Consider a steam power plant operating on an ideal Rankine cycle. Superheated steam enters the turbine with a mass flow rate of 120 kg/s at 16 MPa and 560°C (State 1) and exits at a pressure of 8 kPa (State 2). Assume that saturated liquid water exits the condenser (State 3) and compressed liquid water exits the pump (State 4).

(a) Determine the net power output (kW) of the cycle.
(b) Find the rate of heat transfer (kW) to the steam passing through the boiler.
(c) Calculate thermal efficiency (%) of the cycle.
(d) Show the cycle on T-s diagram. Label states, show temperature values, and indicate appropriate lines of constant pressure.

Consider the same cycle with non-isentropic turbine and pump.

(e) Plot the variation of net power output, rate of heat transfer through the boiler, and cycle thermal efficiency for equal values of isentropic efficiency for turbine and pump ranging from 70% to 100%.

Suppose the cooling water circulating through the condenser enters at 22°C (State 5) and exits at 35°C (State 6) for the ideal cycle. The cooling water leaving the condenser enters a wet cooling tower and is cooled back to 22°C by using ambient air entering the cooling tower at total pressure of 1 atm, dry bulb temperature of 20°C, and relative humidity of 60% (State 7). The air exiting at the top of the cooling tower is saturated at 30°C (State 8). There is make-up water supply at 22°C (State 9).

(f) Calculate the mass flow rate (kg/s) of cooling water.
(g) Find the mass flow rate (kg/s) of air entering the cooling tower.
(h) Determine the mass flow rate (kg/s) required of the make-up water in the cooling tower.

Note:
Please include your EES code, parametric tables, and plots for (e). Assumptions, basic equation(s), system sketch can be either submitted separately or included within the EES code. EES code should contain variable definitions, comments, etc.

**Given:**
Part (a) – (d): A steam power plant operating on an ideal Rankine cycle with a mass flow rate of 120 kg/s
- State 1: Superheated steam at 16 MPa and 560°C
- State 2: 8 kPa
- State 3: Saturated liquid water
State 4: Compressed liquid water

Part (e): Consider the same cycle with non-isentropic turbine and pump.

Part (f) – (h): A cooling tower connecting to the condenser of the steam power plant operating on an ideal Rankine cycle

State 5: Entering to the condenser at 22°C
State 6: Exiting from the condenser at 35°C
State 7: Entering to the cooling tower at total pressure of 1 atm, dry bulb temperature of 20°C, and relative humidity of 60%
State 8:Exiting from the cooling tower at 30°C and relative humidity of 100%
State 9: Make-up water supply at 22°C

Find:

(a) Net power output (kW) of the cycle.
(b) Rate of heat transfer (kW) to the steam passing through the boiler.
(c) Thermal efficiency (%) of the cycle.
(d) T-s diagram.
(e) Variation of net power output, rate of heat transfer through the boiler, and cycle thermal efficiency for isentropic efficiency for turbine and pump ranging from 80% to 100%.
(f) Mass flow rate (kg/s) of cooling water.
(g) Mass flow rate (kg/s) of air entering the cooling tower.
(h) Mass flow rate (kg/s) required of the make-up water in the cooling tower.

System sketch:
Assumptions:
1. Each component on the accompanying sketch is analyzed as a control volume at steady state.
2. Steady, one-dimensional, uniform flow.
3. No work for the boiler and condenser.
4. No heat transfer for the turbine and pump.
5. No work and heat transfer for the cooling tower.
6. Water vapor and dry air are treated as ideal gases.
7. Dalton’s model applicable.
8. Neglect ∆KE and ∆PE.
9. There are no pressure drops for flow through the boiler and condenser.

Basic equations:
\[
\frac{dm_{cv}}{dt} = \sum \dot{m}_i - \sum \dot{m}_e
\]
\[
\frac{dE_{cv}}{dt} = Q_{cv} - W_t + \sum \dot{m}_i (h_i + \frac{v_i^2}{2} + g z_i) - \sum \dot{m}_e (h_e + \frac{v_e^2}{2} + g z_e)
\]

Solution:

State 1: Superheated steam at 16 MPa and 560°C
From the superheated water table, \( h_1 = 3465.4 \frac{kJ}{kg} \) and \( s_1 = 6.5132 \frac{kJ}{kg K} \)

State 2s: 8 kPa and \( s_{2s} = s_1 \)
From the saturate water table, \( \frac{h_{2s} - h_f}{h_g - h_f} = \frac{s_{2s} - s_f}{s_g - s_f} = \frac{h_{2s} - 173.88}{2577 - 173.88} = \frac{6.5132 - 0.5926}{8.2287 - 0.5926} \Rightarrow h_{2s} = 2037.12 \frac{kJ}{kg} \)

State 3: Saturated liquid water at 8 kPa
\( h_3 = 173.88 \frac{kJ}{kg} \) and \( s_3 = 0.5926 \frac{kJ}{kg K} \)

State 4s: Compressed liquid water at 16 MPa and \( s_{4s} = s_3 \)
\( h_{4s} = h_f + v_f (P_{4s} - P_3) = 173.88 \frac{kJ}{kg} + 0.0010084 \frac{m^3}{kg} \ast (160 \ast 10^2 - 8) kPa = 190.01 \frac{kJ}{kg} \)

(a)
Mass balance for the turbine:
\( \dot{m}_1 = \dot{m}_2 = \dot{m}_{steam} \)

Energy balance for the turbine:
\[ \frac{dE_{el}}{dt} = Q_{cv} - \dot{W}_t + \dot{m}_{steam} h_1 - \dot{m}_{steam} h_{2s} \]
\[ \dot{W}_t = \dot{m}_{steam}(h_1 - h_{2s}) = 120 \frac{kg}{s} \times (3465.4 - 2037.12) \frac{kJ}{kg} = 171,394 \frac{kJ}{s} \]

Mass balance for the pump:
\[ \dot{m}_3 = \dot{m}_4 = \dot{m}_{water} = \dot{m}_{steam} \]

Energy balance for the pump:
\[ \frac{dE_{cw}}{dt} = q_{cw} - \dot{W}_p + \dot{m}_{water} h_3 - \dot{m}_{water} h_{4s} \]
\[ \dot{W}_p = \dot{m}_{water}(h_3 - h_{4s}) = 120 \frac{kg}{s} \times (173.88 - 190.01) \frac{kJ}{kg} = -1935.6 \frac{kJ}{s} \]

Net power output of the cycle can be determined by
\[ \dot{W}_{net} = \dot{W}_t + \dot{W}_p = 171,394 \frac{kJ}{s} - 1935.6 \frac{kJ}{s} = 169,458 \frac{kJ}{s} \]

(b)
Mass balance for the boiler:
\[ \dot{m}_4 = \dot{m}_1 = \dot{m}_{steam} \]

Energy balance for the boiler:
\[ \frac{dE_{cw}}{dt} = q_{boiler} - \dot{W}_p + \dot{m}_{steam} h_{4s} - \dot{m}_{steam} h_1 \]
\[ q_{boiler} = \dot{m}_{steam} \times (h_1 - h_{4s}) = 120 \frac{kg}{s} \times (3465.4 - 190.01) \frac{kJ}{kg} = 393,047 \frac{kJ}{s} \]

(c)
Thermal efficiency of the cycle is
\[ \eta = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{169,458 \frac{kJ}{s}}{393,047 \frac{kJ}{s}} = 0.431 \text{ or } 43.1\% \]

(d)
T-s diagram
(e) See the attached EES code.

(f) Mass balance for the condenser:
\[ m_2 = m_3 = m_{\text{steam}} \]
\[ m_5 = m_6 = m_{\text{cooling water}} \]

Energy balance for the condenser:
\[ \frac{dE_{\text{cv}}}{dt} = c_{\text{cv}} \dot{W}_{\text{cv}} + \dot{m}_{\text{steam}} h_{2s} - \dot{m}_{\text{steam}} h_3 + \dot{m}_{\text{cooling water}} h_5 - \dot{m}_{\text{cooling water}} h_6 \]
\[ m_{\text{cooling water}} = \frac{\dot{m}_{\text{steam}} (h_{2s} - h_3)}{h_6 - h_5} = \frac{\dot{m}_{\text{steam}} (h_{2s} - h_3)}{h_f(T_6) - h_f(T_5)} = \frac{\frac{120 \text{kg}}{s} (2037.12 - 173.88)}{146.68 - 92.33} \frac{\text{kJ}}{\text{kg}} \]
\[ = 4113.9 \text{ kg/s} \]

(g) Mass balance for the cooling tower:
Dry air: \[ \dot{m}_{a, 7} = \dot{m}_{a, 8} = \dot{m}_a \]
Water: \[ \dot{m}_6 + \dot{m}_9 + \dot{m}_{v, 7} = \dot{m}_{5} + \dot{m}_{v, 8} \]
\[ \dot{m}_9 = \dot{m}_{v, 8} - \dot{m}_{v, 7} = w_8 \dot{m}_{a, 8} - w_7 \dot{m}_{a, 7} \]
\[ = \dot{m}_a (w_8 - w_7) \]

Energy balance for the cooling tower:
\[ \frac{dE_{\text{cv}}}{dt} = \dot{Q}_{\text{cv}} - \dot{W}_{\text{cv}} + \dot{m}_a h_7 - \dot{m}_a h_8 + \dot{m}_{\text{cooling water}} h_6 - \dot{m}_{\text{cooling water}} h_5 + \dot{m}_9 h_9 \]

Combine the two equations above, then we can have
\[ \dot{m}_a = \frac{m_{\text{cooling water}} (h_f(T_6) - h_f(T_5))}{(h_8 - h_7) - (w_8 - w_7) h_f(T_5)} \]

Find properties from the psychrometric chart for 1 atm.
State 7: Dry bulb temperature of 20°C and relative humidity of 60%
\[ h_7 = 42 \frac{\text{kJ}}{\text{kg dry air}} \text{ and } w_7 = 0.009 \frac{\text{kg water}}{\text{kg dry air}} \]
State 8: Dry bulb temperature of 30°C and relative humidity of 100%
\[ h_8 = 100 \frac{\text{kJ}}{\text{kg dry air}} \text{ and } w_8 = 0.0275 \frac{\text{kg water}}{\text{kg dry air}} \]

Plug into the equation for \( \dot{m}_a \).
We found \( \dot{m}_9 = \dot{m}_a (w_8 - w_7) \) from part (g).

\[
\dot{m}_{\text{makeupwater}} = \dot{m}_9 = \dot{m}_a (w_8 - w_7) = 3972 \frac{kg \text{ dry air}}{s} \times (0.0275 - 0.009) \frac{kg \text{ water}}{kg \text{ dry air}}
= 73.5 \frac{kg}{s}
\]
Given: Rankin vapor power cycle with steam as working fluid

Find: Variation of net power output, rate of heat transfer in the boiler, and cycle thermal efficiency for equal turbine and pump isentropic efficiency from 70% to 100%

State 1: P1, T1
P1=160 [bar]
T1=560 [C]
h1=enthalpy(Steam, P=P1, T=T1)
s1=entropy(Steam, P=P1, T=T1)

State 2s: P2S, s2s=s1
s2s=s1
P2s=0.08 [bar]
h2s=enthalpy(Steam, P=P2s, s=s2s)
T2s=temperature(Steam, P=P2s, s=s2s)

State 2: P2=P2s, turbine efficiency
\[ \eta_{turbine} = \frac{h1-h2}{h1-h2s} \]

State 3: P3, saturated liquid
P3=0.08 [bar]
x3=0
h3=enthalpy(Steam, P=P3, x=x3)
s3=entropy(Steam, P=P3, x=x3)

State 4s: P4s, s4s=s3
s4s=s3
P4s=160 [bar]
h4s=enthalpy(Steam, P=P4s, s=s4s)
T4s=temperature(Steam, P=P4s, s=s4s)

(Turbine Power Output)
Wdot_turbine=mdot*(h1-h2)

(Pump Power Input)
Wdot_pump=mdot*(h4-h3)

(Boiler Heat Transfer Rate)
Qdot_boiler=mdot*(h1-h4)

(Termal Efficiency)
\[ \eta_{thermal} = \frac{Wdot_{net}/Qdot_{boiler}}{100} \]

(Mass Flow Rate)
mdot=120 [kg/s]

Parametric Table: Table 1

<table>
<thead>
<tr>
<th>Run</th>
<th>( \eta_{pump} )</th>
<th>( \eta_{turbine} )</th>
<th>Wdot_{net}</th>
<th>Qdot_{boiler}</th>
<th>( \eta_{thermal} )</th>
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<td>393205</td>
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<td>1</td>
<td>1</td>
<td>169593</td>
<td>393271</td>
<td>43.12</td>
</tr>
</tbody>
</table>
Isentropic pump/turbine efficiency

Net power output (kW)

Boiler heat transfer rate (kW)
Isentropic pump/turbine efficiency

Thermal efficiency (%)
Consider a reheat cycle steam power plant. Superheated steam enters the high-pressure turbine with a mass flow rate of 120 kg/s at 16 MPa and 560°C (State 1) and exits at a pressure of 4 MPa (State 2). The isentropic efficiency of the high-pressure turbine is 85%. Steam leaving the high-pressure turbine is routed back to the boiler and heated to 4 MPa and 560°C (State 3). Steam then expands to 8 kPa (State 4) in the low-pressure turbine which has an isentropic efficiency of 85%. Assume that saturated liquid water exits the condenser (State 5) and compressed liquid water exits the pump (State 6). The pump has an isentropic efficiency of 85%.

(a) Determine the net power output (kW) of the cycle.
(b) Find the rate of heat transfer (kW) to the steam passing through the boiler.
(c) Calculate thermal efficiency (%) of the cycle.
(d) Show the cycle on T-s diagram. Label states, show temperature values, and indicate appropriate lines of constant pressure.
(e) Plot the variation of net power output and cycle thermal efficiency reheat pressure values ranging from 5 bar to 120 bar.

Note:
Please include your EES code, parametric tables, and plots for (e). Assumptions, basic equation(s), system sketch can be either submitted separately or included within the EES code. EES code should contain variable definitions, comments, etc.

**System sketch:**
Given:
See the attached EES code.

Find:
See the attached EES code.

Assumption:
See the attached EES code.

Solution:
(a), (b), (c), and (e): See the attached EES code.

(d)
T-s diagram
{Given: Steam Power Cycle With Reheat}

{Find:}
(a) Net power output (kW) of the cycle
(b) Rate of heat transfer (kW) in boiler
(c) Thermal efficiency (%) of the cycle
(d) T-s diagram
(e) Variation of net power output and thermal efficiency for reheat pressures ranging from 5 bar to 120 bar

{Assumptions:}
Each component on the accompanying sketch is analyzed as a control volume at steady state.
Steady uniform 1-D flow
Neglect KE and PE changes
Ignore pressure drop in boiler, condenser, and reheater
No work for boiler, condenser, open feedwater heater, closed feedwater heater, throttling valve
No heat transfer for turbines, pumps, and throttling valve

{State 1: Boiler Exit = HP Turbine Inet}
P1=160 [bar]
T1=560 [C]
h1=enthalpy(Steam, P=P1, T=T1)
s1=entropy(Steam, P=P1, T=T1)
{State 2s: HP Turbine Exit = Reheat Inet}
s2s=s1
P2s=P3
h2s=enthalpy(Steam, P=P2s, s=s2s)
T2s=temperature(Steam, P=P2s, s=s2s)
{State 2: HP Turbine Isentropic Efficiency}
eta_hpturbine=0.85
eta_hpturbine=(h1-h2)/(h1-h2s)
{State 3: Reheat Exit = LP Turbine Inet}
T3=560 [C]
h3=enthalpy(Steam, P=P3, T=T3)
s3=entropy(Steam, P=P3, T=T3)
{State 4s: LP Turbine Exit = Condenser Inlet}
s4s=s3
P4s=0.08 [bar]
h4s=enthalpy(Steam, P=P4s, s=s4s)
T4s=temperature(Steam, P=P4s, s=s4s)
{State 4: LP Turbine Isentropic Efficiency}
eta_lpturbine=0.85
eta_lpturbine=(h3-h4)/(h3-h4s)
{State 5: Condenser Exit = Pump Inlet}
P5=P4s
x5=0
h5=enthalpy(Steam, P=P5, x=x5)
s5=entropy(Steam, P=P5, x=x5)
{State 6s: Pump Exit = Boiler Inet}
s6s=s5
P6s=P1
h6s=enthalpy(Steam, P=P6s, s=s6s)
T6s=temperature(Steam, P=P6s, s=s6s)
{State 4: Pump Isentropic Efficiency}
eta_pump=0.85
\[
\eta_{pump} = \frac{(h_6s-h_5)}{(h_6-h_5)}
\]

\{Mass Flow Rate\}

\[
\dot{m}_{dot} = 120 \text{ [kg/s]}
\]

\{Turbine Power Output\}

\[
W_{dot \text{ turbine}} = \dot{m}_{dot} \times ((h_1-h_2) + (h_3-h_4))
\]

\{Pump Power Input\}

\[
W_{dot \text{ pump}} = \dot{m}_{dot} \times (h_6-h_5)
\]

\{Net Power Input\}

\[
W_{dot \text{ net}} = W_{dot \text{ turbine}} - W_{dot \text{ pump}}
\]

\{Boiler Heat Transfer Rate\}

\[
Q_{dot \text{ boiler}} = \dot{m}_{dot} \times ((h_1-h_6) + (h_3-h_2))
\]

\{Thermal Efficiency\}

\[
\eta_{thermal} = \frac{W_{dot \text{ net}}}{Q_{dot \text{ boiler}}} \times 100
\]

### Parametric Table: Table 1

<table>
<thead>
<tr>
<th>P3</th>
<th>Qdot(_{\text{boiler}})</th>
<th>Wdot(_{\text{net}})</th>
<th>(\eta_{\text{thermal}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Run 1</td>
<td>5</td>
<td>497149</td>
<td>189404</td>
</tr>
<tr>
<td>Run 2</td>
<td>10</td>
<td>483644</td>
<td>186379</td>
</tr>
<tr>
<td>Run 3</td>
<td>20</td>
<td>468192</td>
<td>181611</td>
</tr>
<tr>
<td>Run 4</td>
<td>30</td>
<td>457619</td>
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</tr>
<tr>
<td>Run 5</td>
<td>40</td>
<td>449250</td>
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<tr>
<td>Run 6</td>
<td>50</td>
<td>442773</td>
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<tr>
<td>Run 7</td>
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<td>Run 8</td>
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<td>Run 9</td>
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</tr>
<tr>
<td>Run 10</td>
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<tr>
<td>Run 13</td>
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</tbody>
</table>

Answers of (a), (b), and (c)
Consider a regenerative steam power plant with one open feedwater heater and one closed feedwater heater. Superheated steam enters the turbine with a mass flow rate of 120 kg/s at 16 MPa and 560°C (State 1). Some fraction of the steam is extracted at 40 bar (State 2) and is supplied to the closed feedwater heater. The remaining steam expands to a pressure of 3 bar (State 3), another fraction is extracted at this pressure, and is supplied to the open feedwater heater. Steam then expands to the condenser pressure of 8 kPa (State 4). The isentropic efficiency of the turbine is 85%. Assume that saturated liquid water exits the condenser (State 5) and compressed liquid water exits the low-pressure pump (State 6). The low-pressure pump has an isentropic efficiency of 85%. Saturated liquid at 3 bar (State 7) exits the open feedwater heater, is compressed to 160 bar (State 8) in the high-pressure pump, and flows into the closed feedwater heater. The high-pressure pump has an isentropic efficiency of 85%. Compressed liquid leaves the closed feedwater heater at 160 bar with a temperature equal to the saturation temperature at 40 bar (State 9) and enters the boiler. Saturated liquid condensate at 40 bar (State 10) drains from the closed feedwater heater and is throttled to 3 bar (State 11) into the open feedwater heater.

(a) Determine the net power output (kW) of the cycle.
(b) Find the rate of heat transfer (kW) to the steam passing through the boiler.
(c) Calculate thermal efficiency (%) of the cycle.
(d) Compare the thermal efficiency of this cycle with the value calculated in SP-23(c) and comment about regeneration.

**Given:**
A regenerative steam power plant with one open feedwater heater and one closed feedwater heater.

States and given properties are as shown in the system sketch.

**Find:**
(a) Net power output (kW) of the cycle.
(b) Rate of heat transfer (kW) to the steam passing through the boiler.
(c) Thermal efficiency (%) of the cycle.
(d) Compare the thermal efficiency of this cycle with the value calculated in SP-23(c) and comment about regeneration.

**System sketch:**

**Assumptions:**
1. Each component on the accompanying sketch is analyzed as a control volume at steady state.
2. Steady, one-dimensional, uniform flow
3. No work for the boiler and condenser.
4. No heat transfer for the turbine and pumps.
5. Neglect ΔKE and ΔPE.
6. There are no pressure drops for flow through the boiler, heaters and condenser.

**Basic equations:**

\[
\frac{dm_{cv}}{dt} = \sum m_i - \sum m_e \\
\frac{dE_{cv}}{dt} = Q_{cv} - W_{cv} + \sum_i m_i \left( h_i + \frac{v_i^2}{2} + gz_i \right) - \sum_e m_e (h_e + \frac{v_e^2}{2} + gz_e) \\
\]

**Solution:**

State 1: Superheated steam at \( P_1 = 16 \) MPa and \( T_1 = 560^\circ\text{C} \)

From the superheated water table, \( h_1 = 3465.4 \text{ kJ/kg} \) and \( s_1 = 6.5132 \text{ kJ/kgK} \)

State 2s: \( P_{2s} = 4 \) MPa and \( s_{2s} = s_1 \)

From the superheated water table, \( h_{2s}=3015.4 \text{ kJ/kg} \) → \( h_{2s} = 3050.9 \text{ kJ/kg} \)

State 2: \( P_2 = P_{2s} \)

Using the first-stage turbine efficiency, \( \eta_{t1} = 0.85 = \frac{h_1-h_2}{h_1-h_{2s}} = \frac{3465.4-h_2}{3465.4-3050.9} \) → \( h_2 = 3113.1 \text{ kJ/kg} \)

From the superheated water table, \( \frac{3113.1-3015.4}{3117.2-3015.4} = \frac{s_2-6.4553}{6.6215-6.4553} \) → \( s_2 = 6.6148 \text{ kJ/kg} \)

State 3s: \( P_{3s} = 0.3 \) MPa and \( s_{3s} = s_2 \)

From the saturate water table, \( \frac{h_{3s}-h_f}{h_g-h_f} = \frac{s_{3s}-s_f}{s_g-s_f} = \frac{h_{3s}-561.47}{2725.3-561.47} = \frac{s_{3s}-1.6718}{6.6215-1.6718} \) → \( h_{3s} = 2517.9 \text{ kJ/kg} \)

State 3: \( P_3 = P_{3s} \)

Using the second-stage turbine efficiency, \( \eta_{t2} = 0.85 = \frac{h_2-h_3}{h_2-h_{3s}} = \frac{3113.1-h_3}{3113.1-2517.9} \) → \( h_3 = 2653.1 \text{ kJ/kg} \)

From the saturate water table, \( \frac{h_3-h_f}{h_g-h_f} = \frac{s_3-s_f}{s_g-s_f} = \frac{2653.1-561.47}{2725.3-561.47} = \frac{s_3-1.6718}{6.6215-1.6718} \) → \( s_3 = 6.8144 \text{ kJ/kg} \)

State 4s: \( P_{4s} = 8 \) kPa and \( s_{4s} = s_3 \)

From the saturate water table, \( \frac{h_{4s}-h_f}{h_g-h_f} = \frac{s_{4s}-s_f}{s_g-s_f} = \frac{h_{4s}-173.88}{2577-173.88} = \frac{s_{4s}-1.6718}{8.2287-1.6718} \) → \( h_{4s} = 2131.9 \text{ kJ/kg} \)

State 3: \( P_4 = P_{4s} \)

Using the third-stage turbine efficiency, \( \eta_{t3} = 0.85 = \frac{h_3-h_4}{h_3-h_{4s}} = \frac{2653.1-h_4}{2653.1-2131.9} \) → \( h_4 = 2210.1 \text{ kJ/kg} \)

State 5: Saturated liquid water at 8 kPa
\[ h_5 = 173.88 \frac{kJ}{kg} \]

State 6: \( P_{6s} = 0.3 \) MPa
\[ h_{6s} \approx h_5 + v_5 (P_{6s} - P_6) = 173.88 \frac{kJ}{kg} + 0.0010084 \frac{m^3}{kg} \times (3 \times 10^2 - 8) \text{ kPa} = 174.17 \frac{kJ}{kg} \]

State 6: \( P_6 = P_{6s} \)
Using the pump efficiency, \( \eta_{p1} = 0.85 = \frac{h_{6s} - h_5}{h_6 - h_5} = \frac{174.17 - 173.88}{h_6 - 173.88} \Rightarrow h_6 = 174.22 \frac{kJ}{kg} \)

State 7: Saturated liquid water and 0.3 MPa
\[ h_7 = 561.47 \frac{kJ}{kg} \]

State 8: \( P_{8s} = 16 \) MPa
\[ h_{8s} \approx h_7 + v_7 (P_{8s} - P_7) = 561.47 \frac{kJ}{kg} + 0.0010732 \frac{m^3}{kg} \times (16 \times 10^3 - 3 \times 10^2) \text{ kPa} \]
\[ = 578.32 \frac{kJ}{kg} \]

State 8: \( P_8 = P_{8s} \)
Using the pump efficiency, \( \eta_{p2} = 0.85 = \frac{h_{8s} - h_7}{h_8 - h_7} = \frac{578.32 - 561.47}{h_8 - 561.47} \Rightarrow h_8 = 581.29 \frac{kJ}{kg} \]

State 9: Compressed liquid water at \( T_{sat} @ 4 \) MPa and \( P_9 = 16 \) MPa
\[ h_9 \approx h_f(T_{sat}) + v_f(T_{sat}) \times (P_9 - P_{sat}) \]
\[ = 1087.3 \frac{kJ}{kg} + 0.0012522 \frac{m^3}{kg} \times (16 \times 10^3 - 4 \times 10^3) \text{ kPa} \]
\[ = 1102.33 \frac{kJ}{kg} \]

State 10: Saturated liquid water at 4 MPa
\[ h_{10} = 1087.3 \frac{kJ}{kg} \]

State 11: Throttling process
\[ h_{11} = h_{10} \]
(a)
Mass balance for the turbine:
\[ m_1 = m_2 + m_3 + m_4 \]
Divided by \( m_1 \)
\[ 1 = y + y' + (1 - y - y') \] as shown in the system sketch

Mass balance for the closed feedwater heater:
\[ m_2 + m_8 = m_9 + m_{10} \]
Divided by \( m_9 \) (= \( m_8 = m_1 \))
\[ y + 1 = 1 + y \]

Energy balance for the closed feedwater heater:
\[ \frac{dE_{ce}}{dt} = Q_{ce} - W_t + m_2 h_2 + m_8 h_8 - m_9 h_9 - m_{10} h_{10} \]
\[ 0 = m_2 h_2 + m_8 h_8 - m_9 h_9 - m_{10} h_{10} \]
Divided by \( m_9 \)
\[ 0 = y h_2 + h_8 - h_9 - y h_{10} \]
\[ y = \frac{h_9 - h_8}{h_2 - h_{10}} = \frac{1102.33 - 581.29}{3113.1 - 1087.3} = 0.257 \]

Mass balance for the open feedwater heater:
\[ m_3 + m_6 + m_{11} = m_7 \]
Divided by \( m_1 \)
\[ y' + (1 - y - y') + y = 1 \]

Energy balance for the open feedwater heater:
\[ \frac{dE_{ce}}{dt} = Q_{ce} - W_t + m_2 h_2 + m_3 h_3 + m_{11} h_{11} + m_6 h_6 - m_7 h_7 \]
\[ 0 = m_3 h_3 + m_{11} h_{11} + m_6 h_6 - m_7 h_7 \]
Divided by \( m_1 \)
\[ 0 = y' h_3 + (1 - y - y') h_6 + y h_{11} - h_7 \]
\[ y' = \frac{h_7 - h_6 + y(6-h_{11})}{h_3 - h_6} = \frac{561.47 - 174.22 + 0.257(174.22 - 1087.3)}{2653.1 - 174.22} = 0.062 \]

Energy balance for the turbine:
\[ \frac{dE_{ce}}{dt} = Q_{ce} - W_t + m_1 h_1 - m_2 h_2 - m_3 h_3 - m_4 h_4 \]
\[ W_t = m_1 h_1 - m_2 h_2 - m_3 h_3 - m_4 h_4 = m_1 \times [h_1 - y h_2 - y' h_3 - (1 - y - y') h_4] \]
\[ = 120 \times \frac{k_g}{s} [3465.4 - 0.257 \times 3113.1 - 0.062 \times 2653.1 - (1 - 0.257 - 0.062) \times 2210.1] \]
= 119,492 kW

Mass balance for the pump 1:
\[ \dot{m}_5 = \dot{m}_6 = \dot{m}_1 (1 - y - y') \]

Energy balance for the pump 1:
\[
\frac{dE}{dt} = \dot{Q}_{cv} - \dot{W}_{p1} + \dot{m}_5 h_5 - \dot{m}_6 h_6
\]
\[
\dot{W}_{p1} = \dot{m}_1 * (1 - y - y') * (h_5 - h_6) = 120 \frac{kg}{s} * (1 - 0.257 - 0.062) * (173.88 - 174.22)
\]
\[ = -27.8 \text{ kW} \]

Mass balance for the pump 2:
\[ \dot{m}_7 = \dot{m}_8 = \dot{m}_1 \]

Energy balance for the pump 2:
\[
\frac{dE}{dt} = \dot{Q}_{cv} - \dot{W}_{p2} + \dot{m}_7 h_7 - \dot{m}_8 h_8
\]
\[
\dot{W}_{p2} = \dot{m}_1 * (h_7 - h_8) = 120 \frac{kg}{s} * (561.47 - 581.29)
\]
\[ = -2378.4 \text{ kW} \]

\[ \dot{W}_{net} = \dot{W}_t + \dot{W}_{p1} + \dot{W}_{p2} = 117,086 \text{ kW} \quad \text{or} \quad 1.17 \times 10^5 \text{ kW} \]

(b)

Mass balance for the boiler:
\[ \dot{m}_9 = \dot{m}_1 \]

Energy balance for the boiler:
\[
\frac{dE}{dt} = \dot{Q}_{boiler} - \dot{W}_{cv} + \dot{m}_9 h_9 - \dot{m}_1 h_1
\]
\[
\dot{Q}_{boiler} = \dot{m}_1 * (h_1 - h_9) = 120 \frac{kg}{s} * (3465.4 - 1102.33)
\]
\[ = 283,568 \text{ kW} \quad \text{or} \quad 2.83 \times 10^5 \text{ kW} \]

(c)
\[
\eta = \frac{\dot{W}_{net}}{\dot{Q}_{boiler}} = \frac{1.17 \times 10^5 \text{ kW}}{2.83 \times 10^5 \text{ kW}} = 0.413 \quad \text{or} \quad 41.3\%
\]

(d)
Thermal efficiency of the ideal Rankin cycle from SP-23 (C) is
\[ \eta = 43\% \]

Thermal efficiency of the cycle with the isentropic efficiency 0.85 from SP-23 (d) is
\[ \eta = 36\% \]

Thermal efficiency of the cycle with the reheat and the isentropic efficiency 0.85 from SP-24 (c) is
\[ \eta = 38\% \]

Thermal efficiency of the cycle with the regeneration, the reheat, and the isentropic efficiency 0.85 is
\[ \eta = 41\% \]

Thus, the regeneration and reheat improve thermal efficiency of the cycle though the isentropic process drastically decreases the thermal efficiency.