1. An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the rate of heat rejection to the environment, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

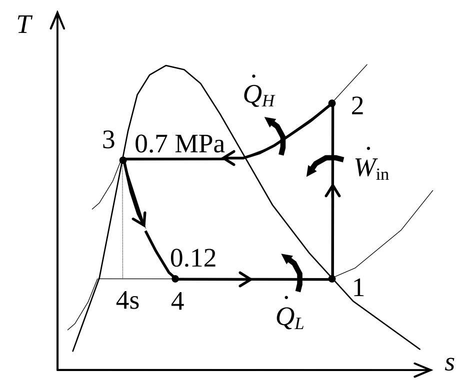
Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 120\text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g\ @\ 120\text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 273.50 \text{ kJ/kg } (T_2 = 34.95°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\ @\ 0.7\text{ MPa}} = 88.82 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 88.82 \text{ kJ/kg (throttling)}$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(236.97 - 88.82) \text{ kJ/kg} = \mathbf{7.41\ \ kW}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(273.50 - 236.97) \text{ kJ/kg} = \mathbf{1.83\ kW}$$

(*b*) The rate of heat rejection to the environment is determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.41 + 1.83 = \mathbf{9.23\ kW}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.41 \text{ kW}}{1.83 \text{ kW}} = \mathbf{4.06}$$

2. 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
26°C
T_{wb} = 21°C

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

$$\omega_2 = \frac{0.622P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(2.488\text{ kPa})}{(100 - 2.488)\text{ kPa}} = 0.01587\text{ kg H}_2\text{O/kg dry air}$$

Thus,

$$\omega_1 = \frac{(1.005\text{ kJ/kg}\cdot°\text{C})(21 - 26)°\text{C} + (0.01587)(2451.2\text{ kJ/kg})}{(2548.3 - 88.10)\text{ kJ/kg}} = \mathbf{0.01377\ kg\ H_2O/kg\ dry\ air}$$

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

$$\phi_1 = \frac{\omega_1 P_1}{(0.622 + \omega_1)P_{g1}} = \frac{(0.01377)(100\text{ kPa})}{(0.622 + 0.01377)(3.3638\text{ kPa})} = 0.644 \quad \text{or} \quad \mathbf{64.4\%}$$

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

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 26°C}} = (0.644)(3.3638\text{ kPa}) = 2.166\text{ kPa}$$

Thus the dew-point temperature of the air is

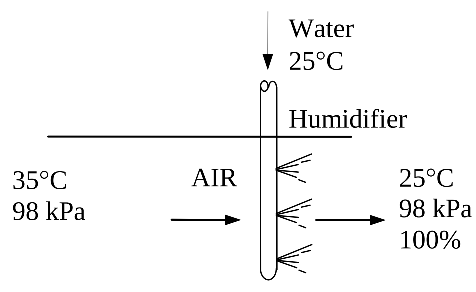
$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 2.166 kPa}} = \mathbf{18.8°C}$$

3. 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

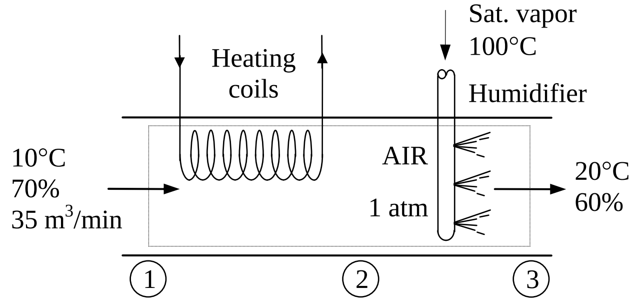
$$h_1 = 77.66\text{ kJ/kg dry air}$$
$$\omega_1 = \mathbf{0.01654\ kg\ H_2O/kg\ dry\ air}$$
$$\phi_1 = \mathbf{0.4511}$$

4. 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 $\omega_2 = 0.0053$ 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,

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

5. 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.

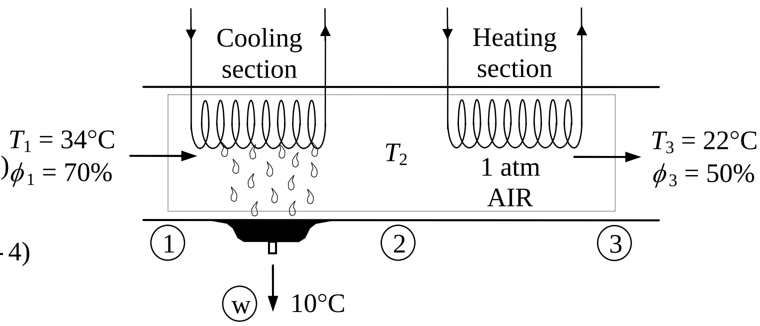
Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

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

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