirH2O,T=Tdb,P=P1,B=Twb)

     h2=enthalpy(AirH2O,T=Tdb,P=P2,B=Twb)
     v2=volume(AirH2O,T=Tdb,P=P2,B=Twb)
     Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,B=Twb)
     w2=humrat(AirH2O,T=Tdb,P=P2,B=Twb)
     Rh2=relhum(AirH2O,T=Tdb,P=P2,B=Twb)
```

SOLUTION

h1=47.61 [kJ/kg]	h2=61.68 [kJ/kg]
P1=101.3 [kPa]	P2=70.11 [kPa]
Rh1=0.4956	Rh2=0.5438
Tdb=24 [C]	Tdp1=12.81 [C]
Tdp2=14.24 [C]	Twb=17 [C]
v1=0.8542 [m^3/kg]	v2=1.245 [m^3/kg]
w1=0.009219 [kg/kg]	w2=0.01475 [kg/kg]
Z=3000 [m]	

14-46 The pressure, temperature, and relative humidity of air are specified. Using the psychrometric chart, the wet-bulb temperature, specific humidity, the enthalpy, the dew-point temperature, and the water vapor pressure are to be determined.

Analysis From the psychrometric chart in Fig. A-31 or using EES psychrometric functions we obtain

(*a*) $T_{\mathrm{wb}} = 27.1°\mathrm{C}$

(*b*) $\omega = 0.0217\ \mathrm{kg\ H_2O / kg\ dry\ air}$

(*c*) $h = 85.5\ \mathrm{kJ/kg\ dry\ air}$

(*d*) $T_{\mathrm{dp}} = 26.2°\mathrm{C}$

(*e*) $P_v = \phi P_g = \phi P_{\mathrm{sat\ @\ 30°C}} = (0.80)(4.2469\ \mathrm{kPa}) = 3.40\ \mathrm{kPa}$

Air
1 atm
30°C
80% RH

14-47E The pressure, temperature, and wet-bulb temperature of air are specified. Using the psychrometric chart, the relative humidity, specific humidity, the enthalpy, the dew-point temperature, and the water vapor pressure are to be determined.

Analysis From the psychrometric chart in Fig. A-31 or using EES psychrometric functions we obtain

(*a*) $\phi = 0.816 = 81.6\%$

(*b*) $\omega = 0.0252\ \mathrm{lbm\ H_2O / lbm\ dry\ air}$

(*c*) $h = 49.4\ \mathrm{Btu/lbm\ dry\ air}$

(*d*) $T_{\mathrm{dp}} = 83.7°\mathrm{F}$

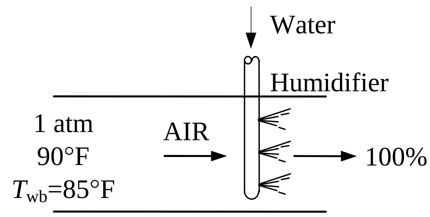
(*e*) $P_v = \phi P_g = \phi P_{\mathrm{sat\ @\ 90°F}} = (0.816)(0.69904\ \mathrm{psia}) = 0.570\ \mathrm{psia}$

Air
1 atm
90°F
T_{wb}=85°F

14-48E The pressure, temperature, and wet-bulb temperature of air are specified. The adiabatic saturation temperature is to be determined.

Analysis For an adiabatic saturation process, we obtained Eq. 14-14 in the text,

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$



This requires a trial-error solution for the adiabatic saturation temperature, T_2. The inlet state properties are

$$\omega_1 = 0.0252 \text{ lbm H}_2\text{O / lbm dry air}$$

$$h_{g1} = h_{g\,@\,90°\text{F}} = 1100.4 \text{ Btu/lbm}$$

As a first estimate, let us take T_2 =85°F (the inlet wet-bulb temperature). Also, at the exit, the relative humidity is 100% ($\phi_2 = 1$) and the pressure is 1 atm. Other properties at the exit state are

$$\omega_2 = 0.0264 \text{ lbm H}_2\text{O / lbm dry air}$$

$$h_{f2} = h_{f\,@\,85°\text{F}} = 53.06 \text{ Btu/lbm} \quad \text{(Table A-4E)}$$

$$h_{fg2} = h_{fg\,@\,85°\text{F}} = 1045.2 \text{ Btu/lbm} \quad \text{(Table A-4E)}$$

Substituting,

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}} = \frac{(0.240)(85-90) + (0.0264)(1045.2)}{1100.4 - 53.06} = 0.0252 \text{ lbm H}_2\text{O / lbm dry air}$$

which is equal to the inlet specific humidity. Therefore, the adiabatic saturation temperature is

T_2 = **85°F**

Discussion This result is not surprising since the wet-bulb and adiabatic saturation temperatures are approximately equal to each other for air-water mixtures at atmospheric pressure.

14-49 The pressure, temperature, and wet-bulb temperature of air are specified. Using the psychrometric chart, the relative humidity, specific humidity, the enthalpy, the dew-point temperature, and the water vapor pressure are to be determined.

Analysis From the psychrometric chart in Fig. A-31 or using EES psychrometric functions we obtain

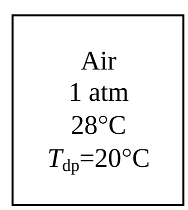
(*a*) $\phi = 0.618 = 61.8\%$

(*b*) $\omega = 0.0148 \text{ kg H}_2\text{O / kg dry air}$

(*c*) $h = 65.8 \text{ kJ/kg dry air}$

(*d*) $T_{wb} = 22.4°\text{C}$

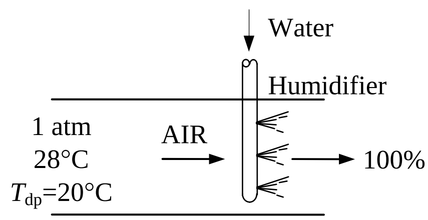
(*e*) $P_v = \phi P_g = \phi P_{\text{sat}\,@\,28°\text{C}} = (0.618)(3.780 \text{ kPa}) = 2.34 \text{ kPa}$



14-50 The pressure, temperature, and wet-bulb temperature of air are specified. The adiabatic saturation temperature is to be determined.

Analysis For an adiabatic saturation process, we obtained Eq. 14-14 in the text,

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$



This requires a trial-error solution for the adiabatic saturation temperature, T_2. The inlet state properties are

$$\omega_1 = 0.0148 \text{ kg H}_2\text{O / kg dry air} \quad \text{(Fig. A-31)}$$

$$h_{g1} = h_{g\,@\,28°\text{C}} = 2551.9 \text{ kJ/kg} \quad \text{(Table A-4)}$$

As a first estimate, let us take T_2 =22°C (the inlet wet-bulb temperature). Also, at the exit, the relative humidity is 100% ($\phi_2 = 1$) and the pressure is 1 atm. Other properties at the exit state are

$$\omega_2 = 0.0167 \text{ kg H}_2\text{O / kg dry air}$$

$$h_{f2} = h_{f\,@\,22°\text{C}} = 92.28 \text{ kJ/kg} \quad \text{(Table A - 4)}$$
$$h_{fg2} = h_{fg\,@\,22°\text{C}} = 2448.8 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

Substituting,

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}} = \frac{(1.005)(22-28) + (0.0167)(2448.8)}{2551.9 - 92.28} = 0.0142 \text{ kg H}_2\text{O / kg dry air}$$

which is sufficiently close to the inlet specific humidity (0.0148). Therefore, the adiabatic saturation temperature is

$$T_2 \cong \mathbf{22°C}$$

Discussion This result is not surprising since the wet-bulb and adiabatic saturation temperatures are approximately equal to each other for air-water mixtures at atmospheric pressure.

Human Comfort and Air-Conditioning

14-51C It humidifies, dehumidifies, cleans and even deodorizes the air.

14-52C (*a*) Perspires more, (*b*) cuts the blood circulation near the skin, and (*c*) sweats excessively.

14-53C It is the direct heat exchange between the body and the surrounding surfaces. It can make a person feel chilly in winter, and hot in summer.

14-54C It affects by removing the warm, moist air that builds up around the body and replacing it with fresh air.

14-55C The spectators. Because they have a lower level of activity, and thus a lower level of heat generation within their bodies.

14-56C Because they have a large skin area to volume ratio. That is, they have a smaller volume to generate heat but a larger area to lose it from.

14-57C It affects a body's ability to perspire, and thus the amount of heat a body can dissipate through evaporation.

14-58C Humidification is to add moisture into an environment, dehumidification is to remove it.

14-59C The metabolism refers to the burning of foods such as carbohydrates, fat, and protein in order to perform the necessary bodily functions. The metabolic rate for an average man ranges from 108 W while reading, writing, typing, or listening to a lecture in a classroom in a seated position to 1250 W at age 20 (730 at age 70) during strenuous exercise. The corresponding rates for women are about 30 percent lower. Maximum metabolic rates of trained athletes can exceed 2000 W. We are interested in metabolic rate of the occupants of a building when we deal with heating and air conditioning because the metabolic rate represents the rate at which a body generates heat and dissipates it to the room. This body heat contributes to the heating in winter, but it adds to the cooling load of the building in summer.

14-60C The metabolic rate is proportional to the size of the body, and the metabolic rate of women, in general, is lower than that of men because of their smaller size. Clothing serves as insulation, and the thicker the clothing, the lower the environmental temperature that feels comfortable.

14-61C Sensible heat is the energy associated with a temperature change. The sensible heat loss from a human body increases as (*a*) the skin temperature increases, (*b*) the environment temperature decreases, and (*c*) the air motion (and thus the convection heat transfer coefficient) increases.

14-62C Latent heat is the energy released as water vapor condenses on cold surfaces, or the energy absorbed from a warm surface as liquid water evaporates. The latent heat loss from a human body increases as (*a*) the skin wetness increases and (*b*) the relative humidity of the environment decreases. The rate of evaporation from the body is related to the rate of latent heat loss by $\dot{Q}_{\text{latent}} = \dot{m}_{\text{vapor}} h_{fg}$ where h_{fg} is the latent heat of vaporization of water at the skin temperature.

14-63 An average person produces 0.25 kg of moisture while taking a shower. The contribution of showers of a family of four to the latent heat load of the air-conditioner per day is to be determined.

Assumptions All the water vapor from the shower is condensed by the air-conditioning system.

Properties The latent heat of vaporization of water is given to be 2450 kJ/kg.

Analysis The amount of moisture produced per day is

$$\begin{aligned}\dot{m}_{\text{vapor}} &= (\text{Moisture produced per person})(\text{No. of persons})\\ &= (0.25\text{ kg / person})(4\text{ persons / day}) = 1\text{ kg / day}\end{aligned}$$

Then the latent heat load due to showers becomes

$$\dot{Q}_{\text{latent}} = \dot{m}_{\text{vapor}} h_{fg} = (1\text{ kg / day})(2450\text{ kJ / kg}) = \mathbf{2450\text{ kJ / day}}$$

14-64 There are 100 chickens in a breeding room. The rate of total heat generation and the rate of moisture production in the room are to be determined.

Assumptions All the moisture from the chickens is condensed by the air-conditioning system.

Properties The latent heat of vaporization of water is given to be 2430 kJ/kg. The average metabolic rate of chicken during normal activity is 10.2 W (3.78 W sensible and 6.42 W latent).

Analysis The total rate of heat generation of the chickens in the breeding room is

$$\dot{Q}_{\text{gen, total}} = \dot{q}_{\text{gen, total}}(\text{No. of chickens}) = (10.2\text{ W / chicken})(100\text{ chickens}) = \mathbf{1020\text{ W}}$$

The latent heat generated by the chicken and the rate of moisture production are

$$\begin{aligned}\dot{Q}_{\text{gen, latent}} &= \dot{q}_{\text{gen, latent}}(\text{No. of chickens})\\ &= (6.42\text{ W/chicken})(100\text{ chickens}) = 642\text{ W}\\ &= 0.642\text{ kW}\end{aligned}$$

$$\dot{m}_{\text{moisture}} = \frac{\dot{Q}_{\text{gen, latent}}}{h_{fg}} = \frac{0.642\text{ kJ / s}}{2430\text{ kJ / kg}} = 0.000264\text{ kg / s} = \mathbf{0.264\text{ g / s}}$$

14-65 A department store expects to have a specified number of people at peak times in summer. The contribution of people to the sensible, latent, and total cooling load of the store is to be determined.

Assumptions There is a mix of men, women, and children in the classroom.

Properties The average rate of heat generation from people doing light work is 115 W, and 70% of is in sensible form (see Sec. 14-6).

Analysis The contribution of people to the sensible, latent, and total cooling load of the store are

$$\dot{Q}_{\text{people, total}} = (\text{No. of people}) \times \dot{Q}_{\text{person, total}} = 135 \times (115\text{ W}) = \mathbf{15{,}525\ W}$$

$$\dot{Q}_{\text{people, sensible}} = (\text{No. of people}) \times \dot{Q}_{\text{person, sensible}} = 135 \times (0.7 \times 115\text{ W}) = \mathbf{10{,}868\ W}$$

$$\dot{Q}_{\text{people, latent}} = (\text{No. of people}) \times \dot{Q}_{\text{person, latent}} = 135 \times (0.3 \times 115\text{ W}) = \mathbf{4658\ W}$$

14-66E There are a specified number of people in a movie theater in winter. It is to be determined if the theater needs to be heated or cooled.

Assumptions There is a mix of men, women, and children in the classroom.

Properties The average rate of heat generation from people in a movie theater is 105 W, and 70 W of it is in sensible form and 35 W in latent form.

Analysis Noting that only the sensible heat from a person contributes to the heating load of a building, the contribution of people to the heating of the building is

$$\dot{Q}_{\text{people,sensible}} = (\text{No. of people}) \times \dot{Q}_{\text{person,sensible}} = 500 \times (70\text{ W}) = 35{,}000\text{ W} = \mathbf{119{,}420\ Btu/h}$$

since 1 W = 3.412 Btu/h. The building needs to be heated since the heat gain from people is less than the rate of heat loss of 130,000 Btu/h from the building.

14-67 The infiltration rate of a building is estimated to be 1.2 ACH. The sensible, latent, and total infiltration heat loads of the building at sea level are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air infiltrates at the outdoor conditions, and exfiltrates at the indoor conditions. **3** Excess moisture condenses at room temperature of 24°C. **4** The effect of water vapor on air density is negligible.

Properties The gas constant and the specific heat of air are R = 0.287 kPa.m^3/kg.K and c_p = 1.005 kJ/kg·°C (Table A-2). The heat of vaporization of water at 24°C is $h_{fg} = h_{fg\,@\,24°\text{C}} = 2444.1\ \text{kJ/kg}$ (Table A-4). The properties of the ambient and room air are determined from the psychrometric chart (Fig. A-31) to be

$$\left.\begin{array}{l} T_{\text{ambient}} = 32^\circ\text{C} \\ \phi_{\text{ambient}} = 50\% \end{array}\right\} w_{\text{ambient}} = 0.0150\ \text{kg/kg dryair}$$

$$\left.\begin{array}{l} T_{\text{room}} = 24^\circ\text{C} \\ \phi_{\text{room}} = 50\% \end{array}\right\} w_{\text{room}} = 0.0093\ \text{kg/kg dryair}$$

Analysis Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 1.2 times every hour, the air will enter the room at a mass flow rate of

$$\rho_{\text{ambient}} = \frac{P_0}{RT_0} = \frac{101.325\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(32+273\ \text{K})} = 1.158\ \text{kg/m}^3$$

$$\dot{m}_{\text{air}} = \rho_{\text{ambient}} \boldsymbol{V}_{\text{room}} \text{ACH} = (1.158\ \text{kg/m}^3)(20\times 13\times 3\ \text{m}^3)(1.2\ \text{h}^{-1}) = 1084\ \text{kg/h} = 0.301\ \text{kg/s}$$

Then the sensible, latent, and total infiltration heat loads of the room are determined to be

$$\dot{Q}_{\text{infiltration, sensible}} = \dot{m}_{\text{air}} c_p (T_{\text{ambient}} - T_{\text{room}}) = (0.301\ \text{kg/s})(1.005\ \text{kJ/kg.°C})(32-24)\text{°C} = \mathbf{2.42\ kW}$$

$$\dot{Q}_{\text{infiltration, latent}} = \dot{m}_{\text{air}} (w_{\text{ambient}} - w_{\text{room}}) h_{fg} = (0.301\ \text{kg/s})(0.0150-0.0093)(2444.1\ \text{kJ/kg}) = \mathbf{4.16\ kW}$$

$$\dot{Q}_{\text{infiltration, total}} = \dot{Q}_{\text{infiltration, sensible}} + \dot{Q}_{\text{infiltration, latent}} = 2.42 + 4.16 = \mathbf{6.58\ kW}$$

Discussion The specific volume of the dry air at the ambient conditions could also be determined from the psychrometric chart at ambient conditions.

14-68 The infiltration rate of a building is estimated to be 1.8 ACH. The sensible, latent, and total infiltration heat loads of the building at sea level are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air infiltrates at the outdoor conditions, and exfiltrates at the indoor conditions. **3** Excess moisture condenses at room temperature of 24°C. **4** The effect of water vapor on air density is negligible.

Properties The gas constant and the specific heat of air are R = 0.287 kPa.m^3/kg.K and c_p = 1.005 kJ/kg·°C (Table A-2). The heat of vaporization of water at 24°C is $h_{fg} = h_{fg\,@\,24°C} = 2444.1\,\text{kJ/kg}$ (Table A-4). The properties of the ambient and room air are determined from the psychrometric chart (Fig. A-31) to be

$$\left.\begin{array}{l} T_{\text{ambient}} = 32^\circ\text{C} \\ \phi_{\text{ambient}} = 50\% \end{array}\right\} w_{\text{ambient}} = 0.0150\,\text{kg/kg dryair}$$

$$\left.\begin{array}{l} T_{\text{room}} = 24^\circ\text{C} \\ \phi_{\text{room}} = 50\% \end{array}\right\} w_{\text{room}} = 0.0093\,\text{kg/kg dryair}$$

Analysis Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 1.8 times every hour, the air will enter the room at a mass flow rate of

$$\rho_{\text{ambient}} = \frac{P_0}{RT_0} = \frac{101.325\,\text{kPa}}{(0.287\,\text{kPa.m}^3/\text{kg.K})(32+273\,\text{K})} = 1.158\,\text{kg/m}^3$$

$$\dot{m}_{\text{air}} = \rho_{\text{ambient}}\boldsymbol{V}_{\text{room}}\text{ACH} = (1.158\,\text{kg/m}^3)(20\times13\times3\,\text{m}^3)(1.8\,\text{h}^{-1}) = 1084\,\text{kg/h} = 0.4514\,\text{kg/s}$$

Then the sensible, latent, and total infiltration heat loads of the room are determined to be

$$\dot{Q}_{\text{infiltration, sensible}} = \dot{m}_{\text{air}} c_p (T_{\text{ambient}} - T_{\text{room}}) = (0.4514\,\text{kg/s})(1.005\,\text{kJ/kg.°C})(32-24)°\text{C} = \mathbf{3.63\,kW}$$

$$\dot{Q}_{\text{infiltration, latent}} = \dot{m}_{\text{air}} (w_{\text{ambient}} - w_{\text{room}}) h_{fg} = (0.4514\,\text{kg/s})(0.0150-0.0093)(2444.1\,\text{kJ/kg}) = \mathbf{6.24\,kW}$$

$$\dot{Q}_{\text{infiltration, total}} = \dot{Q}_{\text{infiltration, sensible}} + \dot{Q}_{\text{infiltration, latent}} = 3.63 + 6.24 = \mathbf{9.87\,kW}$$

Discussion The specific volume of the dry air at the ambient conditions could also be determined from the psychrometric chart at ambient conditions.

Simple Heating and cooling

14-69C Relative humidity decreases during a simple heating process and increases during a simple cooling process. Specific humidity, on the other hand, remains constant in both cases.

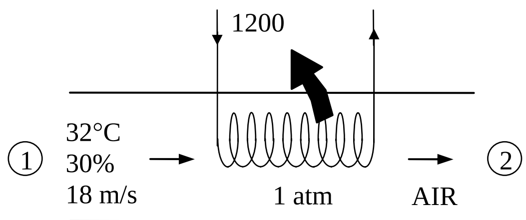
14-70C Because a horizontal line on the psychrometric chart represents a ω = constant process, and the moisture content ω of air remains constant during these processes.

14-71 Air enters a cooling section at a specified pressure, temperature, velocity, and relative humidity. The exit temperature, the exit relative humidity of the air, and the exit velocity 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) 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 state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 55.0 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0089 \text{ kg H}_2\text{O/kg dry air} \ (= \omega_2)$$
$$v_1 = 0.877 \text{ m}^3/\text{kg dry air}$$



The mass flow rate of dry air through the cooling section is

$$\begin{aligned}\dot{m}_a &= \frac{1}{v_1} V_1 A_1 \\ &= \frac{1}{(0.877 \text{ m}^3/\text{kg})}(18 \text{ m/s})(\pi \times 0.4^2/4 \text{ m}^2) \\ &= 2.58 \text{ kg/s}\end{aligned}$$

From the energy balance on air in the cooling section,

$$\begin{aligned}-\dot{Q}_{\text{out}} &= \dot{m}_a(h_2 - h_1) \\ -1200/60 \text{ kJ/s} &= (2.58 \text{ kg/s})(h_2 - 55.0) \text{ kJ/kg} \\ h_2 &= 47.2 \text{ kJ/kg dry air}\end{aligned}$$

The exit state of the air is fixed now since we know both h_2 and ω_2. From the psychrometric chart at this state we read

$$T_2 = \mathbf{24.4°C}$$
(*b*) $$\phi_2 = \mathbf{46.6\%}$$
$$v_2 = 0.856 \text{ m}^3/\text{kg dry air}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_2}{v_2} \longrightarrow \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

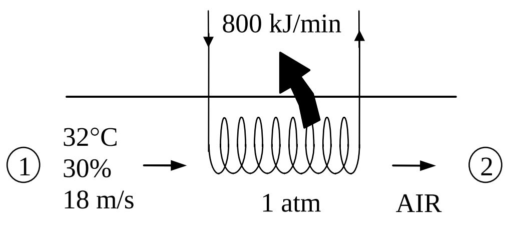
$$V_2 = \frac{v_2}{v_1} V_1 = \frac{0.856}{0.877}(18 \text{ m/s}) = \mathbf{17.6 \text{ m/s}}$$

14-72 Air enters a cooling section at a specified pressure, temperature, velocity, and relative humidity. The exit temperature, the exit relative humidity of the air, and the exit velocity 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) 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 state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 55.0 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0089 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$
$$v_1 = 0.877 \text{ m}^3/\text{kg dry air}$$



The mass flow rate of dry air through the cooling section is

$$\dot{m}_a = \frac{1}{v_1} V_1 A_1$$
$$= \frac{1}{(0.877 \text{ m}^3/\text{kg})}(18 \text{ m/s})(\pi \times 0.4^2/4 \text{ m}^2)$$
$$= 2.58 \text{ kg/s}$$

From the energy balance on air in the cooling section,

$$-\dot{Q}_{\text{out}} = \dot{m}_a(h_2 - h_1)$$
$$-800/60 \text{ kJ/s} = (2.58 \text{ kg/s})(h_2 - 55.0) \text{ kJ/kg}$$
$$h_2 = 49.8 \text{ kJ/kg dry air}$$

The exit state of the air is fixed now since we know both h_2 and ω_2. From the psychrometric chart at this state we read

$$T_2 = \mathbf{26.9°C}$$

(*b*) $\phi_2 = \mathbf{40.0\%}$

$$v_2 = 0.862 \text{ m}^3/\text{kg dry air}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_2}{v_2} \longrightarrow \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

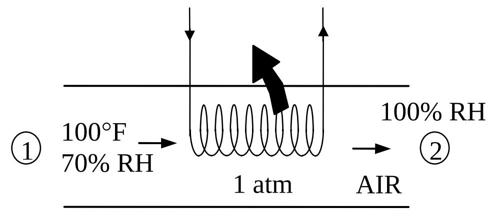
$$V_2 = \frac{v_2}{v_1} V_1 = \frac{0.862}{0.877}(18 \text{ m/s}) = \mathbf{17.7 \text{ m/s}}$$

14-73E 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 (Figure A-31E) to be

$$h_1 = 56.7 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0296 \text{ lbm } H_2O\text{/lbm dry air } (= \omega_2)$$
$$T_{dp,1} = 88.4°F$$



The exit state enthalpy is

$$\left.\begin{aligned} P &= 1 \text{ atm} \\ T_2 &= T_{dp,1} = 88.4°F \\ \phi_1 &= 1 \end{aligned}\right\} h_2 = 53.8 \text{ Btu/lbm dry air}$$

From the energy balance on air in the cooling section,

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

14-74 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.

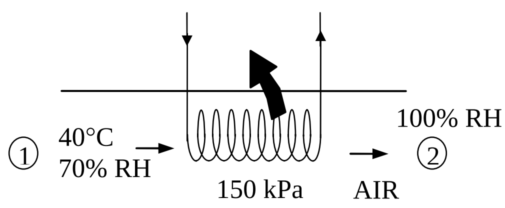
Analysis The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the cooling section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 150 kPa. The properties of the air at the inlet and exit states are determined to be

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 40°C}} = (0.70)(7.3851\text{ kPa}) = 5.1696\text{ kPa}$$

$$h_{g1} = h_{g\text{ @ 40°C}} = 2573.5\text{ kJ/kg}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(5.1696\text{ kPa})}{(150 - 5.1696)\text{ kPa}} = 0.02220\text{ kg H}_2\text{O/kg dry air}$$

$$\begin{aligned} h_1 &= c_p T_1 + \omega_1 h_{g1} \\ &= (1.005\text{ kJ/kg}\cdot°\text{C})(40°\text{C}) + (0.02220)(2573.5\text{ kJ/kg}) \\ &= 97.33\text{ kJ/kg dry air} \end{aligned}$$



$$P_{v2} = P_{v1} = 5.1696\text{ kPa}$$

$$P_{g2} = \frac{P_{v2}}{\phi_2} = \frac{5.1696\text{ kPa}}{1} = 5.1696\text{ kPa}$$

$$T_2 = T_{\text{sat @ 5.1695 kPa}} = 33.5°\text{C}$$

$$h_{g2} = h_{g\text{ @ 33.5°C}} = 2561.9\text{ kJ/kg}$$

$$\omega_2 = \omega_1$$

$$\begin{aligned} h_2 &= c_p T_2 + \omega_2 h_{g2} \\ &= (1.005\text{ kJ/kg}\cdot°\text{C})(33.5°\text{C}) + (0.02220)(2561.9\text{ kJ/kg}) \\ &= 90.55\text{ kJ/kg dry air} \end{aligned}$$

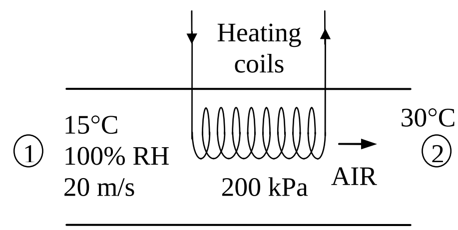
From the energy balance on air in the cooling section,

$$q_{\text{out}} = h_1 - h_2 = 97.33 - 90.55 = \mathbf{6.78\text{ kJ/kg dry air}}$$

14-75 Saturated humid air at a specified state is heated to a specified temperature. The relative humidity at the exit and the rate of heat transfer 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **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 heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 200 kPa. The properties of the air at the inlet and exit states are determined to be



$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 15°C}} = (1.0)(1.7057\text{ kPa}) = 1.7057\text{ kPa}$$

$$h_{g1} = h_{g\text{ @ 15°C}} = 2528.3\text{ kJ/kg}$$

$$P_{a1} = P_1 - P_{v1} = 200 - 1.7057 = 198.29\text{ kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\text{ K})}{198.29\text{ kPa}} = 0.4168\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622 P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.7057\text{ kPa})}{(200 - 1.7057)\text{ kPa}} = 0.005350\text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(15°\text{C}) + (0.005350)(2528.3\text{ kJ/kg}) = 28.60\text{ kJ/kg dry air}$$

$$P_{v2} = P_{v1} = 1.7057\text{ kPa}$$

$$P_{g2} = P_{\text{sat @ 30°C}} = 4.2469\text{ kPa}$$

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{1.7057\text{ kPa}}{4.2469\text{ kPa}} = 0.402 = \mathbf{40.2\%}$$

$$h_{g2} = h_{g\text{ @ 30°C}} = 2555.6\text{ kJ/kg}$$

$$\omega_2 = \omega_1$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot°\text{C})(30°\text{C}) + (0.005350)(2555.6\text{ kJ/kg}) = 43.82\text{ kJ/kg dry air}$$

Then,

$$\dot{\boldsymbol{V}}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (20\text{ m/s})\left(\frac{\pi(0.04\text{ m})^2}{4}\right) = 0.02513\text{ m}^3/\text{s}$$

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.02513\text{ m}^3/\text{s}}{0.4168\text{ m}^3/\text{kg dry air}} = 0.06029\text{ kg/s}$$

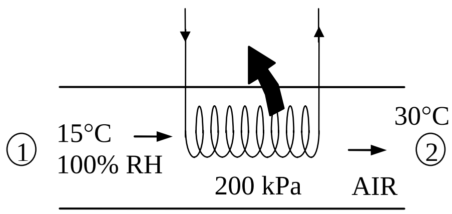
From the energy balance on air in the heating section,

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1) = (0.06029\text{ kg/s})(43.82 - 28.60)\text{kJ/kg} = \mathbf{0.918\text{ kW}}$$

14-76 Saturated humid air at a specified state is heated to a specified temperature. The rate at which the exergy of the humid air is increased 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 heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 200 kPa. The properties of the air at the inlet and exit states are determined to be



$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 15°C}} = (1.0)(1.7057\text{ kPa}) = 1.7057\text{ kPa}$$

$$h_{g1} = h_{g\,@\,15°\text{C}} = 2528.3\text{ kJ/kg}$$

$$s_{g1} = s_{g\,@\,15°\text{C}} = 8.7803\text{ kJ/kg}\cdot\text{K}$$

$$P_{a1} = P_1 - P_{v1} = 200 - 1.7057 = 198.29\text{ kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\text{ K})}{198.29\text{ kPa}} = 0.4168\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.7057\text{ kPa})}{(200 - 1.7057)\text{ kPa}} = 0.005350\text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(15°\text{C}) + (0.005350)(2528.3\text{ kJ/kg}) = 28.60\text{ kJ/kg dry air}$$

$$P_{v2} = P_{v1} = 1.7057\text{ kPa}$$

$$P_{g2} = P_{\text{sat @ 30°C}} = 4.2469\text{ kPa}$$

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{1.7057\text{ kPa}}{4.2469\text{ kPa}} = 0.402 = \mathbf{40.2\%}$$

$$P_{a2} = P_2 - P_{v2} = 200 - 1.7057 = 198.29\text{ kPa}$$

$$h_{g2} = h_{g\,@\,30°\text{C}} = 2555.6\text{ kJ/kg}$$

$$s_{g2} = s_{g\,@\,30°\text{C}} = 8.4520\text{ kJ/kg}\cdot\text{K}$$

$$\omega_2 = \omega_1$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot°\text{C})(30°\text{C}) + (0.005350)(2555.6\text{ kJ/kg}) = 43.82\text{ kJ/kg dry air}$$

The entropy change of the dry air is

$$(s_2 - s_1)_{\text{dry air}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_{a2}}{P_{a1}} = (1.005)\ln\frac{303}{288} - (0.287)\ln\frac{198.29}{198.29} = 0.05103\text{ kJ/kg}\cdot\text{K}$$

The entropy change of the air-water mixture is

$$s_2 - s_1 = (s_2 - s_1)_{\text{dry air}} + \omega(s_2 - s_1)_{\text{water vapor}} = 0.05103 + (0.005350)(8.4520 - 8.7803) = 0.04927\text{ kJ/kg}\cdot\text{K}$$

The mass flow rate of the dry air is

$$\dot{\boldsymbol{V}}_1 = V_1 A_1 = V_1\frac{\pi D^2}{4} = (20\text{ m/s})\left(\frac{\pi(0.04\text{ m})^2}{4}\right) = 0.02513\text{ m}^3/\text{s}$$

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.02513\text{ m}^3/\text{s}}{0.4168\text{ m}^3/\text{kg dry air}} = 0.06029\text{ kg/s}$$

The exergy increase of the humid air during this process is then,

$$\Delta\Phi = \dot{m}_a(\psi_2 - \psi_1) = \dot{m}_a\left[(h_2 - h_1) - T_0(s_2 - s_1)\right]$$
$$= (0.06029\text{ kg/s})\left[(43.82 - 28.60)\text{kJ/kg - }(288\text{ K})(0.04927\text{ kJ/kg}\cdot\text{K})\right] = \mathbf{0.062\ kW/K}$$

Heating with Humidification

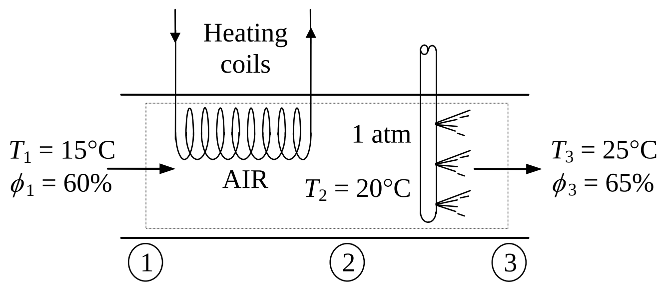
14-77C To achieve a higher level of comfort. Very dry air can cause dry skin, respiratory difficulties, and increased static electricity.

14-78 Air is first heated and then humidified by water vapor. The amount of steam added to the air and the amount of heat transfer to the air 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 31.1 \text{ kJ / kg dry air}$$
$$\omega_1 = 0.0064 \text{ kg H}_2\text{O / kg dry air } (= \omega_2)$$
$$h_2 = 36.2 \text{ kJ / kg dry air}$$
$$h_3 = 58.1 \text{ kJ / kg dry air}$$
$$\omega_3 = 0.0129 \text{ kg H}_2\text{O / kg dry air}$$



Analysis (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The amount of steam added to the air in the heating section is

$$\Delta\omega = \omega_3 - \omega_2 = 0.0129 - 0.0064 = \mathbf{0.0065 \text{ kg H}_2\text{O / kg dry air}}$$

(*b*) The heat transfer to the air in the heating section per unit mass of air is

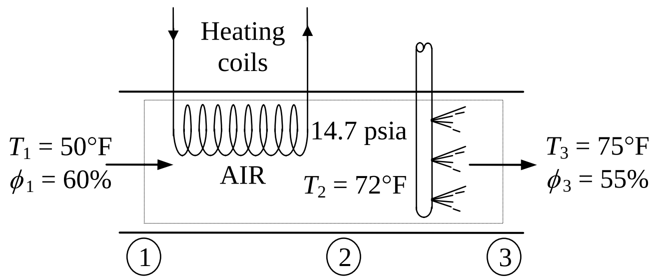
$$q_{\text{in}} = h_2 - h_1 = 36.2 - 31.1 = \mathbf{5.1 \text{ kJ / kg dry air}}$$

14-79E Air is first heated and then humidified by water vapor. The amount of steam added to the air and the amount of heat transfer to the air 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31E) to be

$$h_1 = 17.0 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0046 \text{ lbm H}_2\text{O/lbm dry air}$$
$$h_2 = 22.3 \text{ Btu/lbm dry air}$$
$$\omega_2 = \omega_1 = 0.0046 \text{ lbm H}_2\text{O/lbm dry air}$$
$$h_3 = 29.2 \text{ Btu/lbm dry air}$$
$$\omega_3 = 0.0102 \text{ lbm H}_2\text{O/lbm dry air}$$



Analysis (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The amount of steam added to the air in the heating section is

$$\Delta\omega = \omega_3 - \omega_2 = 0.0102 - 0.0046 = \mathbf{0.0056 \text{ lbm H}_2\text{O/lbm dry air}}$$

(*b*) The heat transfer to the air in the heating section per unit mass of air is

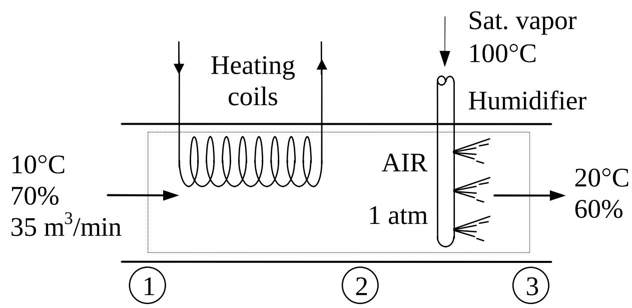
$$q_{\text{in}} = h_2 - h_1 = 22.3 - 17.0 = \mathbf{5.3 \text{ Btu/lbm dry air}}$$

14-80 Air is first heated and then humidified by wet steam. The temperature and relative humidity of air at the exit of heating section, the rate of heat transfer, and the rate at which water is added to the air 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 23.5 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0053 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$
$$v_1 = 0.809 \text{ m}^3\text{/kg dry air}$$
$$h_3 = 42.3 \text{ kJ/kg dry air}$$
$$\omega_3 = 0.0087 \text{ kg H}_2\text{O/kg dry air}$$



Analysis (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{35 \text{ m}^3/\text{min}}{0.809 \text{ m}^3/\text{kg}} = 43.3 \text{ kg/min}$$

Noting that $Q = W = 0$, the energy balance on the humidifying section can be expressed as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_w h_w + \dot{m}_{a2} h_2 = \dot{m}_a h_3$$
$$(\omega_3 - \omega_2) h_w + h_2 = h_3$$

Solving for h_2,

$$h_2 = h_3 - (\omega_3 - \omega_2) h_{g\,@100°\text{C}} = 42.3 - (0.0087 - 0.0053)(2675.6) = 33.2 \text{ kJ/kg dry air}$$

Thus at the exit of the heating section we have ω_2 = 0.0053 kg H_2O dry air and h_2 = 33.2 kJ/kg dry air, which completely fixes the state. Then from the psychrometric chart we read

$$T_2 = \mathbf{19.5°C}$$
$$\phi_2 = \mathbf{37.8\%}$$

(*b*) The rate of heat transfer to the air in the heating section is

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (43.3 \text{ kg/min})(33.2 - 23.5) \text{ kJ/kg} = \mathbf{420 \text{ kJ/min}}$$

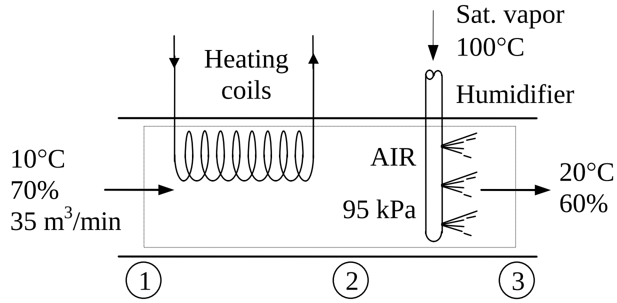
(*c*) The amount of water added to the air in the humidifying section is determined from the conservation of mass equation of water in the humidifying section,

$$\dot{m}_w = \dot{m}_a (\omega_3 - \omega_2) = (43.3 \text{ kg/min})(0.0087 - 0.0053) = \mathbf{0.15 \text{ kg/min}}$$

14-81 Air is first heated and then humidified by wet steam. The temperature and relative humidity of air at the exit of heating section, the rate of heat transfer, and the rate at which water is added to the air 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The amount of moisture in the air also remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. The properties of the air at various states are determined to be



$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 10°C}} = (0.70)(1.2281\text{ kPa}) = 0.860\text{ kPa} \ (= P_{v2})$$

$$P_{a1} = P_1 - P_{v1} = 95 - 0.860 = 94.14\text{ kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\text{ K})}{94.14\text{ kPa}} = 0.863\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\, P_{v1}}{P_1 - P_{v1}} = \frac{0.622(0.86\text{ kPa})}{(95 - 0.86)\text{ kPa}} = 0.00568\text{ kg H}_2\text{O/kg dry air} \ (= \omega_2)$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(10°\text{C}) + (0.00568)(2519.2\text{ kJ/kg}) = 24.36\text{ kJ/kg dry air}$$

$$P_{v3} = \phi_3 P_{g3} = \phi_3 P_{\text{sat @ 20°C}} = (0.60)(2.3392\text{ kPa}) = 1.40\text{ kPa}$$

$$\omega_3 = \frac{0.622\, P_{v3}}{P_3 - P_{v3}} = \frac{0.622(1.40\text{ kPa})}{(95 - 1.40)\text{ kPa}} = 0.00930\text{ kg H}_2\text{O/kg dry air}$$

$$h_3 = c_p T_3 + \omega_3 h_{g3} = (1.005\text{ kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0093)(2537.4\text{ kJ/kg}) = 43.70\text{ kJ/kg dry air}$$

Also,

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{35\text{ m}^3/\text{min}}{0.863\text{ m}^3/\text{kg}} = 40.6\text{ kg/min}$$

Noting that $Q = W = 0$, the energy balance on the humidifying section gives

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i \longrightarrow \dot{m}_w h_w + \dot{m}_{a2} h_2 = \dot{m}_a h_3 \longrightarrow (\omega_3 - \omega_2) h_w + h_2 = h_3$$

$$h_2 = h_3 - (\omega_3 - \omega_2) h_{g\text{ @ 100°C}} = 43.7 - (0.0093 - 0.00568) \times 2675.6 = 34.0\text{ kJ/kg dry air}$$

Thus at the exit of the heating section we have $\omega = 0.00568$ kg H_2O dry air and $h_2 = 34.0$ kJ/kg dry air, which completely fixes the state. The temperature of air at the exit of the heating section is determined from the definition of enthalpy,

$$h_2 = c_p T_2 + \omega_2 h_{g2} \cong c_p T_2 + \omega_2 (2500.9 + 1.82 T_2)$$
$$34.0 = (1.005) T_2 + (0.00568)(2500.9 + 1.82 T_2)$$

Solving for h_2, yields

$$T_2 = \mathbf{19.5°C}$$

The relative humidity at this state is

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{P_{v2}}{P_{\text{sat @ 19.5°C}}} = \frac{0.859\ \text{kPa}}{2.2759\ \text{kPa}} = 0.377 \quad \text{or} \quad \mathbf{37.7\%}$$

(*b*) The rate of heat transfer to the air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1) = (40.6\ \text{kg/min})(34.0 - 24.36)\ \text{kJ/kg} = \mathbf{391\ kJ/min}$$

(*c*) The amount of water added to the air in the humidifying section is determined from the conservation of mass equation of water in the humidifying section,

$$\dot{m}_w = \dot{m}_a(\omega_3 - \omega_2) = (40.6\ \text{kg / min})(0.0093 - 0.00568) = \mathbf{0.147\ kg / min}$$

Cooling with Dehumidification

14-82C To drop its relative humidity to more desirable levels.

14-83 Air is first cooled, then dehumidified, and finally heated. The temperature of air before it enters the heating section, the amount of heat removed in the cooling section, and the amount of heat supplied in the heating section are to be determined.

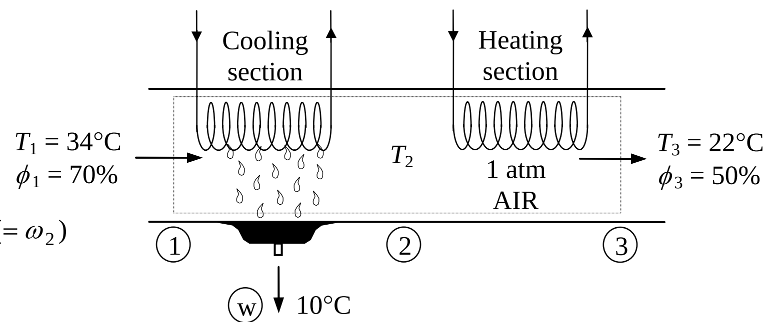
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (a) The amount of moisture in the air decreases due to dehumidification ($\omega_3 < \omega_1$), and remains constant during heating ($\omega_3 = \omega_2$). The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The intermediate state (state 2) is also known since ϕ_2 = 100% and $\omega_2 = \omega_3$. Therefore, we can determine the properties of the air at all three states from the psychrometric chart (Fig. A-31) to be

$$h_1 = 95.2 \text{ kJ / kg dry air}$$
$$\omega_1 = 0.0238 \text{ kg H}_2\text{O / kg dry air}$$

and

$$h_3 = 43.1 \text{ kJ / kg dry air}$$
$$\omega_3 = 0.0082 \text{ kg H}_2\text{O / kg dry air} \ (= \omega_2)$$



Also,

$$h_w \cong h_{f\,@10°\text{C}} = 42.02 \text{ kJ/kg (Table A-4)}$$
$$h_2 = 31.8 \text{ kJ/kg dry air}$$
$$T_2 = \mathbf{11.1°C}$$

(b) The amount of heat removed in the cooling section is determined from the energy balance equation applied to the cooling section,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{\text{out,cooling}}$$
$$\dot{Q}_{\text{out,cooling}} = \dot{m}_{a1} h_1 - (\dot{m}_{a2} h_2 + \dot{m}_w h_w) = \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w$$

or, per unit mass of dry air,

$$q_{\text{out,cooling}} = (h_1 - h_2) - (\omega_1 - \omega_2) h_w$$
$$= (95.2 - 31.8) - (0.0238 - 0.0082)42.02$$
$$= \mathbf{62.7 \text{ kJ/kg dry air}}$$

(c) The amount of heat supplied in the heating section per unit mass of dry air is

$$q_{in,\text{heating}} = h_3 - h_2 = 43.1 - 31.8 = \mathbf{11.3 \text{ kJ / kg dry air}}$$

14-84 [*Also solved by EES on enclosed CD*] Air is cooled by passing it over a cooling coil through which chilled water flows. The rate of heat transfer, the mass flow rate of water, and the exit velocity of airstream are to be determined.

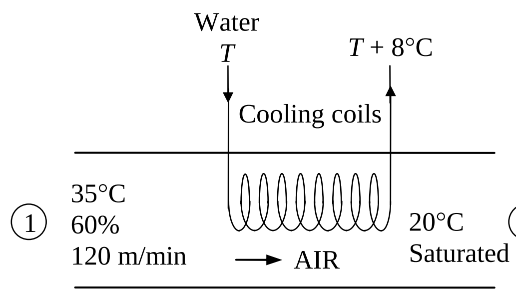
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The saturation pressure of water at 35ºC is 5. 6291 kPa (Table A-4). Then the dew point temperature of the incoming air stream at 35°C becomes

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ } 0.6\times 5.6291\,\text{kPa}} = 26°\text{C} \quad \text{(Table A-5)}$$

since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense. The amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. Then the properties of the air at both states are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 90.3\ \text{kJ/kg dry air}$$
$$\omega_1 = 0.0215\ \text{kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.904\ \text{m}^3\text{/kg dry air}$$



and

$$h_2 = 57.5\ \text{kJ/kg dry air}$$
$$\omega_2 = 0.0147\ \text{kg H}_2\text{O/kg dry air}$$
$$v_2 = 0.851\ \text{m}^3\text{/kg dry air}$$

Also, $h_w \cong h_{f\text{ @ }20°\text{C}} = 83.93\ \text{kJ/kg} \quad \text{(Table A-4)}$

Then,

$$\dot{V}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (120\ \text{m/min})\left(\frac{\pi (0.3\ \text{m})^2}{4}\right) = 8.48\ \text{m}^3\ /\ \text{min}$$

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{8.48\ \text{m}^3\ /\ \text{min}}{0.904\ \text{m}^3\ /\ \text{kg dry air}} = 9.38\ \text{kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the water),

Water Mass Balance: $\quad \sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (9.38\ \text{kg/min})(0.0215 - 0.0147) = 0.064\ \ \text{kg/min}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{\text{out}} \longrightarrow Q_{out} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (9.38\ \text{kg/min})(90.3 - 57.5)\text{kJ/kg} - (0.064\ \text{kg/min})(83.93\ \text{kJ/kg}) = \mathbf{302.3\ kJ/min}$$

(*b*) Noting that the heat lost by the air is gained by the cooling water, the mass flow rate of the cooling water is determined from

$$\dot{Q}_{\text{cooling water}} = \dot{m}_{\text{cooling water}}\Delta h = \dot{m}_{\text{cooling water}} c_p \Delta T$$

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}_w}{c_p \Delta T} = \frac{302.3\ \text{kJ/min}}{(4.18\ \text{kJ/kg}\cdot°\text{C})(8°\text{C})} = \mathbf{9.04\ kg/min}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_2}{v_2} \longrightarrow \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

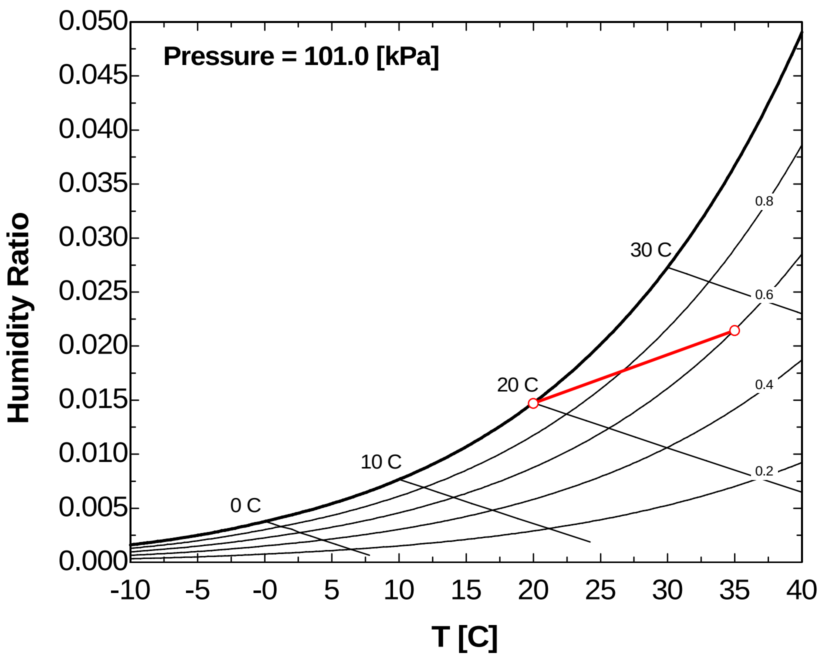
$$V_2 = \frac{v_2}{v_1} V_1 = \frac{0.851}{0.904}(120\ \ \text{m/min}) = \mathbf{113\ \ m/min}$$

14-85 EES Problem 14-84 is reconsidered. A general solution of the problem in which the input variables may be supplied and parametric studies performed is to be developed and the process is to be shown in the psychrometric chart for each set of input variables.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data from the Diagram Window"
{D=0.3
P[1] =101.32 [kPa]
T[1] = 35 [C]
RH[1] = 60/100   "%, relative humidity"
Vel[1] = 120/60 "[m/s]"
DELTAT_cw =8 [C]
P[2] = 101.32 [kPa]
T[2] = 20 [C]}
RH[2] = 100/100 "%"

"Dry air flow rate, m_dot_a, is constant"
Vol_dot[1]= (pi * D^2)/4*Vel[1]
v[1]=VOLUME(AirH2O,T=T[1],P=P[1],R=RH[1])
m_dot_a = Vol_dot[1]/v[1]

"Exit vleocity"
Vol_dot[2]= (pi * D^2)/4*Vel[2]
v[2]=VOLUME(AirH2O,T=T[2],P=P[2],R=RH[2])
m_dot_a = Vol_dot[2]/v[2]

"Mass flow rate of the condensed water"
m_dot_v[1]=m_dot_v[2]+m_dot_w
w[1]=HUMRAT(AirH2O,T=T[1],P=P[1],R=RH[1])
m_dot_v[1] = m_dot_a*w[1]
w[2]=HUMRAT(AirH2O,T=T[2],P=P[2],R=RH[2])
m_dot_v[2] = m_dot_a*w[2]

"SSSF conservation of energy for the air"
m_dot_a *(h[1] + (1+w[1])*Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)) + Q_dot = m_dot_a*(h[2]
+(1+w[2])*Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)) +m_dot_w*h_liq_2
h[1]=ENTHALPY(AirH2O,T=T[1],P=P[1],w=w[1])
h[2]=ENTHALPY(AirH2O,T=T[2],P=P[2],w=w[2])
h_liq_2=ENTHALPY(Water,T=T[2],P=P[2])

"SSSF conservation of energy for the cooling water"
-Q_dot =m_dot_cw*Cp_cw*DELTAT_cw  "Note: Q_netwater=-Q_netair"
Cp_cw = SpecHeat(water,T=10,P=P[2])"kJ/kg-K"
```

RH_1	ma	mw	mcw	Q [kW]	Vel_1 [m/s]	Vel_2 [m/s]	T_1 [C]	T_2 [C]	w_1	w_2
0.5	0.1574	0.0004834	0.1085	-3.632	2	1.894	35	20	0.01777	0.0147
0.6	0.1565	0.001056	0.1505	-5.039	2	1.883	35	20	0.02144	0.0147
0.7	0.1556	0.001629	0.1926	-6.445	2	1.872	35	20	0.02516	0.0147
0.8	0.1547	0.002201	0.2346	-7.852	2	1.861	35	20	0.02892	0.0147
0.9	0.1538	0.002774	0.2766	-9.258	2	1.85	35	20	0.03273	0.0147

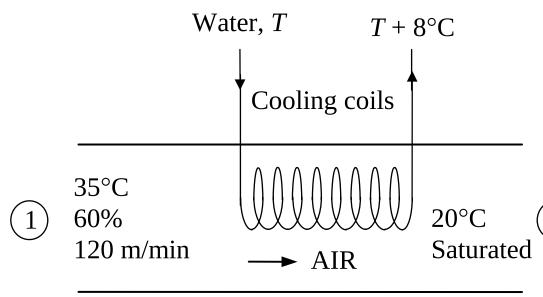


14-86 Air is cooled by passing it over a cooling coil. The rate of heat transfer, the mass flow rate of water, and the exit velocity of airstream are to be determined.

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.

Analysis (*a*) The dew point temperature of the incoming air stream at 35°C is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 35°C}}$$
$$= (0.6)(5.6291\,\text{kPa}) = 3.38\,\text{kPa}$$
$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 3.38 kPa}} = 25.9°\text{C}$$



Since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense.

The amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. Then the properties of the air at both states are determined to be

$$P_{a1} = P_1 - P_{v1} = 95 - 3.38 = 91.62\,\text{kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\,\text{K})}{91.62\,\text{kPa}} = 0.965\,\text{m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(3.38\,\text{kPa})}{(95-3.38)\,\text{kPa}} = 0.0229\,\text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\,\text{kJ/kg}\cdot°\text{C})(35°\text{C}) + (0.0229)(2564.6\,\text{kJ/kg})$$
$$= 93.90\,\text{kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = (1.00)P_{\text{sat @ 20°C}} = 2.3392\,\text{kPa}$$

$$\boldsymbol{v}_2 = \frac{R_a T_2}{P_{a2}} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}{(95-2.339)\,\text{kPa}} = 0.908\,\text{m}^3/\text{kg dry air}$$

$$\omega_2 = \frac{0.622\,P_{v2}}{P_2 - P_{v2}} = \frac{0.622(2.339\,\text{kPa})}{(95-2.339)\,\text{kPa}} = 0.0157\,\text{kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\,\text{kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0157)(2537.4\,\text{kJ/kg})$$
$$= 59.95\,\text{kJ/kg dry air}$$

Also,

$$h_w \cong h_{f\text{ @ 20°C}} = 83.915\,\text{kJ/kg} \qquad \text{(Table A-4)}$$

Then,

$$\dot{\boldsymbol{V}}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (120\,\text{m/min})\left(\frac{\pi(0.3\,\text{m})^2}{4}\right) = 8.48\,\text{m}^3/\text{min}$$

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{8.48\,\text{m}^3/\text{min}}{0.965\,\text{m}^3/\text{kg dry air}} = 8.79\,\text{kg/min}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section (excluding the water),

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (8.79 \text{ kg / min})(0.0229 - 0.0157) = 0.0633 \text{ kg / min}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{\text{out}} \rightarrow \dot{Q}_{out} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (8.79 \text{ kg/min})(93.90 - 59.94)\text{kJ/kg} - (0.0633 \text{ kg/min})(83.915 \text{ kJ/kg}) = \mathbf{293.2\ kJ/min}$$

(*b*) Noting that the heat lost by the air is gained by the cooling water, the mass flow rate of the cooling water is determined from

$$\dot{Q}_{\text{cooling water}} = \dot{m}_{\text{cooling water}} \Delta h = \dot{m}_{\text{cooling water}} c_p \Delta T$$

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}_w}{c_p \Delta T} = \frac{293.2 \text{ kJ/min}}{(4.18 \text{ kJ/kg} \cdot {}^\circ\text{C})(8^\circ\text{C})} = \mathbf{8.77\ kg/min}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{\dot{\boldsymbol{V}}_2}{\boldsymbol{v}_2} \longrightarrow \frac{V_1 A}{\boldsymbol{v}_1} = \frac{V_2 A}{\boldsymbol{v}_2}$$

$$V_2 = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1} V_1 = \frac{0.908}{0.965}(120 \text{ m/min}) = \mathbf{113\ m/min}$$

14-87 Air is cooled and dehumidified at constant pressure. The amount of water removed from the air and the cooling requirement are to be determined.

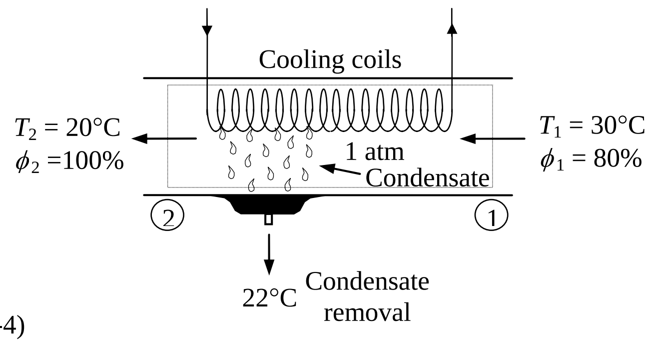
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 85.5 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0217 \text{ kg } H_2O\text{/kg dry air}$$

and

$$\phi_2 = 1.0$$
$$h_2 = 57.6 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0148 \text{ kg } H_2O\text{/kg dry air}$$



Also,

$$h_w \cong h_{f\,@\,22°C} = 92.28 \text{ kJ/kg} \quad \text{(Table A-4)}$$

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\Delta\omega = \omega_1 - \omega_2 = 0.0217 - 0.0148 = \mathbf{0.0069 \text{ kg } H_2O\text{/kg dry air}}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$q_{\text{out}} = h_1 - h_2 - (\omega_1 - \omega_2)h_w$$
$$= (85.5 - 57.6)\text{kJ/kg} - (0.0069)(92.28)$$
$$= \mathbf{27.3 \text{ kJ/kg dry air}}$$

14-88E Air is cooled and dehumidified at constant pressure. The amount of water removed from the air and the rate of cooling are to be determined.

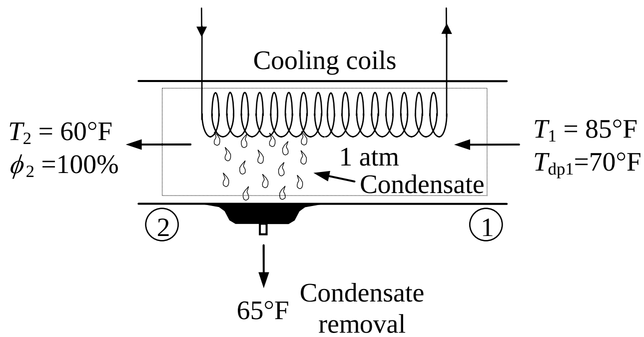
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31E) to be

$$h_1 = 37.8 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0158 \text{ lbm H}_2\text{O/lbm dry air}$$
$$\boldsymbol{v}_1 = 14.08 \text{ ft}^3\text{/lbm dry air}$$

and

$$\phi_2 = 1.0$$
$$h_2 = 26.5 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0111 \text{ lbm H}_2\text{O/lbm dry air}$$



Also,

$$h_w \cong h_{f\,@\,65°\text{F}} = 33.08 \text{ Btu/lbm} \quad \text{(Table A-4E)}$$

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass flow rate of air is

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{(10{,}000/3600)\text{ ft}^3/\text{s}}{14.08 \text{ ft}^3/\text{lbm dry air}} = 0.1973 \text{ lbm/s}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (0.1973 \text{ lbm/s})(0.0158 - 0.0111) = \mathbf{0.000927 \text{ lbm/s}}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,\text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (0.1973 \text{ lbm/s})(37.8 - 26.5)\text{Btu/lbm} - (0.000927 \text{ lbm/s})(33.08 \text{ Btu/lbm})$$

$$= \mathbf{2.20 \text{ Btu/s}}$$

14-89 Air is cooled and dehumidified at constant pressure. The amount of water removed from the air and the rate of cooling are to be determined.

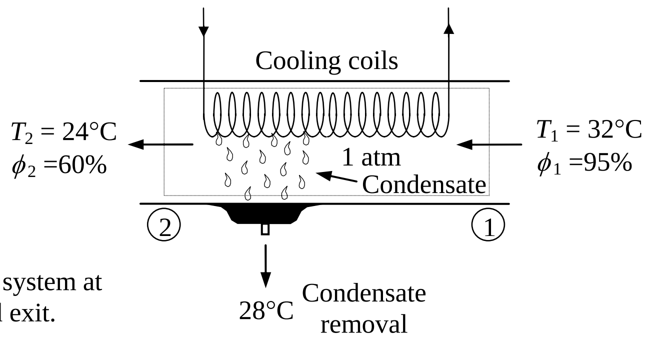
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 106.8 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0292 \text{ kg H}_2\text{O/kg dry air}$$
$$\boldsymbol{v}_1 = 0.905 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 52.7 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0112 \text{ kg H}_2\text{O/kg dry air}$$



We assume that the condensate leaves this system at the average temperature of the air inlet and exit. Then,

$$h_w \cong h_{f\,@\,28°\text{C}} = 117.4 \text{ kJ/kg} \quad \text{(Table A-4)}$$

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass of air is

$$m_a = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{1000 \text{ m}^3}{0.905 \text{ m}^3 \text{ / kg dry air}} = 1105 \text{ kg}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$m_w = m_a(\omega_1 - \omega_2) = (1105 \text{ kg})(0.0292 - 0.0112) = \mathbf{19.89 \text{ kg}}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$
$$Q_{\text{out}} = m_a(h_1 - h_2) - m_w h_w$$
$$Q_{\text{out}} = (1105 \text{ kg})(106.8 - 52.7)\text{kJ/kg} - (19.89 \text{ kg})(117.4 \text{ kJ/kg})$$
$$= \mathbf{57{,}450 \text{ kJ}}$$

14-90 The humid air of the previous problem is reconsidered. The exit temperature of the air to produce the desired dehumidification is to be determined.

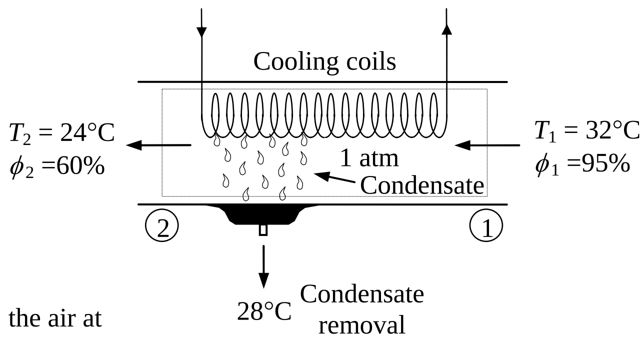
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be (from the data of the previous problem)

$$h_1 = 106.8 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0292 \text{ kg H}_2\text{O/kg dry air}$$
$$\boldsymbol{v}_1 = 0.905 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 52.7 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0112 \text{ kg H}_2\text{O/kg dry air}$$



Analysis For the desired dehumidification, the air at the exit should be saturated with a specific humidity of 0.0112 kg water/kg dry air. That is,

$$\phi_2 = 1.0$$
$$\omega_2 = 0.0112 \text{ kg H}_2\text{O/kg dry air}$$

The temperature of the air at this state is

$$T_2 = \mathbf{15.8°C}$$

14-91 Air is cooled and dehumidified at constant pressure. The cooling required is provided by a simple ideal vapor-compression refrigeration system using refrigerant-134a as the working fluid. The exergy destruction in the total system per 1000 m^3 of dry air is to be determined.

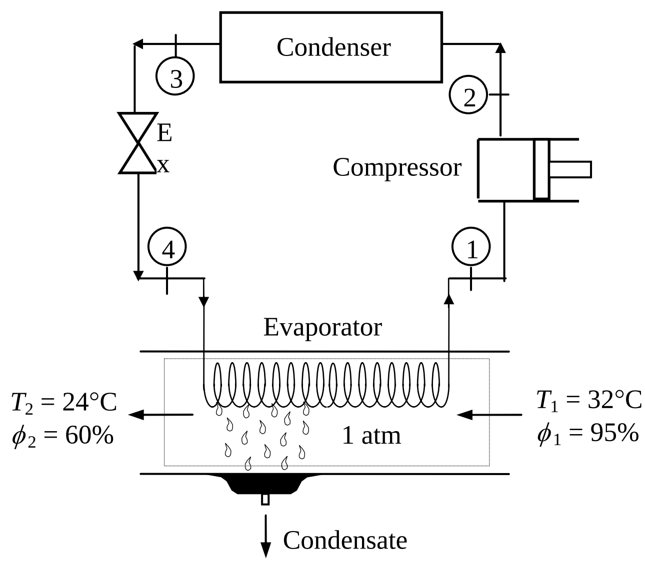
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 106.8 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0292 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.905 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 52.7 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0112 \text{ kg H}_2\text{O/kg dry air}$$



We assume that the condensate leaves this system at the average temperature of the air inlet and exit. Then, from Table A-4,

$$h_w \cong h_{f\,@\,28°\text{C}} = 117.4 \text{ kJ/kg}$$

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass of air is

$$m_a = \frac{V_1}{v_1} = \frac{1000 \text{ m}^3}{0.905 \text{ m}^3 \text{ / kg dry air}} = 1105 \text{ kg}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$m_w = m_a(\omega_1 - \omega_2) = (1105 \text{ kg})(0.0292 - 0.0112) = 19.89 \text{ kg}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$
$$Q_{\text{out}} = m_a(h_1 - h_2) - m_w h_w$$
$$Q_{\text{out}} = (1105 \text{ kg})(106.8 - 52.7)\text{kJ/kg} - (19.89 \text{ kg})(117.4 \text{ kJ/kg}) = 57{,}450 \text{ kJ}$$

We obtain the properties for the vapor-compression refrigeration cycle as follows (Tables A-11,through A-13):

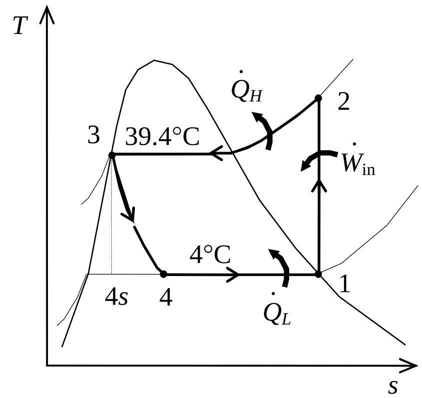
$$\left.\begin{array}{l} T_1 = 4°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,4°\text{C}} = 252.77 \text{ kJ/kg} \\ s_1 = s_{g\,@\,4°\text{C}} = 0.92927 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = P_{\text{sat}\,@\,39.4°\text{C}} = 1\,\text{MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 275.29 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1\,\text{MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,1\,\text{MPa}} = 107.32 \text{ kJ/kg} \\ s_3 = s_{f\,@\,1\,\text{MPa}} = 0.39189 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$h_4 \cong h_3 = 107.32 \text{ kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_4 = 4°\text{C} \\ h_4 = 107.32 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} x_4 = 0.2561 \\ s_4 = 0.4045 \text{ kJ/kg}\cdot\text{K} \end{array}$$



The mass flow rate of refrigerant-134a is

$$m_R = \frac{Q_L}{h_1 - h_4} = \frac{57{,}450 \text{ kJ}}{(252.77 - 107.32)\text{kJ/kg}} = 395.0 \text{ kg}$$

The amount of heat rejected from the condenser is

$$Q_H = m_R(h_2 - h_3) = (395.0 \text{ kg})(275.29 - 107.32) \text{ kJ/kg} = 66{,}350 \text{ kg}$$

Next, we calculate the exergy destruction in the components of the refrigeration cycle:

$$X_{\text{destroyed},12} = m_R T_0 (s_2 - s_1) = 0 \quad \text{(since the process is isentropic)}$$

$$X_{\text{destroyed},23} = T_0\left(m_R(s_3 - s_2) + \frac{Q_H}{T_H}\right)$$

$$= (305\text{ K})\left((395\text{ kg})(0.39189 - 0.92927)\text{ kJ/kg}\cdot\text{K} + \frac{66{,}350\text{ kJ}}{305\text{ K}}\right) = 1609\text{ kJ}$$

$$X_{\text{destroyed},34} = m_R T_0 (s_4 - s_3) = (395\text{ kg})(305\text{ K})(0.4045 - 0.39189)\text{ kJ/kg}\cdot\text{K} = 1519\text{ kJ}$$

The entropies of water vapor in the air stream are

$$s_{g1} = s_{g\,@\,32°\text{C}} = 8.4114 \text{ kJ/kg}\cdot\text{K}$$
$$s_{g2} = s_{g\,@\,24°\text{C}} = 8.5782 \text{ kJ/kg}\cdot\text{K}$$

The entropy change of water vapor in the air stream is

$$\Delta S_{\text{vapor}} = m_a(\omega_2 s_{g2} - \omega_1 s_{g1}) = (1105)(0.0112 \times 8.5782 - 0.0292 \times 8.4114) = -165.2 \text{ kJ/K}$$

The entropy of water leaving the cooling section is

$$S_w = m_w s_{f\,@\,28°\text{C}} = (19.89\text{ kg})(0.4091\text{ kJ/kg}\cdot\text{K}) = 8.14\text{ kJ/K}$$

The partial pressures of water vapor and dry air for air streams are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat}\,@\,32°\text{C}} = (0.95)(4.760\text{ kPa}) = 4.522\text{ kPa}$$
$$P_{a1} = P_1 - P_{v1} = 101.325 - 4.522 = 96.80\text{ kPa}$$
$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat}\,@\,24°\text{C}} = (0.60)(2.986\text{ kPa}) = 1.792\text{ kPa}$$
$$P_{a2} = P_2 - P_{v2} = 101.325 - 1.792 = 99.53\text{ kPa}$$

The entropy change of dry air is

$$\Delta S_a = m_a(s_2 - s_1) = m_a\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_{a2}}{P_{a1}}\right)$$
$$= (1105)\left[(1.005)\ln\frac{297}{305} - (0.287)\ln\frac{99.53}{96.80}\right] = -38.34 \text{ kJ/kg dry air}$$

The entropy change of R-134a in the evaporator is

$$\Delta S_{R,\,41} = m_R(s_1 - s_4) = (395\text{ kg})(0.92927 - 0.4045) = 207.3\text{ kJ/K}$$

An entropy balance on the evaporator gives

$$S_{\text{gen,evaporator}} = \Delta S_{R,41} + \Delta S_{\text{vapor}} + \Delta S_a + S_w = 207.3 + (-165.2) + (-38.34) + 8.14 = 11.90\text{ kJ/K}$$

Then, the exergy destruction in the evaporator is

$$X_{\text{dest}} = T_0 S_{\text{gen, evaporator}} = (305\text{ K})(11.90\text{ kJ/K}) = 3630\text{ kJ}$$

Finally the total exergy destruction is

$$X_{\text{dest, total}} = X_{\text{dest, compressor}} + X_{\text{dest, condenser}} + X_{\text{dest, throttle}} + X_{\text{dest, evaporator}}$$
$$= 0 + 1609 + 1519 + 3630$$
$$= \mathbf{6758\ kJ}$$

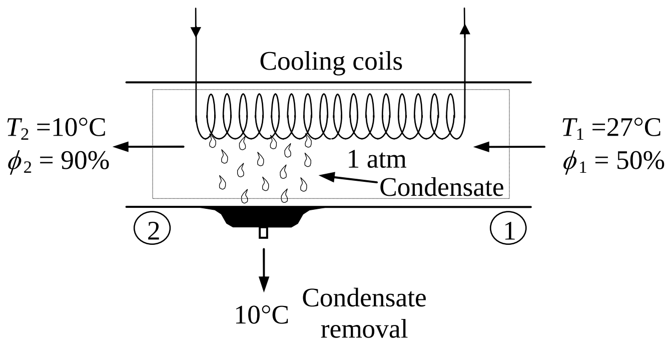
The greatest exergy destruction occurs in the evaporator. Note that heat is absorbed from humid air and rejected to the ambient air at 32°C (305 K), which is also taken as the dead state temperature.

14-92 Atmospheric air enters the evaporator of an automobile air conditioner at a specified pressure, temperature, and relative humidity. The dew point and wet bulb temperatures at the inlet to the evaporator section, the required heat transfer rate from the atmospheric air to the evaporator fluid, and the rate of condensation of water vapor in the evaporator section 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The 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 and exit states may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$T_{\text{dp1}} = \mathbf{15.7°C}$$
$$T_{\text{wb1}} = \mathbf{19.5°C}$$
$$h_1 = 55.60 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01115 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.8655 \text{ m}^3 / \text{kg dry air}$$
$$h_2 = 27.35 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.00686 \text{ kg H}_2\text{O/kg dry air}$$



The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{V_{\text{car}}\text{ACH}}{v_1} = \frac{(2 \text{ m}^3\text{/change})(5 \text{ changes/min})}{0.8655 \text{ m}^3} = 11.55 \text{ kg/min}$$

The mass flow rates of vapor at the inlet and exit are

$$\dot{m}_{v1} = \omega_1 \dot{m}_a = (0.01115)(11.55 \text{ kg/min}) = 0.1288 \text{ kg/min}$$

$$\dot{m}_{v2} = \omega_2 \dot{m}_a = (0.00686)(11.55 \text{ kg/min}) = 0.07926 \text{ kg/min}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$

where the the enthalpy of condensate water is

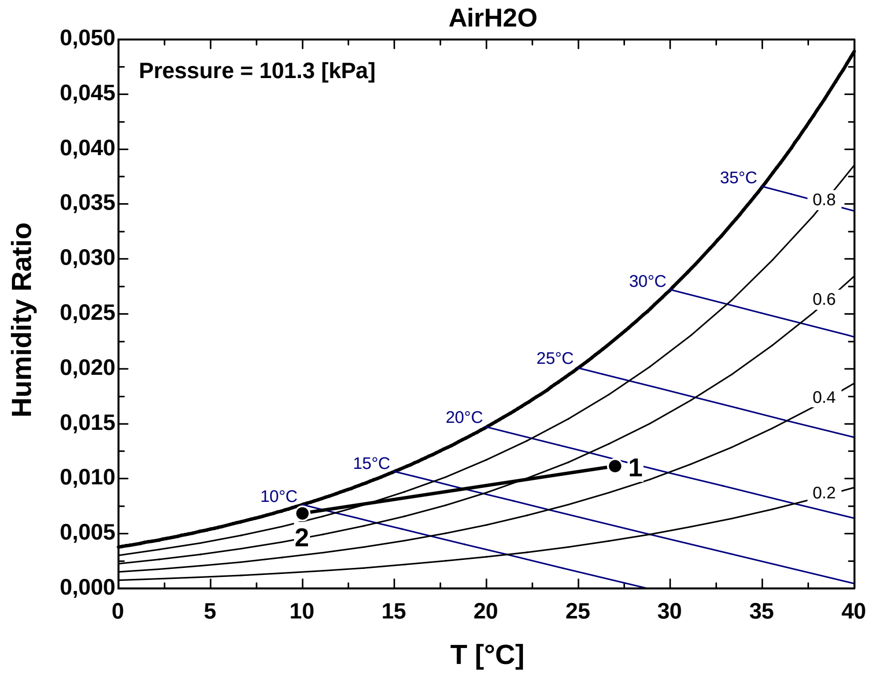
$$h_{w2} = h_{f@10°\text{C}} = 42.02 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

and the rate of condensation of water vapor is

$$\dot{m}_w = \dot{m}_{v1} - \dot{m}_{v2} = 0.1288 - 0.07926 = \mathbf{0.0495 \text{ kg/min}}$$

Substituting,

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(11.55 \text{ kg/min})(55.60 \text{ kJ/kg}) = \dot{Q}_{\text{out}} + (11.55 \text{ kg/min})(27.35 \text{ kJ/kg}) + (0.0495 \text{ kg/min})(42.02 \text{ kJ/kg})$$
$$\dot{Q}_{\text{out}} = 324.4 \text{ kJ/min} = \mathbf{5.41 \text{ kW}}$$

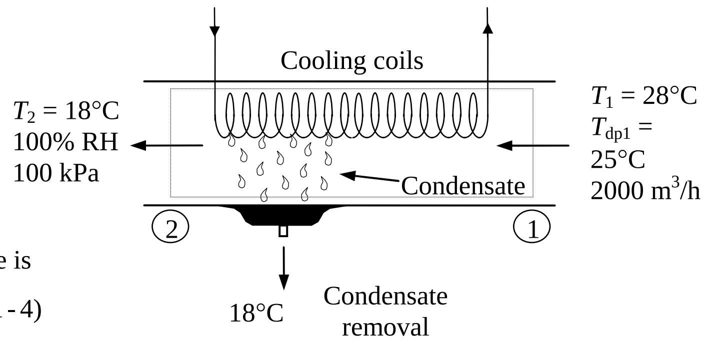

AirH2O
Pressure = 101.3 [kPa]
Humidity Ratio
T [°C]
0,050
0,045
0,040
0,035
0,030
0,025
0,020
0,015
0,010
0,005
0,000
0
5
10
15
20
25
30
35
40
35°C
30°C
25°C
20°C
15°C
10°C
0.8
0.6
0.4
0.2
1
2

14-93 Atmospheric air flows into an air conditioner that uses chilled water as the cooling fluid. The mass flow rate of the condensate water and the volume flow rate of chilled water supplied to the air conditioner 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis We may assume that the exit relative humidity is 100 percent since the exit temperature (18°C) is below the dew-point temperature of the inlet air (25°C). The properties of the air at the exit state may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$h_2 = 51.34 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.01311 \text{ kg H}_2\text{O/kg dry air}$$



The partial pressure of water vapor at the inlet state is (Table A-4)

$$P_{v1} = P_{\text{sat@ 25°C}} = 3.17 \text{ kPa}$$

The saturation pressure at the inlet state is

$$P_{g1} = P_{\text{sat@ 28°C}} = 3.783 \text{ kPa} \quad \text{(Table A - 4)}$$

Then, the relative humidity at the inlet state becomes

$$\phi_1 = \frac{P_{v1}}{P_{g1}} = \frac{3.17}{3.783} = 0.8379$$

Now, the inlet state is also fixed. The properties are obtained from EES to be

$$h_1 = 80.14 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.02036 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.8927 \text{ m}^3\text{/kg}$$

The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{(2000/60) \text{ m}^3\text{/h}}{0.8927 \text{ m}^3\text{/kg}} = 37.34 \text{ kg/min}$$

The mass flow rate of condensate water is

$$\begin{aligned}\dot{m}_w &= \dot{m}_a(\omega_1 - \omega_2) \\ &= (37.34 \text{ kg/min})(0.02036 - 0.01311) \\ &= 0.2707 \text{ kg/min} \\ &= \mathbf{16.24 \text{ kg/h}}\end{aligned}$$

The enthalpy of condensate water is

$$h_{w2} = h_{f@18°C} = 75.54 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

An energy balance on the control volume gives

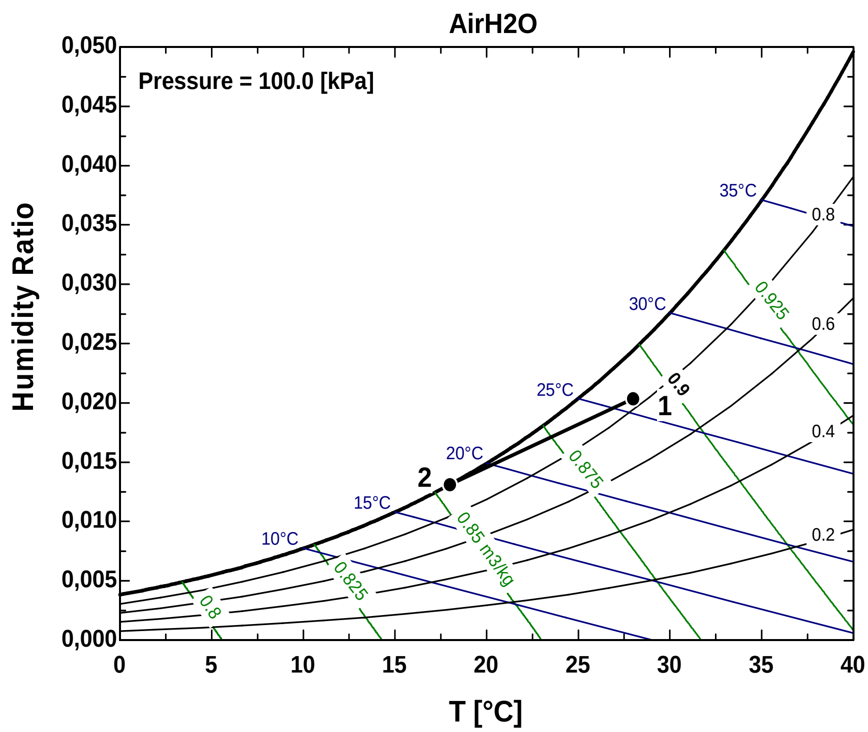
$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(37.34\text{ kg/min})(80.14\text{ kJ/kg}) = \dot{Q}_{\text{out}} + (37.34\text{ kg/min})(51.34\text{ kJ/kg}) + (0.2707\text{ kg/min})(75.54\text{ kJ/kg})$$
$$\dot{Q}_{\text{out}} = 1055\text{ kJ/min} = 17.59\text{ kW}$$

Noting that the rate of heat lost from the air is received by the cooling water, the mass flow rate of the cooling water is determined from

$$\dot{Q}_{\text{in}} = \dot{m}_{cw} c_p \Delta T_{cw} \longrightarrow \dot{m}_{cw} = \frac{\dot{Q}_{\text{in}}}{c_p \Delta T_{cw}} = \frac{1055\text{ kJ/min}}{(4.18\text{ kJ/kg.°C})(10\text{°C})} = 25.24\text{ kg/min}$$

where we used the specific heat of water value at room temperature. Assuming a density of 1000 kg/m^3 for water, the volume flow rate is determined to be

$$\dot{V}_{cw} = \frac{\dot{m}_{cw}}{\rho_{cw}} = \frac{25.24\text{ kg/min}}{1000\text{ kg/m}^3} = \mathbf{0.0252\ m^3/min}$$

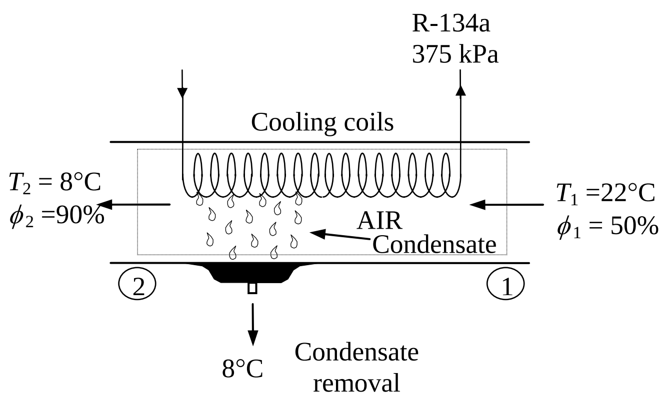


14-94 An automobile air conditioner using refrigerant 134a as the cooling fluid is considered. The inlet and exit states of moist air in the evaporator are specified. The volume flow rate of the air entering the evaporator of the air conditioner is to be determined.

Assumptions **1** All processes are steady flow and 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 We assume that the total pressure of moist air is 100 kPa. Then, the inlet and exit states of the moist air for the evaporator are completely specified. The properties may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 43.33\ \text{kJ/kg dry air}$$
$$\omega_1 = 0.008337\ \text{kg H}_2\text{O/kg dry air}$$
$$\boldsymbol{v}_1 = 0.8585\ \text{m}^3\text{/kg dry air}$$
$$h_2 = 23.31\ \text{kJ/kg dry air}$$
$$\omega_2 = 0.006065\ \text{kg H}_2\text{O/kg dry air}$$



The mass flow rate of dry air is given by

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{\dot{\boldsymbol{V}}_1}{0.8585\ \text{m}^3\text{/kg}}$$

The mass flow rate of condensate water is expressed as

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = \frac{\dot{\boldsymbol{V}}_1}{0.8585}(0.008337 - 0.006065) = 0.002646\dot{\boldsymbol{V}}_1$$

The enthalpy of condensate water is

$$h_{w2} = h_{f@\,8^\circ\text{C}} = 33.63\ \text{kJ/kg} \quad \text{(Table A-4)}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$\frac{\dot{\boldsymbol{V}}_1}{0.8585}(43.05) = \dot{Q}_{\text{out}} + \frac{\dot{\boldsymbol{V}}_1}{0.8585}(23.11) + 0.002646\dot{\boldsymbol{V}}_1(33.63) \tag{1}$$

The properties of the R-134a at the inlet of the compressor and the enthalpy at the exit for the isentropic process are (R-134a tables)

$$\left.\begin{array}{l} P_{R1} = 375\ \text{kPa} \\ x_{R1} = 1 \end{array}\right\} \begin{array}{l} h_{R1} = 254.48\ \text{kJ/kg} \\ s_{R1} = 0.9278\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_{R2} = 1800\ \text{kPa} \\ s_{R2} = s_{R1} \end{array}\right\} h_{R2,s} = 286.90\ \text{kJ/kg}$$

The enthalpies of R-134a at the condenser exit and the throttle exit are

$$h_{R3} = h_{f@\,1800\ \text{kPa}} = 144.07\ \text{kJ/kg}$$
$$h_{R4} = h_{R3} = 144.07\ \text{kJ/kg}$$

The mass flow rate of the refrigerant can be determined from the expression for the compressor power:

$$\dot{W}_C = \dot{m}_R \frac{h_{R2,s} - h_{R1}}{\eta_C}$$

$$6\text{ kW} = \dot{m}_R \frac{(286.90 - 254.48)\text{ kJ/kg}}{0.85}$$

$$\dot{m}_R = 0.1573\text{ kg/s} = 9.439\text{ kg/min}$$

The rate of heat absorbed by the R-134a in the evaporator is

$$\dot{Q}_{R,\text{in}} = \dot{m}_R (h_{R1} - h_{R4}) = (9.439\text{ kg/min})(254.48 - 144.07)\text{ kJ/kg} = 1042.1\text{ kJ/min}$$

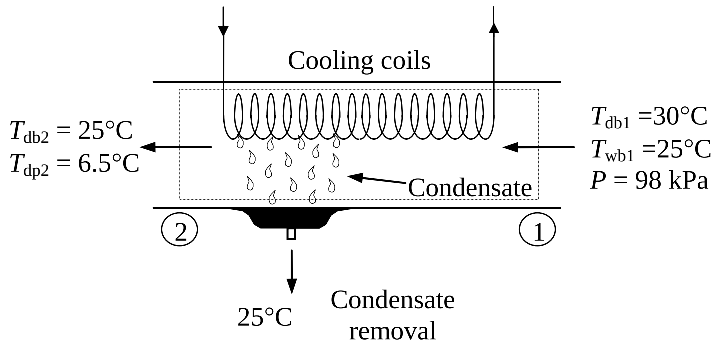
The rate of heat lost from the air in the evaporator is absorbed by the refrigerant-134a. That is, $\dot{Q}_{R,\text{in}} = \dot{Q}_{\text{out}}$. Then, the volume flow rate of the air at the inlet of the evaporator can be determined from Eq. (1) to be

$$\frac{\dot{V}_1}{0.8474}(43.05) = 1042.1 + \frac{\dot{V}_1}{0.8474}(23.11) + 0.002646 V_1 (33.63) \longrightarrow \dot{V}_1 = \mathbf{44.87\ m^3/min}$$

14-95 Air flows through an air conditioner unit. The inlet and exit states are specified. The rate of heat transfer and the mass flow rate of condensate water 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The inlet state of the air is completely specified, and the total pressure is 98 kPa. The properties of the air at the inlet state may be determined from (Fig. A-31) or using EES psychrometric functions to be (we used EES)



$$h_1 = 77.88 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01866 \text{ kg H}_2\text{O/kg dry air}$$
$$\phi_1 = 0.6721$$

The partial pressure of water vapor at the exit state is

$$P_{v2} = P_{\text{sat@ 6.5°C}} = 0.9682 \text{ kPa} \quad \text{(Table A-4)}$$

The saturation pressure at the exit state is

$$P_{g2} = P_{\text{sat@ 25°C}} = 3.17 \text{ kPa} \quad \text{(Table A-4)}$$

Then, the relative humidity at the exit state becomes

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{0.9682}{3.17} = 0.3054$$

Now, the exit state is also fixed. The properties are obtained from EES to be

$$h_2 = 40.97 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.006206 \text{ kg H}_2\text{O/kg dry air}$$
$$v_2 = 0.8820 \text{ m}^3\text{/kg}$$

The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_2}{v_2} = \frac{1000 \text{ m}^3\text{/min}}{0.8820 \text{ m}^3\text{/kg}} = 1133.8 \text{ kg/min}$$

The mass flow rate of condensate water is

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (1133.8 \text{ kg/min})(0.01866 - 0.006206) = 14.12 \text{ kg/min} = \mathbf{847.2\ kg/h}$$

The enthalpy of condensate water is

$$h_{w2} = h_{f@\,25°C} = 104.83 \text{ kJ/kg} \quad \text{(Table A-4)}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(1133.8 \text{ kg/min})(77.88 \text{ kJ/kg}) = \dot{Q}_{\text{out}} + (1133.8 \text{ kg/min})(40.97 \text{ kJ/kg}) + (14.12 \text{ kg/min})(104.83 \text{ kJ/kg})$$
$$\dot{Q}_{\text{out}} = 40{,}377 \text{ kJ/min} = \mathbf{672.9\ kW}$$

Evaporative Cooling

14-96C In steady operation, the mass transfer process does not have to involve heat transfer. However, a mass transfer process that involves phase change (evaporation, sublimation, condensation, melting etc.) must involve heat transfer. For example, the evaporation of water from a lake into air (mass transfer) requires the transfer of latent heat of water at a specified temperature to the liquid water at the surface (heat transfer).

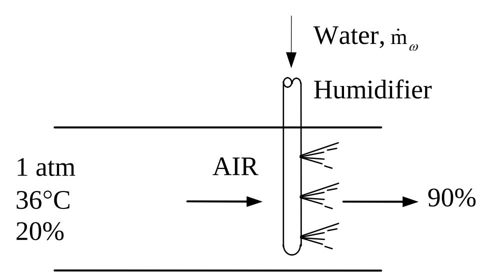
14-97C During evaporation from a water body to air, the latent heat of vaporization will be equal to *convection* heat transfer from the air when *conduction* from the lower parts of the water body to the surface is negligible, and temperature of the surrounding surfaces is at about the temperature of the water surface so that the *radiation* heat transfer is negligible.

14-98C Evaporative cooling is the cooling achieved when water evaporates in dry air. It will not work on humid climates.

14-99 Air is cooled by an evaporative cooler. The exit temperature of the air and the required rate of water supply are to be determined.

Analysis (*a*) From the psychrometric chart (Fig. A-31) at 36°C and 20% relative humidity we read

$$\begin{aligned} T_{\mathrm{wb1}} &= 19.5°\mathrm{C} \\ \omega_1 &= 0.0074 \text{ kg H}_2\text{O/kg dry air} \\ \boldsymbol{v}_1 &= 0.887 \text{ m}^3\text{/kg dry air} \end{aligned}$$



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature. That is,

$$T_{\mathrm{wb2}} \cong T_{\mathrm{wb1}} = 19.5°\mathrm{C}$$

At this wet-bulb temperature and 90% relative humidity we read

$$\begin{aligned} T_2 &= \mathbf{20.5°\,C} \\ \omega_2 &= 0.0137 \text{ kg H}_2\text{O / kg dry air} \end{aligned}$$

Thus air will be cooled to 20.5°C in this evaporative cooler.

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{4 \text{ m}^3\text{ / min}}{0.887 \text{ m}^3\text{ / kg dry air}} = 4.51 \text{ kg/min}$$

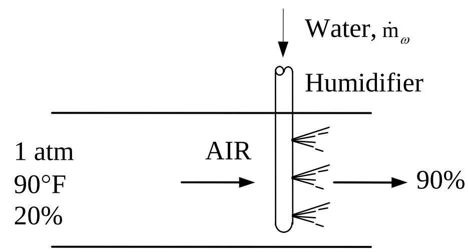
Then the required rate of water supply to the evaporative cooler is determined from

$$\begin{aligned} \dot{m}_{\text{supply}} &= \dot{m}_{w2} - \dot{m}_{w1} = \dot{m}_a(\omega_2 - \omega_1) \\ &= (4.51 \text{ kg/min})(0.0137 - 0.0074) = \mathbf{0.028\,kg/min} \end{aligned}$$

14-100E Air is cooled by an evaporative cooler. The exit temperature of the air and the required rate of water supply are to be determined.

Analysis (*a*) From the psychrometric chart (Fig. A-31E) at 90°F and 20% relative humidity we read

$$T_{\text{wb1}} = 62.8°\text{F}$$
$$\omega_1 = 0.0060 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_1 = 14.0 \text{ ft}^3\text{/lbm dry air}$$



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature. That is,

$$T_{\text{wb2}} \cong T_{\text{wb1}} = 62.8°\text{F}$$

At this wet-bulb temperature and 90% relative humidity we read

$$T_2 = \mathbf{65°F}$$
$$\omega_2 = 0.0116 \text{ lbm H}_2\text{O/lbm dry air}$$

Thus air will be cooled to 64°F in this evaporative cooler.

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{150 \text{ ft}^3 \text{ / min}}{14.0 \text{ ft}^3 \text{ / lbm dry air}} = 10.7 \text{ lbm/min}$$

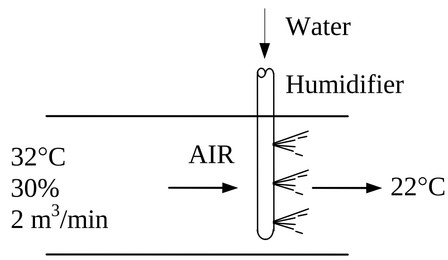
Then the required rate of water supply to the evaporative cooler is determined from

$$\dot{m}_{\text{supply}} = \dot{m}_{w2} - \dot{m}_{w1} = \dot{m}_a(\omega_2 - \omega_1) = (10.7 \text{ lbm/min})(0.0116 - 0.0060) = \mathbf{0.06\ \ lbm/min}$$

14-101 Air is cooled by an evaporative cooler. The final relative humidity and the amount of water added are to be determined.

Analysis (*a*) From the psychrometric chart (Fig. A-31) at 32°C and 30% relative humidity we read

$$T_{wb1} = 19.4°C$$
$$\omega_1 = 0.0089 \text{ kg } H_2O/\text{kg dry air}$$
$$v_1 = 0.877 \text{ m}^3/\text{kg dry air}$$



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature. That is,

$$T_{wb2} \cong T_{wb1} = 19.4°C$$

At this wet-bulb temperature and 22°C temperature we read

$$\phi_2 = \mathbf{79\%}$$
$$\omega_2 = 0.0130 \text{ kg } H_2O/\text{kg dry air}$$

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{5 \text{ m}^3/\text{min}}{0.877 \text{ m}^3/\text{kg dry air}} = 5.70 \text{ kg/min}$$

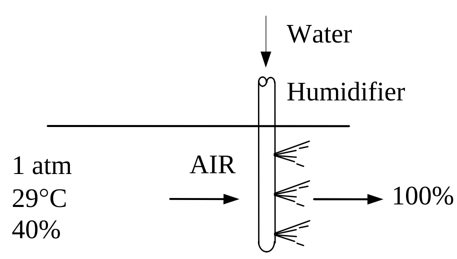
Then the required rate of water supply to the evaporative cooler is determined from

$$\dot{m}_{supply} = \dot{m}_{w2} - \dot{m}_{w1} = \dot{m}_a(\omega_2 - \omega_1) = (5.70 \text{ kg/min})(0.0130 - 0.0089) = \mathbf{0.0234 \text{ kg/min}}$$

14-102 Air enters an evaporative cooler at a specified state and relative humidity. The lowest temperature that air can attain is to be determined.

Analysis From the psychrometric chart (Fig. A-31) at 29°C and 40% relative humidity we read

$$T_{wb1} = 19.3°C$$



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature, which is the lowest temperature that can be obtained in an evaporative cooler. That is,

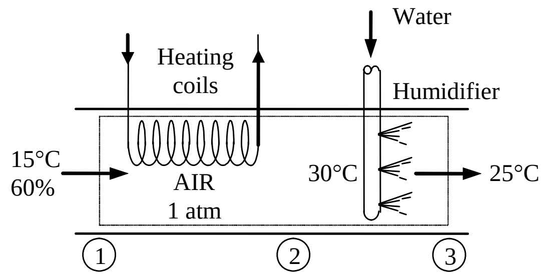
$$T_{min} = T_{wb1} = \mathbf{19.3°C}$$

14-103 Air is first heated in a heating section and then passed through an evaporative cooler. The exit relative humidity and the amount of water added are to be determined.

Analysis (*a*) From the psychrometric chart (Fig. A-31) at 15°C and 60% relative humidity we read

$$\omega_1 = 0.00635 \text{ kg H}_2\text{O / kg dry air}$$

The specific humidity ω remains constant during the heating process. Therefore, $\omega_2 = \omega_1 = 0.00635$ kg H_2O / kg dry air. At this ω value and 30°C we read T_{wb2} = 16.7°C.



Assuming the liquid water is supplied at a temperature not much different than the exit temperature of the air stream, the evaporative cooling process follows a line of constant wet-bulb temperature. That is, $T_{wb3} \cong T_{wb2}$ = 16.7°C. At this T_{wb} value and 25°C we read

$$\phi_3 = \mathbf{42.6\%}$$
$$\omega_3 = 0.00840 \text{ kg H}_2\text{O/kg dry air}$$

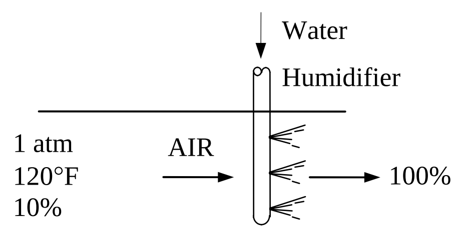
(*b*) The amount of water added to the air per unit mass of air is

$$\Delta\omega_{23} = \omega_3 - \omega_2 = 0.00840 - 0.00635 = \mathbf{0.00205 \ \ kg\,H_2O/kg\,dry\,air}$$

14-104E Desert dwellers often wrap their heads with a water-soaked porous cloth. The temperature of this cloth on a desert with specified temperature and relative humidity is to be determined.

Analysis Since the cloth behaves as the wick on a wet bulb thermometer, the temperature of the cloth will become the wet-bulb temperature. According to the pshchrometric chart, this temperature is

$$T_2 = T_{wb1} = \mathbf{73.7°F}$$



This process can be represented by an evaporative cooling process as shown in the figure.

Adiabatic Mixing of Airstreams

14-105C This will occur when the straight line connecting the states of the two streams on the psychrometric chart crosses the saturation line.

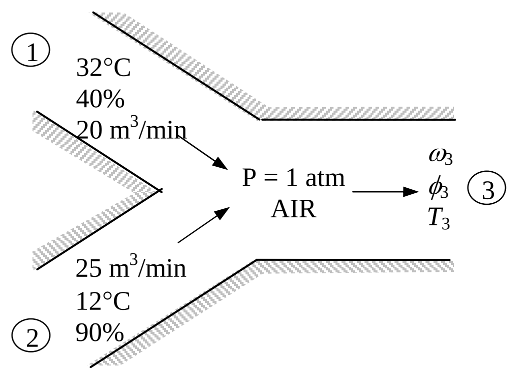
14-106C Yes.

14-107 Two airstreams are mixed steadily. The specific humidity, the relative humidity, the dry-bulb temperature, and the volume 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 Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 62.7 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0119 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.882 \text{ m}^3\text{/kg dry air}$$



and

$$h_2 = 31.9 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0079 \text{ kg H}_2\text{O/kg dry air}$$
$$v_2 = 0.819 \text{ m}^3\text{/kg dry air}$$

Analysis The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{20 \text{ m}^3\text{/min}}{0.882 \text{ m}^3\text{/kg dry air}} = 22.7 \text{ kg/min}$$

$$\dot{m}_{a2} = \frac{\dot{V}_2}{v_2} = \frac{25 \text{ m}^3\text{/min}}{0.819 \text{ m}^3\text{/kg dry air}} = 30.5 \text{ kg/min}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (22.7 + 30.5) \text{ kg/min} = 53.2 \text{ kg/min}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{22.7}{30.5} = \frac{0.0079 - \omega_3}{\omega_3 - 0.0119} = \frac{31.9 - h_3}{h_3 - 62.7}$$

which yields,

$$\omega_3 = \mathbf{0.0096 \text{ kg H}_2\text{O / kg dry air}}$$
$$h_3 = 45.0 \text{ kJ/kg dry air}$$

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

$$T_3 = \mathbf{20.6°C}$$
$$\phi_3 = \mathbf{63.4\%}$$
$$v_3 = 0.845 \text{ m}^3\text{/kg dry air}$$

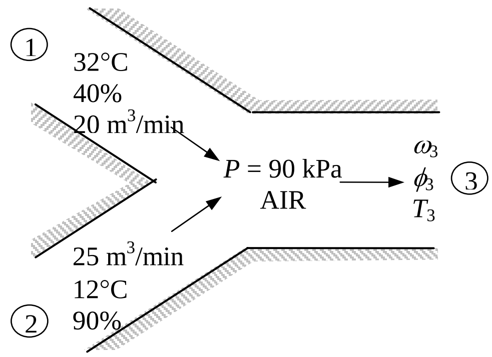
Finally, the volume flow rate of the mixture is determined from

$$\dot{V}_3 = \dot{m}_{a3} v_3 = (53.2 \text{ kg/min})(0.845 \text{ m}^3\text{/kg}) = \mathbf{45.0 \text{ m}^3\text{/min}}$$

14-108 Two airstreams are mixed steadily. The specific humidity, the relative humidity, the dry-bulb temperature, and the volume 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.

Analysis The properties of each inlet stream are determined to be



$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 32°C}} = (0.40)(4.760\text{ kPa}) = 1.90\text{ kPa}$$
$$P_{a1} = P_1 - P_{v1} = 90 - 1.90 = 88.10\text{ kPa}$$
$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(305\text{ K})}{88.10\text{ kPa}}$$
$$= 0.994\text{ m}^3/\text{kg dry air}$$
$$\omega_1 = \frac{0.622 P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.90\text{ kPa})}{(90-1.90)\text{ kPa}}$$
$$= 0.0134\text{ kg H}_2\text{O/kg dry air}$$
$$h_1 = c_p T_1 + \omega_1 h_{g1}$$
$$= (1.005\text{ kJ/kg}\cdot°\text{C})(32°\text{C}) + (0.0134)(2559.2\text{ kJ/kg})$$
$$= 66.45\text{ kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat@12°C}} = (0.90)(1.403\text{ kPa}) = 1.26\text{ kPa}$$
$$P_{a2} = P_2 - P_{v2} = 90 - 1.26 = 88.74\text{ kPa}$$
$$\boldsymbol{v}_2 = \frac{R_a T_2}{P_{a2}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(285\text{ K})}{88.74\text{ kPa}} = 0.922\text{ m}^3/\text{kg dry air}$$
$$\omega_2 = \frac{0.622 P_{v2}}{P_2 - P_{v2}} = \frac{0.622(1.26\text{ kPa})}{(90-1.26)\text{ kPa}} = 0.00883\text{ kg H}_2\text{O/kg dry air}$$
$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot°\text{C})(12°\text{C}) + (0.00883)(2522.9\text{ kJ/kg}) = 34.34\text{ kJ/kg dry air}$$

Then the mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{20\text{ m}^3/\text{min}}{0.994\text{ m}^3/\text{kg dry air}} = 20.12\text{ kg/min}$$
$$\dot{m}_{a2} = \frac{\dot{\boldsymbol{V}}_2}{\boldsymbol{v}_2} = \frac{25\text{ m}^3/\text{min}}{0.922\text{ m}^3/\text{kg dry air}} = 27.11\text{ kg/min}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (20.12 + 27.11)\text{ kg/min} = 47.23\text{ kg/min}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1} \longrightarrow \frac{20.12}{27.11} = \frac{0.00883 - \omega_3}{\omega_3 - 0.0134} = \frac{34.34 - h_3}{h_3 - 66.45}$$

which yields

$$\omega_3 = \mathbf{0.0108\text{ kg H}_2\text{O/kg dry air}}$$
$$h_3 = 48.02\text{ kJ/kg dry air}$$

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

$$h_3 = c_p T_3 + \omega_3 h_{g3} \cong c_p T_3 + \omega_3 (2501.3 + 1.82 T_3)$$

$$48.02 \text{ kJ/kg} = (1.005 \text{ kJ/kg}\cdot{}^\circ\text{C}) T_3 + (0.0108)(2500.9 + 1.82 T_3) \text{ kJ/kg} \longrightarrow T_3 = \mathbf{20.5^\circ C}$$

$$\omega_3 = \frac{0.622\, P_{v3}}{P_3 - P_{v3}}$$

$$0.0108 = \frac{0.622\, P_{v3}}{90 - P_{v3}} \longrightarrow P_{v3} = 1.54 \text{ kPa}$$

$$\phi_3 = \frac{P_{v3}}{P_{g3}} = \frac{P_{v3}}{P_{\text{sat @ } T_3}} = \frac{1.54 \text{ kPa}}{2.41 \text{ kPa}} = 0.639 \quad \text{or} \quad \mathbf{63.9\%}$$

Finally,

$$P_{a3} = P_3 - P_{v3} = 90 - 1.54 = 88.46 \text{ kPa}$$

$$\boldsymbol{v}_3 = \frac{R_a T_3}{P_{a3}} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293.5 \text{ K})}{88.46 \text{ kPa}} = 0.952 \text{ m}^3\text{/kg dry air}$$

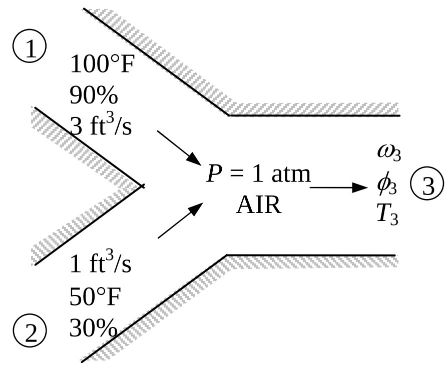
$$\dot{\boldsymbol{V}}_3 = \dot{m}_{a3} \boldsymbol{v}_3 = (47.23 \text{ kg/min})(0.952 \text{ m}^3/\text{kg}) = \mathbf{45.0\ m^3/min}$$

14-109E Two airstreams are mixed steadily. The temperature and the relative humidity 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 Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31E or from EES) to be

$$h_1 = 66.7 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0386 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_1 = 14.98 \text{ ft}^3\text{/lbm dry air}$$



and

$$h_2 = 14.5 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0023 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_2 = 12.90 \text{ ft}^3\text{/lbm dry air}$$

Analysis The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{3 \text{ ft}^3\text{ / s}}{14.98 \text{ ft}^3\text{ / lbmdry air}} = 0.2002 \text{ lbm/s}$$

$$\dot{m}_{a2} = \frac{\dot{V}_2}{v_2} = \frac{1 \text{ ft}^3\text{ / s}}{12.90 \text{ ft}^3\text{ / lbm dry air}} = 0.07755 \text{ lbm/s}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (0.2002 + 0.07755) \text{ lbm/s} = 0.2778 \text{ lbm/s}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{0.2002}{0.07755} = \frac{0.0023 - \omega_3}{\omega_3 - 0.0386} = \frac{14.5 - h_3}{h_3 - 66.7}$$

which yields

$$\omega_3 = 0.0284 \text{ lbm H}_2\text{O/lbm dry air}$$
$$h_3 = 52.1 \text{ Btu/lbm dry air}$$

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

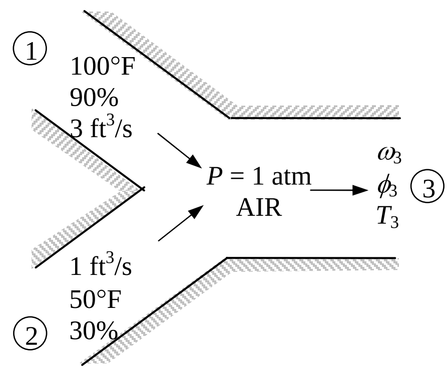
$$T_3 = \mathbf{86.7°F}$$
$$\phi_3 = 1.0 = \mathbf{100\%}$$

14-110E Two airstreams are mixed steadily. The rate of entropy generation is 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 Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31 or from EES) to be

$$h_1 = 66.7 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0386 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_1 = 14.98 \text{ ft}^3\text{/lbm dry air}$$



and

$$h_2 = 14.5 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0023 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_2 = 12.90 \text{ ft}^3\text{/lbm dry air}$$

The entropies of water vapor in the air streams are

$$s_{g1} = s_{g\,@\,100°\text{F}} = 1.9819 \text{ Btu/lbm}\cdot\text{R}$$
$$s_{g2} = s_{g\,@\,50°\text{F}} = 2.1256 \text{ Btu/lbm}\cdot\text{R}$$

Analysis The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{3 \text{ ft}^3\text{ / s}}{14.98 \text{ ft}^3\text{ / lbmdry air}} = 0.2002 \text{ lbm/s}$$

$$\dot{m}_{a2} = \frac{\dot{V}_2}{v_2} = \frac{1 \text{ ft}^3\text{ / s}}{12.90 \text{ ft}^3\text{ / lbm dry air}} = 0.07755 \text{ lbm/s}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (0.2002 + 0.07755) \text{ lbm/s} = 0.2778 \text{ lbm/s}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$
$$\frac{0.2002}{0.07755} = \frac{0.0023 - \omega_3}{\omega_3 - 0.0386} = \frac{14.5 - h_3}{h_3 - 66.7}$$

which yields

$$\omega_3 = 0.0284 \text{ lbm H}_2\text{O/lbm dry air}$$
$$h_3 = 52.1 \text{ Btu/lbm dry air}$$

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

$$T_3 = 86.7°\text{F}$$
$$\phi_3 = 1.0 = 100\%$$

The entropy of water vapor in the mixture is

$$s_{g3} = s_{g\,@\,86.7°\mathrm{F}} = 2.0168\ \mathrm{Btu/lbm\cdot R}$$

An entropy balance on the mixing chamber for the water gives

$$\begin{aligned}\Delta\dot{S}_w &= \dot{m}_{a3}\omega_3 s_3 - \dot{m}_{a1}\omega_1 s_1 - \dot{m}_{a2}\omega_2 s_2\\ &= 0.2778\times 0.0284\times 2.0168 - 0.2002\times 0.0386\times 1.9819 - 0.07755\times 0.0023\times 2.1256\\ &= 2.169\times 10^{-4}\ \mathrm{Btu/s\cdot R}\end{aligned}$$

The partial pressures of water vapor and dry air for all three air streams are

$$\begin{aligned}P_{v1} &= \phi_1 P_{g1} = \phi_1 P_{\mathrm{sat}\,@\,100°\mathrm{F}} = (0.90)(0.95052\ \mathrm{psia}) = 0.8555\ \mathrm{psia}\\ P_{a1} &= P_1 - P_{v1} = 14.696 - 0.8555 = 13.84\ \mathrm{psia}\\ P_{v2} &= \phi_2 P_{g2} = \phi_2 P_{\mathrm{sat}\,@\,50°\mathrm{F}} = (0.30)(0.17812\ \mathrm{psia}) = 0.0534\ \mathrm{psia}\\ P_{a2} &= P_2 - P_{v2} = 14.696 - 0.0534 = 14.64\ \mathrm{psia}\\ P_{v3} &= \phi_3 P_{g3} = \phi_3 P_{\mathrm{sat}\,@\,86.7°\mathrm{C}} = (1.0)(0.6298\ \mathrm{psia}) = 0.6298\ \mathrm{psia}\\ P_{a3} &= P_3 - P_{v3} = 14.696 - 0.6298 = 14.07\ \mathrm{psia}\end{aligned}$$

An entropy balance on the mixing chamber for the dry air gives

$$\begin{aligned}\Delta\dot{S}_a &= \dot{m}_{a1}(s_3 - s_1) + \dot{m}_{a2}(s_3 - s_2)\\ &= \dot{m}_{a1}\left(c_p \ln\frac{T_3}{T_1} - R\ln\frac{P_{a3}}{P_{a1}}\right) + \dot{m}_{a2}\left(c_p \ln\frac{T_3}{T_2} - R\ln\frac{P_{a3}}{P_{a2}}\right)\\ &= 0.2002\left[(0.240)\ln\frac{546.7}{560} - (0.06855)\ln\frac{14.07}{13.84}\right] + 0.07755\left[(0.240)\ln\frac{546.7}{510} - (0.06855)\ln\frac{14.07}{14.64}\right]\\ &= (0.2002)(-0.006899) + (0.07755)(0.01940) = 1.233\times 10^{-4}\ \mathrm{Btu/s\cdot R}\end{aligned}$$

The rate of entropy generation is then

$$\dot{S}_{\mathrm{gen}} = \Delta\dot{S}_a + \Delta\dot{S}_w = 1.233\times 10^{-4} + 2.169\times 10^{-4} = \mathbf{3.402\times 10^{-4}\ Btu/s\cdot R}$$

14-111 Two airstreams are mixed steadily. The mass flow ratio of the two streams for a specified mixture relative humidity and the temperature of the mixture are to be determined.

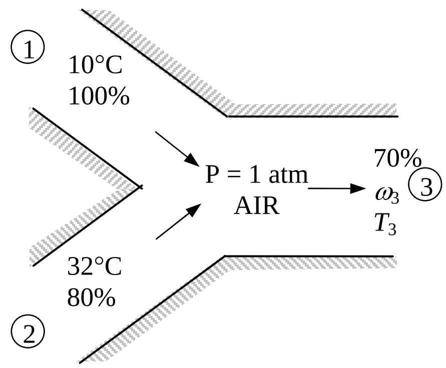
Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31 or from EES) to be

$$h_1 = 29.4 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0077 \text{ kg H}_2\text{O/kg dry air}$$

and

$$h_2 = 94.6 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0244 \text{ kg H}_2\text{O/kg dry air}$$



Analysis An application of Eq. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams gives

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{0.0244 - \omega_3}{\omega_3 - 0.0077} = \frac{94.6 - h_3}{h_3 - 29.4}$$

This equation cannot be solved directly. An iterative solution is needed. A mixture temperature T_3 is selected. At this temperature and given relative humidity (70%), specific humidity and enthalpy are read from the psychrometric chart. These values are substituted into the above equation. If the equation is not satisfied, a new value of T_3 is selected. This procedure is repeated until the equation is satisfied. Alternatively, EES software can be used. We used the following EES program to get the results:

```
"Given"
P=101.325 [kPa]
T_1=10 [C]
phi_1=1.0
T_2=32 [C]
phi_2=0.80
phi_3=0.70
"Analysis"
Fluid$='AirH2O'
"1st stream properties"
h_1=enthalpy(Fluid$, T=T_1, P=P, R=phi_1)
w_1=humrat(Fluid$, T=T_1, P=P, R=phi_1)
"2nd stream properties"
h_2=enthalpy(Fluid$, T=T_2, P=P, R=phi_2)
w_2=humrat(Fluid$, T=T_2, P=P, R=phi_2)
(w_2-w_3)/(w_3-w_1)=(h_2-h_3)/(h_3-h_1)
Ratio=(w_2-w_3)/(w_3-w_1)
"mixture properties"
T_3=temperature(Fluid$, h=h_3, P=P, R=phi_3)
h_3=enthalpy(Fluid$, T=T_3, P=P, R=phi_3)
```

The solution of this EES program is

$$T_3 = \mathbf{24.0°C}, \quad \omega_3 = 0.0149 \text{ kg H}_2\text{O/kg dry air}$$

$$h_3 = 57.6 \text{ kJ/kg dry air}, \quad \frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \mathbf{1.31}$$

14-112 A stream of warm air is mixed with a stream of saturated cool air. The temperature, the specific humidity, and the relative humidity of the mixture are to be determined.

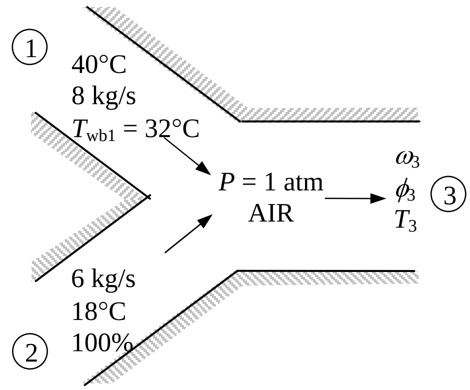
Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties The properties of each inlet stream are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 110.2 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0272 \text{ kg } H_2O\text{/kg dry air}$$

and

$$h_2 = 50.9 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0129 \text{ kg } H_2O\text{/kg dry air}$$



Analysis The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{8.0}{6.0} = \frac{0.0129 - \omega_3}{\omega_3 - 0.0272} = \frac{50.9 - h_3}{h_3 - 110.2}$$

which yields,

(*b*) $\omega_3 = \mathbf{0.0211 \text{ kg } H_2O / \text{kg dry air}}$

$h_3 = 84.8 \text{ kJ / kg dry air}$

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

(*a*) $T_3 = \mathbf{30.7°C}$

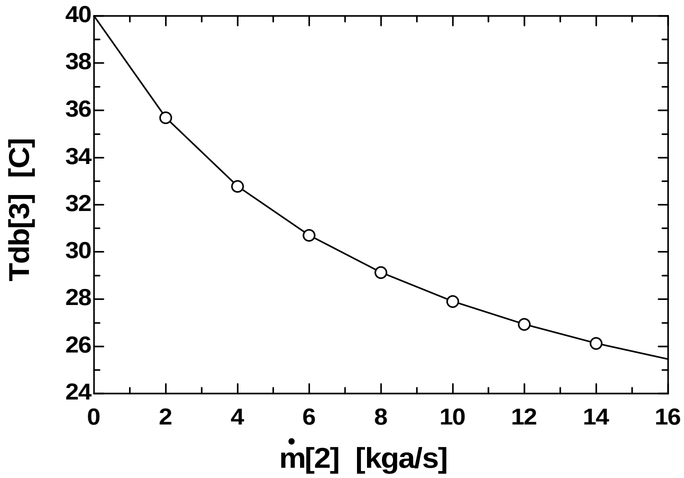
(*c*) $\phi_3 = \mathbf{75.1\%}$

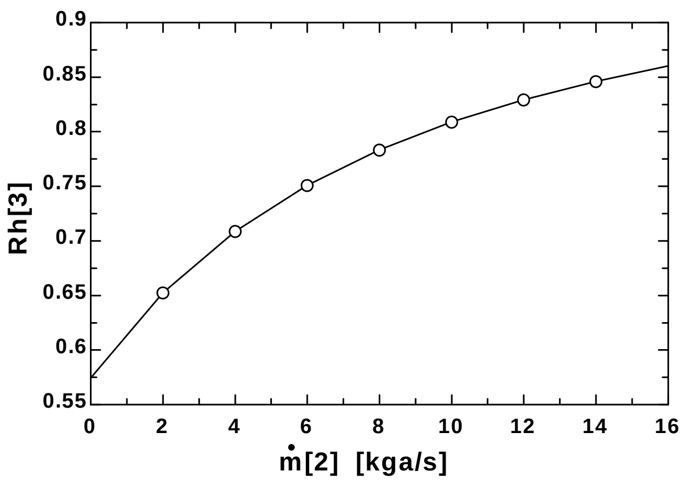
14-113 EES Problem 14-112 is reconsidered. The effect of the mass flow rate of saturated cool air stream on the mixture temperature, specific humidity, and relative humidity is to be investigated.

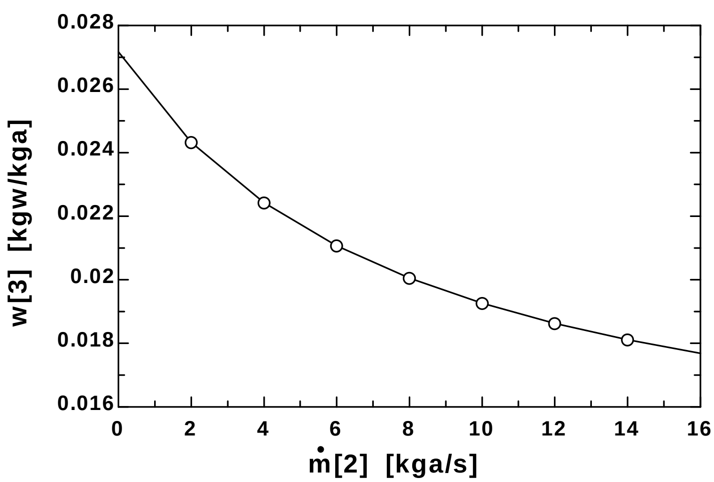
Analysis The problem is solved using EES, and the solution is given below.

```
P=101.325 [kPa]
Tdb[1] =40 [C]
Twb[1] =32 [C]
m_dot[1] = 8 [kg/s]
Tdb[2] =18 [C]
Rh[2] = 1.0
m_dot[2] = 6 [kg/s]
P[1]=P
P[2]=P[1]
P[3]=P[1]

"Energy balance for the steady-flow mixing process:"
"We neglect the PE of the flow. Since we don't know the cross sectional area of the flow streams,
we also neglect theKE of the flow."
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 [kW]
E_dot_in = m_dot[1]*h[1]+m_dot[2]*h[2]
E_dot_out = m_dot[3]*h[3]
"Conservation of mass of dry air during mixing:"
m_dot[1]+m_dot[2] = m_dot[3]
"Conservation of mass of water vapor during mixing:"
m_dot[1]*w[1]+m_dot[2]*w[2] = m_dot[3]*w[3]
m_dot[1]=V_dot[1]/v[1]*convert(1/min,1/s)
m_dot[2]=V_dot[2]/v[2]*convert(1/min,1/s)
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],B=Twb[1])
Rh[1]=RELHUM(AirH2O,T=Tdb[1],P=P[1],B=Twb[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
v[2]=VOLUME(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
w[2]=HUMRAT(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
Tdb[3]=TEMPERATURE(AirH2O,h=h[3],P=P[3],w=w[3])
Rh[3]=RELHUM(AirH2O,T=Tdb[3],P=P[3],w=w[3])
v[3]=VOLUME(AirH2O,T=Tdb[3],P=P[3],w=w[3])
Twb[2]=WETBULB(AirH2O,T=Tdb[2],P=P[2],R=RH[2])
Twb[3]=WETBULB(AirH2O,T=Tdb[3],P=P[3],R=RH[3])
m_dot[3]=V_dot[3]/v[3]*convert(1/min,1/s)
```

m_2 [kga/s]	Tdb_3 [C]	Rh_3	w_3 [kgw/kga]
0	40	0.5743	0.02717
2	35.69	0.6524	0.02433
4	32.79	0.7088	0.02243
6	30.7	0.751	0.02107
8	29.13	0.7834	0.02005
10	27.91	0.8089	0.01926
12	26.93	0.8294	0.01863
14	26.13	0.8462	0.01811
16	25.45	0.8601	0.01768


40
38
36
34
32
30
28
26
24
Tdb[3] [C]
0
2
4
6
8
10
12
14
16
ṁ[2] [kga/s]


0.9
0.85
0.8
0.75
0.7
0.65
0.6
0.55
Rh[3]
0
2
4
6
8
10
12
14
16
ṁ[2] [kga/s]


0.028
0.026
0.024
0.022
0.02
0.018
0.016
w[3] [kgw/kga]
0
2
4
6
8
10
12
14
16
ṁ[2] [kga/s]

Wet Cooling Towers

14-114C The working principle of a natural draft cooling tower is based on buoyancy. The air in the tower has a high moisture content, and thus is lighter than the outside air. This light moist air rises under the influence of buoyancy, inducing flow through the tower.

14-115C A spray pond cools the warm water by spraying it into the open atmosphere. They require 25 to 50 times the area of a wet cooling tower for the same cooling load.

14-116 Water is cooled by air in a cooling tower. The volume flow rate of air and the mass flow rate of the required makeup water are to be determined.

Assumptions **1** Steady operating conditions exist and thus 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. **4** The cooling tower is adiabatic.

Analysis (*a*) The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but 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. Applying the mass and energy balances yields

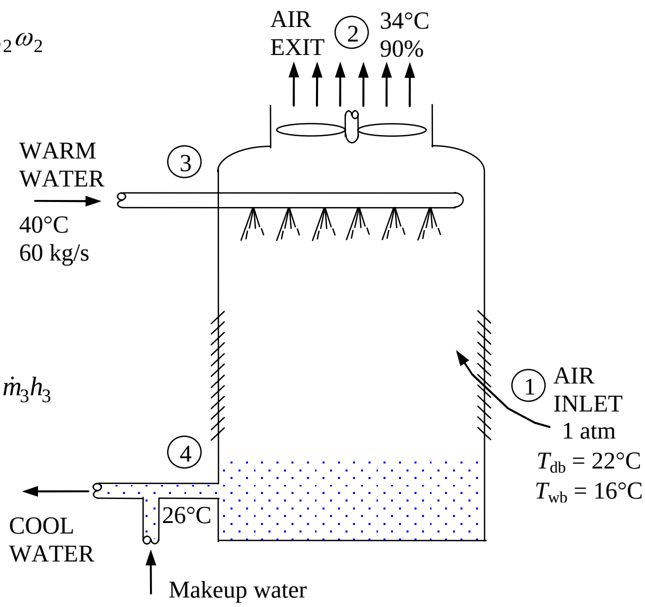
Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \to \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

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



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2}h_2 + \dot{m}_4 h_4 - \dot{m}_{a1}h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$

Solving for $\dot{m}_a$,

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

From the psychrometric chart (Fig. A-31),

$$h_1 = 44.7 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.0089 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.849 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 113.5 \text{ kJ / kg dry air}$$

$$\omega_2 = 0.0309 \text{ kg H}_2\text{O / kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,40°\text{C}} = 167.53 \text{ kJ/kg H}_2\text{O}$$

$$h_4 \cong h_{f\,@\,26°\text{C}} = 109.01 \text{ kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(60 \text{ kg/s})(167.53 - 109.01)\text{kJ/kg}}{(113.5 - 44.7) \text{ kJ/kg} - (0.0309 - 0.0089)(109.01) \text{ kJ/kg}} = 52.9 \text{ kg/s}$$

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

$$\dot{V}_1 = \dot{m}_a v_1 = (52.9 \text{ kg/s})(0.849 \text{ m}^3 \text{ / kg}) = \mathbf{44.9\ m^3/s}$$

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

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (52.9 \text{ kg / s})(0.0309 - 0.0089) = \mathbf{1.16\ kg\,/\,s}$$

14-117 Water is cooled by air in a cooling tower. The mass flow rate of dry air is to be determined.

Assumptions **1** Steady operating conditions exist and thus 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. **4** The cooling tower is adiabatic.

Analysis The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but 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. Applying the mass and energy balances yields

Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

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

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$

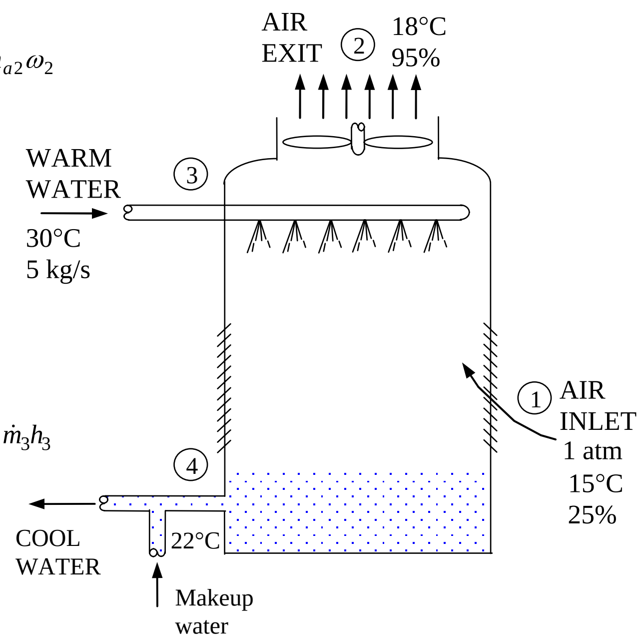
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a (h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}}) h_4 - \dot{m}_3 h_3$$



Solving for $\dot{m}_a$,

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

From the psychrometric chart (Fig. A-31),

$$h_1 = 21.8 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.00264 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.820 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 49.3 \text{ kJ/kg dry air}$$

$$\omega_2 = 0.0123 \text{ kg H}_2\text{O/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,30°\text{C}} = 125.74 \text{ kJ/kg H}_2\text{O}$$

$$h_4 \cong h_{f\,@\,22°\text{C}} = 92.28 \text{ kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(5 \text{ kg/s})(125.74 - 92.28)\text{kJ/kg}}{(49.3 - 21.8) \text{ kJ/kg} - (0.0123 - 0.00264)(92.28) \text{ kJ/kg}} = \mathbf{6.29\ kg/s}$$

14-118 Water is cooled by air in a cooling tower. The exergy lost in the cooling tower is to be determined.

Assumptions **1** Steady operating conditions exist and thus 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. **4** The cooling tower is adiabatic.

Analysis The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but 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. Applying the mass and energy balances yields

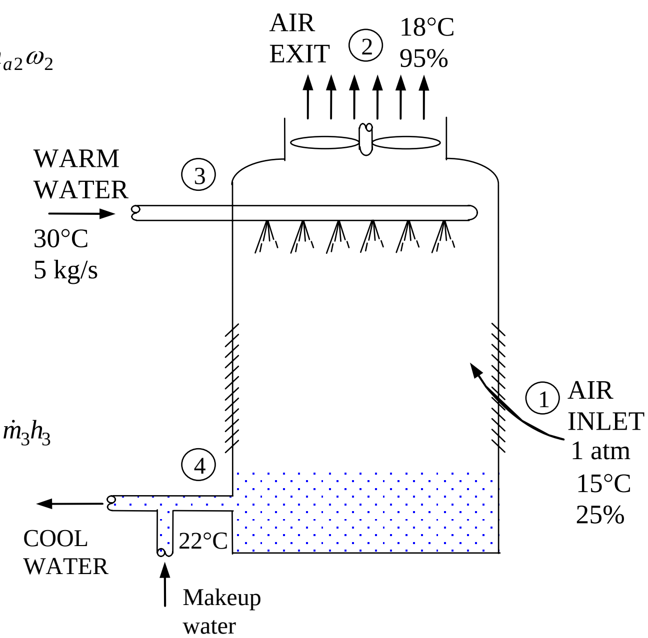
Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

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



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2}h_2 + \dot{m}_4 h_4 - \dot{m}_{a1}h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$

Solving for $\dot{m}_a$,

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

From the psychrometric chart (Fig. A-31),

$$h_1 = 21.8 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.00264 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.820 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 49.3 \text{ kJ/kg dry air}$$

$$\omega_2 = 0.0123 \text{ kg H}_2\text{O/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,30°\text{C}} = 125.74 \text{ kJ/kg H}_2\text{O}$$

$$h_4 \cong h_{f\,@\,22°\text{C}} = 92.28 \text{ kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(5\text{ kg/s})(125.74 - 92.28)\text{kJ/kg}}{(49.3 - 21.8)\text{ kJ/kg} - (0.0123 - 0.00264)(92.28)\text{ kJ/kg}} = 6.29 \text{ kg/s}$$

The mass flow rate of water stream at state 3 per unit mass of dry air is

$$m_3 = \frac{\dot{m}_3}{\dot{m}_a} = \frac{5 \text{ kg water/s}}{6.29 \text{ kg dry air/s}} = 0.7949 \text{ kg water/kg dry air}$$

The mass flow rate of water stream at state 4 per unit mass of dry air is

$$m_4 = m_3 - (\omega_2 - \omega_1) = 0.7949 - (0.0123 - 0.00264) = 0.7852 \text{ kg water/kg dry air}$$

The entropies of water streams are

$$s_3 = s_{f\,@\,30°\text{C}} = 0.4368 \text{ kJ/kg}\cdot\text{K}$$
$$s_4 = s_{f\,@\,22°\text{C}} = 0.3249 \text{ kJ/kg}\cdot\text{K}$$

The entropy change of water stream is

$$\Delta s_{\text{water}} = m_4 s_4 - m_3 s_3 = 0.7852 \times 0.3249 - 0.7949 \times 0.4368 = -0.09210 \text{ kJ/K}\cdot\text{kg dry air}$$

The entropies of water vapor in the air stream are

$$s_{g1} = s_{g\,@\,15°\text{C}} = 8.7803 \text{ kJ/kg}\cdot\text{K}$$
$$s_{g2} = s_{g\,@\,18°\text{C}} = 8.7112 \text{ kJ/kg}\cdot\text{K}$$

The entropy change of water vapor in the air stream is

$$\Delta s_{\text{vapor}} = \omega_2 s_{g2} - \omega_1 s_{g1} = 0.0123 \times 8.7112 - 0.00264 \times 8.7803 = 0.08397 \text{ kJ/K}\cdot\text{kg dry air}$$

The partial pressures of water vapor and dry air for air streams are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat}\,@\,15°\text{C}} = (0.25)(1.7057 \text{ kPa}) = 0.4264 \text{ kPa}$$
$$P_{a1} = P_1 - P_{v1} = 101.325 - 0.4264 = 100.90 \text{ kPa}$$
$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat}\,@\,18°\text{C}} = (0.95)(2.065 \text{ kPa}) = 1.962 \text{ kPa}$$
$$P_{a2} = P_2 - P_{v2} = 101.325 - 1.962 = 99.36 \text{ kPa}$$

The entropy change of dry air is

$$\Delta s_a = s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_{a2}}{P_{a1}}$$
$$= (1.005)\ln\frac{291}{288} - (0.287)\ln\frac{99.36}{100.90} = 0.01483 \text{ kJ/kg dry air}$$

The entropy generation in the cooling tower is the total entropy change:

$$s_{\text{gen}} = \Delta s_{\text{water}} + \Delta s_{\text{vapor}} + \Delta s_a = -0.09210 + 0.08397 + 0.01483 = 0.00670 \text{ kJ/K}\cdot\text{kg dry air}$$

Finally, the exergy destruction per unit mass of dry air is

$$x_{\text{dest}} = T_0 s_{\text{gen}} = (288 \text{ K})(0.00670 \text{ kJ/K}\cdot\text{kg dry air}) = \mathbf{1.93\ kJ/kg\ dry\ air}$$

14-119E Water is cooled by air in a cooling tower. The relative humidity of the air at the exit and the water's exit temperature are to be determined.

Assumptions **1** Steady operating conditions exist and thus 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. **4** The cooling tower is adiabatic.

Analysis The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but 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. Applying the mass and energy balances yields

Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

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

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$

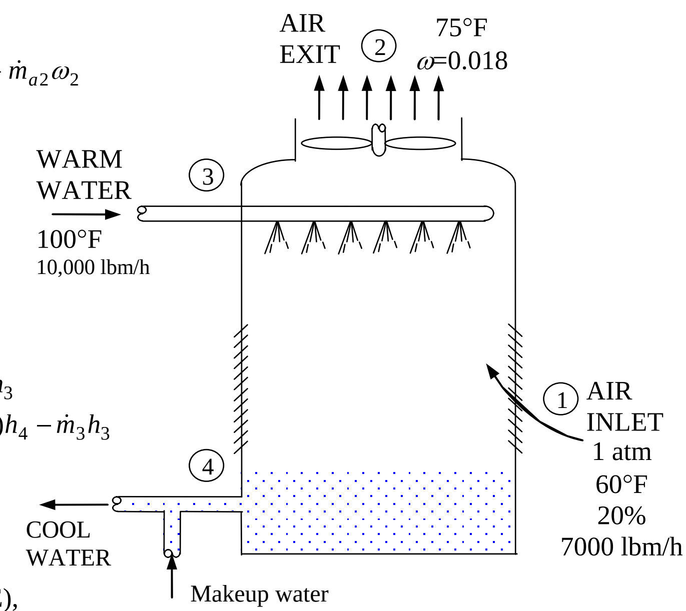
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2}h_2 + \dot{m}_4 h_4 - \dot{m}_{a1}h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$



Solving for h_4,

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

From the psychrometric chart (Fig. A-31E),

$$h_1 = 16.8 \text{ Btu/lbm dry air}$$

$$\omega_1 = 0.00219 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 13.15 \text{ ft}^3\text{/lbm dry air}$$

and

$$h_2 = 37.7 \text{ Btu/lbm dry air}$$

$$\phi_2 = 0.957 = \mathbf{95.7\%}$$

From Table A-4,

$$h_3 \cong h_{f\,@100°\text{F}} = 68.03 \text{ Btu/lbm H}_2\text{O}$$

Also,

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (7000/3600 \text{ lbm/s})(0.018 - 0.00219) = 0.03075 \text{ lbm/s}$$

Substituting,

$$h_4 = \frac{\dot{m}_3 h_3 - \dot{m}_a(h_2 - h_1)}{\dot{m}_3 - \dot{m}_{\text{makeup}}} = \frac{(10{,}000/3600)(68.03) - (7000/3600)(37.7 - 16.8)}{(10{,}000/3600) - 0.03075} = 53.99 \text{ Btu/lbm}$$

The exit temperature of the water is then (Table A-4E)

$$T_4 = T_{\text{sat}\,@\,h_f = 53.99 \text{ Btu/lbm}} = \mathbf{85.9°F}$$

14-120 Water is cooled by air in a cooling tower. The volume flow rate of air and the mass flow rate of the required makeup water are to be determined.

Assumptions **1** Steady operating conditions exist and thus 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. **4** The cooling tower is adiabatic.

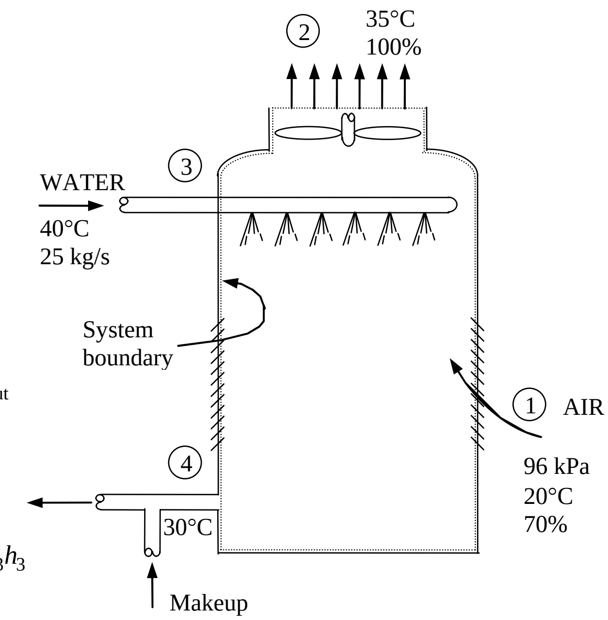
Analysis (*a*) The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e}$$
$$\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e}$$
$$\dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$
$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$
$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$
$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$
$$0 = \dot{m}_a (h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}}) h_4 - \dot{m}_3 h_3$$

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

The properties of air at the inlet and the exit are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 20°C}} = (0.70)(2.3392\text{ kPa}) = 1.637\text{ kPa}$$
$$P_{a1} = P_1 - P_{v1} = 96 - 1.637 = 94.363\text{ kPa}$$
$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{94.363\text{ kPa}} = 0.891\text{ m}^3/\text{kg dry air}$$
$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.637\text{ kPa})}{(96 - 1.637)\text{ kPa}} = 0.0108\text{ kg H}_2\text{O/kg dry air}$$
$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0108)(2537.4\text{ kJ/kg}) = 47.5\text{ kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat @ 35°C}} = (1.00)(5.6291\text{ kPa}) = 5.6291\text{ kPa}$$
$$\omega_2 = \frac{0.622\,P_{v2}}{P_2 - P_{v2}} = \frac{0.622(5.6291\text{ kPa})}{(96 - 5.6291)\text{ kPa}} = 0.0387\text{ kg H}_2\text{O/kg dry air}$$
$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot°\text{C})(35°\text{C}) + (0.0387)(2564.6\text{ kJ/kg}) = 134.4\text{ kJ/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,40°C} = 167.53\ \text{kJ/kg H}_2\text{O}$$
$$h_4 \cong h_{f\,@\,30°C} = 125.74\ \text{kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(25\ \text{kg/s})(167.53 - 125.74)\text{kJ/kg}}{(134.4 - 47.5)\ \text{kJ/kg} - (0.0387 - 0.0108)(125.74)\ \text{kJ/kg}} = 12.53\ \text{kg/s}$$

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

$$\dot{V}_1 = \dot{m}_a v_1 = (12.53\ \text{kg/s})(0.891\ \text{m}^3/\text{kg}) = \mathbf{11.2\ m^3/s}$$

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

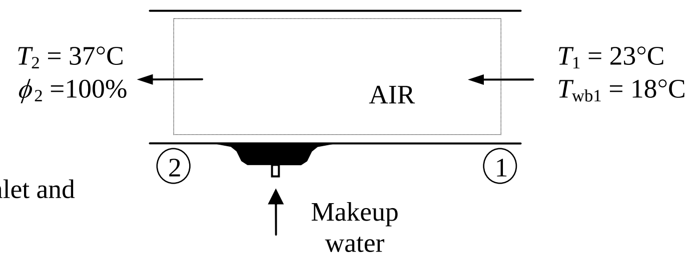
$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (12.53\ \text{kg/s})(0.0387 - 0.0108) = \mathbf{0.35\ kg/s}$$

14-121 A natural-draft cooling tower is used to remove waste heat from the cooling water flowing through the condenser of a steam power plant. The mass flow rate of the cooling water, the volume flow rate of air into the cooling tower, and the mass flow rate of the required makeup water are to be determined.

Assumptions **1** All processes are steady-flow and 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 inlet and exit states of the moist air for the tower are completely specified. The properties may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 50.74 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01085 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.8536 \text{ m}^3\text{/kg dry air}$$
$$h_2 = 142.83 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.04112 \text{ kg H}_2\text{O/kg dry air}$$



The enthalpies of cooling water at the inlet and exit of the condenser are (Table A-4)

$$h_{w3} = h_{f@\,40°C} = 167.53 \text{ kJ/kg}$$
$$h_{w4} = h_{f@\,26°C} = 109.01 \text{ kJ/kg}$$

The steam properties for the condenser are (Steam tables)

$$\left.\begin{array}{l} P_{s1} = 200 \text{ kPa} \\ x_{s1} = 0 \end{array}\right\} h_{s1} = 504.71 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{s2} = 10 \text{ kPa} \\ s_{s2} = 7.962 \text{ kJ/kg.K} \end{array}\right\} h_{s2} = 2524.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{s3} = 10 \text{ kPa} \\ x_{s1} = 0 \end{array}\right\} h_{s3} = 191.81 \text{ kJ/kg}$$

The mass flow rate of dry air is given by

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_1}{0.8536 \text{ m}^3\text{/kg}}$$

The mass flow rates of vapor at the inlet and exit of the cooling tower are

$$\dot{m}_{v1} = \omega_1 \dot{m}_a = (0.01085)\frac{\dot{V}_1}{0.8536} = 0.01271\dot{V}_1$$

$$\dot{m}_{v2} = \omega_2 \dot{m}_a = (0.04112)\frac{\dot{V}_1}{0.8536} = 0.04817\dot{V}_1$$

Mass and energy balances on the cooling tower give

$$\dot{m}_{v1} + \dot{m}_{cw3} = \dot{m}_{v2} + \dot{m}_{cw4}$$
$$\dot{m}_a h_1 + \dot{m}_{cw3} h_{w3} = \dot{m}_a h_2 + \dot{m}_{cw4} h_{w4}$$

The mass flow rate of the makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_{v2} - \dot{m}_{v1} = \dot{m}_{cw3} - \dot{m}_{cw4}$$

An energy balance on the condenser gives

$$0.18\dot{m}_s h_{s1} + 0.82\dot{m}_s h_{s2} + \dot{m}_{cw4} h_{w4} + \dot{m}_{\text{makeup}} h_{w4} = \dot{m}_s h_{s3} + \dot{m}_{cw3} h_{w3}$$

Solving all the above equations simultaneously with known and determined values using EES, we obtain

$$\dot{m}_{cw3} = \mathbf{1413 \ kg/s}$$
$$\dot{V}_1 = \mathbf{47{,}700 \ m^3/min}$$
$$\dot{m}_{\text{makeup}} = \mathbf{28.19 \ kg/s}$$

Review Problems

14-122 Air is compressed by a compressor and then cooled to the ambient temperature at high pressure. It is to be determined if there will be any condensation in the compressed air lines.

Assumptions The air and the water vapor are ideal gases.

Properties The saturation pressure of water at 20°C is 2.3392 kPa (Table A-4)..

Analysis The vapor pressure of air before compression is

$$P_{v1} = \phi_1 P_g = \phi_1 P_{\text{sat @ 25°C}} = (0.50)(2.3392\text{ kPa}) = 1.17\text{ kPa}$$

The pressure ratio during the compression process is (800 kPa)/(92 kPa) = 8.70. That is, the pressure of air and any of its components increases by 8.70 times. Then the vapor pressure of air after compression becomes

$$P_{v2} = P_{v1} \times (\text{Pressure ratio}) = (1.17\text{ kPa})(8.70) = 10.2\text{ kPa}$$

The dew-point temperature of the air at this vapor pressure is

$$T_{\text{dp}} = T_{\text{sat @ }P_{v2}} = T_{\text{sat @10.2 kPa}} = 46.1°\text{C}$$

which is greater than 20°C. Therefore, part of the moisture in the compressed air will **condense** when air is cooled to 20°C.

14-123E The mole fraction of the water vapor at the surface of a lake and the mole fraction of water in the lake are to be determined and compared.

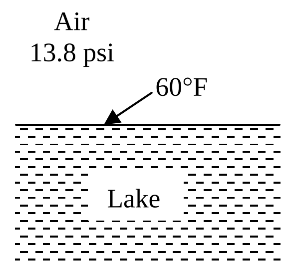
Assumptions **1** Both the air and water vapor are ideal gases. **2** Air is weakly soluble in water and thus Henry's law is applicable.

Properties The saturation pressure of water at 60°F is 0.2564 psia (Table A-4E). Henry's constant for air dissolved in water at 60ºF (289 K) is given in Table 16-2 to be H = 62,000 bar.

Analysis The air at the water surface will be saturated. Therefore, the partial pressure of water vapor in the air at the lake surface will simply be the saturation pressure of water at 60°F,

$$P_{\text{vapor}} = P_{\text{sat @60°F}} = 0.2564\ \text{psia}$$

Assuming both the air and vapor to be ideal gases, the mole fraction of water vapor in the air at the surface of the lake is determined to be



$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{0.2564\ \text{psia}}{13.8\ \text{psia}} = \mathbf{0.0186\ \ (or\ 1.86\ percent)}$$

The partial pressure of dry air just above the lake surface is

$$P_{\text{dry air}} = P - P_{\text{vapor}} = 13.8 - 0.2564 = 13.54\ \text{psia}$$

Then the mole fraction of air in the water becomes

$$y_{\text{dry air, liquid side}} = \frac{P_{\text{dry air, gas side}}}{\text{H}} = \frac{13.54\ \text{psia}(1\ \text{atm} / 14.696\ \text{psia})}{62{,}000\ \text{bar}\,(1\ \text{atm} / 1.01325\ \text{bar})} = 1.51\times 10^{-5}$$

which is very small, as expected. Therefore, the mole fraction of water in the lake near the surface is

$$y_{\text{water,liquid side}} = 1 - y_{\text{dry air, liquid side}} = 1 - 1.51\times 10^{-5} \cong \mathbf{1.0}$$

Discussion The concentration of air in water just below the air-water interface is 1.51 moles per 100,000 moles. The amount of air dissolved in water will decrease with increasing depth.

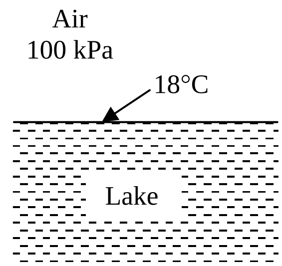
14-124 The mole fraction of the water vapor at the surface of a lake at a specified temperature is to be determined.

Assumptions **1** Both the air and water vapor are ideal gases. **2** Air at the lake surface is saturated.

Properties The saturation pressure of water at 18°C is 2.065 kPa (Table A-4).

Analysis The air at the water surface will be saturated. Therefore, the partial pressure of water vapor in the air at the lake surface will simply be the saturation pressure of water at 18°C,

$$P_{\text{vapor}} = P_{\text{sat @18°C}} = 2.065\text{ kPa}$$



Assuming both the air and vapor to be ideal gases, the partial pressure and mole fraction of dry air in the air at the surface of the lake are determined to be

$$P_{\text{dry air}} = P - P_{\text{vapor}} = 100 - 2.065 = 97.94\text{ kPa}$$

$$y_{\text{dry air}} = \frac{P_{\text{dry air}}}{P} = \frac{97.94\text{ kPa}}{100\text{ kPa}} = \mathbf{0.979}\ \text{(or 97.9\%)}$$

Therefore, the mole fraction of dry air is 97.9 percent just above the air-water interface.

14-125E A room is cooled adequately by a 7500 Btu/h air-conditioning unit. If the room is to be cooled by an evaporative cooler, the amount of water that needs to be supplied to the cooler is to be determined.

Assumptions **1** The evaporative cooler removes heat at the same rate as the air conditioning unit. **2** Water evaporates at an average temperature of 70°F.

Properties The enthalpy of vaporization of water at 70°F is 1053.7 Btu/lbm (Table A-4E).

Analysis Noting that 1 lbm of water removes 1053.7 Btu of heat as it evaporates, the amount of water that needs to evaporate to remove heat at a rate of 7500 Btu/h is determined from $\dot{Q} = \dot{m}_{\text{water}} h_{fg}$ to be

$$\dot{m}_{\text{water}} = \frac{\dot{Q}}{h_{fg}} = \frac{7500\text{ Btu/h}}{1053.7\text{ Btu/lbm}} = \mathbf{7.12\ lbm/h}$$

14-126E The required size of an evaporative cooler in cfm (ft^3/min) for an 8-ft high house is determined by multiplying the floor area of the house by 4. An equivalent rule is to be obtained in SI units.

Analysis Noting that 1 ft = 0.3048 m and thus 1 ft^2 = 0.0929 m^2 and 1 ft^3 = 0.0283 m^3, and noting that a flow rate of 4 ft^3/min is required per ft^2 of floor area, the required flow rate in SI units per m^2 of floor area is determined to

$$1\,\text{ft}^2 \leftrightarrow 4\,\text{ft}^3\ /\ \text{min}$$

$$0.0929\,\text{m}^2 \leftrightarrow 4\times 0.0283\,\text{m}^3\ /\ \text{min}$$

$$1\,\text{m}^2 \leftrightarrow 1.22\,\text{m}^3\ /\ \text{min}$$

Therefore, a flow rate of **1.22 m^3/min** is required per m^2 of floor area.

14-127 A cooling tower with a cooling capacity of 440 kW is claimed to evaporate 15,800 kg of water per day. It is to be determined if this is a reasonable claim.

Assumptions **1** Water evaporates at an average temperature of 30°C. **2** The coefficient of performance of the air-conditioning unit is COP = 3.

Properties The enthalpy of vaporization of water at 30°C is 2429.8 kJ/kg (Table A-4).

Analysis Using the definition of COP, the electric power consumed by the air conditioning unit when running is

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_{\text{cooling}}}{\text{COP}} = \frac{440\text{ kW}}{3} = 146.7\text{ kW}$$

Then the rate of heat rejected at the cooling tower becomes

$$\dot{Q}_{\text{rejected}} = \dot{Q}_{\text{cooling}} + \dot{W}_{in} = 440 + 146.7 = 586.7\text{ kW}$$

Noting that 1 kg of water removes 2429.8 kJ of heat as it evaporates, the amount of water that needs to evaporate to remove heat at a rate of 586.7 kW is determined from $\dot{Q}_{\text{rejected}} = \dot{m}_{\text{water}} h_{fg}$ to be

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{rejected}}}{h_{fg}} = \frac{586.7\text{ kJ/s}}{2429.8\text{ kJ/kg}} = 0.2415\text{ kg/s} = 869.3\text{ kg/h} = 20{,}860\text{ kg/day}$$

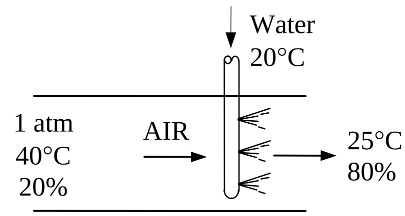
In practice, the air-conditioner will run intermittently rather than continuously at the rated power, and thus the water use will be less. Therefore, the claim amount of 15,800 kg per day is **reasonable**.

14-128 Air is cooled by evaporating water into this air. The amount of water required and the cooling produced 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 64.0 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0092 \text{ kg H}_2\text{O/kg dry air}$$



and

$$h_2 = 66.0 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0160 \text{ kg H}_2\text{O/kg dry air}$$

Also,

$$h_w \cong h_{f\,@\,20°\text{C}} = 83.92 \text{ kJ/kg} \quad \text{(Table A-4)}$$

Analysis The amount of moisture in the air increases due to humidification ($\omega_2 > \omega_1$). Applying the water mass balance and energy balance equations to the combined cooling and humidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\Delta\omega = \omega_2 - \omega_1 = 0.0160 - 0.0092 = \mathbf{0.0068 \text{ kg H}_2\text{O/kg dry air}}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 + \dot{m}_w h_w - \dot{m}_{a2}h_2 = \dot{m}_a(h_1 - h_2) + \dot{m}_w h_w$$
$$q_{\text{out}} = h_1 - h_2 + (\omega_2 - \omega_1)h_w$$
$$= (64.0 - 66.0)\text{kJ/kg} + (0.0068)(83.92)$$
$$= -\mathbf{1.43 \text{ kJ/kg dry air}}$$

The negative sign shows that the heat is actually transferred to the system.

14-129 Air is humidified adiabatically by evaporating water into this air. The temperature of the air at the exit 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.

Properties The inlet state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 64.0 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0092 \text{ kg H}_2\text{O/kg dry air}$$

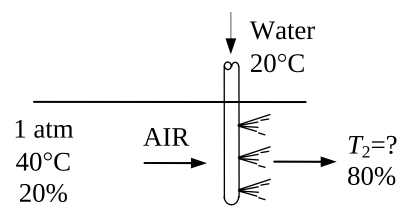
and

$$h_w \cong h_{f\,@\,20°\text{C}} = 83.92 \text{ kJ/kg} \qquad \text{(Table A-4)}$$

Analysis The amount of moisture in the air increases due to humidification ($\omega_2 > \omega_1$). Applying the water mass balance and energy balance equations to the combined cooling and humidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$
$$\dot{m}_w = \dot{m}_a(\omega_2 - \omega_1)$$



Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,\text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_{a1}h_1 + \dot{m}_w h_w = \dot{m}_{a2}h_2$$
$$\dot{m}_w h_w = \dot{m}_a(h_2 - h_1)$$
$$(\omega_2 - \omega_1)h_w = h_2 - h_1$$

Substituting,

$$(\omega_2 - 0.0092)(83.92) = h_2 - 64.0$$

The solution of this equation requires a trial-error method. An air exit temperature is assumed. At this temperature and given relative humidity, the enthalpy and specific humidity values are obtained from psychrometric chart and substituted into this equation. If the equation is not satisfied, a new value of exit temperature is assumed and this continues until the equation is satisfied. Alternatively, an equation solver such as EES may be used for the direct results. We used the following EES program.

```
"Given"
P=101.325 "[kPa]"
T_1=40 "[C]"
phi_1=0.20
phi_2=0.80
"Analysis"
Fluid1$='AirH2O'
Fluid2$='steam_iapws'
h_1=enthalpy(Fluid1$, T=T_1, R=phi_1, P=P)
w_1=humrat(Fluid1$, T=T_1, R=phi_1, P=P)
h_2=enthalpy(Fluid1$, T=T_2, R=phi_2, P=P)
```

```
w_2=humrat(Fluid1$, T=T_2, R=phi_2, P=P)
h_w=enthalpy(Fluid2$, T=20, x=0)
q=0
q=h_1-h_2+(w_2-w_1)*h_w
```

The results of these equations are

$$T_2 = \mathbf{24.6^\circ C}$$
$$h_2 = 64.56 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.01564 \text{ kg H}_2\text{O/kg dry air}$$

14-130E Air is cooled and dehumidified at constant pressure. The rate of cooling and the minimum humid air temperature required to meet this cooling requirement are to be determined.

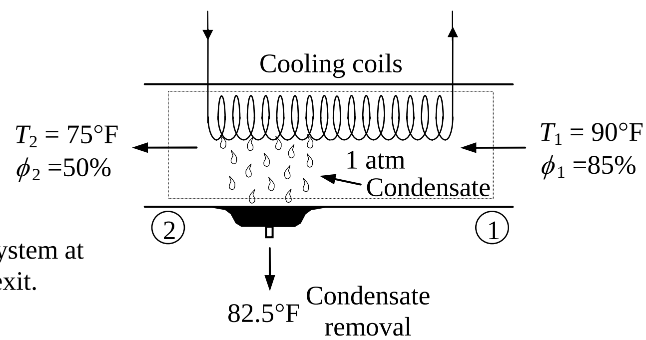
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 50.6 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0263 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_1 = 14.44 \text{ ft}^3\text{/lbm dry air}$$

and

$$h_2 = 28.2 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0093 \text{ lbm H}_2\text{O/lbm dry air}$$



We assume that the condensate leaves this system at the average temperature of the air inlet and exit. Then,

$$h_w \cong h_{f\,@\,82.5°\text{F}} = 50.56 \text{ Btu/lbm} \quad \text{(Table A-4)}$$

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass of air is

$$m_a = \frac{V_1}{v_1} = \frac{1000 \text{ ft}^3}{14.44 \text{ ft}^3 \text{ / lbm dry air}} = 69.25 \text{ lbm}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$m_w = m_a(\omega_1 - \omega_2) = (69.25 \text{ kg})(0.0263 - 0.0093) = 1.177 \text{ lbm}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$
$$Q_{\text{out}} = m_a(h_1 - h_2) - m_w h_w$$
$$Q_{\text{out}} = (69.25 \text{ kg})(50.6 - 28.2)\text{Btu/lbm} - (1.177 \text{ lbm})(50.56 \text{ Btu/lbm})$$
$$= \mathbf{1492 \text{ Btu}}$$

For the desired dehumidification, the air at the exit should be saturated with a specific humidity of 0.0093 lbm water/lbm dry air. That is,

$$\phi_2 = 1.0$$
$$\omega_2 = 0.0093 \text{ lbm H}_2\text{O/lbm dry air}$$

The temperature of the air at this state is the minimum air temperature required during this process:

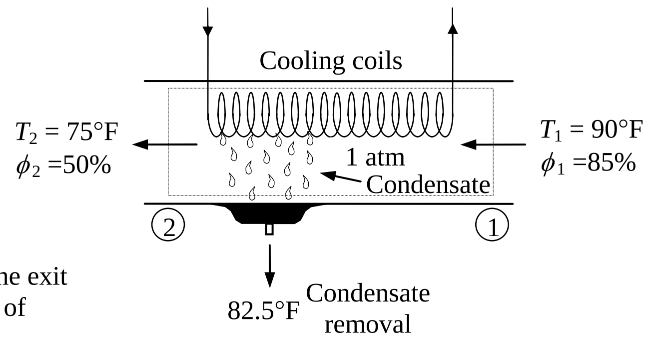
$$T_2 = \mathbf{55.2°F}$$

14-131E Air is cooled and dehumidified at constant pressure by a simple ideal vapor-compression refrigeration system. The system's COP 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 inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 50.6 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0263 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_1 = 14.44 \text{ ft}^3\text{/lbm dry air}$$



and

$$h_2 = 28.2 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0093 \text{ lbm H}_2\text{O/lbm dry air}$$

For the desired dehumidification, the air at the exit should be saturated with a specific humidity of 0.0093 lbm water/lbm dry air. That is,

$$\phi_2 = 1.0$$
$$\omega_2 = 0.0093 \text{ lbm H}_2\text{O/lbm dry air}$$

The temperature of the air at this state is the minimum air temperature required during this process:

$$T_{2,\min} = 55.2°\text{F}$$

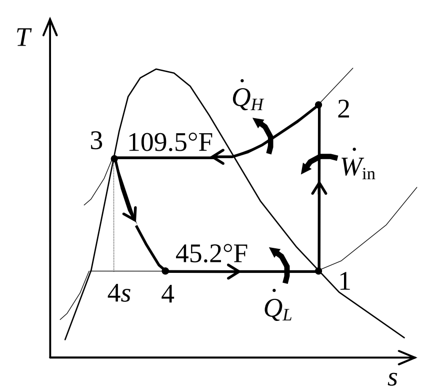
From the problem statement, the properties of R-134a at various states are (Tables A-11E through A-13E or from EES):

$$\left.\begin{array}{l} T_1 = 55.2 - 10 = 45.2°\text{F} \\ P_2 = P_{\text{sat @ 45.2°F}} = 55 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\text{ @ 55 psia}} = 109.49 \text{ Btu/lbm} \\ s_1 = s_{g\text{ @ 55 psia}} = 0.22156 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} T_{\text{sat}} = 90 + 19.5 = 109.5°\text{F} \\ P_2 = P_{\text{sat @ 109.5°F}} = 160 \text{ psia} \\ s_2 = s_1 \end{array}\right\} h_2 = 119.01 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 160 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\text{ @ 160 psia}} = 48.52 \text{ Btu/lbm}$$

$$h_4 \cong h_3 = 48.52 \text{ Btu/lbm} \quad \text{(throttling)}$$



The COP of this system is then

$$\text{COP} = \frac{q_L}{w_{\text{in}}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{109.49 - 48.52}{119.01 - 109.49} = \mathbf{6.40}$$

14-132E Air at a specified state is heated to a specified temperature. The relative humidity after the heating is to be determined.

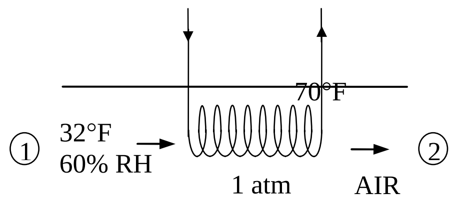
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis There is no correspondence of inlet state from the psychrometric chart. Therefore, we have to use EES psychrometric functions to obtain the specific humidity:

$$\omega_1 = 0.0023\,\text{lbm H}_2\text{O/lbm dry air}$$

As the outside air infiltrates into the dacha, it does not gain or lose any water. Therefore the humidity ratio inside the dacha is the same as that outside,

$$\omega_2 = \omega_1 = 0.0023\,\text{lbm H}_2\text{O/lbm dry air}$$



From EES or Fig. A-31E, at this humidity ratio and the temperature inside the dacha gives

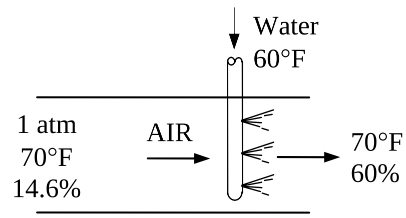
$$\phi_2 = 0.146 = \mathbf{14.6\%}$$

14-133E Air is humidified by evaporating water into this air. The amount of heating 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.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 19.3\ \text{Btu/lbm dry air}$$
$$\omega_1 = 0.0023\ \text{kg H}_2\text{O/kg dry air}$$
$$v_1 = 13.40\ \text{m}^3\text{/kg dry air}$$



and

$$h_2 = 27.1\ \text{Btu/lbm dry air}$$
$$\omega_2 = 0.0094\ \text{lbm H}_2\text{O/lbm dry air}$$

Also,

$$h_w \cong h_{f\,@\,60°\text{F}} = 28.08\ \text{Btu/lbm} \quad \text{(Table A-4E)}$$

Analysis The amount of moisture in the air increases due to humidification ($\omega_2 > \omega_1$). Applying the water mass balance and energy balance equations to the combined cooling and humidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i + \dot{Q}_{\text{in}} = \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{in}} = \dot{m}_{a2}h_2 - \dot{m}_{a1}h_1 - \dot{m}_w h_w = \dot{m}_a(h_2 - h_1) - \dot{m}_w h_w$$
$$q_{\text{in}} = h_2 - h_1 - (\omega_2 - \omega_1)h_w$$
$$= (27.1 - 19.3)\text{Btu/lbm} - (0.0094 - 0.0023)(28.08)$$
$$= 7.59\ \text{Btu/lbm dry air}$$

The mass of air that has to be humidified is

$$m_a = \frac{V}{v_1} = \frac{16{,}000\ \text{ft}^3}{13.40\ \text{ft}^3\text{/lbm dry air}} = 1194\ \text{lbm dry air}$$

The total heat requirement is then

$$Q_{\text{in}} = m_a q_{\text{in}} = (1194\ \text{lbm dry air})(7.59\ \text{Btu/lbm dry air}) = \mathbf{9062\ Btu}$$

14-134E It is estimated that 190,000 barrels of oil would be saved per day if the thermostat setting in residences in summer were raised by 6°F (3.3°C). The amount of money that would be saved per year is to be determined.

Assumptions The average cooling season is given to be 120 days, and the cost of oil to be $20/barrel.

Analysis The amount of money that would be saved per year is determined directly from

$$(190{,}000\text{ barrel/day})(120\text{ days/year})(\$70\text{/barrel}) = \mathbf{\$1{,}596{,}000{,}000}$$

Therefore, the proposed measure will save more than one and half billion dollars a year.

14-135 Shading the condenser can reduce the air-conditioning costs by up to 10 percent. The amount of money shading can save a homeowner per year during its lifetime is to be determined.

Assumptions It is given that the annual air-conditioning cost is $500 a year, and the life of the air-conditioning system is 20 years.

Analysis The amount of money that would be saved per year is determined directly from

$$(\$500\text{ / year})(20\text{ years})(0.10) = \mathbf{\$1000}$$

Therefore, the proposed measure will save about $1000 during the lifetime of the system.

14-136 A tank contains saturated air at a specified state. The mass of the dry air, the specific humidity, and the enthalpy of the air are to be determined.

Assumptions The air and the water vapor are ideal gases.

Analysis (*a*) The air is saturated, thus the partial pressure of water vapor is equal to the saturation pressure at the given temperature,

$$P_v = P_g = P_{\text{sat @ 25°C}} = 3.1698\text{ kPa}$$
$$P_a = P - P_v = 97 - 3.1698 = 93.83\text{ kPa}$$

3 m^3
25°C
97 kPa

Treating air as an ideal gas,

$$m_a = \frac{P_a \mathbf{V}}{R_a T} = \frac{(93.83\text{ kPa})(3\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = \mathbf{3.29\text{ kg}}$$

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(3.1698\text{ kPa})}{(97 - 3.1698)\text{ kPa}} = \mathbf{0.0210\text{ kg H}_2\text{O/kg dry air}}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$\begin{aligned} h &= h_a + \omega h_v \cong c_p T + \omega h_g \\ &= (1.005\text{ kJ/kg}\cdot°\text{C})(25°\text{C}) + (0.0210)(2546.5\text{ kJ/kg}) \\ &= \mathbf{78.6\text{ kJ/kg dry air}} \end{aligned}$$

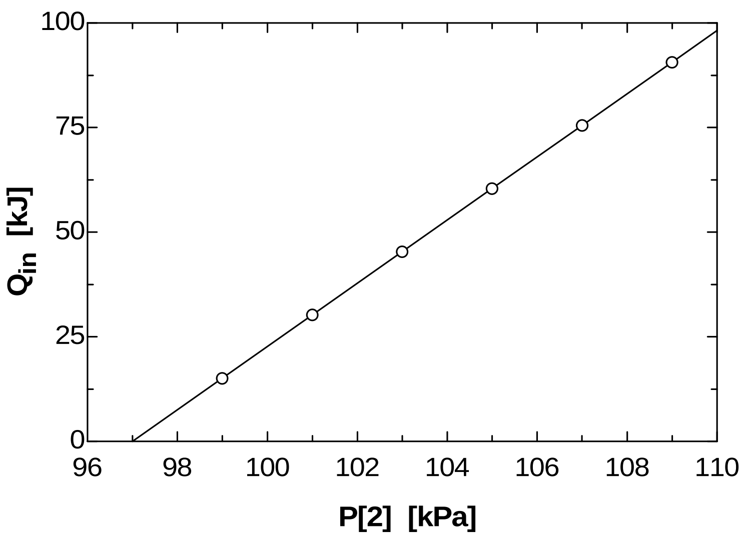
14-137 EES Problem 14-136 is reconsidered. The properties of the air at the initial state are to be determined and the effects of heating the air at constant volume until the pressure is 110 kPa is to be studied.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data:"
Tdb[1] = 25 [C]
P[1]=97 [kPa]
Rh[1]=1.0
P[2]=110 [kPa]
Vol = 3 [m^3]

w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
m_a=Vol/v[1]
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],w=w[1])
"Energy Balance for the constant volume tank:"
E_in - E_out = DELTAE_tank
DELTAE_tank=m_a*(u[2] -u[1])
E_in = Q_in
E_out = 0 [kJ]
u[1]=INTENERGY(AirH2O,T=Tdb[1],P=P[1],w=w[1])
u[2]=INTENERGY(AirH2O,T=Tdb[2],P=P[2],w=w[2])
"The ideal gas mixture assumption applied to the constant volume process yields:"
P[1]/(Tdb[1]+273)=P[2]/(Tdb[2]+273)
"The mass of the water vapor and dry air are constant, thus:"
w[2]=w[1]
Rh[2]=RELHUM(AirH2O,T=Tdb[2],P=P[2],w=w[2])
h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],w=w[2])
v[2]=VOLUME(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
```

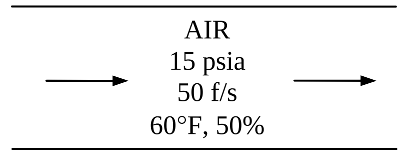
PROPERTIES AT THE INITIAL STATE
h[1]=78.67 [kJ/kga]
m_a=3.289 [kga]
v[1]=0.9121 [m^3/kga]
w[1]=0.02101 [kgw/kga]

P_2 [kPa]	Q_{in} [kJ]
97	0
99	15.12
101	30.23
103	45.34
105	60.45
107	75.55
109	90.65
110	98.2



14-138E Air at a specified state and relative humidity flows through a circular duct. The dew-point temperature, the volume flow rate of air, and the mass flow rate of dry air are to be determined.

Assumptions The air and the water vapor are ideal gases.



Analysis (*a*) The vapor pressure of air is

$$P_v = \phi P_g = \phi P_{\text{sat @ 60°F}} = (0.50)(0.2564\text{ psia}) = 0.128\text{ psia}$$

Thus the dew-point temperature of the air is

$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 0.128 psia}} = \mathbf{41.3°F}\ \ \text{(from EES)}$$

(*b*) The volume flow rate is determined from

$$\dot{V} = VA = V\frac{\pi D^2}{4} = (50\text{ ft/s})\left(\frac{\pi \times (8/12\text{ ft})^2}{4}\right) = \mathbf{17.45\ \ ft^3/s}$$

(*c*) To determine the mass flow rate of dry air, we first need to calculate its specific volume,

$$P_a = P - P_v = 15 - 0.128 = 14.872\text{ psia}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520\text{ R})}{14.872\text{ psia}} = 12.95\text{ ft}^3/\text{lbm dry air}$$

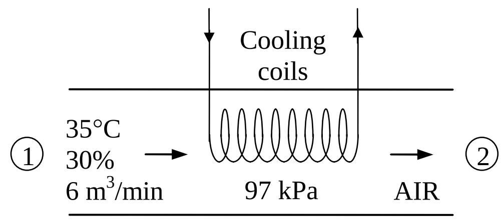
Thus,

$$\dot{m}_{a1} = \frac{\dot{V}_1}{\boldsymbol{v}_1} = \frac{17.45\text{ ft}^3/\text{s}}{12.95\text{ ft}^3/\text{lbm dry air}} = \mathbf{1.35\ lbm/s}$$

14-139 Air enters a cooling section at a specified pressure, temperature, and relative humidity. The temperature of the air at the exit and the rate of heat transfer 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 $(\dot m_{a1} = \dot m_{a2} = \dot m_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The amount of moisture in the air also remains constant $(\omega_1 = \omega_2)$ as it flows through the cooling section since the process involves no humidification or dehumidification. The total pressure is 97 kPa. The properties of the air at the inlet state are



$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 35°C}} = (0.3)(5.629\text{ kPa}) = 1.69\text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 97 - 1.69 = 95.31\text{ kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})}{95.31\text{ kPa}}$$

$$= 0.927\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.69\text{ kPa})}{(97-1.69)\text{ kPa}} = 0.0110\text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg°C})(35\text{°C}) + (0.0110)(2564.6\text{ kJ/kg}) = 63.44\text{ kJ/kg dry air}$$

The air at the final state is saturated and the vapor pressure during this process remains constant. Therefore, the exit temperature of the air must be the dew-point temperature,

$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 1.69 kPa}} = \mathbf{14.8°C}$$

(*b*) The enthalpy of the air at the exit is

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot\text{°C})(14.8\text{°C}) + (0.0110)(2528.1\text{ kJ/kg}) = 42.78\text{ kJ/kg dry air}$$

Also

$$\dot m_a = \frac{\dot V_1}{v_1} = \frac{6\text{ m}^3/\text{s}}{0.927\text{ m}^3/\text{kg dry air}} = 6.47\text{ kg/min}$$

Then the rate of heat transfer from the air in the cooling section becomes

$$\dot Q_{\text{out}} = \dot m_a (h_1 - h_2) = (6.47\text{ kg/min})(63.44 - 42.78)\text{kJ/kg} = \mathbf{134\text{ kJ/min}}$$

14-140 The outdoor air is first heated and then humidified by hot steam in an air-conditioning system. The rate of heat supply in the heating section and the mass flow rate of the steam required in the humidifying section are to be determined.

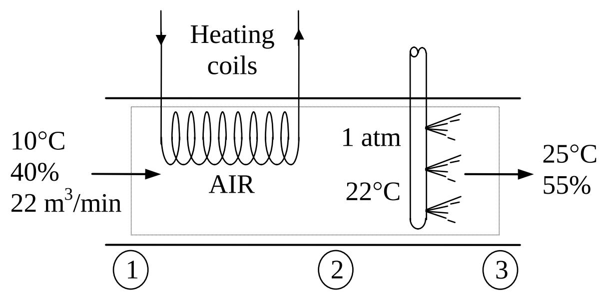
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The amount of moisture in the air also remains constants it flows through the heating section $(\omega_1 = \omega_2)$, but increases in the humidifying section $(\omega_3 > \omega_2)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 17.7 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0030 \text{ kg H}_2\text{O/kg dry air} \ (= \omega_2)$$
$$v_1 = 0.807 \text{ m}^3\text{/kg dry air}$$

$$h_2 = 29.8 \text{ kJ / kg dry air}$$
$$\omega_2 = \omega_1 = 0.0030 \text{ kg H}_2\text{O / kg dry air}$$

$$h_3 = 52.9 \text{ kJ / kg dry air}$$
$$\omega_3 = 0.0109 \text{ kg H}_2\text{O / kg dry air}$$



Analysis (*a*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{22 \text{ m}^3\text{ / min}}{0.807 \text{ m}^3\text{ / kg}} = 27.3 \text{ kg/min}$$

Then the rate of heat transfer to the air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1) = (27.3 \text{ kg / min})(29.8 - 17.7)\text{kJ / kg} = \mathbf{330.3 \text{ kJ / min}}$$

(*b*) The conservation of mass equation for water in the humidifying section can be expressed as

$$\dot{m}_{a2}\omega_2 + \dot{m}_w = \dot{m}_{a3}\omega_3 \quad \text{or} \quad \dot{m}_w = \dot{m}_a(\omega_3 - \omega_2)$$

Thus,

$$\dot{m}_w = (27.3 \text{ kg / min})(0.0109 - 0.0030) = \mathbf{0.216 \text{ kg / min}}$$

14-141 Air is cooled and dehumidified in an air-conditioning system with refrigerant-134a as the working fluid. The rate of dehumidification, the rate of heat transfer, and the mass flow rate of the refrigerant 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) The saturation pressure of water at 30ºC is 4.2469 kPa. Then the dew point temperature of the incoming air stream at 30°C becomes

$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 0.7}\times\text{4.2469 kPa}} = 24°\text{C}$$

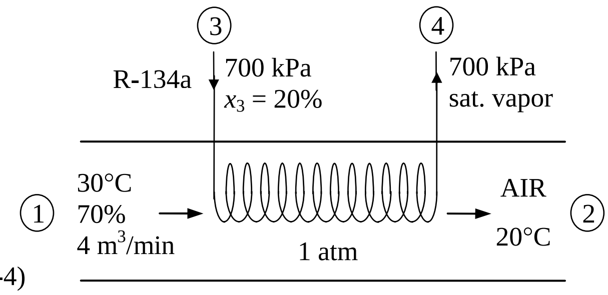
Since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense.

The mass flow rate of dry air remains constant during the entire process, but the amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. Then the properties of the air at both states are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 78.3 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0188 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.885 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 57.5 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0147 \text{ kg H}_2\text{O/kg dry air}$$



Also, $h_w \cong h_{f\text{ @ 20°C}} = 83.915 \text{ kJ/kg}$ (Table A-4)

Then, $$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{4 \text{ m}^3\text{/min}}{0.885 \text{ m}^3\text{/kg dry air}} = 4.52 \text{ kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the refrigerant),

Water Mass Balance: $\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (4.52 \text{ kg/min})(0.0188 - 0.0147) = \mathbf{0.0185 \text{ kg/min}}$$

(*b*) *Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{\text{out}} + \sum \dot{m}_e h_e \longrightarrow \dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$
$$\dot{Q}_{\text{out}} = (4.52 \text{ kg/min})(78.3 - 57.5)\text{kJ/kg} - (0.0185 \text{ kg/min})(83.915 \text{ kJ/kg}) = \mathbf{92.5 \text{ kJ/min}}$$

(*c*) The inlet and exit enthalpies of the refrigerant are

$$h_3 = h_g + x_3 h_{fg} = 88.82 + 0.2 \times 176.21 = 124.06 \text{ kJ/kg}$$
$$h_4 = h_{g\text{ @ 700 kPa}} = 265.03 \text{ kJ/kg}$$

Noting that the heat lost by the air is gained by the refrigerant, the mass flow rate of the refrigerant becomes

$$\dot{Q}_R = \dot{m}_R(h_4 - h_3) \rightarrow \dot{m}_R = \frac{\dot{Q}_R}{h_4 - h_3} = \frac{92.5 \text{ kJ/min}}{(265.03 - 124.06) \text{ kJ/kg}} = \mathbf{0.66 \text{ kg/min}}$$

14-142 Air is cooled and dehumidified in an air-conditioning system with refrigerant-134a as the working fluid. The rate of dehumidification, the rate of heat transfer, and the mass flow rate of the refrigerant 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.

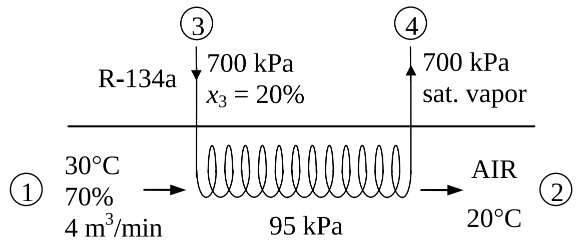
Analysis (*a*) The dew point temperature of the incoming air stream at 30°C is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 30°C}} = (0.7)(4.247\text{ kPa}) = 2.973\text{ kPa}$$

$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 2.973 kPa}} = 24°\text{C}$$

Since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense.

The amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. The properties of the air at both states are determined to be



$$P_{a1} = P_1 - P_{v1} = 95 - 2.97 = 92.03\text{ kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303\text{ K})}{92.03\text{ kPa}} = 0.945\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(2.97\text{ kPa})}{(95-2.97)\text{ kPa}} = 0.0201\text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(30°\text{C}) + (0.0201)(2555.6\text{ kJ/kg})$$
$$= 81.50\text{ kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = (1.00) P_{\text{sat @ 20°C}} = 2.3392\text{ kPa}$$

$$\omega_2 = \frac{0.622\,P_{v2}}{P_2 - P_{v2}} = \frac{0.622(2.3392\text{ kPa})}{(95-2.3392)\text{ kPa}} = 0.0157\text{ kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0157)(2537.4\text{ kJ/kg})$$
$$= 59.94\text{ kJ/kg dry air}$$

Also, $h_w \cong h_{f\text{ @ 20°C}} = 83.915\text{ kJ/kg}$ (Table A-4)

Then,

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{4\text{ m}^3/\text{min}}{0.945\text{ m}^3/\text{kg dry air}} = 4.23\text{ kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the refrigerant),

Water Mass Balance: $\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (4.23\text{ kg/min})(0.0201 - 0.0157) = \mathbf{0.0186\text{ kg/min}}$$

(*b*) *Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

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

$$\sum \dot{m}_i h_i = \dot{Q}_{\text{out}} + \sum \dot{m}_e h_e \longrightarrow \dot{Q}_{\text{out}} = \dot{m}_{a1} h_1 - (\dot{m}_{a2} h_2 + \dot{m}_w h_w) = \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (4.23\text{ kg/min})(81.50 - 59.94)\text{kJ/kg} - (0.0186\text{ kg/min})(83.915\text{ kJ/kg}) = \mathbf{89.7\ kJ/min}$$

(*c*) The inlet and exit enthalpies of the refrigerant are

$$h_3 = h_g + x_3 h_{fg} = 88.82 + 0.2 \times 176.21 = 124.06\text{ kJ/kg}$$

$$h_4 = h_{g\,@\,700\text{ kPa}} = 265.03\text{ kJ/kg}$$

Noting that the heat lost by the air is gained by the refrigerant, the mass flow rate of the refrigerant is determined from

$$\dot{Q}_R = \dot{m}_R (h_4 - h_3)$$

$$\dot{m}_R = \frac{\dot{Q}_R}{h_4 - h_3} = \frac{89.7\text{ kJ/min}}{(265.03 - 124.06)\text{ kJ/kg}} = \mathbf{0.636\ kg/min}$$

14-143 Air is heated and dehumidified in an air-conditioning system consisting of a heating section and an evaporative cooler. The temperature and relative humidity of the air when it leaves the heating section, the rate of heat transfer in the heating section, and the rate of water added to the air in the evaporative cooler are to be determined.

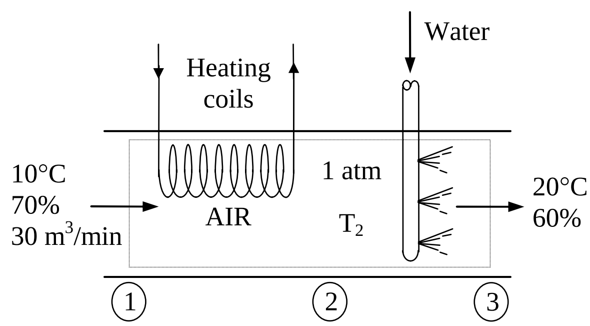
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) Assuming the wet-bulb temperature of the air remains constant during the evaporative cooling process, the properties of air at various states are determined from the psychrometric chart (Fig. A-31) to be

$$\left.\begin{array}{l} T_1 = 10°\text{C} \\ \phi_1 = 70\% \end{array}\right\} \begin{array}{l} h_1 = 23.5 \text{ kJ/kg dry air} \\ \omega_1 = 0.00532 \text{ kg/ H}_2\text{O/kg dry air} \\ \boldsymbol{v}_1 = 0.810 \text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} \omega_2 = \omega_1 \\ T_{\text{wb2}} = T_{\text{wb3}} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{28.3°C} \\ \phi_2 = \mathbf{22.3\%} \\ h_2 \cong h_3 = 42.3 \text{ kJ/kg dry air} \end{array}$$

$$\left.\begin{array}{l} T_3 = 20°\text{C} \\ \phi_3 = 60\% \end{array}\right\} \begin{array}{l} h_3 = 42.3 \text{ kJ/kg dry air} \\ \omega_3 = 0.00875 \text{ kg/ H}_2\text{O/kg dry air} \\ T_{wb3} = 15.1°\text{C} \end{array}$$



(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{30 \text{ m}^3/\text{min}}{0.810 \text{ m}^3/\text{kg dry air}} = 37.0 \text{ kg/min}$$

Then the rate of heat transfer to air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1) = (37.0 \text{ kg/min})(42.3 - 23.5)\text{kJ/kg} = \mathbf{696 \text{ kJ/min}}$$

(*c*) The rate of water added to the air in evaporative cooler is

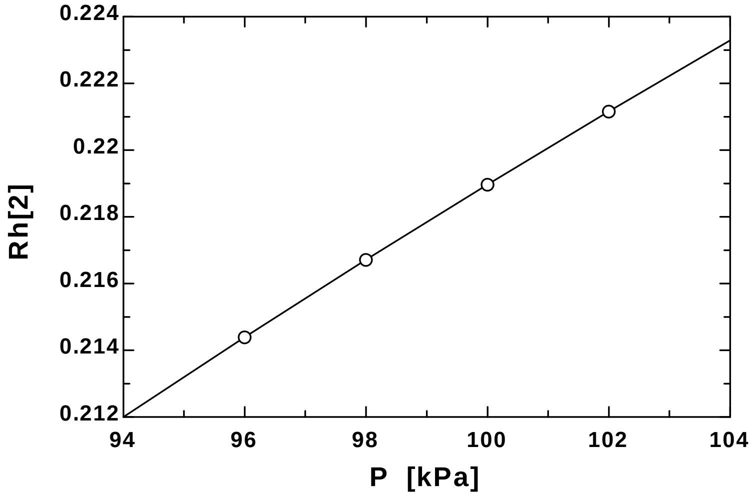
$$\dot{m}_{w,\text{added}} = \dot{m}_{w3} - \dot{m}_{w2} = \dot{m}_a(\omega_3 - \omega_2) = (37.0 \text{ kg/min})(0.00875 - 0.00532) = \mathbf{0.127 \text{ kg/min}}$$

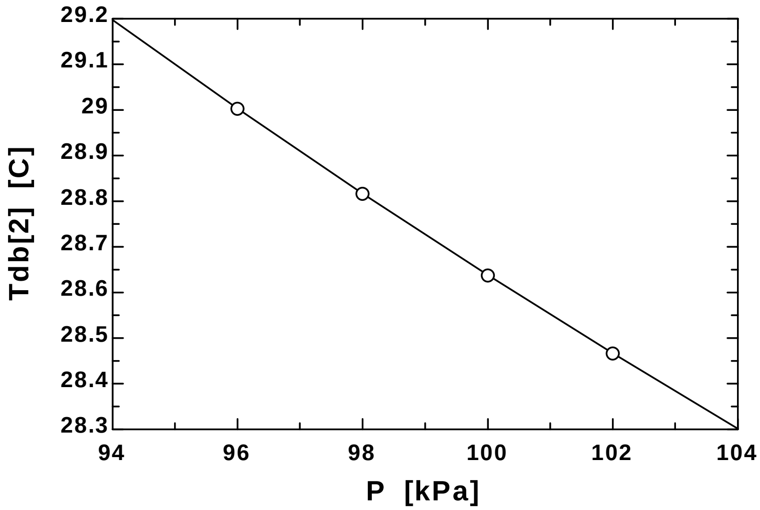
14-144 EES Problem 14-143 is reconsidered. The effect of total pressure in the range 94 to 104 kPa on the results required in the problem is to be studied.

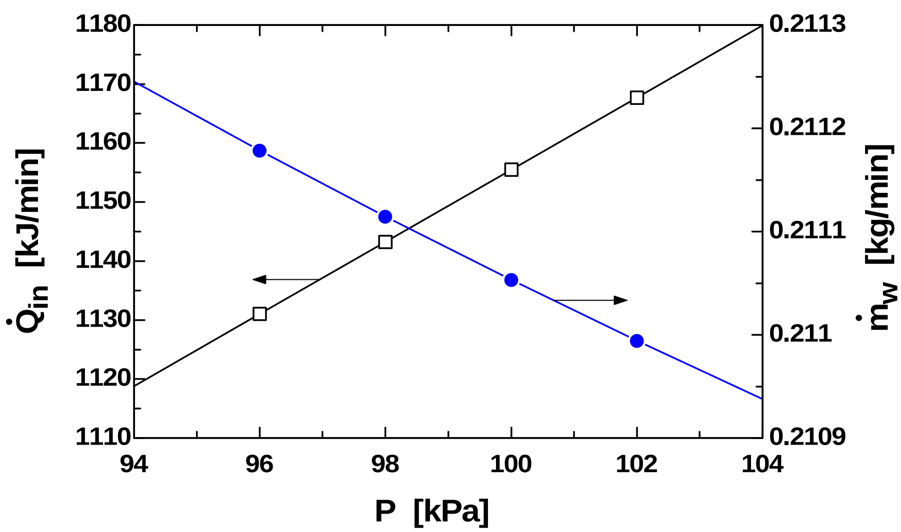
Analysis The problem is solved using EES, and the solution is given below.

```
P=101.325 [kPa]
Tdb[1] =10 [C]
Rh[1] = 0.70
Vol_dot[1]= 50 [m^3/min]
Tdb[3] = 20 [C]
Rh[3] = 0.60
P[1]=P
P[2]=P[1]
P[3]=P[1]

"Energy balance for the steady-flow heating process 1 to 2:"
"We neglect the PE of the flow. Since we don't know the cross sectional area of the flow streams,
we also neglect theKE of the flow."
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 [kJ/min]
E_dot_in = m_dot_a*h[1]+Q_dot_in
E_dot_out = m_dot_a*h[2]
"Conservation of mass of dry air during mixing: m_dot_a = constant"
m_dot_a = Vol_dot[1]/v[1]
"Conservation of mass of water vapor during the heating process:"
m_dot_a*w[1] = m_dot_a*w[2]
"Conservation of mass of water vapor during the evaporative cooler process:"
m_dot_a*w[2]+m_dot_w = m_dot_a*w[3]

"During the evaporative cooler process:"
Twb[2] = Twb[3]
Twb[3] =WETBULB(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
{h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],B=Twb[2])}
h[2]=h[3]
Tdb[2]=TEMPERATURE(AirH2O,h=h[2],P=P[2],w=w[2])

w[2]=HUMRAT(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
h[3]=ENTHALPY(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
w[3]=HUMRAT(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
```

m_w [kg/min]	Q_{in} [kJ/min]	Rh_2	Tdb_2 [C]	P [kPa]
0.2112	1119	0.212	29.2	94
0.2112	1131	0.2144	29	96
0.2111	1143	0.2167	28.82	98
0.2111	1155	0.219	28.64	100
0.211	1168	0.2212	28.47	102
0.2109	1180	0.2233	28.3	104


0.224
0.222
0.22
0.218
0.216
0.214
0.212
Rh[2]
94
96
98
100
102
104
P [kPa]

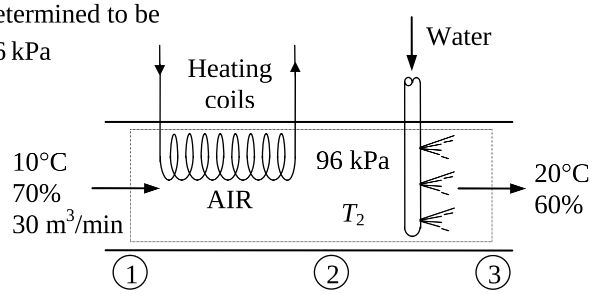

29.2
29.1
29
28.9
28.8
28.7
28.6
28.5
28.4
28.3
Tdb[2] [C]
94
96
98
100
102
104
P [kPa]


1180
1170
1160
1150
1140
1130
1120
1110
$\dot{Q}_{in}$ [kJ/min]
0.2113
0.2112
0.2111
0.211
0.2109
$\dot{m}_w$ [kg/min]
94
96
98
100
102
104
P [kPa]

14-145 Air is heated and dehumidified in an air-conditioning system consisting of a heating section and an evaporative cooler. The temperature and relative humidity of the air when it leaves the heating section, the rate of heat transfer in the heating section, and the rate at which water is added to the air in the evaporative cooler 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis (*a*) Assuming the wet-bulb temperature of the air remains constant during the evaporative cooling process, the properties of air at various states are determined to be



$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ }10°\text{C}} = (0.70)(1.2281\text{ kPa}) = 0.86\text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 96 - 0.86 = 95.14\text{ kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\text{ K})}{95.14\text{ kPa}} = 0.854\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(0.86\text{ kPa})}{(96-0.86)\text{ kPa}} = 0.00562\text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(10°\text{C}) + (0.00562)(2519.2\text{ kJ/kg}) = 24.21\text{ kJ/kg dry air}$$

and

$$P_{v3} = \phi_3 P_{g3} = \phi_3 P_{\text{sat @ }20°\text{C}} = (0.60)(2.3392\text{ kPa}) = 1.40\text{ kPa}$$

$$P_{a3} = P_3 - P_{v3} = 96 - 1.40 = 94.60\text{ kPa}$$

$$\omega_3 = \frac{0.622\,P_{v3}}{P_3 - P_{v3}} = \frac{0.622(1.40\text{ kPa})}{(96-1.40)\text{ kPa}} = 0.00923\text{ kg H}_2\text{O/kg dry air}$$

$$h_3 = c_p T_3 + \omega_3 h_{g3} = (1.005\text{ kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.00921)(2537.4\text{ kJ/kg})$$
$$= 43.52\text{ kJ/kg dry air}$$

Also,

$$h_2 \cong h_3 = 43.52\text{ kJ/kg}$$

$$\omega_2 = \omega_1 = 0.00562\text{ kg H}_2\text{O/kg dry air}$$

Thus,

$$h_2 = c_p T_2 + \omega_2 h_{g2} \cong c_p T_2 + \omega_2(2500.9 + 1.82T_2) = (1.005\text{ kJ/kg}\cdot°\text{C})T_2 + (0.00562)(2500.9 + 1.82T_2)$$

Solving for T_2,

$$T_2 = \mathbf{29.0°C} \longrightarrow P_{g2} = P_{\text{sat@29°C}} = 4.013\text{ kPa}$$

Thus, $$\phi_2 = \frac{\omega_2 P_2}{(0.622+\omega_2)P_{g2}} = \frac{(0.00562)(96)}{(0.622+0.00562)(4.013)} = 0.214 \text{ or } \mathbf{21.4\%}$$

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{30\text{ m}^3/\text{min}}{0.854\text{ m}^3/\text{kg dry air}} = 35.1\text{ kg/min}$$

Then the rate of heat transfer to air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1) = (35.1\text{ kg/min})(43.52 - 24.21)\text{kJ/kg} = \mathbf{679\ kJ/min}$$

(*c*) The rate of water addition to the air in evaporative cooler is

$$\dot{m}_{w,\text{added}} = \dot{m}_{w3} - \dot{m}_{w2} = \dot{m}_a(\omega_3 - \omega_2) = (35.1\text{ kg/min})(0.00923 - 0.00562) = \mathbf{0.127\ kg/min}$$

14-146 [*Also solved by EES on enclosed CD***]** Waste heat from the cooling water is rejected to air in a natural-draft cooling tower. The mass flow rate of the cooling water, the volume flow rate of air, and the mass flow rate of the required makeup water 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 cooling tower is adiabatic.

Analysis (*a*) The mass flow rate of dry air through the tower remains constant $(\dot m_{a1} = \dot m_{a2} = \dot m_a)$, but 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 is made up later in the cycle using water at 27°C. Applying the mass balance and the energy balance equations yields

Dry Air Mass Balance:

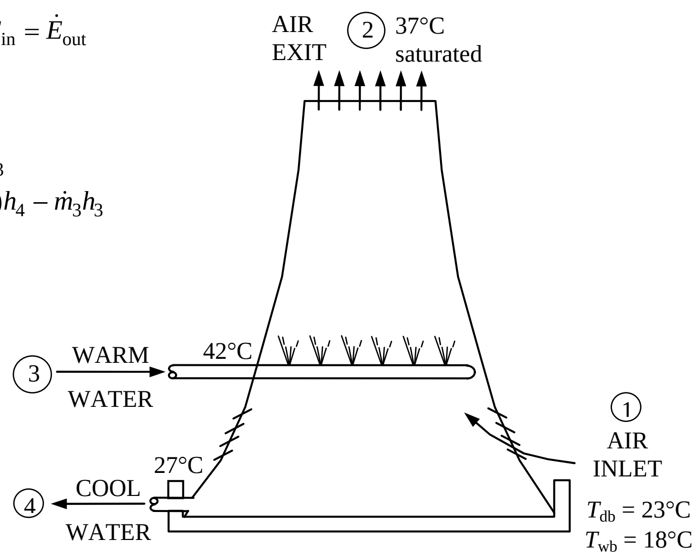
$$\sum \dot m_{a,i} = \sum \dot m_{a,e} \longrightarrow \dot m_{a1} = \dot m_{a2} = \dot m_a$$

Water Mass Balance:

$$\sum \dot m_{w,i} = \sum \dot m_{w,e}$$
$$\dot m_3 + \dot m_{a1}\omega_1 = \dot m_4 + \dot m_{a2}\omega_2$$
$$\dot m_3 - \dot m_4 = \dot m_a(\omega_2 - \omega_1) = \dot m_{\text{makeup}}$$

Energy Balance:

$$\dot E_{\text{in}} - \dot E_{\text{out}} = \Delta \dot E_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0 \longrightarrow \dot E_{\text{in}} = \dot E_{\text{out}}$$
$$\sum \dot m_i h_i = \sum \dot m_e h_e \quad (\text{since } \dot Q = \dot W = 0)$$
$$0 = \sum \dot m_e h_e - \sum \dot m_i h_i$$
$$0 = \dot m_{a2} h_2 + \dot m_4 h_4 - \dot m_{a1} h_1 - \dot m_3 h_3$$
$$0 = \dot m_a (h_2 - h_1) + (\dot m_3 - \dot m_{\text{makeup}}) h_4 - \dot m_3 h_3$$



Solving for $\dot m_a$,

$$\dot m_a = \frac{\dot m_3 (h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1) h_4}$$

From the psychrometric chart (Fig. A-31),

$$h_1 = 50.8 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0109 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.854 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 143.0 \text{ kJ / kg dry air}$$
$$\omega_2 = 0.0412 \text{ kg H}_2\text{O / kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,42°\text{C}} = 175.90 \text{ kJ/kg H}_2\text{O}$$
$$h_4 \cong h_{f\,@\,27°\text{C}} = 113.19 \text{ kJ/kg H}_2\text{O}$$

Substituting

$$\dot m_a = \frac{\dot m_3 (175.90 - 113.19)\text{kJ/kg}}{(143.0 - 50.8)\text{ kJ/kg} - (0.0412 - 0.0109)(113.25)\text{ kJ/kg}} = 0.706\, \dot{\text{m}}_3$$

The mass flow rate of the cooling water is determined by applying the steady flow energy balance equation on the cooling water,

$$\begin{aligned}\dot{Q}_{\text{waste}} &= \dot{m}_3 h_3 - (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 = \dot{m}_3 h_3 - [\dot{m}_3 - \dot{m}_a(\omega_2 - \omega_1)]h_4 \\ &= \dot{m}_3 h_3 - \dot{m}_3[1 - 0.706(0.0412 - 0.0109)]h_4 = \dot{m}_3(h_3 - 0.9786 h_4)\end{aligned}$$

$$50{,}000 \text{ kJ/s} = \dot{m}_3(175.90 - 0.9786 \times 113.19) \text{ kJ/kg} \longrightarrow \dot{m}_3 = \mathbf{768.1 \text{ kg/s}}$$

and

$$\dot{m}_a = 0.706 \dot{m}_3 = (0.706)(768.1 \text{ kg/s}) = 542.3 \text{ kg/s}$$

(*b*) Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a v_1 = (542.3 \text{ kg/s})(0.854 \text{ m}^3/\text{kg}) = \mathbf{463.1 \text{ m}^3/\text{s}}$$

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

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (542.3 \text{ kg/s})(0.0412 - 0.0109) = \mathbf{16.4 \text{ kg/s}}$$

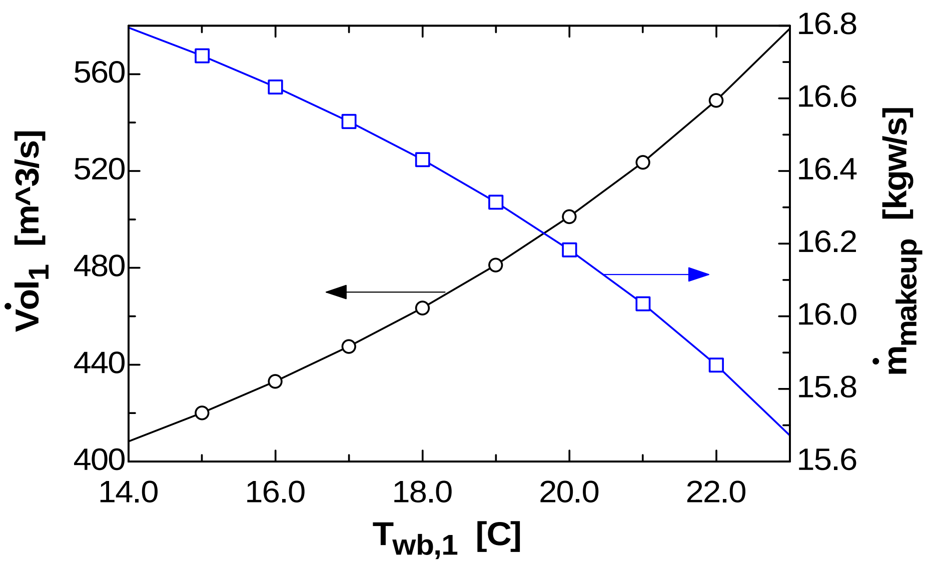
14-147 EES Problem 14-146 is reconsidered. The effect of air inlet wet-bulb temperature on the required air volume flow rate and the makeup water flow rate is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
P_atm =101.325 [kPa]
T_db_1 = 23 [C]
T_wb_1 = 18 [C]
T_db_2 = 37 [C]
RH_2 = 100/100 "%. relative humidity at state 2, saturated condition"
Q_dot_waste = 50 [MW]*Convert(MW, kW)
T_cw_3 = 42 [C] "Cooling water temperature at state 3"
T_cw_4 = 27 [C] "Cooling water temperature at state 4"

"Dry air mass flow rates:"
"RH_1 is the relative humidity at state 1 on a decimal basis"

v_1=VOLUME(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
T_wb_1 = WETBULB(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
m_dot_a_1 = Vol_dot_1/v_1

"Conservaton of mass for the dry air (ma) in the SSSF mixing device:"
m_dot_a_in  - m_dot_a_out = DELTAm_dot_a_cv
m_dot_a_in = m_dot_a_1
m_dot_a_out =  m_dot_a_2
DELTAm_dot_a_cv = 0 "Steady flow requirement"

"Conservation of mass for the water vapor (mv) and cooling water for the SSSF process:"
m_dot_w_in  - m_dot_w_out = DELTAm_dot_w_cv
m_dot_w_in = m_dot_v_1 + m_dot_cw_3
m_dot_w_out =  m_dot_v_2+m_dot_cw_4
DELTAm_dot_w_cv = 0 "Steady flow requirement"
w_1=HUMRAT(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
m_dot_v_1 = m_dot_a_1*w_1
w_2=HUMRAT(AirH2O,T=T_db_2,P=P_atm,R=RH_2)
m_dot_v_2 = m_dot_a_2*w_2

"Conservation of energy for the SSSF cooling tower  process:"
"The process is adiabatic and has no work done, ngelect ke and pe"
E_dot_in_tower - E_dot_out_tower = DELTAE_dot_tower_cv
E_dot_in_tower= m_dot_a_1 *h[1] +  m_dot_cw_3*h_w[3]
E_dot_out_tower =  m_dot_a_2*h[2] + m_dot_cw_4*h_w[4]
DELTAE_dot_tower_cv = 0 "Steady flow requirement"
h[1]=ENTHALPY(AirH2O,T=T_db_1,P=P_atm,w=w_1)
h[2]=ENTHALPY(AirH2O,T=T_db_2,P=P_atm,w=w_2)
h_w[3]=ENTHALPY(steam,T=T_cw_3,x=0)
h_w[4]=ENTHALPY(steam,T=T_cw_4,x=0)


"Energy balance on the external heater determines the cooling water flow rate:"
E_dot_in_heater - E_dot_out_heater = DELTAE_dot_heater_cv
E_dot_in_heater = Q_dot_waste + m_dot_cw_4*h_w[4]
E_dot_out_heater = m_dot_cw_3 * h_w[3]
DELTAE_dot_heater_cv = 0 "Steady flow requirement"
```

"Conservation of mass on the external heater gives the makeup water flow rate."
"Note: The makeup water flow rate equals the amount of water vaporized in the cooling tower."
m_dot_cw_in - m_dot_cw_out = DELTAm_dot_cw_cv
m_dot_cw_in = m_dot_cw_4 + m_dot_makeup
m_dot_cw_out = m_dot_cw_3
DELTAm_dot_cw_cv = 0 "Steady flow requirement"

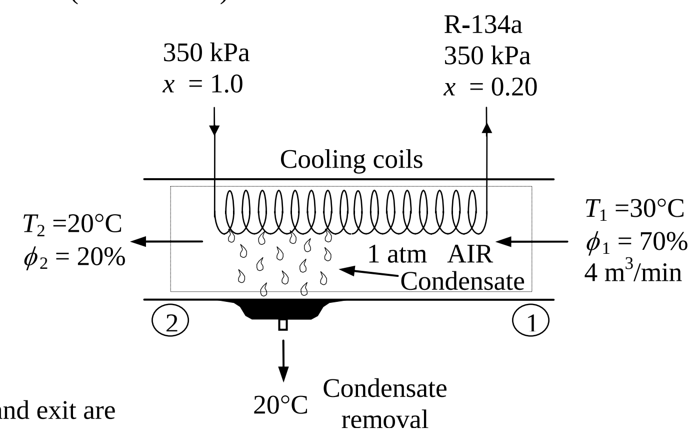
Vol_1 [m^3/s]	m_{makeup} [kgw/s]	m_{cw3} [kgw/s]	m_{a1} [kga/s]	T_{wb1} [C]
408.3	16.8	766.6	481.9	14
420.1	16.72	766.7	495	15
433.2	16.63	766.8	509.4	16
447.5	16.54	767	525.3	17
463.4	16.43	767.2	542.9	18
481.2	16.31	767.4	562.6	19
501.1	16.18	767.7	584.7	20
523.7	16.03	767.9	609.7	21
549.3	15.87	768.2	638.1	22
578.7	15.67	768.6	670.7	23



14-148 Atmospheric air enters an air-conditioning system at a specified pressure, temperature, and relative humidity. The heat transfer, the rate of condensation of water, and the mass flow rate of the refrigerant 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 $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The 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 and exit states may be determined from the psychrometric chart (Figure A-31) or using EES psychrometric functions to be (we used EES)



$$h_1 = 78.24 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01880 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.8847 \text{ m}^3\text{/kg dry air}$$
$$h_2 = 27.45 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.002885 \text{ kg H}_2\text{O/kg dry air}$$

The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{4 \text{ m}^3\text{/min}}{0.8847 \text{ m}^3} = 4.521 \text{ kg/min}$$

The mass flow rates of vapor at the inlet and exit are

$$\dot{m}_{v1} = \omega_1 \dot{m}_a = (0.01880)(4.521 \text{ kg/min}) = 0.0850 \text{ kg/min}$$

$$\dot{m}_{v2} = \omega_2 \dot{m}_a = (0.002885)(4.521 \text{ kg/min}) = 0.01304 \text{ kg/min}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$

where the the enthalpy of condensate water is

$$h_{w2} = h_{f@\,20°\text{C}} = 83.91 \text{ kJ/kg} \quad \text{(Table A - 4)}$$

and the rate of condensation of water vapor is

$$\dot{m}_w = \dot{m}_{v1} - \dot{m}_{v2} = 0.0850 - 0.01304 = \mathbf{0.07196 \text{ kg/min}}$$

Substituting,

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(4.521 \text{ kg/min})(78.24 \text{ kJ/kg}) = \dot{Q}_{\text{out}} + (4.521 \text{ kg/min})(27.45 \text{ kJ/kg}) + (0.07196 \text{ kg/min})(83.91 \text{ kJ/kg})$$
$$\dot{Q}_{\text{out}} = 223.6 \text{ kJ/min} = \mathbf{3.727 \text{ kW}}$$

The properties of the R-134a at the inlet and exit of the cooling section are

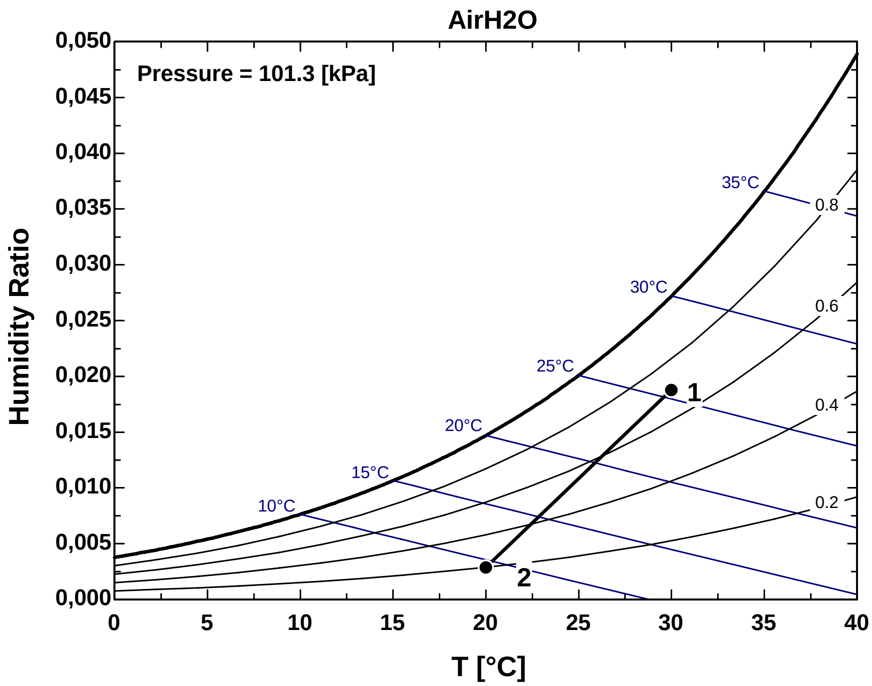
$$\left.\begin{array}{l} P_{R1} = 350 \text{ kPa} \\ x_{R1} = 0.20 \end{array}\right\} h_{R1} = 97.56 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{R2} = 350 \text{ kPa} \\ x_{R2} = 1.0 \end{array}\right\} h_{R2} = 253.34 \text{ kJ/kg}$$

Noting that the rate of heat lost from the air is received by the refrigerant, the mass flow rate of the refrigerant is determined from

$$\dot{m}_R h_{R1} + \dot{Q}_{\text{in}} = \dot{m}_R h_{R2}$$

$$\dot{m}_R = \frac{\dot{Q}_{\text{in}}}{h_{R2} - h_{R1}} = \frac{223.6\text{ kJ/min}}{(253.34 - 97.56)\text{ kJ/kg}} = \mathbf{1.435\ kg/min}$$



14-149 An uninsulated tank contains moist air at a specified state. Water is sprayed into the tank until the relative humidity in the tank reaches a certain value. The amount of water supplied to the tank, the final pressure in the tank, and the heat transfer during the process are to be determined.

Assumptions **1** Dry air and water vapor are ideal gases. **2** The kinetic and potential energy changes are negligible.

Analysis The initial state of the moist air is completely specified. The properties of the air at the inlet state may be determined from the psychrometric chart (Figure A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 49.16 \text{ kJ/kg dry air}, \qquad \omega_1 = 0.005433 \text{ kg H}_2\text{O/kg dry air}$$

$$\boldsymbol{v}_1 = 0.6863 \text{ m}^3 \text{ / kg dry air}$$

The initial mass in the tank is

$$m_a = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.5 \text{ m}^3}{0.6863 \text{ m}^3} = 0.7285 \text{ kg}$$

The partial pressure of dry air in the tank is

$$P_{a2} = \frac{m_a R_a T_2}{\boldsymbol{V}} = \frac{(0.7285 \text{ kg})(0.287 \text{ kJ/kg.K})(35 + 273 \text{ K})}{(0.5 \text{ m}^3)} = 128.8 \text{ kPa}$$

Then, the pressure of moist air in the tank is determined from

$$P_2 = P_{a2}\left(1 + \frac{\omega_2}{0.622}\right) = (128.8 \text{ kPa})\left(1 + \frac{\omega_2}{0.622}\right)$$

We cannot fix the final state explicitly by a hand-solution. However, using EES which has built-in functions for moist air properties, the final state properties are determined to be

$$P_2 = \mathbf{133.87 \text{ kPa}} \qquad \omega_2 = 0.02446 \text{ kg H}_2\text{O/kg dry air}$$

$$h_2 = 97.97 \text{ kJ/kg dry air} \qquad \boldsymbol{v}_2 = 0.6867 \text{ m}^3 \text{ / kg dry air}$$

The partial pressures at the initial and final states are

$$P_{v1} = \phi_1 P_{\text{sat@35°C}} = 0.20(5.6291 \text{ kPa}) = 1.126 \text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 130 - 1.126 = 128.87 \text{ kPa}$$

$$P_{v2} = P_2 - P_{a2} = 133.87 - 128.81 = 5.07 \text{ kPa}$$

The specific volume of water at 35ºC is

$$\boldsymbol{v}_{w1} = \boldsymbol{v}_{w2} = \boldsymbol{v}_{\text{g @35°C}} = 25.205 \text{ m}^3\text{/kg}$$

The internal energies per unit mass of dry air in the tank are

$$u_1 = h_1 - P_{a1}\boldsymbol{v}_1 - w_1 P_{v1}\boldsymbol{v}_{w1} = 49.16 - 128.87 \times 0.6863 - 0.005433 \times 1.126 \times 25.205 = -39.44 \text{ kJ/kg}$$

$$u_2 = h_2 - P_{a2}\boldsymbol{v}_2 - w_2 P_{v2}\boldsymbol{v}_{w2} = 97.97 - 128.81 \times 0.6867 - 0.02446 \times 5.07 \times 25.205 = 6.396 \text{ kJ/kg}$$

The enthalpy of water entering the tank from the supply line is

$$h_{w1} = h_{\text{f @50°C}} = 209.34 \text{ kJ/kg}$$

The internal energy of water vapor at the final state is

$$u_{w2} = u_{\text{g @35°C}} = 2422.7 \text{ kJ/kg}$$

The amount of water supplied to the tank is

$$m_w = m_a(\omega_2 - \omega_1) = (0.7285 \text{ kg})(0.02446 - 0.005433) = \mathbf{0.01386 \text{ kg}}$$

An energy balance on the system gives

$$E_{\text{in}} = \Delta E_{\text{tank}}$$

$$Q_{\text{in}} + m_w h_{w1} = m_a(u_2 - u_1) + m_w u_{w2}$$

$$Q_{\text{in}} + (0.01386 \text{ kg})(209.34 \text{ kJ/kg}) = (0.7285 \text{ kg})\left[6.396 - (-39.44) \text{kJ/kg}\right] + (0.01386 \text{ kg})(2422.7 \text{ kJ/kg})$$

$$\dot{Q}_{\text{in}} = \mathbf{64.1 kJ}$$

Fundamentals of Engineering (FE) Exam Problems

14-150 A room is filled with saturated moist air at 25°C and a total pressure of 100 kPa. If the mass of dry air in the room is 100 kg, the mass of water vapor is

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

Answer (d) 2.04 kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
P=100 "kPa"
m_air=100 "kg"
RH=1
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v
w=0.622*P_v/(P-P_v)
w=m_v/m_air

"Some Wrong Solutions with Common Mistakes:"
W1_vmass=m_air*w1; w1=0.622*P_v/P "Using P instead of P-Pv in w relation"
W2_vmass=m_air "Taking m_vapor = m_air"
W3_vmass=P_v/P*m_air "Using wrong relation"
```

14-151 A room contains 50 kg of dry air and 0.6 kg of water vapor at 25°C and 95 kPa total pressure. The relative humidity of air in the room is

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

Answer (c) 56.7%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
P=95 "kPa"
m_air=50 "kg"
m_v=0.6 "kg"
w=0.622*P_v/(P-P_v)
w=m_v/m_air
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g

"Some Wrong Solutions with Common Mistakes:"
W1_RH=m_v/(m_air+m_v) "Using wrong relation"
W2_RH=P_g/P "Using wrong relation"
```

14-152 A 40-m^3 room contains air at 30°C and a total pressure of 90 kPa with a relative humidity of 75 percent. The mass of dry air in the room is

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

Answer (c) 39.9 kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=40 "m^3"
T1=30 "C"
P=90 "kPa"
RH=0.75
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v
R_air=0.287 "kJ/kg.K"
m_air=P_air*V/(R_air*(T1+273))

"Some Wrong Solutions with Common Mistakes:"
W1_mass=P_air*V/(R_air*T1) "Using C instead of K"
W2_mass=P*V/(R_air*(T1+273)) "Using P instead of P_air"
W3_mass=m_air*RH "Using wrong relation"
```

14-153 A room contains air at 30°C and a total pressure of 96.0 kPa with a relative humidity of 75 percent. The partial pressure of dry air is

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

Answer (c) 92.8 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=30 "C"
P=96 "kPa"
RH=0.75
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v

"Some Wrong Solutions with Common Mistakes:"
W1_Pair=P_v "Using Pv as P_air"
W2_Pair=P-P_g "Using wrong relation"
W3_Pair=RH*P "Using wrong relation"
```

14-154 The air in a house is at 20°C and 50 percent relative humidity. Now the air is cooled at constant pressure. The temperature at which the moisture in the air will start condensing is

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

Answer (d) 9.3°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=20 "C"
RH1=0.50
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH1=P_v/P_g
T_dp=TEMPERATURE(Steam_IAPWS,x=0,P=P_v)

"Some Wrong Solutions with Common Mistakes:"
W1_Tdp=T1*RH1 "Using wrong relation"
W2_Tdp=(T1+273)*RH1-273 "Using wrong relation"
W3_Tdp=WETBULB(AirH2O,T=T1,P=P1,R=RH1); P1=100 "Using wet-bulb temperature"
```

14-155 On the psychrometric chart, a cooling and dehumidification process appears as a line that is

(a) horizontal to the left,
(b) vertical downward,
(c) diagonal upwards to the right (NE direction)
(d) diagonal upwards to the left (NW direction)
(e) diagonal downwards to the left (SW direction)

Answer (e) diagonal downwards to the left (SW direction)

14-156 On the psychrometric chart, a heating and humidification process appears as a line that is

(a) horizontal to the right,
(b) vertical upward,
(c) diagonal upwards to the right (NE direction)
(d) diagonal upwards to the left (NW direction)
(e) diagonal downwards to the right (SE direction)

Answer (c) diagonal upwards to the right (NE direction)

14-157 An air stream at a specified temperature and relative humidity undergoes evaporative cooling by spraying water into it at about the same temperature. The lowest temperature the air stream can be cooled to is

(a) the dry bulb temperature at the given state
(b) the wet bulb temperature at the given state
(c) the dew point temperature at the given state
(d) the saturation temperature corresponding to the humidity ratio at the given state
(e) the triple point temperature of water

Answer (a) the dry bulb temperature at the given state

14-158 Air is cooled and dehumidified as it flows over the coils of a refrigeration system at 85 kPa from 30°C and a humidity ratio of 0.023 kg/kg dry air to 15°C and a humidity ratio of 0.015 kg/kg dry air. If the mass flow rate of dry air is 0.7 kg/s, the rate of heat removal from the air is

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

Answer (e) 25 kJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=85 "kPa"
T1=30 "C"
w1=0.023
T2=15 "C"
w2=0.015
m_air=0.7 "kg/s"
m_water=m_air*(w1-w2)
h1=ENTHALPY(AirH2O,T=T1,P=P,w=w1)
h2=ENTHALPY(AirH2O,T=T2,P=P,w=w2)
h_w=ENTHALPY(Steam_IAPWS,T=T2,x=0)
Q=m_air*(h1-h2)-m_water*h_w

"Some Wrong Solutions with Common Mistakes:"
W1_Q=m_air*(h1-h2) "Ignoring condensed water"
W2_Q=m_air*Cp_air*(T1-T2)-m_water*h_w; Cp_air = 1.005 "Using dry air enthalpies"
W3_Q=m_air*(h1-h2)+m_water*h_w "Using wrong sign"
```

14-159 Air at a total pressure of 90 kPa, 15°C, and 75 percent relative humidity is heated and humidified to 25°C and 75 percent relative humidity by introducing water vapor. If the mass flow rate of dry air is 4 kg/s, the rate at which steam is added to the air is

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

Answer (a) 0.032 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=90 "kPa"
T1=15 "C"
RH1=0.75
T2=25 "C"
RH2=0.75
m_air=4 "kg/s"
w1=HUMRAT(AirH2O,T=T1,P=P,R=RH1)
w2=HUMRAT(AirH2O,T=T2,P=P,R=RH2)
m_water=m_air*(w2-w1)

"Some Wrong Solutions with Common Mistakes:"
W1_mv=0 "sine RH = constant"
W2_mv=w2-w1 "Ignoring mass flow rate of air"
W3_mv=RH1*m_air "Using wrong relation"
```

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