known: Water is the working fluid in a Rankine cycle. The condenser pressure and the turbine inlet state are specified. The mass flow rate of steam is given and the turbine and pump isentropic efficiencies are known. Temperature are given for cooling water entering and exiting the condenser.

**FIND:** Determine (a) the net power, (b) the thermal efficiency, and (c) the mass flow rate of cooling water passing through the condenser.

**SCHEMATIC & GIVEN DATA:**

![Diagram](image)

**ENGINEERING MODEL:** Same as Problem 8.2, except \( \eta_t = 0.88 \) and \( \eta_p = 0.82 \). Also, we assume that the pressure drop is negligible for the cooling water passing through the condenser, and that \( h = h_p(t) \).

**ANALYSIS:** First, fix each of the principal states.

- **State 1:** \( p_1 = 80 \text{ bar}, T_1 = 560 \text{°C} \) \( \Rightarrow \) \( h_1 = 3545.3 \text{ kJ/kg} \), \( s_1 = 6.90172 \text{ kJ/kg K} \)

- **State 2:** \( p_2 = 0.08 \text{ bar}, s_{2s} = s_1 \Rightarrow x_{2s} = \frac{s_{2s} - s_{f2}}{s_{g2} - s_{f2}} = 0.8269 \), \( h_{2s} = 2161.0 \text{ kJ/kg} \)

Using the isentropic turbine efficiency to get \( h_2 \)

\[
\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}} \Rightarrow h_2 = h_1 - \eta_t (h_1 - h_{2s}) = 2327.1 \text{ kJ/kg}
\]

- **State 3:** \( p_3 = 0.08 \text{ bar}, \text{ sat. liquid} \Rightarrow h_3 = 173.88 \text{ kJ/kg} \)

- **State 4:** \( p_4 = 80 \text{ bar}, h_4 \approx h_3 + \frac{h_3}{k_f} (p_4 - p_s) \)

\[
= 173.88 \text{ kJ/kg} + (1.0084 \times 10^5) \frac{m^3}{kg} (80 - 0.08) \text{ bar} = 181.94 \text{ kJ/kg}
\]

Using the isentropic pump efficiency to get \( h_4 \)

\[
\eta_p = \frac{h_{4s} - h_3}{h_4 - h_3} \Rightarrow h_4 = h_3 + \eta_p (h_4 - h_3) = 183.58 \text{ kJ/kg}
\]

(a) The net power developed is

\[
W_{\text{cycle}} = \dot{m} [h_1 - h_2] - (h_4 - h_3)]
\]

\[
= (7.8 \text{ kg/s}) [(3545.3 - 2327.1) - (183.58 - 173.88)] \text{ kJ/kg} = 9426 \text{ kW}
\]

(b) To find the thermal efficiency, first get \( \dot{Q}_{\text{in}} \).

\[
\dot{Q}_{\text{in}} = \dot{m} (h_1 - h_2) = (7.8 \text{ kg/s}) (3545.3 - 183.58) \text{ kJ/kg} = 26221 \text{ kW}
\]

Thus

\[
\eta = \frac{W_{\text{cycle}}}{\dot{Q}_{\text{in}}} = \frac{9426}{26221} = 0.359 (35.9\%)\]

Problem 1 continued

(c) The mass flow rate of cooling water passing through the condenser is

\[ m_{cw} = \frac{m(h_2 - h_3)}{h_{cw,in} - h_{cw, out}} \]

with \( h_{cw} = h_f (CT) \)

\[ m_{cw} = \frac{(7.8 \text{ kg/s}) (2327.1 - 173.88) \text{ kJ/kg}}{(150.86 - 75.58) \text{ kJ/kg}} \]

\[ = 223.1 \text{ kg/s} \]

\( m_{cw} \)
Problem 2

**KNOWN:** Water is the working fluid in an ideal regenerative Rankine cycle with one open feedwater heater. Data at various locations are known.

**FIND:** Determine (a) the rate of heat addition per kg of steam entering the first-stage turbine, (b) the thermal efficiency, and (c) the rate of heat transfer for the condenser per kg of steam entering the first-stage turbine.

**SCHEMATIC & GIVEN DATA:**

**ENGINEERING MODEL:** Same as Example 8.5, except turbine stages and pumps operate in an internally reversible manner.

**ANALYSIS:** First, fix each principal state.

**State 1:** \( p_1 = 100 \text{ bar}, T_1 = 480^\circ \text{C} \Rightarrow h_1 = 3321.4 \text{ kJ/kg}, s_1 = 6.5282 \text{ kJ/kg.K} \)

**State 2:** \( p_2 = 7 \text{ bar}, s_2 = s_1 \Rightarrow x_2 = \frac{s_2 - s_{f1}}{s_{g1} - s_{f1}} = 0.9419, h_2 = 2604.8 \text{ kJ/kg} \)

**State 3:** \( p_3 = 0.06 \text{ bar}, s_3 = s_2 \Rightarrow x_3 = \frac{s_3 - s_{f3}}{s_{g3} - s_{f3}} = 0.7642, h_3 = 2009.8 \text{ kJ/kg} \)

**State 4:** \( p_4 = 0.06 \text{ bar}, \text{sat. liquid} \Rightarrow h_4 = 151.53 \text{ kJ/kg} \)

**State 5:** \( h_5 = h_4 + v_4 (p_5 - p_4) = 151.53 + (1.0064 \times 10^{-3} \text{ m}^3/\text{kg}) (7 - 0.06) \text{ bar} = 151.53 + 0.048 = 152.23 \text{ kJ/kg} \)

**State 6:** \( p_6 = 7 \text{ bar}, \text{sat. liquid} \Rightarrow h_6 = 697.72 \text{ kJ/kg} \)

**State 7:** \( h_7 = h_6 + v_7 (p_7 - p_6) = 697.72 + (1.1050 \times 10^{-3} \text{ m}^3/\text{kg}) (100 - 7) = 707.52 \text{ kJ/kg} \)

(a) For the steam generator
\[
\Delta h_{\text{in}}/m_1 = h_1 - h_7 = (3321.4 - 707.52) = 2613.9 \text{ kJ/kg} \]

(b) Applying mass and energy balances to the control volume enclosing the open feedwater heater
\[
y = \frac{h_6 - h_5}{h_2 - h_5} = \frac{697.72 - 152.23}{2604.8 - 152.23} = 0.2152
\]

For the control volume enclosing the turbine stages
\[
\frac{\Delta h_{\text{in}}}{m_1} = (h_1 - h_2) + (1-y)(h_2 - h_3)
= (3321.4 - 2604.8) + (1-0.2152)(2604.8 - 2009.8) = 1166.3 \text{ kJ/kg}
\]
Problem 2 continued

For the pumps:
\[ \dot{W}_p = \dot{W}_{p1} + \dot{W}_{p2} = \dot{m}_1 [(1-y)(h_5-h_4) + (h_f-h_b)] \]

or
\[ \frac{\dot{W}_p}{\dot{m}_1} = (1-y)(h_5-h_4) + (h_f-h_b) \]
\[ = (1-0.2152)(152.23-151.53)+(707.52-697.22) \]
\[ = 10.85 \text{ kJ/kg} \]

Thus, the net power developed, per unit mass entering the first-stage turbine is
\[ \frac{\dot{W}_{net}}{\dot{m}_1} = \frac{\dot{W}_e}{\dot{m}_1} - \frac{\dot{W}_p}{\dot{m}_1} = 1166.3 - 10.85 = 1155.3 \text{ kJ/kg} \]

And the thermal efficiency is
\[ \eta = \frac{\dot{W}_{net}}{Q_{in}/\dot{m}_1} = \frac{1155.3}{2613.9} = 0.442 \ (44.2\%) \]

(c) For the condenser
\[ \dot{Q}_{out} = \dot{m}_1 (1-y)(h_3-h_4) \]

or
\[ \frac{\dot{Q}_{out}}{\dot{m}_1} = (1-y)(h_3-h_4) = (1-0.2152)(2009.8-151.53) \]
\[ = 1438.4 \text{ kJ/kg} \]
Problem 3

**KNOWN:** Data are given for a vapor-compression heat pump using R-134a as the working fluid. The rate of heat transfer to a dwelling is specified.

**FIND:** Determine, for isentropic compression, (a) the refrigerant mass flow rate, (b) the compressor power, and (c) the coefficient of performance. Repeat for $\eta_c = 0.75$.

**SCHEMATIC & GIVEN DATA:**

**ENGINEERING MODEL:** See Example 10.1 for $\eta_c = 100\%$. See Example 10.3, items 1-4, for $\eta_c = 75\%$.

**ANALYSIS:** First, fix each of the principal states for $\eta_c = 100\%$.

- **State 1** $p_1 = 1.6$ bar, sat. vapor $\Rightarrow h_1 = 237.97$ kJ/kg, $s_1 = 0.9295$ kJ/kg K
- **State 2** $p_2 = 8$ bar, $s_{25} = s_1 \Rightarrow h_{25} = 271.22$ kJ/kg
- **State 3** $p_3 = 8$ bar, sat. liquid $\Rightarrow h_3 = 93.42$ kJ/kg
- **State 4** Throttling process $\Rightarrow h_4 = h_3 = 93.42$ kJ/kg

(a) Using the given value of $Q_{out}$ to get $m$

$$\dot{m} = \frac{Q_{out}}{(h_{25} - h_3)} = \frac{3.5 \text{ kW}}{(271.22 - 93.42) \text{ kJ/kg}} \cdot \frac{1 \text{ kg/s}}{1 \text{ kW}} = 0.197 \text{ kg/s} \quad \frac{\dot{m}}{\eta_c} = 0.197 \text{ kg/s}$$

(b) The compressor power is

$$W_c = \dot{m}(h_{25} - h_1) = (0.197)(271.22 - 237.97) = 6.55 \text{ kW} \quad \frac{W_c}{\eta_c} = 6.55 \text{ kW}$$

(c) The coefficient of performance for the heat pump is

$$\gamma = \frac{Q_{out}}{W_c} = \frac{3.5}{6.55} = 0.534$$

For $\eta_c = 75\%$: $h_2 = h_1 + (h_{25} - h_1)/\eta_c = 237.97 + (271.22 - 237.97)/(0.75) = 282.30$ kJ/kg

$$\dot{m} = \frac{3.5}{(282.30 - 93.42)} = 0.1853 \text{ kg/s}$$

$$W_c = \dot{m}(h_2 - h_1) = (0.1853)(282.30 - 237.97) = 8.214 \text{ kW} \quad \frac{W_c}{\eta_c} = 8.214 \text{ kW}$$

$$\gamma = \frac{3.5}{8.214} = 0.426$$

The presence of irreversibilities in the compressor increases the power input by over 25%. 