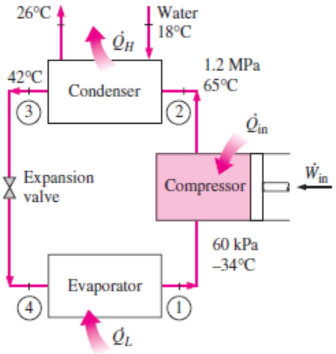
**11–11** A commercial refrigerator with refrigerant-134a as the working fluid is used to keep the refrigerated space at −30°C by rejecting its waste heat to cooling water that enters the condenser at 18°C at a rate of 0.25 kg/s and leaves at 26°C. The refrigerant enters the condenser at 1.2 MPa and 65°C and leaves at 42°C. The inlet state of the compressor is 60 kPa and −34°C and the compressor is estimated to gain a net heat of 450 W from the surroundings. Determine (*a*) the quality of the refrigerant at the evaporator inlet, (*b*) the refrigeration load, (*c*) the COP of the refrigerator, and (*d*) the theoretical maximum refrigeration load for the same power input to the compressor.




**11-11** A commercial refrigerator with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the evaporator inlet, the refrigeration load, the COP of the refrigerator, and the theoretical maximum refrigeration load for the same power input to the compressor are to be determined.

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

***Analysis*** (*a*) From refrigerant-134a tables (Tables A-11 through A-13)

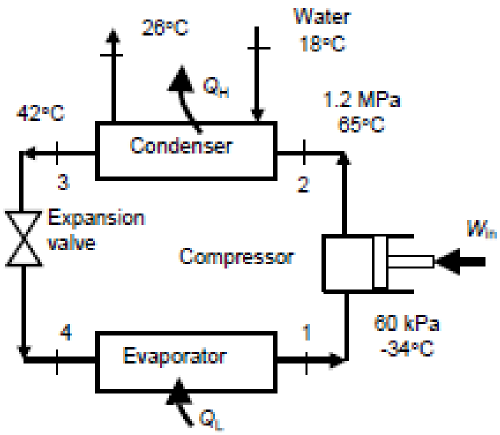
$$\left.\begin{array}{l} P_1 = 60\text{ kPa} \\ T_1 = -34°\text{C} \end{array}\right\} h_1 = 230.03\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1200\text{ kPa} \\ T_2 = 65°\text{C} \end{array}\right\} h_2 = 295.16\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1200\text{ kPa} \\ T_3 = 42°\text{C} \end{array}\right\} h_3 = 111.23\text{ kJ/kg}$$

$$h_4 = h_3 = 111.23\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 60\text{ kPa} \\ h_4 = 111.23\text{ kJ/kg} \end{array}\right\} x_4 = \mathbf{0.4795}$$




Using saturated liquid enthalpy at the given temperature, for water we have (Table A-4)

$$h_{w1} = h_{f@\,18°\text{C}} = 75.47\text{ kJ/kg}$$

$$h_{w2} = h_{f@\,26°\text{C}} = 108.94\text{ kJ/kg}$$

(*b*) The mass flow rate of the refrigerant may be determined from an energy balance on the compressor

$$\dot{m}_R(h_2 - h_3) = \dot{m}_w(h_{w2} - h_{w1})$$

$$\dot{m}_R(295.16 - 111.23)\text{kJ/kg} = (0.25\text{ kg/s})(108.94 - 75.47)\text{kJ/kg}$$

$$\longrightarrow \dot{m}_R = 0.0455\text{ kg/s}$$

The waste heat transferred from the refrigerant, the compressor power input, and the refrigeration load are

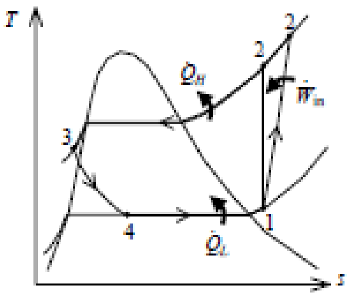
$$\dot{Q}_H = \dot{m}_R(h_2 - h_3) = (0.0455\text{ kg/s})(295.16 - 111.23)\text{kJ/kg} = 8.367\text{ kW}$$

$$\dot{W}_{in} = \dot{m}_R(h_2 - h_1) - \dot{Q}_{in} = (0.0455\text{ kg/s})(295.16 - 230.03)\text{kJ/kg} - 0.45\text{ kW} = 2.513\text{ kW}$$

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{in} = 8.367 - 2.513 = \mathbf{5.85\ kW}$$

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

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{5.85}{2.513} = \mathbf{2.33}$$




(*d*) The reversible COP of the refrigerator for the same temperature limits is

$$\text{COP}_{max} = \frac{1}{T_H / T_L - 1} = \frac{1}{(18 + 273)/(-30 + 273) - 1} = 5.063$$

Then, the maximum refrigeration load becomes

$$\dot{Q}_{L,max} = COP_{max}\dot{W}_{in} = (5.063)(2.513\text{ kW}) = \mathbf{12.72\ kW}$$

**11–12** A refrigerator uses refrigerant-134a as the working fluid and operates on an ideal vapor-compression refrigeration cycle between 0.12 and 0.7 MPa. The mass flow rate of the refrigerant is 0.05 kg/s. Show the cycle on a *T-s* diagram with respect to saturation lines. Determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the rate of heat rejection to the environment, and (*c*) the coefficient of performance. *Answers:* (*a*) 7.41 kW, 1.83 kW, (*b*) 9.23 kW, (*c*) 4.06

**11-12** 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.

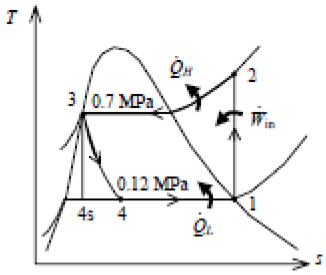
***Analysis*** (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables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

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

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

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

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




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

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

and

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

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

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

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

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

**11–13** Repeat Prob. 11–12 for a condenser pressure of 0.9 MPa.

**11-13** 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.

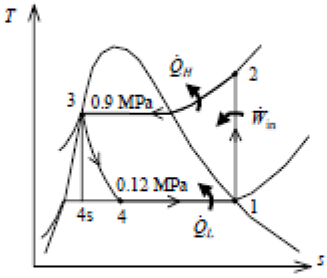
***Analysis*** (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables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

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

$$\left.\begin{array}{l} P_2 = 0.9 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 278.93 \text{ kJ/kg } (T_2 = 44.45°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.9 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.9\text{ MPa}} = 101.61 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 101.61 \text{ kJ/kg (throttling)}$$




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

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(236.97 - 101.61) \text{ kJ/kg} = \mathbf{6.77 \text{ kW}}$$

and

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(278.93 - 236.97) \text{ kJ/kg} = \mathbf{2.10 \text{ kW}}$$

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

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{in} = 6.77 + 2.10 = \mathbf{8.87 \text{ kW}}$$

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

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{6.77 \text{ kW}}{2.10 \text{ kW}} = \mathbf{3.23}$$

**11–14** If the throttling valve in Prob. 11–12 is replaced by an isentropic turbine, determine the percentage increase in the COP and in the rate of heat removal from the refrigerated space. *Answers:* 4.2 percent, 4.2 percent

**11-14** An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The throttling valve in the cycle is replaced by an isentropic turbine. The percentage increase in the COP and in the rate of heat removal from the refrigerated space due to this replacement are to be determined.

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

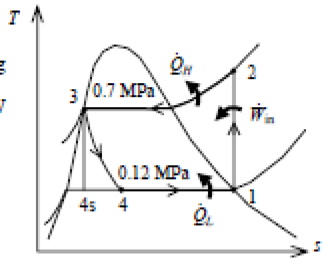
***Analysis*** If the throttling valve in the previous problem is replaced by an isentropic turbine, we would have $s_{4s} = s_3 = s_{f\,@\,0.7\text{ MPa}} = 0.33230$ kJ/kg·K, and the enthalpy at the turbine exit would be

$$x_{4s} = \left(\frac{s_3 - s_f}{s_{fg}}\right)_{@\,120\text{ kPa}} = \frac{0.33230 - 0.09275}{0.85503} = 0.2802$$

$$h_{4s} = (h_f + x_{4s}h_{fg})_{@\,120\text{ kPa}} = 22.49 + (0.2802)(214.48) = 82.58 \text{ kJ/kg}$$

Then, $\dot{Q}_L = \dot{m}(h_1 - h_{4s}) = (0.05 \text{ kg/s})(236.97 - 82.58) \text{ kJ/kg} = 7.72 \text{ kW}$

and $\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{7.72 \text{ kW}}{1.83 \text{ kW}} = 4.23$




Then the percentage increase in $\dot{Q}$ and COP becomes

$$\text{Increase in } \dot{Q}_L = \frac{\Delta\dot{Q}_L}{\dot{Q}_L} = \frac{7.72 - 7.41}{7.41} = \mathbf{4.2\%}$$

$$\text{Increase in COP}_R = \frac{\Delta\text{COP}_R}{\text{COP}_R} = \frac{4.23 - 4.06}{4.06} = \mathbf{4.2\%}$$

**11–15** Consider a 300 kJ/min refrigeration system that operates on an ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid. The refrigerant enters the compressor as saturated vapor at 140 kPa and is compressed to 800 kPa. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the quality of the refrigerant at the end of the throttling process, (*b*) the coefficient of performance, and (*c*) the power input to the compressor.

**11-15** [*Also solved by EES on enclosed CD*] An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the end of the throttling process, the COP, and the power input to the compressor are to be determined.

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

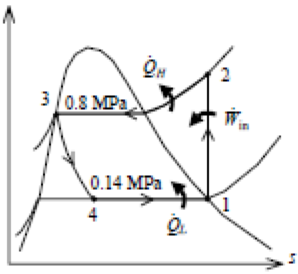
***Analysis*** (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables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

$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,140\text{ kPa}} = 239.16 \text{ kJ/kg} \\ s_1 = s_{g\,@\,140\text{ kPa}} = 0.94456 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 275.37 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.8 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.8\text{ MPa}} = 95.47 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 95.47 \text{ kJ/kg (throttling)}$$




The quality of the refrigerant at the end of the throttling process is

$$x_4 = \left(\frac{h_4 - h_f}{h_{fg}}\right)_{@\,140\text{ kPa}} = \frac{95.47 - 27.08}{212.08} = \mathbf{0.322}$$

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

$$\text{COP}_\text{R} = \frac{q_L}{w_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{239.16 - 95.47}{275.37 - 239.16} = \mathbf{3.97}$$

(*c*) The power input to the compressor is determined from

$$\dot{W}_{in} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{(300/60)\text{kW}}{3.97} = \mathbf{1.26\,kW}$$

**11–17** Repeat Prob. 11–15 assuming an isentropic efficiency of 85 percent for the compressor. Also, determine the rate of exergy destruction associated with the compression process in this case. Take $T_0 = 298$ K.

**11-17** A nonideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the end of the throttling process, the COP, the power input to the compressor, and the irreversibility rate associated with the compression process are to be determined.

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

***Analysis*** (*a*) 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),

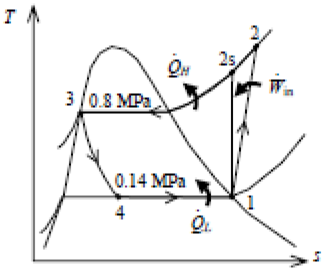
$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,140\text{ kPa}} = 239.16 \text{ kJ/kg} \\ s_1 = s_{g\,@\,140\text{ kPa}} = 0.94456 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 275.37 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} \longrightarrow h_2 = h_1 + (h_{2s}-h_1)/\eta_C = 239.16 + (275.37-239.16)/(0.85) = 281.76 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.8 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.8\text{ MPa}} = 95.47 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 95.47 \text{ kJ/kg (throttling)}$$




The quality of the refrigerant at the end of the throttling process is

$$x_4 = \left(\frac{h_4 - h_f}{h_{fg}}\right)_{@\,140\text{ kPa}} = \frac{95.47-27.08}{212.08} = \mathbf{0.322}$$

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

$$\text{COP}_R = \frac{q_L}{w_{in}} = \frac{h_1-h_4}{h_2-h_1} = \frac{239.16-95.47}{281.76-239.16} = \mathbf{3.37}$$

(*c*) The power input to the compressor is determined from

$$\dot{W}_{in} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{5 \text{ kW}}{3.37} = \mathbf{1.48\ kW}$$

The exergy destruction associated with the compression process is determined from

$$\dot{X}_{destroyed} = T_0 \dot{S}_{gen} = T_0 \dot{m}\left(s_2 - s_1 + \cancelto{0}{\frac{q_{surr}}{T_0}}\right) = T_0 \dot{m}(s_2 - s_1)$$

where

$$\dot{m} = \frac{\dot{Q}_L}{q_L} = \frac{\dot{Q}_L}{h_1-h_4} = \frac{5 \text{ kJ/s}}{(239.16-95.47) \text{ kJ/kg}} = 0.0348 \text{ kg/s}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ h_2 = 281.76 \text{ kJ/kg} \end{array}\right\} s_2 = 0.96483 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\dot{X}_{destroyed} = (298 \text{ K})(0.0348 \text{ kg/s})(0.96483 - 0.94456) \text{ kJ/kg}\cdot\text{K} = \mathbf{0.210\ kW}$$

**11–18** Refrigerant-134a enters the compressor of a refrigerator as superheated vapor at 0.14 MPa and −10°C at a rate of 0.12 kg/s, and it leaves at 0.7 MPa and 50°C. The refrigerant is cooled in the condenser to 24°C and 0.65 MPa, and it is throttled to 0.15 MPa. Disregarding any heat transfer and pressure drops in the connecting lines between the components, show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the isentropic efficiency of the compressor, and (*c*) the COP of the refrigerator. *Answers:* (*a*) 19.4 kW, 5.06 kW, (*b*) 82.5 percent, (*c*) 3.83

**11-18** A refrigerator 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 isentropic efficiency of the compressor, and the COP of the refrigerator are to be determined.

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

***Analysis*** (*a*) From the refrigerant tables (Tables A-12 and A-13),

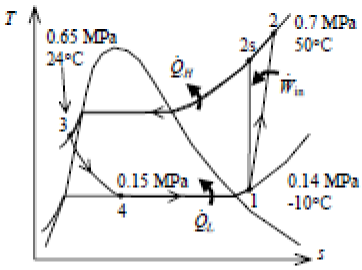
$$\left.\begin{array}{l} P_1 = 0.14 \text{ MPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.97236 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ T_2 = 50°\text{C} \end{array}\right\} h_2 = 288.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{2s} = 0.7 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.16 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.65 \text{ MPa} \\ T_3 = 24°\text{C} \end{array}\right\} h_3 = h_{f\,@\,24°\text{C}} = 84.98 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 84.98 \text{ kJ/kg (throttling)}$$




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

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.12 \text{ kg/s})(246.36 - 84.98) \text{ kJ/kg} = \mathbf{19.4 \text{ kW}}$$

and

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = (0.12 \text{ kg/s})(288.53 - 246.36) \text{ kJ/kg} = \mathbf{5.06 \text{ kW}}$$

(*b*) The adiabatic efficiency of the compressor is determined from

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{281.16 - 246.36}{288.53 - 246.36} = \mathbf{82.5\%}$$

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

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{19.4 \text{ kW}}{5.06 \text{ kW}} = \mathbf{3.83}$$

**11–20** Refrigerant-134a enters the compressor of a refrigerator at 140 kPa and −10°C at a rate of 0.3 m³/min and leaves at 1 MPa. The isentropic efficiency of the compressor is 78 percent. The refrigerant enters the throttling valve at 0.95 MPa and 30°C and leaves the evaporator as saturated vapor at −18.5°C. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the power input to the compressor, (*b*) the rate of heat removal from the refrigerated space, and (*c*) the pressure drop and rate of heat gain in the line between the evaporator and the compressor. *Answers:* (*a*) 1.88 kW, (*b*) 4.99 kW, (*c*) 1.65 kPa, 0.241 kW

**11-20** A refrigerator with refrigerant-134a as the working fluid is considered. The power input to the compressor, the rate of heat removal from the refrigerated space, and the pressure drop and the rate of heat gain in the line between the evaporator and the compressor are to be determined.

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

***Analysis*** (*a*) From the refrigerant tables (Tables A-12 and A-13),

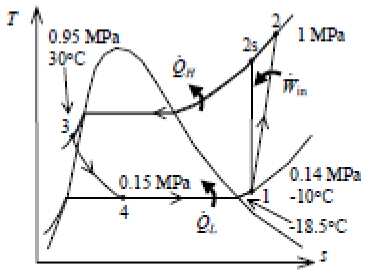
$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.97236 \text{ kJ/kg}\cdot\text{K} \\ v_1 = 0.14605 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.0 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 289.20 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.95 \text{ MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\,@\,30°\text{C}} = 93.58 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 93.58 \text{ kJ/kg (throttling)}$$

$$\left.\begin{array}{l} T_5 = -18.5°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} P_5 = 0.14165 \text{ MPa} \\ h_5 = 239.33 \text{ kJ/kg} \end{array}$$




Then the mass flow rate of the refrigerant and the power input becomes

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.3/60 \text{ m}^3\text{/s}}{0.14605 \text{ m}^3\text{/kg}} = 0.03423 \text{ kg/s}$$

$$\dot{W}_{in} = \dot{m}(h_{2s} - h_1)/\eta_C = (0.03423 \text{ kg/s})[(289.20 - 246.36) \text{ kJ/kg}]/(0.78) = \mathbf{1.88 \text{ kW}}$$

(*b*) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}(h_5 - h_4) = (0.03423 \text{ kg/s})(239.33 - 93.58) \text{ kJ/kg} = \mathbf{4.99 \text{ kW}}$$

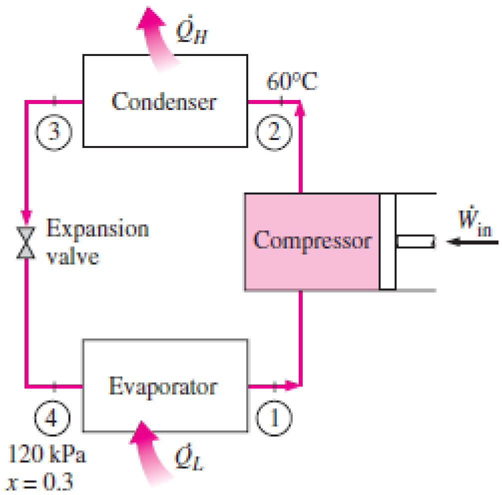
(*c*) The pressure drop and the heat gain in the line between the evaporator and the compressor are

$$\Delta P = P_5 - P_1 = 141.65 - 140 = \mathbf{1.65}$$

and

$$\dot{Q}_{gain} = \dot{m}(h_1 - h_5) = (0.03423 \text{ kg/s})(246.36 - 239.33) \text{ kJ/kg} = \mathbf{0.241 \text{ kW}}$$

**11–22** A refrigerator uses refrigerant-134a as the working fluid and operates on the ideal vapor-compression refrigeration cycle. The refrigerant enters the evaporator at 120 kPa with a quality of 30 percent and leaves the compressor at




60°C. If the compressor consumes 450 W of power, determine (*a*) the mass flow rate of the refrigerant, (*b*) the condenser pressure, and (*c*) the COP of the refrigerator. *Answers:* (*a*) 0.00727 kg/s, (*b*) 672 kPa, (*c*) 2.43

**11-22** A refrigerator uses refrigerant-134a as the working fluid and operates on the ideal vapor-compression refrigeration cycle. The mass flow rate of the refrigerant, the condenser pressure, and the COP of the refrigerator are to be determined.

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

***Analysis*** (*a*) (*b*) From the refrigerant-134a tables (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_4 = 120\text{ kPa} \\ x_4 = 0.30 \end{array}\right\} h_4 = 86.83\text{ kJ/kg}$$

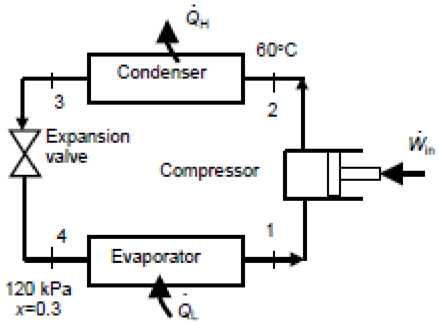
$$h_3 = h_4$$

$$\left.\begin{array}{l} h_3 = 86.83\text{ kJ/kg} \\ x_3 = 0\text{ (sat. liq.)} \end{array}\right\} P_3 = \mathbf{671.8\text{ kPa}}$$

$$P_2 = P_3$$

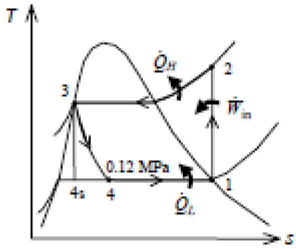
$$\left.\begin{array}{l} P_2 = 671.8\text{ kPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 298.87\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_1 = P_4 = 120\text{ kPa} \\ x_1 = 1\text{ (sat. vap.)} \end{array}\right\} h_1 = 236.97\text{ kJ/kg}$$




The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{W}_{in}}{h_2 - h_1} = \frac{0.45\text{ kW}}{(298.87 - 236.97)\text{kJ/kg}} = \mathbf{0.00727\text{ kg/s}}$$




(*c*) The refrigeration load and the COP are

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.0727\text{ kg/s})(236.97 - 86.83)\text{kJ/kg} = 1.091\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{1.091\text{ kW}}{0.45\text{ kW}} = \mathbf{2.43}$$

**11–31** A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is used to heat water from 15 to 45°C at a rate of 0.12 kg/s. The condenser and evaporator pressures are 1.4 and 0.32 MPa, respectively. Determine the power input to the heat pump.

**11-31** A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. The power input to the heat pump is to be determined.

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

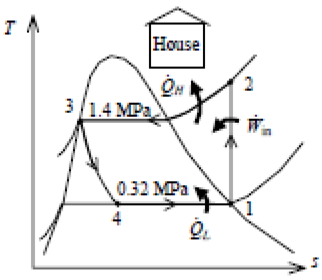
***Analysis*** In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12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

$$\left.\begin{array}{l} P_1 = 320\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,320\text{ kPa}} = 251.88\text{ kJ/kg} \\ s_1 = s_{g\,@\,320\text{ kPa}} = 0.93006\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.4\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 282.54\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.4\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,1.4\text{ MPa}} = 127.22\text{ kJ/kg}$$

$$h_4 \cong h_3 = 127.22\text{ kJ/kg} \quad (\text{throttling})$$




The heating load of this heat pump is determined from

$$\dot{Q}_H = [\dot{m}c(T_2 - T_1)]_{\text{water}}$$
$$= (0.12\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})(45 - 15)°\text{C} = 15.05\text{ kW}$$

and

$$\dot{m}_R = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{15.05\text{ kJ/s}}{(282.54 - 127.22)\text{kJ/kg}} = 0.09688\text{ kg/s}$$

Then,

$$\dot{W}_{in} = \dot{m}_R(h_2 - h_1) = (0.09688\text{ kg/s})(282.54 - 251.88)\text{kJ/kg} = \mathbf{2.97\text{ kW}}$$

**11–32** A heat pump using refrigerant-134a heats a house by using underground water at 8°C as the heat source. The house is losing heat at a rate of 60,000 kJ/h. The refrigerant enters the compressor at 280 kPa and 0°C, and it leaves at 1 MPa and 60°C. The refrigerant exits the condenser at 30°C. Determine (*a*) the power input to the heat pump, (*b*) the rate of heat absorption from the water, and (*c*) the increase in electric power input if an electric resistance heater is used instead of a heat pump. *Answers:* (*a*) 3.55 kW, (*b*) 13.12 kW, (*c*) 13.12 kW

**11-32** A heat pump with refrigerant-134a as the working fluid heats a house by using underground water as the heat source. The power input to the heat pump, the rate of heat absorption from the water, and the increase in electric power input if an electric resistance heater is used instead of a heat pump are to be determined.

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

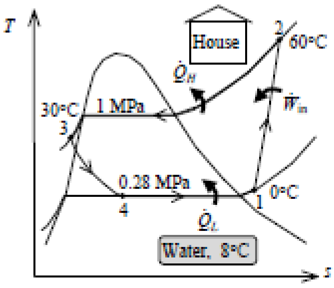
***Analysis*** (*a*) From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 280 \text{ kPa} \\ T_1 = 0°\text{C} \end{array}\right\} h_1 = 250.83 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1.0 \text{ MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 293.38 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.0 \text{ MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\,@\,30°\text{C}} = 93.58 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 93.58 \text{ kJ/kg} \quad (\text{throttling})$$




The mass flow rate of the refrigerant is

$$\dot{m}_R = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{60{,}000/3{,}600 \text{ kJ/s}}{(293.38 - 93.58) \text{ kJ/kg}} = 0.08341 \text{ kg/s}$$

Then the power input to the compressor becomes

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = (0.08341 \text{ kg/s})(293.38 - 250.83) \text{ kJ/kg} = \mathbf{3.55 \text{ kW}}$$

(*b*) The rate of hat absorption from the water is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.08341 \text{ kg/s})(250.83 - 93.58) \text{ kJ/kg} = \mathbf{13.12 \text{ kW}}$$

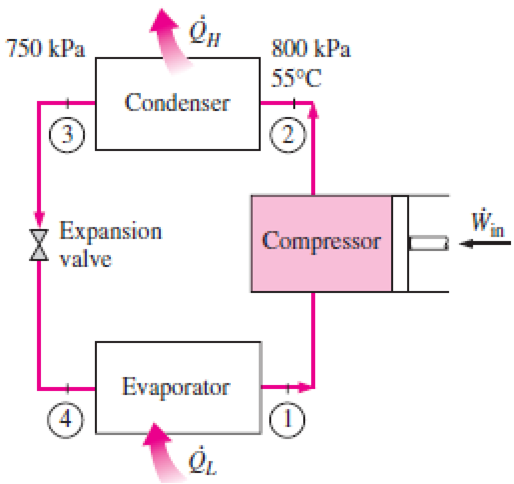
(*c*) The electrical power required without the heat pump is

$$\dot{W}_e = \dot{Q}_H = 60{,}000/3600 \text{ kJ/s} = 16.67 \text{ kW}$$

Thus,

$$\dot{W}_{increase} = \dot{W}_e - \dot{W}_{in} = 16.67 - 3.55 = \mathbf{13.12 \text{ kW}}$$

**11–34** Refrigerant-134a enters the condenser of a residential heat pump at 800 kPa and 55°C at a rate of 0.018 kg/s and leaves at 750 kPa subcooled by 3°C. The refrigerant enters the compressor at 200 kPa superheated by 4°C. Determine (*a*) the isentropic efficiency of the compressor, (*b*) the rate of heat supplied to the heated room, and (*c*) the COP of the heat pump. Also, determine (*d*) the COP and the rate of heat supplied to the heated room if this heat pump operated on the ideal vapor-compression cycle between the pressure limits of 200 and 800 kPa.



**11-34** An actual heat pump cycle with R-134a as the refrigerant is considered. The isentropic efficiency of the compressor, the rate of heat supplied to the heated room, the COP of the heat pump, and the COP and the rate of heat supplied to the heated room if this heat pump operated on the ideal vapor-compression cycle between the same pressure limits are to be determined.

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

***Analysis*** (*a*) The properties of refrigerant-134a are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 55°\text{C} \end{array}\right\} h_2 = 291.76 \text{ kJ/kg}$$

$$T_3 = T_{\text{sat@750 kPa}} = 29.06°\text{C}$$

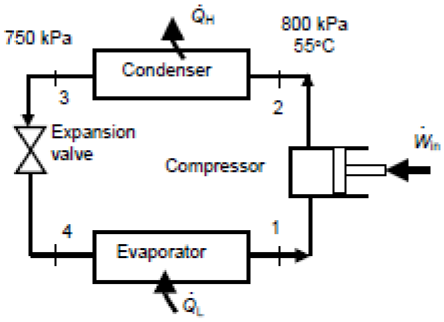
$$\left.\begin{array}{l} P_3 = 750 \text{ kPa} \\ T_3 = (29.06-3)°\text{C} \end{array}\right\} h_3 = 87.91 \text{ kJ/kg}$$

$$h_4 = h_3 = 87.91 \text{ kJ/kg}$$

$$T_{\text{sat@200kPa}} = -10.09°\text{C}$$

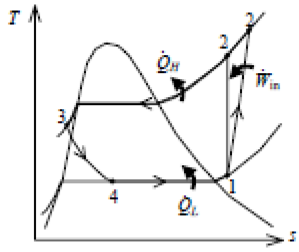
$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = (-10.09+4)°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 247.87 \text{ kJ/kg} \\ s_1 = 0.9506 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 277.26$$




The isentropic efficiency of the compressor is

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{277.26-247.87}{291.76-247.87} = \mathbf{0.670}$$




(*b*) The rate of heat supplied to the room is

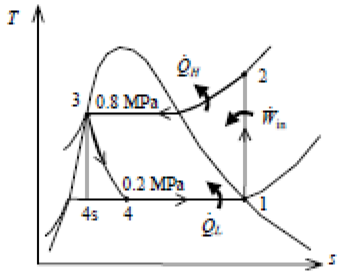
$$\dot{Q}_H = \dot{m}(h_2-h_3) = (0.018 \text{ kg/s})(291.76-87.91)\text{kJ/kg} = \mathbf{3.67 \text{ kW}}$$

(*c*) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}(h_2-h_1) = (0.018 \text{ kg/s})(291.76-247.87)\text{kJ/kg} = 0.790 \text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{3.67}{0.790} = \mathbf{4.64}$$

(*d*) The ideal vapor-compression cycle analysis of the cycle is as follows:




$$h_1 = h_{g@\,200\text{ kPa}} = 244.46 \text{ kJ/kg}$$

$$s_1 = s_{g@\,200\text{ kPa}} = 0.9377 \text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 273.25 \text{ kJ/kg}$$

$$h_3 = h_{f@\,800\text{ kPa}} = 95.47 \text{ kJ/kg}$$

$$h_4 = h_3$$

$$\text{COP} = \frac{h_2-h_3}{h_2-h_1} = \frac{273.25-95.47}{273.25-244.46} = \mathbf{6.18}$$

$$\dot{Q}_H = \dot{m}(h_2-h_3) = (0.018 \text{ kg/s})(273.25-95.47)\text{kJ/kg} = \mathbf{3.20 \text{ kW}}$$

**11–35** A heat pump with refrigerant-134a as the working fluid is used to keep a space at 25°C by absorbing heat from geothermal water that enters the evaporator at 50°C at a rate of 0.065 kg/s and leaves at 40°C. The refrigerant enters the evaporator at 20°C with a quality of 23 percent and leaves at the inlet pressure as saturated vapor. The refrigerant loses 300 W of heat to the surroundings as it flows through the compressor and the refrigerant leaves the compressor at 1.4 MPa at the same entropy as the inlet. Determine (*a*) the degrees of

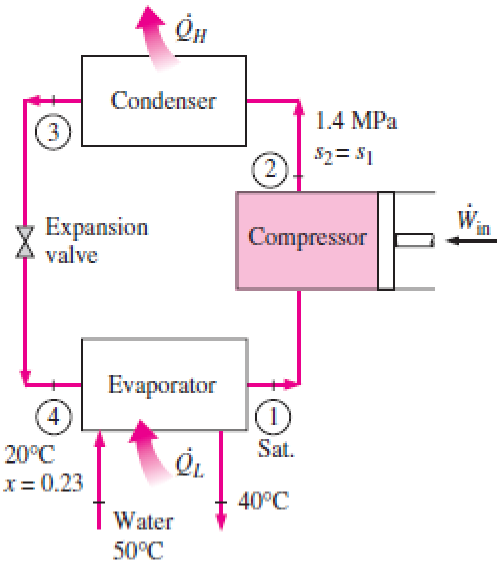


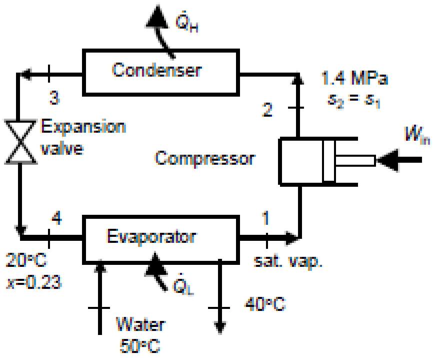

**FIGURE P11–35**

subcooling of the refrigerant in the condenser, (*b*) the mass flow rate of the refrigerant, (*c*) the heating load and the COP of the heat pump, and (*d*) the theoretical minimum power input to the compressor for the same heating load. *Answers:* (*a*) 3.8°C, (*b*) 0.0194 kg/s, (*c*) 3.07 kW, 4.68, (*d*) 0.238 kW

**11-35** A geothermal heat pump is considered. The degrees of subcooling done on the refrigerant in the condenser, the mass flow rate of the refrigerant, the heating load, the COP of the heat pump, the minimum power input are to be determined.

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

***Analysis*** (*a*) From the refrigerant-134a tables (Tables A-11 through A-13)




$$\left.\begin{array}{l} T_4 = 20°\text{C} \\ x_4 = 0.23 \end{array}\right\} \begin{array}{l} P_4 = 572.1\text{ kPa} \\ h_4 = 121.24\text{ kJ/kg} \end{array}$$

$$h_3 = h_4$$

$$\left.\begin{array}{l} P_1 = 572.1\text{ kPa} \\ x_1 = 1\text{ (sat. vap.)} \end{array}\right\} \begin{array}{l} h_1 = 261.59\text{ kJ/kg} \\ s_1 = 0.9223\text{ kJ/kg} \end{array}$$

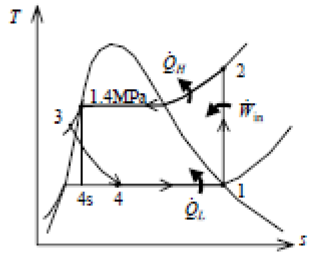
$$\left.\begin{array}{l} P_2 = 1400\text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 280.00\text{ kJ/kg}$$

From the steam tables (Table A-4)

$$h_{w1} = h_{f\,@\,50°\text{C}} = 209.34\text{ kJ/kg}$$

$$h_{w2} = h_{f\,@\,40°\text{C}} = 167.53\text{ kJ/kg}$$

The saturation temperature at the condenser pressure of 1400 kPa and the actual temperature at the condenser outlet are




$$T_{\text{sat}\,@\,1400\text{ kPa}} = 52.40°\text{C}$$

$$\left.\begin{array}{l} P_3 = 1400\text{ kPa} \\ h_3 = 121.24\text{ kJ} \end{array}\right\} T_3 = 48.59°\text{C} \text{ (from EES)}$$

Then, the degrees of subcooling is

$$\Delta T_{\text{subcool}} = T_{\text{sat}} - T_3 = 52.40 - 48.59 = \mathbf{3.81°C}$$

(*b*) The rate of heat absorbed from the geothermal water in the evaporator is

$$\dot{Q}_L = \dot{m}_w(h_{w1} - h_{w2}) = (0.065\text{ kg/s})(209.34 - 167.53)\text{kJ/kg} = 2.718\text{ kW}$$

This heat is absorbed by the refrigerant in the evaporator

$$\dot{m}_R = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{2.718\text{ kW}}{(261.59 - 121.24)\text{kJ/kg}} = \mathbf{0.01936\ kg/s}$$

(*c*) The power input to the compressor, the heating load and the COP are

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) + \dot{Q}_{\text{out}} = (0.01936\text{ kg/s})(280.00 - 261.59)\text{kJ/kg} = 0.6564\text{ kW}$$

$$\dot{Q}_H = \dot{m}_R(h_2 - h_3) = (0.01936\text{ kg/s})(280.00 - 121.24)\text{kJ/kg} = \mathbf{3.074\ kW}$$

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{in}} = \frac{3.074\text{ kW}}{0.6564\text{ kW}} = \mathbf{4.68}$$

(*d*) The reversible COP of the cycle is

$$\text{COP}_{\text{rev}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (25 + 273)/(50 + 273)} = 12.92$$

The corresponding minimum power input is

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{rev}}} = \frac{3.074\text{ kW}}{12.92} = \mathbf{0.238\ kW}$$

## Innovative Refrigeration Systems

**11–42** Consider a two-stage cascade refrigeration system operating between the pressure limits of 0.8 and 0.14 MPa.

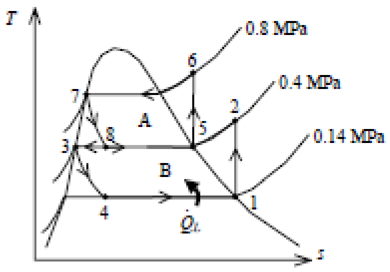
Each stage operates on the ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid. Heat rejection from the lower cycle to the upper cycle takes place in an adiabatic counterflow heat exchanger where both streams enter at about 0.4 MPa. If the mass flow rate of the refrigerant through the upper cycle is 0.24 kg/s, determine (*a*) the mass flow rate of the refrigerant through the lower cycle, (*b*) the rate of heat removal from the refrigerated space and the power input to the compressor, and (*c*) the coefficient of performance of this cascade refrigerator.
*Answers:* (*a*) 0.195 kg/s, (*b*) 34.2 kW, 7.63 kW, (*c*) 4.49

**11-42** A two-stage cascade refrigeration system is considered. Each stage operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid. The mass flow rate of refrigerant through the lower cycle, the rate of heat removal from the refrigerated space, the power input to the compressor, and the COP of this cascade refrigerator are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The heat exchanger is adiabatic.

***Analysis*** (*a*) Each stage of the cascade refrigeration cycle is said to operate on the ideal vapor compression refrigeration cycle. Thus the compression process is isentropic, and the refrigerant enters the compressor as a saturated vapor at the evaporator pressure. Also, the refrigerant leaves the condenser as a saturated liquid at the condenser pressure. The enthalpies of the refrigerant at all 8 states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 239.16 \text{ kJ/kg}, \quad h_2 = 260.58 \text{ kJ/kg}$$
$$h_3 = 63.94 \text{ kJ/kg}, \quad h_4 = 63.94 \text{ kJ/kg}$$
$$h_5 = 255.55 \text{ kJ/kg}, \quad h_6 = 269.91 \text{ kJ/kg}$$
$$h_7 = 95.47 \text{ kJ/kg}, \quad h_8 = 95.47 \text{ kJ/kg}$$




The mass flow rate of the refrigerant through the lower cycle is determined from an energy balance on the heat exchanger:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$\dot{m}_A(h_5 - h_8) = \dot{m}_B(h_2 - h_3)$$
$$\dot{m}_B = \frac{h_5 - h_8}{h_2 - h_3}\dot{m}_A = \frac{255.55 - 95.47}{260.58 - 63.94}(0.24 \text{ kg/s}) = \mathbf{0.1954 \text{ kg/s}}$$

(*b*) The rate of heat removed by a cascade cycle is the rate of heat absorption in the evaporator of the lowest stage. The power input to a cascade cycle is the sum of the power inputs to all of the compressors:

$$\dot{Q}_L = \dot{m}_B(h_1 - h_4) = (0.1954 \text{ kg/s})(239.16 - 63.94) \text{ kJ/kg} = \mathbf{34.24 \text{ kW}}$$
$$\dot{W}_{in} = \dot{W}_{compI,in} + \dot{W}_{compII,in} = \dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1)$$
$$= (0.24 \text{ kg/s})(269.91 - 255.55) \text{ kJ/kg} + (0.1954 \text{ kg/s})(260.58 - 239.16) \text{ kJ/kg}$$
$$= \mathbf{7.63 \text{ kW}}$$

(*c*) The COP of this refrigeration system is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{34.24 \text{ kW}}{7.63 \text{ kW}} = \mathbf{4.49}$$

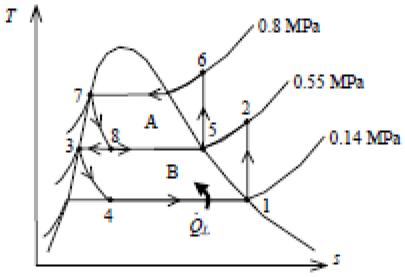
**11–43** Repeat Prob. 11–42 for a heat exchanger pressure of 0.55 MPa.

**11-43** A two-stage cascade refrigeration system is considered. Each stage operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid. The mass flow rate of refrigerant through the lower cycle, the rate of heat removal from the refrigerated space, the power input to the compressor, and the COP of this cascade refrigerator are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The heat exchanger is adiabatic.

***Analysis*** (*a*) Each stage of the cascade refrigeration cycle is said to operate on the ideal vapor compression refrigeration cycle. Thus the compression process is isentropic, and the refrigerant enters the compressor as a saturated vapor at the evaporator pressure. Also, the refrigerant leaves the condenser as a saturated liquid at the condenser pressure. The enthalpies of the refrigerant at all 8 states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 239.16 \text{ kJ/kg}, \quad h_2 = 267.34 \text{ kJ/kg}$$
$$h_3 = 77.54 \text{ kJ/kg}, \quad h_4 = 77.54 \text{ kJ/kg}$$
$$h_5 = 260.92 \text{ kJ/kg}, \quad h_6 = 268.66 \text{ kJ/kg}$$
$$h_7 = 95.47 \text{ kJ/kg}, \quad h_8 = 95.47 \text{ kJ/kg}$$




The mass flow rate of the refrigerant through the lower cycle is determined from an energy balance on the heat exchanger:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$\dot{m}_A(h_5 - h_8) = \dot{m}_B(h_2 - h_3)$$
$$\dot{m}_B = \frac{h_5 - h_8}{h_2 - h_3}\dot{m}_A = \frac{260.92 - 95.47}{267.34 - 77.54}(0.24 \text{ kg/s}) = \mathbf{0.2092 \text{ kg/s}}$$

(*b*) The rate of heat removed by a cascade cycle is the rate of heat absorption in the evaporator of the lowest stage. The power input to a cascade cycle is the sum of the power inputs to all of the compressors:

$$\dot{Q}_L = \dot{m}_B(h_1 - h_4) = (0.2092 \text{ kg/s})(239.16 - 77.54) \text{ kJ/kg} = \mathbf{33.81 \text{ kW}}$$
$$\dot{W}_{in} = \dot{W}_{compI,in} + \dot{W}_{compII,in} = \dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1)$$
$$= (0.24 \text{ kg/s})(268.66 - 260.92) \text{ kJ/kg} + (0.2092 \text{ kg/s})(267.34 - 239.16) \text{ kJ/kg}$$
$$= \mathbf{7.75 \text{ kW}}$$

(*c*) The COP of this refrigeration system is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{33.81 \text{ kW}}{7.75 \text{ kW}} = \mathbf{4.36}$$

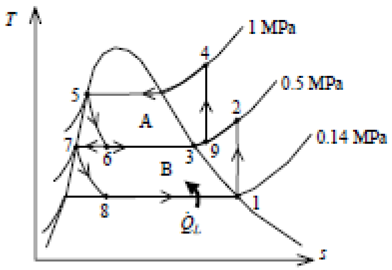
**11–44** A two-stage compression refrigeration system operates with refrigerant-134a between the pressure limits of 1 and 0.14 MPa. The refrigerant leaves the condenser as a saturated liquid and is throttled to a flash chamber operating at 0.5 MPa. The refrigerant leaving the low-pressure compressor at 0.5 MPa is also routed to the flash chamber. The vapor in the flash chamber is then compressed to the condenser pressure by the high-pressure compressor, and the liquid is throttled to the evaporator pressure. Assuming the refrigerant leaves the evaporator as saturated vapor and both compressors are isentropic, determine (*a*) the fraction of the refrigerant that evaporates as it is throttled to the flash chamber, (*b*) the rate of heat removed from the refrigerated space for a mass flow rate of 0.25 kg/s through the condenser, and (*c*) the coefficient of performance.

**11-44** [*Also solved by EES on enclosed CD*] A two-stage compression refrigeration system with refrigerant-134a as the working fluid is considered. The fraction of the refrigerant that evaporates as it is throttled to the flash chamber, the rate of heat removed from the refrigerated space, and the COP are to be determined.

***Assumptions*** 1 Steady operating conditions exist. 2 Kinetic and potential energy changes are negligible. 3 The flash chamber is adiabatic.

***Analysis*** (*a*) The enthalpies of the refrigerant at several states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 239.16 \text{ kJ/kg}, \quad h_2 = 265.31 \text{ kJ/kg}$$
$$h_3 = 259.30 \text{ kJ/kg},$$
$$h_5 = 107.32 \text{ kJ/kg}, \quad h_6 = 107.32 \text{ kJ/kg}$$
$$h_7 = 73.33 \text{ kJ/kg}, \quad h_8 = 73.33 \text{ kJ/kg}$$




The fraction of the refrigerant that evaporates as it is throttled to the flash chamber is simply the quality at state 6,

$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \frac{107.32 - 73.33}{185.98} = \mathbf{0.1828}$$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}^{\;\not{0}\,(steady)} = 0$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$
$$h_9 = (0.1828)(259.30) + (1 - 0.1828)(265.31) = 264.21 \text{ kJ/kg}$$

also,

$$\left.\begin{array}{l} P_4 = 1 \text{ MPa} \\ s_4 = s_9 = 0.94083 \text{ kJ/kg}\cdot\text{K} \end{array}\right\} h_4 = 278.97 \text{ kJ/kg}$$

Then the rate of heat removed from the refrigerated space and the compressor work input per unit mass of refrigerant flowing through the condenser are

$$\dot{m}_B = (1 - x_6)\dot{m}_A = (1 - 0.1828)(0.25 \text{ kg/s}) = 0.2043 \text{ kg/s}$$

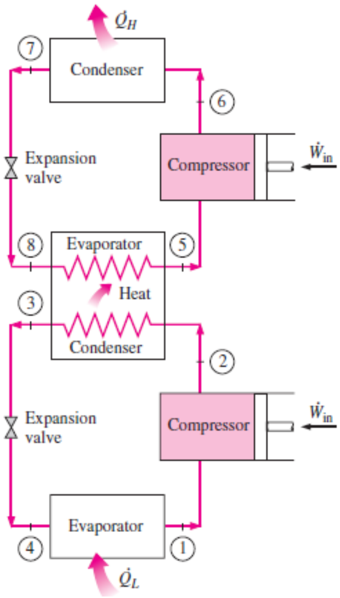
$$\dot{Q}_L = \dot{m}_B (h_1 - h_8) = (0.2043 \text{ kg/s})(239.16 - 73.33) \text{ kJ/kg} = \mathbf{33.88 \text{ kW}}$$

$$\dot{W}_{in} = \dot{W}_{compI,in} + \dot{W}_{compII,in} = \dot{m}_A (h_4 - h_9) + \dot{m}_B (h_2 - h_1)$$
$$= (0.25 \text{ kg/s})(278.97 - 264.21) \text{ kJ/kg} + (0.2043 \text{ kg/s})(265.31 - 239.16) \text{ kJ/kg}$$
$$= 9.03 \text{ kW}$$

(*c*) The coefficient of performance is determined from

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{33.88 \text{ kW}}{9.03 \text{ kW}} = \mathbf{3.75}$$

**11–47** Consider a two-stage cascade refrigeration system operating between the pressure limits of 1.2 MPa and 200 kPa with refrigerant-134a as the working fluid. Heat rejection from the lower cycle to the upper cycle takes place in an adiabatic counterflow heat exchanger where the pressure in the upper and lower cycles are 0.4 and 0.5 MPa, respectively. In both cycles, the refrigerant is a saturated liquid at the condenser exit and a saturated vapor at the compressor inlet, and the isentropic efficiency of the compressor is 80 percent. If the mass flow rate of the refrigerant through the lower cycle is 0.15 kg/s, determine (*a*) the mass flow rate of the refrigerant through the upper cycle, (*b*) the rate of heat removal from the refrigerated space, and (*c*) the COP of this refrigerator. *Answers:* (*a*) 0.212 kg/s, (*b*) 25.7 kW, (*c*) 2.68




**11-47** A two-stage cascade refrigeration cycle is considered. The mass flow rate of the refrigerant through the upper cycle, the rate of heat removal from the refrigerated space, and the COP of the refrigerator are to be determined.

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

***Analysis*** (*a*) The properties are to be obtained from the refrigerant tables (Tables A-11 through A-13):

$$h_1 = h_{g@\,200\text{ kPa}} = 244.46 \text{ kJ/kg}$$
$$s_1 = s_{g@\,200\text{ kPa}} = 0.9377 \text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 263.30 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.80 = \frac{263.30 - 244.46}{h_2 - 244.46} \longrightarrow h_2 = 268.01 \text{ kJ/kg}$$

$$h_3 = h_{f@\,500\text{ kPa}} = 73.33 \text{ kJ/kg}$$
$$h_4 = h_3 = 73.33 \text{ kJ/kg}$$

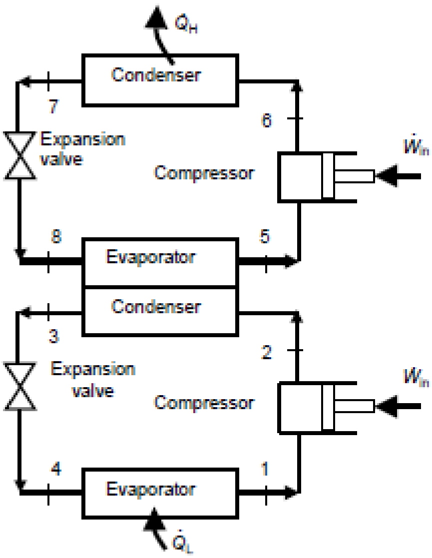
$$h_5 = h_{g@\,400\text{ kPa}} = 255.55 \text{ kJ/kg}$$
$$s_5 = s_{g@\,400\text{ kPa}} = 0.9269 \text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_6 = 1200 \text{ kPa} \\ s_6 = s_5 \end{array}\right\} h_{6s} = 278.33 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{6s} - h_5}{h_6 - h_5}$$

$$0.80 = \frac{278.33 - 255.55}{h_6 - 255.55} \longrightarrow h_6 = 284.02 \text{ kJ/kg}$$

$$h_7 = h_{f@\,1200\text{ kPa}} = 117.77 \text{ kJ/kg}$$
$$h_8 = h_7 = 117.77 \text{ kJ/kg}$$




The mass flow rate of the refrigerant through the upper cycle is determined from an energy balance on the heat exchanger

$$\dot{m}_A(h_5 - h_8) = \dot{m}_B(h_2 - h_3)$$

$$\dot{m}_A(255.55 - 117.77)\text{kJ/kg} = (0.15 \text{ kg/s})(268.01 - 73.33)\text{kJ/kg} \longrightarrow \dot{m}_A = \mathbf{0.212 \text{ kg/s}}$$

(*b*) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}_B(h_1 - h_4) = (0.15 \text{ kg/s})(244.46 - 73.33)\text{kJ/kg} = \mathbf{25.67 \text{ kW}}$$

(*c*) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1)$$
$$= (0.15 \text{ kg/s})(284.02 - 255.55)\text{kJ/kg} + (0.212 \text{ kg/s})(268.01 - 244.46)\text{kJ/kg} = 9.566 \text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{25.67}{9.566} = \mathbf{2.68}$$

**11–55** An ideal gas refrigeration cycle using air as the working fluid is to maintain a refrigerated space at −23°C while rejecting heat to the surrounding medium at 27°C. If the pressure ratio of the compressor is 3, determine (*a*) the maximum and minimum temperatures in the cycle, (*b*) the coefficient of performance, and (*c*) the rate of refrigeration for a mass flow rate of 0.08 kg/s.

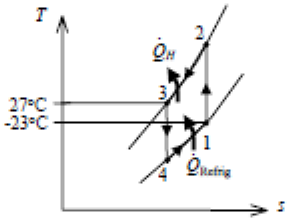
**11-55** An ideal-gas refrigeration cycle with air as the working fluid is considered. The maximum and minimum temperatures in the cycle, the COP, and the rate of refrigeration are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We assume both the turbine and the compressor to be isentropic, the turbine inlet temperature to be the temperature of the surroundings, and the compressor inlet temperature to be the temperature of the refrigerated space. From the air table (Table A-17),

$$T_1 = 250\text{ K} \longrightarrow \begin{array}{l} h_1 = 250.05\text{ kJ/kg} \\ P_{r_1} = 0.7329 \end{array}$$

$$T_1 = 300\text{ K} \longrightarrow \begin{array}{l} h_3 = 300.19\text{ kJ/kg} \\ P_{r_3} = 1.386 \end{array}$$




Thus,

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = (3)(0.7329) = 2.1987 \longrightarrow T_2 = T_{max} = \mathbf{342.2\ K}$$
$$h_2 = 342.60\text{ kJ/kg}$$

$$P_{r_4} = \frac{P_4}{P_3}P_{r_3} = \left(\frac{1}{3}\right)(1.386) = 0.462 \longrightarrow T_4 = T_{min} = \mathbf{219.0\ K}$$
$$h_4 = 218.97\text{ kJ/kg}$$

(*b*) The COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_R = \frac{q_L}{w_{net,in}} = \frac{q_L}{w_{comp,in} - w_{turb,out}}$$

where

$$q_L = h_1 - h_4 = 250.05 - 218.97 = 31.08\text{ kJ/kg}$$
$$w_{comp,in} = h_2 - h_1 = 342.60 - 250.05 = 92.55\text{ kJ/kg}$$
$$w_{turb,out} = h_3 - h_4 = 300.19 - 218.97 = 81.22\text{ kJ/kg}$$

Thus, $$\text{COP}_R = \frac{31.08}{92.55 - 81.22} = \mathbf{2.74}$$

(*c*) The rate of refrigeration is determined to be

$$\dot{Q}_{refrig} = \dot{m}(q_L) = (0.08\text{ kg/s})(31.08\text{ kJ/kg}) = \mathbf{2.49\ \ kJ/s}$$

**11–61** A gas refrigeration system using air as the working fluid has a pressure ratio of 4. Air enters the compressor at −7°C. The high-pressure air is cooled to 27°C by rejecting heat to the surroundings. It is further cooled to −15°C by regenerative cooling before it enters the turbine. Assuming both the turbine and the compressor to be isentropic and using constant specific heats at room temperature, determine (*a*) the lowest temperature that can be obtained by this cycle, (*b*) the coefficient of performance of the cycle, and (*c*) the mass flow rate of air for a refrigeration rate of 12 kW. *Answers:* (*a*) −99.4°C, (*b*) 1.12, (*c*) 0.237 kg/s

**11-61** An ideal-gas refrigeration cycle with air as the working fluid is considered. The lowest temperature that can be obtained by this cycle, the COP, and the mass flow rate of air are to be determined.

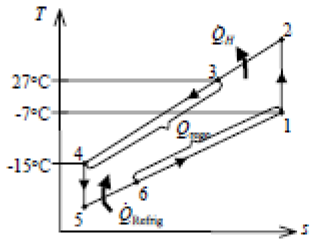
***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) The lowest temperature in the cycle occurs at the turbine exit. From the isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (266\text{ K})(4)^{0.4/1.4} = 395.3\text{ K} = 122.3^\circ\text{C}$$

$$T_5 = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = (258\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 173.6\text{ K} = \mathbf{-99.4^\circ C} = T_{\min}$$




(*b*) From an energy balance on the regenerator,

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

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

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i \longrightarrow \dot{m}(h_3 - h_4) = \dot{m}(h_1 - h_6)$$

or,

$$\dot{m}c_p(T_3 - T_4) = \dot{m}c_p(T_1 - T_6) \longrightarrow T_3 - T_4 = T_1 - T_6$$

or,

$$T_6 = T_1 - T_3 + T_4 = (-7^\circ\text{C}) - 27^\circ\text{C} + (-15^\circ\text{C}) = -49^\circ\text{C}$$

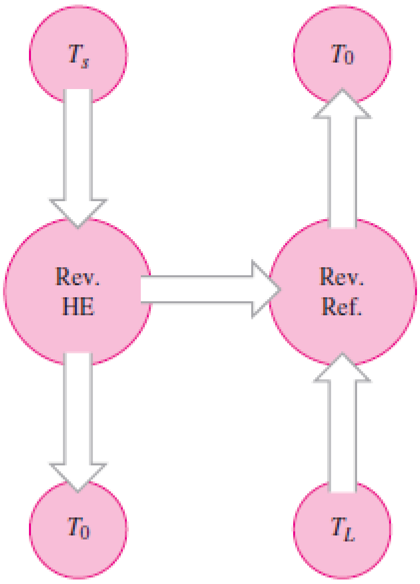
Then the COP of this ideal gas refrigeration cycle is determined from

$$\begin{aligned}\text{COP}_\text{R} &= \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}}\\ &= \frac{h_6 - h_5}{(h_2 - h_1) - (h_4 - h_5)}\\ &= \frac{T_6 - T_5}{(T_2 - T_1) - (T_4 - T_5)}\\ &= \frac{-49^\circ\text{C} - (-99.4^\circ\text{C})}{[122.3 - (-7)]^\circ\text{C} - [-15 - (-99.4)]^\circ\text{C}} = \mathbf{1.12}\end{aligned}$$

(*c*) The mass flow rate is determined from

$$\dot{m} = \frac{\dot{Q}_{\text{refrig}}}{q_L} = \frac{\dot{Q}_{\text{refrig}}}{h_6 - h_5} = \frac{\dot{Q}_{\text{refrig}}}{c_p(T_6 - T_5)} = \frac{12\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot{}^\circ\text{C})[-49 - (-99.4)]^\circ\text{C}} = \mathbf{0.237\ kg/s}$$

**11–74** A reversible absorption refrigerator consists of a reversible heat engine and a reversible refrigerator. The system removes heat from a cooled space at −10°C at a rate of 22 kW. The refrigerator operates in an environment at 25°C. If the heat is supplied to the cycle by condensing saturated steam at 200°C, determine (*a*) the rate at which the steam condenses and (*b*) the power input to the reversible refrigerator. (*c*) If the COP of an actual absorption chiller at the same temperature limits has a COP of 0.7, determine the second law efficiency of this chiller. *Answers:* (*a*) 0.00408 kg/s, (*b*) 2.93 kW, (*c*) 0.252

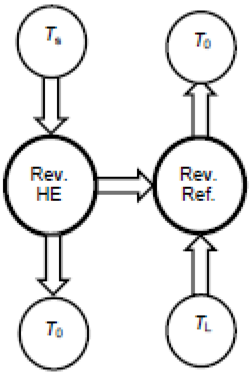


**11-74** A reversible absorption refrigerator consists of a reversible heat engine and a reversible refrigerator. The rate at which the steam condenses, the power input to the reversible refrigerator, and the second law efficiency of an actual chiller are to be determined.

***Properties*** The enthalpy of vaporization of water at 200C is $h_{fg}$ = 1939.8 kJ/kg (Table A-4).

***Analysis*** (*a*) The thermal efficiency of the reversible heat engine is

$$\eta_{\text{th,rev}} = 1 - \frac{T_0}{T_s} = 1 - \frac{(25+273.15)\text{ K}}{(200+273.15)\text{ K}} = 0.370$$




The COP of the reversible refrigerator is

$$\text{COP}_{\text{R,rev}} = \frac{T_L}{T_0 - T_L} = \frac{(-10+273.15)\text{ K}}{(25+273.15)-(-10+273.15)\text{ K}} = 7.52$$

The COP of the reversible absorption refrigerator is

$$\text{COP}_{\text{abs,rev}} = \eta_{\text{th,rev}}\text{COP}_{\text{R,rev}} = (0.370)(7.52) = 2.78$$

The heat input to the reversible heat engine is

$$\dot{Q}_{\text{in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{abs,rev}}} = \frac{22\text{ kW}}{2.78} = 7.911\text{ kW}$$

Then, the rate at which the steam condenses becomes

$$\dot{m}_s = \frac{\dot{Q}_{\text{in}}}{h_{fg}} = \frac{7.911\text{ kJ/s}}{1939.8\text{ kJ/kg}} = \mathbf{0.00408\ kg/s}$$

(*b*) The power input to the refrigerator is equal to the power output from the heat engine

$$\dot{W}_{\text{in,R}} = \dot{W}_{\text{out,HE}} = \eta_{\text{th,rev}}\dot{Q}_{\text{in}} = (0.370)(7.911\text{ kW}) = \mathbf{2.93\ kW}$$

(*c*) The second-law efficiency of an actual absorption chiller with a COP of 0.7 is

$$\eta_{\text{II}} = \frac{\text{COP}_{\text{actual}}}{\text{COP}_{\text{abs,rev}}} = \frac{0.7}{2.78} = \mathbf{0.252}$$