Solucionario Guia 4

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1. (Cengel, 6ª ed)

5-57 Argon gas expands in a turbine. The exit temperature of the argon for a power output of 250 kW is to be determined.

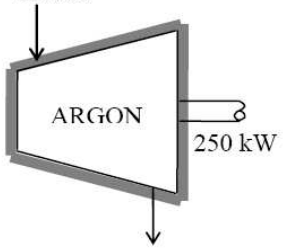
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

Properties The gas constant of Ar is $R = 0.2081$ kPa.m^3/kg.K. The constant pressure specific heat of Ar is $c_p = 0.5203$ kJ/kg·°C (Table A-2a)

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of argon and its mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.2081\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(723\ \text{K})}{900\ \text{kPa}} = 0.167\ \text{m}^3/\text{kg}$$

$A_1 = 60\ \text{cm}^2$
$P_1 = 900\ \text{kPa}$
$T_1 = 450°\text{C}$
$V_1 = 80\ \text{m/s}$



$P_2 = 150\ \text{kPa}$
$V_2 = 150\ \text{m/s}$

Thus,

$$\dot{m} = \frac{1}{v_1}A_1V_1 = \frac{1}{0.167\text{m}^3/\text{kg}}(0.006\ \text{m}^2)(80\ \text{m/s}) = 2.874\ \text{kg/s}$$

We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting,

$$250\ \text{kJ/s} = -(2.874\ \text{kg/s})\left[(0.5203\ \text{kJ/kg}\cdot^\circ\text{C})(T_2 - 450^\circ\text{C}) + \frac{(150\ \text{m/s})^2 - (80\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

It yields

$$T_2 = \mathbf{267.3°C}$$

2. (Cengel, 5ª ed)

5-60 Helium is compressed by a compressor. For a mass flow rate of 90 kg/min, the power input required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of helium is c_p = 5.1926 kJ/kg·K (Table A-2a).

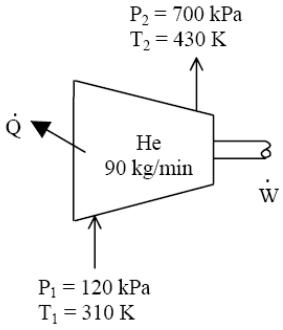
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$



Thus,

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}} + \dot{m}c_p(T_2 - T_1)$$

$$= (90/6\,0 \text{ kg/s})(20 \text{ kJ/kg}) + (90/60 \text{ kg/s})(5.1926 \text{ kJ/kg} \cdot \text{K})(430 - 310)\text{K}$$

$$= \mathbf{965\ kW}$$

3. (Cengel, 5ª ed)

5-77 Feedwater is heated in a chamber by mixing it with superheated steam. If the mixture is saturated liquid, the ratio of the mass flow rates of the feedwater and the superheated vapor is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties Noting that $T < T_{sat\ @\ 1\ MPa} = 179.88°C$, the cold water stream and the mixture exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus, from steam tables (Tables A-4 through A-6)

$$h_1 \cong h_{f@\,50°C} = 209.34\ \text{kJ/kg}$$
$$h_3 \cong h_{f@\,1\ MPa} = 762.51\ \text{kJ/kg}$$

and

$$\left.\begin{matrix} P_2 = 1\,\text{MPa} \\ T_2 = 200°\text{C} \end{matrix}\right\} h_2 = 2828.3\ \text{kJ/kg}$$

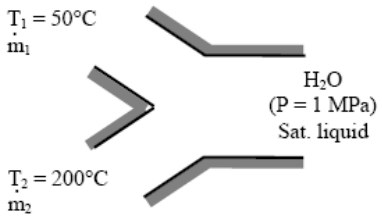
Analysis We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $$\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0\ (steady)} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\not\rightarrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$



Combining the two, $$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

Dividing by $\dot{m}_2$ yields $$y h_1 + h_2 = (y + 1) h_3$$

Solving for y: $$y = \frac{h_3 - h_2}{h_1 - h_3}$$

where $y = \dot{m}_1 / \dot{m}_2$ is the desired mass flow rate ratio. Substituting,

$$y = \frac{762.51 - 2828.3}{209.34 - 762.51} = \mathbf{3.73}$$

4. (Cengel, 5ª ed)

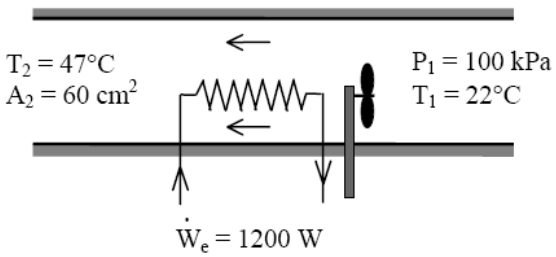
5-114 A hair dryer consumes 1200 W of electric power when running. The inlet volume flow rate and the exit velocity of air are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The power consumed by the fan and the heat losses are negligible.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·K (Table A-2)

Analysis We take the *hair dryer* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



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

$$\dot{W}_{\text{e,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{\text{out}} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Substituting, the mass and volume flow rates of air are determined to be

$$\dot{m} = \frac{\dot{W}_{\text{e,in}}}{c_p(T_2 - T_1)} = \frac{1.2 \text{ kJ/s}}{(1.005 \text{ kJ/kg}\cdot^\circ\text{C})(47 - 22)^\circ\text{C}} = 0.04776 \text{ kg/s}$$

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})}{(100 \text{ kPa})} = 0.8467 \text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}\nu_1 = (0.04776 \text{ kg/s})(0.8467 \text{ m}^3/\text{kg}) = \mathbf{0.0404 \text{ m}^3/\text{s}}$$

(*b*) The exit velocity of air is determined from the mass balance $\dot{m}_1 = \dot{m}_2 = \dot{m}$ to be

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(320\text{K})}{(100 \text{ kPa})} = 0.9184 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 \longrightarrow V_2 = \frac{\dot{m}\nu_2}{A_2} = \frac{(0.04776 \text{ kg/s})(0.9184 \text{ m}^3/\text{kg})}{60\times10^{-4} \text{ m}^2} = \mathbf{7.31 \text{ m/s}}$$

5.

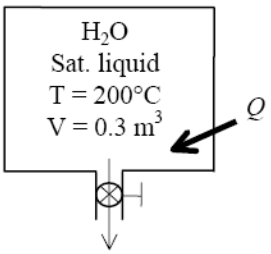
5-133 A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of the mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer is to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 200^\circ\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@200^\circ\text{C}} = 0.001157\ \text{m}^3/\text{kg} \\ u_1 = u_{f@200^\circ\text{C}} = 850.46\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_e = 200^\circ\text{C} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@200^\circ\text{C}} = 852.26\ \text{kJ/kg}$$



Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \ \ (\text{since } W \cong ke \cong pe \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.3\ \text{m}^3}{0.001157\text{m}^3/\text{kg}} = 259.4\ \text{kg}$$

$$m_2 = \tfrac{1}{2} m_1 = \tfrac{1}{2}(259.4\ \text{kg}) = 129.7\ \text{kg}$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 259.4 - 129.7 = 129.7\ \text{kg}$$

Now we determine the final internal energy,

$$v_2 = \frac{V}{m_2} = \frac{0.3\ \text{m}^3}{129.7\ \text{kg}} = 0.002313\ \text{m}^3/\text{kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002313 - 0.001157}{0.12721 - 0.001157} = 0.009171$$

$$\left.\begin{array}{l} T_2 = 200^\circ\text{C} \\ x_2 = 0.009171 \end{array}\right\} u_2 = u_f + x_2 u_{fg} = 850.46 + (0.009171)(1743.7) = 866.46\ \text{kJ/kg}$$

Then the heat transfer during this process is determined from the energy balance by substitution to be

$$Q = (129.7\ \text{kg})(852.26\ \text{kJ/kg}) + (129.7\ \text{kg})(866.46\ \text{kJ/kg}) - (259.4\ \text{kg})(850.46\ \text{kJ/kg})$$
$$= \mathbf{2308\ kJ}$$

6.

5-184 An adiabatic air compressor is powered by a direct-coupled steam turbine, which is also driving a generator. The net power delivered to the generator is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The devices are adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} h_3 = 3343.6 \text{ kJ/kg}$$

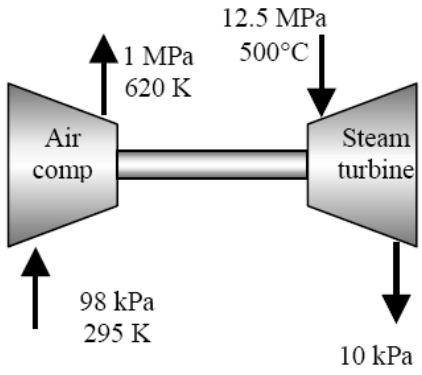
and

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg}$$

From the air table (Table A-17),

$$T_1 = 295 \text{ K} \longrightarrow h_1 = 295.17 \text{ kJ/kg}$$

$$T_2 = 620 \text{ K} \longrightarrow h_2 = 628.07 \text{ kJ/kg}$$



Analysis There is only one inlet and one exit for either device, and thus $\dot{m}_{\text{in}} = \dot{m}_{\text{out}} = \dot{m}$. We take either the turbine or the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for either steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

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

For the turbine and the compressor it becomes

Compressor: $\dot{W}_{\text{comp, in}} + \dot{m}_{\text{air}} h_1 = \dot{m}_{\text{air}} h_2 \quad \rightarrow \quad \dot{W}_{\text{comp, in}} = \dot{m}_{\text{air}} (h_2 - h_1)$

Turbine: $\dot{m}_{\text{steam}} h_3 = \dot{W}_{\text{turb, out}} + \dot{m}_{\text{steam}} h_4 \quad \rightarrow \quad \dot{W}_{\text{turb, out}} = \dot{m}_{\text{steam}} (h_3 - h_4)$

Substituting,

$$\dot{W}_{\text{comp,in}} = (10 \text{ kg/s})(628.07 - 295.17) \text{ kJ/kg} = 3329 \text{ kW}$$

$$\dot{W}_{\text{turb,out}} = (25 \text{ kg/s})(3343.6 - 2392.5) \text{ kJ/kg} = 23{,}777 \text{ kW}$$

Therefore,

$$\dot{W}_{\text{net,out}} = \dot{W}_{\text{turb,out}} - \dot{W}_{\text{comp,in}} = 23{,}777 - 3329 = \mathbf{20{,}448 \text{ kW}}$$

7.

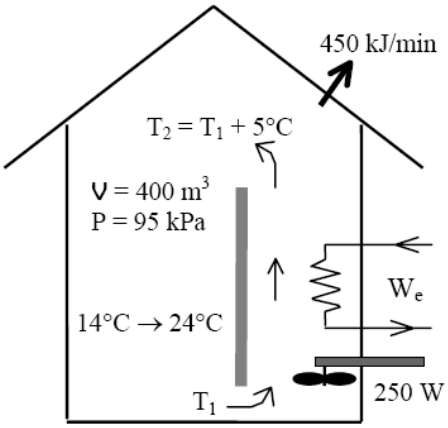
5-193 A building is to be heated by a 30-kW electric resistance heater placed in a duct inside. The time it takes to raise the interior temperature from 14°C to 24°C, and the average mass flow rate of air as it passes through the heater in the duct are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out of the building.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heats of air at room temperature are $c_p = 1.005$ and $c_v = 0.718$ kJ/kg·K (Table A-2).

Analysis (*a*) The total mass of air in the building is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95\ \text{kPa})(400\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(287\ \text{K})} = 461.3\ \text{kg}.$$



We first take the *entire building* as our system, which is a closed system since no mass leaks in or out. The time required to raise the air temperature to 24°C is determined by applying the energy balance to this constant volume closed system:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} + W_{\text{fan,in}} - Q_{\text{out}} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot{W}_{e,\text{in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}}\right) = mc_{v,\text{avg}}(T_2 - T_1)$$

Solving for Δt gives

$$\Delta t = \frac{mc_{v,\text{avg}}(T_2 - T_1)}{\dot{W}_{e,\text{in}} + \dot{W}_{\text{fan,in}} - Q_{\text{out}}} = \frac{(461.3\ \text{kg})(0.718\ \text{kJ/kg}\cdot^\circ\text{C})(24-14)^\circ\text{C}}{(30\ \text{kJ/s}) + (0.25\ \text{kJ/s}) - (450/60\ \text{kJ/s})} = \mathbf{146\ s}$$

(*b*) We now take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this adiabatic steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{W}_{e,\text{in}} + \dot{W}_{\text{fan,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{e,\text{in}} + \dot{W}_{\text{fan,in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Thus,

$$\dot{m} = \frac{\dot{W}_{e,\text{in}} + \dot{W}_{\text{fan,in}}}{c_p \Delta T} = \frac{(30 + 0.25)\ \text{kJ/s}}{(1.005\ \text{kJ/kg}\cdot^\circ\text{C})(5^\circ\text{C})} = \mathbf{6.02\ kg/s}$$

8.

5-196 Air is allowed to leave a piston-cylinder device with a pair of stops. Heat is lost from the cylinder. The amount of mass that has escaped and the work done are to be determined.

Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device is assumed to be constant. **2** Kinetic and potential energies are negligible. 3 Air is an ideal gas with constant specific heats at the average temperature.

Properties The properties of air are $R = 0.287$ kPa.m^3/kg.K (Table A-1), $c_v = 0.733$ kJ/kg.K, $c_p = 1.020$ kJ/kg.K at the anticipated average temperature of 450 K (Table A-2b).

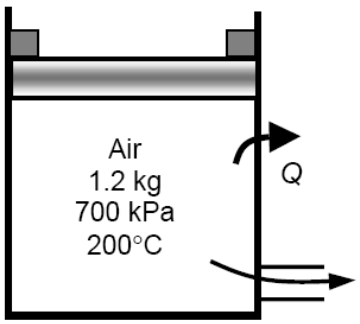
Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $$m_{in} - m_{out} = \Delta m_{\text{system}} \quad \rightarrow \quad m_e = m_1 - m_2$$

Energy balance: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} - m_e h_e = m_2 u_2 - m_1 u_1 \ \ (\text{since ke} \cong \text{pe} \cong 0)$$

or $$W_{\text{b,in}} - Q_{\text{out}} - m_e C_p T_e = m_2 c_v T_2 - m_1 c_v T_1$$



The temperature of the air withdrawn from the cylinder is assumed to be the average of initial and final temperatures of the air in the cylinder. That is,

$$T_e = (1/2)(T_1 + T_2) = (1/2)(473 + T_2)$$

The volumes and the masses at the initial and final states and the mass that has escaped from the cylinder are given by

$$V_1 = \frac{m_1 R T_1}{P_1} = \frac{(1.2\ \text{kg})(0.287\ \text{kPa.m}^3/\text{kg.K})(200 + 273\ \text{K})}{(700\ \text{kPa})} = 0.2327\ \text{m}^3$$

$$V_2 = 0.80 V_1 = (0.80)(0.2327) = 0.1862\ \text{m}^3$$

$$m_2 = \frac{P_2 V_2}{R T_2} = \frac{(600\ \text{kPa})(0.1862\ \text{m}^3)}{(0.287\ \text{kPa.m}^3/\text{kg.K}) T_2} = \frac{389.18}{T_2}\ \text{kg}$$

$$m_e = m_1 - m_2 = \left(1.2 - \frac{389.18}{T_2}\right)\text{kg}$$

Noting that the pressure remains constant after the piston starts moving, the boundary work is determined from

$$W_{\text{b,in}} = P_2 (V_1 - V_2) = (600\ \text{kPa})(0.2327 - 0.1862)\text{m}^3 = \mathbf{27.9\ kJ}$$

Substituting,

$$27.9\ \text{kJ} - 40\ \text{kJ} - \left(1.2 - \frac{389.18}{T_2}\right)(1.020\ \text{kJ/kg.K})(1/2)(473 + T_2)$$
$$= \left(\frac{389.18}{T_2}\right)(0.733\ \text{kJ/kg.K}) T_2 - (1.2\ \text{kg})(0.733\ \text{kJ/kg.K})(473\ \text{K})$$

The final temperature may be obtained from this equation by a trial-error approach or using EES to be

$$T_2 = \mathbf{415.0\ K}$$

Then, the amount of mass that has escaped becomes

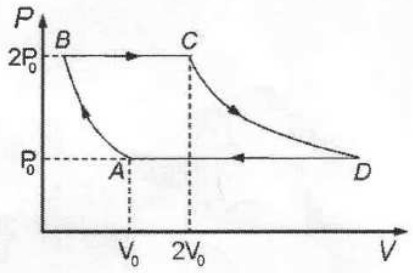
$$m_e = 1.2 - \frac{389.18}{415.0\ \text{K}} = \mathbf{0.262\ kg}$$

9. --

10. Una muy parecida (I1-Ondas, Calor y Óptica,)

Problema 3

El ciclo termodinámico representado por el diagrama $P - V$ de la figura lo realiza un mol de un gas ideal diatómico. El tramo $A \to B$ es isotérmico, el tramo $B \to C$ es isobárico, el tramo $C \to D$ es adiabático, y el tramo $D \to A$ es isobárico. Los valores P_0 y V_0 en el gráfico se suponen conocidos.



a) Encuentre los valores de presión, volumen y temperatura en los puntos A, B, C y D del diagrama.

b) Determine el flujo de calor en cada tramo del ciclo.

c) Determine la eficiencia termodinámica del ciclo.

d) Determine la eficiencia termodinámica del ciclo de Carnot equivalente.

e) Encuentre el trabajo total realizado por el gas en el ciclo.

f) Encuentre el cambio de entropía del gas en cada tramo del ciclo.

Exprese sus resultados en función de los valores P_0 y V_0 conocidos y R, la constante de los gases ideales.

Solución

a) La Ecuacuón de Estado de los gases ideales es

$$PV = nRT \tag{9}$$

En este caso $n = 1$. En el Punto A se tiene $P_A = P_0$ y $V_A = V_0$, por lo que usando la Ec. 9, resulta

$$T_A = \frac{P_A V_A}{R} = \frac{P_0 V_0}{R}$$

El tramo $A \to B$ es isotérmico, por lo que $T_B = T_A = P_0 V_0 / R$ y $P_B = 2P_0$. Usando la Ec. 9, se tiene

$$V_B = \frac{RT_B}{P_B} = \frac{P_0 V_0}{2P_0} = \frac{V_0}{2}$$

El tramo $B \to C$ es isob—'arico, por lo que $P_C = P_B = 2P_0$ y $V_C = 2V_0$. Usando la Ec. 9, se tiene

$$T_C = \frac{P_C V_C}{R} = \frac{4P_0 V_0}{R}$$

El tramo $C \rightarrow D$ es adiabático, por lo que la presión y volumen satisfacen la ecuación

$$PV^\gamma = cte \tag{10}$$

Como el gas es diatómico, γ=7/5=1.4.

Evaluando la constante en el punto C, see tiene

$$PV^\gamma = 2P_0 (2V_0)^\gamma \tag{11}$$

Como $P_D = P_0$, usando la Ec. 11 resulta

$$V_D^\gamma = \frac{2P_0 (2V_0)^\gamma}{P_0}$$

de donde resulta

$$V_D = 2^{(\gamma+1)/\gamma} V_0 = 2^{12/7} V_0$$

Usando la Ec. 9, se tiene

$$T_D = \frac{P_D V_D}{R} = 2^{12/7} \frac{P_0 V_0}{R}$$

b) La Primera Ley de la Termodinámica establece que el cambio en la energía interna dU se relaciona con el flujo de calor dQ y el trabajo realizado dW mediante la ecuación

$$dU = dQ - dW \tag{12}$$

El cambio en la energía interna depende del cambio en la temperatura a través de la relación

$$dU = nc_V dT \tag{13}$$

donde c_V es el calor específico a volumen constante.

El tramo $A \rightarrow B$ es isotérmico, a temperatura constante T_A, por lo que se cumple $dT = 0$ y $dU = 0$. Luego, usando la Ec. 12,

$$dQ = dW = PdV$$

Usando la Ecuación de Estado, Ec. 9,

$$PdV = RT_A \frac{dV}{V} = P_0 V_0 \frac{dV}{V}$$

Reemplazando en la integral,

$$\Delta Q_{AB} = \int_A^B dQ = P_0 V_0 \int_{V_0}^{V_0/2} \frac{dV}{V} = P_0 V_0 ln\left(\frac{1}{2}\right) = -ln(2) P_0 V_0$$

$\Delta Q_{AB} < 0$ indica que en el tramo el gas cede calor.

El tramo $B \to C$ es isobárico. Usando las Ecs. 12 y 13, con $c_V = (5/2)R$,

$$\Delta Q_{BC} = \int_B^C dQ = c_V \int_B^C dT + 2P_0 \int_B^C dV = \frac{5}{2}R\left(T_C - T_B\right) + 2P_0\left(V_C - V_B\right)$$

Reemplazando temperaturas y volúmenes, se obtiene

$$\Delta Q_{BC} = \frac{5}{2}R\left(\frac{4P_0V_0}{R} - \frac{P_0V_0}{R}\right) + 2P_0\left(2V_0 - \frac{V_0}{2}\right) = \frac{21}{2}P_0V_0$$

$\Delta Q_{BC} > 0$ indica que en el tramo el gas absorbe calor.

El tramo $C \to D$ es adiabático, por lo que $\Delta Q_{CD} = 0$.

El tramo $D \to A$ es isobárico. Usando las Ecs. 12 y 13,

$$\Delta Q_{DA} = \int_D^A dQ = c_V \int_D^A dT + P_0 \int_D^A dV = \frac{5}{2}R\left(T_A - T_D\right) + P_0\left(V_D - V_A\right)$$

Reemplazando temperaturas y volúmenes, se obtiene

$$\Delta Q_{DA} = \frac{5}{2}R\left(\frac{P_0V_0}{R} - \frac{2^{12/7}P_0V_0}{R}\right) + P_0\left(2^{12/7}V_0 - V_0\right) = \frac{3}{2}\left(1 - 2^{12/7}\right)P_0V_0$$

$\Delta Q_{DA} < 0$ indica que en el tramo el gas cede calor.

c) La eficiencia termodinámica está dada por la expresión

$$\eta = 1 - \frac{\mid Q_{out} \mid}{Q_{in}} \tag{14}$$

En este caso

$$Q_{in} = \Delta Q_{BC} = \frac{21}{2}P_0V_0$$

y

$$\mid Q_{out} \mid = \mid \Delta Q_{AB} + \Delta Q_{DA} \mid = ln(2)P_0V_0 + \frac{3}{2}\left(2^{12/7} - 1\right)P_0V_0 = \left[ln(2) + \frac{3}{2}\left(2^{12/7} - 1\right)\right]P_0V_0$$

Reemplazando en la Ec. 14,

$$\eta = 1 - \frac{ln(2) + (3/2)\left(2^{12/7} - 1\right)}{21/2} = 1 - \frac{2ln(2) + 3\left(2^{12/7} - 1\right)}{21} = 0.608$$

NOTA: el número final no es necesario.

d) La eficiencia del ciclo de Carnot equivalente está dada por la expresión

$$\eta_C = 1 - \frac{T_{min}}{T_{max}}$$

donde T_{min} y T_{max} son las temperaturas extremas de operación del ciclo. En este caso, $T_{min} = T_A = P_0V_0/R$ y $T_{max} = T_C = 4P_0V_0/R$. Esto porque $4 = 2^2 = 2^{14/7} > 2^{12/7}$. Así,

$$\eta_C = 1 - \frac{1}{4} = 0.75 > \eta$$

como era de esperar.

e) El trabajo realizado por el gas en un ciclo se puede obtener usando la expresión

$$W = \eta Q_{in}$$

así,

$$W = \left[1 - \frac{2ln(2) + 3\left(2^{12/7} - 1\right)}{21}\right] \frac{21}{2} P_0V_0$$

f) La entropía se define mediante la relación

$$dS = \frac{dQ}{T} \tag{15}$$

En el tramo $A \rightarrow B$ la temperatura es constante, luego

$$\Delta S_{AB} = \int_A^B \frac{dQ}{T} = \frac{\Delta QAB}{T_A} = -\frac{ln(2)P_0V_0}{P_0V_0/R} = -Rln(2)$$

En el tramo $B \rightarrow C$ la presión es constante,

$$\Delta S_{BC} = \int_B^C \frac{dQ}{T} = \int_B^C \frac{dU + dW}{T} = \frac{5}{2}R\int_B^C \frac{dT}{T} + R\int_B^C \frac{dT}{T}$$

Integrando,

$$\Delta S_{BC} = \frac{7}{2}Rln\left(\frac{T_C}{T_B}\right) = \frac{7}{2}Rln\left(\frac{4P_0V_0/R}{P_0V_0/R}\right) = 7Rln(2)$$

El tramo $C \rightarrow D$ es adiabático, luego $\Delta Q_{CD} = 0$ y $\Delta S_{CD} = 0$.

En el tramo $D \rightarrow A$ la presión es constante,

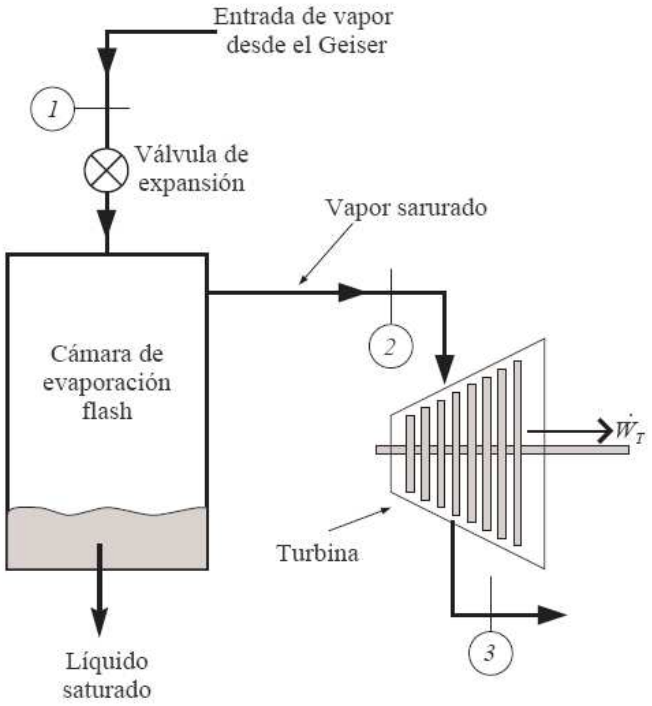
$$\Delta S_{DA} = \int_D^A \frac{dQ}{T} = \int_D^A \frac{dU + dW}{T} = \frac{5}{2}R\int_D^A \frac{dT}{T} + R\int_D^A \frac{dT}{T}$$

Integrando,

$$\Delta S_{DA} = \frac{7}{2}Rln\left(\frac{T_A}{T_D}\right) = \frac{7}{2}Rln\left(\frac{P_0V_0/R}{2^{12/7}P_0V_0/R}\right) = -6Rln(2)$$

Notar que $\Delta S_T = \Delta S_{AB} + \Delta S_{BC} + \Delta S_{CD} + \Delta S_{DA} = 0$, como era de esperar para un ciclo cerrado.

11. (I2, 2010-1)



En la entrada de la valvula, (etiquetada con el número 1) el estado termodinámico del agua está dado por la presión $P_1 = 2.0MPa$ y la temperatura $T_1 = 180^oC$. Usando las tablas de agua comprimida, encontramos que

$$h_1 = 763.71KJ/Kg.$$

Esta entalpía se conserva a través de la válvula de expansión, de donde sabemos que dentro de la cámara de evaporación,

$$h_c = h_1 = 763.71KJ/Kg.$$

La presión dentro de la cámara la conocemos, $P_2 = 400KPa$, y sabemos además que el agua está en un estado de mezcla saturada. Usamos las tablas de vapor para encontrar

que a esta presión $h_f(400KPa) = 604.73KJ/Kg$, $h_g(400KPa) = 2738.53KJ/Kg$, y por lo tanto, la calidad de la mezcla en la cámara es,

$$x_c = \frac{h_c - h_f(400KPa)}{h_g(400KPa) - h_f(400KPa)} = 0.0745.$$

La calidad mide la proporción de la masa de la mezcla que está en estado gaseoso, y por lo tanto, a la entrada de la turbina tenemos un flujo $x_c\dot{m}$ de vapor, en que $\dot{m}$ es el flujo entrante de agua desde el Geiser. Este es vapor saturado a $P_2 = 400KPa$, con $x_2 = 1$, ya que el agua fue descartada. Así, el estado termodinámico del gas que entra a la turbina lo conocemos completamente. En particular, su entalpía es,

$$h_2 = h_g(400KPa) = 2738.53KJ/Kg.$$

Utilizando ahora la primera ley en el volumen de control definido por la turbina encontramos que,

$$0 = x_c\dot{m}(h_2 - h_3) - \dot{W},$$

en que las entalpías están etiquetadas con los números de la figura y $\dot{W} = 1MW$ es el trabajo que realiza la turbina. El gas que sale de la turbina está saturado a $P = 10KPa$ y $x_3 = 0.9$. De este modo, de las tablas,

$$h_3 = h_f(10KPa) + x_3\left(h_g(10KPa) - h_f(10KPa)\right) = 2345.35KJ/Kg.$$

Así, de la primera ley,

$$\dot{m} = \frac{\dot{W}}{x_c(h_2 - h_3)} = 34Kg/seg,$$

lo que equivale a unas 120 toneladas por hora de agua.

12.

6-80 A geothermal power plant uses geothermal liquid water at 160°C at a specified rate as the heat source. The actual and maximum possible thermal efficiencies and the rate of heat rejected from this power plant are to be determined.

Assumptions **1** The power plant operates steadily. **2** The kinetic and potential energy changes are zero. **3** Steam properties are used for geothermal water.

Properties Using saturated liquid properties, the source and the sink state enthalpies of geothermal water are (Table A-4)

$$\left.\begin{aligned} T_{\text{source}} &= 160°\text{C} \\ x_{\text{source}} &= 0 \end{aligned}\right\} h_{\text{source}} = 675.47 \text{ kJ/kg}$$

$$\left.\begin{aligned} T_{\text{sink}} &= 25°\text{C} \\ x_{\text{sink}} &= 0 \end{aligned}\right\} h_{\text{sink}} = 104.83 \text{ kJ/kg}$$

Analysis (*a*) The rate of heat input to the plant may be taken as the enthalpy difference between the source and the sink for the power plant

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{geo}}(h_{\text{source}} - h_{\text{sink}}) = (440 \text{ kg/s})(675.47 - 104.83) \text{ kJ/kg} = 251{,}083 \text{ kW}$$

The actual thermal efficiency is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_{\text{in}}} = \frac{22 \text{ MW}}{251.083 \text{ MW}} = \mathbf{0.0876 = 8.8\%}$$

(*b*) The maximum thermal efficiency is the thermal efficiency of a reversible heat engine operating between the source and sink temperatures

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(25 + 273) \text{ K}}{(160 + 273) \text{ K}} = \mathbf{0.312 = 31.2\%}$$

(*c*) Finally, the rate of heat rejection is

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net,out}} = 251.1 - 22 = \mathbf{229.1 MW}$$

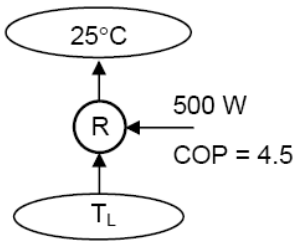
13.

6-90 The refrigerated space temperature, the COP, and the power input of a Carnot refrigerator are given. The rate of heat removal from the refrigerated space and its temperature are to be determined.

Assumptions The refrigerator operates steadily.

Analysis (*a*) The rate of heat removal from the refrigerated space is determined from the definition of the COP of a refrigerator,

$$\dot{Q}_L = \text{COP}_{\text{R}} \dot{W}_{\text{net,in}} = (4.5)(0.5 \text{ kW}) = 2.25 \text{ kW} = \mathbf{135 \text{ kJ/min}}$$



(*b*) The temperature of the refrigerated space T_L is determined from the coefficient of performance relation for a Carnot refrigerator,

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H / T_L) - 1} \longrightarrow 4.5 = \frac{1}{(25 + 273 \text{ K})/T_L - 1}$$

It yields

$$T_{\text{L}} = 243.8 \text{ K} = \mathbf{-29.2°C}$$

14.

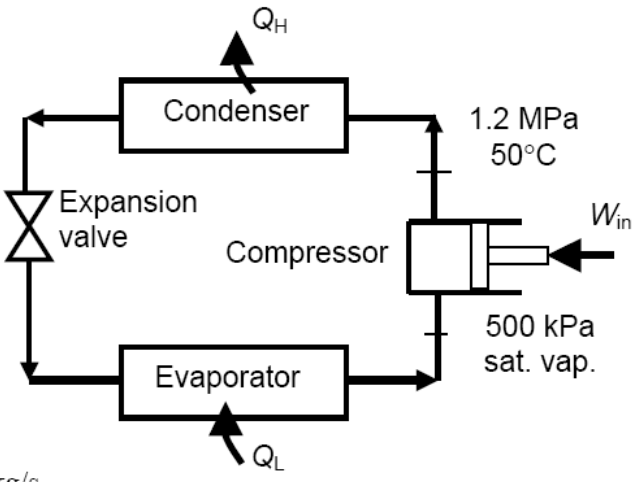
6-103 An air-conditioner with R-134a as the working fluid is considered. The compressor inlet and exit states are specified. The actual and maximum COPs and the minimum volume flow rate of the refrigerant at the compressor inlet are to be determined.

Assumptions **1** The air-conditioner operates steadily. **2** The kinetic and potential energy changes are zero.

Properties The properties of R-134a at the compressor inlet and exit states are (Tables A-11 through A-13)

$$\left.\begin{matrix} P_1 = 500\ \text{kPa} \\ x_1 = 1 \end{matrix}\right\} \begin{matrix} h_1 = 259.30\ \text{kJ/kg} \\ v_1 = 0.04112\ \text{m}^3/\text{kg} \end{matrix}$$

$$\left.\begin{matrix} P_2 = 1.2\ \text{MPa} \\ T_2 = 50°\text{C} \end{matrix}\right\} h_2 = 278.27\ \text{kJ/kg}$$



Analysis (*a*) The mass flow rate of the refrigerant and the power consumption of the compressor are

$$\dot{m}_R = \frac{\dot{V}_1}{v_1} = \frac{100\ \text{L/min}\left(\frac{1\ \text{m}^3}{1000\ \text{L}}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right)}{0.04112\ \text{m}^3/\text{kg}} = 0.04053\ \text{kg/s}$$

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) = (0.04053\ \text{kg/s})(278.27 - 259.30)\ \text{kJ/kg} = 0.7686\ \text{kW}$$

The heat gains to the room must be rejected by the air-conditioner. That is,

$$\dot{Q}_L = \dot{Q}_{\text{heat}} + \dot{Q}_{\text{equipment}} = (250\ \text{kJ/min})\left(\frac{1\ \text{min}}{60\ \text{s}}\right) + 0.9\ \text{kW} = 5.067\ \text{kW}$$

Then, the actual COP becomes

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.067\ \text{kW}}{0.7686\ \text{kW}} = \mathbf{6.59}$$

(*b*) The COP of a reversible refrigerator operating between the same temperature limits is

$$\text{COP}_{\max}\ \frac{1}{T_H / T_L - 1} = \frac{1}{(34 + 273)/(26 + 273) - 1} = \mathbf{37.4}$$

(*c*) The minimum power input to the compressor for the same refrigeration load would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\max}} = \frac{5.067\ \text{kW}}{37.38} = 0.1356\ \text{kW}$$

The minimum mass flow rate is

$$\dot{m}_{R,\min} = \frac{\dot{W}_{\text{in,min}}}{h_2 - h_1} = \frac{0.1356\ \text{kW}}{(278.27 - 259.30)\ \text{kJ/kg}} = 0.007149\ \text{kg/s}$$

Finally, the minimum volume flow rate at the compressor inlet is

$$\dot{V}_{\text{min},1} = \dot{m}_{R,\min} v_1 = (0.007149\ \text{kg/s})(0.04112\ \text{m}^3/\text{kg}) = 0.000294\ \text{m}^3/\text{s} = \mathbf{17.64\ L/min}$$

15. --

16.

(Ver cengel, 6a ed, Capítulo 6)

Sean COP_R y COP_{BC}, los coeficientes de funcionamiento del refrigerador y de la bomba de calor, respectivamente.

Sabemos que:

$COP_R = \dfrac{1}{\dfrac{Q_H}{Q_L} - 1}$, y que $COP_{BC} = \dfrac{1}{1 - \dfrac{Q_L}{Q_H}}$.

Podemos demostrar que: $COP_{BC} = COP_R + 1$.

En efecto:

$$COP_R + 1 = \frac{1}{\frac{Q_H}{Q_L} - 1} + 1 = \frac{1 + \frac{Q_H}{Q_L} - 1}{\frac{Q_H}{Q_L} - 1} = \frac{\frac{Q_H}{Q_L}}{\frac{Q_H}{Q_L} - 1} = \left(\frac{1}{\frac{Q_L}{Q_H}} \right) \cdot \left(\frac{1}{\frac{Q_H}{Q_L} - 1} \right) = \frac{1}{\frac{Q_H Q_L}{Q_L Q_H} - \frac{Q_L}{Q_H}} = \frac{1}{1 - \frac{Q_L}{Q_H}} = COP_{BC}$$

17.

7-32 A rigid tank is initially filled with a saturated mixture of R-134a. Heat is transferred to the tank from a source until the pressure inside rises to a specified value. The entropy change of the refrigerant, entropy change of the source, and the total entropy change for this process are to be determined. √

Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

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

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.4 \end{array}\right\} \begin{array}{l} u_1 = u_f + x_1 u_{fg} = 38.28 + (0.4)(186.21) = 112.76 \text{ kJ/kg} \\ s_1 = s_f + x_1 s_{fg} = 0.15457 + (0.4)(0.78316) = 0.4678 \text{ kJ/kg} \cdot \text{K} \\ v_1 = v_f + x_1 v_{fg} = 0.0007533 + (0.4)(0.099867 - 0.0007533) = 0.04040 \text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 400 \text{ kPa} \\ v_2 = v_1 \end{array}\right\} \begin{array}{l} x_2 = \dfrac{v_2 - v_f}{v_{fg}} = \dfrac{0.04040 - 0.0007907}{0.051201 - 0.0007907} = 0.7857 \\ u_2 = u_f + x_2 u_{fg} = 63.62 + (0.7857)(171.45) = 198.34 \text{ kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 0.24761 + (0.7857)(0.67929) = 0.7813 \text{ kJ/kg} \cdot \text{K} \end{array}$$

The mass of the refrigerant is

$$m = \frac{V}{v_1} = \frac{0.5 \text{ m}^3}{0.04040 \text{ m}^3/\text{kg}} = 12.38 \text{ kg}$$

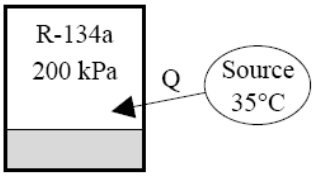
Then the entropy change of the refrigerant becomes

$$\Delta S_{\text{system}} = m(s_2 - s_1) = (12.38 \text{ kg})(0.7813 - 0.4678) \text{ kJ/kg} \cdot \text{K} = \mathbf{3.880 \text{ kJ/K}}$$

(*b*) We take the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1)$$



Substituting,

$$Q_{\text{in}} = m(u_2 - u_1) = (12.38 \text{ kg})(198.34 - 112.76) = 1059 \text{ kJ}$$

The heat transfer for the source is equal in magnitude but opposite in direction. Therefore,

$$Q_{\text{source, out}} = -Q_{\text{tank, in}} = -1059 \text{ kJ}$$

and

$$\Delta S_{\text{source}} = -\frac{Q_{\text{source,out}}}{T_{\text{source}}} = -\frac{1059 \text{ kJ}}{308 \text{ K}} = \mathbf{-3.439 \text{ kJ/K}}$$

(*c*) The total entropy change for this process is

$$\Delta S_{\text{total}} = \Delta S_{\text{system}} + \Delta S_{\text{source}} = 3.880 + (-3.439) = \mathbf{0.442 \text{ kJ/K}}$$

18.

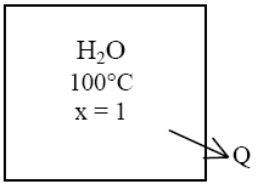
7-49 A rigid tank contains saturated water vapor at a specified temperature. Steam is cooled to ambient temperature. The process is to be sketched and entropy changes for the steam and for the process are to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible.

Analysis (b) From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} T_1 = 100°\text{C} \\ x = 1 \end{array}\right\} \begin{array}{l} v_1 = v_g = 1.6720 \text{ kJ/kg} \\ u_1 = u_g = 2506.0 \text{ kJ/kg} \\ s_1 = 7.3542 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ v_2 = v_1 \end{array}\right\} \begin{array}{l} x_2 = 0.0386 \\ u_2 = 193.78 \text{ kJ/kg} \\ s_2 = 1.0715 \text{ kJ/kg} \cdot \text{K} \end{array}$$



The entropy change of steam is determined from

$$\Delta S_w = m(s_2 - s_1) = (5 \text{ kg})(1.0715 - 7.3542)\text{kJ/kg} \cdot \text{K} = \mathbf{-31.41 \text{ kJ/K}}$$

(c) We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

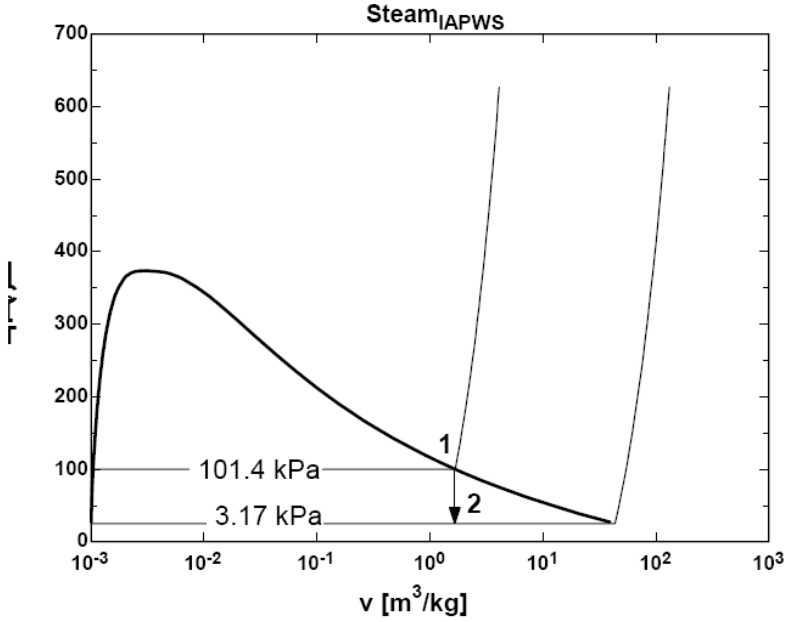
$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

That is,

$$Q_{\text{out}} = m(u_1 - u_2) = (5 \text{ kg})(2506.0 - 193.78)\text{kJ/kg} = 11{,}511 \text{ kJ}$$

The total entropy change for the process is

$$S_{\text{gen}} = \Delta S_w + \frac{Q_{\text{out}}}{T_{\text{surr}}} = -31.41 \text{ kJ/K} + \frac{11{,}511 \text{ kJ}}{298 \text{ K}} = \mathbf{7.39 \text{ kJ/K}}$$



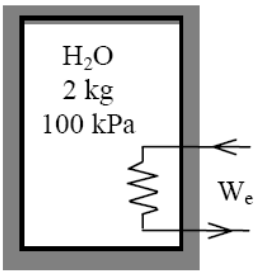
19.

7-34 An insulated rigid tank contains a saturated liquid-vapor mixture of water at a specified pressure. An electric heater inside is turned on and kept on until all the liquid vaporized. The entropy change of the water during this process is to be determined.

Analysis From the steam tables (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.001 + (0.25)(1.6941 - 0.001) = 0.4243 \text{ m}^3/\text{kg} \\ s_1 = s_f + x_1 s_{fg} = 1.3028 + (0.25)(6.0562) = 2.8168 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} v_2 = v_1 \\ \text{sat. vapor} \end{array}\right\} s_2 = 6.8649 \text{ kJ/kg} \cdot \text{K}$$



Then the entropy change of the steam becomes

$$\Delta S = m(s_2 - s_1) = (2 \text{ kg})(6.8649 - 2.8168) \text{ kJ/kg} \cdot \text{K} = \mathbf{8.10\ kJ/K}$$

20.

7-57 An iron block and a copper block are dropped into a large lake. The total amount of entropy change when both blocks cool to the lake temperature is to be determined.

Assumptions **1** Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** Kinetic and potential energies are negligible.

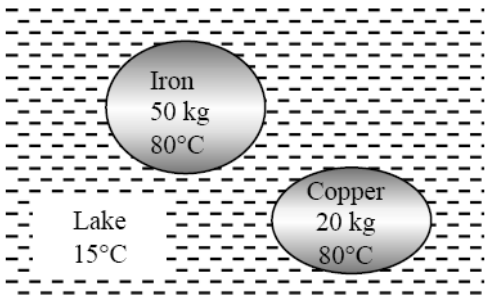
Properties The specific heats of iron and copper at room temperature are $c_{\text{iron}} = 0.45$ kJ/kg.°C and $c_{\text{copper}} = 0.386$ kJ/kg.°C (Table A-3).

Analysis The thermal-energy capacity of the lake is very large, and thus the temperatures of both the iron and the copper blocks will drop to the lake temperature (15°C) when the thermal equilibrium is established. Then the entropy changes of the blocks become

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (50\ \text{kg})(0.45\ \text{kJ/kg}\cdot\text{K})\ln\left(\frac{288\ \text{K}}{353\ \text{K}}\right) = -4.579\ \text{kJ/K}$$

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (20\ \text{kg})(0.386\ \text{kJ/kg}\cdot\text{K})\ln\left(\frac{288\ \text{K}}{353\ \text{K}}\right) = -1.571\ \text{kJ/K}$$

We take both the iron and the copper blocks, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = \Delta U_{\text{iron}} + \Delta U_{\text{copper}}$$

or,

$$Q_{\text{out}} = [mc(T_1 - T_2)]_{\text{iron}} + [mc(T_1 - T_2)]_{\text{copper}}$$

Substituting,

$$\begin{aligned} Q_{\text{out}} &= (50\ \text{kg})(0.45\ \text{kJ/kg}\cdot\text{K})(353 - 288)\text{K} + (20\ \text{kg})(0.386\ \text{kJ/kg}\cdot\text{K})(353 - 288)\text{K} \\ &= 1964\ \text{kJ} \end{aligned}$$

Thus,

$$\Delta S_{\text{lake}} = \frac{Q_{\text{lake,in}}}{T_{\text{lake}}} = \frac{1964\ \text{kJ}}{288\ \text{K}} = 6.820\ \text{kJ/K}$$

Then the total entropy change for this process is

$$\Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{copper}} + \Delta S_{\text{lake}} = -4.579 - 1.571 + 6.820 = \mathbf{0.670\ kJ/K}$$