Ideal and Actual Vapor-Compression Cycles

10-4C Yes; the throttling process is an internally irreversible process.

10-5C To make the ideal vapor-compression refrigeration cycle more closely approximate the actual cycle.

10-6C No. Assuming the water is maintained at 10°C in the evaporator, the evaporator pressure will be the saturation pressure corresponding to this pressure, which is 1.2 kPa. It is not practical to design refrigeration or air-conditioning devices that involve such extremely low pressures.

10-7C Allowing a temperature difference of 10°C for effective heat transfer, the condensation temperature of the refrigerant should be 25°C. The saturation pressure corresponding to 25°C is 0.67 MPa. Therefore, the recommended pressure would be 0.7 MPa.

10-8C The area enclosed by the cyclic curve on a T-s diagram represents the net work input for the reversed Carnot cycle, but not so for the ideal vapor-compression refrigeration cycle. This is because the latter cycle involves an irreversible process for which the process path is not known.

10-9C The cycle that involves saturated liquid at 30°C will have a higher COP because, judging from the T-s diagram, it will require a smaller work input for the same refrigeration capacity.

10-10C The minimum temperature that the refrigerant can be cooled to before throttling is the temperature of the sink (the cooling medium) since heat is transferred from the refrigerant to the cooling medium.

10-11 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.

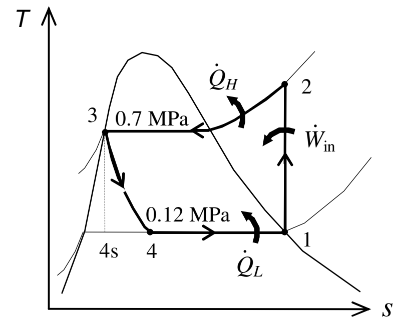
Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12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

$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 120\text{ kPa}} = 233.86 \text{ kJ/kg} \\ s_1 = s_{g\ @\ 120\text{ kPa}} = 0.9354 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 270.22 \text{ kJ/kg } (T_2 = 34.6°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7 \text{ MPa} \\ \text{sat.liquid} \end{array}\right\} h_3 = h_{f\ @\ 0.7\text{ MPa}} = 86.78 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 86.78 \text{ kJ/kg (throttling)}$$



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

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(233.86 - 86.78) \text{ kJ/kg} = \mathbf{7.35 \text{ kW}}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(270.22 - 233.86) \text{ kJ/kg} = \mathbf{1.82 \text{ kW}}$$

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

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.35 + 1.82 = \mathbf{9.17 \text{ kW}}$$

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

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.35 \text{ kW}}{1.82 \text{ kW}} = \mathbf{4.04}$$

10-19 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),

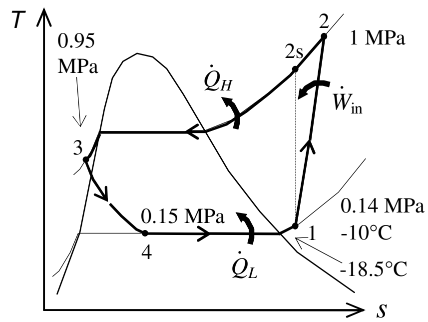
$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 243.40 \text{ kJ/kg} \\ s_1 = 0.9606 \text{ kJ/kg}\cdot\text{K} \\ v_1 = 0.14549 \text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.0 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 286.04 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.95 \text{ MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\ @\ 30\ °\text{C}} = 91.49 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 91.49 \text{ kJ/kg (throttling)}$$

$$\left.\begin{array}{l} T_5 = -18.5°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} P_5 = 0.14187 \text{ MPa} \\ h_5 = 236.23 \text{ kJ/kg} \end{array}$$



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

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.3/60 \text{ m}^3/\text{s}}{0.14549 \text{ m}^3/\text{kg}} = 0.0344 \text{ kg/s}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_{2s} - h_1)/\eta_C = (0.0344 \text{ kg/s})[(286.04 - 243.40) \text{ kJ/kg}]/(0.78) = \mathbf{1.88\ kW}$$

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

$$\dot{Q}_L = \dot{m}(h_5 - h_4) = (0.0344 \text{ kg/s})(236.23 - 91.49) \text{ kJ/kg} = \mathbf{4.98\ kW}$$

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

$$\Delta P = P_5 - P_1 = 141.87 - 140 = \mathbf{1.87}$$

and

$$\dot{Q}_{\text{gain}} = \dot{m}(h_1 - h_5) = (0.0344 \text{ kg/s})(243.40 - 236.23) \text{ kJ/kg} = \mathbf{0.247\ kW}$$

10-20 EES solution of this (and other comprehensive problems designated with the *computer icon*) is available to instructors at the *Instructor Manual* section of the *Online Learning Center* (OLC) at www.mhhe.com/cengel-boles. See the Preface for access information.
