AREN 2110 SOLUTIONS
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HOMEWORK ASSIGNMENTS 6, 7 and 8

SOLUTIONS: HOMEWORK #6

Chapter 5 Problems

5-45 A number of brass balls are to be quenched in a water bath at a specified rate. The rate at which heat needs to be removed from the water in order to keep its temperature constant is to be determined.

Assumptions **1** The thermal properties of the balls are constant. **2** The balls are at a uniform temperature before and after quenching. **3** The changes in kinetic and potential energies are negligible.

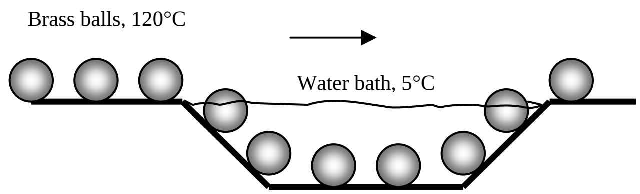
Properties The density and specific heat of the brass balls are given to be ρ = 8522 kg/m^3 and C_p = 0.385 kJ/kg.°C.

Analysis We take a single ball as the system. The energy balance for this closed system can be expressed as

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

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mC(T_1 - T_2)$$



The total amount of heat transfer from a ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (8522 \text{ kg/m}^3)\frac{\pi(0.05 \text{ m})^3}{6} = 0.558 \text{ kg}$$

$$Q_{out} = mC(T_1 - T_2) = (0.558 \text{ kg})(0.385 \text{ kJ/kg.°C})(120 - 74)\text{°C} = 9.88 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{total} = \dot{n}_{\text{ball}} Q_{\text{ball}} = (100 \text{ balls/min}) \times (9.88 \text{ kJ/ball}) = \mathbf{988 \text{ kJ/min}}$$

Therefore, heat must be removed from the water at a rate of 988 kJ/min in order to keep its temperature constant at 50°C since energy input must be equal to energy output for a system whose energy level remains constant. That is, $E_{in} = E_{out}$ when $\Delta E_{\text{system}} = 0$.

5-58C It is mostly converted to internal energy as shown by a rise in the fluid temperature.

5-59C The kinetic energy of a fluid increases at the expense of the internal energy as evidenced by a decrease in the fluid temperature.

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5-61 Air is accelerated in a nozzle from 30 m/s to 180 m/s. The mass flow rate, the exit temperature, and the exit area of the nozzle 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. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heat of air at the anticipated average temperature of 450 K is C_p = 1.02 kJ/kg.°C (Table A-2).

Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume and the mass flow rate of air are determined to be

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473\ \text{K})}{300\ \text{kPa}} = 0.4525\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_1}A_1\mathbf{V}_1 = \frac{1}{0.4525\text{m}^3/\text{kg}}(0.008\text{m}^2)(30\text{m/s}) = \mathbf{0.5304\ kg/s}$$

(*b*) We take nozzle 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

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

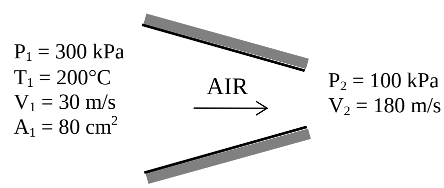
$$0 = h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} \longrightarrow 0 = C_{p,ave}\left(T_2 - T_1\right) + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}$$

Substituting,

$$0 = (1.02\ \text{kJ/kg}\cdot\text{K})(T_2 - 200^\circ\text{C}) + \frac{(180\ \text{m/s})^2 - (30\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

It yields T_2 = **184.6°C**

(*c*) The specific volume of air at the nozzle exit is



$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(184.6 + 273\ \text{K})}{100\ \text{kPa}} = 1.313\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_2}A_2\mathbf{V}_2 \longrightarrow 0.5304\text{kg/s} = \frac{1}{1.313\text{m}^3/\text{kg}}A_2\left(180\text{m/s}\right)$$

$$A_2 = 0.00387\ \text{m}^2 = \mathbf{38.7\ cm^2}$$

5-77C Yes. Because energy (in the form of shaft work) is being added to the air.

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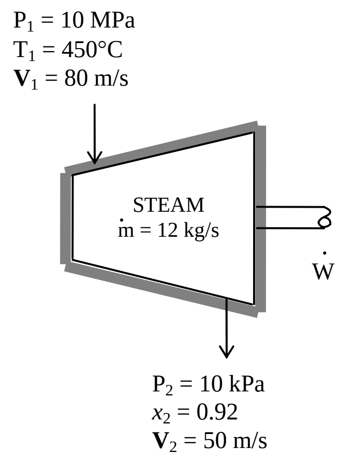
5-79 Steam expands in a turbine. The change in kinetic energy, the power output, and the turbine inlet area are to be determined.

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.

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

$$\left.\begin{array}{l} P_1 = 10\text{MPa} \\ T_1 = 450^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.02975\text{m}^3/\text{kg} \\ h_1 = 3240.9\text{kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 10\text{ kPa} \\ x_2 = 0.92 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 191.83 + 0.92 \times 2392.8 = 2393.2\text{kJ/kg}$$

Analysis (a) The change in kinetic energy is determined from

$$\Delta ke = \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2} = \frac{(50\text{m/s})^2 - (80\text{m/s})^2}{2}\left(\frac{1\text{kJ/kg}}{1000\text{m}^2/\text{s}^2}\right) = \mathbf{-1.95kJ/kg}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. 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

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

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

$$\dot{m}(h_1 + \mathbf{V}_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + \mathbf{V}_2^2/2) \quad (\text{since } \dot{\text{Q}} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{\mathbf{V}_2^2 - \mathbf{V}_1^2}{2}\right)$$

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{\text{out}} = -(12\text{ kg/s})(2393.2 - 3240.9 - 1.95)\text{kJ/kg} = \mathbf{10.2\ MW}$$

(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{v_1}A_1\mathbf{V}_1 \longrightarrow A_1 = \frac{\dot{m}v_1}{\mathbf{V}_1} = \frac{(12\text{ kg/s})(0.02975\text{ m}^3/\text{kg})}{80\text{ m/s}} = \mathbf{0.00446\ m^2}$$

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5-90 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).

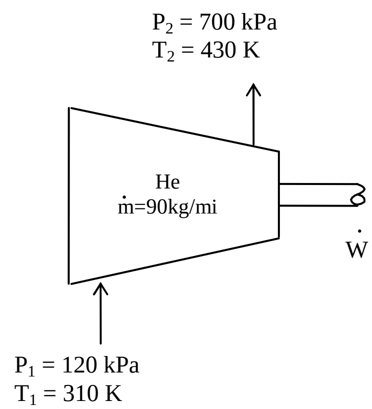
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

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

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



Thus,

$$\begin{aligned}\dot{W}_{in} &= \dot{Q}_{out} + \dot{m}C_p\left(T_2 - T_1\right)\\ &= (90/60\text{kg/s})(20\text{ kJ/kg}) + (90/60\text{kg/s})(5.1926\text{kJ/kg}\cdot\text{K})(430 - 310)\text{K}\\ &= \mathbf{965kW}\end{aligned}$$

5-104 A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There are no work interactions.

Properties Noting that T < $T_{\text{sat @ 250 kPa}}$ = 127.44°C, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus,

$$\begin{aligned} h_1 &\cong h_{f\,@\,80°\text{C}} = 334.91 \text{ kJ/kg}\\ h_2 &\cong h_{f\,@\,20°\text{C}} = 83.96 \text{ kJ/kg}\\ h_3 &\cong h_{f\,@\,42°\text{C}} = 175.92 \text{ kJ/kg}\end{aligned}$$

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

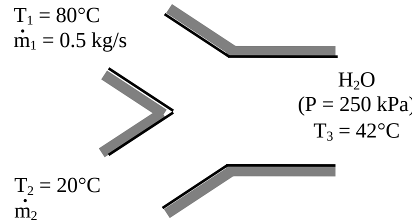
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

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

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

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



5-104 CONTINUED

Combining the two relations and solving for $\dot{m}_2$ gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2} \dot{m}_1$$

Substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{(334.91 - 175.92)\text{kJ/kg}}{(175.92 - 83.96)\text{kJ/kg}} (0.5 \text{ kg/s}) = \mathbf{0.864 \ kg/s}$$

5-106 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 < $T_{sat\ @\ 800\ kPa}$ = 170.43°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,

$$h_1 \cong h_{f\,@\,50°C} = 209.33 \text{ kJ/kg}$$
$$h_3 \cong h_{f\,@\,800\text{ kPa}} = 721.11 \text{ kJ/kg}$$

and

$$\left.\begin{array}{l} P_2 = 800\text{kPa} \\ T_2 = 200^\circ\text{C} \end{array}\right\} h_2 = 2839.3\text{kJ/kg}$$

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

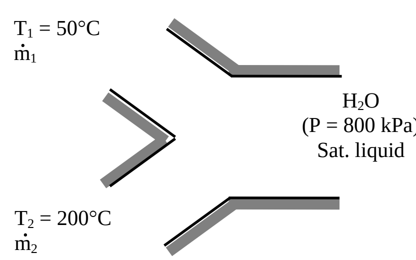
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{\text{system}} \nearrow^{0 \text{ (steady)}} = 0 \quad \rightarrow \quad \dot{m}_{in} = \dot{m}_{out} \quad \rightarrow \quad \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

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

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

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



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*: $y = \dfrac{h_3 - h_2}{h_1 - h_3}$

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

$$y = \frac{721.11 - 2839.3}{209.33 - 721.11} = \mathbf{4.14}$$

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SOLUTIONS: HOMEWORK #7

Chapter 6 Problems

6-2C Transferring 5 kWh of heat to an electric resistance wire and producing 5 kWh of electricity.

6-3C An electric resistance heater which consumes 5 kWh of electricity and supplies 6 kWh of heat to a room.

6-10C Heat engines are cyclic devices that receive heat from a source, convert some of it to work, and reject the rest to a sink.

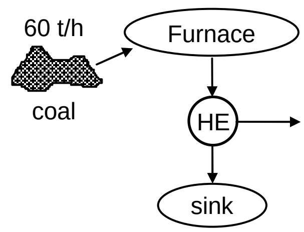
6-12C No. Because 100% of the work can be converted to heat.

6-15C No. Such an engine violates the Kelvin-Planck statement of the second law of thermodynamics.

6-20 The power output and fuel consumption rate of a power plant are given.
The overall efficiency is to be determined.

Assumptions The plant operates steadily.

Properties The heating value of coal is given to be 30,000 kJ/kg.



Analysis The rate of energy supply (in chemical form) to this power plant is

$$\dot{Q}_H = \dot{m}_{coal} u_{coal} = (60{,}000 \text{ kg/h})(30{,}000 \text{ kJ/kg}) = 1.8 \times 10^9 \text{ kJ/h} = 500 \text{ MW}$$

Then the overall efficiency of the plant becomes

$$\eta_{\text{overall}} = \frac{\dot{W}_{net,out}}{\dot{Q}_{\text{H}}} = \frac{150 \text{ MW}}{500 \text{ MW}} = 0.300 = \mathbf{30.0\%}$$

6-40C The difference between the two devices is one of purpose. The purpose of a refrigerator is to remove heat from a cold medium whereas the purpose of a heat pump is to supply heat to a warm medium.

6-47C No. The refrigerator captures energy from a cold medium and carries it to a warm medium. It does not create it.

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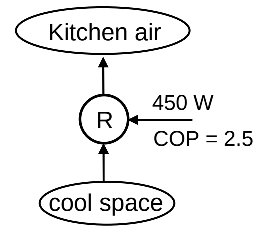
6-54 The COP and the power consumption of a refrigerator are given. The time it will take to cool 5 watermelons is to be determined.

Assumptions **1** The refrigerator operates steadily. **2** The heat gain of the refrigerator through its walls, door, etc. is negligible. **3** The watermelons are the only items in the refrigerator to be cooled.

Properties The specific heat of watermelons is given to be C = 4.2 kJ/kg.°C.

Analysis The total amount of heat that needs to be removed from the watermelons is

$$Q_L = (mC\Delta T)_{watermelons} = 5\times(10\text{kg})(4.2\text{kJ/kg}\cdot{}^\circ\text{C})(20-8)^\circ\text{C} = 2520\text{kJ}$$



The rate at which this refrigerator removes heat is

$$\dot{Q}_L = (COP_R)(\dot{W}_{net,in}) = (2.5)(0.45\text{kW}) = 1.125\text{kW}$$

That is, this refrigerator can remove 1.125 kJ of heat per second. Thus the time required to remove 2520 kJ of heat is

$$\Delta t = \frac{Q_L}{\dot{Q}_L} = \frac{2520\text{kJ}}{1.125\text{kJ/s}} = 2240\text{s} = \mathbf{37.3min}$$

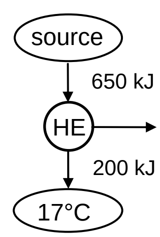
This answer is optimistic since the refrigerated space will gain some heat during this process from the surrounding air, which will increase the work load. Thus, in reality, it will take longer to cool the watermelons.

6-81 The sink temperature of a Carnot heat engine and the rates of heat supply and heat rejection are given. The source temperature and the thermal efficiency of the engine are to be determined.

Assumptions The Carnot heat engine operates steadily.

Analysis (a) For reversible cyclic devices we have $\left(\frac{Q_H}{Q_L}\right)_{rev} = \left(\frac{T_H}{T_L}\right)$

Thus the temperature of the source T_H must be



$$T_H = \left(\frac{Q_H}{Q_L}\right)_{rev} T_L = \left(\frac{650\text{ kJ}}{200\text{ kJ}}\right)(290\text{ K}) = \mathbf{942.5\ \ K}$$

(b) The thermal efficiency of a Carnot heat engine depends on the source and the sink temperatures only, and is determined from

$$\eta_{th,C} = 1 - \frac{T_L}{T_H} = 1 - \frac{290\text{ K}}{942.5\text{ K}} = 0.69 \text{ or } \mathbf{69\%}$$

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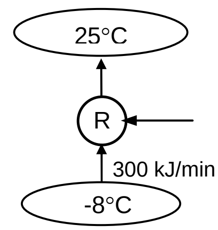
6-95 The refrigerated space and the environment temperatures for a refrigerator and the rate of heat removal from the refrigerated space are given. The minimum power input required is to be determined.

Assumptions The refrigerator operates steadily.

Analysis The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only, and is determined from

$$COP_{R,rev} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(25 + 273\text{K})/(-8 + 273\text{K}) - 1} = 8.03$$



The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

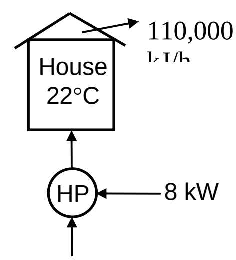
$$\dot{W}_{net,in,\min} = \frac{\dot{Q}_L}{COP_{R,\max}} = \frac{300 \text{ kJ/min}}{8.03} = 37.36 \text{ kJ/min} = \mathbf{0.623\ kW}$$

6-103 A heat pump maintains a house at a specified temperature. The rate of heat loss of the house and the power consumption of the heat pump are given. It is to be determined if this heat pump can do the job.

Assumptions The heat pump operates steadily.

Analysis The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. The coefficient of performance of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{HP,rev} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (2 + 273\text{K})/(22 + 273\text{K})} = 14.75$$



The required power input to this reversible heat pump is determined from the definition of the coefficient of performance to be

$$\dot{W}_{net,in,\min} = \frac{\dot{Q}_H}{COP_{HP}} = \frac{110{,}000\text{kJ/h}}{14.75}\left(\frac{1\text{h}}{3600\text{s}}\right) = \mathbf{2.07kW}$$

This heat pump is **powerful enough** since 8 kW > 2.07 kW.

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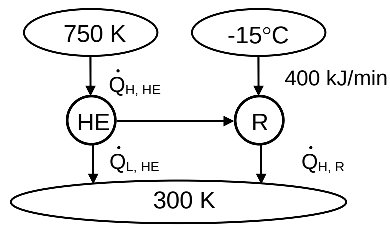
6-125 A Carnot heat engine drives a Carnot refrigerator that removes heat from a cold medium at a specified rate. The rate of heat supply to the heat engine and the total rate of heat rejection to the environment are to be determined.

Analysis (*a*) The coefficient of performance of the Carnot refrigerator is

$$COP_{R,C} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(300\text{K})/(258\text{K}) - 1} = 6.14$$

Then power input to the refrigerator becomes



$$\dot{W}_{net,\,in} = \frac{\dot{Q}_L}{COP_{R,C}} = \frac{400 \text{ kJ/min}}{6.14} = 65.1 \text{ kJ/min}$$

which is equal to the power output of the heat engine, $\dot{W}_{net,\,out}$.

The thermal efficiency of the Carnot heat engine is determined from

$$\eta_{th,C} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{750 \text{ K}} = 0.60$$

Then the rate of heat input to this heat engine is determined from the definition of thermal efficiency to be

$$\dot{Q}_{H,\,HE} = \frac{\dot{W}_{net,\,out}}{\eta_{th,\,HE}} = \frac{65.1 \text{ kJ/min}}{0.60} = \mathbf{108.5\ kJ/min}$$

(*b*) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\,HE}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,\,R}$),

$$\dot{Q}_{L,HE} = \dot{Q}_{H,HE} - \dot{W}_{net,out} = 108.5 - 65.1 = 43.4\text{kJ/min}$$
$$\dot{Q}_{H,R} = \dot{Q}_{L,R} + \dot{W}_{net,in} = 400 + 65.1 = 465.1\text{kJ/min}$$

and

$$\dot{Q}_{Ambient} = \dot{Q}_{L,\,HE} + \dot{Q}_{H,\,R} = 43.4 + 465.1 = \mathbf{508.5\ kJ/min}$$

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Chapter 7 problems

7-10C No. An isothermal process can be irreversible. Example: A system that involves paddle-wheel work while losing an equivalent amount of heat.

7-17C Increase.

7-21C Yes. This will happen when the system is losing heat, and the decrease in entropy as a result of this heat loss is equal to the increase in entropy as a result of irreversibilities.

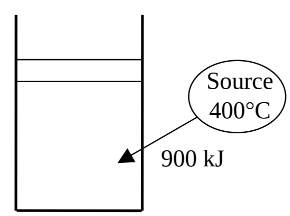
7-26 Heat is transferred isothermally from a source to the working fluid of a Carnot engine. The entropy change of the working fluid, the entropy change of the source, and the total entropy change during this process are to be determined.

Analysis (*a*) This is a reversible isothermal process, and the entropy change during such a process is given by

$$\Delta S = \frac{Q}{T}$$

Noting that heat transferred from the source is equal to the heat transferred to the working fluid, the entropy changes of the fluid and of the source become

$$\Delta S_{fluid} = \frac{Q_{fluid}}{T_{fluid}} = \frac{Q_{in,fluid}}{T_{fluid}} = \frac{900 \text{ kJ}}{673 \text{ K}} = \mathbf{1.337 \text{ kJ/K}}$$



(*b*) $$\Delta S_{source} = \frac{Q_{source}}{T_{source}} = -\frac{Q_{\text{out, source}}}{T_{source}} = -\frac{900 \text{ kJ}}{673 \text{ K}} = \mathbf{-1.337 \text{ kJ/K}}$$

(*c*) Thus the total entropy change of the process is

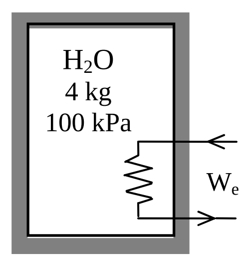
$$S_{gen} = \Delta S_{\text{total}} = \Delta S_{fluid} + \Delta S_{source} = 1.337 - 1.337 = \mathbf{0}$$

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)

$$\left.\begin{array}{l} P_1 = 100\text{kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.001 + (0.25)(1.694 - 0.001) = 0.4243\text{m}^3\text{/kg} \\ s_1 = s_f + x_1 s_{fg} = 1.3026 + (0.25)(6.0568) = 2.8168\text{kJ/kg}\cdot\text{K} \end{array}$$

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



Then the entropy change of the steam becomes

$$\Delta S = m(s_2 - s_1) = (4 \text{ kg})(6.8649 - 2.8168) \text{ kJ/kg}\cdot\text{K} = \mathbf{16.19 \text{ kJ/K}}$$

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7-51 An aluminum block is brought into contact with an iron block in an insulated enclosure. The final equilibrium temperature and the total entropy change for this process are to be determined.

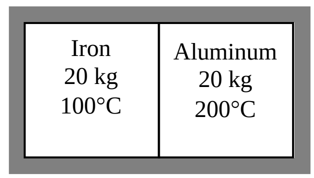
Assumptions **1** Both the aluminum and the iron block are incompressible substances with constant specific heats. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The system is well-insulated and thus there is no heat transfer.

Properties The specific heat of aluminum at the anticipated average temperature of 450 K is C_p = 0.973 kJ/kg.°C. The specific heat of iron at room temperature (the only value available in the tables) is C_p = 0.45 kJ/kg.°C (Table A-3).

Analysis We take the iron+aluminum blocks as the system, which is a closed system. The energy balance for this system can be expressed as

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

$$0 = \Delta U$$



or,

$$\Delta U_{\text{alum}} + \Delta U_{iron} = 0$$

$$[mC(T_2 - T_1)]_{\text{alum}} + [mC(T_2 - T_1)]_{iron} = 0$$

Substituting,

$$(20\text{kg})(0.45\text{kJ/kg}\cdot\text{K})(T_2 - 100^\circ\text{C}) + (20\text{kg})(0.973\text{kJ/kg}\cdot\text{K})(T_2 - 200^\circ\text{C}) = 0$$

$$T_2 = \mathbf{168.4^\circ C} = 441.4\text{K}$$

The total entropy change for this process is determined from

$$\Delta S_{iron} = mC_{ave} \ln\left(\frac{T_2}{T_1}\right) = (20\text{kg})(0.45\text{kJ/kg}\cdot\text{K})\ln\left(\frac{441.4\text{K}}{373\text{K}}\right) = 1.515\text{kJ/K}$$

$$\Delta S_{alum} = mC_{ave} \ln\left(\frac{T_2}{T_1}\right) = (20\text{kg})(0.973\text{kJ/kg}\cdot\text{K})\ln\left(\frac{441.4\text{K}}{473\text{K}}\right) = -1.346\text{kJ/K}$$

Thus,

$$\Delta S_{total} = \Delta S_{iron} + \Delta S_{alum} = 1.515 - 1.346 = \mathbf{0.169\ kJ/K}$$

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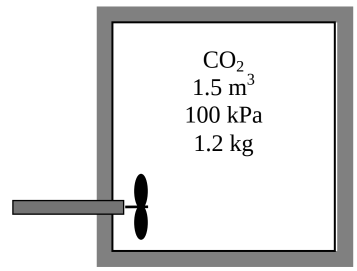
7-62 An insulated tank contains CO_2 gas at a specified pressure and volume. A paddle-wheel in the tank stirs the gas, and the pressure and temperature of CO_2 rises. The entropy change of CO_2 during this process is to be determined using constant specific heats.

Assumptions At specified conditions, CO_2 can be treated as an ideal gas with constant specific heats at room temperature.

Properties The specific heat of CO_2 is C_v = 0.657 kJ/kg.K (Table A-2).

Analysis Using the ideal gas relation, the entropy change is determined to be

$$\frac{P_2 V}{T_2} = \frac{P_1 V}{T_1} \longrightarrow \frac{T_2}{T_1} = \frac{P_2}{P_1} = \frac{120 kPa}{100 kPa} = 1.2$$



Thus,

$$\Delta S = m(s_2 - s_1) = m\left(C_{v,ave} \ln \frac{T_2}{T_1} + \cancelto{0}{R \ln \frac{V_2}{V_1}} \right) = mC_{v,ave} \ln \frac{T_2}{T_1}$$
$$= (2.7 \text{ kg})(0.657 \text{kJ/kg} \cdot \text{K}) \ln(1.2)$$
$$= \mathbf{0.323 \ kJ/K}$$

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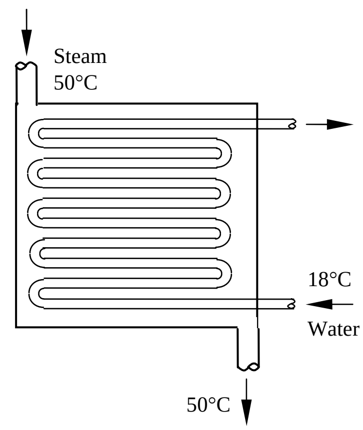
7-110 Steam is condensed by cooling water in the condenser of a power plant. The rate of condensation of steam and the rate of entropy generation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The enthalpy and entropy of vaporization of water at 50°C are h_{fg} =2382.7 kJ/kg and s_{fg}= 7.3725 kJ/kg.K (Table A-4). The specific heat of water at room temperature is C_p = 4.18 kJ/kg.°C (Table A-3).

Analysis (*a*) We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



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

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

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

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$

Then the heat transfer rate to the cooling water in the condenser becomes

$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{cooling water}}$$
$$= (101 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(27\text{°C} - 18\text{°C}) = 3800 \text{ kJ/s}$$

The rate of condensation of steam is determined to be

$$\dot{Q} = (\dot{m}h_{fg})_{steam} \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{3800 \text{kJ/s}}{2382.7 \text{ kJ/kg}} = \mathbf{1.595 \ kg/s}$$

(*b*) The rate of entropy generation within the condenser during this process can be determined by applying the rate form of the entropy balance on the entire condenser. Noting that the condenser is well-insulated and thus heat transfer is negligible, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{steam}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{steam}} s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{steam}}(s_4 - s_3)$$

Noting that water is an incompressible substance and steam changes from saturated vapor to saturated liquid, the rate of entropy generation is determined to be

$$\dot{S}_{gen} = \dot{m}_{\text{water}} C_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{steam}}(s_f - s_g) = \dot{m}_{\text{water}} C_p \ln\frac{T_2}{T_1} - \dot{m}_{\text{steam}} s_{fg}$$

$$= (101 \text{ kg/s})(4.18 \text{ kJ/kg.K})\ln\frac{27+273}{18+273} - (1.595 \text{ kg/s})(7.3725 \text{ kJ/kg.K}) = 1.100 \text{ kW/K}$$

7-132 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of entropy generation are to be determined.

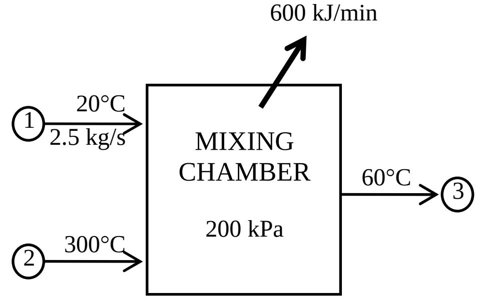
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Properties Noting that $T < T_{sat\ @\ 200\ kPa} = 120.23°C$, the cold water and the exit mixture streams exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. From tables,

$$\left.\begin{array}{l} P_1 = 200\text{kPa} \\ T_1 = 20^\circ\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f@20^\circ\text{C}} = 83.96\text{kJ/kg} \\ s_1 \cong s_{f@20^\circ\text{C}} = 0.2966\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200\text{kPa} \\ T_2 = 300^\circ\text{C} \end{array}\right\} \begin{array}{l} h_2 = 3071.8\text{kJ/kg} \\ s_2 = 7.8926\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 200\text{kPa} \\ T_3 = 60^\circ\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@60^\circ\text{C}} = 251.13\text{kJ/kg} \\ s_3 \cong s_{f@60^\circ\text{C}} = 0.8312\text{kJ/kg}\cdot\text{K} \end{array}$$



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

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

Energy balance:

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

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

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{Q}_{out} + \dot{m}_3 h_3$$

Combining the two relations gives $\dot{Q}_{out} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3 = \dot{m}_1 (h_1 - h_3) + \dot{m}_2 (h_2 - h_3)$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of the superheated steam is determined to be

$$\dot{m}_2 = \frac{\dot{Q}_{out} - \dot{m}_1 (h_1 - h_3)}{h_2 - h_3} = \frac{(600/60\text{kJ/s}) - (2.5\text{kg/s})(83.96 - 251.13)\text{kJ/kg}}{(3071.8 - 251.13)\text{kJ/kg}} = \mathbf{0.152 kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 2.5 + 0.152 = 2.652\ \text{kg/s}$

(*b*) The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings so that the boundary temperature of the extended system is 25°C at all times. It gives

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{out}}{T_{b,surr}} + \dot{S}_{\text{gen}} = 0$$

Substituting, the rate of entropy generation during this process is determined to be

$$\begin{aligned} \dot{S}_{gen} &= \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{out}}{T_{b,surr}} \\ &= (2.652\text{kg/s})(0.8312\text{kJ/kg}\cdot\text{K}) - (0.152\text{kg/s})(7.8926\text{kJ/kg}\cdot\text{K}) \\ &\quad - (2.5\text{kg/s})(0.2966\text{kJ/kg}\cdot\text{K}) + \frac{(600/60\ \text{kJ/s})}{298\,\text{K}} \\ &= \mathbf{0.297 kW/K} \end{aligned}$$

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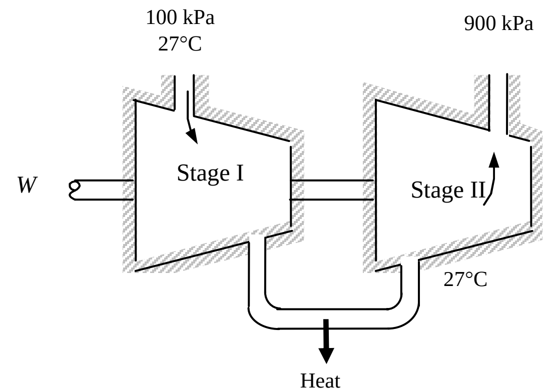
7-143 Air is compressed in a two-stage ideal compressor with intercooling. For a specified mass flow rate of air, the power input to the compressor is to be determined, and it is to be compared to the power input to a single-stage compressor. √

Assumptions **1** The compressor operates steadily. **2** Kinetic and potential energies are negligible. **3** The compression process is reversible adiabatic, and thus isentropic. **4** Air is an ideal gas with constant specific heats at room temperature.

Properties The gas constant of air is R =0.287 kPa.m^3/kg.K (Table A-1). The specific heat ratio of air is k = 1.4 (Table A-2)

Analysis The intermediate pressure between the two stages is

$$P_x = \sqrt{P_1 P_2} = \sqrt{(100\text{kPa})(900\text{kPa})} = 300\text{kPa}$$



The compressor work across each stage is the same, thus total compressor work is twice the compression work for a single stage:

$$\begin{aligned} w_{comp,in} &= (2)(w_{comp,in,I}) = 2\frac{kRT_1}{k-1}\left((P_x/P_1)^{(k-1)/k} - 1\right) \\ &= 2\frac{(1.4)(0.287\text{kJ/kg}\cdot\text{K})(300\text{K})}{1.4-1}\left(\left(\frac{300\text{kPa}}{100\text{kPa}}\right)^{0.4/1.4} - 1\right) \\ &= 222.2\text{kJ/kg} \end{aligned}$$

and

$$\dot{W}_{in} = \dot{m}w_{comp,in} = (0.02\text{kg/s})(222.2\text{kJ/kg}) = \mathbf{4.44kW}$$

The work input to a single-stage compressor operating between the same pressure limits would be

$$w_{\text{comp,in}} = \frac{kRT_1}{k-1}\left((P_2/P_1)^{(k-1)/k} - 1\right) = \frac{(1.4)(0.287\text{kJ/kg}\cdot\text{K})(300\text{K})}{1.4-1}\left(\left(\frac{900\text{kPa}}{100\text{kPa}}\right)^{0.4/1.4} - 1\right) = 263.2\text{kJ/kg}$$

and

$$\dot{W}_{in} = \dot{m}w_{comp,in} = (0.02\text{kg/s})(263.2\text{kJ/kg}) = \mathbf{5.26\ kW}$$

Discussion Note that the power consumption of the compressor decreases significantly by using 2-stage compression with intercooling.

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SOLUTIONS: HOMEWORK #8

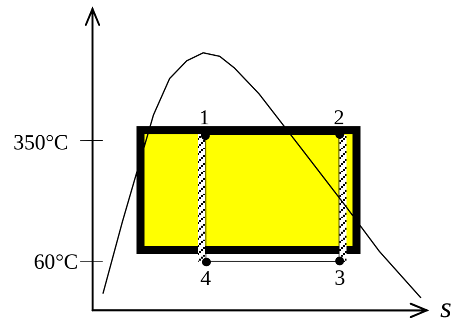
Chapter 8 Problems

8-95 A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the pressure at the turbine inlet, and the net work output are to be determined.

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

Analysis (*a*) The thermal efficiency is determined from

$$\eta_{\text{th, C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{60+273 \text{ K}}{350+273 \text{ K}} = \mathbf{46.5\%}$$



(*b*) Note that $s_2 = s_3 = s_f + x_3 s_{fg}$

$= 0.8312 + 0.891 \times 7.0784 = 7.138$ kJ/kg·K

Thus ,

$$\left.\begin{array}{l} T_2 = 350\,°\text{C} \\ s_2 = 7.138 \text{ kJ/kg}\cdot\text{K} \end{array}\right\} P_2 = 1.40 \text{ MPa}$$

(*c*) The net work can be determined by calculating the enclosed area on the T-s diagram,

$$s_4 = s_f + x_4 s_{fg} = 0.8312 + (0.1)(7.0784) = 1.539 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$w_{\text{net}} = \text{Area} = (T_H - T_L)(s_3 - s_4) = (350-60)(7.138-1.539) = \mathbf{1624 \text{ kJ/kg}}$$

Drawing from solution (1-2-3-4) is incorrect. Yellow-shaded area is correct. Water is compressed liquid at the boiler inlet, not a mixture, since $s_1 < s_g$ at 350 ^{o}C. Numerical answers are okay.

8-97C Heat rejected decreases; everything else increases.

8-103C Yes, because the saturation temperature of steam at 10 kPa is 45.81°C, which is much higher than the temperature of the cooling water.

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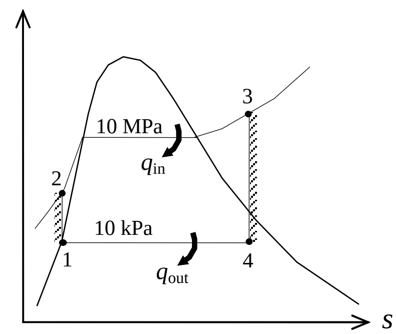
8-105 A steam power plant that operates on a simple ideal Rankine cycle is considered. The quality of the steam at the turbine exit, the thermal efficiency of the cycle, and the mass flow rate of the steam are to be determined.

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

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.83\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$



$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.00101\text{ m}^3\text{/kg})(10{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.09\text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 10.09 = 201.92\text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 10\text{ MPa} \\ T_3 &= 500\,°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3373.7\text{ kJ/kg} \\ s_3 &= 6.5966\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 10\text{ kPa} \\ s_4 &= s_3 \end{aligned}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.5966 - 0.6493}{7.5009} = 0.793$$

$$h_4 = h_f + x_4 h_{fg} = 191.83 + (0.793)(2392.8) = 2089.3\text{ kJ/kg}$$

(*b*)

$$q_{\text{in}} = h_3 - h_2 = 3373.7 - 201.92 = 3171.78\text{ kJ / kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2089.3 - 191.83 = 1897.47\text{ kJ / kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3171.78 - 1897.47 = 1274.31\text{ kJ / kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1274.31\text{kJ/kg}}{3171.78\text{kJ/kg}} = \mathbf{40.2\%}$$

(*c*)

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{210{,}000\text{kJ/s}}{1274.31\text{kJ/kg}} = \mathbf{165\text{ kg/s}}$$

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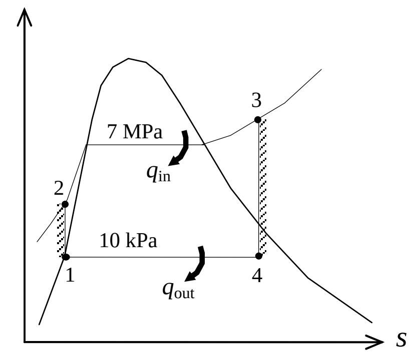
8-111 A steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The thermal efficiency of the cycle, the mass flow rate of the steam, and the temperature rise of the cooling water are to be determined.

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

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.83\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$
$$w_{p,\text{in}} = v_1(P_2 - P_1)$$
$$= (0.00101\ \text{m}^3/\text{kg})(7{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 7.06\text{ kJ/kg}$$



$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 7.06 = 198.89\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 7\text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3410.5\text{ kJ/kg} \\ s_3 = 6.7975\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7975 - 0.6493}{7.5009} = 0.820$$
$$h_4 = h_f + x_4 h_{fg} = 191.83 + (0.820)(2392.8) = 2153.93\text{ kJ/kg}$$

Thus,
$$q_{\text{in}} = h_3 - h_2 = 3410.5 - 198.89 = 3211.61\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2153.93 - 191.83 = 1962.10\text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3211.61 - 1962.10 = 1249.51\text{ kJ/kg}$$

and
$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1249.51\text{kJ/kg}}{3211.61\text{kJ/kg}} = \mathbf{38.9\%}$$

(*b*)
$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000\text{kJ/s}}{1249.51\text{kJ/kg}} = \mathbf{36.0 kg/s}$$

(*c*) The rate of heat rejection to the cooling water and its temperature rise are

$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (36.0\ \text{kg/s})(1962.1\ \text{kJ/kg}) = 70{,}636\ \text{kJ/s}$$
$$\Delta T_{\text{coolingwater}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}C)_{\text{coolingwater}}} = \frac{70{,}636\ \text{kJ/s}}{(2000\ \text{kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})} = \mathbf{8.45°C}$$

AREN 2110 SOLUTIONS
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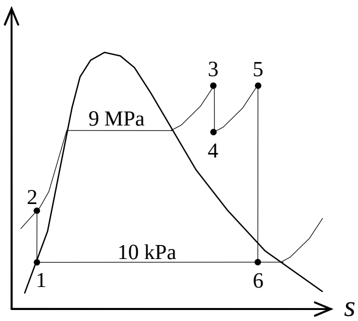
HOMEWORK ASSIGNMENTS 6, 7 and 8

8-121 A steam power plant that operates on an ideal reheat Rankine cycle between the specified pressure limits is considered. The pressure at which reheating takes place, the total rate of heat input in the boiler, and the thermal efficiency of the cycle are to be determined.

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

Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),



$$h_1 = h_{\text{sat@ 10 kPa}} = 191.83 \text{ kJ/kg}$$

$$v_1 = v_{\text{sat@ 10 kPa}} = 0.00101 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.00101 \text{ m}^3\text{/kg})(9{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) \\ &= 9.08 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 9.08 = 200.91 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 9 \text{ MPa} \\ T_3 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3386.1 \text{ kJ/kg} \\ s_3 &= 6.6576 \text{ kJ/kg} \cdot \text{K} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 10 \text{ kPa} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} h_6 &= h_f + x_6 h_{fg} = 191.83 + (0.90)(2392.8) = 2345.4 \text{ kJ/kg} \\ s_6 &= s_f + x_6 s_{fg} = 0.6493 + (0.90)(7.5009) = 7.4001 \text{ kJ/kg} \cdot \text{K} \end{aligned}$$

$$\left.\begin{aligned} T_5 &= 500°\text{C} \\ s_5 &= s_6 \end{aligned}\right\} \begin{aligned} P_5 &= \mathbf{2.146 \text{ MPa}} \text{ (the reheat pressure)} \\ h_5 &= 3466.0 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 2.146 \text{ MPa} \\ s_4 &= s_3 \end{aligned}\right\} h_4 = 2979.5 \text{ kJ/kg}$$

(*b*) The rate of heat supply is

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{m}[(h_3 - h_2) + (h_5 - h_4)] \\ &= (25 \text{ kJ/s})(3386.1 - 200.91 + 3466 - 2979.5) \text{kJ/kg} = \mathbf{91{,}792 \text{ kJ/s}} \end{aligned}$$

(*c*) The thermal efficiency is determined from

$$\dot{Q}_{\text{out}} = \dot{m}(h_6 - h_1) = (25 \text{ kJ/s})(2345.4 - 191.83) \text{kJ/kg} = 53{,}839 \text{ kJ/s}$$

Thus,

$$\eta_{th} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{53{,}839 \text{ kJ/s}}{91{,}792 \text{ kJ/s}} = \mathbf{41.3\%}$$

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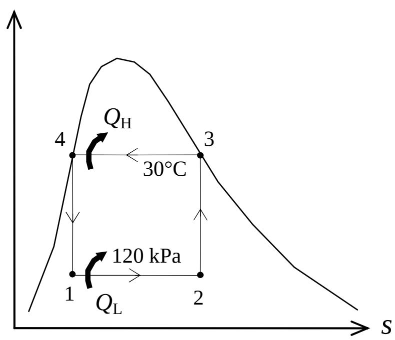
8-123 A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The coefficient of performance, the amount of heat absorbed from the refrigerated space, and the net work input are to be determined.

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

Analysis (*a*) Noting that T_H = 30°C = 303 K and $T_L = T_{\text{sat @ 120 kPa}}$ = -22.36°C = 250.6 K, the COP of this Carnot refrigerator is determined from

$$\text{COP}_{\text{R,C}} = \frac{1}{T_H/T_L - 1} = \frac{1}{(303\text{ K})/(250.6\text{ K}) - 1} = \mathbf{4.78}$$



(*b*) From the refrigerant tables (Table A-11),

$$h_3 = h_{g\,@\,30°\text{C}} = 263.50 \text{ kJ/kg}$$
$$h_4 = h_{f\,@\,30°\text{C}} = 91.49 \text{ kJ/kg}$$

Thus,

$$q_H = h_3 - h_4 = 263.50 - 91.49 = 172.01 \text{ kJ/kg}$$

and

$$\frac{q_H}{q_L} = \frac{T_H}{T_L} \longrightarrow q_L = \frac{T_L}{T_H} q_H = \left(\frac{250.6\text{ K}}{303\text{ K}}\right)(172.01\text{ kJ/kg}) = \mathbf{142.3\ \ kJ/kg}$$

(*c*) The net work input is determined from

$$w_{\text{net}} = q_H - q_L = 172.01 - 142.3 = \mathbf{29.71\ \ kJ/kg}$$

8-132 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.

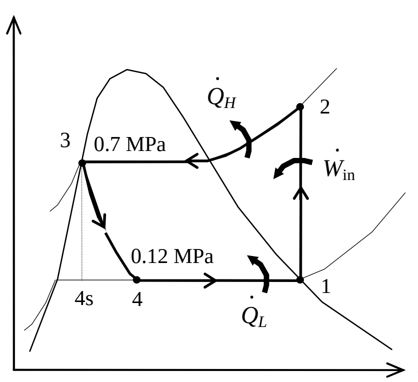
Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12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

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

and

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

8-132 CONTINUED

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

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

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

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

8-146 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.

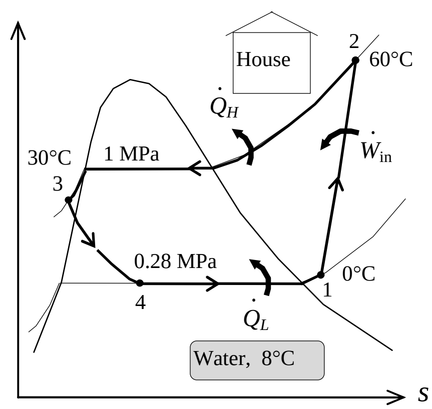
Analysis (*a*) From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 280\ \text{kPa} \\ T_1 = 0°\text{C} \end{array}\right\} h_1 = 247.64\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1.0\ \text{MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 291.36\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.0\ \text{MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\ @\ 30°\text{C}} = 91.49\ \text{kJ/kg}$$

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



The mass flow rate of the refrigerant is

$$\dot{m}_R = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{60{,}000/3{,}600\ \text{kJ/s}}{(291.36 - 91.49)\ \text{kJ/kg}} = 0.0834\ \ \text{kg/s}$$

Then the power input to the compressor becomes

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.0834\ \text{kg/s})(291.36 - 247.64)\ \text{kJ/kg} = \mathbf{3.65\ kW}$$

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

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.0834\ \text{kg/s})(247.64 - 91.49)\ \text{kJ/kg} = \mathbf{13.02\ kW}$$

(*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,

$$\dot{W}_{\text{increase}} = \dot{W}_e - \dot{W}_{\text{in}} = 16.67 - 3.65 = \mathbf{13.02kW}$$

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HOMEWORK ASSIGNMENTS 6, 7 and 8

8-192 A heat pump water heater has a COP of 2.2 and consumes 2 kW when running. It is to be determined if this heat pump can be used to meet the cooling needs of a room by absorbing heat from it.

Assumptions The COP of the heat pump remains constant whether heat is absorbed from the outdoor air or room air.

Analysis The COP of the heat pump is given to be 2.2. Then the COP of the air-conditioning system becomes

$$\mathrm{COP}_{\text{air-cond}} = \mathrm{COP}_{\text{heat pump}} - 1 = 2.2 - 1 = 1.2$$

Then the rate of cooling (heat absorption from the air) becomes

$$\dot{Q}_{\text{cooling}} = \mathrm{COP}_{\text{air-cond}}\dot{W}_{in} = (1.2)(2\text{ kW}) = 2.4\text{ kW} = 8640\text{ kJ/h}$$

since 1 kW = 3600 kJ/h. We conclude that this heat pump **can meet** the cooling needs of the room since its cooling rate is greater than the rate of heat gain of the room.