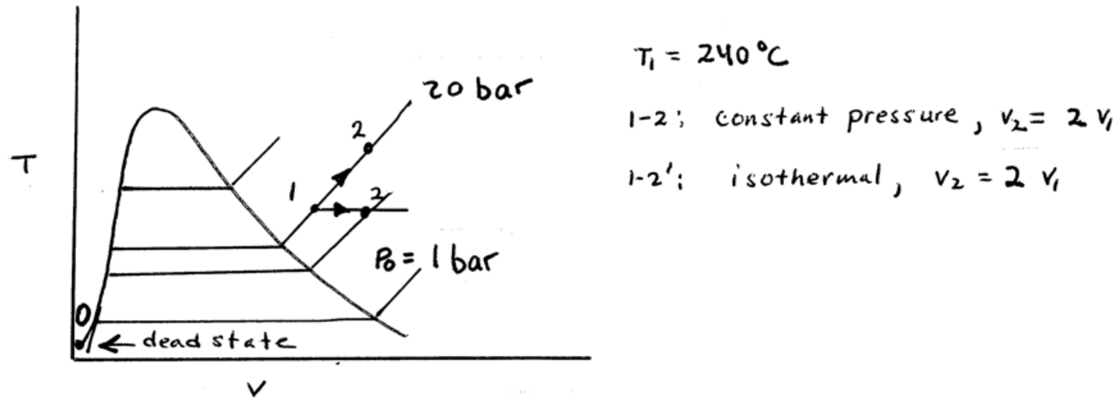
Homework 7 solutions

Problem 1: Exercise 7.20 in text



Known: One kg of steam at 20 bar and 240 ^oC undergoes two different processes.

Find: For each process, determine the change in exergy.

Assumptions:

1. The system is the 1 kg of steam.
2. For the environment, T_0 = 20 ^oC, p_0 = 1 bar.
3. The effects of motion and gravity are negligible.

Analysis:

Equation (7.10) when simplified yields,

$$E_2 - E_1 = m\left[(u_2 - u_1) + P_0(v_2 - v_1) - T_0(s_2 - s_1)\right]$$

(a) From Table A-4, u_1 = 2659.6 kJ/kg, v_1 = 0.1085 m3/kg, s_1 = 6.4952 kJ/kg at 240 ^oC and 20 bar. For each process, the volume is doubled. But mass also remains the same. Therefore, $v_2 = 2v_1$ for each process.

(a) Constant pressure process: Interpolating in table A-4 at 20 bar with v_2 = 0.217 m^3/kg, u_2 = 3423.1 kJ/kg, s_2 = 7.8849 kJ/kg.

$$E_2 - E_1 = (1kg)\left[(3423.1 - 2659.6)\frac{kJ}{kg} + 100\frac{kN}{m^2}(0.217 - 0.1085)\frac{m^3}{kg} - 293K(7.8849 - 6.4952)\frac{kJ}{kg\ K}\right]$$

= 367.2 kJ/kg .

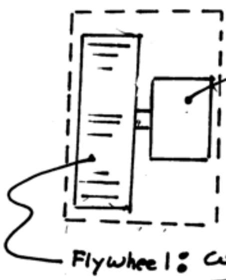
(b) Isothermal process: Interpolating in table A-4 at 240 ^oC with v_2 = 0.217 m^3/kg, u_2 = 2689.96 kJ/kg, s_2 = 6.8476 kJ/kg.

$$E_2 - E_1 = (1kg)\left[(2689.96 - 2659.6) + 100(0.1085) - 293(6.8476 - 6.4952)\right]kJ/kg$$

= -62.04 kJ/kg .

Note: For (a) pressure is kept constant and temperature increases relative to T_0. Thexergy increases in this process. For (b), the temperature is kept constant and pressure decreases relative to p0. The exergy decreases in this process.

Problem 2: Exercise 7.21 in text



Brake lining:
$m = 2.27\ kg$
$c = 4.19\ \frac{kJ}{kg \cdot K}$
$T_1 = 16°C$

Flywheel: $\omega_1 = 3000$ RPM, $\omega_2 = 0$,
$I_{mom} = 6.74\ kg \cdot m^2$

Note:
$\omega_1 = \left(3000 \frac{Rev}{min}\right)\left(2\pi \frac{rad}{rev}\right)\left(\frac{1 min}{60 s}\right)$
$= 100\pi$ rad/s

Known: Data is provided for a flywheel braked to rest.

Find: Determine the final temperature after the flywheel is braked to rest and the maximum theoretical rotational speed that could be attained by the flywheel using the energy stored in the flywheel.

Assumptions:
1. For the system shown in the figure W = Q = 0 and ΔPE = 0.
2. The brake is modeled as an incompressible substance with constant specific heat c.
3. For the environment T_0 = 16 ^{o}C.

Analysis:
(a) With W = Q = ΔPE = 0, the energy balance reduces to ΔU + ΔKE = 0, where ΔU is the change in internal energy of the brake liner and ΔKE, the change in kinetic energy of the flywheel.

$$mc(T_2 - T_1) + \left(0 - \frac{1}{2} \mathrm{I}_{mom}\, \omega^2\right) = 0 \;\; \Rightarrow \;\; T_2 - T_1 = \frac{0.5\ \mathrm{I}_{mom}\, \omega^2}{mc}$$

$$= \frac{0.5\left(100\pi\ \dfrac{rad}{s}\right)^2 (6.74\ kgm^2)}{(2.27\ kg)\left(4.19\dfrac{kJ}{kg\ K}\right)\dfrac{10^3 N.m}{1\ kJ}} = 35\ K$$

Therefore, the final temperature of the brake lining is 324 K or (51 ^{o}C).

(b) One observes from the energy balance that energy is stored in the brake lining. However, it is the exergy stored and not the energy stored that is significant in determining the final rotational speed of the flywheel. At the final state, the flywheel being at rest, makes no significant contribution to the system exergy. The brake lining does contribute exergy, because $T_2 \neq T_0$. The final exergy is then found using the incompressible substance relations

$$\mathrm{E}_2 - \mathrm{E}_0 = m\left[(u_2 - u_0) + P_0(v_2 - v_0) - T_0(s_2 - s_0)\right] = m\ [c(T_2 - T_0) - c\ln\left(T_2 / T_0\right)]$$

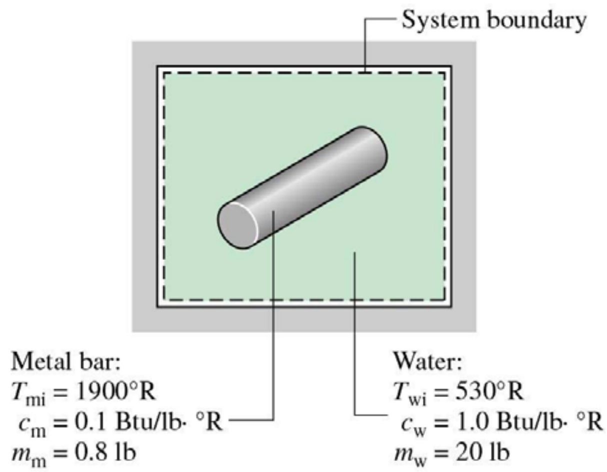
$$= 2.27kg\left[4.19\frac{kJ}{kg\ K}\right]\left[35K - 289\ln\left(\frac{324}{289}\right)\right] = \underline{18.67\ kJ}$$

In principle, the exergy at the final state can be used to bring the flywheel into motion as the brake lining is restored to its initial temperature. Accordingly the maximum theoretical rotational speed, $\omega_{\max}$ must satisfy $\mathrm{E}_2 - \mathrm{E}_0 = 0.5\,\mathrm{I}_{mom}\,\omega_{\max}^2$

$$\Rightarrow \quad \omega_{\max} = \sqrt{\frac{2(\mathrm{E}_2 - \mathrm{E}_0)}{\mathrm{I}_{mom}}} = \sqrt{\frac{2\ (18.67kJ)}{6.74\ kgm^2}\left|\frac{10^3\,Nm}{1\ kJ}\right|\left|\frac{1\ kgms^{-2}}{1\ N}\right|}\left|\frac{1\ rev}{2\pi\ \mathrm{rad}}\right|\left|\frac{60\ s}{1\mathrm{min}}\right| = \underline{711\ \mathrm{RPM}}$$

Problem 3: Exercise 7.30 in text

Known: A hot metal bar is quenched by immersing it in a tank of water
Find: Determine the exergy destruction



Assumptions:
1. As shown in the schematic, the metal bar and water form a closed system.
2. For the system, Q = W = 0 and the effects of motion and gravity are negligible.
3. The metal bar and water are each modeled as incompressible.
4. T_o = 537 °R (77 °F)

Analysis:
An energy balance of the system reduces to give $\Delta U|_{water} + \Delta U|_{metal} = 0$…………(1)

Using assumption 2 and the fact that both metal and water have constant specific heats over this temperature range, Eq. (1) is expressed as

$$m_w c_w (T_f - T_{wi}) + m_m c_m (T_f - T_{mi}) = 0 \quad \Rightarrow \quad T_f = \frac{m_w (c_w / c_m) T_{wi} + m_m T_{mi}}{m_w (c_w / c_m) + m_m}$$

After substituting all the given values, we get T_f = 535 °R.

An exergy balance for this system reduces to give

$$\Delta \mathrm{E} = \int_1^2 \left[1 - \frac{T_0}{T_b}\right]\cancelto{0}{\delta Q} - [\cancelto{0}{W} - p_0 \cancelto{0}{\Delta V}] - \mathsf{E}_\mathrm{d} \ \Rightarrow\ \mathsf{E}_\mathrm{d} = -\Delta \mathrm{E}$$

Since energy is an extensive property, $\Delta\mathrm{E} = \Delta\mathrm{E}|_{water} + \Delta\mathrm{E}|_{metal}$. Using Equation (7.10)

0 0

$$\mathsf{E}_d = -\left[[\Delta U + P_0\Delta V - T_0\Delta S]_{water} + [\Delta U + P_0\Delta V - T_0\Delta S]_{metal}\right]$$

$$\mathsf{E}_d = \left[T_0\Delta S\,|_{water} + T_0\Delta S\,|_{metal}\right] = T_0\left[m_w c_w \ln\left(\frac{T_f}{T_{wi}}\right) + m_m c_m \ln\left(\frac{T_f}{T_{mi}}\right)\right]$$

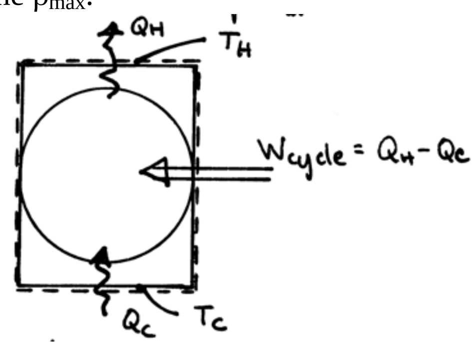
$$= 537\ ^{o}\mathrm{R}[\,(20\ lb)\left(1.0\frac{Btu}{lb\ ^{o}\mathrm{R}}\right)\ln\frac{535}{530} + (0.8\ lb)\left(0.1\frac{Btu}{lb\ ^{o}\mathrm{R}}\right)\ln\frac{535}{1900}\,] = \underline{46.4\ \mathrm{Btu}}$$

Note: (1)Under the current assumptions the system is isolated. Eq. (1) indicates that the energy is conserve. However, the total exergy of the system does not remain constant since exergy is destroyed. (2) $\mathrm{E_d}$ *can also be expressed as* $T_0\,\sigma$, *where* σ *is the entropy generated in the system. In example 6.5 of the text,* σ *is calculated for the same system.*

Problem 4: Exercise 7.43 in text

Known: A system undergoes a refrigeration cycle while receiving heat Q_C at temperature T_C and discharging Q_H at T_H, where $T_H > T_C$. Q_H and Q_C are the only heat transfers.

Find: (a) Show that W_{cycle} cannot be zero using exergy balance. (b) Obtain a specified expression for coefficient of performance β. (c) Determine β_{max}.



Assumptions:
1. The system undergoes a refrigeration cycle steady
2. Q_H and Q_C are the only heat transfers
3. T_H and T_C are constant and $T_H > T_C$.
4. The environment temperature is T_0.

Analysis:
(a) An exergy balance for the cycle reads

$$\cancelto{0}{\Delta E_{cycle}} = \left(1-\frac{T_0}{T_C}\right)Q_C - \left(1-\frac{T_0}{T_H}\right)Q_H - \left[W - p_0\cancelto{0}{\Delta V}\,|_{cycle}\right] - \mathsf{E}_d$$

, where $\Delta \mathrm{E}_{cycle}$ and ΔV_{cycle} are both zero. Introducing the energy balance, $W_{cycle} + Q_C = Q_H$ we get,

$$0 = \left(1-\frac{T_0}{T_C}\right)Q_C - \left(1-\frac{T_0}{T_H}\right)(Q_C + W_{cycle}) + W_{cycle} - \mathsf{E}_d \text{, (where } W = -W_{cycle}\text{)}$$

$$0 = \left(\frac{T_0}{T_H} - \frac{T_0}{T_C}\right)Q_C + \frac{T_0}{T_H}W_{cycle} - \mathsf{E}_d \quad \ldots\ldots(1)$$

Solving for E_d and letting $W_{cycle} = 0$, $\mathsf{E}_d = T_0\left(\frac{1}{T_H} - \frac{1}{T_C}\right)Q_C < 0$ which is impossible.

Therefore W_{cycle} cannot be zero.

(b) Re-expressing Eq.(1), we have $\left(\frac{T_H - T_C}{T_C T_H}\right) Q_C = \frac{W_{cycle}}{T_H} - \frac{\mathrm{E}_{\mathrm{d}}}{T_0}$

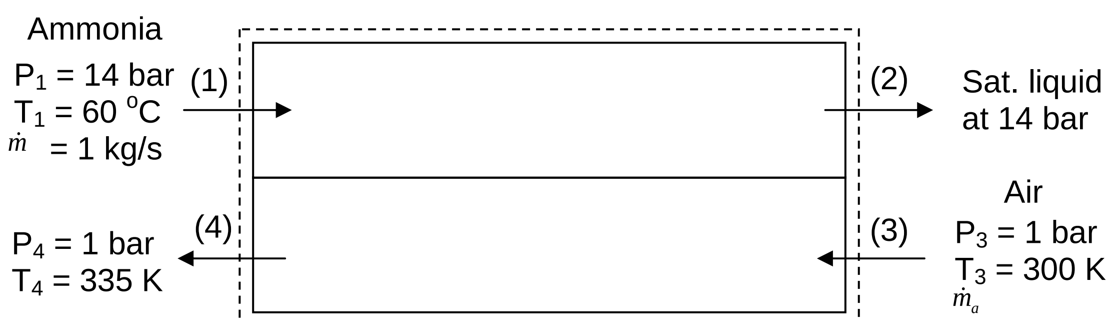
Therefore, $\beta = \frac{Q_C}{W_{cycle}} = \left(\frac{T_C}{T_H - T_C}\right)\left(1 - \frac{T_H \mathrm{E}_{\mathrm{d}}}{T_0 W_{cycle}}\right) = \left(\frac{T_C}{T_H - T_C}\right)\left(1 - \frac{T_H \mathrm{E}_{\mathrm{d}}}{T_0 (Q_H - Q_C)}\right)$

(c) From the result of (b), β increases as E_{d} goes to zero, therefore $\beta_{\max} = \left(\frac{T_C}{T_H - T_C}\right)$

Problem 5: Exercise 7.71 of text

Known: Ammonia and air flow on opposite sides of a counterflow heat exchanger and operating at steady state. Operating data are given.

Find: (a) The change in flow exergy of each stream and (b) the rate of exergy destruction.



Assumptions: (1) The control volume shown is at a steady state (2) For the control volume $\dot{W}_{cv} = \dot{Q}_{cv} = 0$, and the kinetic and potential energy effects are negligible. (3) The air is modeled as an ideal gas. (4) For the environment, T_0 = 300 K and P_0 = 1 bar.

Analysis: (a) First we use mass and energy rate balances to determine $\dot{m}_a$. From the mass balance we have, $\dot{m}_1 = \dot{m}_2 = \dot{m}$ and $\dot{m}_3 = \dot{m}_4 = \dot{m}_a$. From energy balance, and assumptions (1) and (2), we have $0 = \cancelto{0}{\dot{Q}} - \cancelto{0}{\dot{W}}_{cv} + \dot{m}(h_1 - h_2) + \dot{m}_a(h_3 - h_4)$

Inserting the enthalpy values from tables A-14, A-15 and A-22 and solving for $\dot{m}_a$, we have $\dot{m}_a = \frac{h_1 - h_2}{h_4 - h_3}\dot{m} = \frac{1542.89 - 352.97}{335.38 - 300.19}(1\ kg/s) = 33.81\ kg/s$

The change in flow exergy rate for ammonia

$$\dot{\mathrm{E}}_{f2} - \dot{\mathrm{E}}_{f1} = \dot{m}(e_{f2} - e_{f1}) = \dot{m}\left[(h_2 - h_1) - T_0(s_2 - s_1)\right]$$

$$= 1.0\frac{kg}{s}\left[1542.89 - 352.97 - 300(1.2987 - 5.136)\right]\frac{kJ}{kg\ K}\left|\frac{1\ kW}{1\ kJ/s}\right| = -38.73\text{ kW}$$

The change in exergy flow rate of the air stream is calculated as

$$\dot{E}_{f4} - \dot{E}_{f3} = \dot{m}(e_{f4} - e_{f3}) = \dot{m}\left[(h_4 - h_3) - T_0(s_4 - s_3)\right]$$

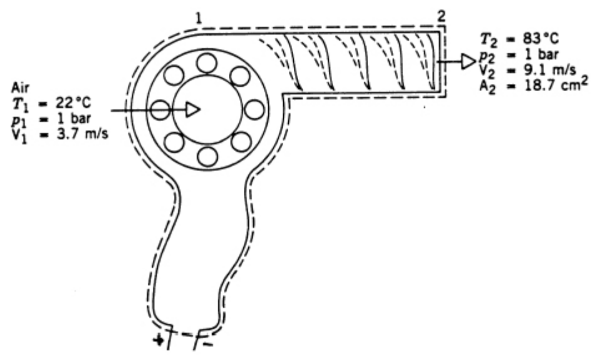
$$= 33.81\frac{kg}{s}\left[335.38 - 330.19 - 300(1.8129 - 1.70203)\right]\frac{kJ}{kg\ K}\left|\frac{1\ kW}{1\ kJ/s}\right| = 65.22 \text{ kW}$$

(b) At steady state, the exergy rate balance for the counterflow as a heat exchanger reduces to

$$0 = \sum_j \left(1 - \cancelto{0}{\frac{T_0}{T_j}}\right)\dot{Q}_j - \left[\cancelto{0}{\dot{W}_{cv}} - P_0 \cancelto{0}{\frac{dV}{dt}}\right] + \dot{m}(e_{f1} - e_{f2}) + \dot{m}_a(e_{f3} - e_{f4}) - \dot{E}_d$$

Thus, $\dot{E}_d = -\left(\dot{E}_{f2} - \dot{E}_{f1}\right) - \left(\dot{E}_{f4} - \dot{E}_{f3}\right) = 38.73\ kW - 65.22\ kW = -26.49\ kW$. Clearly, a negative value of $\dot{E}_d$ is not possible for this process. This implies that this hypothetical heat exchanger does not exist.

Problem 6(*): Exercise 7.92 of text



Known: Steady-state operating data are provided for a hand held hair dryer.

Find: (a)Evaluate the power input. (b)Devise and evaluate an exergetic efficiency.

Assumptions:

(1) The control volume shown in the accompanying figure is at steady state.

(2) For the control volume, $\dot{Q}_{cv} = 0$ and potential energy effects can be ignored.

(3) Air is modeled as air as an ideal gas.

(4) Environment temperature, T_o = 295K(22^{o}C)

Analysis:

(a) The power $\dot{W}_{cv}$ for the control volume in the accompanying figure can be evaluated using mass and energy rate balances. At steady state, the mass rate balance reduces to give $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using this relationship between the mass flow rates together with assumption 2, the energy rate balance reduces at steady state to give:

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

The mass flow rate $\dot{m}$ can be evaluated from the expression

$$\dot{m} = \frac{A_2V_2}{v_2} = \frac{A_2V_2}{RT_2/p_2} = \frac{\left[\frac{18.7cm^2}{(100cm/m)^2}\right]\left(9.1\frac{m}{s}\right)\left(10^5\ \frac{N}{m^2}\right)}{\left(\frac{8314}{28.97}\ \frac{Nm}{kgK}\right)(356K)}$$

$\dot{m}$ = 0.0167 kg/s

Then with specific enthalpy values from Table A-22, h_1 = 295kJ/kg and h_2 = 356.5 kJ/kg, $\dot{W}_{cv}$ is obtained as follows*:*

$$\dot{W}_{cv} = 0.0167\frac{kg}{s}\left[295.2\frac{kJ}{kg} - 356.5\frac{kJ}{kg} + \frac{(3.7m/s)^2 - (9.1m/s)^2}{2}\left|\frac{1N}{1kg\cdot m/s^2}\right|\left|\frac{1kJ}{1000N\cdot m}\right|\right]$$

$$\dot{W}_{cv} = 0.0167\frac{kg}{s}\left[-61.3\frac{kJ}{kg} - 0.03\frac{kJ}{kg}\right]\left|\frac{1kW}{1kJ/s}\right|$$

$\dot{W}_{cv}$ = -1.02 kW

The minus sign reminds us that an electrical power input is required to operate the hair dryer.

(b) Since $\dot{Q}_{cv} = 0$ and $\dot{m}_1 = \dot{m}_2$, the steady state exergy rate balance reduces as follows*:*

$$o = \sum_j \left[1 - \frac{T_o}{T_j}\right]\dot{Q}_j - \dot{W}_{cv} + \dot{m}\left(e_{f1} - e_{f2}\right) - \dot{E}_d \Rightarrow -\dot{W}_{cv} = \dot{m}\left(e_{f2} - e_{f1}\right) - \dot{E}_d$$

This result shows that the exergy input, $-\dot{W}_{cv}$, either goes into increasing the exergy of the air stream, $\dot{m}(e_{f2} - e_{f1})$, or is destroyed, $\dot{E}_d$. Thus, an exergetic efficiency can be defined as

$$\varepsilon = \frac{\dot{m}\left(e_{f2} - e_{f1}\right)}{\left(-\dot{W}_{cv}\right)} = \frac{\dot{m}\left[\left(h_2 - h_1\right) - T_o\left(s_2 - s_1\right) + \frac{V_2^2 - V_1^2}{2}\right]}{\left(-\dot{W}_{cv}\right)}$$

$$= \frac{\dot{m}\left[\left(h_2 - h_1\right) - T_o\left(s_2^o - s_1^o - R\ln\left(p_2/p_1\right)\right) + \frac{V_2^2 - V_1^2}{2}\right]}{\left(-\dot{W}_{cv}\right)}$$

With data from table A-22

$$\varepsilon = \frac{(0.0167kg/s)\left[(356.5 - 295.2) - 295(1.874 - 1.685 - 0) + \frac{(9.1)^2 - (3.7)^2}{2(1000)}\right]}{(1.02kW)}\frac{kJ}{kg}\left|\frac{1kW}{1kJ/s}\right|$$

$\varepsilon =$ 0.091 (9.1%)

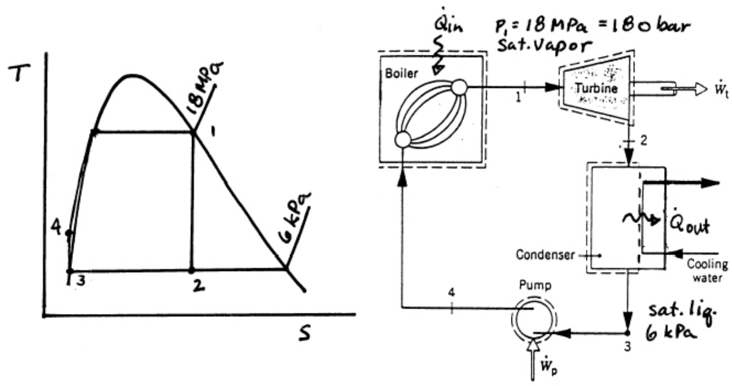
In this device a relatively high potential for use electrical input is used to obtain a stream of slightly warm air. The low value of the efficiency ε emphasizes that the source and end use are not well matched.

Problem 7: Exercise 8.6 of text

Known: Water is the working fluid in an ideal Rankine cycle. The condenser pressure and the turbine inlet state are specified.

Find:

(a) the net work per unit mass of steam flow, (b) the heat transfer per unit mass of steam flow through the boiler, (c) the thermal efficiency, and (d) the heat transfer to cooling water passing through the condenser per unit mass of steam condensed.



Assumptions:

(1) Each component of the cycle is analyzed as a control volume at steady state. The control volumes are shown on the accompanying sketch above by dashed lines.

(2) All processes of the working fluid are internally reversible.

(3) The turbine and pump operate adiabatically.

(4) Kinetic and potential energy effects are negligible.

(5) Saturated vapor enters the turbine. Condensate exits the condenser as saturated liquid.

Analysis:

First, fix each of the principal states.

State 1: p_1 = 18 MPa, sat. vapor → h_1 = 2509.1 kJ/kg, s_1 = 5.1044 kJ/kgK

State 2: p_2 = 6 kPa, $s_2 = s_1$ → $x_2 = \dfrac{s_2 - s_{f2}}{s_{fg2}} = 0.5869$, h_2 =1569.4 kJ/kg

State 3: p_3 = 6kPa, sat. liquid → h_3 = 151.53kJ/kg

State 4: $h_4 = h_3 + v_3(p_4 - p_3)$ $= 151.53\dfrac{kJ}{kg} + \left(1.0064x10^{-3}\dfrac{m^3}{kg}\right)(180-0.06)bar\left|\dfrac{10^5 N/m^2}{1bar}\right|\left|\dfrac{1kJ}{1000Nm}\right|$

$$h_4 = 151.53 + 18.11 = 169.64 kJ/kg$$

(a) The net work developed per unit mass of steam flow is:

$$\frac{\dot{W}_{cycle}}{\dot{m}} = \frac{\dot{W}_t}{\dot{m}} - \frac{\dot{W}_p}{\dot{m}} = (h_1 - h_2) - (h_4 - h_3) = (2509.1 - 1569.4) - (169.64 - 151.53)$$

$$\frac{\dot{W}_{cycle}}{\dot{m}} = \underline{921.59\ kJ/kg}$$

(b) The rate of heat transfer per unit mass of steam flow through the boiler is:

$$\frac{\dot{Q}_{in}}{\dot{m}} = (h_1 - h_4) = 2509.1 - 169.64 \qquad \Rightarrow \qquad \frac{\dot{Q}_{in}}{\dot{m}} = \underline{2339.46}\ kJ/kg$$

(c) The thermal efficiency is:

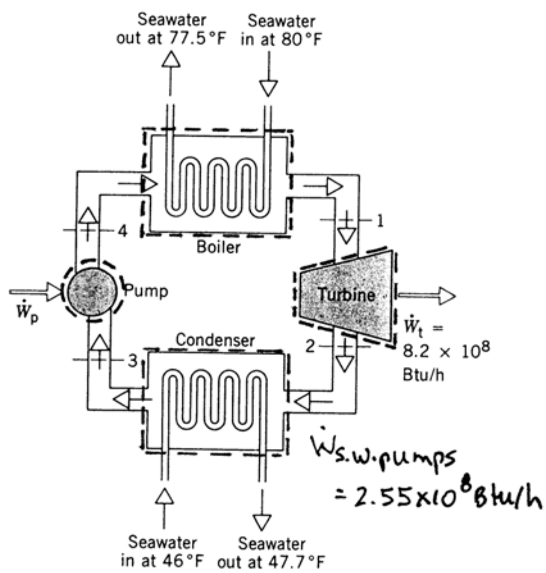
$$\eta = \frac{\dot{W}_{cycle}/\dot{m}}{\dot{Q}_{in}/\dot{m}} = \frac{921.59}{2339.46} \qquad \Rightarrow \quad \eta = 0.394(39.4\%)$$

(d) The rate of heat transfer to cooling water passing through the condenser is:

$$\frac{\dot{Q}_{out}}{\dot{m}} = (h_2 - h_3) = 1569.4 - 151.53 \qquad \Rightarrow \quad \frac{\dot{Q}_{out}}{\dot{m}} = \underline{1417.9\ kJ/kg}$$

Problem 8: Exercise 8.14 of text

Known: A Rankine power plant, using Ammonia as the working fluid has been proposed to utilize the naturally occurring temperature gradient within the ocean to provide electrical power.



Find: (a) the thermal efficiency and compare it to that of a reversible power cycle operating between reservoirs at the sea-water temperatures (b) the net power output, and (c) the sea-water flow rates through the boiler and condenser.

Assumptions: (1) Each control volume is at steady state, and $\dot{Q}_{cv}$ = 0. (2) kinetic and potential energy effects can be neglected. (3) the sea-water has properties of pure water assumed to be an incompressible liquid with c = 1.0 Btu/lb ^oR . (4) The isentropic efficiencies of the turbine and pump are 80 % and 75 % respectively. (5) The evaporating temperature of ammonia inside the boiler is 75 ^oF and the condensing temperature of ammonia inside the condenser is 50 ^oF.

Analysis:

The evaporating temperature of ammonia in the boiler and condensing temperature in the condenser are based on the sea-water temperatures in the boiler and condenser respectively.

First, fix each of the principal states.

State 1: T_1 = 75 ^oF, sat. vapor → h_1 = 629.20 Btu/lb, s_1 = 1.2048 *Btu/lb^{o}R*, $p_1 = p_{sat@T1}$ = 140.60 *lb/in^2*.

State 2: T_2 = 50 ^oF , $s_{2s} = s_1$ → $x_{2s} = \frac{s_{2s} - s_{f2}}{s_{fg2}} = 0.9625$, h_{2s} =604.81 *Btu/lb^{o}R*, $p_2 = p_{sat@T1}$ = 89.242 *lb/in^2*.

The isentropic efficiency of the turbine is 80%. Therefore, $h_2 = h_1 - \eta_f \left(h_1 - h_{2s}\right)$ $= 609.7\ Btu/lb$.

State 3: $p_3 = p_2$, sat. liquid $\Rightarrow h_3 = 97.55\ Btu/lb$

State 4: $p_4 = p_1$,

$$h_{4s} = h_3 + v_3(p_4 - p_3) = 97.55\ Btu/lb + (0.02564\ ft^3/lb)(140.6 - 89.242)\frac{lbf}{in^2}\left|\frac{144\ in^2}{1\ ft^2}\right|\left|\frac{1\ Btu}{778\ ft \cdot lbf}\right|$$

= 97.79 *Btu/lb*.

The isentropic efficiency of the pump is 70%. Therefore $h_4 = h_3 + (h_{4s} - h_3)/\eta_p$ $= 97.87\ Btu/lb$

(a) For the ammonia passing through the boiler, the heat transfer from the seawater is

$$\frac{\dot{Q}_{in}}{\dot{m}} = \left(h_1 - h_4\right) = 629.20 - 97.87 = 531.33\ Btu/lb$$

Similarly, for the ammonia passing through the condenser, the heat transfer to the seawater is $\frac{\dot{Q}_{out}}{\dot{m}} = \left(h_2 - h_3\right) = 609.7 - 97.55 = 512.15\ Btu/lb$

Thus, the thermal efficiency is $\eta = 1 - \frac{Q_C}{Q_H} = 1 - \frac{512.15}{531.33} = 0.0361 = \underline{3.61\%}$

For a reversible cycle with T_C = 506 ^oR and T_H = 540 ^oR, $\eta_{\max} = 1 - \frac{T_C}{T_H} = 0.063 = \underline{6.3\%}$

(b) The mass flow rate of ammonia is:

$$\dot{m}_{ammonia} = \frac{\dot{W}_t}{\left(h_1 - h_2\right)} = \frac{8.2x10^8\ \ Btu/hr}{(629.2 - 609.7)\ Btu/lb} = 4.205x10^7\ \ lb/hr$$

Therefore, $-\dot{W}_p = \dot{m}_{ammonia}(h_4 - h_3) = 1.346x10^7\ \ Btu/h$.

The net power output is given by, $\dot{W}_{net} = \dot{W}_t - \dot{W}_p - \dot{W}_{s.w.pumps} = \underline{5.515x10^8\ \ Btu/h}$

(c) Taking the boiler control volume overall:

$$\dot{m}_{s.w.boiler} = \frac{\dot{m}_{ammonia}(h_1 - h_4)}{c(T_{w,in} - T_{w,out})} = \frac{4.205x10^7(531.33)}{1(80 - 77.5)} = \underline{8.94x10^9\ \ lb/h}$$

Similarly for the condenser, we have

$$\dot{m}_{s.w.condensor} = \frac{\dot{m}_{ammonia}(h_2 - h_3)}{c(T_{w,in} - T_{w,out})} = \frac{4.205x10^7(512.15)}{1(47.7 - 46)} = \underline{1.267x10^{10}\ \ lb/h}$$