Chapter 10
VAPOR AND COMBINED POWER CYCLES

Carnot Vapor Cycle

10-1C Because excessive moisture in steam causes erosion on the turbine blades. The highest moisture content allowed is about 10%.

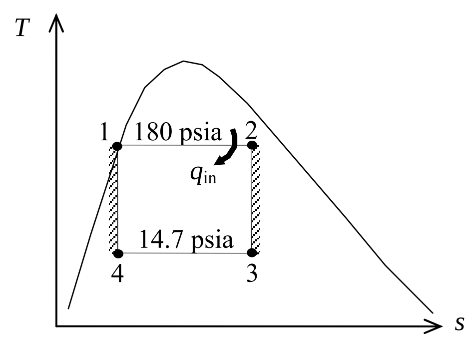
10-2C The Carnot cycle is not a realistic model for steam power plants because (1) limiting the heat transfer processes to two-phase systems to maintain isothermal conditions severely limits the maximum temperature that can be used in the cycle, (2) the turbine will have to handle steam with a high moisture content which causes erosion, and (3) it is not practical to design a compressor that will handle two phases.

10-3E A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the quality at the end of the heat rejection process, 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*) We note that

$$T_H = T_{\text{sat@180 psia}} = 373.1°\text{F} = 833.1 \text{ R}$$
$$T_L = T_{\text{sat@14.7psia}} = 212.0°\text{F} = 672.0 \text{ R}$$



and

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{672.0 \text{ R}}{833.1 \text{ R}} = \mathbf{19.3\%}$$

(*b*) Noting that $s_4 = s_1 = s_{f @ 180 \text{ psia}} = 0.53274$ Btu/lbm·R,

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{0.53274 - 0.31215}{1.44441} = \mathbf{0.153}$$

(*c*) The enthalpies before and after the heat addition process are

$$h_1 = h_{f @ 180 \text{ psia}} = 346.14 \text{ Btu/lbm}$$
$$h_2 = h_f + x_2 h_{fg} = 346.14 + (0.90)(851.16) = 1112.2 \text{ Btu/lbm}$$

Thus,

$$q_{\text{in}} = h_2 - h_1 = 1112.2 - 346.14 = 766.0 \text{ Btu/lbm}$$

and,

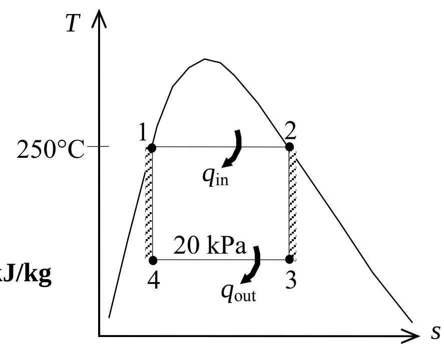
$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.1934)(766.0 \text{ Btu/lbm}) = \mathbf{148.1 \text{ Btu/lbm}}$$

10-4 A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the amount of heat rejected, 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*) Noting that T_H = 250°C = 523 K and $T_L = T_{\text{sat @ 20 kPa}}$ = 60.06°C = 333.1 K, the thermal efficiency becomes

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{333.1\ \text{K}}{523\ \text{K}} = 0.3632 = \mathbf{36.3\%}$$



(*b*) The heat supplied during this cycle is simply the enthalpy of vaporization,

$$q_{\text{in}} = h_{fg@\,250^\circ C} = 1715.3\ \text{kJ/kg}$$

Thus,

$$q_{\text{out}} = q_L = \frac{T_L}{T_H} q_{\text{in}} = \left(\frac{333.1\ \text{K}}{523\ \text{K}}\right)(1715.3\ \text{kJ/kg}) = \mathbf{1092.3\ kJ/kg}$$

(*c*) The net work output of this cycle is

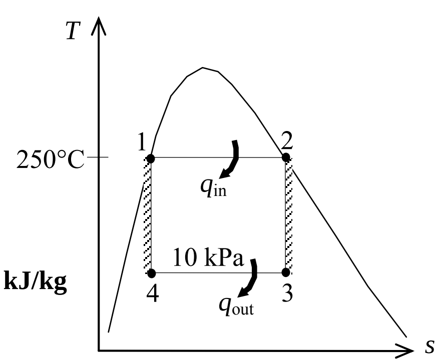
$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.3632)(1715.3\ \text{kJ/kg}) = \mathbf{623.0\ kJ/kg}$$

10-5 A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the amount of heat rejected, 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*) Noting that T_H = 250°C = 523 K and $T_L = T_{\text{sat @ 10 kPa}}$ = 45.81°C = 318.8 K, the thermal efficiency becomes

$$\eta_{\text{th, C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{318.8\ \text{K}}{523\ \text{K}} = \mathbf{39.04\%}$$



(*b*) The heat supplied during this cycle is simply the enthalpy of vaporization,

$$q_{\text{in}} = h_{fg@\,250°\text{C}} = 1715.3\ \text{kJ/kg}$$

Thus,

$$q_{\text{out}} = q_L = \frac{T_L}{T_H} q_{\text{in}} = \left(\frac{318.8\ \text{K}}{523\ \text{K}}\right)(1715.3\ \text{kJ/kg}) = \mathbf{1045.6\ kJ/kg}$$

(*c*) The net work output of this cycle is

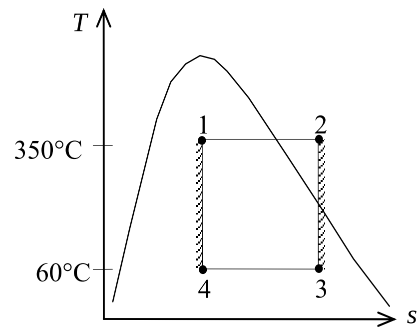
$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.3904)(1715.3\ \text{kJ/kg}) = \mathbf{669.7\ kJ/kg}$$

10-6 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

$$s_2 = s_3 = s_f + x_3 s_{fg}$$
$$= 0.8313 + 0.891 \times 7.0769 = 7.1368 \text{ kJ/kg·K}$$

Thus,

$$\left.\begin{array}{l} T_2 = 350°\text{C} \\ s_2 = 7.1368 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} P_2 \cong \mathbf{1.40\ MPa} \text{ (Table A-6)}$$

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

$$s_4 = s_f + x_4 s_{fg} = 0.8313 + (0.1)(7.0769) = 1.5390 \text{ kJ/kg} \cdot \text{K}$$

Thus,

$$w_{\text{net}} = \text{Area} = (T_H - T_L)(s_3 - s_4) = (350 - 60)(7.1368 - 1.5390) = \mathbf{1623\ kJ/kg}$$

The Simple Rankine Cycle

10-7C The four processes that make up the simple ideal cycle are (1) Isentropic compression in a pump, (2) P = constant heat addition in a boiler, (3) Isentropic expansion in a turbine, and (4) P = constant heat rejection in a condenser.

10-8C Heat rejected decreases; everything else increases.

10-9C Heat rejected decreases; everything else increases.

10-10C The pump work remains the same, the moisture content decreases, everything else increases.

10-11C The actual vapor power cycles differ from the idealized ones in that the actual cycles involve friction and pressure drops in various components and the piping, and heat loss to the surrounding medium from these components and piping.

10-12C The boiler exit pressure will be (a) lower than the boiler inlet pressure in actual cycles, and (b) the same as the boiler inlet pressure in ideal cycles.

10-13C We would reject this proposal because $w_{turb} = h_1 - h_2 - q_{out}$, and any heat loss from the steam will adversely affect the turbine work output.

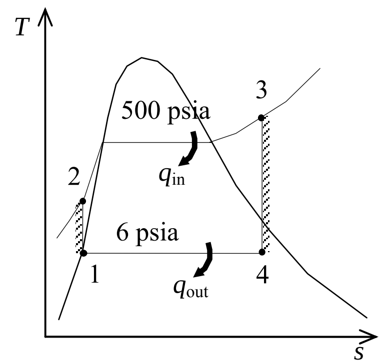
10-14C Yes, because the saturation temperature of steam at 10 kPa is 45.81°C, which is much higher than the temperature of the cooling water.

10-15E A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The rates of heat addition and rejection, 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 From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f\,@\,6\text{ psia}} = 138.02\text{ Btu/lbm}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,6\text{ psia}} = 0.01645\text{ ft}^3\text{/lbm}$$



$$\begin{aligned} w_{p,in} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01645\text{ ft}^3\text{/lbm})(500-6)\text{psia}\left(\frac{1\text{ Btu}}{5.404\text{ psia}\cdot\text{ft}^3}\right) \\ &= 1.50\text{ Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{p,in} = 138.02 + 1.50 = 139.52\text{ Btu/lbm}$$

$$\left.\begin{aligned} P_3 &= 500\text{ psia} \\ T_3 &= 1200°\text{F} \end{aligned}\right\} \begin{aligned} h_3 &= 1630.0\text{ Btu/lbm} \\ s_3 &= 1.8075\text{ Btu/lbm}\cdot\text{R} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 6\text{ psia} \\ s_4 &= s_3 \end{aligned}\right\} \begin{aligned} x_4 &= \frac{s_4 - s_f}{s_{fg}} = \frac{1.8075 - 0.24739}{1.58155} = 0.9864 \\ h_4 &= h_f + x_4 h_{fg} = 138.02 + (0.9864)(995.88) = 1120.4\text{ Btu/lbm} \end{aligned}$$

Knowing the power output from the turbine the mass flow rate of steam in the cycle is determined from

$$\dot{W}_{T,out} = \dot{m}(h_3 - h_4) \longrightarrow \dot{m} = \frac{\dot{W}_{T,out}}{h_3 - h_4} = \frac{500\text{ kJ/s}}{(1630.0 - 1120.4)\text{Btu/lbm}}\left(\frac{0.94782\text{ Btu}}{1\text{ kJ}}\right) = 0.9300\text{ lbm/s}$$

The rates of heat addition and rejection are

$$\dot{Q}_{in} = \dot{m}(h_3 - h_2) = (0.9300\text{ lbm/s})(1630.0 - 139.52)\text{Btu/lbm} = \mathbf{1386\ Btu/s}$$
$$\dot{Q}_{out} = \dot{m}(h_4 - h_1) = (0.9300\text{ lbm/s})(1120.4 - 138.02)\text{Btu/lbm} = \mathbf{913.6\ Btu/s}$$

and the thermal efficiency of the cycle is

$$\eta_{th} = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}} = 1 - \frac{913.6}{1386} = \mathbf{0.341}$$

10-16 A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The maximum thermal efficiency of the cycle for a given quality at the turbine exit is to be determined.

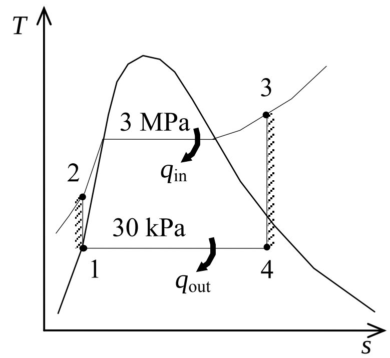
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis For maximum thermal efficiency, the quality at state 4 would be at its minimum of 85% (most closely approaches the Carnot cycle), and the properties at state 4 would be (Table A-5)

$$\left.\begin{array}{l} P_4 = 30\text{ kPa} \\ x_4 = 0.85 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 289.27 + (0.85)(2335.3) = 2274.3\text{ kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 0.9441 + (0.85)(6.8234) = 6.7440\text{ kJ/kg}\cdot\text{K} \end{array}$$

Since the expansion in the turbine is isentropic,

$$\left.\begin{array}{l} P_3 = 3000\text{ kPa} \\ s_3 = s_4 = 6.7440\text{ kJ/kg}\cdot\text{K} \end{array}\right\} h_3 = 3115.5\text{ kJ/kg}$$



Other properties are obtained as follows (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,30\text{ kPa}} = 289.27\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,30\text{ kPa}} = 0.001022\text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001022\text{ m}^3\text{/kg})(3000-30)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 3.04\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 289.27 + 3.04 = 292.31\text{ kJ/kg}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3115.5 - 292.31 = 2823.2\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2274.3 - 289.27 = 1985.0\text{ kJ/kg}$$

and the thermal efficiency of the cycle is

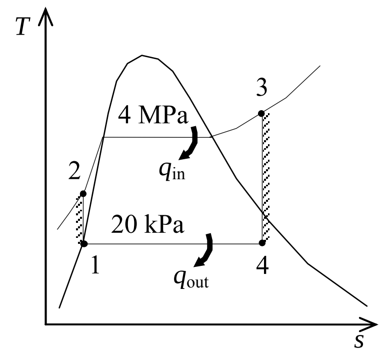
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1985.0}{2823.2} = \mathbf{0.297}$$

10-17 A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The power produced by the turbine and consumed by the pump are to be determined.

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

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

$$h_1 = h_{f@\,20\,\text{kPa}} = 251.42\ \text{kJ/kg}$$
$$v_1 = v_{f@\,20\,\text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{\text{p,in}} &= v_1(P_2 - P_1) \\ &= (0.001017\ \text{m}^3/\text{kg})(4000-20)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 4.05\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 4.05 = 255.47\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 4000\ \text{kPa} \\ T_3 = 700°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3906.3\ \text{kJ/kg} \\ s_3 = 7.6214\ \text{kJ/kg}\cdot\text{K} \end{array}$$

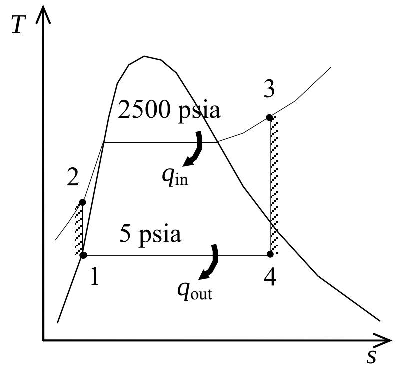
$$\left.\begin{array}{l} P_4 = 20\ \text{kPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{7.6214 - 0.8320}{7.0752} = 0.9596 \\ h_4 = h_f + x_4 h_{fg} = 251.42 + (0.9596)(2357.5) = 2513.7\ \text{kJ/kg} \end{array}$$

The power produced by the turbine and consumed by the pump are

$$\dot{W}_{\text{T,out}} = \dot{m}(h_3 - h_4) = (50\ \text{kg/s})(3906.3 - 2513.7)\text{kJ/kg} = \mathbf{69{,}630\ kW}$$
$$\dot{W}_{\text{P,in}} = \dot{m}w_{\text{P,in}} = (50\ \text{kg/s})(4.05\ \text{kJ/kg}) = \mathbf{203\ kW}$$

10-18E A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The turbine inlet temperature 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 From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f\,@\,5\text{ psia}} = 130.18\text{ Btu/lbm}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,5\text{ psia}} = 0.01641\text{ ft}^3/\text{lbm}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01641\text{ ft}^3/\text{lbm})(2500-5)\text{psia}\left(\frac{1\text{ Btu}}{5.404\text{ psia}\cdot\text{ft}^3}\right) \\ &= 7.58\text{ Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 130.18 + 7.58 = 137.76\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_4 = 5\text{ psia} \\ x_4 = 0.80 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 130.18 + (0.80)(1000.5) = 930.58\text{ Btu/lbm} \\ s_4 = s_f + x_4 s_{fg} = 0.23488 + (0.80)(1.60894) = 1.52203\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_3 = 2500\text{ psia} \\ s_3 = s_4 = 1.52203\text{ Btu/lbm}\cdot\text{R} \end{array}\right\} \begin{array}{l} h_3 = 1450.8\text{ Btu/lbm} \\ T_3 = \mathbf{989.2°F} \end{array}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 1450.8 - 137.76 = 1313.0\text{ Btu/lbm}$$
$$q_{\text{out}} = h_4 - h_1 = 930.58 - 130.18 = 800.4\text{ Btu/lbm}$$

The thermal efficiency of the cycle is

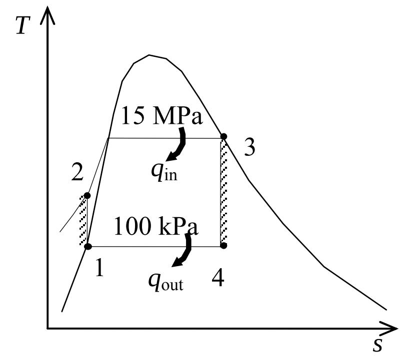
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{800.4}{1313.0} = \mathbf{0.390}$$

10-19 A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The power produced by the turbine, the heat added 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 From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,100\,\text{kPa}} = 417.51\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,100\,\text{kPa}} = 0.001043\ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001043\ \text{m}^3/\text{kg})(15{,}000 - 100)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 15.54\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 417.51 + 15.54 = 433.05\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 15{,}000\ \text{kPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2610.8\ \text{kJ/kg} \\ s_3 = 5.3108\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 100\ \text{kPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{5.3108 - 1.3028}{6.0562} = 0.6618 \\ h_4 = h_f + x_4 h_{fg} = 417.51 + (0.6618)(2257.5) = 1911.5\ \text{kJ/kg} \end{array}$$

Thus,

$$w_{\text{T,out}} = h_3 - h_4 = 2610.8 - 1911.5 = \mathbf{699.3\ kJ/kg}$$
$$q_{\text{in}} = h_3 - h_2 = 2610.8 - 433.05 = \mathbf{2177.8\ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 1911.5 - 417.51 = 1494.0\ \text{kJ/kg}$$

The thermal efficiency of the cycle is

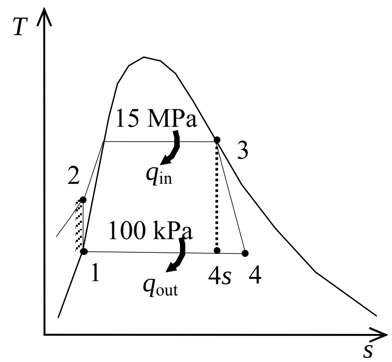
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1494.0}{2177.8} = \mathbf{0.314}$$

10-20 A simple Rankine cycle with water as the working fluid operates between the specified pressure limits. The isentropic efficiency of the turbine, 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 From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,100\,\text{kPa}} = 417.51\ \text{kJ/kg}$$
$$v_1 = v_{f\,@\,100\,\text{kPa}} = 0.001043\ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{\text{p,in}} &= v_1(P_2 - P_1) \\ &= (0.001043\ \text{m}^3/\text{kg})(15{,}000 - 100)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 15.54\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 417.51 + 15.54 = 433.05\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 15{,}000\ \text{kPa} \\ x_3 &= 1 \end{aligned}\right\} \begin{aligned} h_3 &= 2610.8\ \text{kJ/kg} \\ s_3 &= 5.3108\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 100\ \text{kPa} \\ s_4 &= s_3 \end{aligned}\right\} \begin{aligned} x_{4s} &= \frac{s_4 - s_f}{s_{fg}} = \frac{5.3108 - 1.3028}{6.0562} = 0.6618 \\ h_{4s} &= h_f + x_{4s}h_{fg} = 417.51 + (0.6618)(2257.5) = 1911.5\ \text{kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 100\ \text{kPa} \\ x_4 &= 0.70 \end{aligned}\right\} h_4 = h_f + x_4 h_{fg} = 417.51 + (0.70)(2257.5) = 1997.8\ \text{kJ/kg}$$

The isentropic efficiency of the turbine is

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{2610.8 - 1997.8}{2610.8 - 1911.5} = \mathbf{0.877}$$

Thus,

$$q_{in} = h_3 - h_2 = 2610.8 - 433.05 = 2177.8\ \text{kJ/kg}$$
$$q_{out} = h_4 - h_1 = 1997.8 - 417.51 = 1580.3\ \text{kJ/kg}$$

The thermal efficiency of the cycle is

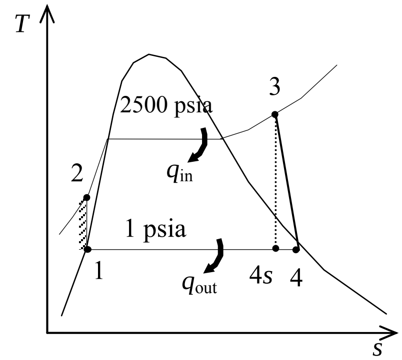
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1580.3}{2177.8} = \mathbf{0.274}$$

10-21E A simple steam Rankine cycle operates between the specified pressure limits. The mass flow rate, the power produced by the turbine, the rate of heat addition, 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 From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f\,@\,1\,\text{psia}} = 69.72\ \text{Btu/lbm}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,6\,\text{psia}} = 0.01614\ \text{ft}^3/\text{lbm}$$



$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01614\ \text{ft}^3/\text{lbm})(2500-1)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) \\ &= 7.46\ \text{Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 69.72 + 7.46 = 77.18\ \text{Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 2500\ \text{psia} \\ T_3 = 800°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1302.0\ \text{Btu/lbm} \\ s_3 = 1.4116\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 1\ \text{psia} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.4116 - 0.13262}{1.84495} = 0.6932 \\ h_{4s} = h_f + x_{4s}h_{fg} = 69.72 + (0.6932)(1035.7) = 787.70\ \text{Btu/lbm} \end{array}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s}) = 1302.0 - (0.90)(1302.0 - 787.70) = 839.13\ \text{kJ/kg}$$

Thus,

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 1302.0 - 77.18 = 1224.8\ \text{Btu/lbm} \\ q_{\text{out}} &= h_4 - h_1 = 839.13 - 69.72 = 769.41\ \text{Btu/lbm} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 1224.8 - 769.41 = 455.39\ \text{Btu/lbm} \end{aligned}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{1000\ \text{kJ/s}}{455.39\ \text{Btu/lbm}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = \mathbf{2.081\ lbm/s}$$

The power output from the turbine and the rate of heat addition are

$$\dot{W}_{\text{T,out}} = \dot{m}(h_3 - h_4) = (2.081\ \text{lbm/s})(1302.0 - 839.13)\text{Btu/lbm}\left(\frac{1\ \text{kJ}}{0.94782\ \text{Btu}}\right) = \mathbf{1016\ kW}$$

$$\dot{Q}_{\text{in}} = \dot{m}q_{\text{in}} = (2.081\ \text{lbm/s})(1224.8\ \text{Btu/lbm}) = \mathbf{2549\ Btu/s}$$

and the thermal efficiency of the cycle is

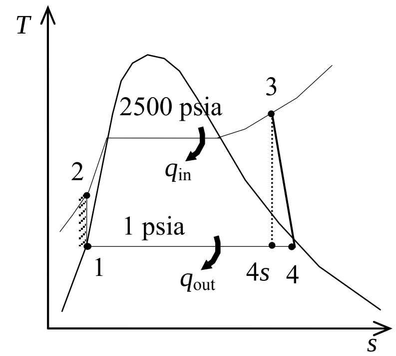
$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1000\ \text{kJ/s}}{2549\ \text{Btu/s}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = \mathbf{0.3718}$$

10-22E A simple steam Rankine cycle operates between the specified pressure limits. The mass flow rate, the power produced by the turbine, the rate of heat addition, 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 From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f\,@\,1\,\text{psia}} = 69.72\ \text{Btu/lbm}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,6\,\text{psia}} = 0.01614\ \text{ft}^3/\text{lbm}$$



$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01614\ \text{ft}^3/\text{lbm})(2500-1)\text{psia}\left(\frac{1\ \text{Btu}}{5.404\ \text{psia}\cdot\text{ft}^3}\right) \\ &= 7.46\ \text{Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 69.72 + 7.46 = 77.18\ \text{Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 2500\ \text{psia} \\ T_3 = 800°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1302.0\ \text{Btu/lbm} \\ s_3 = 1.4116\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 1\ \text{psia} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.4116 - 0.13262}{1.84495} = 0.6932 \\ h_{4s} = h_f + x_{4s}h_{fg} = 69.72 + (0.6932)(1035.7) = 787.70\ \text{Btu/lbm} \end{array}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s}) = 1302.0 - (0.90)(1302.0 - 787.70) = 839.13\ \text{kJ/kg}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m}(h_3 - h_4) \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{h_3 - h_4} = \frac{1000\ \text{kJ/s}}{(1302.0 - 839.13)\ \text{Btu/lbm}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = 2.048\ \text{lbm/s}$$

The rate of heat addition is

$$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (2.048\ \text{lbm/s})(1302.0 - 77.18)\text{Btu/lbm}\left(\frac{1\ \text{kJ}}{0.94782\ \text{Btu}}\right) = 2508\ \text{Btu/s}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1000\ \text{kJ/s}}{2508\ \text{Btu/s}}\left(\frac{0.94782\ \text{Btu}}{1\ \text{kJ}}\right) = 0.3779$$

The thermal efficiency in the previous problem was determined to be 0.3718. The error in the thermal efficiency caused by neglecting the pump work is then

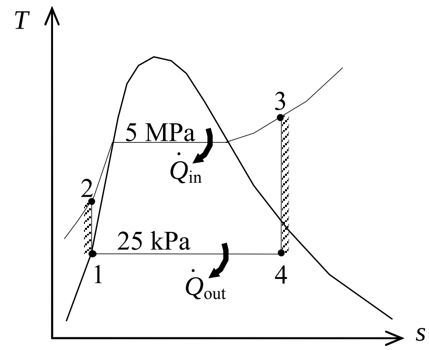
$$\text{Error} = \frac{0.3779 - 0.3718}{0.3718} \times 100 = \mathbf{1.64\%}$$

10-23 A 300-MW coal-fired steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The overall plant efficiency and the required rate of the coal supply 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),

$$h_1 = h_{f\,@\,25\text{ kPa}} = 271.96\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,25\text{ kPa}} = 0.001020\text{ m}^3/\text{kg}$$



$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00102\text{ m}^3/\text{kg})(5000 - 25\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 5.07\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 271.96 + 5.07 = 277.03\text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 5\text{ MPa} \\ T_3 &= 450°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3317.2\text{ kJ/kg} \\ s_3 &= 6.8210\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 25\text{ kPa} \\ s_4 &= s_3 \end{aligned}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8210 - 0.8932}{6.9370} = 0.8545$$
$$h_4 = h_f + x_4 h_{fg} = 271.96 + (0.8545)(2345.5) = 2276.2\text{ kJ/kg}$$

The thermal efficiency is determined from

$$q_{\text{in}} = h_3 - h_2 = 3317.2 - 277.03 = 3040.2\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2276.2 - 271.96 = 2004.2\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2004.2}{3040.2} = 0.3407$$

Thus,

$$\eta_{\text{overall}} = \eta_{\text{th}} \times \eta_{\text{comb}} \times \eta_{\text{gen}} = (0.3407)(0.75)(0.96) = \mathbf{24.5\%}$$

(*b*) Then the required rate of coal supply becomes

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{overall}}} = \frac{300{,}000\text{ kJ/s}}{0.2453} = 1{,}222{,}992\text{ kJ/s}$$

and

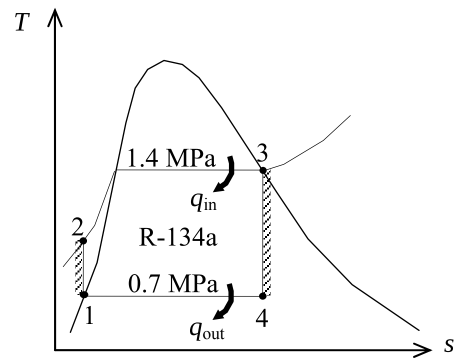
$$\dot{m}_{\text{coal}} = \frac{\dot{Q}_{\text{in}}}{C_{\text{coal}}} = \frac{1{,}222{,}992\text{ kJ/s}}{29{,}300\text{ kJ/kg}}\left(\frac{1\text{ ton}}{1000\text{ kg}}\right) = 0.04174\text{ tons/s} = \mathbf{150.3\ tons/h}$$

10-24 A solar-pond power plant that operates on a simple ideal Rankine cycle with refrigerant-134a as the working fluid is considered. The thermal efficiency of the cycle and the power output of the plant 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-11, A-12, and A-13),

$$h_1 = h_{f@\,0.7\text{ MPa}} = 88.82 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,0.7\text{ MPa}} = 0.0008331 \text{ m}^3\text{/kg}$$
$$w_{p,\text{in}} = \boldsymbol{v}_1(P_2 - P_1)$$
$$= (0.0008331 \text{ m}^3\text{/kg})(1400 - 700 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 0.58 \text{ kJ/kg}$$



$$h_2 = h_1 + w_{p,\text{in}} = 88.82 + 0.58 = 89.40 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.4 \text{ MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_3 = h_{g@\,1.4\text{ MPa}} = 276.12 \text{ kJ/kg} \\ s_3 = s_{g@\,1.4\text{ MPa}} = 0.9105 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 0.7 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{0.9105 - 0.33230}{0.58763} = 0.9839$$
$$h_4 = h_f + x_4 h_{fg} = 88.82 + (0.9839)(176.21) = 262.20 \text{ kJ/kg}$$

Thus ,

$$q_{\text{in}} = h_3 - h_2 = 276.12 - 89.40 = 186.72 \text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 262.20 - 88.82 = 173.38 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 186.72 - 173.38 = 13.34 \text{ kJ/kg}$$

and

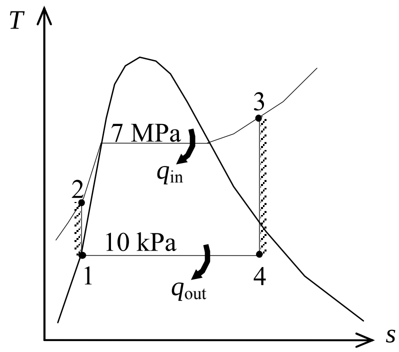
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{13.34 \text{ kJ/kg}}{186.72 \text{ kJ/kg}} = \mathbf{7.1\%}$$

(*b*) $\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} = (3 \text{ kg/s})(13.34 \text{ kJ/kg}) = \mathbf{40.02\ kW}$

10-25 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),



$$h_1 = h_{f @ 10\text{ kPa}} = 191.81 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f @ 10\text{ kPa}} = 0.00101 \text{ m}^3\text{/kg}$$
$$w_{p,\text{in}} = \boldsymbol{v}_1(P_2 - P_1)$$
$$= (0.00101 \text{ m}^3\text{/kg})(7{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 7.06 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 7.06 = 198.87 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 7 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3411.4 \text{ kJ/kg} \\ s_3 = 6.8000 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201$$
$$h_4 = h_f + x_4 h_{fg} = 191.81 + (0.8201)(2392.1) = 2153.6 \text{ kJ/kg}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3411.4 - 198.87 = 3212.5 \text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2153.6 - 191.81 = 1961.8 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3212.5 - 1961.8 = 1250.7 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1250.7 \text{ kJ/kg}}{3212.5 \text{ kJ/kg}} = \mathbf{38.9\%}$$

(*b*) $$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000 \text{ kJ/s}}{1250.7 \text{ kJ/kg}} = \mathbf{36.0 \text{ kg/s}}$$

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

$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (35.98 \text{ kg/s})(1961.8 \text{ kJ/kg}) = 70{,}586 \text{ kJ/s}$$
$$\Delta T_{\text{coolingwater}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}c)_{\text{coolingwater}}} = \frac{70{,}586 \text{ kJ/s}}{(2000 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot°\text{C})} = \mathbf{8.4°C}$$

10-26 A steam power plant operates on a simple nonideal 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.

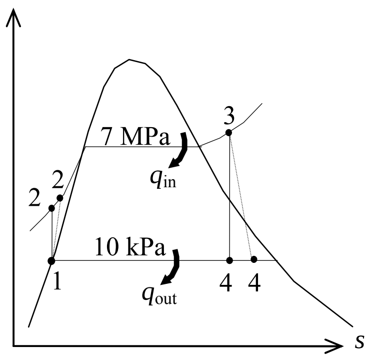
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

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

$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,10\text{ kPa}} = 0.00101\ \text{m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= \left(0.00101\ \text{m}^3\text{/kg}\right)\left(7{,}000 - 10\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/(0.87) \\ &= 8.11\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 8.11 = 199.92\text{ kJ/kg}$$



$$\left.\begin{aligned} P_3 &= 7\text{ MPa} \\ T_3 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3411.4\text{ kJ/kg} \\ s_3 &= 6.8000\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 10\text{ kPa} \\ s_4 &= s_3 \end{aligned}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201$$

$$h_{4s} = h_f + x_4 h_{fg} = 191.81 + (0.820)(2392.1) = 2153.6\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s})$$
$$= 3411.4 - (0.87)(3411.4 - 2153.6) = 2317.1\text{ kJ/kg}$$

Thus,

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 3411.4 - 199.92 = 3211.5\text{ kJ/kg} \\ q_{\text{out}} &= h_4 - h_1 = 2317.1 - 191.81 = 2125.3\text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 3211.5 - 2125.3 = 1086.2\text{ kJ/kg} \end{aligned}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1086.2\text{ kJ/kg}}{3211.5\text{ kJ/kg}} = \mathbf{33.8\%}$$

(*b*) $$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000\text{ kJ/s}}{1086.2\text{ kJ/kg}} = \mathbf{41.43\ kg/s}$$

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

$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (41.43\text{ kg/s})(2125.3\text{ kJ/kg}) = 88{,}051\text{ kJ/s}$$

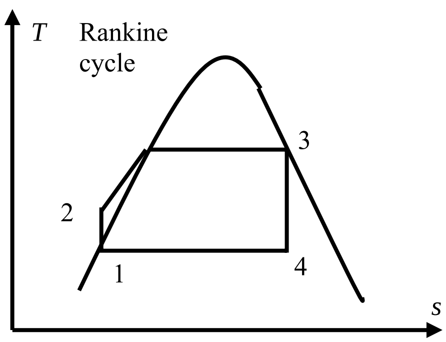
$$\Delta T_{\text{cooling water}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}c)_{\text{cooling water}}} = \frac{88{,}051\ \text{kJ/s}}{(2000\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})} = \mathbf{10.5°C}$$

10-27 The net work outputs and the thermal efficiencies for a Carnot cycle and a simple ideal Rankine cycle are to be determined.

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

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

$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.42\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,20\text{ kPa}} = 0.001017\text{ m}^3/\text{kg}$$



$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017\text{ m}^3/\text{kg})(10{,}000 - 20)\text{ kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.15\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 251.42 + 10.15 = 261.57\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10\text{ MPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2725.5\text{ kJ/kg} \\ s_3 = 5.6159\text{ kJ/kg}\cdot\text{K} \end{array}$$

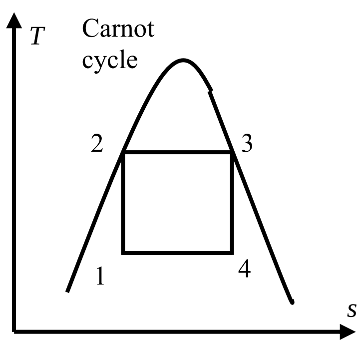
$$\left.\begin{array}{l} P_4 = 20\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{5.6159 - 0.8320}{7.0752} = 0.6761$$

$$\begin{aligned} h_4 &= h_f + x_4 h_{fg} = 251.42 + (0.6761)(2357.5) \\ &= 1845.3\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 2725.5 - 261.57 = 2463.9\text{ kJ/kg} \\ q_{\text{out}} &= h_4 - h_1 = 1845.3 - 251.42 = 1594.0\text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 2463.9 - 1594.0 = \mathbf{869.9\ kJ/kg} \end{aligned}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1594.0}{2463.9} = \mathbf{0.353}$$

(*b*) Carnot Cycle analysis:



$$\left.\begin{array}{l} P_3 = 10\text{ MPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2725.5\text{ kJ/kg} \\ T_3 = 311.0\,°\text{C} \end{array}$$

$$\left.\begin{array}{l} T_2 = T_3 = 311.0\,°\text{C} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} h_2 = 1407.8\text{ kJ/kg} \\ s_2 = 3.3603\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_1 = 20\text{ kPa} \\ s_1 = s_2 \end{array}\right\} \begin{array}{l} x_1 = \dfrac{s_1 - s_f}{s_{fg}} = \dfrac{3.3603 - 0.8320}{7.0752} = 0.3574 \\ h_1 = h_f + x_1 h_{fg} \end{array}$$

$$= 251.42 + (0.3574)(2357.5) = 1093.9\text{ kJ/kg}$$

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 2725.5 - 1407.8 = 1317.7\text{ kJ/kg} \\ q_{\text{out}} &= h_4 - h_1 = 1845.3 - 1093.9 = 751.4\text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 1317.7 - 752.3 = \mathbf{565.4\ kJ/kg} \end{aligned}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{751.4}{1317.7} = \mathbf{0.430}$$

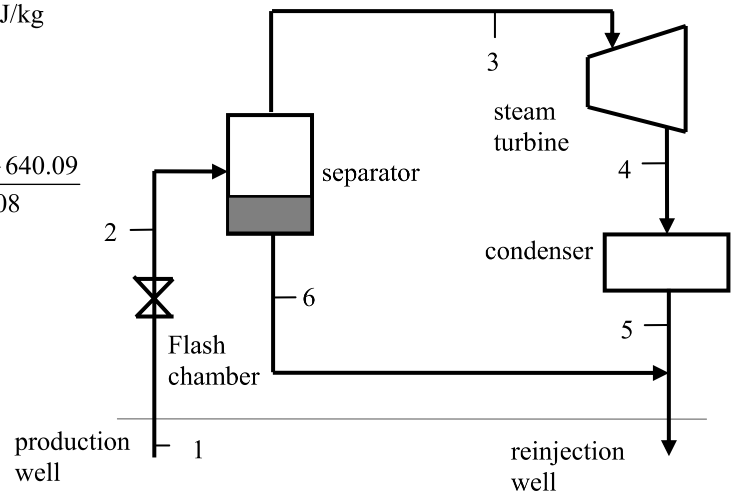
10-28 A single-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The mass flow rate of steam through the turbine, the isentropic efficiency of the turbine, the power output from the turbine, and the thermal efficiency of the plant are to be determined.

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

Analysis (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{aligned} T_1 &= 230°\text{C} \\ x_1 &= 0 \end{aligned}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 500 \text{ kPa} \\ h_2 &= h_1 = 990.14 \text{ kJ/kg} \end{aligned}\right\} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{990.14 - 640.09}{2108} = 0.1661$$



The mass flow rate of steam through the turbine is

$$\dot{m}_3 = x_2 \dot{m}_1 = (0.1661)(230 \text{ kg/s}) = \mathbf{38.20 \text{ kg/s}}$$

(*b*) Turbine:

$$\left.\begin{aligned} P_3 &= 500 \text{ kPa} \\ x_3 &= 1 \end{aligned}\right\} \begin{aligned} h_3 &= 2748.1 \text{ kJ/kg} \\ s_3 &= 6.8207 \text{ kJ/kg} \cdot \text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 10 \text{ kPa} \\ s_4 &= s_3 \end{aligned}\right\} h_{4s} = 2160.3 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_4 &= 10 \text{ kPa} \\ x_4 &= 0.90 \end{aligned}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.90)(2392.1) = 2344.7 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{2748.1 - 2344.7}{2748.1 - 2160.3} = \mathbf{0.686}$$

(*c*) The power output from the turbine is

$$\dot{W}_{\text{T,out}} = \dot{m}_3 (h_3 - h_4) = (38.20 \text{ kJ/kg})(2748.1 - 2344.7) \text{kJ/kg} = \mathbf{15{,}410 \text{ kW}}$$

(*d*) We use saturated liquid state at the standard temperature for dead state enthalpy

$$\left.\begin{aligned} T_0 &= 25°\text{C} \\ x_0 &= 0 \end{aligned}\right\} h_0 = 104.83 \text{ kJ/kg}$$

$$\dot{E}_{\text{in}} = \dot{m}_1 (h_1 - h_0) = (230 \text{ kJ/kg})(990.14 - 104.83) \text{kJ/kg} = 203{,}622 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{T,out}}}{\dot{E}_{\text{in}}} = \frac{15{,}410}{203{,}622} = 0.0757 = \mathbf{7.6\%}$$

10-29 A double-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The temperature of the steam at the exit of the second flash chamber, the power produced from the second turbine, and the thermal efficiency of the plant are to be determined.

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

Analysis (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 230°\text{C} \\ x_1 = 0 \end{array}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{array}\right\} x_2 = 0.1661$$

$$\dot{m}_3 = x_2\dot{m}_1 = (0.1661)(230 \text{ kg/s}) = 38.20 \text{ kg/s}$$

$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 0.1661 = 191.80 \text{ kg/s}$$

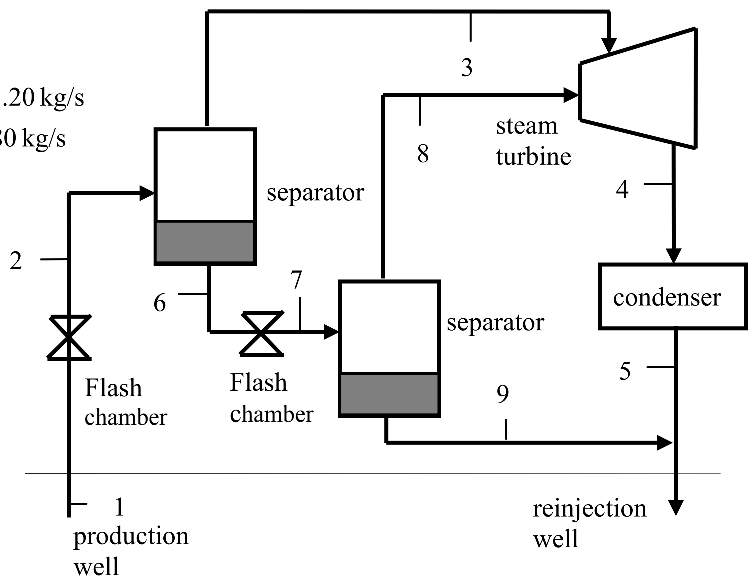
$$\left.\begin{array}{l} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{array}\right\} h_3 = 2748.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.90 \end{array}\right\} h_4 = 2344.7 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 500 \text{ kPa} \\ x_6 = 0 \end{array}\right\} h_6 = 640.09 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 150 \text{ kPa} \\ h_7 = h_6 \end{array}\right\} \begin{array}{l} T_7 = \mathbf{111.35\ °C} \\ x_7 = 0.0777 \end{array}$$

$$\left.\begin{array}{l} P_8 = 150 \text{ kPa} \\ x_8 = 1 \end{array}\right\} h_8 = 2693.1 \text{ kJ/kg}$$



(*b*) The mass flow rate at the lower stage of the turbine is

$$\dot{m}_8 = x_7\dot{m}_6 = (0.0777)(191.80 \text{ kg/s}) = 14.90 \text{ kg/s}$$

The power outputs from the high and low pressure stages of the turbine are

$$\dot{W}_{\text{T1,out}} = \dot{m}_3(h_3 - h_4) = (38.20 \text{ kJ/kg})(2748.1 - 2344.7)\text{kJ/kg} = 15{,}410 \text{ kW}$$

$$\dot{W}_{\text{T2,out}} = \dot{m}_8(h_8 - h_4) = (14.90 \text{ kJ/kg})(2693.1 - 2344.7)\text{kJ/kg} = \mathbf{5191\ kW}$$

(*c*) We use saturated liquid state at the standard temperature for the dead state enthalpy

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} h_0 = 104.83 \text{ kJ/kg}$$

$$\dot{E}_{\text{in}} = \dot{m}_1(h_1 - h_0) = (230 \text{ kg/s})(990.14 - 104.83)\text{kJ/kg} = 203{,}621 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{T,out}}}{\dot{E}_{\text{in}}} = \frac{15{,}410 + 5193}{203{,}621} = 0.101 = \mathbf{10.1\%}$$

10-30 A combined flash-binary geothermal power plant uses hot geothermal water at 230ºC as the heat source. The mass flow rate of isobutane in the binary cycle, the net power outputs from the steam turbine and the binary cycle, and the thermal efficiencies for the binary cycle and the combined plant are to be determined.

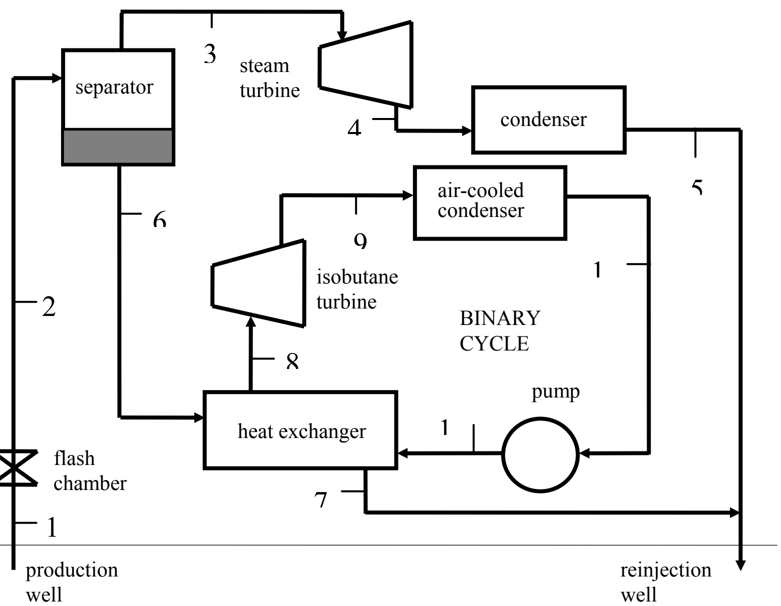
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 230°\text{C} \\ x_1 = 0 \end{array}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{array}\right\} x_2 = 0.1661$$

$$\dot{m}_3 = x_2\dot{m}_1 = (0.1661)(230 \text{ kg/s}) = 38.20 \text{ kg/s}$$
$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 38.20 = 191.80 \text{ kg/s}$$



$$\left.\begin{array}{l} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{array}\right\} h_3 = 2748.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.90 \end{array}\right\} h_4 = 2344.7 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 500 \text{ kPa} \\ x_6 = 0 \end{array}\right\} h_6 = 640.09 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_7 = 90°\text{C} \\ x_7 = 0 \end{array}\right\} h_7 = 377.04 \text{ kJ/kg}$$

The isobutane properties are obtained from EES:

$$\left.\begin{array}{l} P_8 = 3250 \text{ kPa} \\ T_8 = 145°\text{C} \end{array}\right\} h_8 = 755.05 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 400 \text{ kPa} \\ T_9 = 80°\text{C} \end{array}\right\} h_9 = 691.01 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{10} = 400 \text{ kPa} \\ x_{10} = 0 \end{array}\right\} \begin{array}{l} h_{10} = 270.83 \text{ kJ/kg} \\ \boldsymbol{v}_{10} = 0.001839 \text{ m}^3\text{/kg} \end{array}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_{10}(P_{11} - P_{10})/\eta_p \\ &= (0.001819 \text{ m}^3\text{/kg})(3250 - 400)\text{kPa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)/0.90 \\ &= 5.82 \text{ kJ/kg.} \end{aligned}$$

$$h_{11} = h_{10} + w_{p,\text{in}} = 270.83 + 5.82 = 276.65 \text{ kJ/kg}$$

An energy balance on the heat exchanger gives

$$\dot{m}_6(h_6 - h_7) = \dot{m}_{\text{iso}}(h_8 - h_{11})$$

$$(191.81 \text{ kg/s})(640.09 - 377.04)\text{kJ/kg} = \dot{m}_{\text{iso}}(755.05 - 276.65)\text{kJ/kg} \longrightarrow \dot{m}_{\text{iso}} = \mathbf{105.46 \text{ kg/s}}$$

(*b*) The power outputs from the steam turbine and the binary cycle are

$$\dot{W}_{\text{T,steam}} = \dot{m}_3(h_3 - h_4) = (38.19 \text{ kJ/kg})(2748.1 - 2344.7)\text{kJ/kg} = \mathbf{15{,}410 \text{ kW}}$$

$$\dot{W}_{\text{T,iso}} = \dot{m}_{iso}(h_8 - h_9) = (105.46 \text{ kJ/kg})(755.05 - 691.01)\text{kJ/kg} = 6753 \text{ kW}$$

$$\dot{W}_{\text{net,binary}} = \dot{W}_{\text{T,iso}} - \dot{m}_{\text{iso}} w_{p,in} = 6753 - (105.46 \text{ kg/s})(5.82 \text{ kJ/kg}) = \mathbf{6139\ kW}$$

(*c*) The thermal efficiencies of the binary cycle and the combined plant are

$$\dot{Q}_{\text{in,binary}} = \dot{m}_{\text{iso}}(h_8 - h_{11}) = (105.46 \text{ kJ/kg})(755.05 - 276.65)\text{kJ/kg} = 50{,}454 \text{ kW}$$

$$\eta_{\text{th,binary}} = \frac{\dot{W}_{\text{net,binary}}}{\dot{Q}_{\text{in,binary}}} = \frac{6139}{50{,}454} = 0.122 = \mathbf{12.2\%}$$

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} h_0 = 104.83 \text{ kJ/kg}$$

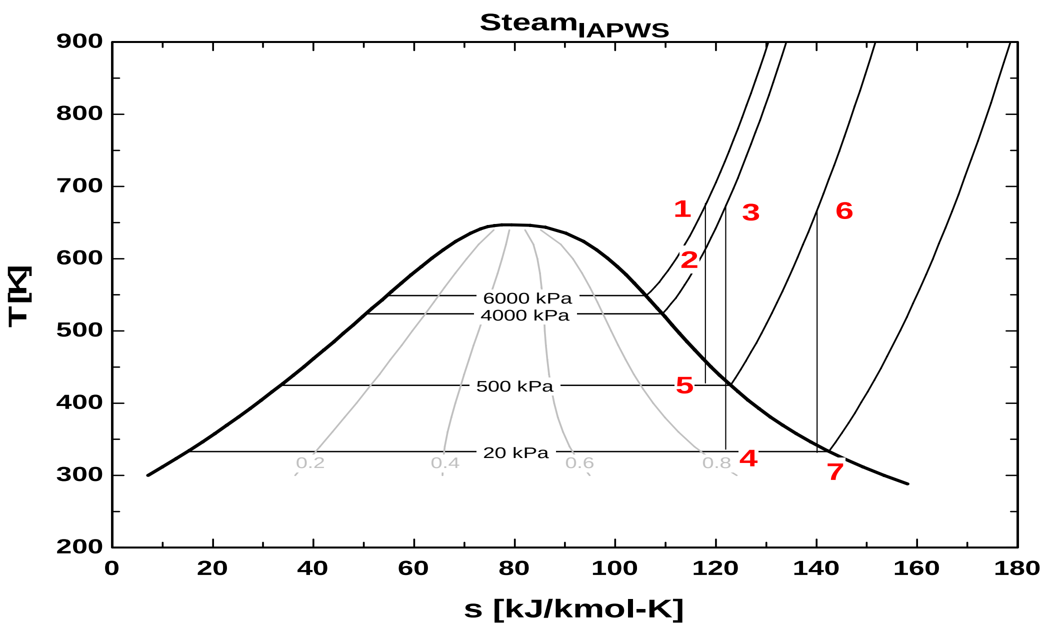
$$\dot{E}_{\text{in}} = \dot{m}_1(h_1 - h_0) = (230 \text{ kJ/kg})(990.14 - 104.83)\text{kJ/kg} = 203{,}622 \text{ kW}$$

$$\eta_{\text{th,plant}} = \frac{\dot{W}_{\text{T,steam}} + \dot{W}_{\text{net,binary}}}{\dot{E}_{\text{in}}} = \frac{15{,}410 + 6139}{203{,}622} = 0.106 = \mathbf{10.6\%}$$

The Reheat Rankine Cycle

10-31C The pump work remains the same, the moisture content decreases, everything else increases.

10-32C The *T-s* diagram shows two reheat cases for the reheat Rankine cycle similar to the one shown in Figure 10-11. In the first case there is expansion through the high-pressure turbine from 6000 kPa to 4000 kPa between states 1 and 2 with reheat at 4000 kPa to state 3 and finally expansion in the low-pressure turbine to state 4. In the second case there is expansion through the high-pressure turbine from 6000 kPa to 500 kPa between states 1 and 5 with reheat at 500 kPa to state 6 and finally expansion in the low-pressure turbine to state 7. Increasing the pressure for reheating increases the average temperature for heat addition makes the energy of the steam more available for doing work, see the reheat process 2 to 3 versus the reheat process 5 to 6. Increasing the reheat pressure will increase the cycle efficiency. However, as the reheating pressure increases, the amount of condensation increases during the expansion process in the low-pressure turbine, state 4 versus state 7. An optimal pressure for reheating generally allows for the moisture content of the steam at the low-pressure turbine exit to be in the range of 10 to 15% and this corresponds to quality in the range of 85 to 90%.



10-33C The thermal efficiency of the simple ideal Rankine cycle will probably be higher since the average temperature at which heat is added will be higher in this case.

10-34 [*Also solved by EES on enclosed CD*] A steam power plant that operates on the ideal reheat Rankine cycle is considered. The turbine work output 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.

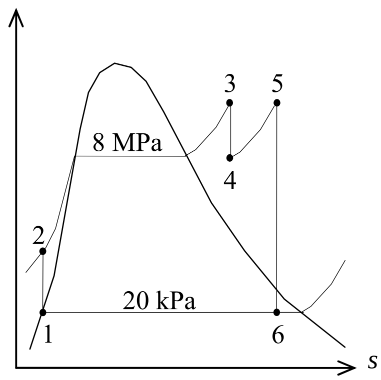
Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.42 \text{ kJ/kg}$$

$$v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.001017 \text{ m}^3\text{/kg})(8000 - 20 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 8.12 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 251.42 + 8.12 = 259.54 \text{ kJ/kg}$$



$$\left.\begin{array}{l} P_3 = 8 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3399.5 \text{ kJ/kg} \\ s_3 = 6.7266 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 3 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 3105.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 3 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3457.2 \text{ kJ/kg} \\ s_5 = 7.2359 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 20 \text{ kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{7.2359 - 0.8320}{7.0752} = 0.9051 \\ h_6 = h_f + x_6 h_{fg} = 251.42 + (0.9051)(2357.5) = 2385.2 \text{ kJ/kg} \end{array}$$

The turbine work output and the thermal efficiency are determined from

$$w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3399.5 - 3105.1 + 3457.2 - 2385.2 = \mathbf{1366.4 \text{ kJ/kg}}$$

and

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3399.5 - 259.54 + 3457.2 - 3105.1 = 3492.0 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{T,out} - w_{p,\text{in}} = 1366.4 - 8.12 = 1358.3 \text{ kJ/kg}$$

Thus,

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1358.3 \text{ kJ/kg}}{3492.5 \text{ kJ/kg}} = \mathbf{38.9\%}$$

10-35 EES Problem 10-34 is reconsidered. The problem is to be solved by the diagram window data entry feature of EES by including the effects of the turbine and pump efficiencies and reheat on the steam quality at the low-pressure turbine exit Also, the *T*-*s* diagram is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data - from diagram window"
{P[6] = 20 [kPa]
P[3] = 8000 [kPa]
T[3] = 500 [C]
P[4] = 3000 [kPa]
T[5] = 500 [C]
Eta_t = 100/100 "Turbine isentropic efficiency"
Eta_p = 100/100 "Pump isentropic efficiency"}

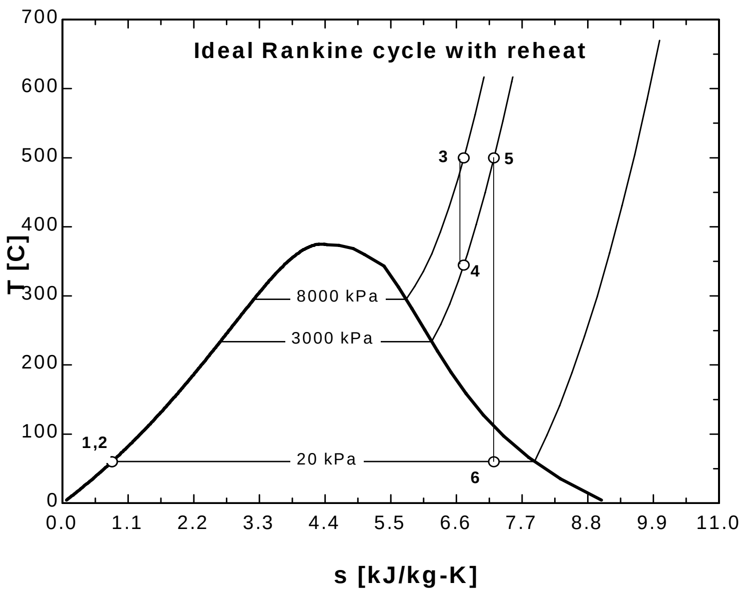
"Pump analysis"
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end

Fluid$='Steam_IAPWS'

P[1] = P[6]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,T=T[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"
"Low Pressure Turbine analysis"
P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
```

```
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
"Boiler analysis"
Q_in + h[2]+h[4]=h[3]+h[5]"SSSF First Law for the Boiler"
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])

"Cycle Statistics"
W_net=W_t_hp+W_t_lp-W_p
Eff=W_net/Q_in
```



SOLUTION

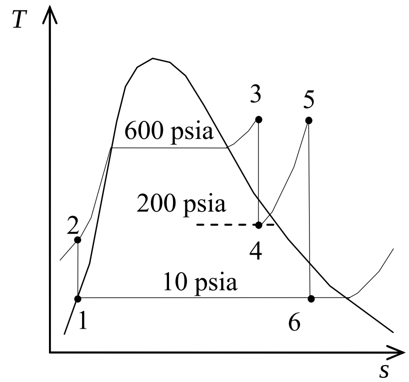
Eff=0.389	Eta_p=1	Eta_t=1
Fluid$='Steam_IAPWS'	h[1]=251.4 [kJ/kg]	h[2]=259.5 [kJ/kg]
h[3]=3400 [kJ/kg]	h[4]=3105 [kJ/kg]	h[5]=3457 [kJ/kg]
h[6]=2385 [kJ/kg]	hs[4]=3105 [kJ/kg]	hs[6]=2385 [kJ/kg]
P[1]=20 [kPa]	P[2]=8000 [kPa]	P[3]=8000 [kPa]
P[4]=3000 [kPa]	P[5]=3000 [kPa]	P[6]=20 [kPa]
Q_in=3493 [kJ/kg]	Q_out=2134 [kJ/kg]	s[1]=0.832 [kJ/kg-K]
s[2]=0.8321 [kJ/kg-K]	s[3]=6.727 [kJ/kg-K]	s[4]=6.727 [kJ/kg-K]
s[5]=7.236 [kJ/kg-K]	s[6]=7.236 [kJ/kg-K]	s_s[4]=6.727 [kJ/kg-K]
s_s[6]=7.236 [kJ/kg-K]	T[1]=60.06 [C]	T[2]=60.4 [C]
T[3]=500 [C]	T[4]=345.2 [C]	T[5]=500 [C]
T[6]=60.06 [C]	Ts[4]=345.2 [C]	Ts[6]=60.06 [C]
v[1]=0.001017 [m^3/kg]	v[2]=0.001014 [m^3/kg]	v[3]=0.04177 [m^3/kg]
v[4]=0.08968 [m^3/kg]	vs[6]=6.922 [m^3/kg]	W_net=1359 [kJ/kg]
W_p=8.117 [kJ/kg]	W_p_s=8.117 [kJ/kg]	W_t_hp=294.8 [kJ/kg]
W_t_lp=1072 [kJ/kg]	x6s$=''	x[1]=0
x[6]=0.9051		

10-36E An ideal reheat steam Rankine cycle produces 5000 kW power. The rates of heat addition and rejection, 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 From the steam tables (Tables A-4E, A-5E, and A-6E or EES),

$$h_1 = h_{f\,@\,10\text{ psia}} = 161.25 \text{ Btu/lbm}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,10\text{ psia}} = 0.01659 \text{ ft}^3/\text{lbm}$$



$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01659 \text{ ft}^3/\text{lbm})(600-10)\text{psia}\left(\frac{1 \text{ Btu}}{5.404 \text{ psia}\cdot\text{ft}^3}\right) \\ &= 1.81 \text{ Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 161.25 + 1.81 = 163.06 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 600 \text{ psia} \\ T_3 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1289.9 \text{ Btu/lbm} \\ s_3 = 1.5325 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 200 \text{ psia} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.5325 - 0.54379}{1.00219} = 0.9865 \\ h_4 = h_f + x_4 h_{fg} = 355.46 + (0.9865)(843.33) = 1187.5 \text{ Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_5 = 200 \text{ psia} \\ T_5 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_5 = 1322.3 \text{ Btu/lbm} \\ s_5 = 1.6771 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ psia} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{1.6771 - 0.28362}{1.50391} = 0.9266 \\ h_6 = h_f + x_6 h_{fg} = 161.25 + (0.9266)(981.82) = 1071.0 \text{ Btu/lbm} \end{array}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 1289.9 - 163.06 + 1322.3 - 1187.5 = 1261.7 \text{ Btu/lbm}$$
$$q_{\text{out}} = h_6 - h_1 = 1071.0 - 161.25 = 909.7 \text{ Btu/lbm}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1261.7 - 909.8 = 352.0 \text{ Btu/lbm}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{5000 \text{ kJ/s}}{352.0 \text{ Btu/lbm}}\left(\frac{0.94782 \text{ Btu}}{1 \text{ kJ}}\right) = 13.47 \text{ lbm/s}$$

The rates of heat addition and rejection are

$$\dot{Q}_{\text{in}} = \dot{m}q_{\text{in}} = (13.47 \text{ lbm/s})(1261.7 \text{ Btu/lbm}) = \mathbf{16{,}995\ Btu/s}$$
$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (13.47 \text{ lbm/s})(909.7 \text{ Btu/lbm}) = \mathbf{12{,}250\ Btu/s}$$

and the thermal efficiency of the cycle is

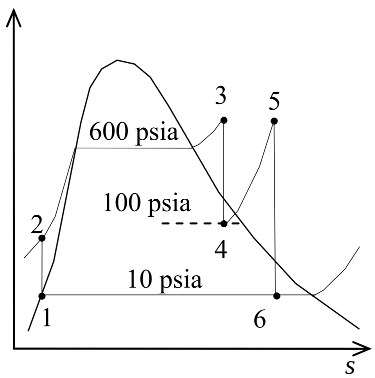
$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{5000 \text{ kJ/s}}{16{,}990 \text{ Btu/s}}\left(\frac{0.94782 \text{ Btu}}{1 \text{ kJ}}\right) = \mathbf{0.2790}$$

10-37E An ideal reheat steam Rankine cycle produces 5000 kW power. The rates of heat addition and rejection, and the thermal efficiency of the cycle are to be determined for a reheat pressure of 100 psia.

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

Analysis From the steam tables (Tables A-4E, A-5E, and A-6E or EES),

$$h_1 = h_{f\,@\,10\text{ psia}} = 161.25 \text{ Btu/lbm}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,6\text{ psia}} = 0.01659 \text{ ft}^3\text{/lbm}$$



$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01659 \text{ ft}^3\text{/lbm})(600-10)\text{psia}\left(\frac{1 \text{ Btu}}{5.404 \text{ psia}\cdot\text{ft}^3}\right) \\ &= 1.81 \text{ Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 161.25 + 1.81 = 163.06 \text{ Btu/lbm}$$

$$\left.\begin{aligned} P_3 &= 600 \text{ psia} \\ T_3 &= 600°\text{F} \end{aligned}\right\} \begin{aligned} h_3 &= 1289.9 \text{ Btu/lbm} \\ s_3 &= 1.5325 \text{ Btu/lbm}\cdot\text{R} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 100 \text{ psia} \\ s_4 &= s_3 \end{aligned}\right\} \begin{aligned} x_4 &= \frac{s_4 - s_f}{s_{fg}} = \frac{1.5325 - 0.47427}{1.12888} = 0.9374 \\ h_4 &= h_f + x_4 h_{fg} = 298.51 + (0.9374)(888.99) = 1131.9 \text{ Btu/lbm} \end{aligned}$$

$$\left.\begin{aligned} P_5 &= 100 \text{ psia} \\ T_5 &= 600°\text{F} \end{aligned}\right\} \begin{aligned} h_5 &= 1329.4 \text{ Btu/lbm} \\ s_5 &= 1.7586 \text{ Btu/lbm}\cdot\text{R} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 10 \text{ psia} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} x_6 &= \frac{s_6 - s_f}{s_{fg}} = \frac{1.7586 - 0.28362}{1.50391} = 0.9808 \\ h_6 &= h_f + x_6 h_{fg} = 161.25 + (0.9808)(981.82) = 1124.2 \text{ Btu/lbm} \end{aligned}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 1289.9 - 163.07 + 1329.4 - 1131.9 = 1324.4 \text{ Btu/lbm}$$
$$q_{\text{out}} = h_6 - h_1 = 1124.2 - 161.25 = 962.9 \text{ Btu/lbm}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1324.4 - 962.9 = 361.5 \text{ Btu/lbm}$$

The mass flow rate of steam in the cycle is determined from

$$\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} \longrightarrow \dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{5000 \text{ kJ/s}}{361.5 \text{ Btu/lbm}}\left(\frac{0.94782 \text{ Btu}}{1 \text{ kJ}}\right) = 13.11 \text{ lbm/s}$$

The rates of heat addition and rejection are

$$\dot{Q}_{\text{in}} = \dot{m}q_{\text{in}} = (13.11 \text{ lbm/s})(1324.4 \text{ Btu/lbm}) = \mathbf{17{,}360 \text{ Btu/s}}$$
$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (13.11 \text{ lbm/s})(962.9 \text{ Btu/lbm}) = \mathbf{12{,}620 \text{ Btu/s}}$$

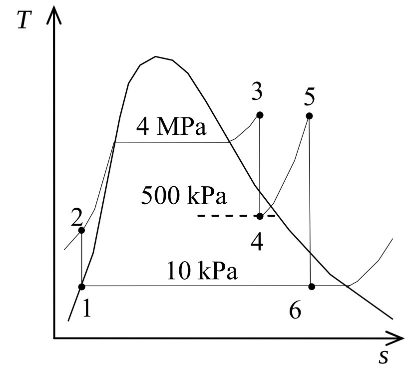
and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{5000 \text{ kJ/s}}{17{,}360 \text{ Btu/s}}\left(\frac{0.94782 \text{ Btu}}{1 \text{ kJ}}\right) = \mathbf{0.2729}$$

Discussion The thermal efficiency for 200 psia reheat pressure was determined in the previous problem to be 0.2790. Thus, operating the reheater at 100 psia causes a slight decrease in the thermal efficiency.

10-38 An ideal reheat Rankine with water as the working fluid is considered. The temperatures at the inlet of both turbines, 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 From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\,\text{kPa}} = 191.81\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,10\,\text{kPa}} = 0.001010\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001010\ \text{m}^3/\text{kg})(4000-10)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 4.03\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 191.81 + 4.03 = 195.84\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 500\ \text{kPa} \\ x_4 = 0.90 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 640.09 + (0.90)(2108.0) = 2537.3\ \text{kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 1.8604 + (0.90)(4.9603) = 6.3247\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 4000\ \text{kPa} \\ s_3 = s_4 \end{array}\right\} \begin{array}{l} h_3 = 2939.4\ \text{kJ/kg} \\ T_3 = \mathbf{292.2^\circ C} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10\ \text{kPa} \\ x_6 = 0.90 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.81 + (0.90)(2392.1) = 2344.7\ \text{kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6492 + (0.90)(7.4996) = 7.3989\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 500\ \text{kPa} \\ s_5 = s_6 \end{array}\right\} \begin{array}{l} h_5 = 3029.2\ \text{kJ/kg} \\ T_5 = \mathbf{282.9^\circ C} \end{array}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 2939.4 - 195.84 + 3029.2 - 2537.3 = 3235.4\ \text{kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2344.7 - 191.81 = 2152.9\ \text{kJ/kg}$$

and

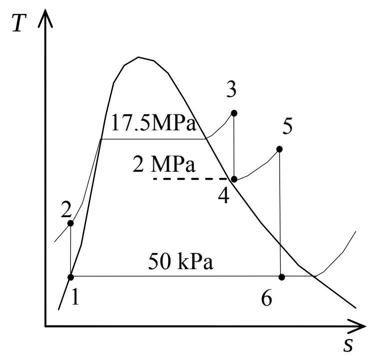
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2152.9}{3235.4} = \mathbf{0.335}$$

10-39 An ideal reheat Rankine cycle with water as the working fluid is considered. The thermal efficiency of the cycle is to be determined.

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

Analysis From the steam tables (Tables A-4, A-5, and A-6 or EES),

$$h_1 = h_{f\,@\,50\,\text{kPa}} = 340.54\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,50\,\text{kPa}} = 0.001030\ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001030\ \text{m}^3/\text{kg})(17500 - 50)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 17.97\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 340.54 + 17.97 = 358.51\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 17{,}500\ \text{kPa} \\ T_3 &= 550°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3423.6\ \text{kJ/kg} \\ s_3 &= 6.4266\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 2000\ \text{kPa} \\ s_4 &= s_3 \end{aligned}\right\} h_4 = 2841.5\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_5 &= 2000\ \text{kPa} \\ T_5 &= 300°\text{C} \end{aligned}\right\} \begin{aligned} h_5 &= 3024.2\ \text{kJ/kg} \\ s_5 &= 6.7684\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 50\ \text{kPa} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} x_6 &= \frac{s_6 - s_f}{s_{fg}} = \frac{6.7684 - 1.0912}{6.5019} = 0.8732 \\ h_6 &= h_f + x_6 h_{fg} = 340.54 + (0.8732)(2304.7) = 2352.9\ \text{kJ/kg} \end{aligned}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3423.6 - 358.51 + 3024.2 - 2841.5 = 3247.8\ \text{kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2352.9 - 340.54 = 2012.4\ \text{kJ/kg}$$

and

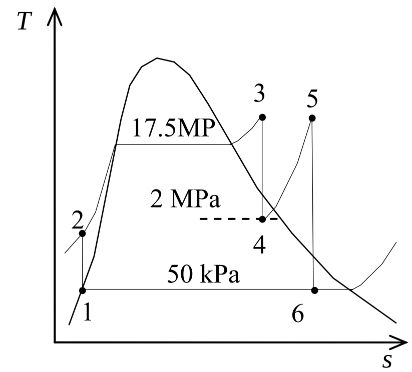
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2012.4}{3247.8} = \mathbf{0.380}$$

10-40 An ideal reheat Rankine cycle with water as the working fluid is considered. The thermal efficiency of the cycle is to be determined.

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

Analysis From the steam tables (Tables A-4, A-5, and A-6 or EES),

$$h_1 = h_{f\,@\,50\,\text{kPa}} = 340.54\ \text{kJ/kg}$$
$$v_1 = v_{f\,@\,50\,\text{kPa}} = 0.001030\ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{\text{p,in}} &= v_1(P_2 - P_1) \\ &= (0.001030\ \text{m}^3/\text{kg})(17500-50)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 17.97\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 340.54 + 17.97 = 358.52\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 17{,}500\ \text{kPa} \\ T_3 &= 550°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3423.6\ \text{kJ/kg} \\ s_3 &= 6.4266\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 2000\ \text{kPa} \\ s_4 &= s_3 \end{aligned}\right\} h_4 = 2841.5\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_5 &= 2000\ \text{kPa} \\ T_5 &= 550°\text{C} \end{aligned}\right\} \begin{aligned} h_5 &= 3579.0\ \text{kJ/kg} \\ s_5 &= 7.5725\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 50\ \text{kPa} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} x_6 &= \frac{s_6 - s_f}{s_{fg}} = \frac{7.5725 - 1.0912}{6.5019} = 0.9968 \\ h_6 &= h_f + x_6 h_{fg} = 340.54 + (0.9968)(2304.7) = 2638.0\ \text{kJ/kg} \end{aligned}$$

Thus,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3423.6 - 358.52 + 3579.0 - 2841.5 = 3802.6\ \text{kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2638.0 - 340.54 = 2297.4\ \text{kJ/kg}$$

and

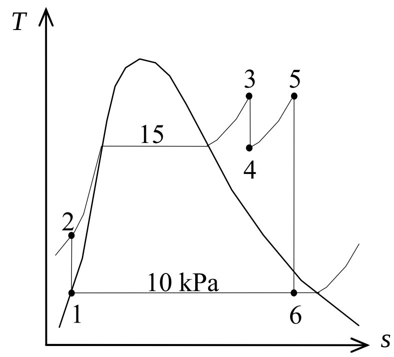
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2297.4}{3802.6} = \mathbf{0.396}$$

The thermal efficiency was determined to be 0.380 when the temperature at the inlet of low-pressure turbine was 300°C. When this temperature is increased to 550°C, the thermal efficiency becomes 0.396. This corresponding to a percentage increase of 4.2% in thermal efficiency.

10-41 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),



$$h_1 = h_{\text{sat@ 10 kPa}} = 191.81 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{\text{sat@ 10 kPa}} = 0.00101 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00101 \text{ m}^3\text{/kg})(15{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 15.14 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 15.14 = 206.95 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 15 \text{ MPa} \\ T_3 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3310.8 \text{ kJ/kg} \\ s_3 &= 6.3480 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 10 \text{ kPa} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} h_6 &= h_f + x_6 h_{fg} = 191.81 + (0.90)(2392.1) = 2344.7 \text{ kJ/kg} \\ s_6 &= s_f + x_6 s_{fg} = 0.6492 + (0.90)(7.4996) = 7.3988 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} T_5 &= 500°\text{C} \\ s_5 &= s_6 \end{aligned}\right\} \begin{aligned} P_5 &= \mathbf{2150 \text{ kPa}} \text{ (the reheat pressure)} \\ h_5 &= 3466.61 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 2.15 \text{ MPa} \\ s_4 &= s_3 \end{aligned}\right\} h_4 = 2817.2 \text{ kJ/kg}$$

(*b*) The rate of heat supply is

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{m}[(h_3 - h_2) + (h_5 - h_4)] \\ &= (12 \text{ kg/s})(3310.8 - 206.95 + 3466.61 - 2817.2) \text{kJ/kg} = \mathbf{45{,}039 \text{ kW}} \end{aligned}$$

(*c*) The thermal efficiency is determined from

$$\dot{Q}_{\text{out}} = \dot{m}(h_6 - h_1) = (12 \text{ kJ/s})(2344.7 - 191.81) \text{kJ/kg} = 25{,}835 \text{ kJ/s}$$

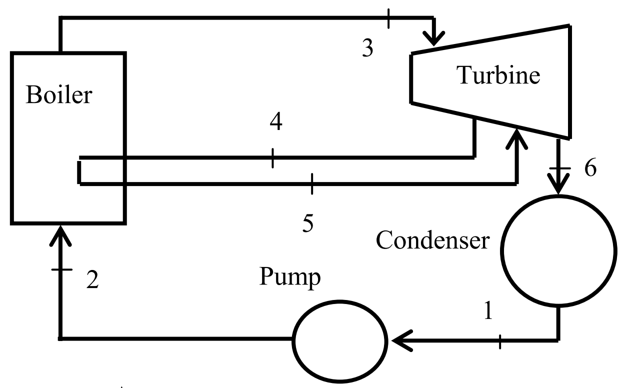
Thus,

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{25{,}834 \text{ kJ/s}}{45{,}039 \text{ kJ/s}} = \mathbf{42.6\%}$$

10-42 A steam power plant that operates on a reheat Rankine cycle is considered. The condenser pressure, the net power output, and the thermal efficiency 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),

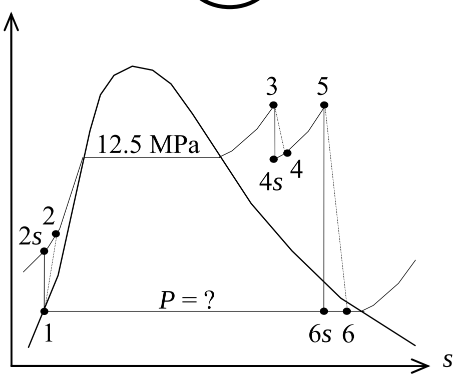


$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3476.5 \text{ kJ/kg} \\ s_3 = 6.6317 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 2 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} h_{4s} = 2948.1 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$\begin{aligned} \rightarrow h_4 &= h_3 - \eta_T (h_3 - h_{4s}) \\ &= 3476.5 - (0.85)(3476.5 - 2948.1) \\ &= 3027.3 \text{ kJ/kg} \end{aligned}$$



$$\left.\begin{array}{l} P_5 = 2 \text{ MPa} \\ T_5 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3358.2 \text{ kJ/kg} \\ s_5 = 7.2815 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = ? \\ x_6 = 0.95 \end{array}\right\} h_6 =$$

$$\left.\begin{array}{l} P_6 = ? \\ s_6 = s_5 \end{array}\right\} h_{6s} =$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow \begin{aligned} h_6 &= h_5 - \eta_T (h_5 - h_{6s}) \\ &= 3358.2 - (0.85)(3358.2 - 2948.1) \\ &= 3027.3 \text{ kJ/kg} \end{aligned}$$

The pressure at state 6 may be determined by a trial-error approach from the steam tables or by using EES from the above equations:

$$P_6 = \mathbf{9.73\ kPa}, \quad h_6 = 2463.3 \text{ kJ/kg},$$

(*b*) Then,

$$h_1 = h_{f\,@\,9.73 \text{ kPa}} = 189.57 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,10 \text{ kPa}} = 0.001010 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1 (P_2 - P_1)/\eta_p \\ &= (0.00101 \text{ m}^3\text{/kg})(12{,}500 - 9.73 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)/(0.90) \\ &= 14.02 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 189.57 + 14.02 = 203.59 \text{ kJ/kg}$$

Cycle analysis:

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3476.5 - 3027.3 + 3358.2 - 2463.3 = 3603.8 \text{ kJ/kg}$$

$$q_{\text{out}} = h_6 - h_1 = 3027.3 - 189.57 = 2273.7 \text{ kJ/kg}$$

$$\dot{W}_{\text{net}} = \dot{m}(q_{\text{in}} - q_{\text{out}}) = (7.7 \text{ kg/s})(3603.8 - 2273.7)\text{kJ/kg} = \mathbf{10{,}242\ kW}$$

(*c*) The thermal efficiency is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2273.7 \text{ kJ/kg}}{3603.8 \text{ kJ/kg}} = 0.369 = \mathbf{36.9\%}$$

Regenerative Rankine Cycle

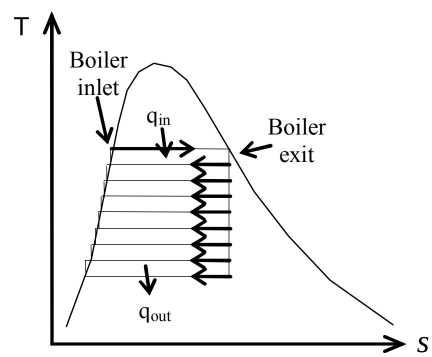
10-43C Moisture content remains the same, everything else decreases.

10-44C This is a smart idea because we waste little work potential but we save a lot from the heat input. The extracted steam has little work potential left, and most of its energy would be part of the heat rejected anyway. Therefore, by regeneration, we utilize a considerable amount of heat by sacrificing little work output.

10-45C In open feedwater heaters, the two fluids actually mix, but in closed feedwater heaters there is no mixing.

10-46C Both cycles would have the same efficiency.

10-47C To have the same thermal efficiency as the Carnot cycle, the cycle must receive and reject heat isothermally. Thus the liquid should be brought to the saturated liquid state at the boiler pressure isothermally, and the steam must be a saturated vapor at the turbine inlet. This will require an infinite number of heat exchangers (feedwater heaters), as shown on the *T-s* diagram.



10-48 Feedwater is heated by steam in a feedwater heater of a regenerative The required mass flow rate of the steam 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 From the steam tables (Tables A-4 through A-6 or EES),

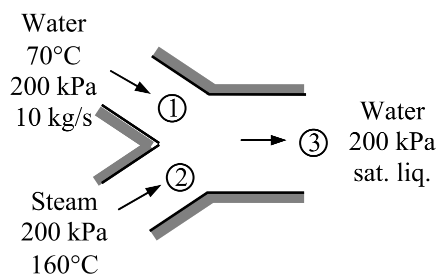
$$h_1 \cong h_{f\,@\,70°\text{C}} = 293.07 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ T_2 = 160°\text{C} \end{array}\right\} h_2 = 2789.7 \text{ kJ/kg}$$

$$h_3 = h_{f\,@\,200 \text{ kPa}} = 504.71 \text{ kJ/kg}$$

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:



$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

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

$$\dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

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

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

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

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

Solving for $\dot{m}_2$, and substituting gives

$$\dot{m}_2 = \dot{m}_1 \frac{h_1 - h_3}{h_3 - h_2} = (10 \text{ kg/s}) \frac{(293.07 - 504.71) \text{ kJ/kg}}{(504.71 - 2789.7) \text{ kJ/kg}} = \mathbf{0.926 \text{ kg/s}}$$

10-49E In a regenerative Rankine cycle, the closed feedwater heater with a pump as shown in the figure is arranged so that the water at state 5 is mixed with the water at state 2 to form a feedwater which is a saturated liquid. The mass flow rate of bleed steam required to operate this unit is to be determined.

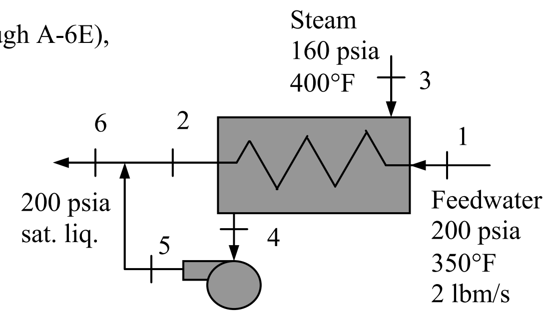
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

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

$$\left.\begin{array}{l} P_1 = 200\text{ psia} \\ T_1 = 350°\text{F} \end{array}\right\} h_1 \cong h_{f\,@\,350°\text{F}} = 321.73\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 160\text{ psia} \\ T_3 = 400°\text{F} \end{array}\right\} h_3 = 1218.0\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_6 = 200\text{ psia} \\ x_6 = 0 \end{array}\right\} h_6 = h_{f\,@\,200\text{ psia}} = 355.46\text{ Btu/lbm}$$



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

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

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

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 + \dot{m}_3 w_{\text{P,in}} = \dot{m}_6 h_6$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 + \dot{m}_3 w_{\text{P,in}} = (\dot{m}_1 + \dot{m}_3) h_6$$

Solving this for $\dot{m}_3$,

$$\dot{m}_3 = \dot{m}_1 \frac{h_6 - h_1}{(h_3 - h_6) + w_{\text{P,in}}} = (2\text{ lbm/s}) \frac{355.46 - 321.73}{1218.0 - 355.46 + 0.1344} = \mathbf{0.0782\ lbm/s}$$

where

$$w_{\text{P,in}} = \boldsymbol{v}_4 (P_5 - P_4) = \boldsymbol{v}_{f\,@\,160\text{ psia}} (P_5 - P_4)$$

$$= (0.01815\text{ ft}^3/\text{lbm})(200 - 160)\text{ psia} \left(\frac{1\text{ Btu}}{5.404\text{ psia} \cdot \text{ft}^3} \right) = 0.1344\text{ Btu/lbm}$$

10-50E The closed feedwater heater of a regenerative Rankine cycle is to heat feedwater to a saturated liquid. The required mass flow rate of bleed steam 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** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

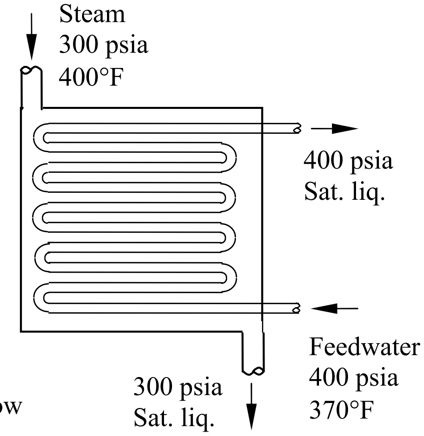
Properties From the steam tables (Tables A-4E through A-6E),

$$\left.\begin{array}{l} P_1 = 400\text{ psia} \\ T_1 = 370°\text{F} \end{array}\right\} h_1 \cong h_{f\,@\,370°\text{F}} = 342.88\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_2 = 400\text{ psia} \\ x_2 = 0 \end{array}\right\} h_2 = h_{f\,@\,400\text{ psia}} = 424.13\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 300\text{ psia} \\ T_3 = 500°\text{F} \end{array}\right\} h_3 = 1257.9\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_4 = 300\text{ psia} \\ x_4 = 0 \end{array}\right\} h_4 = h_{f\,@\,300\text{ psia}} = 393.94\text{ Btu/lbm}$$



Analysis We take the heat exchanger 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 (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_{fw} \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

Energy balance (for the heat exchanger):

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

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

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two,

$$\dot{m}_{fw}(h_2 - h_1) = \dot{m}_s(h_3 - h_4)$$

Solving for $\dot{m}_s$:

$$\dot{m}_s = \frac{h_2 - h_1}{h_3 - h_4}\dot{m}_{fw}$$

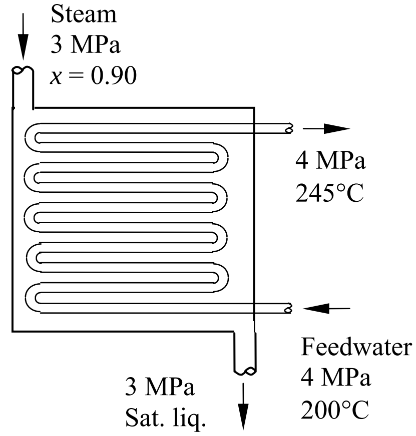
Substituting,

$$\dot{m}_s = \frac{424.13 - 342.88}{1257.9 - 393.94}(1\text{ lbm/s}) = \mathbf{0.0940\ lbm/s}$$

10-51 The closed feedwater heater of a regenerative Rankine cycle is to heat feedwater to a saturated liquid. The required mass flow rate of bleed steam 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** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

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



$$\left.\begin{array}{l} P_1 = 4000\text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} h_1 \cong h_{f\,@\,200°\text{C}} = 852.26\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 4000\text{ kPa} \\ T_2 = 245°\text{C} \end{array}\right\} h_2 \cong h_{f\,@\,245°\text{C}} = 1061.5\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 3000\text{ kPa} \\ x_3 = 0.90 \end{array}\right\} h_3 = h_f + x_3 h_{fg}$$

$$= 1008.3 + (0.9)(1794.9) = 2623.7\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 3000\text{ kPa} \\ x_4 = 0 \end{array}\right\} h_4 = h_{f\,@\,3000\text{ kPa}} = 1008.3\text{ kJ/kg}$$

Analysis We take the heat exchanger 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 (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_{fw} \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

Energy balance (for the heat exchanger):

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

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

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two,

$$\dot{m}_{fw}(h_2 - h_1) = \dot{m}_s(h_3 - h_4)$$

Solving for $\dot{m}_s$:

$$\dot{m}_s = \frac{h_2 - h_1}{h_3 - h_4}\dot{m}_{fw}$$

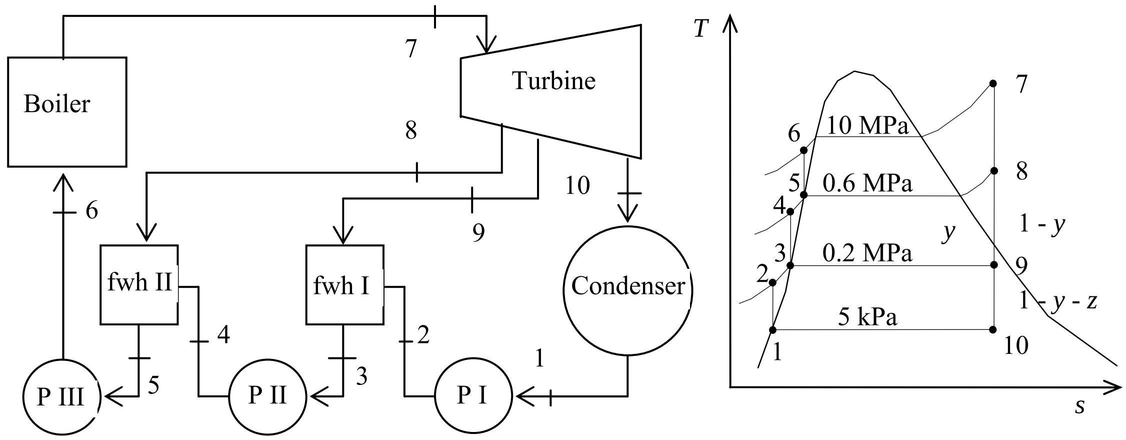
Substituting,

$$\dot{m}_s = \frac{1061.5 - 852.26}{2623.7 - 1008.3}(6\text{ kg/s}) = \mathbf{0.777\ kg/s}$$

10-52 A steam power plant operates on an ideal regenerative Rankine cycle with two open feedwater heaters. The net power output of the power plant 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),

$$h_1 = h_{f\,@\,5\text{ kPa}} = 137.75 \text{ kJ/kg}$$
$$v_1 = v_{f\,@\,5\text{ kPa}} = 0.001005 \text{ m}^3\text{/kg}$$
$$w_{pI,\text{in}} = v_1(P_2 - P_1) = (0.001005 \text{ m}^3\text{/kg})(200 - 5 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = 0.20 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{pI,\text{in}} = 137.75 + 0.20 = 137.95 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.2 \text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.2\text{ MPa}} = 504.71 \text{ kJ/kg} \\ v_3 = v_{f\,@\,0.2\text{ MPa}} = 0.001061 \text{ m}^3\text{/kg} \end{array}$$

$$w_{pII,\text{in}} = v_3(P_4 - P_3) = (0.001061 \text{ m}^3\text{/kg})(600 - 200 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 0.42 \text{ kJ/kg}$$
$$h_4 = h_3 + w_{pII,\text{in}} = 504.71 + 0.42 = 505.13 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 0.6 \text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_5 = h_{f\,@\,0.6\text{ MPa}} = 670.38 \text{ kJ/kg} \\ v_5 = v_{f\,@\,0.6\text{ MPa}} = 0.001101 \text{ m}^3\text{/kg} \end{array}$$

$$w_{pIII,\text{in}} = v_5(P_6 - P_5) = (0.001101 \text{ m}^3\text{/kg})(10{,}000 - 600 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.35 \text{ kJ/kg}$$

$$h_6 = h_5 + w_{pIII,\text{in}} = 670.38 + 10.35 = 680.73 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 10 \text{ MPa} \\ T_7 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3625.8 \text{ kJ/kg} \\ s_7 = 6.9045 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 0.6 \text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_8 = 2821.8 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 0.2 \text{ MPa} \\ s_9 = s_7 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{6.9045 - 1.5302}{5.5968} = 0.9602 \\ h_9 = h_f + x_9 h_{fg} = 504.71 + (0.9602)(2201.6) = 2618.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_{10} = 5 \text{ kPa} \\ s_{10} = s_7 \end{array}\right\} \begin{array}{l} x_{10} = \dfrac{s_{10} - s_f}{s_{fg}} = \dfrac{6.9045 - 0.4762}{7.9176} = 0.8119 \\ h_{10} = h_f + x_{10} h_{fg} = 137.75 + (0.8119)(2423.0) = 2105.0 \text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

FWH-2:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_4 h_4 = \dot{m}_5 h_5 \longrightarrow y h_8 + (1-y) h_4 = 1(h_5)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_5$). Solving for y,

$$y = \frac{h_5 - h_4}{h_8 - h_4} = \frac{670.38 - 505.13}{2821.8 - 505.13} = 0.07133$$

FWH-1:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_9 h_9 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow z h_9 + (1-y-z) h_2 = (1-y) h_3$$

where z is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for z,

$$z = \frac{h_3 - h_2}{h_9 - h_2}(1-y) = \frac{504.71 - 137.95}{2618.7 - 137.95}(1 - 0.07136) = 0.1373$$

Then,

$$q_{\text{in}} = h_7 - h_6 = 3625.8 - 680.73 = 2945.0 \text{ kJ/kg}$$

$$q_{\text{out}} = (1-y-z)(h_{10} - h_1) = (1 - 0.07133 - 0.1373)(2105.0 - 137.75) = 1556.8 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2945.0 - 1556.8 = 1388.2 \text{ kJ/kg}$$

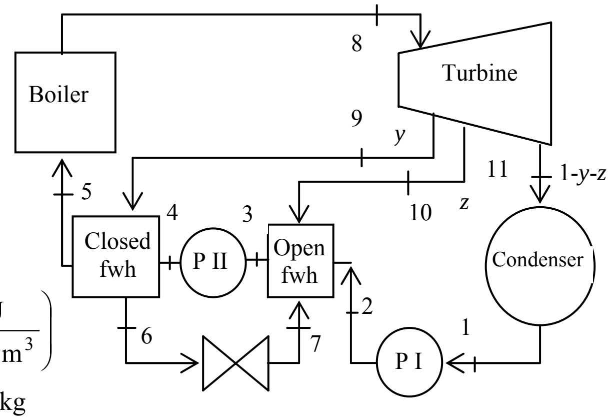
and

$$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (22 \text{ kg/s})(1388.2 \text{ kJ/kg}) = 30{,}540 \text{ kW} \cong \mathbf{30.5 \text{ MW}}$$

(*b*)
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1556.8 \text{ kJ/kg}}{2945.0 \text{ kJ/kg}} = \mathbf{47.1\%}$$

10-53 [*Also solved by EES on enclosed CD*] A steam power plant operates on an ideal regenerative Rankine cycle with two feedwater heaters, one closed and one open. The mass flow rate of steam through the boiler for a net power output of 250 MW 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.



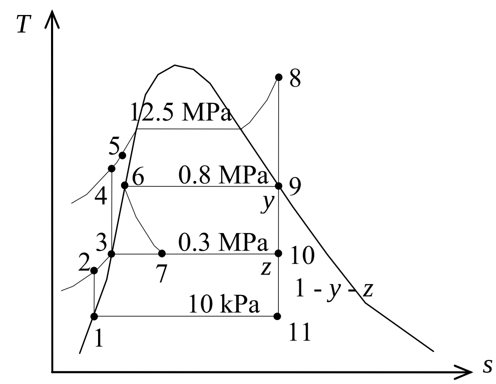
Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f @ 10\text{ kPa}} = 191.81\text{ kJ/kg}$$

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

$$\begin{aligned} w_{pI,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.00101\ \text{m}^3\text{/kg})(300 - 10\ \text{kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.29\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 191.81 + 0.29 = 192.10\ \text{kJ/kg}$$



$$\left.\begin{array}{l} P_3 = 0.3\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f @ 0.3\text{ MPa}} = 561.43\text{ kJ/kg} \\ v_3 = v_{f @ 0.3\text{ MPa}} = 0.001073\ \text{m}^3\text{/kg} \end{array}$$

$$w_{pII,\text{in}} = v_3(P_4 - P_3)$$

$$= (0.001073\ \text{m}^3\text{/kg})(12{,}500 - 300\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 13.09\text{ kJ/kg}$$

$$h_4 = h_3 + w_{pII,\text{in}} = 561.43 + 13.09 = 574.52\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 0.8\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_6 = h_7 = h_{f @ 0.8\text{ MPa}} = 720.87\text{ kJ/kg} \\ v_6 = v_{f @ 0.8\text{ MPa}} = 0.001115\ \text{m}^3\text{/kg} \\ T_6 = T_{\text{sat} @ 0.8\text{ MPa}} = 170.4°\text{C} \end{array}$$

$$T_6 = T_5,\ P_5 = 12.5\ \text{MPa} \rightarrow h_5 = 727.83\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 12.5\text{ MPa} \\ T_8 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_8 = 3476.5\text{ kJ/kg} \\ s_8 = 6.6317\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_9 = 0.8\text{ MPa} \\ s_9 = s_8 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{6.6317 - 2.0457}{4.6160} = 0.9935 \\ h_9 = h_f + x_9 h_{fg} = 720.87 + (0.9935)(2047.5) = 2755.0\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_{10} = 0.3\text{ MPa} \\ s_{10} = s_8 \end{array}\right\} \begin{array}{l} x_{10} = \dfrac{s_{10} - s_f}{s_{fg}} = \dfrac{6.6317 - 1.6717}{5.3200} = 0.9323 \\ h_{10} = h_f + x_{10} h_{fg} = 561.43 + (0.9323)(2163.5) = 2578.5\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_{11} = 10\text{ kPa} \\ s_{11} = s_8 \end{array}\right\} \begin{array}{l} x_{11} = \dfrac{s_{11} - s_f}{s_{fg}} = \dfrac{6.6317 - 0.6492}{7.4996} = 0.7977 \\ h_{11} = h_f + x_{11} h_{fg} = 191.81 + (0.7977)(2392.1) = 2100.0\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0\,\text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_9(h_9 - h_6) = \dot{m}_5(h_5 - h_4) \longrightarrow y(h_9 - h_6) = (h_5 - h_4)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_{10} / \dot{m}_5$). Solving for y,

$$y = \frac{h_5 - h_4}{h_9 - h_6} = \frac{727.83 - 574.52}{2755.0 - 720.87} = 0.0753$$

For the open FWH,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0\,\text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_7 h_7 + \dot{m}_2 h_2 + \dot{m}_{10} h_{10} = \dot{m}_3 h_3 \longrightarrow y h_7 + (1 - y - z) h_2 + z h_{10} = (1) h_3$$

where z is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for z,

$$z = \frac{(h_3 - h_2) - y(h_7 - h_2)}{h_{10} - h_2} = \frac{561.43 - 192.10 - (0.0753)(720.87 - 192.10)}{2578.5 - 192.10} = 0.1381$$

Then,

$$q_{\text{in}} = h_8 - h_5 = 3476.5 - 727.36 = 2749.1 \text{ kJ/kg}$$
$$q_{\text{out}} = (1 - y - z)(h_{11} - h_1) = (1 - 0.0753 - 0.1381)(2100.0 - 191.81) = 1500.1 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2749.1 - 1500.1 = 1249 \text{ kJ/kg}$$

and

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{250{,}000 \text{ kJ/s}}{1249 \text{ kJ/kg}} = \mathbf{200.2\ kg/s}$$

(*b*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1500.1 \text{ kJ/kg}}{2749.1 \text{ kJ/kg}} = \mathbf{45.4\%}$$

10-54 EES Problem 10-53 is reconsidered. The effects of turbine and pump efficiencies on the mass flow rate and thermal efficiency are to be investigated. Also, the *T-s* diagram is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
P[8] = 12500 [kPa]
T[8] = 550 [C]
P[9] = 800 [kPa]
"P_cfwh=300 [kPa]"
P[10] = P_cfwh
P_cond=10 [kPa]
P[11] = P_cond
W_dot_net=250 [MW]*Convert(MW, kW)
Eta_turb= 100/100 "Turbine isentropic efficiency"
Eta_turb_hp = Eta_turb "Turbine isentropic efficiency for high pressure stages"
Eta_turb_ip = Eta_turb "Turbine isentropic efficiency for intermediate pressure stages"
Eta_turb_lp = Eta_turb "Turbine isentropic efficiency for low pressure stages"
Eta_pump = 100/100 "Pump isentropic efficiency"

"Condenser exit pump  or Pump 1 analysis"

Fluid$='Steam_IAPWS'
P[1] = P[11]
P[2]=P[10]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Open Feedwater Heater analysis"
z*h[10] + y*h[7] + (1-y-z)*h[2] = 1*h[3] "Steady-flow conservation of energy"
h[3]=enthalpy(Fluid$,P=P[3],x=0)
T[3]=temperature(Fluid$,P=P[3],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[3]=entropy(Fluid$,P=P[3],x=0)

"Boiler condensate pump  or Pump 2 analysis"
P[5]=P[8]
P[4] = P[5]
P[3]=P[10]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])

"Closed Feedwater Heater analysis"
P[6]=P[9]
 y*h[9] + 1*h[4] = 1*h[5] + y*h[6] "Steady-flow conservation of energy"
```

```
h[5]=enthalpy(Fluid$,P=P[6],x=0) "h[5] = h(T[5], P[5]) where T[5]=Tsat at P[9]"
T[5]=temperature(Fluid$,P=P[5],h=h[5]) "Condensate leaves heater as sat. liquid at P[6]"
s[5]=entropy(Fluid$,P=P[6],h=h[5])
h[6]=enthalpy(Fluid$,P=P[6],x=0)
T[6]=temperature(Fluid$,P=P[6],x=0) "Condensate leaves heater as sat. liquid at P[6]"
s[6]=entropy(Fluid$,P=P[6],x=0)

"Trap analysis"
P[7] = P[10]
y*h[6] = y*h[7]  "Steady-flow conservation of energy for the trap operating as a throttle"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])

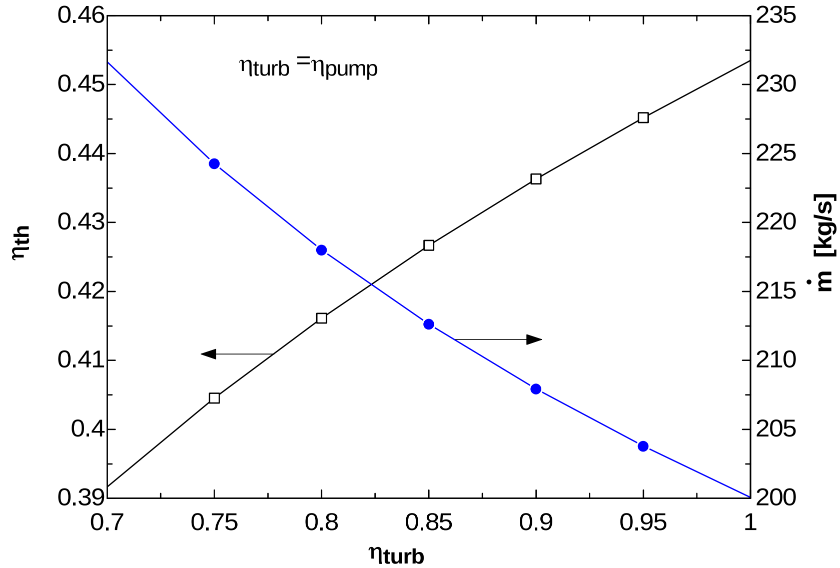
"Boiler analysis"
q_in + h[5]=h[8]"SSSF conservation of energy for the Boiler"
h[8]=enthalpy(Fluid$, T=T[8], P=P[8])
s[8]=entropy(Fluid$, T=T[8], P=P[8])

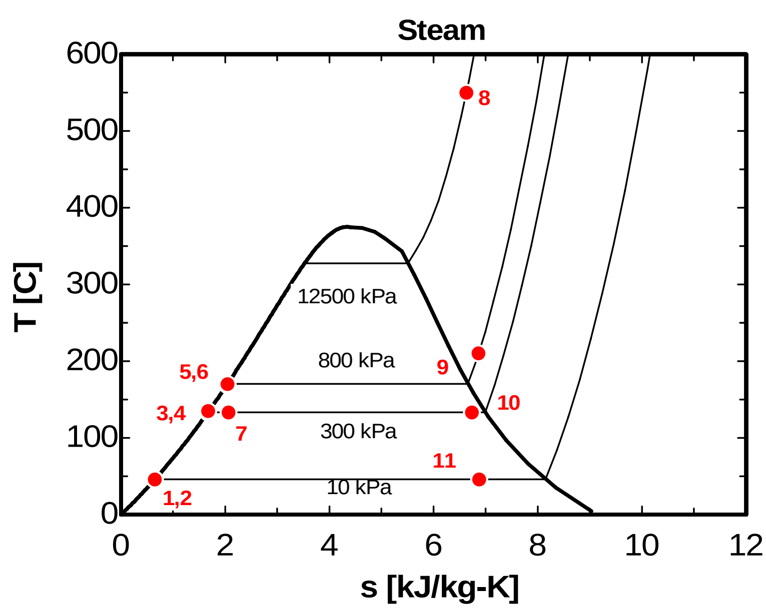
"Turbine analysis"
ss[9]=s[8]
hs[9]=enthalpy(Fluid$,s=ss[9],P=P[9])
Ts[9]=temperature(Fluid$,s=ss[9],P=P[9])
h[9]=h[8]-Eta_turb_hp*(h[8]-hs[9])"Definition of turbine efficiency for high pressure stages"
T[9]=temperature(Fluid$,P=P[9],h=h[9])
s[9]=entropy(Fluid$,P=P[9],h=h[9])
ss[10]=s[8]
hs[10]=enthalpy(Fluid$,s=ss[10],P=P[10])
Ts[10]=temperature(Fluid$,s=ss[10],P=P[10])
h[10]=h[9]-Eta_turb_ip*(h[9]-hs[10])"Definition of turbine efficiency for Intermediate pressure stages"
T[10]=temperature(Fluid$,P=P[10],h=h[10])
s[10]=entropy(Fluid$,P=P[10],h=h[10])
ss[11]=s[8]
hs[11]=enthalpy(Fluid$,s=ss[11],P=P[11])
Ts[11]=temperature(Fluid$,s=ss[11],P=P[11])
h[11]=h[10]-Eta_turb_lp*(h[10]-hs[11])"Definition of turbine efficiency for low pressure stages"
T[11]=temperature(Fluid$,P=P[11],h=h[11])
s[11]=entropy(Fluid$,P=P[11],h=h[11])
h[8] =y*h[9] + z*h[10] + (1-y-z)*h[11] + w_turb "SSSF conservation of energy for turbine"

"Condenser analysis"
(1-y-z)*h[11]=q_out+(1-y-z)*h[1]"SSSF First Law for the Condenser"

"Cycle Statistics"
w_net=w_turb - ((1-y-z)*w_pump1+ w_pump2)
Eta_th=w_net/q_in
W_dot_net = m_dot * w_net
```

η_{turb}	η_{turb}	η_{th}	m [kg/s]
0.7	0.7	0.3916	231.6
0.75	0.75	0.4045	224.3
0.8	0.8	0.4161	218
0.85	0.85	0.4267	212.6
0.9	0.9	0.4363	207.9
0.95	0.95	0.4452	203.8
1	1	0.4535	200.1

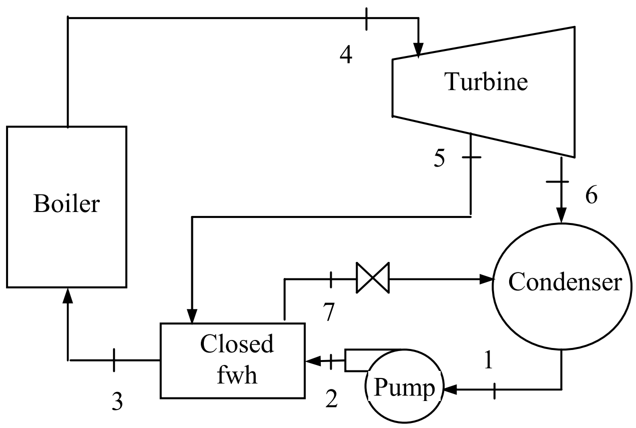


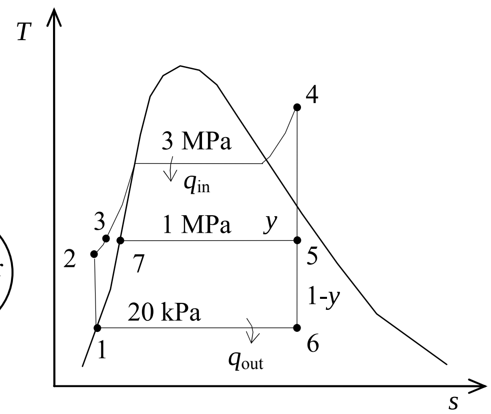


10-55 An ideal regenerative Rankine cycle with a closed feedwater heater is considered. The work produced by the turbine, the work consumed by the pumps, and the heat added in the boiler are to be determined.

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

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





$$h_1 = h_{f @ 20\,\text{kPa}} = 251.42 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f @ 20\,\text{kPa}} = 0.001017 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,in} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017 \text{ m}^3\text{/kg})(3000-20)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 3.03 \text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{p,in} = 251.42 + 3.03 = 254.45 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 3000 \text{ kPa} \\ T_4 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_4 = 3116.1 \text{ kJ/kg} \\ s_4 = 6.7450 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 1000 \text{ kPa} \\ s_5 = s_4 \end{array}\right\} h_5 = 2851.9 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 20 \text{ kPa} \\ s_6 = s_4 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.7450 - 0.8320}{7.0752} = 0.8357 \\ h_6 = h_f + x_6 h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7 \text{ kJ/kg} \end{array}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.

$$\left.\begin{array}{l} P_7 = 1000 \text{ kPa} \\ x_7 = 0 \end{array}\right\} \begin{array}{l} h_7 = 762.51 \text{ kJ/kg} \\ T_7 = 179.9°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 3000 \text{ kPa} \\ T_3 = T_7 = 209.9°\text{C} \end{array}\right\} h_3 = 763.53 \text{ kJ/kg}$$

An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5 / \dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$
$$y h_5 + 1 h_2 = 1 h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{763.53 - 254.45}{2851.9 - 762.51} = 0.2437$$

Then,

$$w_{\mathrm{T,out}} = h_4 - h_5 + (1-y)(h_5 - h_6) = 3116.1 - 2851.9 + (1 - 0.2437)(2851.9 - 2221.7) = \mathbf{740.9\ kJ/kg}$$
$$w_{\mathrm{P,in}} = \mathbf{3.03\ kJ/kg}$$
$$q_{\mathrm{in}} = h_4 - h_3 = 3116.1 - 763.53 = \mathbf{2353\ kJ/kg}$$

Also,

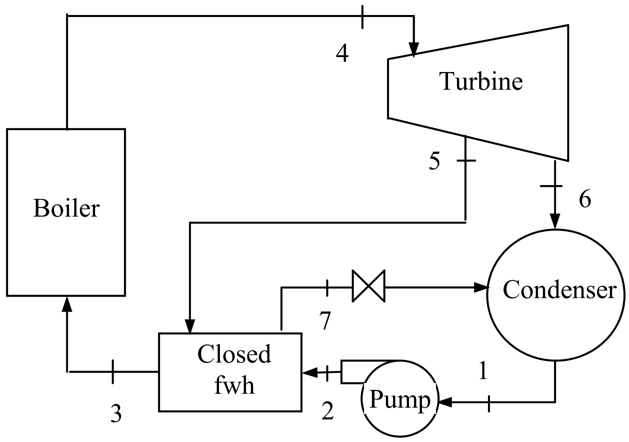
$$w_{\mathrm{net}} = w_{\mathrm{T,out}} - w_{\mathrm{P,in}} = 740.9 - 3.03 = 737.8\ \mathrm{kJ/kg}$$

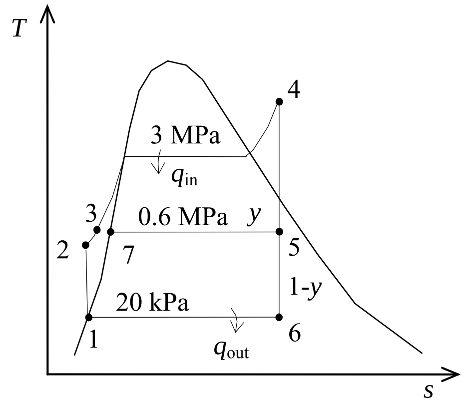
$$\eta_{\mathrm{th}} = \frac{w_{\mathrm{net}}}{q_{\mathrm{in}}} = \frac{737.8}{2353} = 0.3136$$

10-56 An ideal regenerative Rankine cycle with a closed feedwater heater is considered. The change in thermal efficiency when the steam serving the closed feedwater heater is extracted at 600 kPa rather than 1000 kPa is to be determined.

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

Analysis From the steam tables (Tables A-4, A-5, and A-6 or EES),





$$h_1 = h_{f\,@\,20\,\text{kPa}} = 251.42 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,20\,\text{kPa}} = 0.001017 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017 \text{ m}^3\text{/kg})(3000-20)\text{kPa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 3.03 \text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 3.03 = 254.45 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 3000 \text{ kPa} \\ T_4 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_4 = 3116.1 \text{ kJ/kg} \\ s_4 = 6.7450 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 600 \text{ kPa} \\ s_5 = s_4 \end{array}\right\} \begin{array}{l} x_5 = \dfrac{s_5 - s_f}{s_{fg}} = \dfrac{6.7450-1.9308}{4.8285} = 0.9970 \\ h_5 = h_f + x_5 h_{fg} = 670.38 + (0.9970)(2085.8) = 2750.0 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_6 = 20 \text{ kPa} \\ s_6 = s_4 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.7450-0.8320}{7.0752} = 0.8357 \\ h_6 = h_f + x_6 h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7 \text{ kJ/kg} \end{array}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.

$$\left.\begin{array}{l} P_7 = 600 \text{ kPa} \\ x_7 = 0 \end{array}\right\} \begin{array}{l} h_7 = 670.38 \text{ kJ/kg} \\ T_7 = 158.8°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 3000 \text{ kPa} \\ T_3 = T_7 = 158.8°\text{C} \end{array}\right\} h_3 = 671.79 \text{ kJ/kg}$$

An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5 / \dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$
$$y h_5 + 1 h_2 = 1 h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{671.79 - 254.45}{2750.0 - 670.38} = 0.2007$$

Then,

$$w_{\text{T,out}} = h_4 - h_5 + (1-y)(h_5 - h_6) = 3116.1 - 2750.0 + (1 - 0.2007)(2750.0 - 2221.7) = 788.4 \text{ kJ/kg}$$
$$w_{\text{P,in}} = 3.03 \text{ kJ/kg}$$
$$q_{\text{in}} = h_4 - h_3 = 3116.1 - 671.79 = 2444 \text{ kJ/kg}$$

Also,

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}} = 788.4 - 3.03 = 785.4 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{785.4}{2444} = \mathbf{0.3213}$$

When the steam serving the closed feedwater heater is extracted at 600 kPa rather than 1000 kPa, the thermal efficiency increases from **0.3136** to **0.3213**. This is an increase of **2.5%**.

10-57 EES The optimum bleed pressure for the open feedwater heater that maximizes the thermal efficiency of the cycle is to be determined by EES.

Analysis The EES program used to solve this problem as well as the solutions are given below.

```
"Given"
P[4]=3000 [kPa]
T[4]=350 [C]
P[5]=600 [kPa]
P[6]=20 [kPa]

P[3]=P[4]
P[2]=P[3]
P[7]=P[5]
P[1]=P[6]

"Analysis"
Fluid$='steam_iapws'

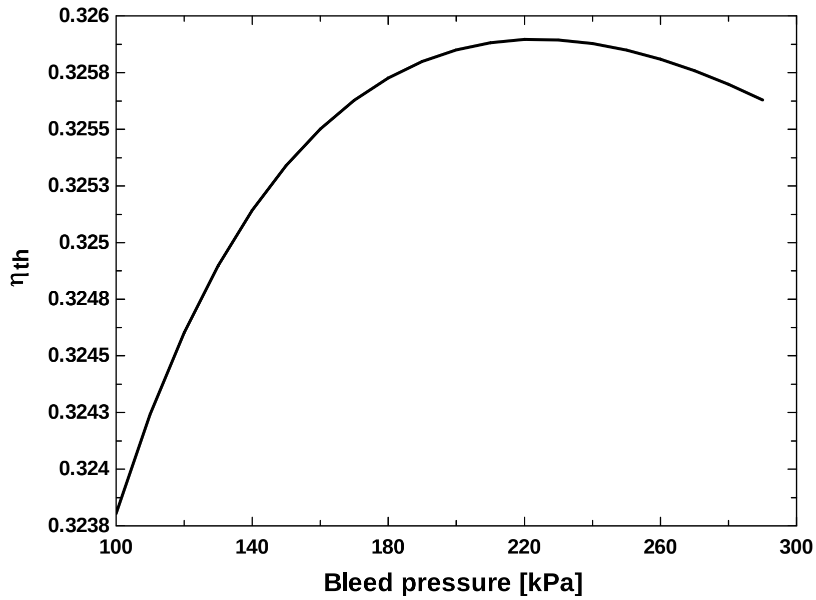
"pump I"
x[1]=0
h[1]=enthalpy(Fluid$, P=P[1], x=x[1])
v[1]=volume(Fluid$, P=P[1], x=x[1])
w_p_in=v[1]*(P[2]-P[1])
h[2]=h[1]+w_p_in

"turbine"
h[4]=enthalpy(Fluid$, P=P[4], T=T[4])
s[4]=entropy(Fluid$, P=P[4], T=T[4])
s[5]=s[4]
h[5]=enthalpy(Fluid$, P=P[5], s=s[5])
T[5]=temperature(Fluid$, P=P[5], s=s[5])
x[5]=quality(Fluid$, P=P[5], s=s[5])
s[6]=s[4]
h[6]=enthalpy(Fluid$, P=P[6], s=s[6])
x[6]=quality(Fluid$, P=P[6], s=s[6])

"closed feedwater heater"
x[7]=0
h[7]=enthalpy(Fluid$, P=P[7], x=x[7])
T[7]=temperature(Fluid$, P=P[7], x=x[7])
T[3]=T[7]
h[3]=enthalpy(Fluid$, P=P[3], T=T[3])
y=(h[3]-h[2])/(h[5]-h[7])  "y=m_dot_5/m_dot_4"

"cycle"
q_in=h[4]-h[3]
w_T_out=h[4]-h[5]+(1-y)*(h[5]-h[6])
w_net=w_T_out-w_p_in
Eta_th=w_net/q_in
```

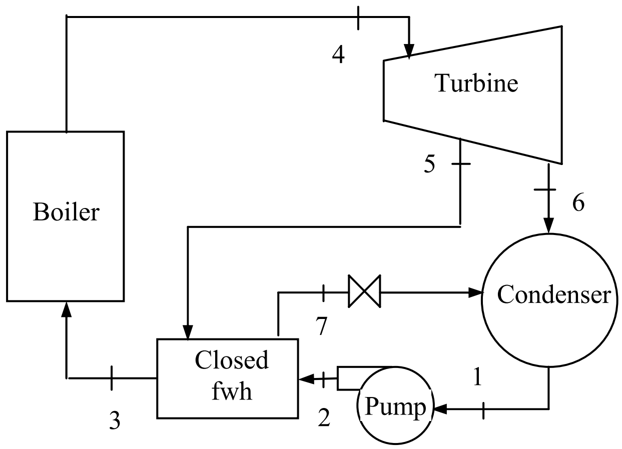
P_6 [kPa]	η_{th}
100	0.32380
110	0.32424
120	0.32460
130	0.32490
140	0.32514
150	0.32534
160	0.32550
170	0.32563
180	0.32573
190	0.32580
200	0.32585
210	0.32588
220	**0.32590**
230	0.32589
240	0.32588
250	0.32585
260	0.32581
270	0.32576
280	0.32570
290	0.32563

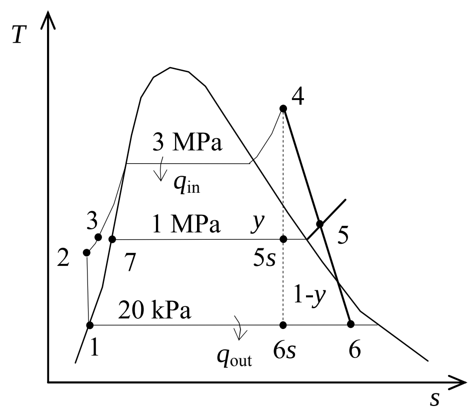


10-58 A regenerative Rankine cycle with a closed feedwater heater is considered. The thermal efficiency is to be determined.

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

Analysis From the steam tables (Tables A-4, A-5, and A-6 or EES),





$$h_1 = h_{f@\,20\,\text{kPa}} = 251.42\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,20\,\text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017\ \text{m}^3/\text{kg})(3000-20)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 3.03\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 3.03 = 254.45\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 3000\ \text{kPa} \\ T_4 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_4 = 3116.1\ \text{kJ/kg} \\ s_4 = 6.7450\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 1000\ \text{kPa} \\ s_{5s} = s_4 \end{array}\right\} h_{5s} = 2851.9\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 20\ \text{kPa} \\ s_{6s} = s_4 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{6.7450 - 0.8320}{7.0752} = 0.8357 \\ h_{6s} = h_f + x_{6s} h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7\ \text{kJ/kg} \end{array}$$

$$\eta_{\text{T}} = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow h_5 = h_4 - \eta_{\text{T}}(h_4 - h_{5s}) = 3116.1 - (0.90)(3116.1 - 2851.9) = 2878.3\ \text{kJ/kg}$$

$$\eta_{\text{T}} = \frac{h_4 - h_6}{h_4 - h_{6s}} \longrightarrow h_6 = h_4 - \eta_{\text{T}}(h_4 - h_{6s}) = 3116.1 - (0.90)(3116.1 - 2221.7) = 2311.1\ \text{kJ/kg}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.

$$\left.\begin{array}{l} P_7 = 1000 \text{ kPa} \\ x_7 = 0 \end{array}\right\} \begin{array}{l} h_7 = 762.51 \text{ kJ/kg} \\ T_7 = 179.9°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 3000 \text{ kPa} \\ T_3 = T_7 = 209.9°\text{C} \end{array}\right\} h_3 = 763.53 \text{ kJ/kg}$$

An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5 / \dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$

$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$

$$y h_5 + 1 h_2 = 1 h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{763.53 - 254.45}{2878.3 - 762.51} = 0.2406$$

Then,

$$w_{\text{T,out}} = h_4 - h_5 + (1 - y)(h_5 - h_6) = 3116.1 - 2878.3 + (1 - 0.2406)(2878.3 - 2311.1) = 668.5 \text{ kJ/kg}$$

$$w_{\text{P,in}} = 3.03 \text{ kJ/kg}$$

$$q_{\text{in}} = h_4 - h_3 = 3116.1 - 763.53 = 2353 \text{ kJ/kg}$$

Also,

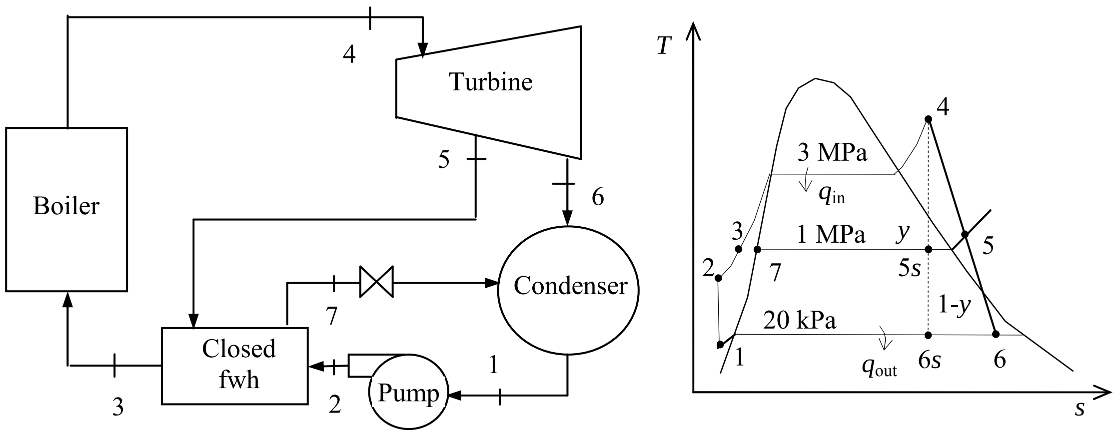
$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}} = 668.5 - 3.03 = 665.5 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{665.5}{2353} = \mathbf{0.2829}$$

10-59 A regenerative Rankine cycle with a closed feedwater heater is considered. The thermal efficiency is to be determined.

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

Analysis From the steam tables (Tables A-4, A-5, and A-6 or EES),



When the liquid enters the pump 10°C cooler than a saturated liquid at the condenser pressure, the enthalpies become

$$\left.\begin{array}{l} P_1 = 20\text{ kPa} \\ T_1 = T_{\text{sat @ 20 kPa}} - 10 = 60.06 - 10 \cong 50°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f\,@\,50°\text{C}} = 209.34\text{ kJ/kg} \\ \boldsymbol{v}_1 \cong \boldsymbol{v}_{f\,@\,50°\text{C}} = 0.001012\text{ m}^3\text{/kg} \end{array}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001012\text{ m}^3\text{/kg})(3000 - 20)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 3.02\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 209.34 + 3.02 = 212.36\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 3000\text{ kPa} \\ T_4 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_4 = 3116.1\text{ kJ/kg} \\ s_4 = 6.7450\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 1000\text{ kPa} \\ s_{5s} = s_4 \end{array}\right\} h_{5s} = 2851.9\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 20\text{ kPa} \\ s_{6s} = s_4 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{6.7450 - 0.8320}{7.0752} = 0.8357 \\ h_{6s} = h_f + x_{6s}h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7\text{ kJ/kg} \end{array}$$

$$\eta_{\text{T}} = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow h_5 = h_4 - \eta_{\text{T}}(h_4 - h_{5s}) = 3116.1 - (0.90)(3116.1 - 2851.9) = 2878.3\text{ kJ/kg}$$

$$\eta_{\text{T}} = \frac{h_4 - h_6}{h_4 - h_{6s}} \longrightarrow h_6 = h_4 - \eta_{\text{T}}(h_4 - h_{6s}) = 3116.1 - (0.90)(3116.1 - 2221.7) = 2311.1\text{ kJ/kg}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.

$$\left.\begin{array}{l} P_7 = 1000 \text{ kPa} \\ x_7 = 0 \end{array}\right\} \begin{array}{l} h_7 = 762.51 \text{ kJ/kg} \\ T_7 = 179.9°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 3000 \text{ kPa} \\ T_3 = T_7 = 209.9°\text{C} \end{array}\right\} h_3 = 763.53 \text{ kJ/kg}$$

An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5 / \dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$
$$y h_5 + 1 h_2 = 1 h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{763.53 - 212.36}{2878.3 - 762.51} = 0.2605$$

Then,

$$w_{\text{T,out}} = h_4 - h_5 + (1-y)(h_5 - h_6) = 3116.1 - 2878.3 + (1 - 0.2605)(2878.3 - 2311.1) = 657.2 \text{ kJ/kg}$$
$$w_{\text{P,in}} = 3.03 \text{ kJ/kg}$$
$$q_{\text{in}} = h_4 - h_3 = 3116.1 - 763.53 = 2353 \text{ kJ/kg}$$

Also,

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}} = 657.2 - 3.03 = 654.2 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{654.2}{2353} = \mathbf{0.2781}$$

10-60 EES The effect of pressure drop and non-isentropic turbine on the rate of heat input is to be determined for a given power plant.

Analysis The EES program used to solve this problem as well as the solutions are given below.

```
"Given"
P[3]=3000 [kPa]
DELTAP_boiler=10 [kPa]
P[4]=P[3]-DELTAP_boiler
T[4]=350 [C]
P[5]=1000 [kPa]
P[6]=20 [kPa]
eta_T=0.90

P[2]=P[3]
P[7]=P[5]
P[1]=P[6]

"Analysis"
Fluid$='steam_iapws'
"(a)"
"pump I"
x[1]=0
h[1]=enthalpy(Fluid$, P=P[1], x=x[1])
v[1]=volume(Fluid$, P=P[1], x=x[1])
w_p_in=v[1]*(P[2]-P[1])
h[2]=h[1]+w_p_in
"turbine"
h[4]=enthalpy(Fluid$, P=P[4], T=T[4])
s[4]=entropy(Fluid$, P=P[4], T=T[4])
s[5]=s[4]
h_s[5]=enthalpy(Fluid$, P=P[5], s=s[5])
T[5]=temperature(Fluid$, P=P[5], s=s[5])
x_s[5]=quality(Fluid$, P=P[5], s=s[5])
s[6]=s[4]
h_s[6]=enthalpy(Fluid$, P=P[6], s=s[6])
x_s[6]=quality(Fluid$, P=P[6], s=s[6])

h[5]=h[4]-eta_T*(h[4]-h_s[5])
h[6]=h[4]-eta_T*(h[4]-h_s[6])
x[5]=quality(Fluid$, P=P[5], h=h[5])
x[6]=quality(Fluid$, P=P[6], h=h[6])

"closed feedwater heater"
x[7]=0
h[7]=enthalpy(Fluid$, P=P[7], x=x[7])
T[7]=temperature(Fluid$, P=P[7], x=x[7])
T[3]=T[7]
h[3]=enthalpy(Fluid$, P=P[3], T=T[3])
y=(h[3]-h[2])/(h[5]-h[7])  "y=m_dot_5/m_dot_4"

"cycle"
q_in=h[4]-h[3]
w_T_out=h[4]-h[5]+(1-y)*(h[5]-h[6])
w_net=w_T_out-w_p_in
Eta_th=w_net/q_in
```

Solution with 10 kPa pressure drop in the boiler:

DELTAP_boiler=10 [kPa]
Eta_th=0.2827
P[3]=3000 [kPa]
q_in=2352.8 [kJ/kg]
w_p_in=3.031 [m^3-kPa/kg]
y=0.2405
eta_T=0.9
Fluid$='steam_iapws'
P[4]=2990 [kPa]
w_net=665.1 [kJ/kg]
w_T_out=668.1 [kJ/kg]

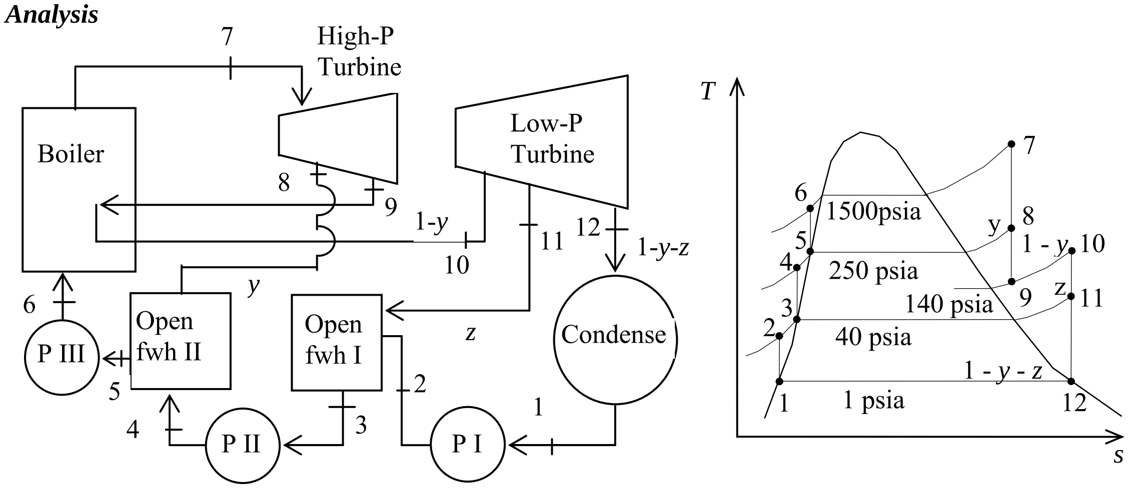
Solution without any pressure drop in the boiler:

DELTAP_boiler=0 [kPa]
Eta_th=0.3136
P[3]=3000 [kPa]
q_in=2352.5 [kJ/kg]
w_p_in=3.031 [m^3-kPa/kg]
y=0.2437
eta_T=1
Fluid$='steam_iapws'
P[4]=3000 [kPa]
w_net=737.8 [kJ/kg]
w_T_out=740.9 [kJ/kg]

10-61E A steam power plant operates on an ideal reheat-regenerative Rankine cycle with one reheater and two open feedwater heaters. The mass flow rate of steam through the boiler, the net power output of the plant, 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-4E, A-5E, and A-6E),

$$h_1 = h_{f\,@\,1\text{ psia}} = 69.72\text{ Btu/lbm}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,1\text{ psia}} = 0.01614\text{ ft}^3/\text{lbm}$$

$$\begin{aligned} w_{pI,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01614\text{ ft}^3/\text{lbm})(40-1\text{ psia})\left(\frac{1\text{ Btu}}{5.4039\text{ psia}\cdot\text{ft}^3}\right) \\ &= 0.12\text{ Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 69.72 + 0.12 = 69.84\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 40\text{ psia} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,40\text{ psia}} = 236.14\text{ Btu/lbm} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f\,@\,40\text{ psia}} = 0.01715\text{ ft}^3/\text{lbm} \end{array}$$

$$\begin{aligned} w_{pII,\text{in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= (0.01715\text{ ft}^3/\text{lbm})(250-40\text{ psia})\left(\frac{1\text{ Btu}}{5.4039\text{ psia}\cdot\text{ft}^3}\right) \\ &= 0.67\text{ Btu/lbm} \end{aligned}$$

$$h_4 = h_3 + w_{pII,in} = 236.14 + 0.67 = 236.81\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_5 = 250\text{ psia} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_5 = h_{f\,@\,250\text{ psia}} = 376.09\text{ Btu/lbm} \\ \boldsymbol{v}_5 = \boldsymbol{v}_{f\,@\,250\text{ psia}} = 0.01865\text{ ft}^3/\text{lbm} \end{array}$$

$$\begin{aligned} w_{pIII,\text{in}} &= \boldsymbol{v}_5(P_6 - P_5) \\ &= (0.01865\text{ ft}^3/\text{lbm})(1500-250\text{ psia})\left(\frac{1\text{ Btu}}{5.4039\text{ psia}\cdot\text{ft}^3}\right) \\ &= 4.31\text{ Btu/lbm} \end{aligned}$$

$$h_6 = h_5 + w_{pIII,\text{in}} = 376.09 + 4.31 = 380.41\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_7 = 1500\text{ psia} \\ T_7 = 1100°\text{F} \end{array}\right\} \begin{array}{l} h_7 = 1550.5\text{ Btu/lbm} \\ s_7 = 1.6402\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_8 = 250\text{ psia} \\ s_8 = s_7 \end{array}\right\} h_8 = 1308.5\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_9 = 140 \text{ psia} \\ s_9 = s_7 \end{array}\right\} h_9 = 1248.8 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_{10} = 140 \text{ psia} \\ T_{10} = 1000°\text{F} \end{array}\right\} \begin{array}{l} h_{10} = 1531.3 \text{ Btu/lbm} \\ s_{10} = 1.8832 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_{11} = 40 \text{ psia} \\ s_{11} = s_{10} \end{array}\right\} h_{11} = 1356.0 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_{12} = 1 \text{ psia} \\ s_{12} = s_{10} \end{array}\right\} \begin{array}{l} x_{12} = \dfrac{s_{12} - s_f}{s_{fg}} = \dfrac{1.8832 - 0.13262}{1.84495} = 0.9488 \\ h_{12} = h_f + x_{12} h_{fg} = 69.72 + (0.9488)(1035.7) \\ \quad = 1052.4 \text{ Btu/lbm} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0$$

FWH-2: $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_4 h_4 = \dot{m}_5 h_5 \longrightarrow y h_8 + (1-y) h_4 = 1(h_5)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_5$). Solving for y,

$$y = \frac{h_5 - h_4}{h_8 - h_4} = \frac{376.09 - 236.81}{1308.5 - 236.81} = 0.1300$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0$$

FWH-1 $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_{11} h_{11} + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow z h_{11} + (1-y-z) h_2 = (1-y) h_3$$

where z is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for z,

$$z = \frac{h_3 - h_2}{h_{11} - h_2}(1-y) = \frac{236.14 - 69.84}{1356.0 - 69.84}(1 - 0.1300) = 0.1125$$

Then,

$$q_{\text{in}} = h_7 - h_6 + (1-y)(h_{10} - h_9) = 1550.5 - 380.41 + (1 - 0.1300)(1531.3 - 1248.8) = 1415.8 \text{ Btu/lbm}$$
$$q_{\text{out}} = (1-y-z)(h_{12} - h_1) = (1 - 0.1300 - 0.1125)(1052.4 - 69.72) = 744.4 \text{ Btu/lbm}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1415.8 - 744.4 = 671.4 \text{ Btu/lbm}$$

and

$$\dot{m} = \frac{\dot{Q}_{\text{in}}}{q_{\text{in}}} = \frac{4 \times 10^5 \text{ Btu/s}}{1415.8 \text{ Btu/lbm}} = \mathbf{282.5\ lbm/s}$$

(*b*) $$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (282.5 \text{ lbm/s})(671.4 \text{ Btu/lbm})\left(\frac{1.055 \text{ kJ}}{1 \text{ Btu}}\right) = \mathbf{200.1\ MW}$$

(*c*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{744.4 \text{ Btu/lbm}}{1415.8 \text{ Btu/lbm}} = \mathbf{47.4\%}$$

10-62 A steam power plant that operates on an ideal regenerative Rankine cycle with a closed feedwater heater is considered. The temperature of the steam at the inlet of the closed feedwater heater, the mass flow rate of the steam extracted from the turbine for the closed feedwater heater, the net power output, and the thermal efficiency are to be determined.

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

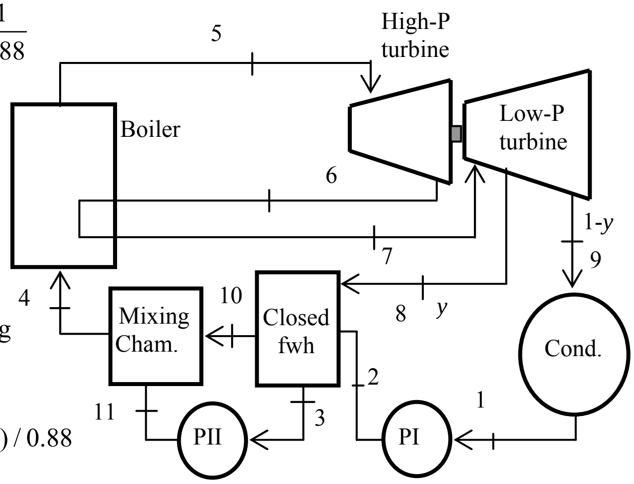
Analysis (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f @ 20\text{ kPa}} = 251.42 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f @ 20\text{ kPa}} = 0.001017 \text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{pI,\text{in}} &= \boldsymbol{v}_1 (P_2 - P_1)/\eta_p \\ &= (0.001017 \text{ m}^3/\text{kg})(12{,}500 - 20 \text{ kPa})\frac{1}{0.88} \\ &= 14.43 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} h_2 &= h_1 + w_{pI,\text{in}} \\ &= 251.42 + 14.43 \\ &= 265.85 \text{ kJ/kg} \end{aligned}$$



$$\left.\begin{array}{l} P_3 = 1 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f @ 1\text{ MPa}} = 762.51 \text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f @ 1\text{ MPa}} = 0.001127 \text{ m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{pII,\text{in}} &= \boldsymbol{v}_3 (P_{11} - P_3)/\eta_p \\ &= (0.001127 \text{ m}^3/\text{kg})(12{,}500 - 1000 \text{ kPa})/0.88 \\ &= 14.73 \text{ kJ/kg} \end{aligned}$$

$$h_{11} = h_3 + w_{pII,\text{in}} = 762.51 + 14.73 = 777.25 \text{ kJ/kg}$$

Also, $h_4 = h_{10} = h_{11} = 777.25$ kJ/kg since the two fluid streams which are being mixed have the same enthalpy.

$$\left.\begin{array}{l} P_5 = 12.5 \text{ MPa} \\ T_5 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3476.5 \text{ kJ/kg} \\ s_5 = 6.6317 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 5 \text{ MPa} \\ s_6 = s_5 \end{array}\right\} h_{6s} = 3185.6 \text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow \begin{aligned}[t] h_6 &= h_5 - \eta_T (h_5 - h_{6s}) \\ &= 3476.5 - (0.88)(3476.5 - 3185.6) = 3220.5 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_7 = 5 \text{ MPa} \\ T_7 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3550.9 \text{ kJ/kg} \\ s_7 = 7.1238 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 1 \text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_{8s} = 3051.1 \text{ kJ/kg}$$

$$\eta_T = \frac{h_7 - h_8}{h_7 - h_{8s}} \longrightarrow \begin{aligned}[t] h_8 &= h_7 - \eta_T (h_7 - h_{8s}) \\ &= 3550.9 - (0.88)(3550.9 - 3051.1) = 3111.1 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_8 = 1 \text{ MPa} \\ h_8 = 3111.1 \text{ kJ/kg} \end{array}\right\} T_8 = \mathbf{328°C}$$

$$\left.\begin{array}{l} P_9 = 20 \text{ kPa} \\ s_9 = s_7 \end{array}\right\} h_{9s} = 2347.9 \text{ kJ/kg}$$

$$\eta_T = \frac{h_7 - h_9}{h_7 - h_{9s}} \longrightarrow h_9 = h_7 - \eta_T (h_7 - h_{9s})$$
$$= 3550.9 - (0.88)(3550.9 - 2347.9) = 2492.2 \text{ kJ/kg}$$

The fraction of steam extracted from the low pressure turbine for closed feedwater heater is determined from the steady-flow energy balance equation applied to the feedwater heater. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

$$(1 - y)(h_{10} - h_2) = y(h_8 - h_3)$$
$$(1 - y)(777.25 - 265.85) = y(3111.1 - 762.51) \longrightarrow y = 0.1788$$

The corresponding mass flow rate is

$$\dot{m}_8 = y\dot{m}_5 = (0.1788)(24 \text{ kg/s}) = \mathbf{4.29 \text{ kg/s}}$$

(*c*) Then,

$$q_{\text{in}} = h_5 - h_4 + h_7 - h_6 = 3476.5 - 777.25 + 3550.9 - 3220.5 = 3029.7 \text{ kJ/kg}$$
$$q_{\text{out}} = (1 - y)(h_9 - h_1) = (1 - 0.1788)(2492.2 - 251.42) = 1840.1 \text{ kJ/kg}$$

and

$$\dot{W}_{\text{net}} = \dot{m}(q_{\text{in}} - q_{\text{out}}) = (24 \text{ kg/s})(3029.7 - 1840.1)\text{kJ/kg} = \mathbf{28{,}550 \text{ kW}}$$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1840.1 \text{ kJ/kg}}{3029.7 \text{ kJ/kg}} = 0.393 = \mathbf{39.3\%}$$

Second-Law Analysis of Vapor Power Cycles

10-63C In the simple ideal Rankine cycle, irreversibilities occur during heat addition and heat rejection processes in the boiler and the condenser, respectively, and both are due to temperature difference. Therefore, the irreversibilities can be decreased and thus the 2nd law efficiency can be increased by minimizing the temperature differences during heat transfer in the boiler and the condenser. One way of doing that is regeneration.

10-64 The exergy destruction associated with the heat rejection process in Prob. 10-25 is to be determined for the specified source and sink temperatures. The exergy of the steam at the boiler exit is also to be determined.

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

Analysis From Problem 10-25,

$$\begin{aligned} s_1 &= s_2 = s_{f@10\text{kPa}} = 0.6492 \text{ kJ/kg}\cdot\text{K} \\ s_3 &= s_4 = 6.8000 \text{ kJ/kg}\cdot\text{K} \\ h_3 &= 3411.4 \text{ kJ/kg} \\ q_{\text{out}} &= 1961.8 \text{ kJ/kg} \end{aligned}$$

The exergy destruction associated with the heat rejection process is

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = (290 \text{ K})\left(0.6492 - 6.8000 + \frac{1961.8 \text{ kJ/kg}}{290 \text{ K}}\right) = \mathbf{178.0 \text{ kJ/kg}}$$

The exergy of the steam at the boiler exit is simply the flow exergy,

$$\begin{aligned} \psi_3 &= (h_3 - h_0) - T_0(s_3 - s_0) + \frac{V_3^2}{2}^{\nearrow 0} + qz_3^{\nearrow 0} \\ &= (h_3 - h_0) - T_0(s_3 - s_0) \end{aligned}$$

where

$$\begin{aligned} h_0 &= h_{@(290 \text{ K, } 100 \text{ kPa})} \cong h_{f\ @\ 290 \text{ K}} = 71.95 \text{ kJ/kg} \\ s_0 &= s_{@(290 \text{ K,} 100 \text{ kPa})} \cong s_{f\ @\ 290 \text{ K}} = 0.2533 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

Thus,

$$\psi_3 = (3411.4 - 71.95) \text{ kJ/kg} - (290 \text{ K})(6.800 - 0.2532) \text{ kJ/kg}\cdot\text{K} = \mathbf{1440.9\ \ kJ/kg}$$

10-65E The exergy destructions associated with each of the processes of the Rankine cycle described in Prob. 10-15E are to be determined for the specified source and sink temperatures.

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

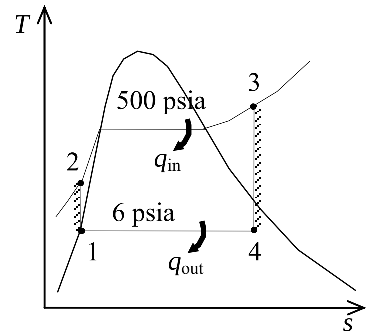
Analysis From Problem 10-15E,

$$s_1 = s_2 = s_{f\,@\,6\,\text{psia}} = 0.24739\ \text{Btu/lbm}\cdot\text{R}$$

$$s_3 = s_4 = 1.8075\ \text{Btu/lbm}\cdot\text{R}$$

$$q_{\text{in}} = h_3 - h_2 = 1630.0 - 139.52 = 1490.5\ \text{Btu/lbm}$$

$$q_{\text{out}} = h_4 - h_2 = 1120.4 - 138.02 = 982.38\ \text{Btu/lbm}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{destroyed},23} = T_0\left(s_3 - s_2 - \frac{q_{\text{in}}}{T_{\text{source}}}\right) = (500\ \text{R})\left(1.8075 - 0.24739 - \frac{1490.5\ \text{Btu/lbm}}{1960\ \text{R}}\right) = \mathbf{399.8\ Btu/lbm}$$

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{\text{out}}}{T_{\text{sink}}}\right) = (500\ \text{R})\left(0.24739 - 1.8075 + \frac{982.38\ \text{Btu/lbm}}{500\ \text{R}}\right) = \mathbf{202.3\ Btu/lbm}$$

Processes 1-2 and 3-4 are isentropic, and thus

$$x_{\text{destroyed},12} = \mathbf{0}$$

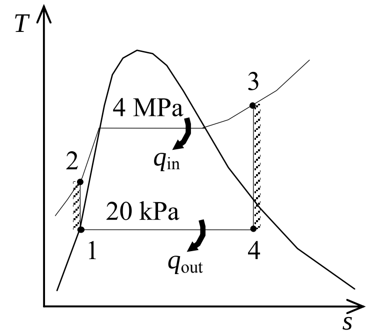
$$x_{\text{destroyed},34} = \mathbf{0}$$

10-66 The exergy destructions associated with each of the processes of the Rankine cycle described in Prob. 10-17 are to be determined for the specified source and sink temperatures.

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

Analysis From Problem 10-17,

$$s_1 = s_2 = s_{f\,@\,20\,\text{kPa}} = 0.8320\ \text{kJ/kg}\cdot\text{K}$$
$$s_3 = s_4 = 7.6214\ \text{kJ/kg}\cdot\text{K}$$
$$q_{\text{in}} = h_3 - h_2 = 3906.3 - 255.47 = 3650.8\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2513.7 - 251.42 = 2262.3\ \text{kJ/kg}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{destroyed},23} = T_0\left(s_3 - s_2 - \frac{q_{\text{in}}}{T_{\text{source}}}\right) = (288\ \text{K})\left(7.6214 - 0.8320 - \frac{3650.8\ \text{kJ/kg}}{1023\ \text{K}}\right) = \mathbf{927.6\ kJ/kg}$$

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{\text{out}}}{T_{\text{sink}}}\right) = (288\ \text{K})\left(0.8320 - 7.6214 + \frac{2262.3\ \text{kJ/kg}}{288\ \text{K}}\right) = \mathbf{307.0\ kJ/kg}$$

Processes 1-2 and 3-4 are isentropic, and thus

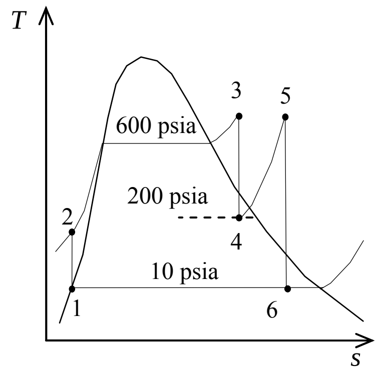
$$x_{\text{destroyed},12} = \mathbf{0}$$
$$x_{\text{destroyed},34} = \mathbf{0}$$

10-67E The exergy destructions associated with each of the processes of the ideal reheat Rankine cycle described in Prob. 10-36E are to be determined for the specified source and sink temperatures.

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

Analysis From Problem 10-36E,

$$s_1 = s_2 = s_{f\,@\,10\,\text{psia}} = 0.28362\ \text{Btu/lbm}\cdot\text{R}$$
$$s_3 = s_4 = 1.5325\ \text{Btu/lbm}\cdot\text{R}$$
$$s_5 = s_6 = 1.6771\ \text{Btu/lbm}\cdot\text{R}$$
$$q_{\text{in,2-3}} = h_3 - h_2 = 1289.9 - 163.06 = 1126.8\ \text{Btu/lbm}$$
$$q_{\text{in, 4-5}} = h_5 - h_4 = 1322.3 - 1187.5 = 134.8\ \text{Btu/lbm}$$
$$q_{\text{out}} = 909.7\ \text{Btu/lbm}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$x_{\text{destroyed, 23}} = T_0\left(s_3 - s_2 - \frac{q_{\text{in, 2-3}}}{T_{\text{source}}}\right) = (537\ \text{R})\left(1.5325 - 0.28362 - \frac{1126.8\ \text{Btu/lbm}}{1160\ \text{R}}\right) = \mathbf{149.0\ Btu/lbm}$$

$$x_{\text{destroyed, 45}} = T_0\left(s_5 - s_4 - \frac{q_{\text{in, 4-5}}}{T_{\text{source}}}\right) = (537\ \text{R})\left(1.6771 - 1.5325 - \frac{134.8\ \text{Btu/lbm}}{1160\ \text{R}}\right) = \mathbf{15.2\ Btu/lbm}$$

$$x_{\text{destroyed, 61}} = T_0\left(s_1 - s_6 + \frac{q_{\text{out}}}{T_{\text{sink}}}\right) = (537\ \text{R})\left(0.28362 - 1.6771 + \frac{909.7\ \text{Btu/lbm}}{537\ \text{R}}\right) = \mathbf{161.4\ Btu/lbm}$$

Processes 1-2, 3-4, and 5-6 are isentropic, and thus,

$$x_{\text{destroyed, 12}} = \mathbf{0}$$
$$x_{\text{destroyed, 34}} = \mathbf{0}$$
$$x_{\text{destroyed, 56}} = \mathbf{0}$$

10-68 The exergy destructions associated with each of the processes of the reheat Rankine cycle described in Prob. 10-34 are to be determined for the specified source and sink temperatures.

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

Analysis From Problem 10-34,

$$s_1 = s_2 = s_{f\,@20\text{kPa}} = 0.8320 \text{ kJ/kg}\cdot\text{K}$$
$$s_3 = s_4 = 6.7266 \text{ kJ/kg}\cdot\text{K}$$
$$s_5 = s_6 = 7.2359 \text{ kJ/kg}\cdot\text{K}$$
$$q_{23,\text{in}} = 3399.5 - 259.54 = 3140.0 \text{ kJ/kg}$$
$$q_{45,\text{in}} = 3457.2 - 3105.1 = 352.1 \text{ kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2385.2 - 251.42 = 2133.8 \text{ kJ/kg}$$

Processes 1-2, 3-4, and 5-6 are isentropic. Thus, $i_{12} = i_{34} = i_{56} = \mathbf{0}$. Also,

$$x_{\text{destroyed},23} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = (300\text{ K})\left(6.7266 - 0.8320 + \frac{-3140.0\text{ kJ/kg}}{1800\text{ K}}\right) = \mathbf{1245.0\ kJ/kg}$$

$$x_{\text{destroyed},45} = T_0\left(s_5 - s_4 + \frac{q_{R,45}}{T_R}\right) = (300\text{ K})\left(7.2359 - 6.7266 + \frac{-352.5\text{ kJ/kg}}{1800\text{ K}}\right) = \mathbf{94.1\ kJ/kg}$$

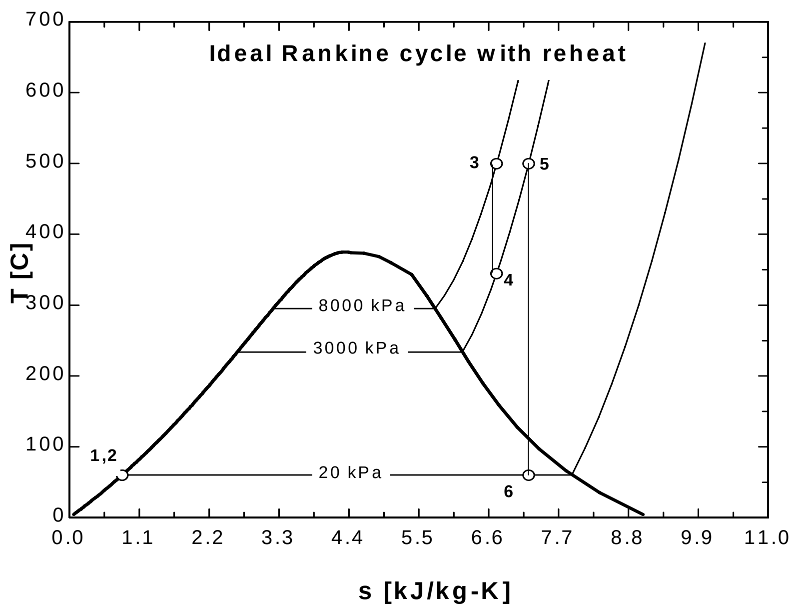
$$x_{\text{destroyed},61} = T_0\left(s_1 - s_6 + \frac{q_{R,61}}{T_R}\right) = (300\text{ K})\left(0.8320 - 7.2359 + \frac{2133.8\text{ kJ/kg}}{300\text{ K}}\right) = \mathbf{212.6\ kJ/kg}$$

10-69 EES Problem 10-68 is reconsidered. The problem is to be solved by the diagram window data entry feature of EES by including the effects of the turbine and pump efficiencies. Also, the *T-s* diagram is to be plotted.

Analysis The problem is solved using EES, and the solution is given below.

```
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end
"Input Data - from diagram window"
{P[6] = 20 [kPa]
P[3] = 8000 [kPa]
T[3] = 500 [C]
P[4] = 3000 [kPa]
T[5] = 500 [C]
Eta_t = 100/100 "Turbine isentropic efficiency"
Eta_p = 100/100 "Pump isentropic efficiency"}
"Data for the irreversibility calculations:"
T_o = 300 [K]
T_R_L = 300 [K]
T_R_H = 1800 [K]
"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[6]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,T=T[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"
"Low Pressure Turbine analysis"
P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
```

```
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
"Boiler analysis"
Q_in + h[2]+h[4]=h[3]+h[5]"SSSF First Law for the Boiler"
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])
"Cycle Statistics"
W_net=W_t_hp+W_t_lp-W_p
Eff=W_net/Q_in
"The irreversibilities (or exergy destruction) for each of the processes are:"
q_R_23 = - (h[3] - h[2]) "Heat transfer for the high temperature reservoir to process 2-3"
i_23 = T_o*(s[3] -s[2] + q_R_23/T_R_H)
q_R_45 = - (h[5] - h[4]) "Heat transfer for the high temperature reservoir to process 4-5"
i_45 = T_o*(s[5] -s[4] + q_R_45/T_R_H)
q_R_61 =  (h[6] - h[1]) "Heat transfer to the low temperature reservoir in process 6-1"
i_61 = T_o*(s[1] -s[6] + q_R_61/T_R_L)
i_34 = T_o*(s[4] -s[3])
i_56 = T_o*(s[6] -s[5])
i_12 = T_o*(s[2] -s[1])
```



SOLUTION

Eff=0.389
Eta_p=1
Eta_t=1
Fluid$='Steam_IAPWS'
h[1]=251.4 [kJ/kg]
h[2]=259.5 [kJ/kg]
h[3]=3400 [kJ/kg]
h[4]=3105 [kJ/kg]
h[5]=3457 [kJ/kg]
h[6]=2385 [kJ/kg]
hs[4]=3105 [kJ/kg]
hs[6]=2385 [kJ/kg]
i_12=0.012 [kJ/kg]
i_23=1245.038 [kJ/kg]
i_34=-0.000 [kJ/kg]
i_45=94.028 [kJ/kg]
i_56=0.000 [kJ/kg]
i_61=212.659 [kJ/kg]
P[1]=20 [kPa]
P[2]=8000 [kPa]
P[3]=8000 [kPa]
P[4]=3000 [kPa]
P[5]=3000 [kPa]
P[6]=20 [kPa]
Q_in=3493 [kJ/kg]
Q_out=2134 [kJ/kg]
q_R_23=-3140 [kJ/kg]
q_R_45=-352.5 [kJ/kg]
q_R_61=2134 [kJ/kg]
s[1]=0.832 [kJ/kg-K]
s[2]=0.8321 [kJ/kg-K]
s[3]=6.727 [kJ/kg-K]
s[4]=6.727 [kJ/kg-K]
s[5]=7.236 [kJ/kg-K]
s[6]=7.236 [kJ/kg-K]
s_s[4]=6.727 [kJ/kg-K]
s_s[6]=7.236 [kJ/kg-K]
T[1]=60.06 [C]
T[2]=60.4 [C]
T[3]=500 [C]
T[4]=345.2 [C]
T[5]=500 [C]
T[6]=60.06 [C]
Ts[4]=345.2 [C]
Ts[6]=60.06 [C]
T_o=300 [K]
T_R_H=1800 [K]
T_R_L=300 [K]
v[1]=0.001017 [m^3/kg]
v[2]=0.001014 [m^3/kg]
v[3]=0.04177 [m^3/kg]
v[4]=0.08968 [m^3/kg]
vs[6]=6.922 [m^3/kg]
W_net=1359 [kJ/kg]
W_p=8.117 [kJ/kg]
W_p_s=8.117 [kJ/kg]
W_t_hp=294.8 [kJ/kg]
W_t_lp=1072 [kJ/kg]
x6s$=''
x[1]=0
x[6]=0.9051

10-70 A single-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The power output from the turbine, the thermal efficiency of the plant, the exergy of the geothermal liquid at the exit of the flash chamber, and the exergy destructions and exergy efficiencies for the flash chamber, the turbine, and the entire plant are to be determined.

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

Analysis (*a*) We use properties of water for geothermal water (Tables A-4, A-5, and A-6)

$$\left.\begin{array}{l} T_1 = 230°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 990.14 \text{ kJ/kg} \\ s_1 = 2.6100 \text{ kJ/kg.K} \end{array}$$

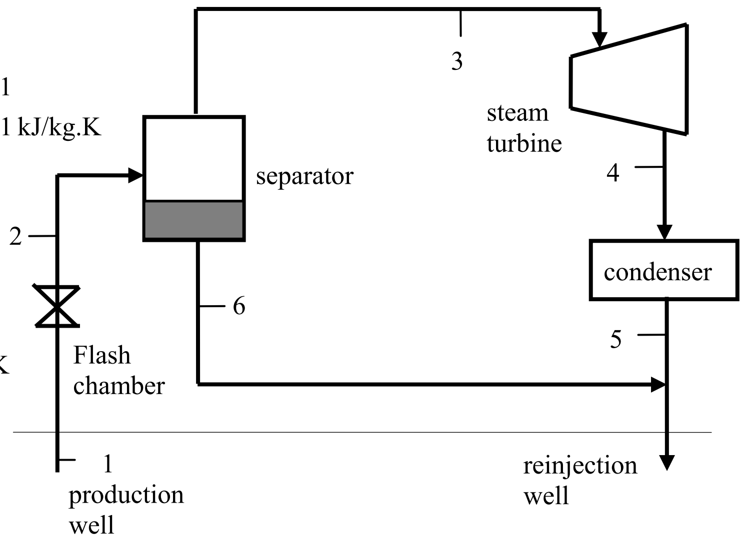
$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} x_2 = 0.1661 \\ s_2 = 2.6841 \text{ kJ/kg.K} \end{array}$$

$$\begin{aligned} \dot{m}_3 &= x_2 \dot{m}_1 \\ &= (0.1661)(230 \text{ kg/s}) \\ &= 38.19 \text{ kg/s} \end{aligned}$$

$$\left.\begin{array}{l} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2748.1 \text{ kJ/kg} \\ s_3 = 6.8207 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.95 \end{array}\right\} \begin{array}{l} h_4 = 2464.3 \text{ kJ/kg} \\ s_4 = 7.7739 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 500 \text{ kPa} \\ x_6 = 0 \end{array}\right\} \begin{array}{l} h_6 = 640.09 \text{ kJ/kg} \\ s_6 = 1.8604 \text{ kJ/kg}\cdot\text{K} \end{array}$$



$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 38.19 = 191.81 \text{ kg/s}$$

The power output from the turbine is

$$\dot{W}_\text{T} = \dot{m}_3 (h_3 - h_4) = (38.19 \text{ kJ/kg})(2748.1 - 2464.3)\text{kJ/kg} = \mathbf{10{,}842 \text{ kW}}$$

We use saturated liquid state at the standard temperature for dead state properties

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} \begin{array}{l} h_0 = 104.83 \text{ kJ/kg} \\ s_0 = 0.3672 \text{ kJ/kg} \end{array}$$

$$\dot{E}_\text{in} = \dot{m}_1 (h_1 - h_0) = (230 \text{ kJ/kg})(990.14 - 104.83)\text{kJ/kg} = 203{,}622 \text{ kW}$$

$$\eta_\text{th} = \frac{\dot{W}_\text{T,out}}{\dot{E}_\text{in}} = \frac{10{,}842}{203{,}622} = 0.0532 = \mathbf{5.3\%}$$

(*b*) The specific exergies at various states are

$$\psi_1 = h_1 - h_0 - T_0(s_1 - s_0) = (990.14 - 104.83)\text{kJ/kg} - (298 \text{ K})(2.6100 - 0.3672)\text{kJ/kg.K} = 216.53 \text{ kJ/kg}$$

$$\psi_2 = h_2 - h_0 - T_0(s_2 - s_0) = (990.14 - 104.83)\text{kJ/kg} - (298 \text{ K})(2.6841 - 0.3672)\text{kJ/kg.K} = 194.44 \text{ kJ/kg}$$

$$\psi_3 = h_3 - h_0 - T_0(s_3 - s_0) = (2748.1 - 104.83)\text{kJ/kg} - (298 \text{ K})(6.8207 - 0.3672)\text{kJ/kg.K} = 719.10 \text{ kJ/kg}$$

$$\psi_4 = h_4 - h_0 - T_0(s_4 - s_0) = (2464.3 - 104.83)\text{kJ/kg} - (298 \text{ K})(7.7739 - 0.3672)\text{kJ/kg.K} = 151.05 \text{ kJ/kg}$$

$$\psi_6 = h_6 - h_0 - T_0(s_6 - s_0) = (640.09 - 104.83)\text{kJ/kg} - (298 \text{ K})(1.8604 - 0.3672)\text{kJ/kg.K} = 89.97 \text{ kJ/kg}$$

The exergy of geothermal water at state 6 is

$$\dot{X}_6 = \dot{m}_6\psi_6 = (191.81\,\text{kg/s})(89.97\,\text{kJ/kg}) = \mathbf{17{,}257\ kW}$$

(c) Flash chamber:

$$\dot{X}_{\text{dest, FC}} = \dot{m}_1(\psi_1 - \psi_2) = (230\,\text{kg/s})(216.53 - 194.44)\text{kJ/kg} = \mathbf{5080\ kW}$$

$$\eta_{\text{II,FC}} = \frac{\psi_2}{\psi_1} = \frac{194.44}{216.53} = 0.898 = \mathbf{89.8\%}$$

(d) Turbine:

$$\dot{X}_{\text{dest,T}} = \dot{m}_3(\psi_3 - \psi_4) - \dot{W}_\text{T} = (38.19\,\text{kg/s})(719.10 - 151.05)\text{kJ/kg} - 10{,}842\,\text{kW} = \mathbf{10{,}854\ kW}$$

$$\eta_{\text{II,T}} = \frac{\dot{W}_\text{T}}{\dot{m}_3(\psi_3 - \psi_4)} = \frac{10{,}842\,\text{kW}}{(38.19\,\text{kg/s})(719.10 - 151.05)\text{kJ/kg}} = 0.500 = \mathbf{50.0\%}$$

(e) Plant:

$$\dot{X}_{\text{in,Plant}} = \dot{m}_1\psi_1 = (230\,\text{kg/s})(216.53\,\text{kJ/kg}) = 49{,}802\,\text{kW}$$

$$\dot{X}_{\text{dest,Plant}} = \dot{X}_{\text{in,Plant}} - \dot{W}_\text{T} = 49{,}802 - 10{,}842 = \mathbf{38{,}960\ kW}$$

$$\eta_{\text{II,Plant}} = \frac{\dot{W}_\text{T}}{\dot{X}_{\text{in,Plant}}} = \frac{10{,}842\,\text{kW}}{49{,}802\,\text{kW}} = 0.2177 = \mathbf{21.8\%}$$

Cogeneration

10-71C The utilization factor of a cogeneration plant is the ratio of the energy utilized for a useful purpose to the total energy supplied. It could be unity for a plant that does not produce any power.

10-72C No. A cogeneration plant may involve throttling, friction, and heat transfer through a finite temperature difference, and still have a utilization factor of unity.

10-73C Yes, if the cycle involves no irreversibilities such as throttling, friction, and heat transfer through a finite temperature difference.

10-74C Cogeneration is the production of more than one useful form of energy from the same energy source. Regeneration is the transfer of heat from the working fluid at some stage to the working fluid at some other stage.

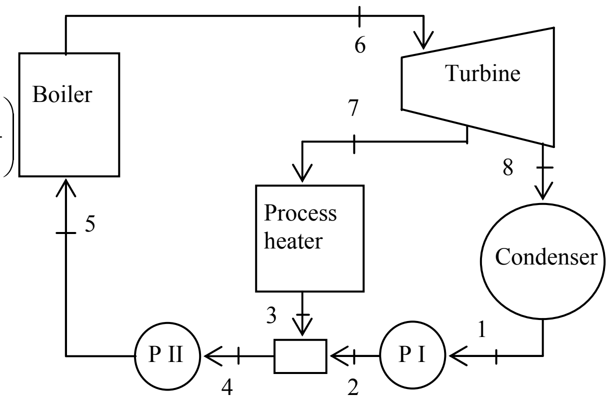
10-75 A cogeneration plant is to generate power and process heat. Part of the steam extracted from the turbine at a relatively high pressure is used for process heating. The net power produced and the utilization factor of the plant are to be determined.

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

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

$$h_1 = h_{f\ @\ 10\ \text{kPa}} = 191.81\ \text{kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 10\ \text{kPa}} = 0.00101\ \ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00101\ \ \text{m}^3/\text{kg})(600 - 10\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 0.60\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.60 = 192.41\ \text{kJ/kg}$$

$$h_3 = h_{f\ @\ 0.6\ \text{MPa}} = 670.38\ \text{kJ/kg}$$

Mixing chamber:

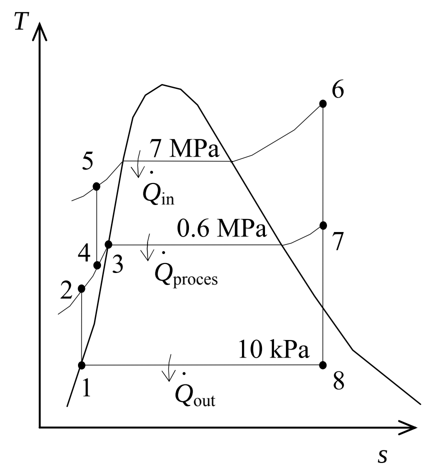
$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0\ (\text{steady})} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_4 h_4 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$

or,

$$h_4 = \frac{\dot{m}_2 h_2 + \dot{m}_3 h_3}{\dot{m}_4} = \frac{(22.50)(192.41) + (7.50)(670.38)}{30} = 311.90\ \text{kJ/kg}$$

$$\boldsymbol{v}_4 \cong \boldsymbol{v}_{f\ @\ h_f = 311.90\ \text{kJ/kg}} = 0.001026\ \ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{p\text{II,in}} &= \boldsymbol{v}_4(P_5 - P_4) \\ &= (0.001026\ \ \text{m}^3/\text{kg})(7000 - 600\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 6.57\ \text{kJ/kg} \end{aligned}$$

$$h_5 = h_4 + w_{p\text{II,in}} = 311.90 + 6.57 = 318.47\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 7\ \text{MPa} \\ T_6 &= 500°\text{C} \end{aligned}\right\}\ \begin{aligned} h_6 &= 3411.4\ \text{kJ/kg} \\ s_6 &= 6.8000\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 0.6\ \text{MPa} \\ s_7 &= s_6 \end{aligned}\right\}\ h_7 = 2774.6\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_8 &= 10\ \text{kPa} \\ s_8 &= s_6 \end{aligned}\right\}\ \begin{aligned} x_8 &= \frac{s_8 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201 \\ h_8 &= h_f + x_8 h_{fg} = 191.81 + (0.8201)(2392.1) = 2153.6\ \text{kJ/kg} \end{aligned}$$

Then,

$$\begin{aligned} \dot{W}_{\text{T,out}} &= \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_7 - h_8) \\ &= (30\ \text{kg/s})(3411.4 - 2774.6)\text{kJ/kg} + (22.5\ \text{kg/s})(2774.6 - 2153.6)\text{kJ/kg} = 33{,}077\ \text{kW} \end{aligned}$$

$$\dot{W}_{\text{p,in}} = \dot{m}_1 w_{\text{pI,in}} + \dot{m}_4 w_{\text{pII,in}} = (22.5\ \text{kg/s})(0.60\ \text{kJ/kg}) + (30\ \text{kg/s})(6.57\ \text{kJ/kg}) = 210.6\ \text{kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{p,in}} = 33{,}077 - 210.6 = \mathbf{32{,}866\ \ kW}$$

Also, $\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (7.5\ \text{kg/s})(2774.6 - 670.38)\ \text{kJ/kg} = 15{,}782\ \text{kW}$

$$\dot{Q}_{\text{in}} = \dot{m}_5(h_6 - h_5) = (30\ \text{kg/s})(3411.4 - 318.47) = 92{,}788\ \text{kW}$$

and

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{32{,}866 + 15{,}782}{92{,}788} = \mathbf{52.4\%}$$

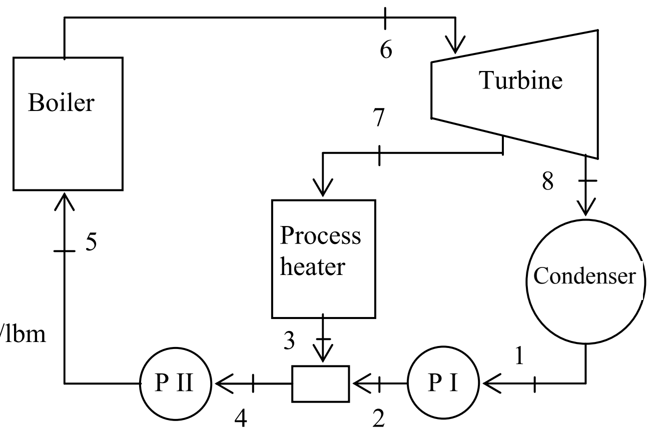
10-76E A large food-processing plant requires steam at a relatively high pressure, which is extracted from the turbine of a cogeneration plant. The rate of heat transfer to the boiler and the power output of the cogeneration plant are to be determined.

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

Analysis

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

$$h_1 = h_{f\ @\ 2\ \text{psia}} = 94.02\ \text{Btu/lbm}$$
$$v_1 = v_{f\ @\ 2\ \text{psia}} = 0.01623\ \text{ft}^3/\text{lbm}$$



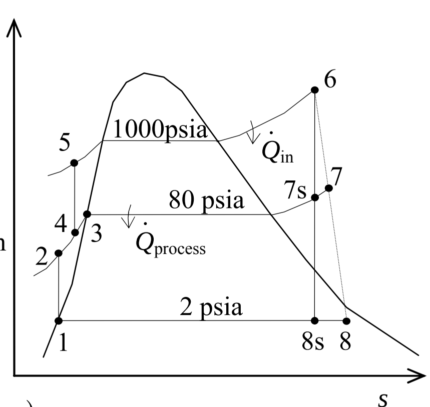
$$\begin{aligned} w_{\text{pI,in}} &= v_1(P_2 - P_1)/\eta_p \\ &= \frac{1}{0.86}(0.01623\ \text{ft}^3/\text{lbm})(80-2)\text{psia} \\ &\times\left(\frac{1\ \text{Btu}}{5.4039\ \text{psia}\cdot\text{ft}^3}\right) \\ &= 0.27\ \text{Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 94.02 + 0.27 = 94.29\ \text{Btu/lbm}$$

$$h_3 = h_{f\ @\ 80\ \text{psia}} = 282.13\ \text{Btu/lbm}$$

Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_4 h_4 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$



or,

$$h_4 = \frac{\dot{m}_2 h_2 + \dot{m}_3 h_3}{\dot{m}_4} = \frac{(3)(94.29)+(2)(282.13)}{5} = 169.43\ \text{Btu/lbm}$$
$$v_4 \cong v_{f\ @\ h_f = 169.43\ \text{Btu/lbm}} = 0.01664\ \text{ft}^3/\text{lbm}$$

$$\begin{aligned} w_{\text{pII,in}} &= v_4(P_5 - P_4)/\eta_p \\ &= (0.01664\ \text{ft}^3/\text{lbm})(1000-80\ \text{psia})\left(\frac{1\ \text{Btu}}{5.4039\ \text{psia}\cdot\text{ft}^3}\right)/(0.86) \\ &= 3.29\ \text{Btu/lbm} \end{aligned}$$

$$h_5 = h_4 + w_{pII,\text{in}} = 169.43 + 3.29 = 172.72\ \text{Btu/lbm}$$

$$\left.\begin{array}{l} P_6 = 1000\ \text{psia} \\ T_6 = 1000°\text{F} \end{array}\right\} \begin{array}{l} h_6 = 1506.2\ \text{Btu/lbm} \\ s_6 = 1.6535\ \text{Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_{7s} = 80\ \text{psia} \\ s_{7s} = s_6 \end{array}\right\} h_{7s} = 1209.0\ \text{Btu/lbm}$$

$$\left.\begin{array}{l} P_{8s} = 2\ \text{psia} \\ s_{8s} = s_6 \end{array}\right\} \begin{array}{l} x_{8s} = \dfrac{s_{8s} - s_f}{s_{fg}} = \dfrac{1.6535 - 0.17499}{1.74444} = 0.8475 \\ h_{8s} = h_f + x_{8s} h_{fg} = 94.02 + (0.8475)(1021.7) = 959.98\ \text{Btu/lbm} \end{array}$$

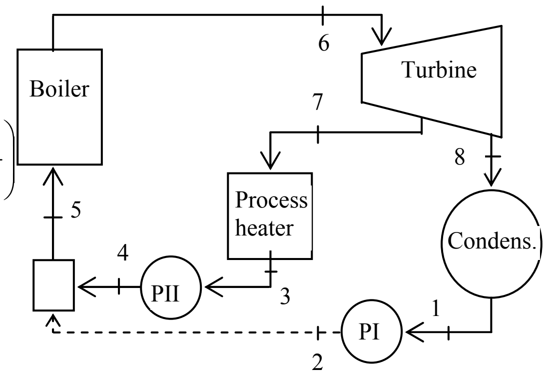
Then, $\dot{Q}_{\text{in}} = \dot{m}_5(h_6 - h_5) = (5\ \text{lbm/s})(1506.2 - 172.72)\text{Btu/lbm} = \mathbf{6667\ Btu/s}$

(*b*)
$$\begin{aligned} \dot{W}_{\text{T,out}} &= \eta_T \dot{W}_{T,s} = \eta_T[\dot{m}_6(h_6 - h_{7s}) + \dot{m}_8(h_{7s} - h_{8s})] \\ &= (0.86)[(5\ \text{lbm/s})(1506.2 - 1209.0)\ \text{Btu/lbm} + (3\ \text{lbm/s})(1209.0 - 959.98)\ \text{Btu/lbm}] \\ &= 1921\ \text{Btu/s} = \mathbf{2026\ kW} \end{aligned}$$

10-77 A cogeneration plant has two modes of operation. In the first mode, all the steam leaving the turbine at a relatively high pressure is routed to the process heater. In the second mode, 60 percent of the steam is routed to the process heater and remaining is expanded to the condenser pressure. The power produced and the rate at which process heat is supplied in the first mode, and the power produced and the rate of process heat supplied in the second mode 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),



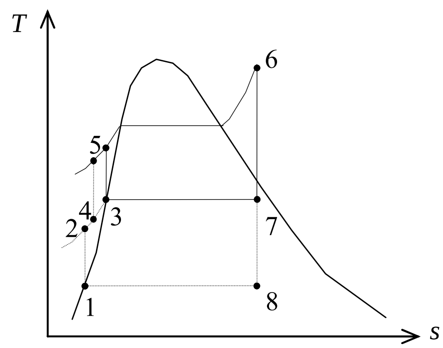
$$h_1 = h_{f\ @\ 20\ \text{kPa}} = 251.42\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 20\ \text{kPa}} = 0.001017\ \ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= \left(0.001017\ \ \text{m}^3/\text{kg}\right)(10{,}000 - 20\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 10.15\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 10.15 = 261.57\ \text{kJ/kg}$$
$$h_3 = h_{f\ @\ 0.5\ \text{MPa}} = 640.09\ \ \text{kJ/kg}$$
$$\boldsymbol{v}_3 = \boldsymbol{v}_{f\ @\ 0.5\ \text{MPa}} = 0.001093\ \ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= \left(0.001093\ \ \text{m}^3/\text{kg}\right)(10{,}000 - 500\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 10.38\ \text{kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.38 = 650.47\ \text{kJ/kg}$$



Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \quad\rightarrow\quad \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad\longrightarrow\quad \dot{m}_5 h_5 = \dot{m}_2 h_2 + \dot{m}_4 h_4$$

or,
$$h_5 = \frac{\dot{m}_2 h_2 + \dot{m}_4 h_4}{\dot{m}_5} = \frac{(2)(261.57) + (3)(650.47)}{5} = 494.91\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 10\ \text{MPa} \\ T_6 &= 450°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3242.4\ \text{kJ/kg} \\ s_6 &= 6.4219\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 0.5\ \text{MPa} \\ s_7 &= s_6 \end{aligned}\right\} \begin{aligned} x_7 &= \frac{s_7 - s_f}{s_{fg}} = \frac{6.4219 - 1.8604}{4.9603} = 0.9196 \\ h_7 &= h_f + x_7 h_{fg} = 640.09 + (0.9196)(2108.0) = 2578.6\ \text{kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_8 &= 20\ \text{kPa} \\ s_8 &= s_6 \end{aligned}\right\} \begin{aligned} x_8 &= \frac{s_8 - s_f}{s_{fg}} = \frac{6.4219 - 0.8320}{7.0752} = 0.7901 \\ h_8 &= h_f + x_8 h_{fg} = 251.42 + (0.7901)(2357.5) = 2114.0\ \text{kJ/kg} \end{aligned}$$

When the entire steam is routed to the process heater,

$$\dot{W}_{\text{T,out}} = \dot{m}_6(h_6 - h_7) = (5\ \text{kg/s})(3242.4 - 2578.6)\text{kJ/kg} = \mathbf{3319\ \ kW}$$

$$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (5\ \text{kg/s})(2578.6 - 640.09)\text{kJ/kg} = \mathbf{9693\ \ kW}$$

(*b*) When only 60% of the steam is routed to the process heater,

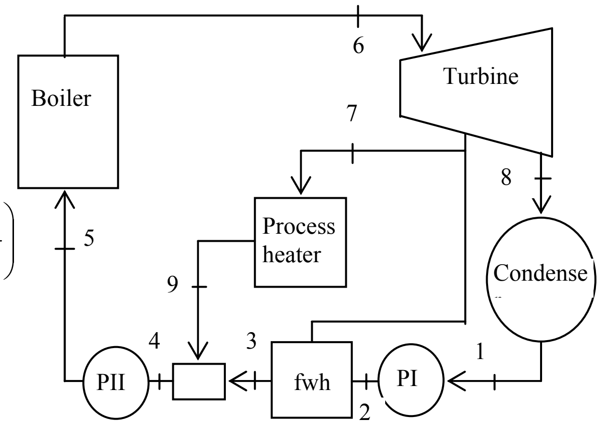
$$\begin{aligned} \dot{W}_{\text{T,out}} &= \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_7 - h_8) \\ &= (5\ \text{kg/s})(3242.4 - 2578.6)\ \text{kJ/kg} + (2\ \text{kg/s})(2578.6 - 2114.0)\ \text{kJ/kg} = \mathbf{4248\ kW} \end{aligned}$$

$$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (3\ \text{kg/s})(2578.6 - 640.09)\ \text{kJ/kg} = \mathbf{5816\ \ kW}$$

10-78 A cogeneration plant modified with regeneration is to generate power and process heat. The mass flow rate of steam through the boiler for a net power output of 15 MW is to be determined.

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

Analysis

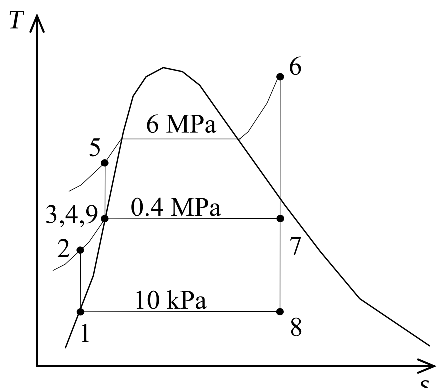


From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\ @\ 10\ \text{kPa}} = 191.81\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 10\ \text{kPa}} = 0.00101\ \ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00101\ \ \text{m}^3/\text{kg})(400 - 10\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 0.39\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.39 = 192.20\ \text{kJ/kg}$$
$$h_3 = h_4 = h_9 = h_{f\ @\ 0.4\ \text{MPa}} = 604.66\ \text{kJ/kg}$$
$$\boldsymbol{v}_4 = \boldsymbol{v}_{f\ @\ 0.4\ \text{MPa}} = 0.001084\ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_4(P_5 - P_4) \\ &= (0.001084\ \ \text{m}^3/\text{kg})(6000 - 400\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 6.07\ \text{kJ/kg} \end{aligned}$$

$$h_5 = h_4 + w_{\text{pII,in}} = 604.66 + 6.07 = 610.73\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 6\ \text{MPa} \\ T_6 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_6 = 3302.9\ \text{kJ/kg} \\ s_6 = 6.7219\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_7 = 0.4\ \text{MPa} \\ s_7 = s_6 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.7219 - 1.7765}{5.1191} = 0.9661 \\ h_7 = h_f + x_7 h_{fg} = 604.66 + (0.9661)(2133.4) = 2665.7\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_8 = 10\ \text{kPa} \\ s_8 = s_6 \end{array}\right\} \begin{array}{l} x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{6.7219 - 0.6492}{7.4996} = 0.8097 \\ h_8 = h_f + x_8 h_{fg} = 191.81 + (0.8097)(2392.1) = 2128.7\ \text{kJ/kg} \end{array}$$

Then, per kg of steam flowing through the boiler, we have

$$\begin{aligned} w_{\text{T,out}} &= (h_6 - h_7) + 0.4(h_7 - h_8) \\ &= (3302.9 - 2665.7)\ \text{kJ/kg} + (0.4)(2665.7 - 2128.7)\ \text{kJ/kg} \\ &= 852.0\ \text{kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{p,in}} &= 0.4 w_{\text{pI,in}} + w_{\text{pII,in}} \\ &= (0.4)(0.39\ \text{kJ/kg}) + (6.07\ \text{kJ/kg}) \\ &= 6.23\ \text{kJ/kg} \end{aligned}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{p,in}} = 852.0 - 6.23 = 845.8\ \text{kJ/kg}$$

Thus,

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{15{,}000\ \text{kJ/s}}{845.8\ \text{kJ/kg}} = \mathbf{17.73\ kg/s}$$

10-79 EES Problem 10-78 is reconsidered. The effect of the extraction pressure for removing steam from the turbine to be used for the process heater and open feedwater heater on the required mass flow rate is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input Data"
y = 0.6 "fraction of steam extracted from turbine for feedwater heater and process heater"
P[6] = 6000 [kPa]
T[6] = 450 [C]
P_extract=400 [kPa]
P[7] = P_extract
P_cond=10 [kPa]
P[8] = P_cond
W_dot_net=15 [MW]*Convert(MW, kW)
Eta_turb= 100/100 "Turbine isentropic efficiency"
Eta_pump = 100/100 "Pump isentropic efficiency"
P[1] = P[8]
P[2]=P[7]
P[3]=P[7]
P[4] = P[7]
P[5]=P[6]
P[9] = P[7]

"Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'

h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Open Feedwater Heater analysis:"
z*h[7]  + (1- y)*h[2] = (1- y + z)*h[3] "Steady-flow conservation of energy"
h[3]=enthalpy(Fluid$,P=P[3],x=0)
T[3]=temperature(Fluid$,P=P[3],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[3]=entropy(Fluid$,P=P[3],x=0)

"Process heater analysis:"
(y - z)*h[7]  = q_process + (y - z)*h[9] "Steady-flow conservation of energy"
Q_dot_process = m_dot*(y - z)*q_process"[kW]"
h[9]=enthalpy(Fluid$,P=P[9],x=0)
T[9]=temperature(Fluid$,P=P[9],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[9]=entropy(Fluid$,P=P[9],x=0)

"Mixing chamber at 3, 4, and 9:"
(y-z)*h[9]  + (1-y+z)*h[3] = 1*h[4] "Steady-flow conservation of energy"
T[4]=temperature(Fluid$,P=P[4],h=h[4]) "Condensate leaves heater as sat. liquid at P[3]"
s[4]=entropy(Fluid$,P=P[4],h=h[4])

"Boiler condensate pump  or Pump 2 analysis"
```

```
v4=volume(Fluid$,P=P[4],x=0)
w_pump2_s=v4*(P[5]-P[4])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[4]+w_pump2= h[5]  "Steady-flow conservation of energy"
s[5]=entropy(Fluid$,P=P[5],h=h[5])
T[5]=temperature(Fluid$,P=P[5],h=h[5])

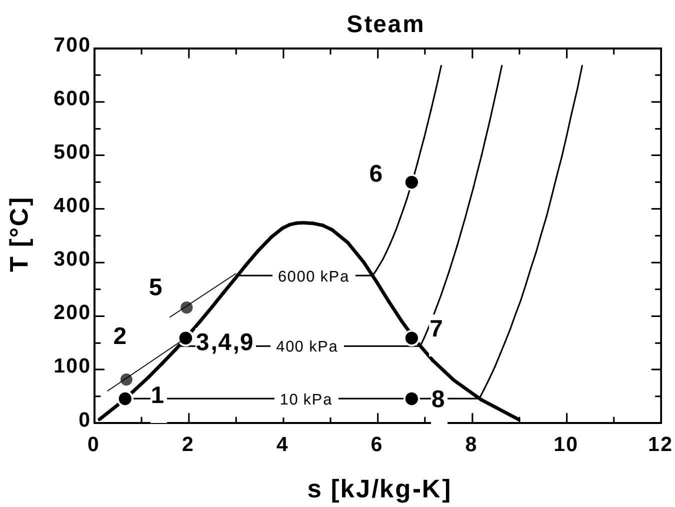
"Boiler analysis"
q_in + h[5]=h[6]"SSSF conservation of energy for the Boiler"
h[6]=enthalpy(Fluid$, T=T[6], P=P[6])
s[6]=entropy(Fluid$, T=T[6], P=P[6])

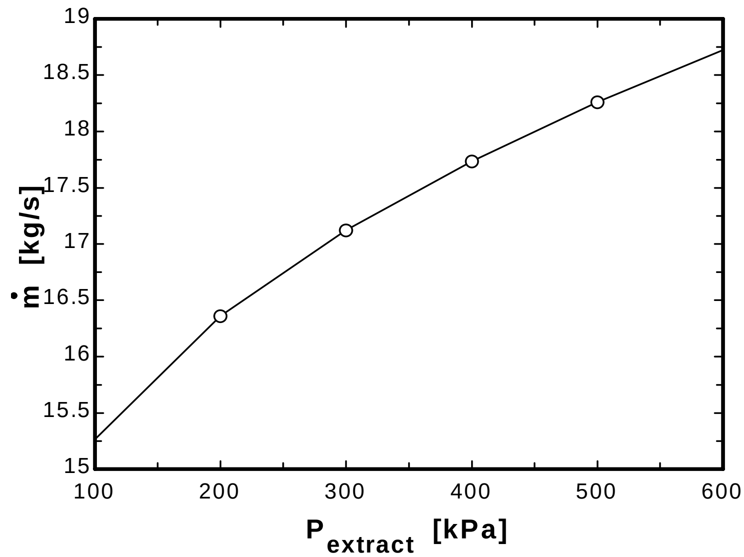
"Turbine analysis"
ss[7]=s[6]
hs[7]=enthalpy(Fluid$,s=ss[7],P=P[7])
Ts[7]=temperature(Fluid$,s=ss[7],P=P[7])
h[7]=h[6]-Eta_turb*(h[6]-hs[7])"Definition of turbine efficiency for high pressure stages"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])
ss[8]=s[7]
hs[8]=enthalpy(Fluid$,s=ss[8],P=P[8])
Ts[8]=temperature(Fluid$,s=ss[8],P=P[8])
h[8]=h[7]-Eta_turb*(h[7]-hs[8])"Definition of turbine efficiency for low pressure stages"
T[8]=temperature(Fluid$,P=P[8],h=h[8])
s[8]=entropy(Fluid$,P=P[8],h=h[8])
h[6] =y*h[7] + (1- y)*h[8] + w_turb "SSSF conservation of energy for turbine"

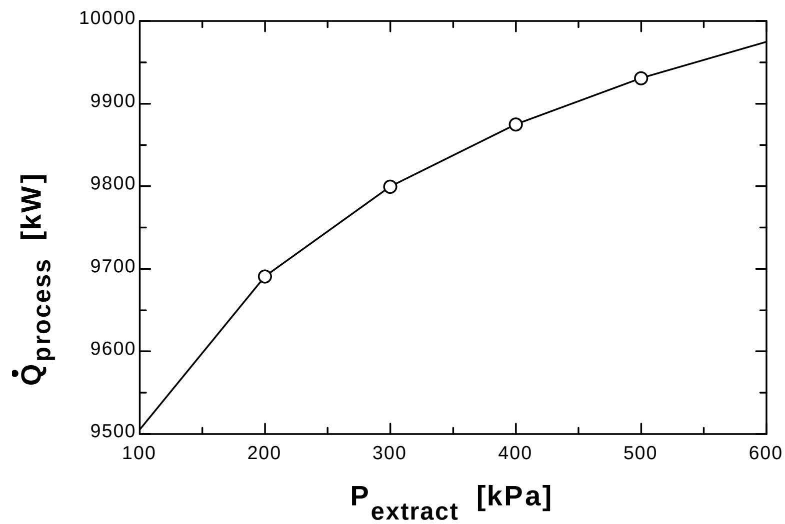
"Condenser analysis"
(1- y)*h[8]=q_out+(1- y)*h[1]"SSSF First Law for the Condenser"

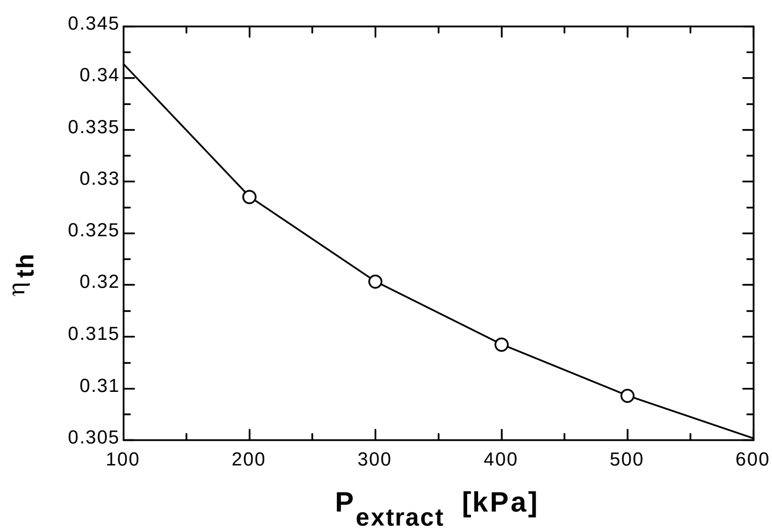
"Cycle Statistics"
w_net=w_turb - ((1- y)*w_pump1+ w_pump2)
Eta_th=w_net/q_in
W_dot_net = m_dot * w_net
```

$P_{extract}$ [kPa]	η_{th}	m [kg/s]	$Q_{process}$ [kW]
100	0.3413	15.26	9508
200	0.3284	16.36	9696
300	0.3203	17.12	9806
400	0.3142	17.74	9882
500	0.3092	18.26	9939
600	0.305	18.72	9984




ṁ [kg/s]
19
18.5
18
17.5
17
16.5
16
15.5
15
100
200
300
400
500
600
P extract [kPa]


Q̇ process [kW]
10000
9900
9800
9700
9600
9500
100
200
300
400
500
600
P extract [kPa]


ηth
0.345
0.34
0.335
0.33
0.325
0.32
0.315
0.31
0.305
100
200
300
400
500
600
P extract [kPa]

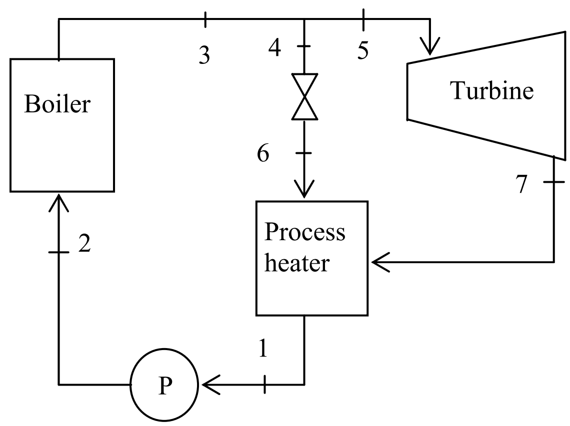
10-80E A cogeneration plant is to generate power while meeting the process steam requirements for a certain industrial application. The net power produced, the rate of process heat supply, and the utilization factor of this plant are to be determined.

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

Analysis

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

$$h_1 \cong h_{f\ @\ 240°\text{F}} = 208.49 \text{ Btu/lbm}$$
$$h_2 \cong h_1$$

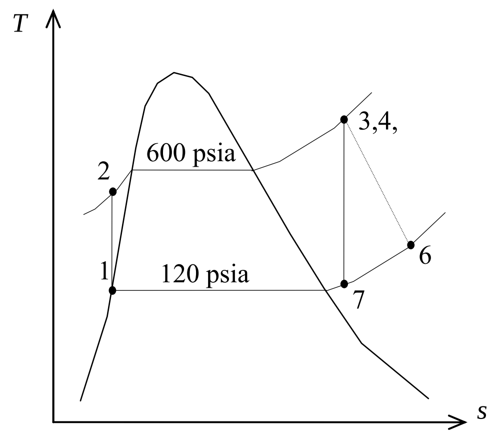


$$\left.\begin{array}{l} P_3 = 600 \text{ psia} \\ T_3 = 800°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1408.0 \text{ Btu/lbm} \\ s_3 = s_5 = s_7 = 1.6348 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$h_3 = h_4 = h_5 = h_6$$

$$\left.\begin{array}{l} P_7 = 120 \text{ psia} \\ s_7 = s_3 \end{array}\right\} h_7 = 1229.5 \text{ Btu/lbm}$$

$$\begin{aligned} \dot{W}_{\text{net}} &= \dot{m}_5 (h_5 - h_7) \\ &= (12 \text{ lbm/s})(1408.0 - 1229.5) \text{ Btu/lbm} \\ &= 2142 \text{ Btu/s} = \mathbf{2260\ kW} \end{aligned}$$



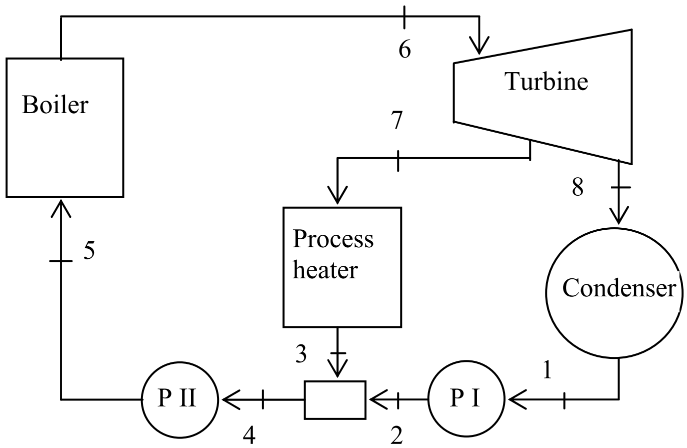
(*b*) $$\begin{aligned} \dot{Q}_{\text{process}} &= \sum \dot{m}_i h_i - \sum \dot{m}_e h_e \\ &= \dot{m}_6 h_6 + \dot{m}_7 h_7 - \dot{m}_1 h_1 - \\ &= (6)(1408.0) + (12)(1229.5) - (18)(208.49) \\ &= \mathbf{19{,}450\ Btu/s} \end{aligned}$$

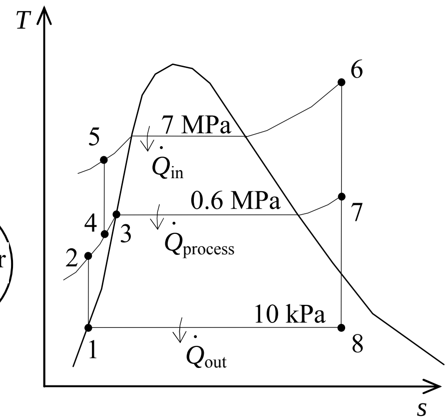
$$\begin{aligned} \dot{Q}_{\text{process}} &= \sum \dot{m}_e h_e - \sum \dot{m}_i h_i = \dot{m}_1 h_1 - \dot{m}_6 h_6 - \dot{m}_7 h_7 \\ &= (18)(208.49) - (6)(1408.0) - (12)(1229.5) \\ &= \mathbf{-19{,}450\ Btu/s} \end{aligned}$$

(*c*) $\varepsilon_u = \mathbf{1}$ since all the energy is utilized.

10-81 A cogeneration plant is to generate power and process heat. Part of the steam extracted from the turbine at a relatively high pressure is used for process heating. The mass flow rate of steam that must be supplied by the boiler, the net power produced, and the utilization factor of the plant are to be determined.

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





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

$$h_1 = h_{f\ @\ 10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 10\text{ kPa}} = 0.00101\ \text{m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00101\ \text{m}^3\text{/kg})(600 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.596\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.596 = 192.40\text{ kJ/kg}$$

$$h_3 = h_{f\ @\ 0.6\text{ MPa}} = 670.38\text{ kJ/kg}$$

Mixing chamber:

$$\dot{m}_3 h_3 + \dot{m}_2 h_2 = \dot{m}_4 h_4$$
$$(0.25)(670.38\text{ kJ/kg}) + (0.75)(192.40\text{ kJ/kg})) = (1)h_4 \longrightarrow h_4 = 311.90\text{ kJ/kg}$$

$$\boldsymbol{v}_4 \cong \boldsymbol{v}_{f\ @\ h_f = 311.90\text{ kJ/kg}} = 0.001026\ \text{m}^3\text{/kg}$$

$$\begin{aligned} w_{p\text{II,in}} &= \boldsymbol{v}_4(P_5 - P_4) \\ &= (0.001026\ \text{m}^3\text{/kg})(7000 - 600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 6.563\text{ kJ/kg} \end{aligned}$$
$$h_5 = h_4 + w_{p\text{II,in}} = 311.90 + 6.563 = 318.47\text{ kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 7\text{ MPa} \\ T_6 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3411.4\text{ kJ/kg} \\ s_6 &= 6.8000\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 0.6\text{ MPa} \\ s_7 &= s_6 \end{aligned}\right\} h_7 = 2773.9\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 10 \text{ kPa} \\ s_8 = s_6 \end{array}\right\} h_8 = 2153.6 \text{ kJ/kg}$$

$$\begin{aligned} \dot{Q}_{\text{process}} &= \dot{m}_7 (h_7 - h_3) \\ 8600 \text{ kJ/s} &= \dot{m}_7 (2773.9 - 670.38) \text{kJ/kg} \\ \dot{m}_7 &= 4.088 \text{ kg/s} \end{aligned}$$

This is one-fourth of the mass flowing through the boiler. Thus, the mass flow rate of steam that must be supplied by the boiler becomes

$$\dot{m}_6 = 4\dot{m}_7 = 4(4.088 \text{ kg/s}) = \mathbf{16.35\ kg/s}$$

(*b*) Cycle analysis:

$$\begin{aligned} \dot{W}_{\text{T,out}} &= \dot{m}_7 (h_6 - h_7) + \dot{m}_8 (h_6 - h_8) \\ &= (4.088 \text{ kg/s})(3411.4 - 2773.9) \text{kJ/kg} + (16.35 \text{ - } 4.088 \text{ kg/s})(3411.4 - 2153.6) \text{kJ/kg} \\ &= 18{,}033 \text{ kW} \end{aligned}$$

$$\begin{aligned} \dot{W}_{\text{p,in}} &= \dot{m}_1 w_{\text{pI,in}} + \dot{m}_4 w_{\text{pII,in}} \\ &= (16.35 \text{ - } 4.088 \text{ kg/s})(0.596 \text{ kJ/kg}) + (16.35 \text{ kg/s})(6.563 \text{ kJ/kg}) = 114.6 \text{ kW} \end{aligned}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{p,in}} = 18{,}033 - 115 = \mathbf{17{,}919\ kW}$$

(*c*) Then,

$$\dot{Q}_{\text{in}} = \dot{m}_5 (h_6 - h_5) = (16.35 \text{ kg/s})(3411.4 - 318.46) = 50{,}581 \text{ kW}$$

and

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{17{,}919 + 8600}{50{,}581} = 0.524 = \mathbf{52.4\%}$$

Combined Gas-Vapor Power Cycles

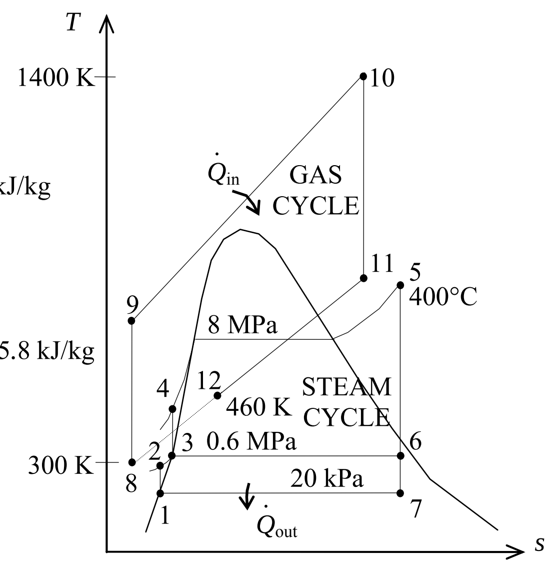
10-82C The energy source of the steam is the waste energy of the exhausted combustion gases.

10-83C Because the combined gas-steam cycle takes advantage of the desirable characteristics of the gas cycle at high temperature, and those of steam cycle at low temperature, and combines them. The result is a cycle that is more efficient than either cycle executed operated alone.

10-84 [*Also solved by EES on enclosed CD***]** A 450-MW combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is an ideal Rankine cycle with an open feedwater heater. The mass flow rate of air to steam, the required rate of heat input in the combustion chamber, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) The analysis of gas cycle yields (Table A-17)



$$T_8 = 300\text{ K} \longrightarrow h_8 = 300.19\text{ kJ/kg}$$
$$P_{r_8} = 1.386$$

$$P_{r_9} = \frac{P_9}{P_8}P_{r_8} = (14)(1.386) = 19.40 \longrightarrow h_9 = 635.5\text{ kJ/kg}$$

$$T_{10} = 1400\text{ K} \longrightarrow h_{10} = 1515.42\text{ kJ/kg}$$
$$P_{r_{10}} = 450.5$$

$$P_{r_{11}} = \frac{P_{11}}{P_{10}}P_{r_{10}} = \left(\frac{1}{14}\right)(450.5) = 32.18 \longrightarrow h_{11} = 735.8\text{ kJ/kg}$$

$$T_{12} = 460\text{ K} \longrightarrow h_{12} = 462.02\text{ kJ/kg}$$

From the steam tables (Tables A-4, A-5, A-6),

$$h_1 = h_{f\text{ @ 20 kPa}} = 251.42\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\text{ @ 20 kPa}} = 0.001017\ \text{m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017\ \text{m}^3\text{/kg})(600 - 20\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.59\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 0.59 = 252.01\text{ kJ/kg}$$

$$h_3 = h_{f\text{ @ 0.6 MPa}} = 670.38\text{ kJ/kg}$$
$$\boldsymbol{v}_3 = \boldsymbol{v}_{f\text{ @ 0.6 MPa}} = 0.001101\ \text{m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= (0.001101\ \text{m}^3\text{/kg})(8{,}000 - 600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 8.15\text{ kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pI,in}} = 670.38 + 8.15 = 678.53\text{ kJ/kg}$$

$$\left.\begin{aligned} P_5 &= 8\text{ MPa} \\ T_5 &= 400°\text{C} \end{aligned}\right\} \begin{aligned} h_5 &= 3139.4\text{ kJ/kg} \\ s_5 &= 6.3658\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 0.6\text{ MPa} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} x_6 &= \frac{s_6 - s_f}{s_{fg}} = \frac{6.3658 - 1.9308}{4.8285} = 0.9185 \\ h_6 &= h_f + x_6 h_{fg} = 670.38 + (0.9185)(2085.8) = 2586.1\text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 20\text{ kPa} \\ s_7 &= s_5 \end{aligned}\right\} \begin{aligned} x_7 &= \frac{s_7 - s_f}{s_{fg}} = \frac{6.3658 - 0.8320}{7.0752} = 0.7821 \\ h_7 &= h_f + x_7 h_{fg} = 251.42 + (0.7821)(2357.5) = 2095.2\text{ kJ/kg} \end{aligned}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\rm in} - \dot{E}_{\rm out} = \Delta\dot{E}_{\rm system}^{\nearrow 0\,(\rm steady)} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s(h_5 - h_4) = \dot{m}_{\rm air}(h_{11} - h_{12})$$

$$\frac{\dot{m}_{\rm air}}{\dot{m}_s} = \frac{h_5 - h_4}{h_{11} - h_{12}} = \frac{3139.4 - 678.53}{735.80 - 462.02} = \textbf{8.99 kg air / kg steam}$$

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$ for the open FWH, the steady-flow energy balance equation yields

$$\dot{E}_{\rm in} - \dot{E}_{\rm out} = \Delta\dot{E}_{\rm system}^{\nearrow 0\,(\rm steady)} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y) h_2 = (1) h_3$$

Thus,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{670.38 - 252.01}{2586.1 - 252.01} = 0.1792 \quad (\text{the fraction of steam extracted})$$

$$w_T = h_5 - h_6 + (1-y)(h_6 - h_7)$$
$$= 3139.4 - 2586.1 + (1 - 0.1792)(2586.1 - 2095.2) = 956.23 \text{ kJ/kg}$$

$$w_{\rm net,steam} = w_T - w_{p,\rm in} = w_T - (1-y)w_{p,I} - w_{p,II}$$
$$= 956.23 - (1 - 0.1792)(0.59) - 8.15 = 948.56 \text{ kJ/kg}$$

$$w_{\rm net,gas} = w_T - w_{C,\rm in} = (h_{10} - h_{11}) - (h_9 - h_8)$$
$$= 1515.42 - 735.8 - (635.5 - 300.19) = 444.3 \text{ kJ/kg}$$

The net work output per unit mass of gas is

$$w_{\rm net} = w_{\rm net,gas} + \tfrac{1}{8.99} w_{\rm net,steam} = 444.3 + \tfrac{1}{8.99}(948.56) = 549.8 \text{ kJ/kg}$$

$$\dot{m}_{\rm air} = \frac{\dot{W}_{\rm net}}{w_{\rm net}} = \frac{450{,}000 \text{ kJ/s}}{549.7 \text{ kJ/kg}} = 818.7 \text{ kg/s}$$

and

$$\dot{Q}_{\rm in} = \dot{m}_{\rm air}(h_{10} - h_9) = (818.5 \text{ kg/s})(1515.42 - 635.5) \text{ kJ/kg} = \textbf{720,215 kW}$$

(*c*) $$\eta_{th} = \frac{\dot{W}_{\rm net}}{\dot{Q}_{\rm in}} = \frac{450{,}000 \text{ kW}}{720{,}215 \text{ kW}} = \textbf{62.5\%}$$

10-85 EES Problem 10-84 is reconsidered. The effect of the gas cycle pressure ratio on the ratio of gas flow rate to steam flow rate and cycle thermal efficiency is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input data"
T[8] = 300 [K]            "Gas compressor inlet"
P[8] = 14.7 [kPa]         "Assumed air inlet pressure"
"Pratio = 14"             "Pressure ratio for gas compressor"
T[10] = 1400 [K]          "Gas turbine inlet"
T[12] = 460 [K]           "Gas exit temperature from Gas-to-steam heat exchanger "
P[12] = P[8]              "Assumed air exit pressure"
W_dot_net=450 [MW]
Eta_comp = 1.0
Eta_gas_turb = 1.0
Eta_pump = 1.0
Eta_steam_turb = 1.0
P[5] = 8000 [kPa]          "Steam turbine inlet"
T[5] =(400+273) "[K]"      "Steam turbine inlet"
P[6] = 600 [kPa]          "Extraction pressure for steam open feedwater heater"
P[7] = 20 [kPa]           "Steam condenser pressure"

"GAS POWER CYCLE ANALYSIS"

"Gas Compressor anaysis"
s[8]=ENTROPY(Air,T=T[8],P=P[8])
ss9=s[8] "For the ideal case the entropies are constant across the compressor"
P[9] = Pratio*P[8]
Ts9=temperature(Air,s=ss9,P=P[9])"Ts9 is the isentropic value of T[9] at compressor exit"
Eta_comp = w_gas_comp_isen/w_gas_comp "compressor adiabatic efficiency, w_comp >
w_comp_isen"
h[8] + w_gas_comp_isen =hs9"SSSF conservation of energy for the isentropic compressor,
assuming: adiabatic, ke=pe=0 per unit gas mass flow rate in kg/s"
h[8]=ENTHALPY(Air,T=T[8])
hs9=ENTHALPY(Air,T=Ts9)
h[8] + w_gas_comp = h[9]"SSSF conservation of energy for the actual compressor, assuming:
adiabatic, ke=pe=0"
T[9]=temperature(Air,h=h[9])
s[9]=ENTROPY(Air,T=T[9],P=P[9])

"Gas Cycle External heat exchanger analysis"
h[9] + q_in = h[10]"SSSF conservation of energy for the external heat exchanger, assuming W=0,
ke=pe=0"
h[10]=ENTHALPY(Air,T=T[10])
P[10]=P[9]                          "Assume process 9-10 is SSSF constant pressure"
Q_dot_in"MW"*1000"kW/MW"=m_dot_gas*q_in

"Gas Turbine analysis"
s[10]=ENTROPY(Air,T=T[10],P=P[10])
ss11=s[10] "For the ideal case the entropies are constant across the turbine"
P[11] = P[10] /Pratio
Ts11=temperature(Air,s=ss11,P=P[11])"Ts11 is the isentropic value of T[11] at gas turbine exit"
Eta_gas_turb = w_gas_turb /w_gas_turb_isen "gas turbine adiabatic efficiency, w_gas_turb_isen
> w_gas_turb"
```

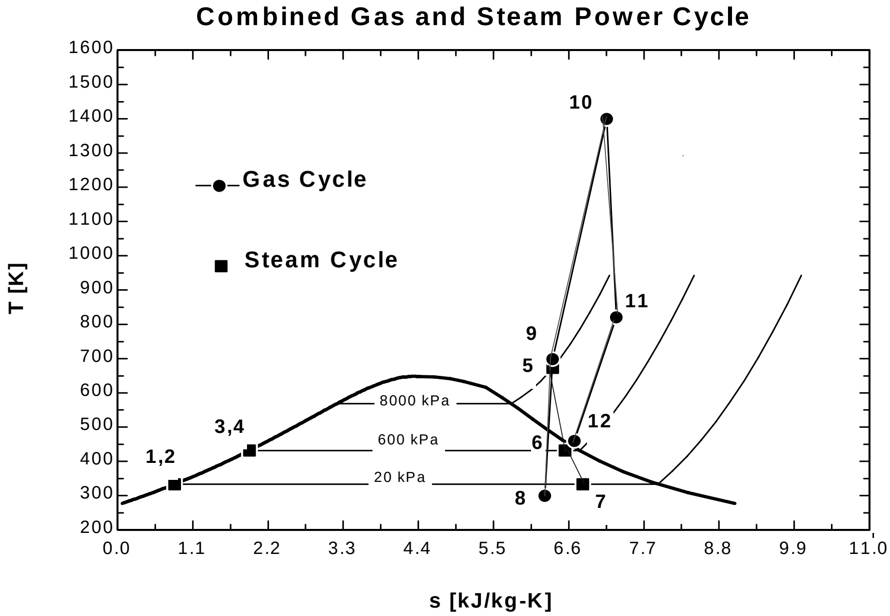
```
h[10] = w_gas_turb_isen + hs11"SSSF conservation of energy for the isentropic gas turbine, assuming: adiabatic, ke=pe=0"
hs11=ENTHALPY(Air,T=Ts11)
h[10] = w_gas_turb + h[11]"SSSF conservation of energy for the actual gas turbine, assuming: adiabatic, ke=pe=0"
T[11]=temperature(Air,h=h[11])
s[11]=ENTROPY(Air,T=T[11],P=P[11])

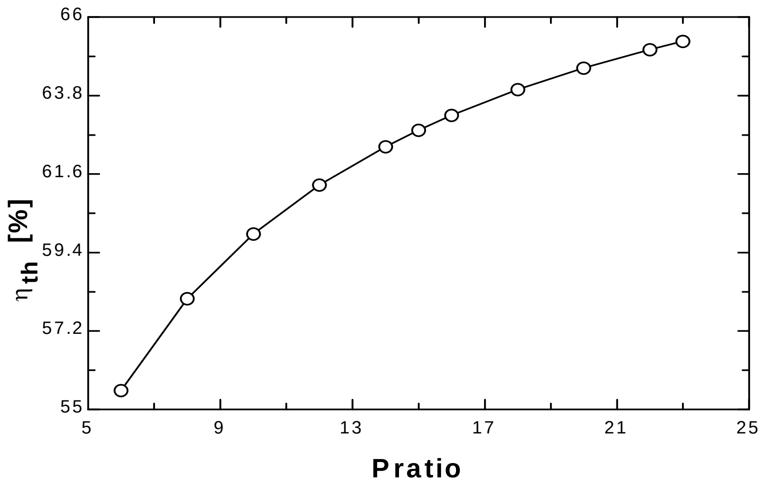
"Gas-to-Steam Heat Exchanger"
"SSSF conservation of energy for the gas-to-steam heat exchanger, assuming: adiabatic, W=0, ke=pe=0"
m_dot_gas*h[11] + m_dot_steam*h[4] = m_dot_gas*h[12] + m_dot_steam*h[5]
h[12]=ENTHALPY(Air, T=T[12])
s[12]=ENTROPY(Air,T=T[12],P=P[12])

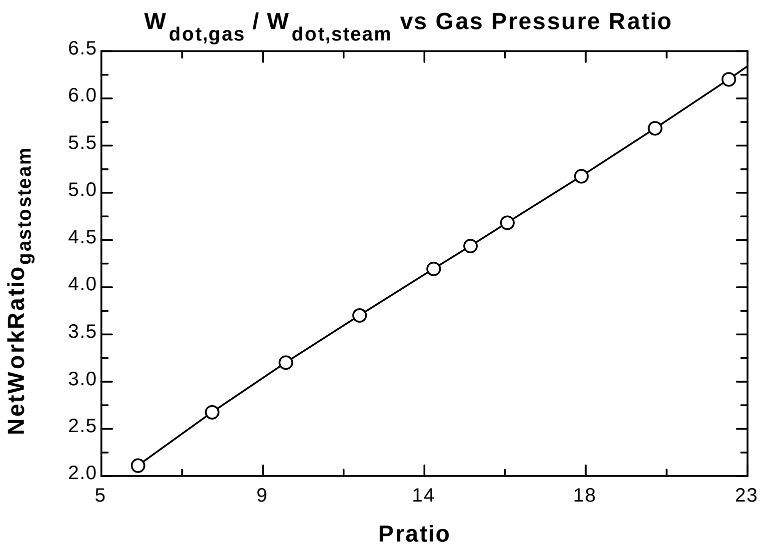
"STEAM CYCLE ANALYSIS"
"Steam Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'
P[1] = P[7]
P[2]=P[6]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Saturated liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"Open Feedwater Heater analysis"
y*h[6] + (1-y)*h[2] = 1*h[3] "Steady-flow conservation of energy"
P[3]=P[6]
h[3]=enthalpy(Fluid$,P=P[3],x=0)      "Condensate leaves heater as sat. liquid at P[3]"
T[3]=temperature(Fluid$,P=P[3],x=0)
s[3]=entropy(Fluid$,P=P[3],x=0)
"Boiler condensate pump  or Pump 2 analysis"
P[4] = P[5]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])
w_steam_pumps = (1-y)*w_pump1+ w_pump2 "Total steam pump work input/ mass steam"
"Steam Turbine analysis"
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s[5]=entropy(Fluid$,P=P[5],T=T[5])
ss6=s[5]
hs6=enthalpy(Fluid$,s=ss6,P=P[6])
Ts6=temperature(Fluid$,s=ss6,P=P[6])
h[6]=h[5]-Eta_steam_turb*(h[5]-hs6)"Definition of steam turbine efficiency"
T[6]=temperature(Fluid$,P=P[6],h=h[6])
s[6]=entropy(Fluid$,P=P[6],h=h[6])
ss7=s[5]
hs7=enthalpy(Fluid$,s=ss7,P=P[7])
Ts7=temperature(Fluid$,s=ss7,P=P[7])
```

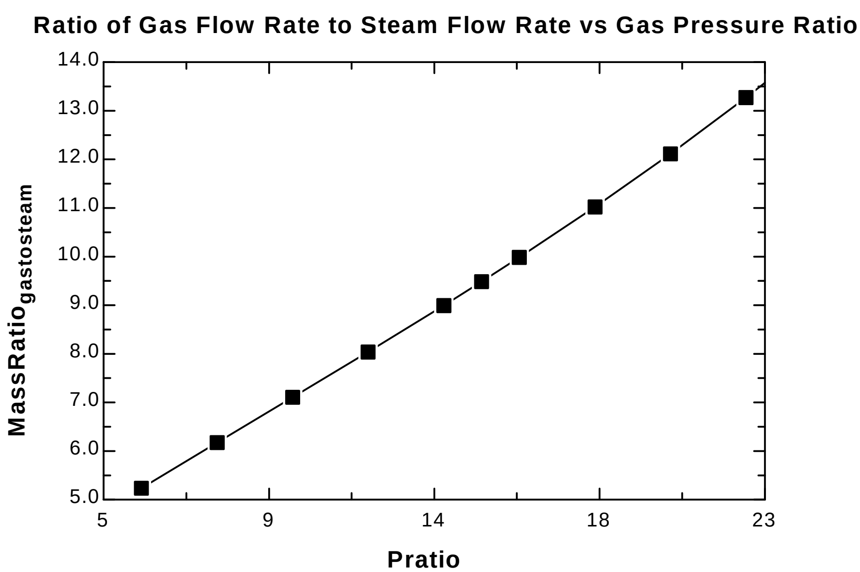
```
h[7]=h[5]-Eta_steam_turb*(h[5]-hs7)"Definition of steam turbine efficiency"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])
"SSSF conservation of energy for the steam turbine: adiabatic, neglect ke and pe"
h[5] = w_steam_turb + y*h[6] +(1-y)*h[7]
"Steam Condenser analysis"
(1-y)*h[7]=q_out+(1-y)*h[1]"SSSF conservation of energy for the Condenser per unit mass"
Q_dot_out*Convert(MW, kW)=m_dot_steam*q_out
"Cycle Statistics"
MassRatio_gastosteam =m_dot_gas/m_dot_steam
W_dot_net*Convert(MW, kW)=m_dot_gas*(w_gas_turb-w_gas_comp)+
m_dot_steam*(w_steam_turb - w_steam_pumps)"definition of the net cycle work"
Eta_th=W_dot_net/Q_dot_in*Convert(, %) "Cycle thermal efficiency, in percent"
Bwr=(m_dot_gas*w_gas_comp + m_dot_steam*w_steam_pumps)/(m_dot_gas*w_gas_turb +
m_dot_steam*w_steam_turb) "Back work ratio"
W_dot_net_steam = m_dot_steam*(w_steam_turb - w_steam_pumps)
W_dot_net_gas = m_dot_gas*(w_gas_turb - w_gas_comp)
NetWorkRatio_gastosteam = W_dot_net_gas/W_dot_net_steam
```

Pratio	$MassRatio_{gastosteam}$	W_{netgas} [kW]	$W_{netsteam}$ [kW]	η_{th} [%]	$NetWorkRatio_{gastosteam}$
10	7.108	342944	107056	59.92	3.203
11	7.574	349014	100986	60.65	3.456
12	8.043	354353	95647	61.29	3.705
13	8.519	359110	90890	61.86	3.951
14	9.001	363394	86606	62.37	4.196
15	9.492	367285	82715	62.83	4.44
16	9.993	370849	79151	63.24	4.685
17	10.51	374135	75865	63.62	4.932
18	11.03	377182	72818	63.97	5.18
19	11.57	380024	69976	64.28	5.431
20	12.12	382687	67313	64.57	5.685




66
63.8
61.6
59.4
57.2
55
5
9
13
17
21
25
ηth [%]
Pratio


Wdot,gas / Wdot,steam vs Gas Pressure Ratio
6.5
6.0
5.5
5.0
4.5
4.0
3.5
3.0
2.5
2.0
5
9
14
18
23
NetWorkRatiogastosteam
Pratio

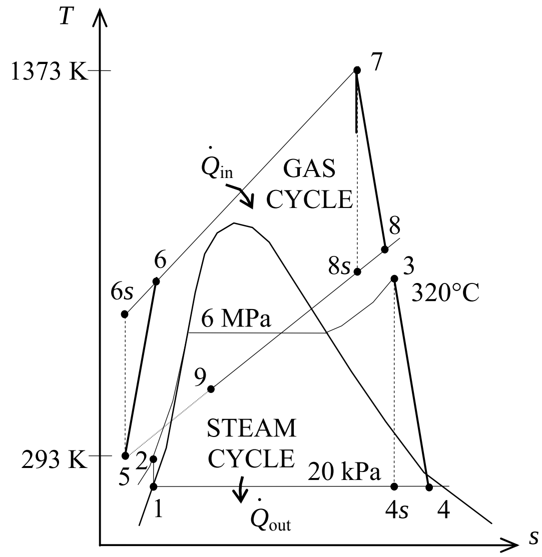

Ratio of Gas Flow Rate to Steam Flow Rate vs Gas Pressure Ratio
14.0
13.0
12.0
11.0
10.0
9.0
8.0
7.0
6.0
5.0
5
9
14
18
23
MassRatiogastosteam
Pratio

10-86 A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The mass flow rate of air for a specified power output is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable fo Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis Working around the topping cycle gives the following results:



$$T_{6s} = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (293\text{ K})(8)^{0.4/1.4} = 530.8\text{ K}$$

$$\eta_C = \frac{h_{6s}-h_5}{h_6-h_5} = \frac{c_p(T_{6s}-T_5)}{c_p(T_6-T_5)}$$

$$\longrightarrow T_6 = T_5 + \frac{T_{6s}-T_5}{\eta_C} = 293 + \frac{530.8-293}{0.85} = 572.8\text{ K}$$

$$T_{8s} = T_7\left(\frac{P_8}{P_7}\right)^{(k-1)/k} = (1373\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 758.0\text{ K}$$

$$\eta_T = \frac{h_7-h_8}{h_7-h_{8s}} = \frac{c_p(T_7-T_8)}{c_p(T_7-T_{8s})} \longrightarrow T_8 = T_7 - \eta_T(T_7-T_{8s}) = 1373-(0.90)(1373-758.0) = 819.5\text{ K}$$

$$T_9 = T_{\text{sat @ 6000 kPa}} = 275.6°\text{C} = 548.6\text{ K}$$

Fixing the states around the bottom steam cycle yields (Tables A-4, A-5, A-6):

$$h_1 = h_{f\text{ @ 20 kPa}} = 251.42\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\text{ @ 20 kPa}} = 0.001017\text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2-P_1) \\ &= (0.001017\text{ m}^3\text{/kg})(6000-20)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 6.08\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 6.08 = 257.5\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 6000\text{ kPa} \\ T_3 = 320°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 2953.6\text{ kJ/kg} \\ s_3 = 6.1871\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 20\text{ kPa} \\ s_4 = s_3 \end{array}\right\} h_{4s} = 2035.8\text{ kJ/kg}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3-h_{4s}) = 2953.6-(0.90)(2953.6-2035.8) = 2127.6\text{ kJ/kg}$$

The net work outputs from each cycle are

$$\begin{aligned} w_{\text{net, gas cycle}} &= w_{\text{T,out}} - w_{\text{C,in}} \\ &= c_p(T_7 - T_8) - c_p(T_6 - T_5) \\ &= (1.005 \text{ kJ/kg} \cdot \text{K})(1373 - 819.5 - 572.7 + 293)\text{K} \\ &= 275.2 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{net, steam cycle}} &= w_{\text{T,out}} - w_{\text{P,in}} \\ &= (h_3 - h_4) - w_{\text{P,in}} \\ &= (2953.6 - 2127.6) - 6.08 \\ &= 819.9 \text{ kJ/kg} \end{aligned}$$

An energy balance on the heat exchanger gives

$$\dot{m}_a c_p (T_8 - T_9) = \dot{m}_w (h_3\text{-}h_2) \longrightarrow \dot{m}_w = \frac{c_p(T_8 - T_9)}{h_3\text{-}h_2}\dot{m}_a = \frac{(1.005)(819.5 - 548.6)}{2953.6 - 257.5} = 0.1010\dot{m}_a$$

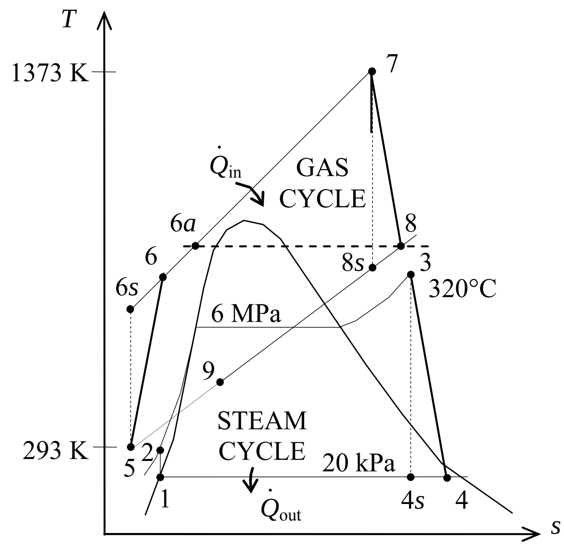
That is, 1 kg of exhaust gases can heat only 0.1010 kg of water. Then, the mass flow rate of air is

$$\dot{m}_a = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{100{,}000 \text{ kJ/s}}{(1 \times 275.2 + 0.1010 \times 819.9) \text{ kJ/kg air}} = \mathbf{279.3\ kg/s}$$

10-87 A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The mass flow rate of air for a specified power output is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable fo Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).



Analysis With an ideal regenerator, the temperature of the air at the compressor exit will be heated to the to the temperature at the turbine exit. Representing this state by "6a"

$$T_{6a} = T_8 = 819.5\ \text{K}$$

The rate of heat addition in the cycle is

$$\begin{aligned}\dot{Q}_{\text{in}} &= \dot{m}_a c_p (T_7 - T_{6a}) \\ &= (279.3\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot{}^\circ\text{C})(1373 - 819.5)\ \text{K} \\ &= 155{,}370\ \text{kW}\end{aligned}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{100{,}000\ \text{kW}}{155{,}370\ \text{kW}} = \mathbf{0.6436}$$

Without the regenerator, the rate of heat addition and the thermal efficiency are

$$\dot{Q}_{\text{in}} = \dot{m}_a c_p (T_7 - T_6) = (279.3\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot{}^\circ\text{C})(1373 - 572.7)\ \text{K} = 224{,}640\ \text{kW}$$

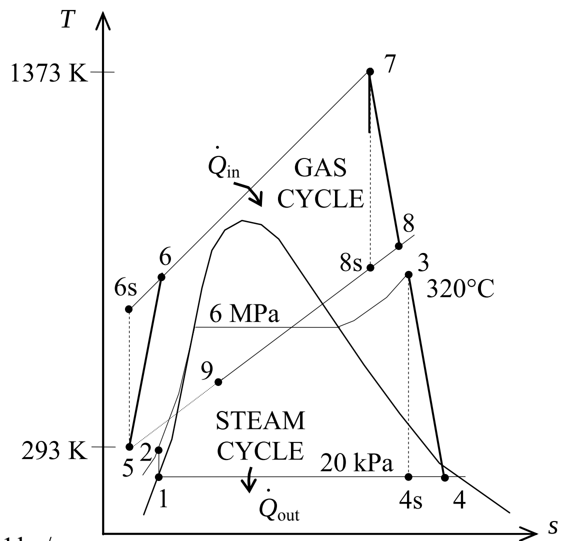
$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{100{,}000\ \text{kW}}{224{,}640\ \text{kW}} = \mathbf{0.4452}$$

The change in the thermal efficiency due to using the ideal regenerator is

$$\Delta\eta_{\text{th}} = 0.6436 - 0.4452 = \mathbf{0.1984}$$

10-88 The component of the combined cycle with the largest exergy destruction of the component of the combined cycle in Prob. 10-86 is to be determined.

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



Analysis From Problem 10-86,

$$T_{\text{source, gas cycle}} = 1373\text{ K}$$

$$T_{\text{source, steam cycle}} = T_8 = 819.5\text{ K}$$

$$T_{\text{sink}} = 293\text{ K}$$

$$s_1 = s_2 = s_{f\,@\,20\,\text{kPa}} = 0.8320\text{ kJ/kg}\cdot\text{K}$$

$$s_3 = 6.1871\text{ kJ/kg}\cdot\text{K}$$

$$s_4 = 6.4627\text{ kJ/kg}\cdot\text{K}$$

$$q_{\text{in},67} = c_p(T_7 - T_6) = 804.3\text{ kJ/kg}$$

$$q_{\text{in},23} = h_3 - h_2 = 2696.1\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 1876.2\text{ kJ/kg}$$

$$\dot{m}_w = 0.1010\dot{m}_a = 0.1010(279.3) = 28.21\text{ kg/s}$$

$$\dot{X}_{\text{destroyed},12} = 0 \quad \text{(isentropic process)}$$

$$\dot{X}_{\text{destroyed},34} = \dot{m}_w T_0 (s_4 - s_3) = (28.21\text{ kg/s})(293\text{ K})(6.4627 - 6.1871) = 2278\text{ kW}$$

$$\dot{X}_{\text{destroyed},41} = \dot{m}_w T_0 \left(s_1 - s_4 + \frac{q_{\text{out}}}{T_{\text{sink}}} \right)$$

$$= (28.21\text{ kg/s})(293\text{ K})\left(0.8320 - 6.1871 + \frac{1876.2\text{ kJ/kg}}{293\text{ K}} \right) = 8665\text{ kW}$$

$$\dot{X}_{\text{destroyed,heat exchanger}} = \dot{m}_a T_0 \Delta s_{89} + \dot{m}_w T_0 \Delta s_{23} = \dot{m}_a T_0 \left(c_p \ln\frac{T_9}{T_8} \right) + \dot{m}_a T_0 (s_3 - s_2)$$

$$= (279.3)(293)\left[(1.005)\ln\frac{548.6}{819.5} \right] + (28.21)(293)(6.1871 - 0.8320)$$

$$= 11260\text{ kW}$$

$$\dot{X}_{\text{destroyed},56} = \dot{m}_a T_0 \left(c_p \ln\frac{T_6}{T_5} - R\ln\frac{P_6}{P_5} \right) = (279.3)(293)\left[(1.005)\ln\frac{572.7}{293} - (0.287)\ln(8) \right] = 6280\text{ kW}$$

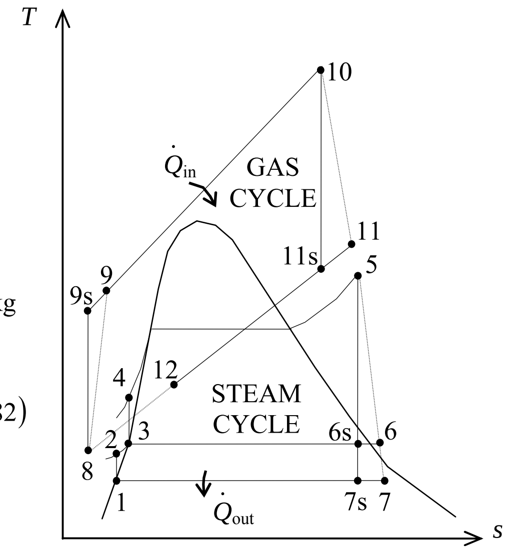
$$\dot{X}_{\text{destroyed},67} = \dot{m}_a T_0 \left(c_p \ln\frac{T_7}{T_6} - \frac{q_{\text{in}}}{T_{\text{source}}} \right) = (279.3)(293)\left[(1.005)\ln\frac{1373}{572.7} - \frac{804.3}{1373} \right] = \mathbf{23{,}970\ kW}$$

$$\dot{X}_{\text{destroyed},78} = \dot{m}_a T_0 \left(c_p \ln\frac{T_8}{T_7} - R\ln\frac{P_8}{P_7} \right) = (279.3)(293)\left[(1.005)\ln\frac{819.5}{1373} - (0.287)\ln\left(\frac{1}{8}\right) \right] = 6396\text{ kW}$$

The largest exergy destruction occurs during the heat addition process in the combustor of the gas cycle.

10-89 A 450-MW combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal Rankine cycle with an open feedwater heater. The mass flow rate of air to steam, the required rate of heat input in the combustion chamber, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.



Analysis (*a*) Using the properties of air from Table A-17, the analysis of gas cycle yields

$$T_8 = 300 \text{ K} \longrightarrow h_8 = 300.19 \text{ kJ/kg}$$
$$P_{r_8} = 1.386$$

$$P_{r_9} = \frac{P_9}{P_8} P_{r_8} = (14)(1.386) = 19.40 \longrightarrow h_{9s} = 635.5 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{9s} - h_8}{h_9 - h_8} \longrightarrow h_9 = h_8 + (h_{9s} - h_8)/\eta_C = 300.19 + (635.5 - 300.19)/(0.82) = 709.1 \text{ kJ/kg}$$

$$T_{10} = 1400 \text{ K} \longrightarrow h_{10} = 1515.42 \text{ kJ/kg}$$
$$P_{r_{10}} = 450.5$$

$$P_{r_{11}} = \frac{P_{11}}{P_{10}} P_{r_{10}} = \left(\frac{1}{14}\right)(450.5) = 32.18 \longrightarrow h_{11s} = 735.8 \text{ kJ/kg}$$

$$\eta_T = \frac{h_{10} - h_{11}}{h_{10} - h_{11s}} \longrightarrow h_{11} = h_{10} - \eta_T (h_{10} - h_{11s}) = 1515.42 - (0.86)(1515.42 - 735.8) = 844.95 \text{ kJ/kg}$$

$$T_{12} = 460 \text{ K} \longrightarrow h_{12} = 462.02 \text{ kJ/kg}$$

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

$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.42 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,20\text{ kPa}} = 0.001017 \text{ m}^3\text{/kg}$$

$$w_{\text{pI,in}} = \boldsymbol{v}_1(P_2 - P_1) = (0.001017 \text{ m}^3\text{/kg})(600 - 20 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = 0.59 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 0.59 = 252.01 \text{ kJ/kg}$$

$$h_3 = h_{f\,@\,0.6\text{ MPa}} = 670.38 \text{ kJ/kg}$$
$$\boldsymbol{v}_3 = \boldsymbol{v}_{f\,@\,0.6\text{ MPa}} = 0.001101 \text{ m}^3\text{/kg}$$

$$w_{\text{pII,in}} = \boldsymbol{v}_3(P_4 - P_3) = (0.001101 \text{ m}^3\text{/kg})(8{,}000 - 600 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = 8.15 \text{ kJ/kg}$$

$$h_4 = h_3 + w_{\text{pI,in}} = 670.38 + 8.15 = 678.52 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 8 \text{ MPa} \\ T_5 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3139.4 \text{ kJ/kg} \\ s_5 = 6.3658 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.6 \text{ MPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{6.3658 - 1.9308}{4.8285} = 0.9184 \\ h_{6s} = h_f + x_{6s} h_{fg} = 670.38 + (0.9184)(2085.8) = 2585.9 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T (h_5 - h_{6s}) = 3139.4 - (0.86)(3139.4 - 2585.9) = 2663.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 20 \text{ kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_{7s} = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.3658 - 0.8320}{7.0752} = 0.7820 \\ h_{7s} = h_f + x_7 h_{fg} = 251.42 + (0.7820)(2357.5) = 2095.1 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_7}{h_5 - h_{7s}} \longrightarrow h_7 = h_5 - \eta_T (h_5 - h_{7s}) = 3139.4 - (0.86)(3139.4 - 2095.1) = 2241.3 \text{ kJ/kg}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s (h_5 - h_4) = \dot{m}_{\text{air}} (h_{11} - h_{12})$$

$$\frac{\dot{m}_{\text{air}}}{\dot{m}_s} = \frac{h_5 - h_4}{h_{11} - h_{12}} = \frac{3139.4 - 678.52}{844.95 - 462.02} = \mathbf{6.425 \text{ kg air / kg steam}}$$

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the open FWH, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1 - y) h_2 = (1) h_3$$

Thus,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{670.38 - 252.01}{2663.3 - 252.01} = 0.1735 \quad \text{(the fraction of steam extracted)}$$

$$\begin{aligned} w_T &= \eta_T \left[h_5 - h_6 + (1 - y)(h_6 - h_7) \right] \\ &= (0.86)\left[3139.4 - 2663.3 + (1 - 0.1735)(2663.3 - 2241.3)\right] = 824.5 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{net,steam}} &= w_T - w_{\text{p,in}} = w_T - (1 - y) w_{\text{p,I}} - w_{\text{p,II}} \\ &= 824.5 - (1 - 0.1735)(0.59) - 8.15 = 815.9 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{net,gas}} &= w_T - w_{C,\text{in}} = (h_{10} - h_{11}) - (h_9 - h_8) \\ &= 1515.42 - 844.95 - (709.1 - 300.19) = 261.56 \text{ kJ/kg} \end{aligned}$$

The net work output per unit mass of gas is

$$w_{\text{net}} = w_{\text{net,gas}} + \frac{1}{6.425} w_{\text{net,steam}} = 261.56 + \frac{1}{6.425}(815.9) = 388.55 \text{ kJ/kg}$$

$$\dot{m}_{\text{air}} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{450{,}000 \text{ kJ/s}}{388.55 \text{ kJ/kg}} = 1158.2 \text{ kg/s}$$

and $\quad \dot{Q}_{\text{in}} = \dot{m}_{\text{air}} (h_{10} - h_9) = (1158.2 \text{ kg/s})(1515.42 - 709.1) \text{ kJ/kg} = \mathbf{933{,}850 \text{ kW}}$

(*c*) $\quad \eta_{th} = \dfrac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \dfrac{450{,}000 \text{ kW}}{933{,}850 \text{ kW}} = \mathbf{48.2\%}$

10-90 EES Problem 10-89 is reconsidered. The effect of the gas cycle pressure ratio on the ratio of gas flow rate to steam flow rate and cycle thermal efficiency is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"Input data"
T[8] = 300 [K]            "Gas compressor inlet"
P[8] = 14.7 [kPa]         "Assumed air inlet pressure"
"Pratio = 14"             "Pressure ratio for gas compressor"
T[10] = 1400 [K]          "Gas turbine inlet"
T[12] = 460 [K]           "Gas exit temperature from Gas-to-steam heat exchanger "
P[12] = P[8]              "Assumed air exit pressure"
W_dot_net=450 [MW]
Eta_comp = 0.82
Eta_gas_turb = 0.86
Eta_pump = 1.0
Eta_steam_turb = 0.86
P[5] = 8000 [kPa]          "Steam turbine inlet"
T[5] =(400+273) "K"        "Steam turbine inlet"
P[6] = 600 [kPa]          "Extraction pressure for steam open feedwater heater"
P[7] = 20 [kPa]           "Steam condenser pressure"

"GAS POWER CYCLE ANALYSIS"

"Gas Compressor anaysis"
s[8]=ENTROPY(Air,T=T[8],P=P[8])
ss9=s[8] "For the ideal case the entropies are constant across the compressor"
P[9] = Pratio*P[8]
Ts9=temperature(Air,s=ss9,P=P[9])"Ts9 is the isentropic value of T[9] at compressor exit"
Eta_comp = w_gas_comp_isen/w_gas_comp "compressor adiabatic efficiency, w_comp >
w_comp_isen"
h[8] + w_gas_comp_isen =hs9"SSSF conservation of energy for the isentropic compressor,
assuming: adiabatic, ke=pe=0 per unit gas mass flow rate in kg/s"
h[8]=ENTHALPY(Air,T=T[8])
hs9=ENTHALPY(Air,T=Ts9)
h[8] + w_gas_comp = h[9]"SSSF conservation of energy for the actual compressor, assuming:
adiabatic, ke=pe=0"
T[9]=temperature(Air,h=h[9])
s[9]=ENTROPY(Air,T=T[9],P=P[9])

"Gas Cycle External heat exchanger analysis"
h[9] + q_in = h[10]"SSSF conservation of energy for the external heat exchanger, assuming W=0,
ke=pe=0"
h[10]=ENTHALPY(Air,T=T[10])
P[10]=P[9]                          "Assume process 9-10 is SSSF constant pressure"
Q_dot_in"MW"*1000"kW/MW"=m_dot_gas*q_in

"Gas Turbine analysis"
s[10]=ENTROPY(Air,T=T[10],P=P[10])
ss11=s[10] "For the ideal case the entropies are constant across the turbine"
P[11] = P[10] /Pratio
Ts11=temperature(Air,s=ss11,P=P[11])"Ts11 is the isentropic value of T[11] at gas turbine exit"
Eta_gas_turb = w_gas_turb /w_gas_turb_isen "gas turbine adiabatic efficiency, w_gas_turb_isen
> w_gas_turb"
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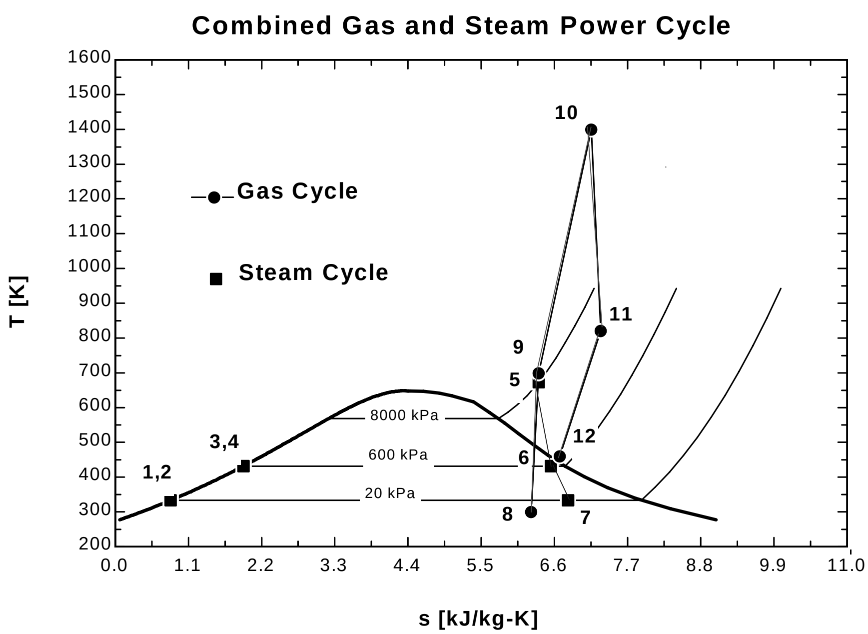
```
h[10] = w_gas_turb_isen + hs11"SSSF conservation of energy for the isentropic gas turbine,
assuming: adiabatic, ke=pe=0"
hs11=ENTHALPY(Air,T=Ts11)
h[10] = w_gas_turb + h[11]"SSSF conservation of energy for the actual gas turbine, assuming:
adiabatic, ke=pe=0"
T[11]=temperature(Air,h=h[11])
s[11]=ENTROPY(Air,T=T[11],P=P[11])

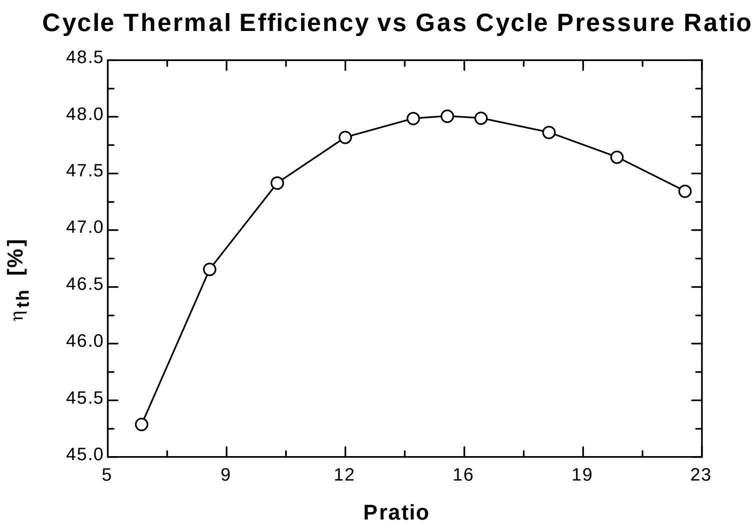
"Gas-to-Steam Heat Exchanger"
"SSSF conservation of energy for the gas-to-steam heat exchanger, assuming: adiabatic,
W=0, ke=pe=0"
m_dot_gas*h[11] + m_dot_steam*h[4] = m_dot_gas*h[12] + m_dot_steam*h[5]
h[12]=ENTHALPY(Air, T=T[12])
s[12]=ENTROPY(Air,T=T[12],P=P[12])

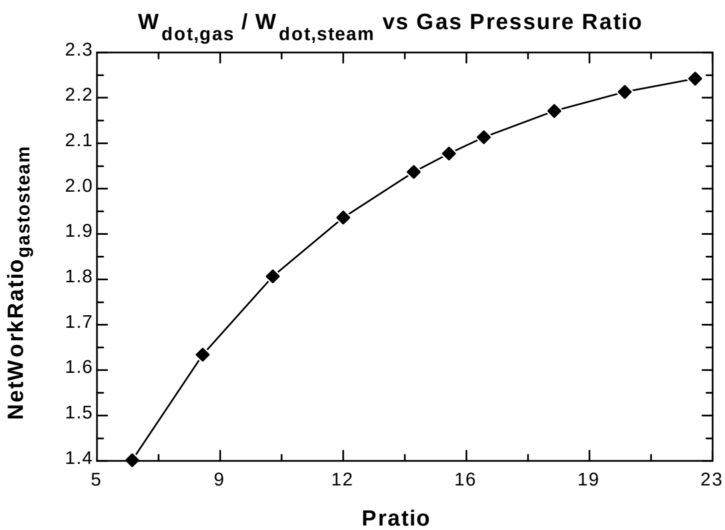
"STEAM CYCLE ANALYSIS"
"Steam Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'
P[1] = P[7]
P[2]=P[6]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Saturated liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"Open Feedwater Heater analysis"
y*h[6] + (1-y)*h[2] = 1*h[3] "Steady-flow conservation of energy"
P[3]=P[6]
h[3]=enthalpy(Fluid$,P=P[3],x=0)      "Condensate leaves heater as sat. liquid at P[3]"
T[3]=temperature(Fluid$,P=P[3],x=0)
s[3]=entropy(Fluid$,P=P[3],x=0)
"Boiler condensate pump  or Pump 2 analysis"
P[4] = P[5]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])
w_steam_pumps = (1-y)*w_pump1+ w_pump2 "Total steam pump work input/ mass steam"
"Steam Turbine analysis"
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s[5]=entropy(Fluid$,P=P[5],T=T[5])
ss6=s[5]
hs6=enthalpy(Fluid$,s=ss6,P=P[6])
Ts6=temperature(Fluid$,s=ss6,P=P[6])
h[6]=h[5]-Eta_steam_turb*(h[5]-hs6)"Definition of steam turbine efficiency"
T[6]=temperature(Fluid$,P=P[6],h=h[6])
s[6]=entropy(Fluid$,P=P[6],h=h[6])
ss7=s[5]
hs7=enthalpy(Fluid$,s=ss7,P=P[7])
Ts7=temperature(Fluid$,s=ss7,P=P[7])
```

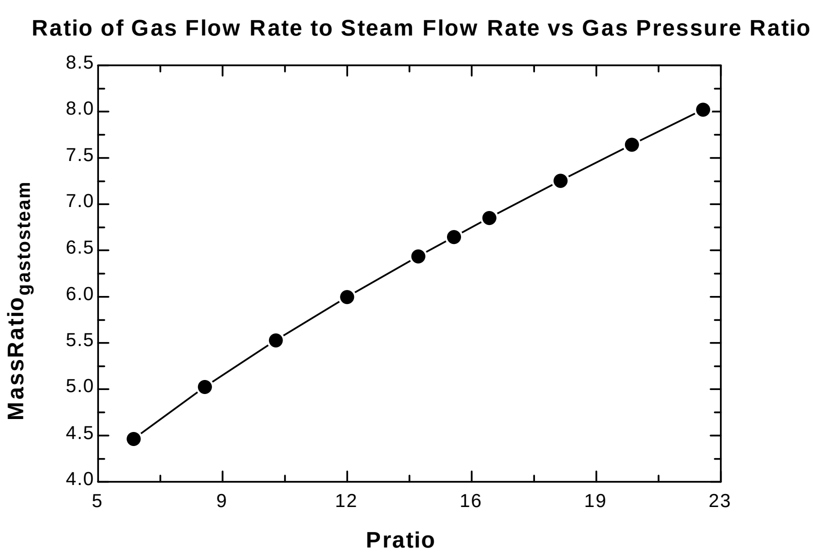
h[7]=h[5]-Eta_steam_turb*(h[5]-hs7)"Definition of steam turbine efficiency"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])
"SSSF conservation of energy for the steam turbine: adiabatic, neglect ke and pe"
h[5] = w_steam_turb + y*h[6] +(1-y)*h[7]
"Steam Condenser analysis"
(1-y)*h[7]=q_out+(1-y)*h[1]"SSSF conservation of energy for the Condenser per unit mass"
Q_dot_out*Convert(MW, kW)=m_dot_steam*q_out
"Cycle Statistics"
MassRatio_gastosteam =m_dot_gas/m_dot_steam
W_dot_net*Convert(MW, kW)=m_dot_gas*(w_gas_turb-w_gas_comp)+
m_dot_steam*(w_steam_turb - w_steam_pumps)"definition of the net cycle work"
Eta_th=W_dot_net/Q_dot_in*Convert(, %) "Cycle thermal efficiency, in percent"
Bwr=(m_dot_gas*w_gas_comp + m_dot_steam*w_steam_pumps)/(m_dot_gas*w_gas_turb +
m_dot_steam*w_steam_turb) "Back work ratio"
W_dot_net_steam = m_dot_steam*(w_steam_turb - w_steam_pumps)
W_dot_net_gas = m_dot_gas*(w_gas_turb - w_gas_comp)
NetWorkRatio_gastosteam = W_dot_net_gas/W_dot_net_steam

Pratio	$MassRatio_{gastosteam}$	W_{netgas} [kW]	$W_{netsteam}$ [kW]	η_{th} [%]	$NetWorkRatio_{gastosteam}$
6	4.463	262595	187405	45.29	1.401
8	5.024	279178	170822	46.66	1.634
10	5.528	289639	160361	47.42	1.806
12	5.994	296760	153240	47.82	1.937
14	6.433	301809	148191	47.99	2.037
15	6.644	303780	146220	48.01	2.078
16	6.851	305457	144543	47.99	2.113
18	7.253	308093	141907	47.87	2.171
20	7.642	309960	140040	47.64	2.213
22	8.021	311216	138784	47.34	2.242




Cycle Thermal Efficiency vs Gas Cycle Pressure Ratio
48.5
48.0
47.5
47.0
46.5
46.0
45.5
45.0
5
9
12
16
19
23
ηth [%]
Pratio

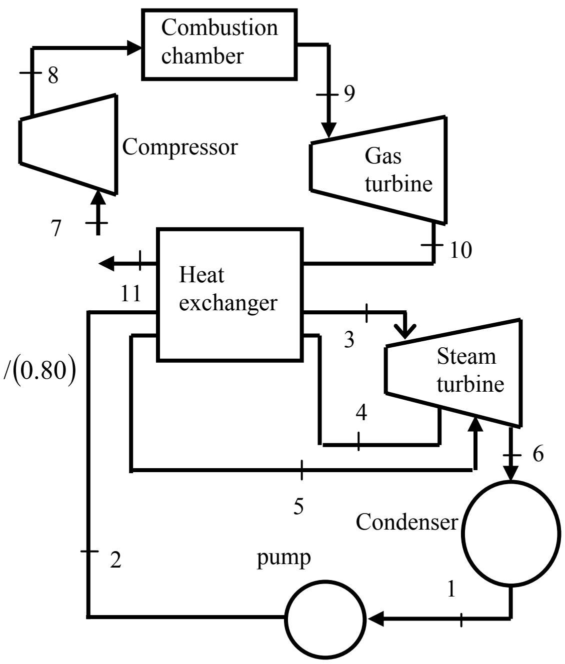

Wdot,gas / Wdot,steam vs Gas Pressure Ratio
2.3
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
5
9
12
16
19
23
NetWorkRatiogastosteam
Pratio


Ratio of Gas Flow Rate to Steam Flow Rate vs Gas Pressure Ratio
8.5
8.0
7.5
7.0
6.5
6.0
5.5
5.0
4.5
4.0
5
9
12
16
19
23
MassRatiogastosteam
Pratio

10-91 A combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal reheat Rankine cycle. The moisture percentage at the exit of the low-pressure turbine, the steam temperature at the inlet of the high-pressure turbine, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) We obtain the air properties from EES. The analysis of gas cycle is as follows



$$T_7 = 15°\text{C} \longrightarrow h_7 = 288.50 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_7 = 15°\text{C} \\ P_7 = 100 \text{ kPa} \end{array}\right\} s_7 = 5.6648 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 700 \text{ kPa} \\ s_8 = s_7 \end{array}\right\} h_{8s} = 503.47 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{8s} - h_7}{h_8 - h_7} \longrightarrow h_8 = h_7 + (h_{8s} - h_7)/\eta_C$$
$$= 290.16 + (503.47 - 290.16)/(0.80)$$
$$= 557.21 \text{ kJ/kg}$$

$$T_9 = 950°\text{C} \longrightarrow h_9 = 1304.8 \text{ kJ/kg}$$

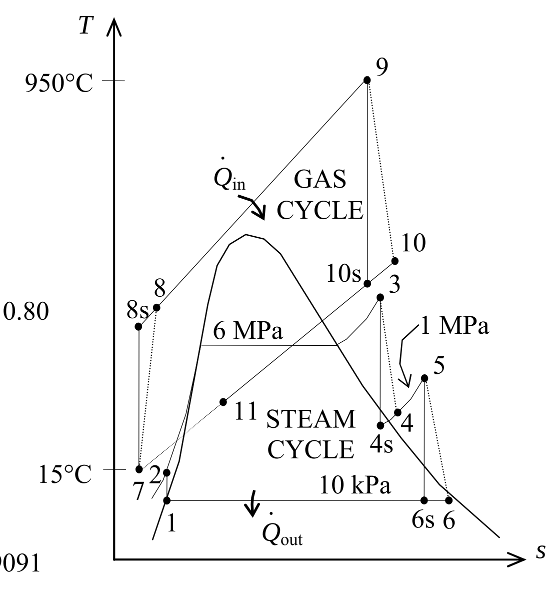
$$\left.\begin{array}{l} T_9 = 950°\text{C} \\ P_9 = 700 \text{ kPa} \end{array}\right\} s_9 = 6.6456 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{10} = 100 \text{ kPa} \\ s_{10} = s_9 \end{array}\right\} h_{10s} = 763.79 \text{ kJ/kg}$$

$$\eta_T = \frac{h_9 - h_{10}}{h_9 - h_{10s}} \longrightarrow h_{10} = h_9 - \eta_T (h_9 - h_{10s})$$
$$= 1304.8 - (0.80)(1304.8 - 763.79)$$
$$= 871.98 \text{ kJ/kg}$$

$$T_{11} = 200°\text{C} \longrightarrow h_{11} = 475.62 \text{ kJ/kg}$$

From the steam tables (Tables A-4, A-5, and A-6 or from EES),



$$h_1 = h_{f\text{ @ 10 kPa}} = 191.81 \text{ kJ/kg}$$
$$v_1 = v_{f\text{ @ 10 kPa}} = 0.00101 \text{ m}^3\text{/kg}$$

$$w_{\text{pI,in}} = v_1(P_2 - P_1)/\eta_p$$
$$= (0.00101 \text{ m}^3\text{/kg})(6000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)/0.80$$
$$= 7.56 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 7.65 = 199.37 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 1 \text{ MPa} \\ T_5 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3264.5 \text{ kJ/kg} \\ s_5 = 7.4670 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{7.4670 - 0.6492}{7.4996} = 0.9091 \\ h_{6s} = h_f + x_{6s} h_{fg} = 191.81 + (0.9091)(2392.1) = 2366.4 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T(h_5 - h_{6s}) = 3264.5 - (0.80)(3264.5 - 2366.4) = 2546.0 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ h_6 = 2546.5 \text{ kJ/kg} \end{array}\right\} x_6 = 0.9842$$

Moisture Percentage $= 1 - x_6 = 1 - 0.9842 = 0.0158 =$ **1.6%**

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$

$$\dot{m}_s(h_3 - h_2) + \dot{m}_s(h_5 - h_4) = \dot{m}_{\text{air}}(h_{10} - h_{11})$$

$$(1.15)[(3346.5 - 199.37) + (3264.5 - h_4)] = (10)(871.98 - 475.62) \longrightarrow h_4 = 2965.0 \text{ kJ/kg}$$

Also,

$$\left.\begin{array}{l} P_3 = 6 \text{ MPa} \\ T_3 = ? \end{array}\right\} \begin{array}{l} h_3 = \\ s_3 = \end{array} \qquad \left.\begin{array}{l} P_4 = 1 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} h_{4s} =$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s})$$

The temperature at the inlet of the high-pressure turbine may be obtained by a trial-error approach or using EES from the above relations. The answer is T_3 = **468.0ºC**. Then, the enthalpy at state 3 becomes: h_3 = 3346.5 kJ/kg

(*c*) $\dot{W}_{\text{T,gas}} = \dot{m}_{\text{air}}(h_9 - h_{10}) = (10 \text{ kg/s})(1304.8 - 871.98) \text{ kJ/kg} = 4328 \text{ kW}$

$$\dot{W}_{\text{C,gas}} = \dot{m}_{\text{air}}(h_8 - h_7) = (10 \text{ kg/s})(557.21 - 288.50) \text{ kJ/kg} = 2687 \text{ kW}$$

$$\dot{W}_{\text{net,gas}} = \dot{W}_{\text{T,gas}} - \dot{W}_{\text{C,gas}} = 4328 - 2687 = 1641 \text{ kW}$$

$$\dot{W}_{\text{T,steam}} = \dot{m}_s(h_3 - h_4 + h_5 - h_6) = (1.15 \text{ kg/s})(3346.5 - 2965.0 + 3264.5 - 2546.0) \text{ kJ/kg} = 1265 \text{ kW}$$

$$\dot{W}_{\text{P,steam}} = \dot{m}_s w_{pump} = (1.15 \text{ kg/s})(7.564) \text{ kJ/kg} = 8.7 \text{ kW}$$

$$\dot{W}_{\text{net,steam}} = \dot{W}_{\text{T,steam}} - \dot{W}_{\text{P,steam}} = 1265 - 8.7 = 1256 \text{ kW}$$

$$\dot{W}_{\text{net,plant}} = \dot{W}_{\text{net,gas}} + \dot{W}_{\text{net,steam}} = 1641 + 1256 = \mathbf{2897 \text{ kW}}$$

(*d*) $\dot{Q}_{\text{in}} = \dot{m}_{\text{air}}(h_9 - h_8) = (10 \text{ kg/s})(1304.8 - 557.21) \text{ kJ/kg} = 7476 \text{ kW}$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,plant}}}{\dot{Q}_{\text{in}}} = \frac{2897 \text{ kW}}{7476 \text{ kW}} = 0.388 = \mathbf{38.8\%}$$

Special Topic: Binary Vapor Cycles

10-92C Binary power cycle is a cycle which is actually a combination of two cycles; one in the high temperature region, and the other in the low temperature region. Its purpose is to increase thermal efficiency.

10-93C Consider the heat exchanger of a binary power cycle. The working fluid of the topping cycle (cycle A) enters the heat exchanger at state 1 and leaves at state 2. The working fluid of the bottoming cycle (cycle B) enters at state 3 and leaves at state 4. Neglecting any changes in kinetic and potential energies, and assuming the heat exchanger is well-insulated, the steady-flow energy balance relation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0\,\text{(steady)}} = 0$$

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

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$

$$\dot{m}_A h_2 + \dot{m}_B h_4 = \dot{m}_A h_1 + \dot{m}_B h_3 \;\; or \;\; \dot{m}_A (h_2 - h_1) = \dot{m}_B (h_3 - h_4)$$

Thus,

$$\frac{\dot{m}_A}{\dot{m}_B} = \frac{h_3 - h_4}{h_2 - h_1}$$

10-94C Steam is not an ideal fluid for vapor power cycles because its critical temperature is low, its saturation dome resembles an inverted V, and its condenser pressure is too low.

10-95C Because mercury has a high critical temperature, relatively low critical pressure, but a very low condenser pressure. It is also toxic, expensive, and has a low enthalpy of vaporization.

10-96C In binary vapor power cycles, both cycles are vapor cycles. In the combined gas-steam power cycle, one of the cycles is a gas cycle.

Review Problems

10-97 It is to be demonstrated that the thermal efficiency of a combined gas-steam power plant η_{cc} can be expressed as $\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s$ where $\eta_g = W_g / Q_{in}$ and $\eta_s = W_s / Q_{g,out}$ are the thermal efficiencies of the gas and steam cycles, respectively, and the efficiency of a combined cycle is to be obtained.

Analysis The thermal efficiencies of gas, steam, and combined cycles can be expressed as

$$\eta_{cc} = \frac{W_{total}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

$$\eta_g = \frac{W_g}{Q_{in}} = 1 - \frac{Q_{g,out}}{Q_{in}}$$

$$\eta_s = \frac{W_s}{Q_{g,out}} = 1 - \frac{Q_{out}}{Q_{g,out}}$$

where Q_{in} is the heat supplied to the gas cycle, where Q_{out} is the heat rejected by the steam cycle, and where $Q_{g,out}$ is the heat rejected from the gas cycle and supplied to the steam cycle.

Using the relations above, the expression $\eta_g + \eta_s - \eta_g \eta_s$ can be expressed as

$$\begin{aligned}
\eta_g + \eta_s - \eta_g \eta_s &= \left(1 - \frac{Q_{g,out}}{Q_{in}}\right) + \left(1 - \frac{Q_{out}}{Q_{g,out}}\right) - \left(1 - \frac{Q_{g,out}}{Q_{in}}\right)\left(1 - \frac{Q_{out}}{Q_{g,out}}\right) \\
&= 1 - \frac{Q_{g,out}}{Q_{in}} + 1 - \frac{Q_{out}}{Q_{g,out}} - 1 + \frac{Q_{g,out}}{Q_{in}} + \frac{Q_{out}}{Q_{g,out}} - \frac{Q_{out}}{Q_{in}} \\
&= 1 - \frac{Q_{out}}{Q_{in}} \\
&= \eta_{cc}
\end{aligned}$$

Therefore, the proof is complete. Using the relation above, the thermal efficiency of the given combined cycle is determined to be

$$\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s = 0.4 + 0.30 - 0.40 \times 0.30 = \mathbf{0.58}$$

10-98 The thermal efficiency of a combined gas-steam power plant η_{cc} can be expressed in terms of the thermal efficiencies of the gas and the steam turbine cycles as $\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s$. It is to be shown that the value of η_{cc} is greater than either of η_g or η_s.

Analysis By factoring out terms, the relation $\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s$ can be expressed as

$$\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s = \eta_g + \underbrace{\eta_s (1 - \eta_g)}_{\substack{\text{Positive since} \\ \eta_g < 1}} > \eta_g$$

or

$$\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s = \eta_s + \underbrace{\eta_g (1 - \eta_s)}_{\substack{\text{Positive since} \\ \eta_s < 1}} > \eta_s$$

Thus we conclude that the combined cycle is more efficient than either of the gas turbine or steam turbine cycles alone.

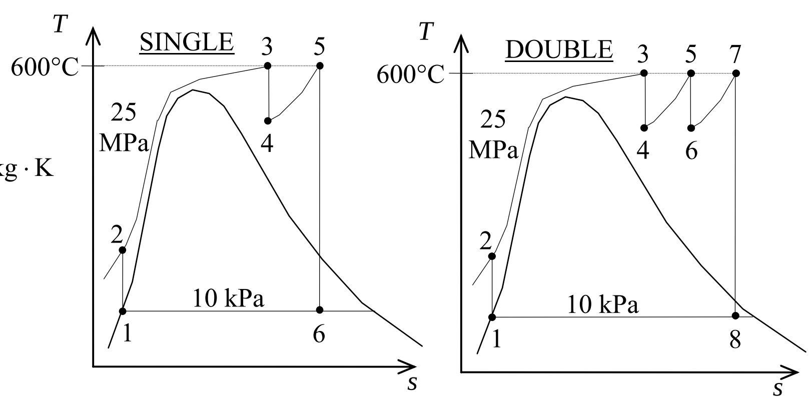
10-99 A steam power plant operating on the ideal Rankine cycle with reheating is considered. The reheat pressures of the cycle are to be determined for the cases of single and double reheat.

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

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

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ x_6 = 0.92 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6492 + (0.92)(7.4996) = 7.5488 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} T_5 = 600°\text{C} \\ s_5 = s_6 \end{array}\right\} P_5 = \mathbf{2780\ kPa}$$



(*b*) Double Reheat :

$$\left.\begin{array}{l} P_3 = 25 \text{ MPa} \\ T_3 = 600°\text{C} \end{array}\right\} s_3 = 6.3637 \text{ kJ/kg} \cdot \text{K}$$

$$\begin{array}{l} P_4 = P_x \\ s_4 = s_3 \end{array} \quad \text{and} \quad \begin{array}{l} P_5 = P_x \\ T_5 = 600°\text{C} \end{array}$$

Any pressure P_x selected between the limits of 25 MPa and 2.78 MPa will satisfy the requirements, and can be used for the double reheat pressure.

10-100E A geothermal power plant operating on the simple Rankine cycle using an organic fluid as the working fluid is considered. The exit temperature of the geothermal water from the vaporizer, the rate of heat rejection from the working fluid in the condenser, the mass flow rate of geothermal water at the preheater, and the thermal efficiency of the Level I cycle of this plant are to be determined.

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

Analysis (*a*) The exit temperature of geothermal water from the vaporizer is determined from the steady-flow energy balance on the geothermal water (brine),

$$\dot{Q}_{\text{brine}} = \dot{m}_{\text{brine}} c_p (T_2 - T_1)$$
$$-22{,}790{,}000 \text{ Btu/h} = (384{,}286 \text{ lbm/h})(1.03 \text{ Btu/lbm}\cdot°\text{F})(T_2 - 325°\text{F})$$
$$T_2 = \mathbf{267.4°F}$$

(*b*) The rate of heat rejection from the working fluid to the air in the condenser is determined from the steady-flow energy balance on air,

$$\dot{Q}_{\text{air}} = \dot{m}_{\text{air}} c_p (T_9 - T_8)$$
$$= (4{,}195{,}100 \text{ lbm/h})(0.24 \text{ Btu/lbm}\cdot°\text{F})(84.5 - 55°\text{F})$$
$$= \mathbf{29.7\ MBtu/h}$$

(*c*) The mass flow rate of geothermal water at the preheater is determined from the steady-flow energy balance on the geothermal water,

$$\dot{Q}_{\text{geo}} = \dot{m}_{\text{geo}} c_p (T_{\text{out}} - T_{\text{in}})$$
$$-11{,}140{,}000 \text{ Btu/h} = \dot{m}_{\text{geo}} (1.03 \text{ Btu/lbm}\cdot°\text{F})(154.0 - 211.8°\text{F})$$
$$\dot{m}_{\text{geo}} = \mathbf{187{,}120\ lbm/h}$$

(*d*) The rate of heat input is

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{vaporizer}} + \dot{Q}_{\text{reheater}} = 22{,}790{,}000 + 11{,}140{,}000$$
$$= 33{,}930{,}000 \text{ Btu/h}$$

and

$$\dot{W}_{\text{net}} = 1271 - 200 = 1071 \text{ kW}$$

Then,

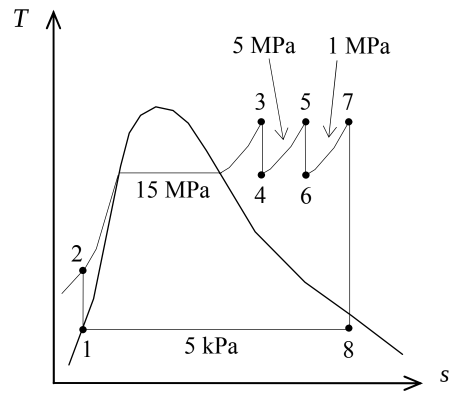
$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1071 \text{ kW}}{33{,}930{,}000 \text{ Btu/h}}\left(\frac{3412.14 \text{ Btu}}{1 \text{ kWh}}\right) = \mathbf{10.8\%}$$

10-101 A steam power plant operating on an ideal Rankine cycle with two stages of reheat is considered. 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),

$$h_1 = h_{f\ @\ 5\text{ kPa}} = 137.75\text{ kJ/kg}$$
$$v_1 = v_{f\ @\ 5\text{ kPa}} = 0.001005\ \text{m}^3/\text{kg}$$
$$w_{p,in} = v_1(P_2 - P_1) = (0.001005\ \text{m}^3/\text{kg})(15{,}000 - 5\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 15.07\text{ kJ/kg}$$
$$h_2 = h_1 + w_{p,in} = 137.75 + 15.07 = 152.82\text{ kJ/kg}$$



$$\left.\begin{array}{l} P_3 = 15\text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3310.8\text{ kJ/kg} \\ s_3 = 6.3480\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 5\text{ MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 3007.4\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 5\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3434.7\text{ kJ/kg} \\ s_5 = 6.9781\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 1\text{ MPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 2971.3\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 1\text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3479.1\text{ kJ/kg} \\ s_7 = 7.7642\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 5\text{ kPa} \\ s_8 = s_7 \end{array}\right\} \begin{array}{l} x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{7.7642 - 0.4762}{7.9176} = 0.9204 \\ h_8 = h_f + x_8 h_{fg} = 137.75 + (0.9204)(2423.0) = 2367.9\text{ kJ/kg} \end{array}$$

Then,

$$q_{in} = (h_3 - h_2) + (h_5 - h_4) + (h_7 - h_6)$$
$$= 3310.8 - 152.82 + 3434.7 - 3007.4 + 3479.1 - 2971.3 = 4093.1\text{ kJ/kg}$$
$$q_{out} = h_8 - h_1 = 2367.9 - 137.75 = 2230.2\text{ kJ/kg}$$
$$w_{net} = q_{in} - q_{out} = 4093.1 - 2230.2 = 1862.9\text{ kJ/kg}$$

Thus,

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{1862.9\text{ kJ/kg}}{4093.1\text{ kJ/kg}} = \mathbf{45.5\%}$$

(*b*) $$\dot{m} = \frac{\dot{W}_{net}}{w_{net}} = \frac{120{,}000\text{ kJ/s}}{1862.9\text{ kJ/kg}} = \mathbf{64.4\ kg/s}$$

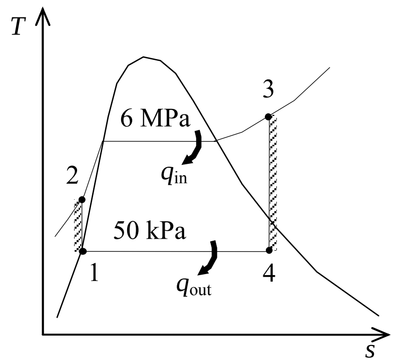
10-102 A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The thermal efficiency of the cycle is to be compared when it is operated so that the liquid enters the pump as a saturated liquid against that when the liquid enters as a subcooled liquid.

determined power produced by the turbine and consumed by the pump are to be determined.

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

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

$$h_1 = h_{f\,@\,50\,\text{kPa}} = 340.54\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,20\,\text{kPa}} = 0.001030\ \text{m}^3/\text{kg}$$



$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001030\ \text{m}^3/\text{kg})(6000-50)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 6.13\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 340.54 + 6.13 = 346.67\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 6000\ \text{kPa} \\ T_3 &= 600°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3658.8\ \text{kJ/kg} \\ s_3 &= 7.1693\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 50\ \text{kPa} \\ s_4 &= s_3 \end{aligned}\right\} \begin{aligned} x_4 &= \frac{s_4 - s_f}{s_{fg}} = \frac{7.1693-1.0912}{6.5019} = 0.9348 \\ h_4 &= h_f + x_4 h_{fg} = 340.54 + (0.9348)(2304.7) = 2495.0\ \text{kJ/kg} \end{aligned}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3658.8 - 346.67 = 3312.1\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2495.0 - 340.54 = 2154.5\ \text{kJ/kg}$$

and the thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2154.5}{3312.1} = \mathbf{0.3495}$$

When the liquid enters the pump 11.3°C cooler than a saturated liquid at the condenser pressure, the enthalpies become

$$\left.\begin{aligned} P_1 &= 50\ \text{kPa} \\ T_1 &= T_{\text{sat}\,@\,50\,\text{kPa}} - 11.3 = 81.3 - 11.3 = 70°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &\cong h_{f\,@\,70°\text{C}} = 293.07\ \text{kJ/kg} \\ \boldsymbol{v}_1 &\cong \boldsymbol{v}_{f\,@\,70°\text{C}} = 0.001023\ \text{m}^3/\text{kg} \end{aligned}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001023\ \text{m}^3/\text{kg})(6000-50)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 6.09\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 293.07 + 6.09 = 299.16\ \text{kJ/kg}$$

Then,

$$q_{\text{in}} = h_3 - h_2 = 3658.8 - 299.16 = 3359.6\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2495.0 - 293.09 = 2201.9\ \text{kJ/kg}$$

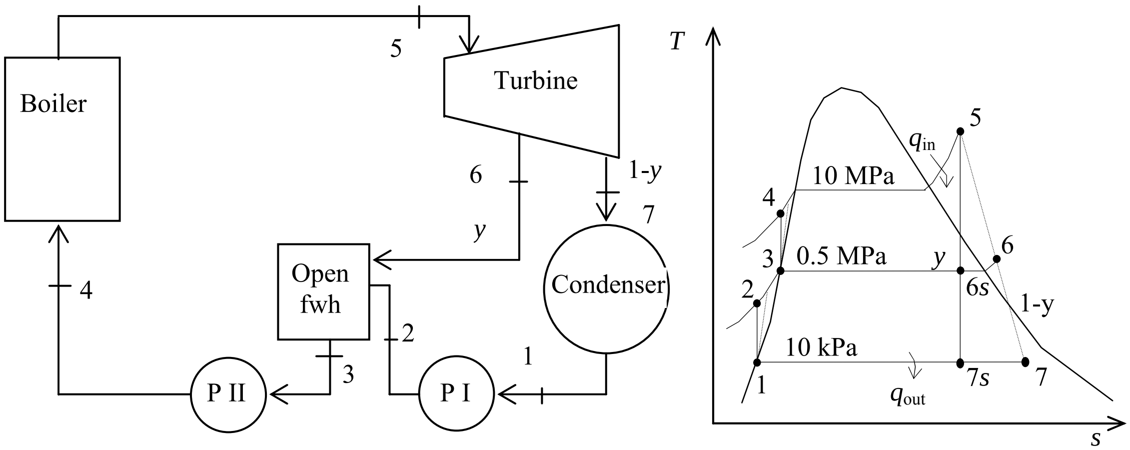
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2201.9}{3359.6} = \mathbf{0.3446}$$

The thermal efficiency slightly decreases as a result of subcooling at the pump inlet.

10-103 An 150-MW steam power plant operating on a regenerative Rankine cycle with an open feedwater heater is considered. The mass flow rate of steam through the boiler, the thermal efficiency of the cycle, and the irreversibility associated with the regeneration process 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),

$$h_1 = h_{f\ @10\,\text{kPa}} = 191.81\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @10\,\text{kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= (0.00101\ \text{m}^3/\text{kg})(500 - 10\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 0.52\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.52 = 192.33\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.5\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\ @0.5\,\text{MPa}} = 640.09\ \text{kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f\ @0.5\,\text{MPa}} = 0.001093\ \text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3)/\eta_p \\ &= (0.001093\ \text{m}^3/\text{kg})(10{,}000 - 500\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 10.93\ \text{kJ/kg} \end{aligned}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.93 = 651.02\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\ \text{MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\ \text{kJ/kg} \\ s_5 = 6.5995\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{6s} = 0.5\ \text{MPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{6.5995 - 1.8604}{4.9603} = 0.9554 \\ h_{6s} = h_f + x_{6s}h_{fg} = 640.09 + (0.9554)(2108.0) \\ \quad = 2654.1\ \text{kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T(h_5 - h_{6s}) = 3375.1 - (0.80)(3375.1 - 2654.1) = 2798.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{7s} = 10 \text{ kPa} \\ s_{7s} = s_5 \end{array}\right\} \begin{array}{l} x_{7s} = \dfrac{s_{7s} - s_f}{s_{fg}} = \dfrac{6.5995 - 0.6492}{7.4996} = 0.7934 \\ h_{7s} = h_f + x_{7s} h_{fg} = 191.81 + (0.7934)(2392.1) \\ \quad = 2089.7 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_7}{h_5 - h_{7s}} \longrightarrow h_7 = h_5 - \eta_T(h_5 - h_{7s}) = 3375.1 - (0.80)(3375.1 - 2089.7) = 2346.8 \text{ kJ/kg}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y) h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{640.09 - 192.33}{2798.3 - 192.33} = 0.1718$$

Then,
$$q_{\text{in}} = h_5 - h_4 = 3375.1 - 651.02 = 2724.1 \text{ kJ/kg}$$
$$q_{\text{out}} = (1-y)(h_7 - h_1) = (1 - 0.1718)(2346.8 - 191.81) = 1784.7 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2724.1 - 1784.7 = 939.4 \text{ kJ/kg}$$

and

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150{,}000 \text{ kJ/s}}{939.4 \text{ kJ/kg}} = \mathbf{159.7 \text{ kg/s}}$$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1784.7 \text{ kJ/kg}}{2724.1 \text{ kJ/kg}} = \mathbf{34.5\%}$$

Also,

$$\left.\begin{array}{l} P_6 = 0.5 \text{ MPa} \\ h_6 = 2798.3 \text{ kJ/kg} \end{array}\right\} s_6 = 6.9453 \text{ kJ/kg}\cdot\text{K}$$

$$s_3 = s_{f\,@\,0.5\text{ MPa}} = 1.8604 \text{ kJ/kg}\cdot\text{K}$$
$$s_2 = s_1 = s_{f\,@\,10\text{ kPa}} = 0.6492 \text{ kJ/kg}\cdot\text{K}$$

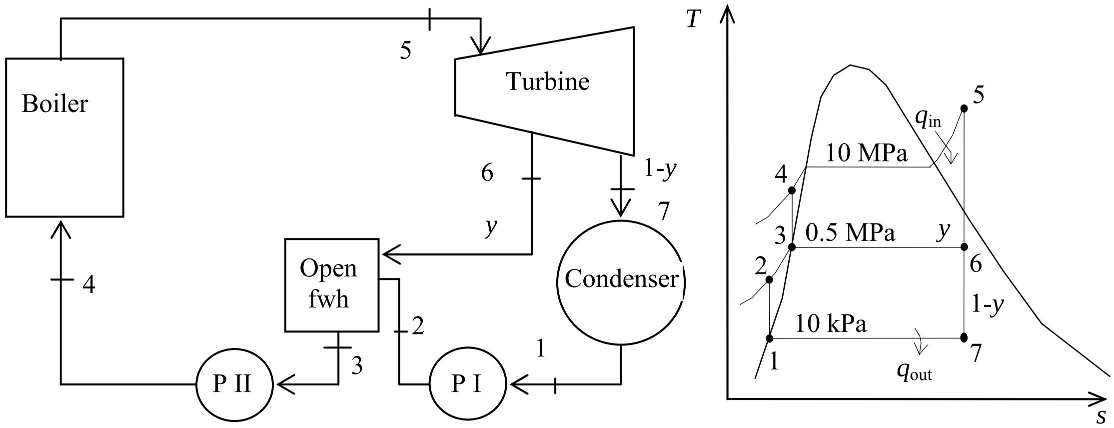
Then the irreversibility (or exergy destruction) associated with this regeneration process is

$$i_{\text{regen}} = T_0 s_{\text{gen}} = T_0\left(\sum m_e s_e - \sum m_i s_i + \frac{q_{\text{surr}}}{T_L}^{\nearrow 0}\right) = T_0[s_3 - y s_6 - (1-y) s_2]$$
$$= (303 \text{ K})[1.8604 - (0.1718)(6.9453) - (1 - 0.1718)(0.6492)] = \mathbf{39.25 \text{ kJ/kg}}$$

10-104 An 150-MW steam power plant operating on an ideal regenerative Rankine cycle with an open feedwater heater is considered. The mass flow rate of steam through the boiler, the thermal efficiency of the cycle, and the irreversibility associated with the regeneration process 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),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,10\text{ kPa}} = 0.00101\ \text{m}^3\text{/kg}$$

$$w_{\text{pI,in}} = \boldsymbol{v}_1(P_2 - P_1)$$
$$= (0.00101\ \text{m}^3\text{/kg})(500 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 0.50\text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.50 = 192.30\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.5\text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.5\text{ MPa}} = 640.09\text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f\,@\,0.5\text{ MPa}} = 0.001093\ \text{m}^3\text{/kg} \end{array}$$

$$w_{\text{pII,in}} = \boldsymbol{v}_3(P_4 - P_3)$$
$$= (0.001093\ \text{m}^3\text{/kg})(10{,}000 - 500\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 10.38\text{ kJ/kg}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.38 = 650.47\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\text{ kJ/kg} \\ s_5 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.5\text{ MPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.5995 - 1.8604}{4.9603} = 0.9554 \\ h_6 = h_f + x_6 h_{fg} = 640.09 + (0.9554)(2108.0) = 2654.1\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_7 = 10\text{ kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.5995 - 0.6492}{7.4996} = 0.7934 \\ h_7 = h_f + x_7 h_{fg} = 191.81 + (0.7934)(2392.1) = 2089.7\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y)h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{640.09 - 192.31}{2654.1 - 192.31} = 0.1819$$

Then, $q_{\text{in}} = h_5 - h_4 = 3375.1 - 650.47 = 2724.6 \text{ kJ/kg}$

$$q_{\text{out}} = (1-y)(h_7 - h_1) = (1 - 0.1819)(2089.7 - 191.81) = 1552.7 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2724.6 - 1552.7 = 1172.0 \text{ kJ/kg}$$

and $\dot{m} = \dfrac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \dfrac{150{,}000 \text{ kJ/s}}{1171.9 \text{ kJ/kg}} = \mathbf{128.0\ kg/s}$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1552.7 \text{ kJ/kg}}{2724.7 \text{ kJ/kg}} = \mathbf{43.0\%}$$

Also,

$$s_6 = s_5 = 6.5995 \text{ kJ/kg}\cdot\text{K}$$

$$s_3 = s_{f\,@\,0.5\text{ MPa}} = 1.8604 \text{ kJ/kg}\cdot\text{K}$$

$$s_2 = s_1 = s_{f\,@\,10\text{ kPa}} = 0.6492 \text{ kJ/kg}\cdot\text{K}$$

Then the irreversibility (or exergy destruction) associated with this regeneration process is

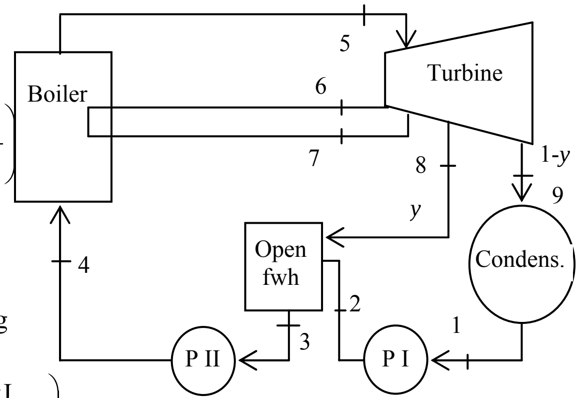
$$i_{\text{regen}} = T_0 s_{\text{gen}} = T_0\left(\sum m_e s_e - \sum m_i s_i + \frac{q_{\text{surr}}}{T_L}^{\nearrow 0}\right) = T_0\left[s_3 - y s_6 - (1-y)s_2\right]$$

$$= (303 \text{ K})\left[1.8604 - (0.1819)(6.5995) - (1 - 0.1819)(0.6492)\right] = \mathbf{39.0\ kJ/kg}$$

10-105 An ideal reheat-regenerative Rankine cycle with one open feedwater heater is considered. The fraction of steam extracted for regeneration 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),



$$h_1 = h_{f\,@\,15\text{ kPa}} = 225.94\text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,15\text{ kPa}} = 0.001014\ \text{m}^3/\text{kg}$$

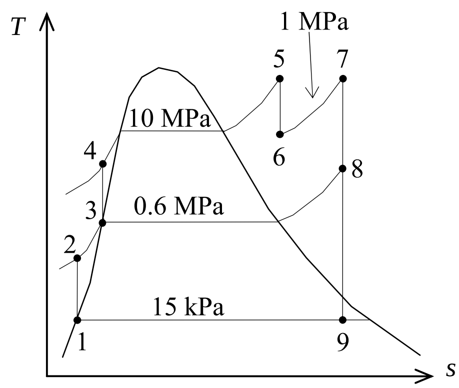
$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001014\ \text{m}^3/\text{kg})(600 - 15\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.59\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 225.94 + 0.59 = 226.53\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.6\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.6\text{ MPa}} = 670.38\text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f\,@\,0.6\text{ MPa}} = 0.001101\ \text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= (0.001101\ \text{m}^3/\text{kg})(10{,}000 - 600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.35\text{ kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pII,in}} = 670.38 + 10.35 = 680.73\text{ kJ/kg}$$



$$\left.\begin{array}{l} P_5 = 10\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\text{ kJ/kg} \\ s_5 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 1.0\text{ MPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 2783.8\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 1.0\text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3479.1\text{ kJ/kg} \\ s_7 = 7.7642\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 0.6\text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_8 = 3310.2\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 15\text{ kPa} \\ s_9 = s_7 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{7.7642 - 0.7549}{7.2522} = 0.9665 \\ h_9 = h_f + x_9 h_{fg} = 225.94 + (0.9665)(2372.3) = 2518.8\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_8 + (1-y) h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_8 - h_2} = \frac{670.38 - 226.53}{3310.2 - 226.53} = \mathbf{0.144}$$

(*b*) The thermal efficiency is determined from

$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6) = (3375.1 - 680.73) + (3479.1 - 2783.8) = 3389.7\text{ kJ/kg}$$

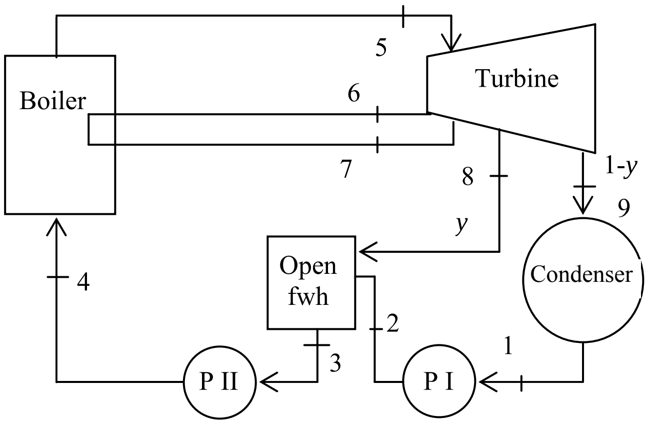
$$q_{\text{out}} = (1-y)(h_9 - h_1) = (1 - 0.1440)(2518.8 - 225.94) = 1962.7\text{ kJ/kg}$$

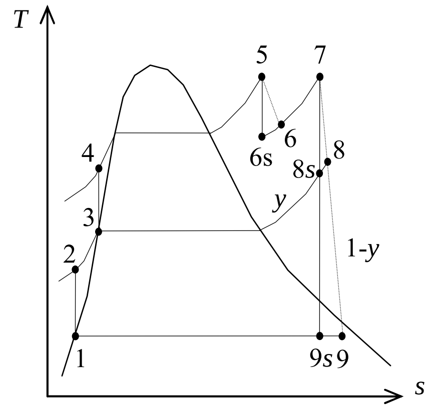
and $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1962.7\text{ kJ/kg}}{3389.7\text{ kJ/kg}} = \mathbf{42.1\%}$$

10-106 A nonideal reheat-regenerative Rankine cycle with one open feedwater heater is considered. The fraction of steam extracted for regeneration 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),

$$h_1 = h_{f@\,15\text{ kPa}} = 225.94\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,15\text{ kPa}} = 0.001014\text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{pI,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001014\text{ m}^3\text{/kg})(600 - 15\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.59\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{pI,\text{in}} = 225.94 + 0.59 = 226.54\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.6\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,0.6\text{ MPa}} = 670.38\text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f@\,0.6\text{ MPa}} = 0.001101\text{ m}^3\text{/kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= (0.001101\text{ m}^3\text{/kg})(10{,}000 - 600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.35\text{ kJ/kg} \end{aligned}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 670.38 + 10.35 = 680.73\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\text{ kJ/kg} \\ s_5 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{6s} = 1.0\text{ MPa} \\ s_{6s} = s_5 \end{array}\right\} h_{6s} = 2783.8\text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow \begin{aligned} h_6 &= h_5 - \eta_T(h_5 - h_{6s}) \\ &= 3375.1 - (0.84)(3375.1 - 2783.8) \\ &= 2878.4\text{ kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_7 = 1.0 \text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3479.1 \text{ kJ/kg} \\ s_7 = 7.7642 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_{8s} = 0.6 \text{ MPa} \\ s_{8s} = s_7 \end{array}\right\} h_{8s} = 3310.2 \text{ kJ/kg}$$

$$\eta_T = \frac{h_7 - h_8}{h_7 - h_{8s}} \longrightarrow h_8 = h_7 - \eta_T (h_7 - h_{8s}) = 3479.1 - (0.84)(3479.1 - 3310.2) = 3337.2 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{9s} = 15 \text{ kPa} \\ s_{9s} = s_7 \end{array}\right\} \begin{array}{l} x_{9s} = \dfrac{s_{9s} - s_f}{s_{fg}} = \dfrac{7.7642 - 0.7549}{7.2522} = 0.9665 \\ h_{9s} = h_f + x_{9s} h_{fg} = 225.94 + (0.9665)(2372.3) = 2518.8 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_7 - h_9}{h_7 - h_{9s}} \longrightarrow h_9 = h_7 - \eta_T (h_7 - h_{9s}) = 3479.1 - (0.84)(3479.1 - 2518.8) = 2672.5 \text{ kJ/kg}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_8 + (1 - y) h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_8 - h_2} = \frac{670.38 - 226.53}{3335.3 - 226.53} = \mathbf{0.1427}$$

(*b*) The thermal efficiency is determined from

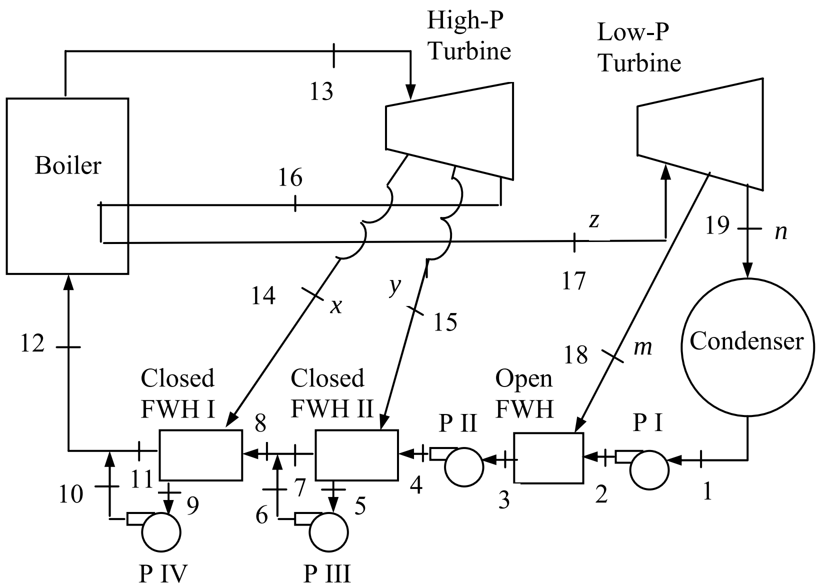
$$\begin{aligned} q_{\text{in}} &= (h_5 - h_4) + (h_7 - h_6) \\ &= (3375.1 - 680.73) + (3479.1 - 2878.4) = 3295.1 \text{ kJ/kg} \\ q_{\text{out}} &= (1 - y)(h_9 - h_1) = (1 - 0.1427)(2672.5 - 225.94) = 2097.2 \text{ kJ/kg} \end{aligned}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2097.2 \text{ kJ/kg}}{3295.1 \text{ kJ/kg}} = \mathbf{36.4\%}$$

10-107 A steam power plant operating on the ideal reheat-regenerative Rankine cycle with three feedwater heaters is considered. Various items for this system per unit of mass flow rate through the boiler are to be determined.

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



Analysis The compression processes in the pumps and the expansion processes in the turbines are isentropic. Also, the state of water at the inlet of pumps is saturated liquid. Then, from the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = 191.81\ \text{kJ/kg} \qquad h_{13} = 3139.4\ \text{kJ/kg}$$
$$h_2 = 191.90\ \text{kJ/kg} \qquad h_{14} = 3062.8\ \text{kJ/kg}$$
$$h_3 = 417.51\ \text{kJ/kg} \qquad h_{15} = 2931.8\ \text{kJ/kg}$$
$$h_4 = 421.05\ \text{kJ/kg} \qquad h_{16} = 2470.4\ \text{kJ/kg}$$
$$h_5 = 1049.7\ \text{kJ/kg} \qquad h_{17} = 3275.5\ \text{kJ/kg}$$
$$h_6 = 1052.8\ \text{kJ/kg} \qquad h_{18} = 2974.6\ \text{kJ/kg}$$
$$h_9 = 1213.8\ \text{kJ/kg} \qquad h_{19} = 2547.5\ \text{kJ/kg}$$
$$h_{10} = 1216.2\ \text{kJ/kg}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure. Then,

$$\left.\begin{array}{l} P_7 = 3500\ \text{kPa} \\ T_7 = T_5 = 242.6°\text{C} \end{array}\right\} h_7 = 1050.0\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_{11} = 6000\ \text{kPa} \\ T_{11} = T_9 = 275.6°\text{C} \end{array}\right\} h_{11} = 1213.1\ \text{kJ/kg}$$

Enthalpies at other states and the fractions of steam extracted from the turbines can be determined from mass and energy balances on cycle components as follows:

Mass Balances:

$$x + y + z = 1$$
$$m + n = z$$

Open feedwater heater:

$$mh_{18} + nh_2 = zh_3$$

Closed feedwater heater-II:

$$zh_4 + yh_{15} = zh_7 + yh_5$$

Closed feedwater heater-I:

$$(y+z)h_8 + xh_{14} = (y+z)h_{11} + xh_9$$

Mixing chamber after closed feedwater heater II:

$$zh_7 + yh_6 = (y+z)h_8$$

Mixing chamber after closed feedwater heater I:

$$xh_{10} + (y+z)h_{11} = 1h_{12}$$

Substituting the values and solving the above equations simultaneously using EES, we obtain

$$h_8 = 1050.7 \text{ kJ/kg}$$
$$h_{12} = 1213.3 \text{ kJ/kg}$$
$$x = \mathbf{0.08072}$$
$$y = \mathbf{0.2303}$$
$$z = 0.6890$$
$$m = \mathbf{0.05586}$$
$$n = \mathbf{0.6332}$$

Note that these values may also be obtained by a hand solution by using the equations above with some rearrangements and substitutions. Other results of the cycle are

$$w_{\text{T,out,HP}} = x(h_{13} - h_{14}) + y(h_{13} - h_{15}) + z(h_{13} - h_{16}) = \mathbf{514.9 \text{ kJ/kg}}$$
$$w_{\text{T,out,LP}} = m(h_{17} - h_{18}) + n(h_{17} - h_{19}) = \mathbf{477.8 \text{ kJ/kg}}$$
$$q_{\text{in}} = h_{13} - h_{12} + z(h_{17} - h_{16}) = \mathbf{2481 \text{ kJ/kg}}$$
$$q_{\text{out}} = n(h_{19} - h_1) = \mathbf{1492 \text{ kJ/kg}}$$
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1492}{2481} = \mathbf{0.3986}$$

10-108 EES The optimum bleed pressure for the open feedwater heater that maximizes the thermal efficiency of the cycle is to be determined using EES.

Analysis The EES program used to solve this problem as well as the solutions are given below.

```
"Given"
P_boiler=8000 [kPa]
P_cfwh1=6000 [kPa]
P_cfwh2=3500 [kPa]
P_reheat=300 [kPa]
P_ofwh=100 [kPa]
P_condenser=10 [kPa]
T_turbine=400 [C]

"Analysis"
Fluid$='steam_iapws'

"turbines"
h[13]=enthalpy(Fluid$, P=P_boiler, T=T_turbine)
s[13]=entropy(Fluid$, P=P_boiler, T=T_turbine)
h[14]=enthalpy(Fluid$, P=P_cfwh1, s=s[13])
h[15]=enthalpy(Fluid$, P=P_cfwh2, s=s[13])
h[16]=enthalpy(Fluid$, P=P_reheat, s=s[13])
h[17]=enthalpy(Fluid$, P=P_reheat, T=T_turbine)
s[17]=entropy(Fluid$, P=P_reheat, T=T_turbine)
h[18]=enthalpy(Fluid$, P=P_ofwh, s=s[17])
h[19]=enthalpy(Fluid$, P=P_condenser, s=s[17])

"pump I"
h[1]=enthalpy(Fluid$, P=P_condenser, x=0)
v[1]=volume(Fluid$, P=P_condenser, x=0)
w_pI_in=v[1]*(P_ofwh-P_condenser)
h[2]=h[1]+w_pI_in

"pump II"
h[3]=enthalpy(Fluid$, P=P_ofwh, x=0)
v[3]=volume(Fluid$, P=P_ofwh, x=0)
w_pII_in=v[3]*(P_cfwh2-P_ofwh)
h[4]=h[3]+w_pII_in

"pump III"
h[5]=enthalpy(Fluid$, P=P_cfwh2, x=0)
T[5]=temperature(Fluid$, P=P_cfwh2, x=0)
v[5]=volume(Fluid$, P=P_cfwh2, x=0)
w_pIII_in=v[5]*(P_cfwh1-P_cfwh2)
h[6]=h[5]+w_pIII_in

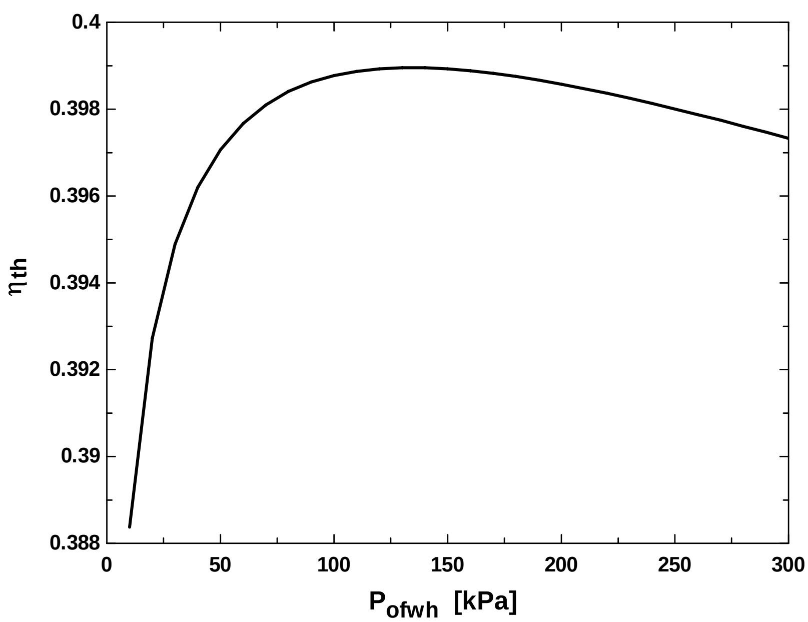
"pump IV"
h[9]=enthalpy(Fluid$, P=P_cfwh1, x=0)
T[9]=temperature(Fluid$, P=P_cfwh1, x=0)
v[9]=volume(Fluid$, P=P_cfwh1, x=0)
w_p4_in=v[5]*(P_boiler-P_cfwh1)
h[10]=h[9]+w_p4_in

"Mass balances"
x+y+z=1
```

```
m+n=z
"Open feedwater heater"
m*h[18]+n*h[2]=z*h[3]
"closed feedwater heater 2"
T[7]=T[5]
h[7]=enthalpy(Fluid$, P=P_cfwh1, T=T[7])
z*h[4]+y*h[15]=z*h[7]+y*h[5]
"closed feedwater heater 1"
T[11]=T[9]
h[11]=enthalpy(Fluid$, P=P_boiler, T=T[11])
(y+z)*h[8]+x*h[14]=(y+z)*h[11]+x*h[9]
"Mixing chamber after closed feedwater heater 2"
z*h[7]+y*h[6]=(y+z)*h[8]
"Mixing chamber after closed feedwater heater 1"
x*h[10]+(y+z)*h[11]=1*h[12]

"cycle"
w_T_out_high=x*(h[13]-h[14])+y*(h[13]-h[15])+z*(h[13]-h[16])
w_T_out_low=m*(h[17]-h[18])+n*(h[17]-h[19])
q_in=h[13]-h[12]+z*(h[17]-h[16])
q_out=n*(h[19]-h[1])
Eta_th=1-q_out/q_in
```

$P_{open\ fwh}$ [kPa]	η_{th}
10	0.388371
20	0.392729
30	0.394888
40	0.396199
50	0.397068
60	0.397671
70	0.398099
80	0.398406
90	0.398624
100	0.398774
110	0.398872
120	0.398930
130	**0.398954**
140	0.398952
150	0.398927
160	0.398883
170	0.398825
180	0.398752
190	0.398669
200	0.398576

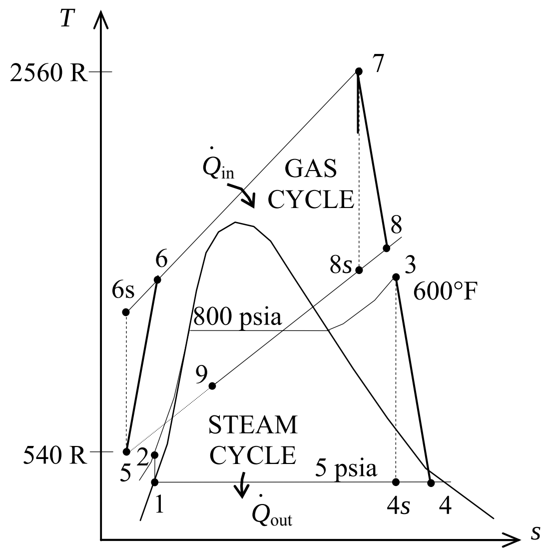


10-109E A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The thermal efficiency of the cycle is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable for Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis Working around the topping cycle gives the following results:



$$T_{6s} = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (540\text{ R})(10)^{0.4/1.4} = 1043\text{ R}$$

$$\eta_C = \frac{h_{6s}-h_5}{h_6-h_5} = \frac{c_p(T_{6s}-T_5)}{c_p(T_6-T_5)}$$

$$\longrightarrow T_6 = T_5 + \frac{T_{6s}-T_5}{\eta_C} = 540 + \frac{1043-540}{0.90} = 1099\text{ R}$$

$$T_{8s} = T_7\left(\frac{P_8}{P_7}\right)^{(k-1)/k} = (2560\text{ R})\left(\frac{1}{10}\right)^{0.4/1.4} = 1326\text{ R}$$

$$\eta_T = \frac{h_7-h_8}{h_7-h_{8s}} = \frac{c_p(T_7-T_8)}{c_p(T_7-T_{8s})} \longrightarrow T_8 = T_7 - \eta_T(T_7-T_{8s}) = 2560-(0.90)(2560-1326) = 1449\text{ R}$$

$$T_9 = T_{\text{sat @ 800 psia}} + 50 = 978.3\text{ R} + 50 = 1028\text{ R}$$

Fixing the states around the bottom steam cycle yields (Tables A-4E, A-5E, A-6E):

$$h_1 = h_{f\text{@ 5 psia}} = 130.18\text{ Btu/lbm}$$
$$v_1 = v_{f\text{@ 5 psia}} = 0.01641\text{ ft}^3\text{/lbm}$$

$$w_{\text{p,in}} = v_1(P_2-P_1) = (0.01641\text{ ft}^3\text{/lbm})(800-5)\text{psia}\left(\frac{1\text{ Btu}}{5.404\text{ psia}\cdot\text{ft}^3}\right) = 2.41\text{ Btu/lbm}$$

$$h_2 = h_1 + w_{\text{p,in}} = 130.18 + 2.41 = 132.59\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 800\text{ psia} \\ T_3 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1270.9\text{ Btu/lbm} \\ s_3 = 1.4866\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 5\text{ psia} \\ s_4 = s_3 \end{array}\right\} h_{4s} = 908.6\text{ Btu/lbm}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3-h_{4s}) = 1270.9-(0.95)(1270.9-908.6) = 926.7\text{ Btu/lbm}$$

The net work outputs from each cycle are

$$\begin{aligned} w_{\text{net, gas cycle}} &= w_{\text{T,out}} - w_{\text{C,in}} \\ &= c_p(T_7 - T_8) - c_p(T_6 - T_5) \\ &= (0.240\ \text{Btu/lbm}\cdot\text{R})(2560 - 1449 - 1099 + 540)\text{R} \\ &= 132.5\ \text{Btu/lbm} \end{aligned}$$

$$\begin{aligned} w_{\text{net, steam cycle}} &= w_{\text{T,out}} - w_{\text{P,in}} \\ &= (h_3 - h_4) - w_{\text{P,in}} \\ &= (1270.9 - 926.7) - 2.41 \\ &= 341.8\ \text{Btu/lbm} \end{aligned}$$

An energy balance on the heat exchanger gives

$$\dot{m}_a c_p(T_8 - T_9) = \dot{m}_w(h_3\text{-}h_2) \longrightarrow \dot{m}_w = \frac{c_p(T_8 - T_9)}{h_3\text{-}h_2}\dot{m}_a = \frac{(0.240)(1449 - 1028)}{1270.9 - 132.59} = 0.08876\dot{m}_a$$

That is, 1 lbm of exhaust gases can heat only 0.08876 lbm of water. Then the heat input, the heat output and the thermal efficiency are

$$q_{\text{in}} = \frac{\dot{m}_a}{\dot{m}_a} c_p(T_7 - T_6) = (0.240\ \text{Btu/lbm}\cdot\text{R})(2560 - 1099)\text{R} = 350.6\ \text{Btu/lbm}$$

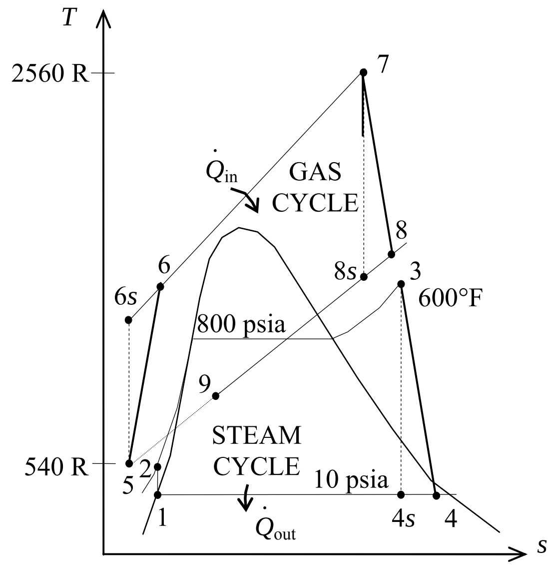
$$\begin{aligned} q_{\text{out}} &= \frac{\dot{m}_a}{\dot{m}_a} c_p(T_9 - T_1) + \frac{\dot{m}_w}{\dot{m}_a}(h_4 - h_1) \\ &= 1 \times (0.240\ \text{Btu/lbm}\cdot\text{R})(1028 - 540)\text{R} + 0.08876 \times (926.7 - 130.18)\ \text{Btu/lbm} \\ &= 187.8\ \text{Btu/lbm} \end{aligned}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{187.8}{350.6} = \mathbf{0.4643}$$

10-110E A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The thermal efficiency of the cycle is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable fo Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are $c_p = 0.240$ Btu/lbm·R and $k = 1.4$ (Table A-2Ea).



Analysis Working around the topping cycle gives the following results:

$$T_{6s} = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (540\text{ R})(10)^{0.4/1.4} = 1043\text{ R}$$

$$\eta_C = \frac{h_{6s} - h_5}{h_6 - h_5} = \frac{c_p(T_{6s} - T_5)}{c_p(T_6 - T_5)}$$

$$\longrightarrow T_6 = T_5 + \frac{T_{6s} - T_5}{\eta_C} = 540 + \frac{1043 - 540}{0.90} = 1099\text{ R}$$

$$T_{8s} = T_7\left(\frac{P_8}{P_7}\right)^{(k-1)/k} = (2560\text{ R})\left(\frac{1}{10}\right)^{0.4/1.4} = 1326\text{ R}$$

$$\eta_T = \frac{h_7 - h_8}{h_7 - h_{8s}} = \frac{c_p(T_7 - T_8)}{c_p(T_7 - T_{8s})} \longrightarrow T_8 = T_7 - \eta_T(T_7 - T_{8s}) = 2560 - (0.90)(2560 - 1326) = 1449\text{ R}$$

$$T_9 = T_{\text{sat @ 800 psia}} + 50 = 978.3\text{ R} + 50 = 1028\text{ R}$$

Fixing the states around the bottom steam cycle yields (Tables A-4E, A-5E, A-6E):

$$h_1 = h_{f\text{@ 10 psia}} = 161.25\text{ Btu/lbm}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\text{@ 10 psia}} = 0.01659\text{ ft}^3\text{/lbm}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01659\text{ ft}^3\text{/lbm})(800 - 10)\text{psia}\left(\frac{1\text{ Btu}}{5.404\text{ psia}\cdot\text{ft}^3}\right) \\ &= 2.43\text{ Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 161.25 + 2.43 = 163.7\text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 800\text{ psia} \\ T_3 = 600°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1270.9\text{ Btu/lbm} \\ s_3 = 1.4866\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ psia} \\ s_4 = s_3 \end{array}\right\} h_{4s} = 946.6\text{ Btu/lbm}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 1270.9 - (0.95)(1270.9 - 946.6)$$
$$= 962.8 \text{ Btu/lbm}$$

The net work outputs from each cycle are

$$\begin{aligned} w_{\text{net, gas cycle}} &= w_{\text{T,out}} - w_{\text{C,in}} \\ &= c_p (T_7 - T_8) - c_p (T_6 - T_5) \\ &= (0.240 \text{ Btu/lbm}\cdot\text{R})(2560 - 1449 - 1099 + 540)\text{R} \\ &= 132.5 \text{ Btu/lbm} \end{aligned}$$

$$\begin{aligned} w_{\text{net, steam cycle}} &= w_{\text{T,out}} - w_{\text{P,in}} \\ &= (h_3 - h_4) - w_{\text{P,in}} \\ &= (1270.9 - 962.8) - 2.43 \\ &= 305.7 \text{ Btu/lbm} \end{aligned}$$

An energy balance on the heat exchanger gives

$$\dot{m}_a c_p (T_8 - T_9) = \dot{m}_w (h_3 - h_2) \longrightarrow \dot{m}_w = \frac{c_p (T_8 - T_9)}{h_3 - h_2} \dot{m}_a = \frac{(0.240)(1449 - 1028)}{1270.9 - 163.7} = 0.09126 \dot{m}_a$$

That is, 1 lbm of exhaust gases can heat only 0.09126 lbm of water. Then the heat input, the heat output and the thermal efficiency are

$$q_{\text{in}} = \frac{\dot{m}_a}{\dot{m}_a} c_p (T_7 - T_6) = (0.240 \text{ Btu/lbm}\cdot\text{R})(2560 - 1099)\text{R} = 350.6 \text{ Btu/lbm}$$

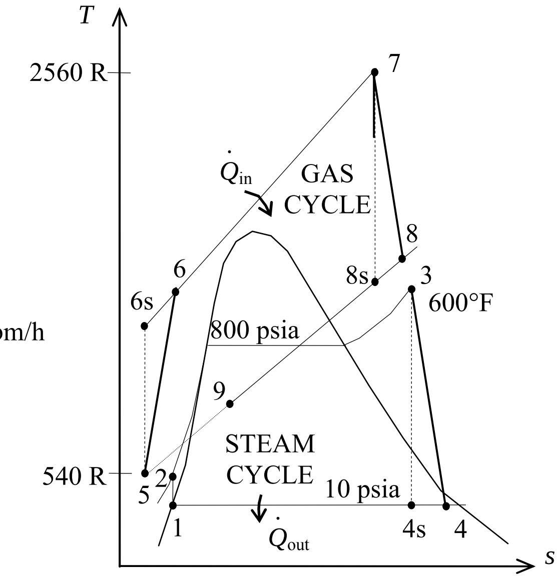
$$\begin{aligned} q_{\text{out}} &= \frac{\dot{m}_a}{\dot{m}_a} c_p (T_9 - T_1) + \frac{\dot{m}_w}{\dot{m}_a} (h_4 - h_1) \\ &= 1 \times (0.240 \text{ Btu/lbm}\cdot\text{R})(1028 - 540)\text{R} + 0.09126 \times (962.8 - 161.25) \text{ Btu/lbm} \\ &= 190.3 \text{ Btu/lbm} \end{aligned}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{190.3}{350.6} = \mathbf{0.4573}$$

When the condenser pressure is increased from 5 psia to 10 psia, the thermal efficiency is decreased from 0.4643 to 0.4573.

10-111E A combined gas-steam power cycle uses a simple gas turbine for the topping cycle and simple Rankine cycle for the bottoming cycle. The cycle supplies a specified rate of heat to the buildings during winter. The mass flow rate of air and the net power output from the cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable to Brayton cycle. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.



Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis The mass flow rate of water is

$$\dot{m}_w = \frac{\dot{Q}_{\text{buildings}}}{h_4 - h_1} = \frac{2\times10^6 \text{ Btu/h}}{(962.8\text{-}161.25)\text{ Btu/lbm}} = 2495\text{ lbm/h}$$

The mass flow rate of air is then

$$\dot{m}_w = \frac{\dot{m}_a}{0.09126} = \frac{2495}{0.09126} = \mathbf{27{,}340\text{ lbm/h}}$$

The power outputs from each cycle are

$$\begin{aligned}\dot{W}_{\text{net, gas cycle}} &= \dot{m}_a(w_{\text{T,out}} - w_{\text{C,in}}) \\ &= c_p(T_7 - T_8) - c_p(T_6 - T_5) \\ &= (27{,}340\text{ lbm/h})(0.240\text{ Btu/lbm}\cdot\text{R})(2560 - 1449 - 1099 + 540)\text{R}\left(\frac{1\text{ kW}}{3412.14\text{ Btu/h}}\right) \\ &= 1062\text{ kW}\end{aligned}$$

$$\begin{aligned}\dot{W}_{\text{net, steam cycle}} &= \dot{m}_a(w_{\text{T,out}} - w_{\text{P,in}}) \\ &= \dot{m}_a(h_3 - h_4 - w_{\text{P,in}}) \\ &= (2495\text{ lbm/h})(1270.9 - 962.8 - 2.43)\left(\frac{1\text{ kW}}{3412.14\text{ Btu/h}}\right) \\ &= 224\text{ kW}\end{aligned}$$

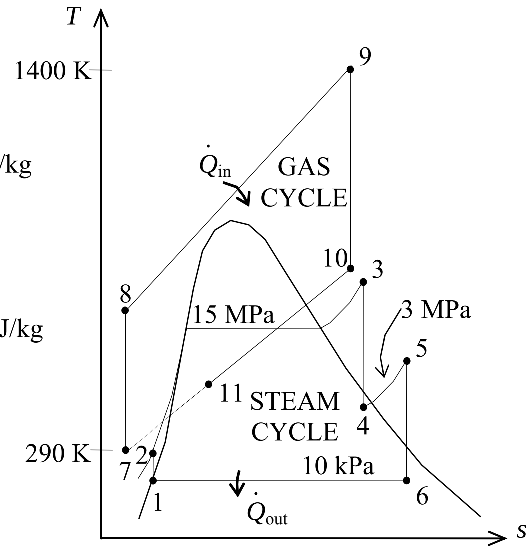
The net electricity production by this cycle is then

$$\dot{W}_{\text{net}} = 1062 + 224 = \mathbf{1286\text{ kW}}$$

10-112 A combined gas-steam power plant is considered. The topping cycle is an ideal gas-turbine cycle and the bottoming cycle is an ideal reheat Rankine cycle. The mass flow rate of air in the gas-turbine cycle, the rate of total heat input, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) The analysis of gas cycle yields



$$T_7 = 290 \text{ K} \longrightarrow h_7 = 290.16 \text{ kJ/kg}$$
$$P_{r_7} = 1.2311$$

$$P_{r_8} = \frac{P_8}{P_7} P_{r_7} = (8)(1.2311) = 9.849 \longrightarrow h_8 = 526.12 \text{ kJ/kg}$$

$$T_9 = 1400 \text{ K} \longrightarrow h_9 = 1515.42 \text{ kJ/kg}$$
$$P_{r_9} = 450.5$$

$$P_{r_{10}} = \frac{P_{10}}{P_9} P_{r_9} = \left(\frac{1}{8}\right)(450.5) = 56.3 \longrightarrow h_{10} = 860.35 \text{ kJ/kg}$$

$$T_{11} = 520 \text{ K} \longrightarrow h_{11} = 523.63 \text{ kJ/kg}$$

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

$$h_1 = h_{f\ @\ 10\text{ kPa}} = 191.81 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 10\text{ kPa}} = 0.00101 \text{ m}^3\text{/kg}$$

$$w_{\text{pI,in}} = \boldsymbol{v}_1(P_2 - P_1) = (0.00101 \text{ m}^3\text{/kg})(15{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = 15.14 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 15.14 = 206.95 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 15 \text{ MPa} \\ T_3 &= 450°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3157.9 \text{ kJ/kg} \\ s_3 &= 6.1434 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 3 \text{ MPa} \\ s_4 &= s_3 \end{aligned}\right\} \begin{aligned} x_4 &= \frac{s_4 - s_f}{s_{fg}} = \frac{6.1434 - 2.6454}{3.5402} = 0.9880 \\ h_4 &= h_f + x_4 h_{fg} = 1008.3 + (0.9880)(1794.9) = 2781.7 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_5 &= 3 \text{ MPa} \\ T_5 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_5 &= 3457.2 \text{ kJ/kg} \\ s_5 &= 7.2359 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 10 \text{ kPa} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} x_6 &= \frac{s_6 - s_f}{s_{fg}} = \frac{7.2355 - 0.6492}{7.4996} = 0.8783 \\ h_6 &= h_f + x_6 h_{fg} = 191.81 + (0.8783)(2392.1) = 2292.8 \text{ kJ/kg} \end{aligned}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow \sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s(h_3 - h_2) = \dot{m}_{\text{air}}(h_{10} - h_{11})$$

$$\dot{m}_{\text{air}} = \frac{h_3 - h_2}{h_{10} - h_{11}} \dot{m}_s = \frac{3157.9 - 206.95}{860.35 - 523.63}(30 \text{ kg/s}) = \mathbf{262.9\ kg/s}$$

(*b*) $\dot{Q}_{\text{in}} = \dot{Q}_{\text{air}} + \dot{Q}_{\text{reheat}} = \dot{m}_{\text{air}}(h_9 - h_8) + \dot{m}_{\text{reheat}}(h_5 - h_4)$

$$= (262.9\ \text{kg/s})(1515.42 - 526.12)\ \text{kJ/kg} + (30\ \text{kg/s})(3457.2 - 2781.7)\text{kJ/kg}$$
$$= 280{,}352\ \text{kW}$$
$$\cong \mathbf{2.80 \times 10^5\ \ kW}$$

(*c*) $\dot{Q}_{\text{out}} = \dot{Q}_{\text{out,air}} + \dot{Q}_{\text{out,steam}} = \dot{m}_{\text{air}}(h_{11} - h_7) + \dot{m}_s(h_6 - h_1)$

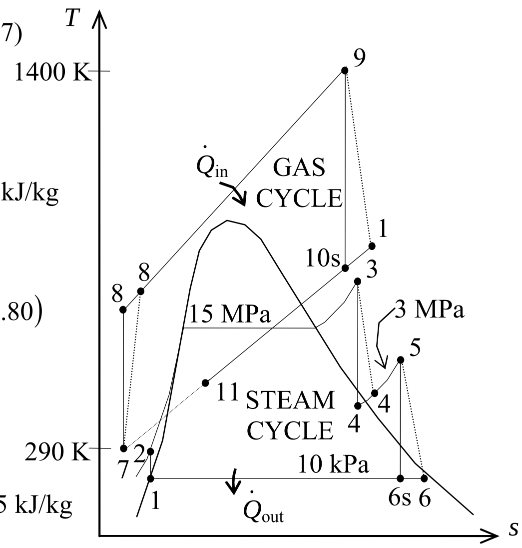
$$= (262.9\ \text{kg/s})(523.63 - 290.16)\ \text{kJ/kg} + (30\ \text{kg/s})(2292.8 - 191.81)\ \text{kJ/kg}$$
$$= 124{,}409\ \text{kW}$$

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{124{,}409\ \text{kW}}{280{,}352\ \text{kW}} = \mathbf{55.6\%}$$

10-113 A combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal reheat Rankine cycle. The mass flow rate of air in the gas-turbine cycle, the rate of total heat input, and the thermal efficiency of the combined cycle are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Analysis (*a*) The analysis of gas cycle yields (Table A-17)



$$T_7 = 290 \text{ K} \longrightarrow h_7 = 290.16 \text{ kJ/kg}$$
$$P_{r_7} = 1.2311$$

$$P_{r_{8s}} = \frac{P_{8s}}{P_7} P_{r_7} = (8)(1.2311) = 9.849 \longrightarrow h_{8s} = 526.12 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{8s} - h_7}{h_8 - h_7} \longrightarrow h_8 = h_7 + (h_{8s} - h_7)/\eta_C$$
$$= 290.16 + (526.12 - 290.16)/(0.80)$$
$$= 585.1 \text{ kJ/kg}$$

$$T_9 = 1400 \text{ K} \longrightarrow h_9 = 1515.42 \text{ kJ/kg}$$
$$P_{r_9} = 450.5$$

$$P_{r_{10s}} = \frac{P_{10s}}{P_9} P_{r_9} = \left(\frac{1}{8}\right)(450.5) = 56.3 \longrightarrow h_{10s} = 860.35 \text{ kJ/kg}$$

$$\eta_T = \frac{h_9 - h_{10}}{h_9 - h_{10s}} \longrightarrow h_{10} = h_9 - \eta_T (h_9 - h_{10s})$$
$$= 1515.42 - (0.85)(1515.42 - 860.35)$$
$$= 958.4 \text{ kJ/kg}$$

$$T_{11} = 520 \text{ K} \longrightarrow h_{11} = 523.63 \text{ kJ/kg}$$

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

$$h_1 = h_{f\ @\ 10\text{ kPa}} = 191.81 \text{ kJ/kg}$$
$$v_1 = v_{f\ @\ 10\text{ kPa}} = 0.00101 \text{ m}^3/\text{kg}$$

$$w_{\text{pI,in}} = v_1 (P_2 - P_1)$$
$$= (0.00101 \text{ m}^3/\text{kg})(15{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$
$$= 15.14 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 15.14 = 206.95 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 15 \text{ MPa} \\ T_3 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3157.9 \text{ kJ/kg} \\ s_3 = 6.1428 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 3 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{s_{4s} - s_f}{s_{fg}} = \dfrac{6.1434 - 2.6454}{3.5402} = 0.9880 \\ h_{4s} = h_f + x_{4s} h_{fg} = 1008.3 + (0.9879)(1794.9) = 2781.7 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 3157.9 - (0.85)(3157.9 - 2781.7)$$
$$= 2838.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l}P_5 = 3\text{ MPa}\\ T_5 = 500°\text{C}\end{array}\right\}\begin{array}{l}h_5 = 3457.2\text{ kJ/kg}\\ s_5 = 7.2359\text{ kJ/kg}\cdot\text{K}\end{array}$$

$$\left.\begin{array}{l}P_6 = 10\text{ kPa}\\ s_{6s} = s_5\end{array}\right\}\begin{array}{l}x_{6s} = \dfrac{s_{6s}-s_f}{s_{fg}} = \dfrac{7.2359-0.6492}{7.4996} = 0.8783\\ h_{6s} = h_f + x_{6s}h_{fg} = 191.81 + (0.8782)(2392.1) = 2292.8\text{ kJ/kg}\end{array}$$

$$\eta_T = \frac{h_5-h_6}{h_5-h_{6s}} \longrightarrow h_6 = h_5 - \eta_T(h_5-h_{6s})$$
$$= 3457.2-(0.85)(3457.2-2292.8)$$
$$= 2467.5\text{ kJ/kg}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s(h_3-h_2) = \dot{m}_{\text{air}}(h_{10}-h_{11})$$
$$\dot{m}_{\text{air}} = \frac{h_3-h_2}{h_{10}-h_{11}}\dot{m}_s = \frac{3157.9-206.95}{958.4-523.63}(30\text{ kg/s}) = \mathbf{203.6\text{ kg/s}}$$

(*b*) $\dot{Q}_{\text{in}} = \dot{Q}_{\text{air}} + \dot{Q}_{\text{reheat}} = \dot{m}_{\text{air}}(h_9-h_8) + \dot{m}_{\text{reheat}}(h_5-h_4)$
$= (203.6\text{ kg/s})(1515.42-585.1)\text{ kJ/kg} + (30\text{ kg/s})(3457.2-2838.1)\text{ kJ/kg}$
$=$ **207,986 kW**

(*c*) $\dot{Q}_{\text{out}} = \dot{Q}_{\text{out,air}} + \dot{Q}_{\text{out,steam}} = \dot{m}_{\text{air}}(h_{11}-h_7) + \dot{m}_s(h_6-h_1)$
$= (203.6\text{ kg/s})(523.63-290.16)\text{ kJ/kg} + (30\text{ kg/s})(2467.5-191.81)\text{ kJ/kg}$
$= 115{,}805\text{ kW}$

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{115{,}805\text{ kW}}{207{,}986\text{ kW}} = \mathbf{44.3\%}$$

10-114 It is to be shown that the exergy destruction associated with a simple ideal Rankine cycle can be expressed as $x_{\text{destroyed}} = q_{in}(\eta_{\text{th,Carnot}} - \eta_{th})$, where η_{th} is efficiency of the Rankine cycle and $\eta_{\text{th, Carnot}}$ is the efficiency of the Carnot cycle operating between the same temperature limits.

Analysis The exergy destruction associated with a cycle is given on a unit mass basis as

$$x_{\text{destroyed}} = T_0 \sum \frac{q_R}{T_R}$$

where the direction of q_{in} is determined with respect to the reservoir (positive if to the reservoir and negative if from the reservoir). For a cycle that involves heat transfer only with a source at T_{H} and a sink at T_0, the irreversibility becomes

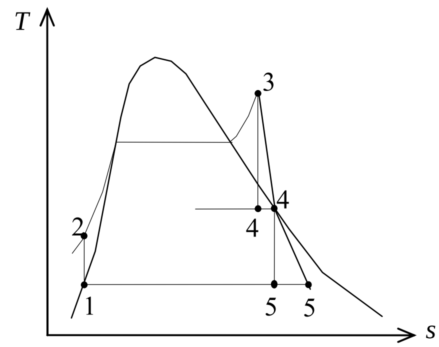
$$\begin{aligned} x_{\text{destroyed}} &= T_0\left(\frac{q_{\text{out}}}{T_0} - \frac{q_{\text{in}}}{T_H}\right) = q_{\text{out}} - \frac{T_0}{T_H} q_{\text{in}} = q_{\text{in}}\left(\frac{q_{\text{out}}}{q_{\text{in}}} - \frac{T_0}{T_H}\right) \\ &= q_{\text{in}}\left[(1-\eta_{th}) - (1-\eta_{th,C})\right] = q_{\text{in}}(\eta_{th,C} - \eta_{th}) \end{aligned}$$

10-115 A cogeneration plant is to produce power and process heat. There are two turbines in the cycle: a high-pressure turbine and a low-pressure turbine. The temperature, pressure, and mass flow rate of steam at the inlet of high-pressure turbine are to be determined.

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

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

$$\left.\begin{array}{l} P_4 = 1.4 \text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_4 = h_{g\,@\,1.4\text{ MPa}} = 2788.9 \text{ kJ/kg} \\ s_4 = s_{g\,@\,1.4\text{ MPa}} = 6.4675 \text{ kJ/kg}\cdot\text{K} \end{array}$$



$$\left.\begin{array}{l} P_5 = 10 \text{ kPa} \\ s_{5s} = s_4 \end{array}\right\} \begin{array}{l} x_{5s} = \dfrac{s_{4s} - s_f}{s_{fg}} = \dfrac{6.4675 - 0.6492}{7.4996} = 0.7758 \\ h_{5s} = h_f + x_{5s} h_{fg} \\ = 191.81 + (0.7758)(2392.1) = 2047.6 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow \begin{array}{l} h_5 = h_4 - \eta_T (h_4 - h_{5s}) \\ = 2788.9 - (0.60)(2788.9 - 2047.6) \\ = 2344.1 \text{ kJ/kg} \end{array}$$

and

$$w_{\text{turb,low}} = h_4 - h_5 = 2788.9 - 2344.1 = 444.8 \text{ kJ/kg}$$

$$\dot{m}_{\text{low turb}} = \frac{\dot{W}_{\text{turb,II}}}{w_{\text{turb,low}}} = \frac{800 \text{ kJ/s}}{444.8 \text{ kJ/kg}} = 1.799 \text{ kg/s} = 107.9 \text{ kg/min}$$

Therefore ,

$$\dot{m}_{\text{total}} = 1000 + 108 = 1108 \text{ kg/min} = \mathbf{18.47\ kg/s}$$

$$w_{\text{turb,high}} = \frac{\dot{W}_{\text{turb},I}}{\dot{m}_{\text{high,turb}}} = \frac{1000 \text{ kJ/s}}{18.47 \text{ kg/s}} = 54.15 \text{ kJ/kg} = h_3 - h_4$$

$$h_3 = w_{\text{turb,high}} + h_4 = 54.15 + 2788.9 = 2843.0 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow \begin{array}{l} h_{4s} = h_3 - (h_3 - h_4)/\eta_T \\ = 2843.0 - (2843.0 - 2788.9)/(0.75) \\ = 2770.8 \text{ kJ/kg} \end{array}$$

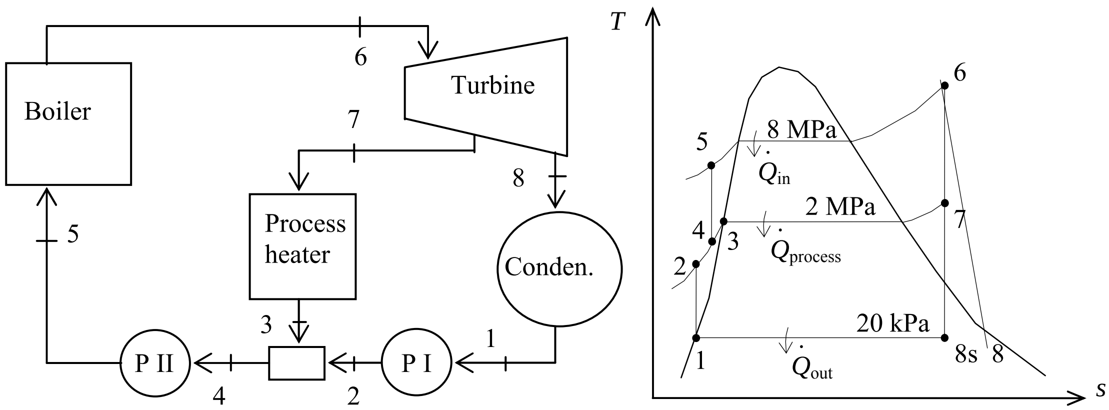
$$\left.\begin{array}{l} P_{4s} = 1.4 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{h_{4s} - h_f}{h_{fg}} = \dfrac{2770.8 - 829.96}{1958.9} = 0.9908 \\ s_{4s} = s_f + x_{4s} s_{fg} = 2.2835 + (0.9908)(4.1840) = 6.4289 \text{ kJ/kg}\cdot\text{K} \end{array}$$

Then from the tables or the software, the turbine inlet temperature and pressure becomes

$$\left.\begin{array}{l} h_3 = 2843.0 \text{ kJ/kg} \\ s_3 = 6.4289 \text{ kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} P_3 = \mathbf{2\ MPa} \\ T_3 = \mathbf{227.5°C} \end{array}$$

10-116 A cogeneration plant is to generate power and process heat. Part of the steam extracted from the turbine at a relatively high pressure is used for process heating. The rate of process heat, the net power produced, and the utilization factor of the plant are to be determined.

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



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

$$h_1 = h_{f\ @\ 20\ \text{kPa}} = 251.42\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 20\ \text{kPa}} = 0.001017\ \ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= \left(0.001017\ \ \text{m}^3/\text{kg}\right)\left(2000 - 20\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/0.88 \\ &= 2.29\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 2.29 = 253.71\ \text{kJ/kg}$$

$$h_3 = h_{f\ @\ 2\ \text{MPa}} = 908.47\ \text{kJ/kg}$$

Mixing chamber:

$$\dot{m}_3 h_3 + \dot{m}_2 h_2 = \dot{m}_4 h_4$$
$$(4\ \text{kg/s})(908.47\ \text{kJ/kg}) + (11 - 4\ \text{kg/s})(253.71\ \text{kJ/kg})) = (11\ \text{kg/s})h_4 \longrightarrow h_4 = 491.81\ \text{kJ/kg}$$

$$\boldsymbol{v}_4 \cong \boldsymbol{v}_{f\ @\ h_f = 491.81\ \text{kJ/kg}} = 0.001058\ \ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{p\text{II,in}} &= \boldsymbol{v}_4(P_5 - P_4)/\eta_p \\ &= \left(0.001058\ \ \text{m}^3/\text{kg}\right)\left(8000 - 2000\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/0.88 \\ &= 7.21\ \text{kJ/kg} \end{aligned}$$
$$h_5 = h_4 + w_{p\text{II,in}} = 491.81 + 7.21 = 499.02\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 8\ \text{MPa} \\ T_6 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3399.5\ \text{kJ/kg} \\ s_6 &= 6.7266\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 2\ \text{MPa} \\ s_7 &= s_6 \end{aligned}\right\} h_{7s} = 3000.4\ \text{kJ/kg}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} \longrightarrow h_7 = h_6 - \eta_T (h_6 - h_{7s}) = 3399.5 - (0.88)(3399.5 - 3000.4) = 3048.3 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_8 = 20 \text{ kPa} \\ s_8 = s_6 \end{array} \right\} h_{8s} = 2215.5 \text{ kJ/kg}$$

$$\eta_T = \frac{h_6 - h_8}{h_6 - h_{8s}} \longrightarrow h_8 = h_6 - \eta_T (h_6 - h_{8s}) = 3399.5 - (0.88)(3399.5 - 2215.5) = 2357.6 \text{ kJ/kg}$$

Then,

$$\dot{Q}_{\text{process}} = \dot{m}_7 (h_7 - h_3) = (4 \text{ kg/s})(3048.3 - 908.47) \text{ kJ/kg} = \mathbf{8559\ kW}$$

(*b*) Cycle analysis:

$$\begin{aligned} \dot{W}_{\text{T,out}} &= \dot{m}_7 (h_6 - h_7) + \dot{m}_8 (h_6 - h_8) \\ &= (4 \text{ kg/s})(3399.5 - 3048.3)\text{kJ/kg} + (7 \text{ kg/s})(3399.5 - 2357.6)\text{kJ/kg} \\ &= 8698 \text{ kW} \end{aligned}$$

$$\dot{W}_{\text{p,in}} = \dot{m}_1 w_{\text{pI,in}} + \dot{m}_4 w_{\text{pII,in}} = (7 \text{ kg/s})(2.29 \text{ kJ/kg}) + (11 \text{ kg/s})(7.21 \text{ kJ/kg}) = 95 \text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{p,in}} = 8698 - 95 = \mathbf{8603\ kW}$$

(*c*) Then,

$$\dot{Q}_{\text{in}} = \dot{m}_5 (h_6 - h_5) = (11 \text{ kg/s})(3399.5 - 499.02) = 31{,}905 \text{ kW}$$

and

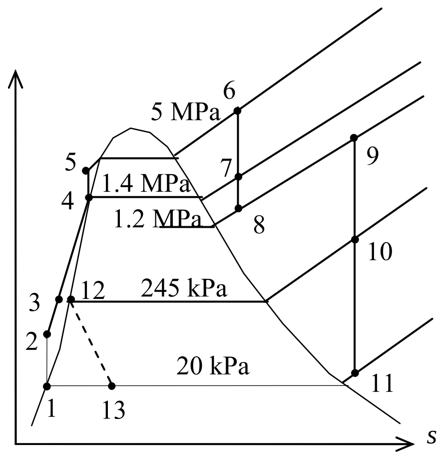
$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{8603 + 8559}{31{,}905} = 0.538 = \mathbf{53.8\%}$$

10-117 A Rankine steam cycle modified for reheat, a closed feedwater heater, and an open feedwater heater is considered. The *T-s* diagram for the ideal cycle is to be sketched. The net power output of the cycle and the minimum flow rate of the cooling water required are to be determined.

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

Analysis (*b*) Using the data from the problem statement, the enthalpies at various states are

$$h_1 = h_{f\ @\ 20\ \text{kPa}} = 251.4\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 20\ \text{kPa}} = 0.00102\ \text{m}^3/\text{kg}$$
$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00102\ \text{m}^3/\text{kg})(1400 - 20\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 1.41\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 251.4 + 1.41 = 252.8\ \text{kJ/kg}$$
$$h_4 = h_{f\ @1400\ \text{kPa}} = 830\ \text{kJ/kg}$$
$$\boldsymbol{v}_4 = \boldsymbol{v}_{f\ @\ 1400\ \text{kPa}} = 0.00115\ \text{m}^3/\text{kg}$$
$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_1(P_5 - P_4) \\ &= (0.00115\ \text{m}^3/\text{kg})(5000 - 1400\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 4.14\ \text{kJ/kg} \end{aligned}$$
$$h_5 = h_4 + w_{\text{pII,in}} = 830 + 4.14 = 834.1\ \text{kJ/kg}$$



Also,

$$h_3 = h_{12} = h_{f\ @\ 245\ \text{kPa}} = 532\ \text{kJ/kg}$$
$$h_{13} = h_{12} \quad \text{(throttle valve operation)}$$

An energy balance on the open feedwater heater gives

$$yh_7 + (1-y)h_3 = 1(h_4)$$

where *y* is the fraction of steam extracted from the high-pressure turbine. Solving for *y*,

$$y = \frac{h_4 - h_3}{h_7 - h_3} = \frac{830 - 532}{3400 - 532} = 0.1039$$

An energy balance on the closed feedwater heater gives

$$zh_{10} + (1-y)h_2 = (1-y)h_3 + zh_{12}$$

where *z* is the fraction of steam extracted from the low-pressure turbine. Solving for *z*,

$$z = \frac{(1-y)(h_3 - h_2)}{h_{10} - h_{12}} = \frac{(1-0.1039)(532 - 252.8)}{3154 - 532} = 0.09542$$

The heat input in the boiler is

$$\begin{aligned} q_{\text{in}} &= (h_6 - h_5) + (1-y)(h_9 - h_8) \\ &= (3894 - 834.1) + (1 - 0.1039)(3692 - 3349) = 3367\ \text{kJ/kg} \end{aligned}$$

The work output from the turbines is

$$
\begin{aligned}
w_{\mathrm{T,out}} &= h_6 - yh_7 - (1-y)h_8 + (1-y)h_9 - zh_{10} - (1-y-z)h_{11} \\
&= 3894 - (0.1039)(3400) - (1-0.1039)(3349) \\
&\quad + (1-0.1039)(3692) - (0.09542)(3154) - (1-0.1039-0.09542)(2620) \\
&= 1449 \text{ kJ/kg}
\end{aligned}
$$

The net work output from the cycle is

$$
\begin{aligned}
w_{\mathrm{net}} &= w_{\mathrm{T,out}} - (1-y)w_{\mathrm{PI,in}} - w_{\mathrm{PII,in}} \\
&= 1449 - (1-0.1039)(1.41) - 4.14 \\
&= 1444 \text{ kJ/kg}
\end{aligned}
$$

The net power output is

$$
\begin{aligned}
\dot{W}_{\mathrm{net}} &= \dot{m}w_{\mathrm{net}} = (100 \text{ kg/s})(1444 \text{ kJ/kg}) \\
&= 144{,}400 \text{ kW} = \mathbf{144.4\ MW}
\end{aligned}
$$

(*c*) The heat rejected from the condenser is

$$
\begin{aligned}
q_{\mathrm{out}} &= (1-y-z)h_{11} + zh_{13} - (1-y)h_1 \\
&= (1-0.1039-0.09542)(2620) + (0.09542)(532) - (1-0.1039)(251.4) \\
&= 1923 \text{ kJ/kg}
\end{aligned}
$$

The mass flow rate of cooling water will be minimum when the cooling water exit temperature is a maximum. That is,

$$
T_{w,2} = T_1 = T_{sat\ @\ 20\ \mathrm{kPa}} = 60.1°\mathrm{C}
$$

Then an energy balance on the condenser gives

$$
\dot{m}q_{\mathrm{out}} = \dot{m}_w c_{\mathrm{p,w}}(T_{\mathrm{w,2}} - T_{\mathrm{w,1}})
$$

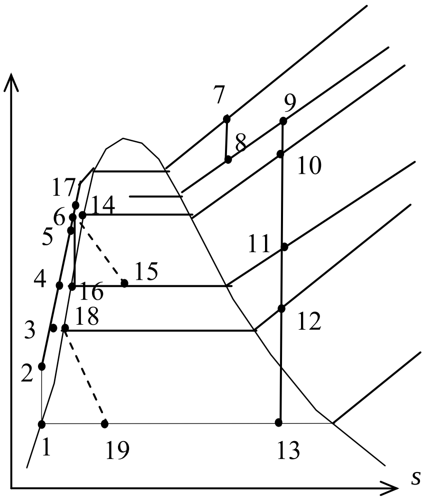
$$
\dot{m}_w = \frac{\dot{m}q_{\mathrm{out}}}{c_{\mathrm{p,w}}(T_{\mathrm{w,2}} - T_{\mathrm{w,1}})} = \frac{(100 \text{ kg/s})(1923 \text{ kJ/kg})}{(4.18 \text{ kJ/kg}\cdot\text{K})(60.1-25) \text{ K}} = \mathbf{1311\,kg/s}
$$

10-118 A Rankine steam cycle modified for reheat and three closed feedwater heaters is considered. The *T*-*s* diagram for the ideal cycle is to be sketched. The net power output of the cycle and the flow rate of the cooling water required are to be determined.

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

Analysis (*b*) Using the data from the problem statement, the enthalpies at various states are

$$h_1 = h_{f\ @\ 10\ \text{kPa}} = 191.8\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 10\ \text{kPa}} = 0.001010\ \ \text{m}^3/\text{kg}$$
$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001010\ \ \text{m}^3/\text{kg})(5000 - 10\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 5.04\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.8 + 5.04 = 196.8\ \text{kJ/kg}$$
$$h_{16} = h_{f\ @300\ \text{kPa}} = 561.4\ \text{kJ/kg}$$
$$\boldsymbol{v}_{16} = \boldsymbol{v}_{f\ @300\ \text{kPa}} = 0.001073\ \ \text{m}^3/\text{kg}$$
$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_{16}(P_{17} - P_{16}) \\ &= (0.001073\ \ \text{m}^3/\text{kg})(5000 - 300\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 5.04\ \text{kJ/kg} \end{aligned}$$
$$h_{17} = h_{16} + w_{\text{pII,in}} = 561.4 + 5.04 = 566.4\ \text{kJ/kg}$$



Also,

$$h_3 = h_{18} = h_{f\ @\ 75\ \text{kPa}} = 384.4\ \text{kJ/kg}$$
$$h_4 = h_{16} = h_{f\ @\ 300\ \text{kPa}} = 561.4\ \text{kJ/kg}$$
$$h_5 = h_{14} = h_{f\ @\ 925\ \text{kPa}} = 747.7\ \text{kJ/kg}$$
$$h_{15} = h_{14} \quad \text{(throttle valve operation)}$$
$$h_{19} = h_{18} \quad \text{(throttle valve operation)}$$

Energy balances on three closed feedwater heaters give

$$yh_{10} + (1-y-z)h_4 = (1-y-z)h_5 + yh_{15}$$
$$zh_{11} + (1-y-z)h_3 + yh_{15} = (1-y-z)h_4 + (y+z)h_{16}$$
$$wh_{12} + (1-y-z)h_2 = (1-y-z)h_3 + wh_{18}$$

The enthalpies are known, and thus there are three unknowns (y, z, w) and three equations. Solving these equations using EES, we obtain

$$y = 0.06335$$
$$z = 0.05863$$
$$w = 0.07063$$

The enthalpy at state 6 may be determined from an energy balance on mixing chamber:

$$\begin{aligned} h_6 &= (1-y-z)h_5 + (y+z)h_{17} \\ &= (1 - 0.06335 - 0.05863)(747.7) + (0.06335 + 0.05863)(566.4) = 725.6\ \text{kJ/kg} \end{aligned}$$

The heat input in the boiler is

$$\begin{aligned} q_{\text{in}} &= (h_7 - h_6) + (h_9 - h_8) \\ &= (3900 - 725.6) + (3687 - 3615) = 3246\ \text{kJ/kg} \end{aligned}$$

The work output from the turbines is

$$\begin{aligned} w_{\text{T,out}} &= h_7 - h_8 + h_9 - yh_{10} - zh_{11} - wh_{12} - (1-y-z-w)h_{13} \\ &= 3900 - 3615 + 3687 - (0.06335)(3330) - (0.05863)(3011) \\ &\quad -(0.07063)(2716) - (1 - 0.06335 - 0.05863 - 0.07063)(2408) \\ &= 1448\ \text{kJ/kg} \end{aligned}$$

The net work output from the cycle is

$$\begin{aligned} w_{\text{net}} &= w_{\text{T,out}} - (1-y-z)w_{\text{PI,in}} - (y+z)w_{\text{PII,in}} \\ &= 1448 - (1 - 0.06335 - 0.05863)(5.04) - (0.06335 + 0.05863)(5.04) \\ &= 1443\ \text{kJ/kg} \end{aligned}$$

The net power output is

$$\begin{aligned} \dot{W}_{\text{net}} &= \dot{m}w_{\text{net}} = (100\ \text{kg/s})(1443\ \text{kJ/kg}) \\ &= 144{,}300\ \text{kW} = \mathbf{144.3\ MW} \end{aligned}$$

(c) The heat rejected from the condenser is

$$\begin{aligned} q_{\text{out}} &= (1-y-z-w)h_{13} + wh_{19} - (1-y-z)h_1 \\ &= (1 - 0.06335 - 0.05863 - 0.07063)(2408) + (0.07063)(384.4) - (1 - 0.06335 - 0.05863)(191.8) \\ &= 1803\ \text{kJ/kg} \end{aligned}$$

Then an energy balance on the condenser gives

$$\dot{m}q_{\text{out}} = \dot{m}_w c_{\text{p,w}}(T_{\text{w,2}} - T_{\text{w,1}})$$

$$\dot{m}_w = \frac{\dot{m}q_{\text{out}}}{c_{\text{p,w}}(T_{\text{w,2}} - T_{\text{w,1}})} = \frac{(100\ \text{kg/s})(1803\ \text{kJ/kg})}{(4.18\ \text{kJ/kg}\cdot\text{K})(10\ \text{K})} = \mathbf{4313\ kg/s}$$

10-119 EES The effect of the condenser pressure on the performance a simple ideal Rankine cycle is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
        x4$=''
        if (x4>1) then x4$='(superheated)'
        if (x4<0) then x4$='(compressed)'
end

P[3] = 5000 [kPa]
T[3] = 500 [C]
"P[4] = 5 [kPa]"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"

"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[4]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

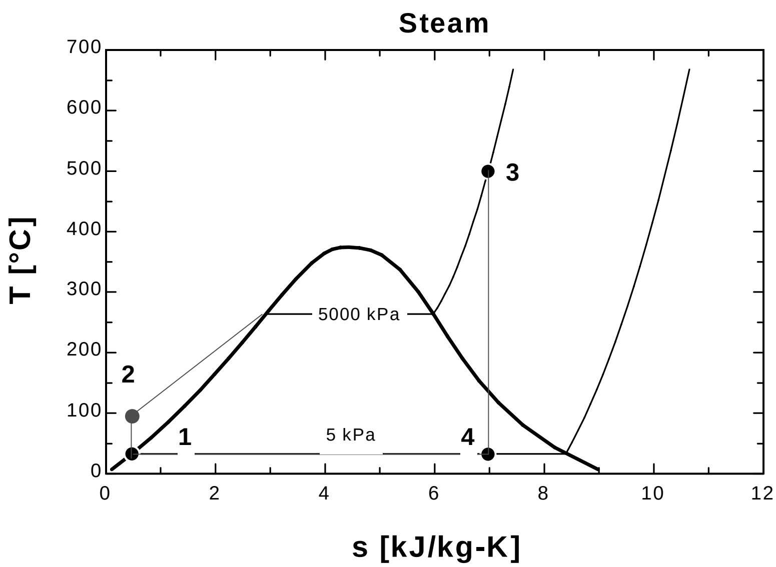
"Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
x[4]=quality(Fluid$,h=h[4],P=P[4])
h[3] =W_t+h[4]"SSSF First Law for the turbine"
x4s$=x4$(x[4])

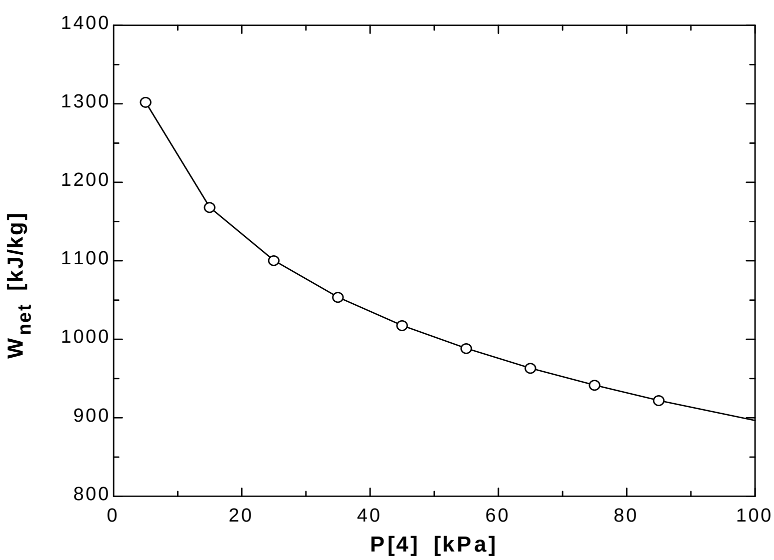
"Boiler analysis"
Q_in + h[2]=h[3]"SSSF First Law for the Boiler"

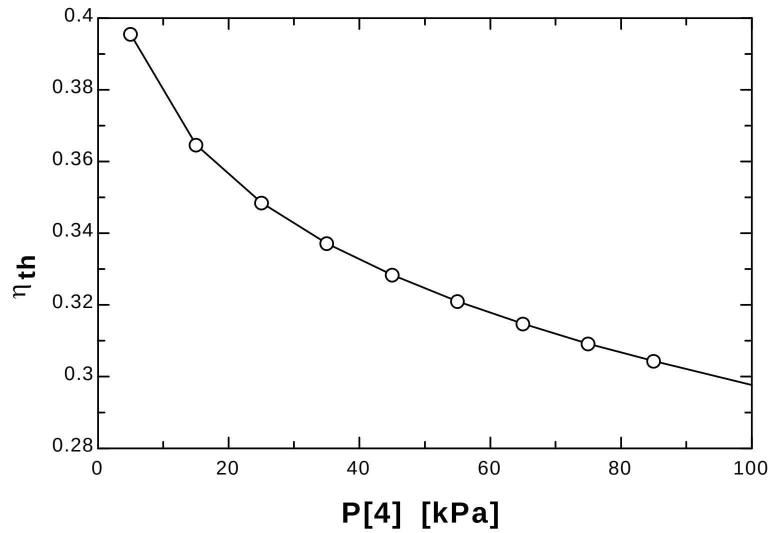
"Condenser analysis"
h[4]=Q_out+h[1]"SSSF First Law for the Condenser"

"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

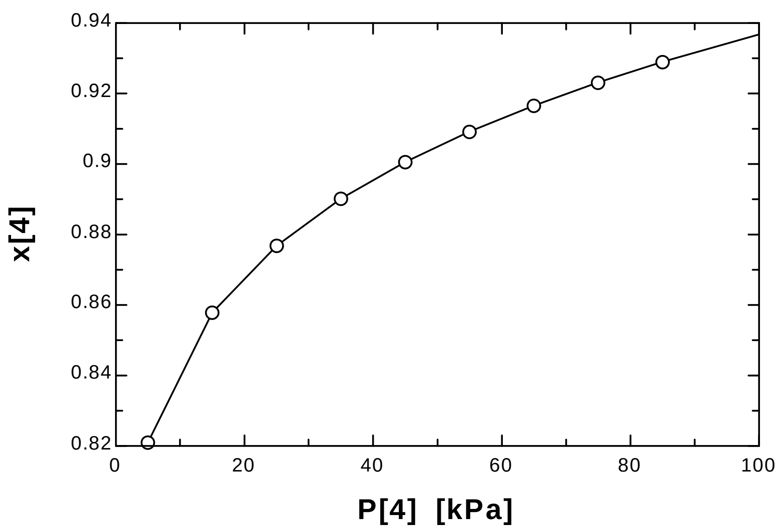
η_{th}	P_4 [kPa]	W_{net} [kJ/kg]	x_4	Q_{in} [kJ/kg]	Q_{out} [kJ/kg]
0.3956	5	1302	0.8212	3292	1990
0.3646	15	1168	0.8581	3204	2036
0.3484	25	1100	0.8772	3158	2057
0.3371	35	1054	0.8905	3125	2072
0.3283	45	1018	0.9009	3100	2082
0.321	55	988.3	0.9096	3079	2091
0.3147	65	963.2	0.917	3061	2098
0.3092	75	941.5	0.9235	3045	2104
0.3042	85	922.1	0.9293	3031	2109
0.2976	100	896.5	0.9371	3012	2116







0.4
0.38
0.36
0.34
0.32
0.3
0.28
η_{th}
0
20
40
60
80
100
P[4] [kPa]



0.94
0.92
0.9
0.88
0.86
0.84
0.82
x[4]
0
20
40
60
80
100
P[4] [kPa]

10-120 EES The effect of superheating the steam on the performance a simple ideal Rankine cycle is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
          x4$=''
          if (x4>1) then x4$='(superheated)'
          if (x4<0) then x4$='(compressed)'
end

P[3] = 3000 [kPa]
{T[3] = 600 [C]}
P[4] = 10 [kPa]
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"

"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[4]
P[2]=P[3]
x[1]=0 "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

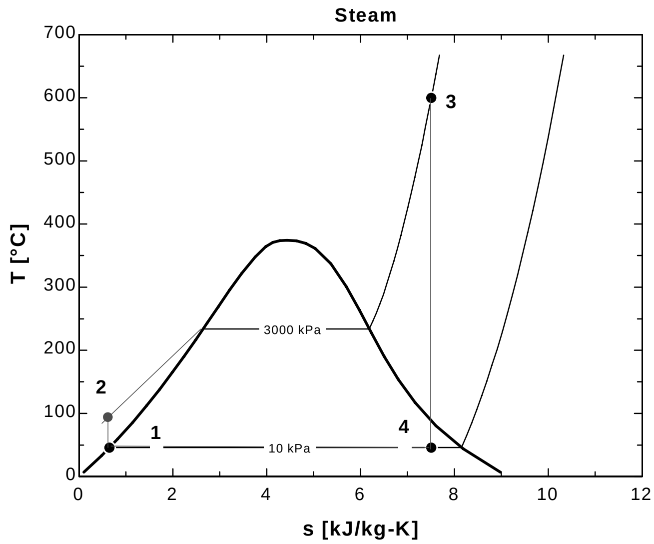
"Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
x[4]=quality(Fluid$,h=h[4],P=P[4])
h[3] =W_t+h[4]"SSSF First Law for the turbine"
x4s$=x4$(x[4])

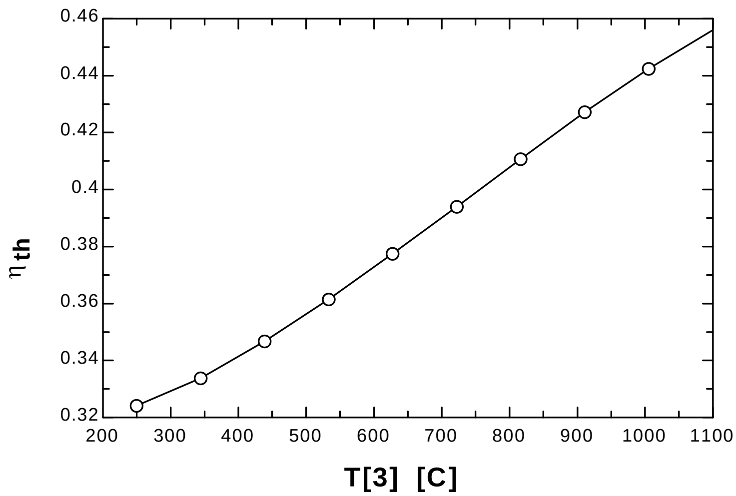
"Boiler analysis"
Q_in + h[2]=h[3]"SSSF First Law for the Boiler"

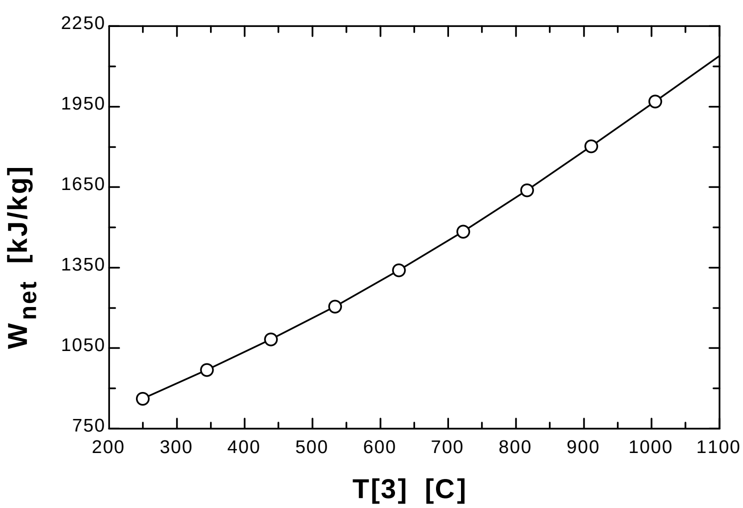
"Condenser analysis"
h[4]=Q_out+h[1]"SSSF First Law for the Condenser"

"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

T_3 [C]	η_{th}	W_{net} [kJ/kg]	x_4
250	0.3241	862.8	0.752
344.4	0.3338	970.6	0.81
438.9	0.3466	1083	0.8536
533.3	0.3614	1206	0.8909
627.8	0.3774	1340	0.9244
722.2	0.3939	1485	0.955
816.7	0.4106	1639	0.9835
911.1	0.4272	1803	100
1006	0.4424	1970	100
1100	0.456	2139	100







10-121 EES The effect of number of reheat stages on the performance an ideal Rankine cycle is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end

Procedure Reheat(P[3],T[3],T[5],h[4],NoRHStages,Pratio,Eta_t:Q_in_reheat,W_t_lp,h6)
P3=P[3]
T5=T[5]
h4=h[4]
Q_in_reheat =0
W_t_lp = 0
R_P=(1/Pratio)^(1/(NoRHStages+1))

 imax:=NoRHStages - 1
 i:=0

REPEAT
i:=i+1

P4 = P3*R_P

P5=P4
P6=P5*R_P

Fluid$='Steam_IAPWS'
s5=entropy(Fluid$,T=T5,P=P5)
h5=enthalpy(Fluid$,T=T5,P=P5)
s_s6=s5
hs6=enthalpy(Fluid$,s=s_s6,P=P6)
Ts6=temperature(Fluid$,s=s_s6,P=P6)
vs6=volume(Fluid$,s=s_s6,P=P6)
"Eta_t=(h5-h6)/(h5-hs6)""Definition of turbine efficiency"
h6=h5-Eta_t*(h5-hs6)
W_t_lp=W_t_lp+h5-h6"SSSF First Law for the low pressure turbine"
x6=QUALITY(Fluid$,h=h6,P=P6)
Q_in_reheat =Q_in_reheat + (h5 - h4)
P3=P4

UNTIL (i>imax)

END

"NoRHStages = 2"
P[6] = 10"kPa"
P[3] = 15000"kPa"
P_extract = P[6]  "Select a lower limit on the reheat pressure"
T[3] = 500"C"
T[5] = 500"C"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"
```

```
Pratio = P[3]/P_extract
P[4] = P[3]*(1/Pratio)^(1/(NoRHStages+1))"kPa"

Fluid$='Steam_IAPWS'

"Pump analysis"
P[1] = P[6]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"

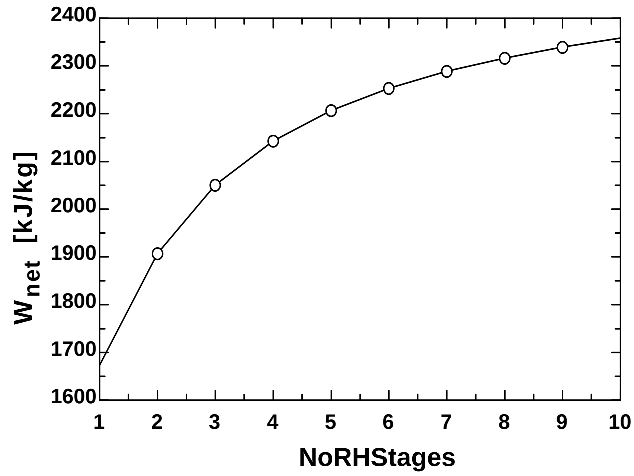
"Low Pressure Turbine analysis"
Call Reheat(P[3],T[3],T[5],h[4],NoRHStages,Pratio,Eta_t:Q_in_reheat,W_t_lp,h6)
h[6]=h6

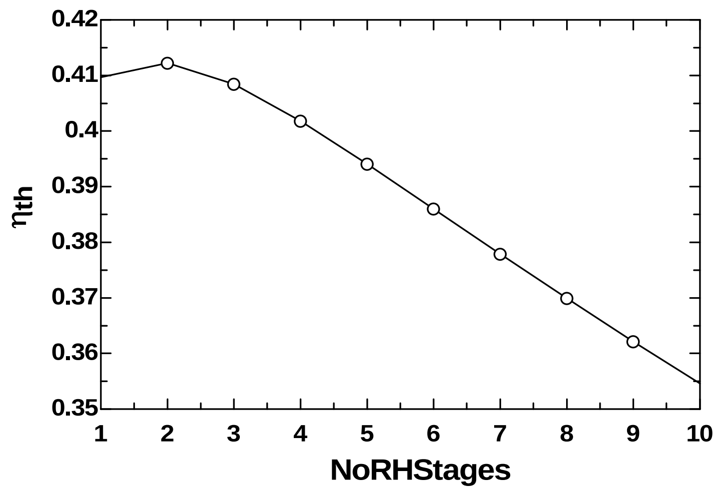
{P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
W_t_lp_total = NoRHStages*W_t_lp
Q_in_reheat = NoRHStages*(h[5] - h[4])}

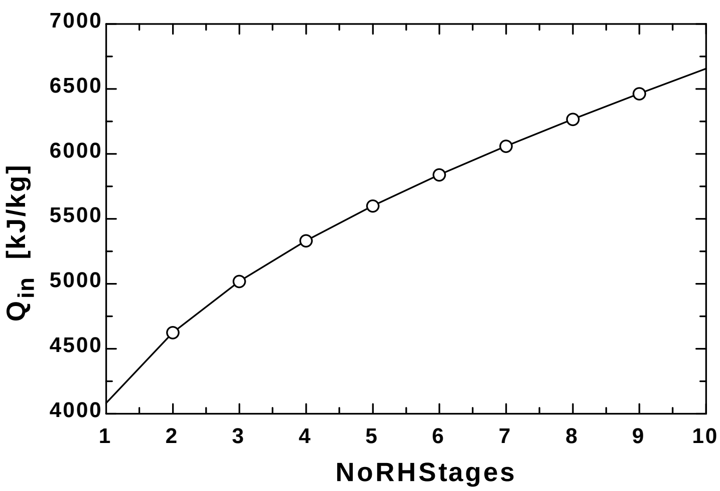
"Boiler analysis"
Q_in_boiler + h[2]=h[3]"SSSF First Law for the Boiler"
Q_in = Q_in_boiler+Q_in_reheat
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
```

```
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])
"Cycle Statistics"
W_net=W_t_hp+W_t_lp - W_p
Eta_th=W_net/Q_in
```

η_{th}	NoRH Stages	Q_{in} [kJ/kg]	W_{net} [kJ/kg]
0.4097	1	4085	1674
0.4122	2	4628	1908
0.4085	3	5020	2051
0.4018	4	5333	2143
0.3941	5	5600	2207
0.386	6	5838	2253
0.3779	7	6058	2289
0.3699	8	6264	2317
0.3621	9	6461	2340
0.3546	10	6651	2358







10-122 EES The effect of number of regeneration stages on the performance an ideal regenerative Rankine cycle with one open feedwater heater is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
Procedure Reheat(NoFwh,T[5],P[5],P_cond,Eta_turb,Eta_pump:q_in,w_net)

Fluid$='Steam_IAPWS'
Tcond = temperature(Fluid$,P=P_cond,x=0)
Tboiler = temperature(Fluid$,P=P[5],x=0)
P[7] = P_cond
s[5]=entropy(Fluid$, T=T[5], P=P[5])
h[5]=enthalpy(Fluid$, T=T[5], P=P[5])
h[1]=enthalpy(Fluid$, P=P[7],x=0)

P4[1] = P[5]  "NOTICE THIS IS P4[i] WITH i = 1"

DELTAT_cond_boiler = Tboiler - Tcond

If NoFWH = 0 Then

        "the following are h7, h2, w_net, and q_in for zero feedwater heaters, NoFWH = 0"
        h7=enthalpy(Fluid$, s=s[5],P=P[7])
        h2=h[1]+volume(Fluid$, P=P[7],x=0)*(P[5] - P[7])/Eta_pump
        w_net = Eta_turb*(h[5]-h7)-(h2-h[1])
        q_in = h[5] - h2

else

i=0
REPEAT
i=i+1
"The following maintains the same temperature difference between any two regeneration stages."
T_FWH[i] = (NoFWH +1 - i)*DELTAT_cond_boiler/(NoFWH + 1)+Tcond"[C]"
P_extract[i] = pressure(Fluid$,T=T_FWH[i],x=0)"[kPa]"
P3[i]=P_extract[i]
P6[i]=P_extract[i]
If i > 1 then P4[i] = P6[i - 1]

UNTIL i=NoFWH

P4[NoFWH+1]=P6[NoFWH]
h4[NoFWH+1]=h[1]+volume(Fluid$, P=P[7],x=0)*(P4[NoFWH+1] - P[7])/Eta_pump

i=0
REPEAT
i=i+1

"Boiler condensate pump  or the Pumps 2 between feedwater heaters analysis"
h3[i]=enthalpy(Fluid$,P=P3[i],x=0)
v3[i]=volume(Fluid$,P=P3[i],x=0)
w_pump2_s=v3[i]*(P4[i]-P3[i])"SSSF isentropic pump work assuming constant specific volume"
w_pump2[i]=w_pump2_s/Eta_pump "Definition of pump efficiency"
h4[i]= w_pump2[i] +h3[i]  "Steady-flow conservation of energy"
s4[i]=entropy(Fluid$,P=P4[i],h=h4[i])
T4[i]=temperature(Fluid$,P=P4[i],h=h4[i])
```

```
Until i = NoFWH
i=0
REPEAT
i=i+1
"Open Feedwater Heater analysis:"
{h2[i] = h6[i]}
s5[i] = s[5]
ss6[i]=s5[i]
hs6[i]=enthalpy(Fluid$,s=ss6[i],P=P6[i])
Ts6[i]=temperature(Fluid$,s=ss6[i],P=P6[i])
h6[i]=h[5]-Eta_turb*(h[5]-hs6[i])"Definition of turbine efficiency for high pressure stages"
If i=1 then y[1]=(h3[1] - h4[2])/(h6[1]  - h4[2])  "Steady-flow conservation of energy for the FWH"
If i > 1 then
  js = i -1
  j = 0
  sumyj = 0
  REPEAT
  j = j+1
  sumyj = sumyj + y[ j ]
  UNTIL j = js
y[i] =(1- sumyj)*(h3[i] - h4[i+1])/(h6[i]  - h4[i+1])

ENDIF
T3[i]=temperature(Fluid$,P=P3[i],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s3[i]=entropy(Fluid$,P=P3[i],x=0)

"Turbine analysis"
T6[i]=temperature(Fluid$,P=P6[i],h=h6[i])
s6[i]=entropy(Fluid$,P=P6[i],h=h6[i])
yh6[i] = y[i]*h6[i]
UNTIL i=NoFWH
ss[7]=s6[i]
hs[7]=enthalpy(Fluid$,s=ss[7],P=P[7])
Ts[7]=temperature(Fluid$,s=ss[7],P=P[7])
h[7]=h6[i]-Eta_turb*(h6[i]-hs[7])"Definition of turbine efficiency for low pressure stages"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])

sumyi = 0
sumyh6i = 0
wp2i =  W_pump2[1]
i=0
REPEAT
i=i+1
sumyi = sumyi + y[i]
sumyh6i = sumyh6i + yh6[i]
If NoFWH > 1 then wp2i = wp2i + (1- sumyi)*W_pump2[i]
UNTIL i = NoFWH

"Condenser Pump---Pump_1 Analysis:"
P[2] = P6 [ NoFWH]
P[1] = P_cond
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
```

```
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[2]=w_pump1+ h[1]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Boiler analysis"
q_in = h[5] - h4[1]"SSSF conservation of energy for the Boiler"
w_turb = h[5] - sumyh6i - (1- sumyi)*h[7]  "SSSF conservation of energy for turbine"

"Condenser analysis"
q_out=(1- sumyi)*(h[7] - h[1])"SSSF First Law for the Condenser"

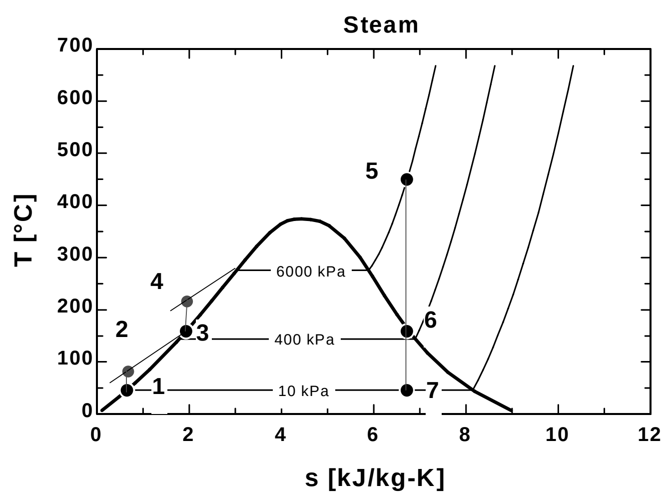
"Cycle Statistics"
w_net=w_turb - ((1- sumyi)*w_pump1+ wp2i)

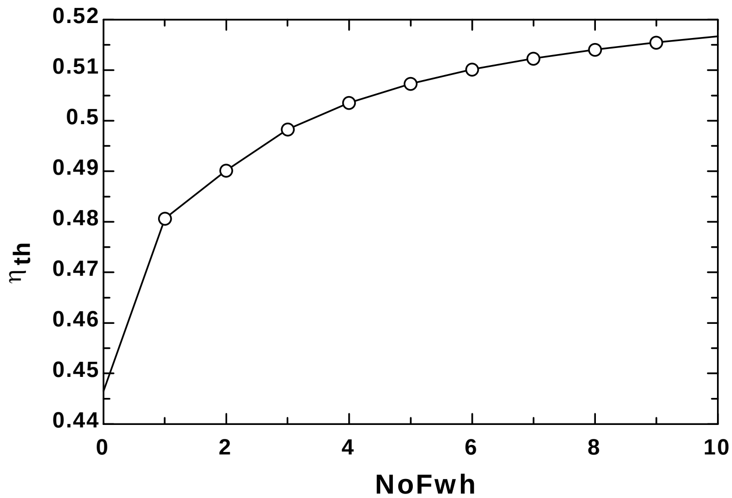
endif
END

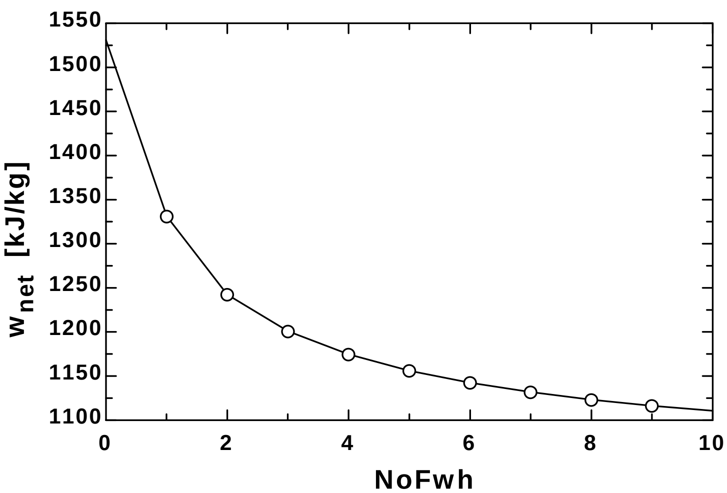
"Input Data"
NoFWH = 2
P[5] = 15000 [kPa]
T[5] = 600 [C]
P_cond=5 [kPa]
Eta_turb= 1.0  "Turbine isentropic efficiency"
Eta_pump = 1.0 "Pump isentropic efficiency"
P[1] = P_cond
P[4] = P[5]

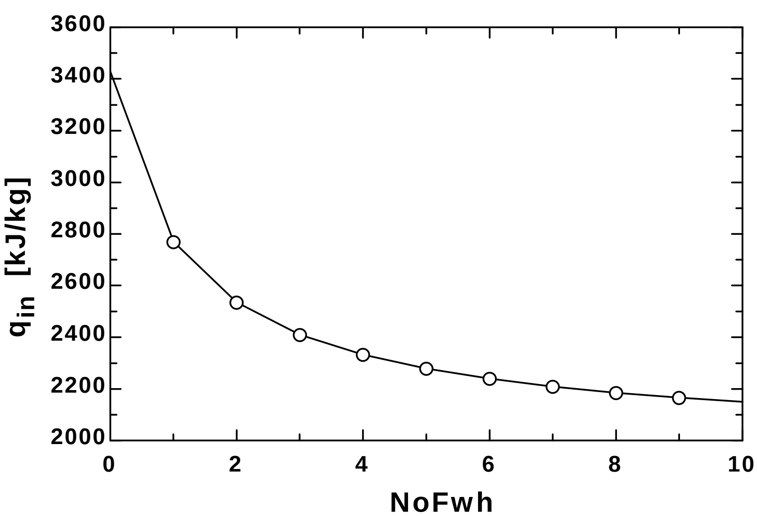
"Condenser exit pump  or Pump 1 analysis"
Call Reheat(NoFwh,T[5],P[5],P_cond,Eta_turb,Eta_pump:q_in,w_net)
Eta_th=w_net/q_in
```

No FWH	η_{th}	w_{net} [kJ/kg]	q_{in} [kJ/kg]
0	0.4466	1532	3430
1	0.4806	1332	2771
2	0.4902	1243	2536
3	0.4983	1202	2411
4	0.5036	1175	2333
5	0.5073	1157	2280
6	0.5101	1143	2240
7	0.5123	1132	2210
8	0.5141	1124	2186
9	0.5155	1117	2167
10	0.5167	1111	2151




0.52
0.51
0.5
0.49
0.48
0.47
0.46
0.45
0.44
η_{th}
0
2
4
6
8
10
NoFwh


1550
1500
1450
1400
1350
1300
1250
1200
1150
1100
w_{net} [kJ/kg]
0
2
4
6
8
10
NoFwh


3600
3400
3200
3000
2800
2600
2400
2200
2000
q_{in} [kJ/kg]
0
2
4
6
8
10
NoFwh

Fundamentals of Engineering (FE) Exam Problems

10-123 Consider a steady-flow Carnot cycle with water as the working fluid executed under the saturation dome between the pressure limits of 8 MPa and 20 kPa. Water changes from saturated liquid to saturated vapor during the heat addition process. The net work output of this cycle is

(a) 494 kJ/kg (b) 975 kJ/kg (c) 596 kJ/kg (d) 845 kJ/kg (e) 1148 kJ/kg

Answer (c) 596 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=8000 "kPa"
P2=20 "kPa"
h_fg=ENTHALPY(Steam_IAPWS,x=1,P=P1)-ENTHALPY(Steam_IAPWS,x=0,P=P1)
T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1)+273
T2=TEMPERATURE(Steam_IAPWS,x=0,P=P2)+273
q_in=h_fg
Eta_Carnot=1-T2/T1
w_net=Eta_Carnot*q_in

"Some Wrong Solutions with Common Mistakes:"
W1_work = Eta1*q_in; Eta1=T2/T1 "Taking Carnot efficiency to be T2/T1"
W2_work = Eta2*q_in; Eta2=1-(T2-273)/(T1-273)  "Using C instead of K"
W3_work = Eta_Carnot*ENTHALPY(Steam_IAPWS,x=1,P=P1)  "Using h_g instead of h_fg"
W4_work = Eta_Carnot*q2; q2=ENTHALPY(Steam_IAPWS,x=1,P=P2)-
ENTHALPY(Steam_IAPWS,x=0,P=P2) "Using h_fg at P2"
```

10-124 A simple ideal Rankine cycle operates between the pressure limits of 10 kPa and 3 MPa, with a turbine inlet temperature of 600°C. Disregarding the pump work, the cycle efficiency is

(a) 24% (b) 37% (c) 52% (d) 63% (e) 71%

Answer (b) 37%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=3000 "kPa"
P3=P2
P4=P1
T3=600 "C"
s4=s3
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1) "kJ/kg"
h2=h1+w_pump
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
q_in=h3-h2
q_out=h4-h1
Eta_th=1-q_out/q_in

"Some Wrong Solutions with Common Mistakes:"
W1_Eff = q_out/q_in "Using wrong relation"
W2_Eff = 1-(h44-h1)/(h3-h2); h44 = ENTHALPY(Steam_IAPWS,x=1,P=P4) "Using h_g for h4"
W3_Eff = 1-(T1+273)/(T3+273); T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1) "Using Carnot efficiency"
W4_Eff = (h3-h4)/q_in  "Disregarding pump work"
```

10-125 A simple ideal Rankine cycle operates between the pressure limits of 10 kPa and 5 MPa, with a turbine inlet temperature of 600°C. The mass fraction of steam that condenses at the turbine exit is

(a) 6% (b) 9% (c) 12% (d) 15% (e) 18%

Answer (c) 12%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=5000 "kPa"
P3=P2
P4=P1
T3=600 "C"
s4=s3
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
x4=QUALITY(Steam_IAPWS,s=s4,P=P4)
moisture=1-x4

"Some Wrong Solutions with Common Mistakes:"
W1_moisture = x4 "Taking quality as moisture"
W2_moisture = 0 "Assuming superheated vapor"
```

10-126 A steam power plant operates on the simple ideal Rankine cycle between the pressure limits of 10 kPa and 10 MPa, with a turbine inlet temperature of 600°C. The rate of heat transfer in the boiler is 800 kJ/s. Disregarding the pump work, the power output of this plant is

(a) 243 kW (b) 284 kW (c) 508 kW (d) 335 kW (e) 800 kW

Answer (d) 335 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=10000 "kPa"
P3=P2
P4=P1
T3=600 "C"
s4=s3
Q_rate=800 "kJ/s"
m=Q_rate/q_in
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
h2=h1  "pump work is neglected"
"v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1)
h2=h1+w_pump"
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
q_in=h3-h2
W_turb=m*(h3-h4)

"Some Wrong Solutions with Common Mistakes:"
W1_power = Q_rate "Assuming all heat is converted to power"
W3_power = Q_rate*Carnot; Carnot = 1-(T1+273)/(T3+273);
T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1) "Using Carnot efficiency"
W4_power = m*(h3-h44); h44 = ENTHALPY(Steam_IAPWS,x=1,P=P4) "Taking h4=h_g"
```

10-127 Consider a combined gas-steam power plant. Water for the steam cycle is heated in a well-insulated heat exchanger by the exhaust gases that enter at 800 K at a rate of 60 kg/s and leave at 400 K. Water enters the heat exchanger at 200°C and 8 MPa and leaves at 350°C and 8 MPa. If the exhaust gases are treated as air with constant specific heats at room temperature, the mass flow rate of water through the heat exchanger becomes

(a) 11 kg/s (b) 24 kg/s (c) 46 kg/s (d) 53 kg/s (e) 60 kg/s

Answer (a) 11 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_gas=60 "kg/s"
Cp=1.005 "kJ/kg.K"
T3=800 "K"
T4=400 "K"
Q_gas=m_gas*Cp*(T3-T4)
P1=8000 "kPa"
T1=200 "C"
P2=8000 "kPa"
T2=350 "C"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
Q_steam=m_steam*(h2-h1)
Q_gas=Q_steam

"Some Wrong Solutions with Common Mistakes:"
m_gas*Cp*(T3 -T4)=W1_msteam*4.18*(T2-T1) "Assuming no evaporation of liquid water"
m_gas*Cv*(T3 -T4)=W2_msteam*(h2-h1); Cv=0.718 "Using Cv for air instead of Cp"
W3_msteam = m_gas "Taking the mass flow rates of two fluids to be equal"
m_gas*Cp*(T3 -T4)=W4_msteam*(h2-h11); h11=ENTHALPY(Steam_IAPWS,x=0,P=P1) "Taking
h1=hf@P1"
```

10-128 An ideal reheat Rankine cycle operates between the pressure limits of 10 kPa and 8 MPa, with reheat occurring at 4 MPa. The temperature of steam at the inlets of both turbines is 500°C, and the enthalpy of steam is 3185 kJ/kg at the exit of the high-pressure turbine, and 2247 kJ/kg at the exit of the low-pressure turbine. Disregarding the pump work, the cycle efficiency is

(a) 29% (b) 32% (c) 36% (d) 41% (e) 49%

Answer (d) 41%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=8000 "kPa"
P3=P2
P4=4000 "kPa"
P5=P4
P6=P1
T3=500 "C"
T5=500 "C"
s4=s3
s6=s5
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
h2=h1
h44=3185 "kJ/kg - for checking given data"
h66=2247 "kJ/kg - for checking given data"
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
h5=ENTHALPY(Steam_IAPWS,T=T5,P=P5)
s5=ENTROPY(Steam_IAPWS,T=T5,P=P5)
h6=ENTHALPY(Steam_IAPWS,s=s6,P=P6)
q_in=(h3-h2)+(h5-h4)
q_out=h6-h1
Eta_th=1-q_out/q_in

"Some Wrong Solutions with Common Mistakes:"
W1_Eff = q_out/q_in "Using wrong relation"
W2_Eff = 1-q_out/(h3-h2) "Disregarding heat input during reheat"
W3_Eff = 1-(T1+273)/(T3+273); T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1) "Using Carnot efficiency"
W4_Eff = 1-q_out/(h5-h2)  "Using wrong relation for q_in"
```

10-129 Pressurized feedwater in a steam power plant is to be heated in an ideal open feedwater heater that operates at a pressure of 0.5 MPa with steam extracted from the turbine. If the enthalpy of feedwater is 252 kJ/kg and the enthalpy of extracted steam is 2665 kJ/kg, the mass fraction of steam extracted from the turbine is

(a) 4% (b) 10% (c) 16% (d) 27% (e) 12%

Answer (c) 16%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_feed=252 "kJ/kg"
h_extracted=2665 "kJ/kg"
P3=500 "kPa"
h3=ENTHALPY(Steam_IAPWS,x=0,P=P3)
"Energy balance on the FWH"
h3=x_ext*h_extracted+(1-x_ext)*h_feed

"Some Wrong Solutions with Common Mistakes:"
W1_ext = h_feed/h_extracted "Using wrong relation"
W2_ext = h3/(h_extracted-h_feed) "Using wrong relation"
W3_ext = h_feed/(h_extracted-h_feed) "Using wrong relation"
```

10-130 Consider a steam power plant that operates on the regenerative Rankine cycle with one open feedwater heater. The enthalpy of the steam is 3374 kJ/kg at the turbine inlet, 2797 kJ/kg at the location of bleeding, and 2346 kJ/kg at the turbine exit. The net power output of the plant is 120 MW, and the fraction of steam bled off the turbine for regeneration is 0.172. If the pump work is negligible, the mass flow rate of steam at the turbine inlet is

(a) 117 kg/s (b) 126 kg/s (c) 219 kg/s (d) 288 kg/s (e) 679 kg/s

Answer (b) 126 kg/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_in=3374 "kJ/kg"
h_out=2346 "kJ/kg"
h_extracted=2797 "kJ/kg"
Wnet_out=120000 "kW"
x_bleed=0.172
w_turb=(h_in-h_extracted)+(1-x_bleed)*(h_extracted-h_out)
m=Wnet_out/w_turb
"Some Wrong Solutions with Common Mistakes:"
W1_mass = Wnet_out/(h_in-h_out) "Disregarding extraction of steam"
W2_mass = Wnet_out/(x_bleed*(h_in-h_out)) "Assuming steam is extracted at trubine inlet"
W3_mass = Wnet_out/(h_in-h_out-x_bleed*h_extracted) "Using wrong relation"
```

10-131 Consider a simple ideal Rankine cycle. If the condenser pressure is lowered while keeping turbine inlet state the same, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the cycle efficiency will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the pump work input will decrease.

Answer (b) the amount of heat rejected will decrease.

10-132 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures. If the steam is superheated to a higher temperature, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the cycle efficiency will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the amount of heat input will decrease.

Answer (d) the moisture content at turbine exit will decrease.

10-133 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures . If the cycle is modified with reheating, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the pump work input will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the amount of heat input will decrease.

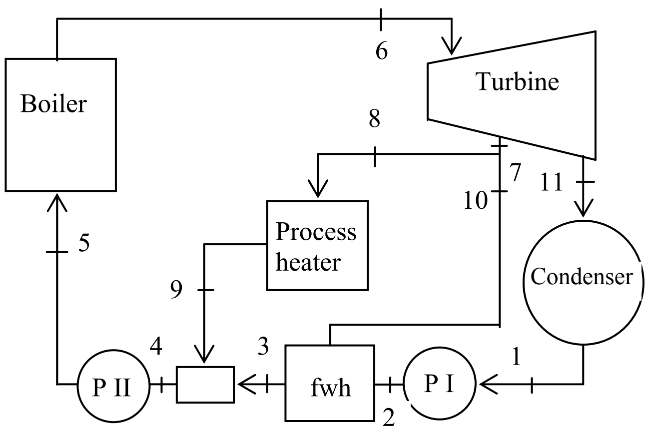
Answer (d) the moisture content at turbine exit will decrease.

10-134 Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures . If the cycle is modified with regeneration that involves one open feed water heater, (select the correct statement per unit mass of steam flowing through the boiler)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will increase.
(c) the cycle thermal efficiency will decrease.
(d) the quality of steam at turbine exit will decrease.
(e) the amount of heat input will increase.

Answer (a) the turbine work output will decrease.

10-135 Consider a cogeneration power plant modified with regeneration. Steam enters the turbine at 6 MPa and 450°C at a rate of 20 kg/s and expands to a pressure of 0.4 MPa. At this pressure, 60% of the steam is extracted from the turbine, and the remainder expands to a pressure of 10 kPa. Part of the extracted steam is used to heat feedwater in an open feedwater heater. The rest of the extracted steam is used for process heating and leaves the process heater as a saturated liquid at 0.4 MPa. It is subsequently mixed with the feedwater leaving the feedwater heater, and the mixture is pumped to the boiler pressure. The steam in the condenser is cooled and condensed by the cooling water from a nearby river, which enters the adiabatic condenser at a rate of 463 kg/s.



$h_1 = 191.81$
$h_2 = 192.20$
$h_3 = h_4 = h_9 = 604.66$
$h_5 = 610.73$
$h_6 = 3302.9$
$h_7 = h_8 = h_{10} = 2665.6$
$h_{11} = 2128.8$

1. The total power output of the turbine is

(*a*) 17.0 MW (*b*) 8.4 MW (*c*) 12.2 MW (*d*) 20.0 MW (*e*) 3.4 MW

Answer (a) 17.0 MW

2. The temperature rise of the cooling water from the river in the condenser is

(*a*) 8.0°C (*b*) 5.2°C (*c*) 9.6°C (*d*) 12.9°C (*e*) 16.2°C

Answer (*a*) 8.0°C

3. The mass flow rate of steam through the process heater is

(*a*) 1.6 kg/s (*b*) 3.8 kg/s (*c*) 5.2 kg/s (*d*) 7.6 kg/s (*e*) 10.4 kg/s

Answer (*e*) 10.4 kg/s

4. The rate of heat supply from the process heater per unit mass of steam passing through it is

(*a*) 246 kJ/kg (*b*) 893 kJ/kg (*c*) 1344 kJ/kg (*d*) 1891 kJ/kg (*e*) 2060 kJ/kg

Answer (*e*) 2060 kJ/kg

5. The rate of heat transfer to the steam in the boiler is

(*a*) 26.0 MJ/s (*b*) 53.8 MJ/s (*c*) 39.5 MJ/s (*d*) 62.8 MJ/s (*e*) 125.4 MJ/s

Answer (*b*) 53.8 MJ/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

Note: The solution given below also evaluates all enthalpies given on the figure.

```
P1=10 "kPa"
P11=P1
P2=400 "kPa"
P3=P2; P4=P2; P7=P2; P8=P2; P9=P2; P10=P2
P5=6000 "kPa"
P6=P5
T6=450 "C"
m_total=20 "kg/s"
m7=0.6*m_total
m_cond=0.4*m_total
C=4.18 "kJ/kg.K"
m_cooling=463 "kg/s"
s7=s6
s11=s6
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1)
h2=h1+w_pump
h3=ENTHALPY(Steam_IAPWS,x=0,P=P3)
h4=h3; h9=h3
v4=VOLUME(Steam_IAPWS,x=0,P=P4)
w_pump2=v4*(P5-P4)
h5=h4+w_pump2
h6=ENTHALPY(Steam_IAPWS,T=T6,P=P6)
s6=ENTROPY(Steam_IAPWS,T=T6,P=P6)
h7=ENTHALPY(Steam_IAPWS,s=s7,P=P7)
h8=h7; h10=h7
h11=ENTHALPY(Steam_IAPWS,s=s11,P=P11)
W_turb=m_total*(h6-h7)+m_cond*(h7-h11)
m_cooling*C*T_rise=m_cond*(h11-h1)
m_cond*h2+m_feed*h10=(m_cond+m_feed)*h3
m_process=m7-m_feed
q_process=h8-h9
Q_in=m_total*(h6-h5)
```

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