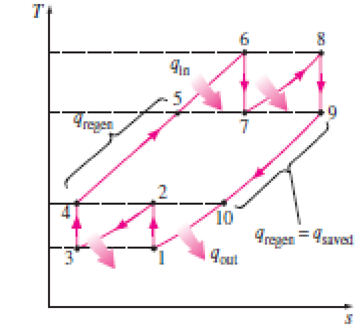
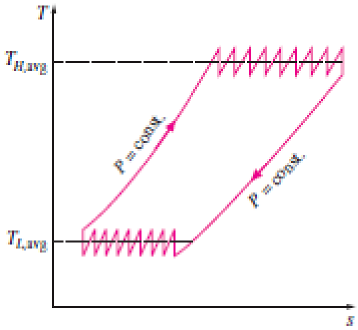
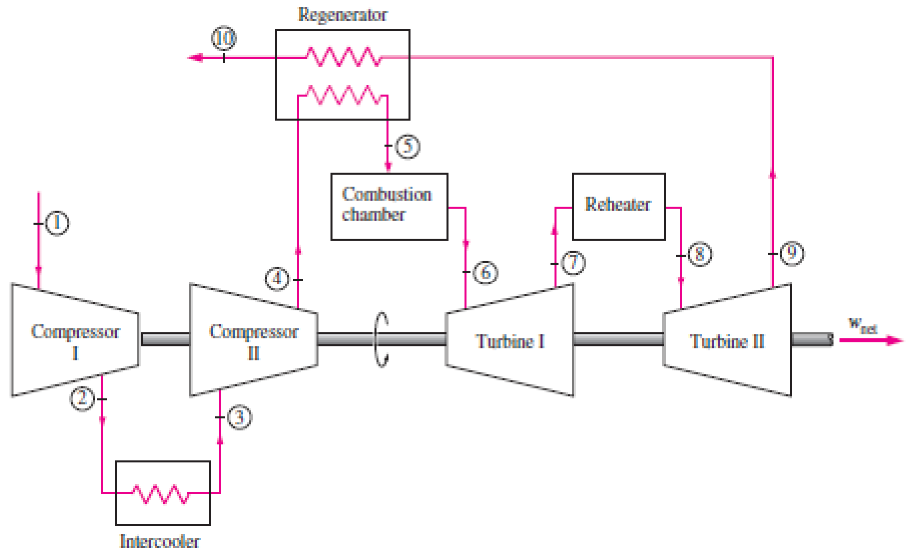






9–131 A gas-turbine power plant operates on the regenerative Brayton cycle between the pressure limits of 100 and 700 kPa. Air enters the compressor at 30°C at a rate of 12.6 kg/s and leaves at 260°C. It is then heated in a regenerator to 400°C by the hot combustion gases leaving the turbine. A diesel fuel with a heating value of 42,000 kJ/kg is burned in the combustion chamber with a combustion efficiency of 97 percent. The combustion gases leave the combustion chamber at 871°C and enter the turbine whose isentropic efficiency is 85 percent. Treating combustion gases as air and using constant specific heats at 500°C, determine (*a*) the isentropic efficiency of the compressor, (*b*) the effectiveness of the regenerator, (*c*) the air–fuel ratio in the combustion chamber, (*d*) the net power output and the back work ratio, (*e*) the thermal efficiency, and (*f*) the second-law efficiency of the plant. Also determine (*g*) the second-law (exergetic) efficiencies of the compressor, the turbine, and the regenerator, and (*h*) the rate of the exergy flow with the combustion gases at the regenerator exit. *Answers:* (*a*) 0.881, (*b*) 0.632, (*c*) 78.1, (*d*) 2267 kW, 0.583, (*e*) 0.345, (*f*) 0.469, (*g*) 0.929, 0.932, 0.890, (*h*) 1351 kW

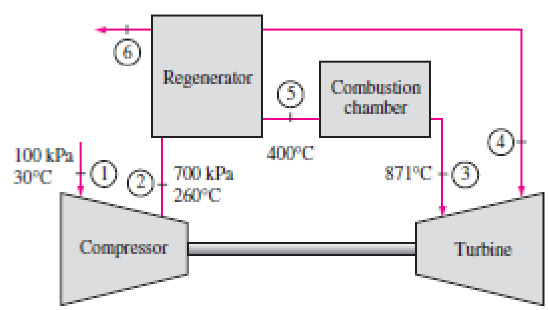
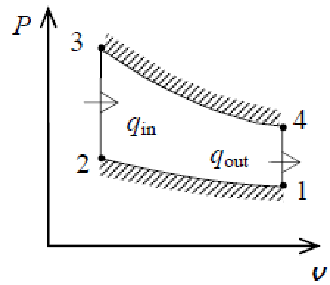


FIGURE P9–131

9-47 An ideal Otto cycle is considered. The heat rejection, the net work production, the thermal efficiency, and the mean effective pressure are to be determined. P1=90kPa, T1=23C, CR=7, m_air=0.0042

$$m = \frac{P_1 V_1}{RT_1} = \frac{(90\,\text{kPa})(0.004\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\,\text{K})} = 0.004181\,\text{kg}$$



The two unknown temperatures are

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (300\,\text{K})(7)^{1.4-1} = 653.4\,\text{K}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1} = (1400\,\text{K})\left(\frac{1}{7}\right)^{1.4-1} = 642.8\,\text{K}$$

Application of the first law to four cycle processes gives

$$W_{1-2} = mc_v(T_2 - T_1) = (0.004181\,\text{kg})(0.718\,\text{kJ/kg}\cdot\text{K})(653.4 - 300)\text{K} = 1.061\,\text{kJ}$$

$$Q_{2-3} = mc_v(T_3 - T_2) = (0.004181\,\text{kg})(0.718\,\text{kJ/kg}\cdot\text{K})(1400 - 653.4)\text{K} = 2.241\,\text{kJ}$$

$$W_{3-4} = mc_v(T_3 - T_4) = (0.004181\,\text{kg})(0.718\,\text{kJ/kg}\cdot\text{K})(1400 - 642.8)\text{K} = 2.273\,\text{kJ}$$

$$Q_{4-1} = mc_v(T_4 - T_1) = (0.004181\,\text{kg})(0.718\,\text{kJ/kg}\cdot\text{K})(642.8 - 300)\text{K} = \mathbf{1.029\,kJ}$$

The net work is

$$W_{\text{net}} = W_{3-4} - W_{1-2} = 2.273 - 1.061 = \mathbf{1.212\,kJ}$$

The thermal efficiency is then

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{1.212\,\text{kJ}}{2.241\,\text{kJ}} = \mathbf{0.541}$$

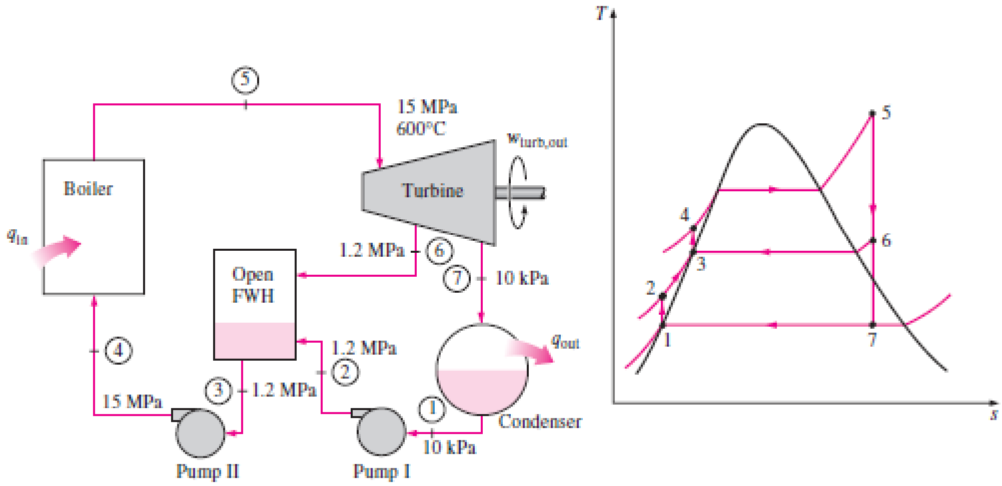
The minimum volume of the cycle occurs at the end of the compression

$$V_2 = \frac{V_1}{r} = \frac{0.004\,\text{m}^3}{7} = 0.0005714\,\text{m}^3$$

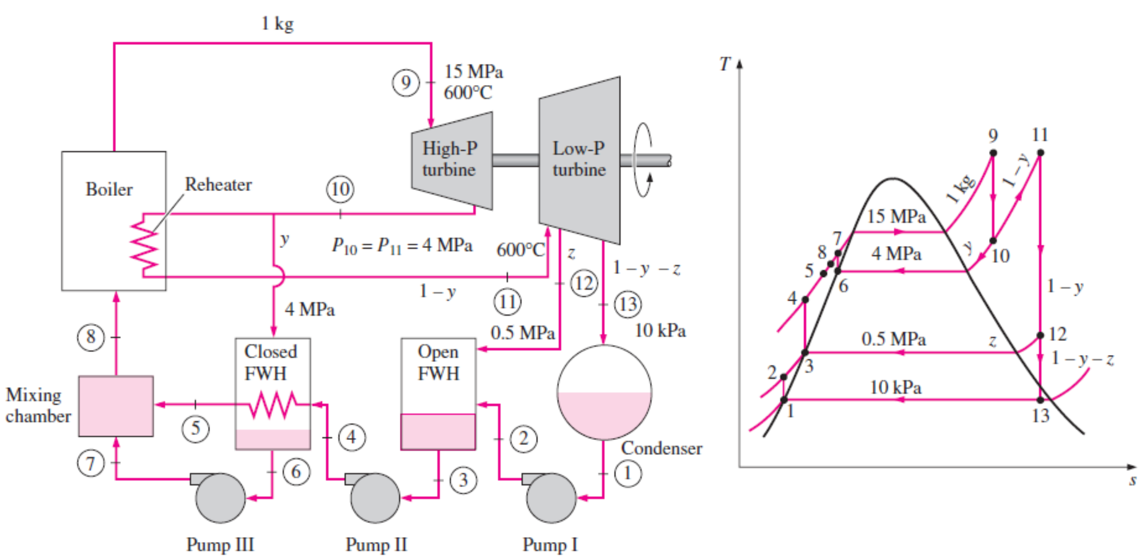
The engine's mean effective pressure is then

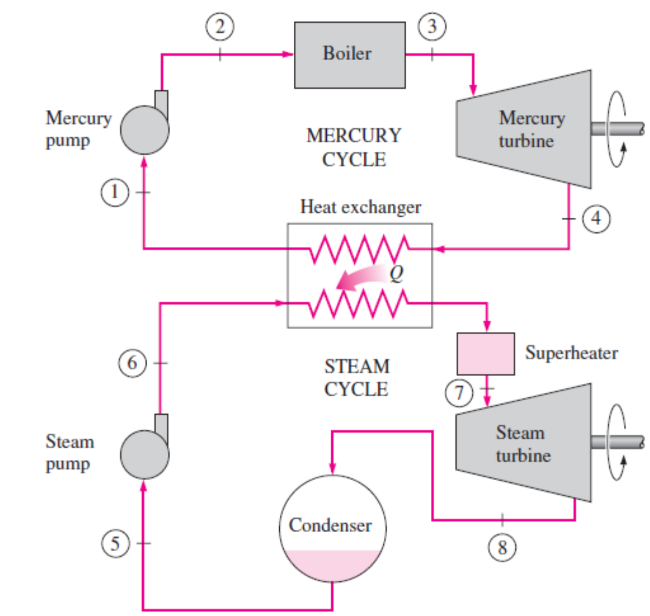
$$\text{MEP} = \frac{W_{\text{net}}}{V_1 - V_2} = \frac{1.212\,\text{kJ}}{(0.004 - 0.0005714)\,\text{m}^3}\left(\frac{1\,\text{kPa}\cdot\text{m}^3}{1\,\text{kJ}}\right) = \mathbf{354\,kPa}$$

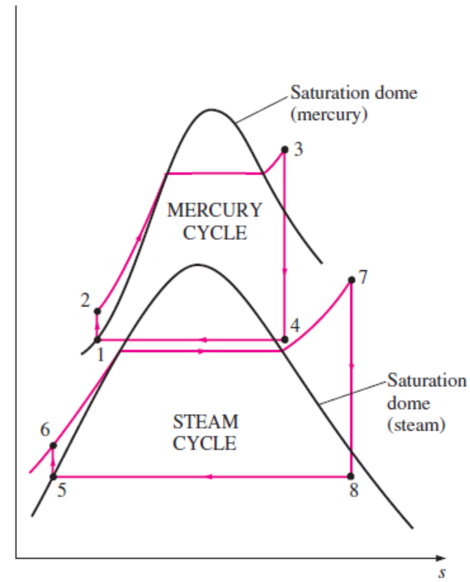
if diameter of the cylinder equals 10cm determine the ….

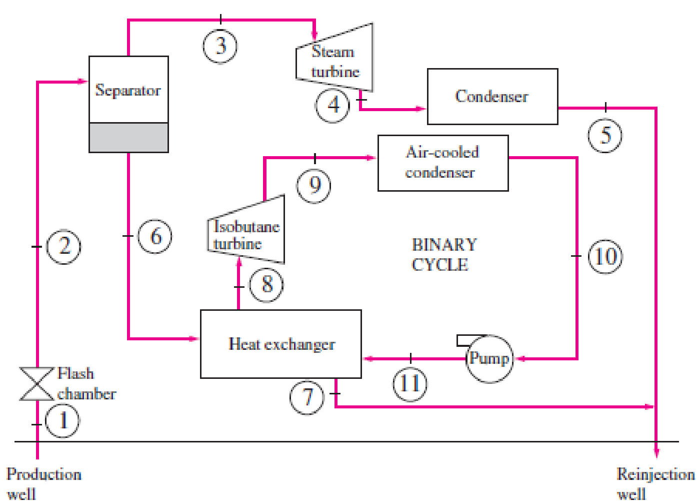


Consider a steam power plant that operates on an ideal reheat–regenerative Rankine cycle with one open feedwater heater, one closed feedwater heater, and one reheater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam is extracted from the turbine at 4 MPa for the closed feedwater heater, and the remaining steam is reheated at the same pressure to 600°C. The extracted steam is completely condensed in the heater and is pumped to 15 MPa before it mixes with the feedwater at the same pressure. Steam for the open feedwater heater is extracted from the low-pressure turbine at a pressure of 0.5 MPa. Determine the fractions of steam extracted from the turbine as well as the thermal efficiency of the cycle.









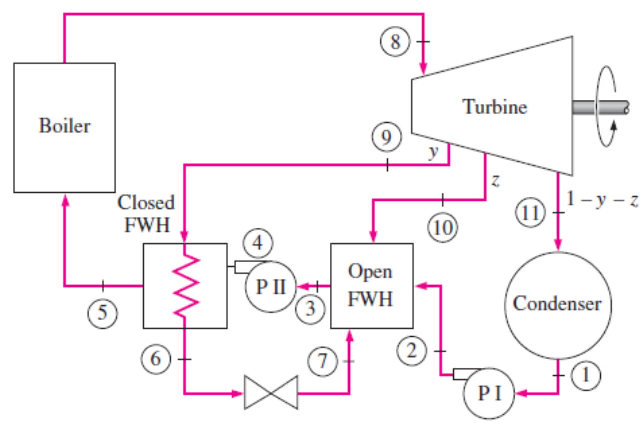


FIGURE P10–47

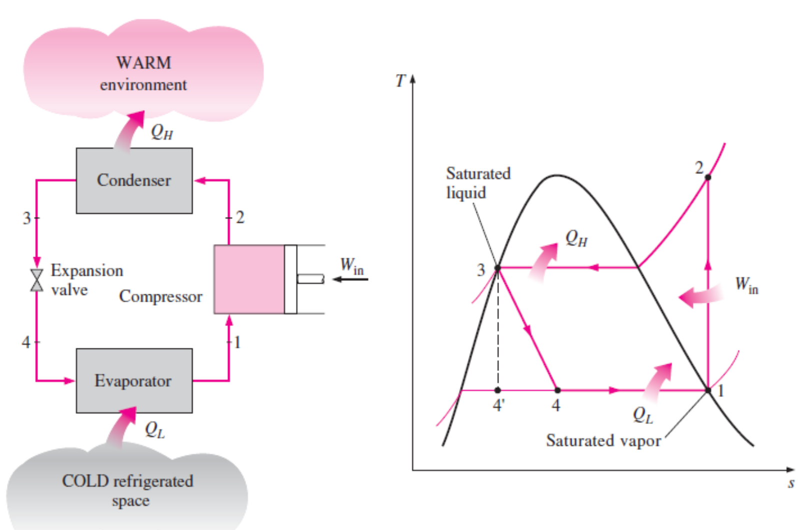
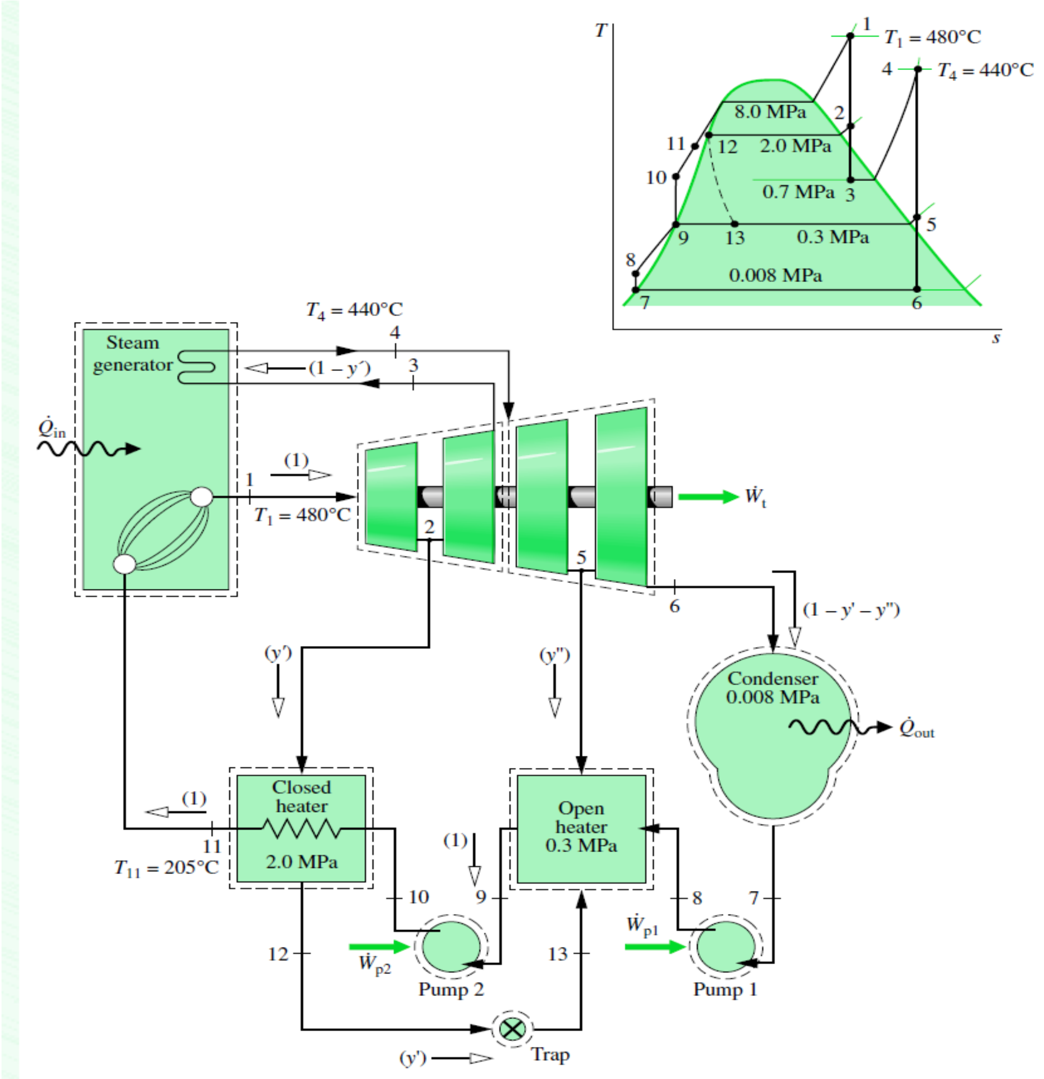


FIGURE 11–3

Schematic and *T-s* diagram for the ideal vapor-compression refrigeration cycle.



T
1
T_1 = 480°C
4
T_4 = 440°C
8.0 MPa
2
11
12
2.0 MPa
10
0.7 MPa
3
9
13
0.3 MPa
5
8
0.008 MPa
7
6
s
T_4 = 440°C
Steam generator
(1 – y′)
$\dot{Q}_{in}$
(1)
T_1 = 480°C
$\dot{W}_t$
(1 – y' – y")
(y')
(y")
Condenser
0.008 MPa
$\dot{Q}_{out}$
Closed heater
2.0 MPa
(1)
T_{11} = 205°C
Open heater
0.3 MPa
$\dot{W}_{p1}$
$\dot{W}_{p2}$
Pump 2
Pump 1
Trap
(y')

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)

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

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

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

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

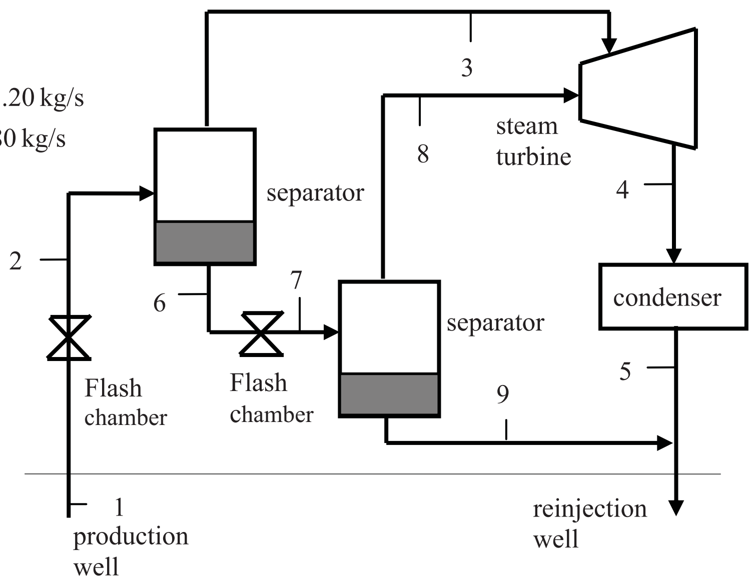
$$\left.\begin{array}{l} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{array}\right\} h_3 = 2748.1 \text{ kJ/kg}$$

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

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

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

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



(*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

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

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

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

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

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

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