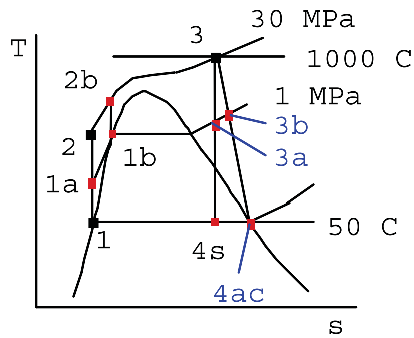
11.62

A supercritical steam power plant has a high pressure of 30 MPa and an exit condenser temperature of 50°C. The maximum temperature in the boiler is 1000°C and the turbine exhaust is saturated vapor There is one open feedwater heater receiving extraction from the turbine at 1MPa, and its exit is saturated liquid flowing to pump 2. The isentropic efficiency for the first section and the overall turbine are both 88.5%. Find the ratio of the extraction mass flow to total flow into turbine. What is the boiler inlet temperature with and without the feedwater heater?

Basically a Rankine Cycle
1: 50°C, 12.35 kPa,
$h = 209.31$ kJ/kg, $s = 0.7037$ kJ/kg K
2: 30 MPa
3: 30 MPa, 1000 °C,
$h = 4554.7$ kJ/kg, $s = 7.2867$ kJ/kg K
4AC: 50°C, $x = 1$, $h = 2592.1$ kJ/kg

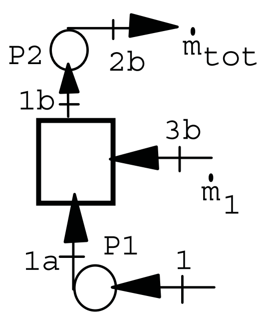


a) C.V. Turbine Ideal: $s_{4S} = s_3 \Rightarrow x_{4S} = 0.8929$,

$h_{4S} = 2336.8$ kJ/kg => $w_{T,S} = h_3 - h_{4S} = 2217.86$ kJ/kg

Actual: $w_{T,AC} = h_3 - h_{4AC} = 1962.6$ kJ/kg, $\eta = w_{T,AC}/w_{T,S} =$ **0.885**

b)



1b: Sat liq. 179.91°C, $h = 762.81$ kJ/kg
3a: 1 MPa, $s = s_3 \rightarrow h_{3a} = 3149.09$ kJ/kg,
$T_{3a} = 345.96 \rightarrow w_{T1s} = 1405.6$ kJ/kg
3b: 1 MPa, $w_{T1ac} = \eta w_{T1s} = 1243.96$ kJ/kg
$w_{T1ac} = h_3 - h_{3b} \Rightarrow h_{3b} = 3310.74$ kJ/kg
1a: $w_{P1} = v_1(P_{1a} - P_1) \approx 1$ kJ/kg
$h_{1a} = h_1 + w_{P1} = 210.31$ kJ/kg

C.V. Feedwater Heater: $\dot{m}_{TOT}h_{1b} = \dot{m}_1 h_{3b} + (\dot{m}_{TOT} - \dot{m}_1)h_{1a}$

$\Rightarrow \dot{m}_1/\dot{m}_{TOT} = x = (h_{1b} - h_{1a})/(h_{3b} - h_{1a}) =$ **0.178**

c) C.V. Turbine: $(\dot{m}_{TOT})_3 = (\dot{m}_1)_{3b} + (\dot{m}_{TOT} - \dot{m}_1)_{4AC}$

$\dot{W}_T = \dot{m}_{TOT}h_3 - \dot{m}_1 h_{3b} - (\dot{m}_{TOT} - \dot{m}_1)h_{4AC} = 25\text{ MW} = \dot{m}_{TOT}w_T$

$w_T = h_3 - xh_{3b} - (1-x)h_{4AC} = 1834.7$ kJ/kg => $\dot{m}_{TOT} =$ **13.63 kg/s**

d) C.V. No FWH, Pump Ideal: $w_P = h_{2S} - h_1$, $s_{2S} = s_1$

Steam table $\Rightarrow h_{2S} = 240.1$ kJ/kg, $T_{2S} =$ **51.2°C**

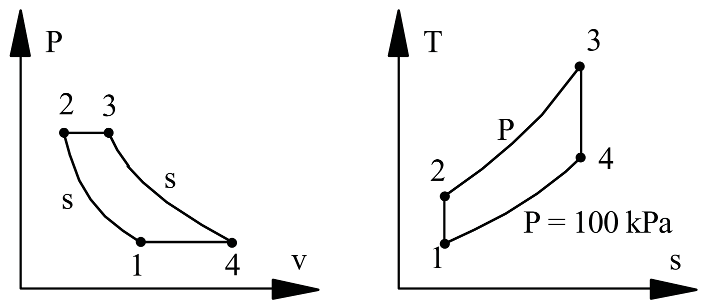
1 FWH, CV: P2. $s_{2b} = s_{1b} = 2.1386$ kJ/kg K => $T_{2b} =$ **183.9°C**

Brayton Cycles, Gas Turbines

11.68

Consider an ideal air-standard Brayton cycle in which the air into the compressor is at 100 kPa, 20°C, and the pressure ratio across the compressor is 12:1. The maximum temperature in the cycle is 1100°C, and the air flow rate is 10 kg/s. Assume constant specific heat for the air, value from Table A.5. Determine the compressor work, the turbine work, and the thermal efficiency of the cycle.

Solution:



Compression ratio $\frac{P_2}{P_1} = 12$

Max temperature $T_3 = 1100^oC$

$\dot{m} = 10$ kg/s

The compression is reversible and adiabatic so constant s. From Eq.8.32

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = 293.2(12)^{0.286} = 596.8 \text{ K}$$

Energy equation with compressor work in

$$w_C = -{}_1w_2 = C_{P0}(T_2 - T_1) = 1.004(596.8 - 293.2) = 304.8 \text{ kJ/kg}$$

The expansion is reversible and adiabatic so constant s. From Eq.8.32

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{\frac{k-1}{k}} = 1373.2\left(\frac{1}{12}\right)^{0.286} = 674.7 \text{ K}$$

Energy equation with turbine work out

$$w_T = C_{P0}(T_3 - T_4) = 1.004(1373.2 - 674.7) = 701.3 \text{ kJ/kg}$$

Scale the work with the mass flow rate

$$\dot{W}_C = \dot{m}w_C = \mathbf{3048\ kW}, \qquad \dot{W}_T = \dot{m}w_T = \mathbf{7013\ kW}$$

Energy added by the combustion process

$$q_H = C_{P0}(T_3 - T_2) = 1.004(1373.2 - 596.8) = 779.5 \text{ kJ/kg}$$

$$\eta_{TH} = w_{NET}/q_H = (701.3 - 304.8)/779.5 = \mathbf{0.509}$$

11.76

Repeat Problem 11.71, but include a regenerator with 75% efficiency in the cycle. A large stationary Brayton cycle gas-turbine power plant delivers a power output of 100 MW to an electric generator. The minimum temperature in the cycle is 300 K, and the maximum temperature is 1600 K. The minimum pressure in the cycle is 100 kPa, and the compressor pressure ratio is 14 to 1. Calculate the power output of the turbine. What fraction of the turbine output is required to drive the compressor? What is the thermal efficiency of the cycle?

Solution:

Both compressor and turbine are reversible and adiabatic so constant s, Eq.8.32 relates then T to P assuming constant heat capacity.

Compressor: $\Rightarrow T_2 = T_1(P_2/P_1)^{\frac{k-1}{k}} = 300(14)^{0.286} = 638.1$ K

$w_C = h_2 - h_1 = C_{P0}(T_2 - T_1) = 1.004\ (638.1 - 300) = 339.5$ kJ/kg

Turbine $s_4 = s_3 \Rightarrow T_4 = T_3(P_4/P_3)^{\frac{k-1}{k}} = 1600\ (1/14)^{0.286} = 752.2$ K

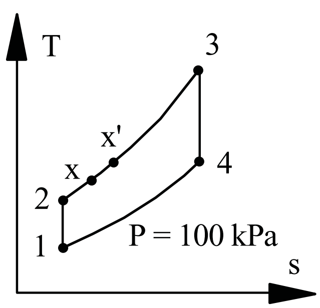
$w_T = h_3 - h_4 = C_{P0}(T_3 - T_4) = 1.004\ (1600 - 752.2) = 851.2$ kJ/kg

$w_{NET} = 851.2 - 339.5 = 511.7$ kJ/kg

$\dot{m} = \dot{W}_{NET}/w_{NET} = 100\ 000/511.7 = 195.4$ kg/s

$\dot{W}_T = \dot{m}w_T = 195.4 \times 851.2 =$ **166.32 MW**

$w_C/w_T = 339.5/851.2 =$ **0.399**



For the regenerator

$$\eta_{REG} = 0.75 = \frac{h_X - h_2}{h_{X'} - h_2} = \frac{T_X - T_2}{T_4 - T_2} = \frac{T_X - 638.1}{752.2 - 638.1}$$

$\Rightarrow T_X = 723.7$ K

Turbine and compressor work not affected by regenerator.

Combustor needs to add less energy with the regenerator as

$q_H = C_{P0}(T_3 - T_X) = 1.004(1600 - 723.7) = 879.8$ kJ/kg

$\eta_{TH} = w_{NET}/q_H = 511.7/879.8 =$ **0.582**

11.78

A two-stage compressor in a gas turbine brings atmospheric air at 100 kPa, 17°C to 500 kPa, then cools it in an intercooler to 27°C at constant P. The second stage brings the air to 1000 kPa. Assume both stages are adiabatic and reversible. Find the combined specific work to the compressor stages. Compare that to the specific work for the case of no intercooler (i.e. one compressor from 100 to 1000 kPa).

Solution:

C.V. Stage 1: 1 => 2

Reversible and adiabatic gives constant s which from Eq.8.32 gives:

$$T_2 = T_1 (P_2/P_1)^{(k-1)/k} = 290 (500/100)^{0.2857} = 459.3 \text{ K}$$

$$w_{c1in} = C_P(T_2 - T_1) = 1.004(459.3 – 290) = 187.0 \text{ kJ/kg}$$

C.V. Stage 2: 3 => 4

Reversible and adiabatic gives constant s which from Eq.8.32 gives:

$$T_4 = T_3 (P_4/P_3)^{(k-1)/k} = 300 (1000/500)^{0.2857} = 365.7 \text{ K}$$

$$w_{c2in} = C_P(T_4 - T_3) = 1.004(365.7 – 300) = 65.96 \text{ kJ/kg}$$

$$w_{tot} = w_{c1} + w_{c2} = 187 + 65.96 = \mathbf{253 \text{ kJ/kg}}$$

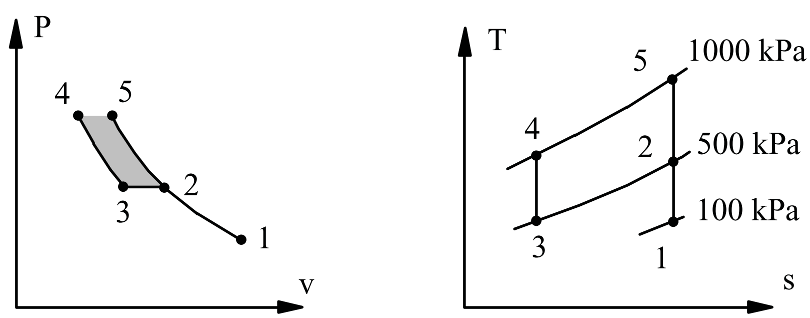
The intercooler reduces the work for stage 2 as T is lower and so is specific volume.

C.V. One compressor 1 => 5

Reversible and adiabatic gives constant s which from Eq.8.32 gives:

$$T_5 = T_1 (P_5/P_1)^{(k-1)/k} = 290 (1000/100)^{0.2857} = 559.88 \text{ K}$$

$$w_{in} = C_P(T_5 - T_1) = 1.004(559.88 – 290) = \mathbf{271 \text{ kJ/kg}}$$



The reduction in work due to the intercooler is shaded in the P-v diagram.

11.94

A gasoline engine has a volumetric compression ratio of 9. The state before compression is 290 K, 90 kPa, and the peak cycle temperature is 1800 K. Find the pressure after expansion, the cycle net work and the cycle efficiency using properties from Table A.5.

Compression 1 to 2: $s_2 = s_1 \;\Rightarrow$ From Eq.8.33 and Eq.8.34

$$T_2 = T_1 \,(v_1/v_2)^{k-1} = 290 \times 9^{0.4} = 698.4 \text{ K}$$

$$P_2 = P_1 \times (v_1/v_2)^{k} = 90 \times 9^{1.4} = 1950.7 \text{ kPa}$$

Combustion 2 to 3 at constant volume: $v_3 = v_2$

$$q_H = u_3 - u_2 = C_v(T_3 - T_2) = 0.717\,(1800 - 698.4) = 789.85 \text{ kJ/kg}$$

$$P_3 = P_2 \times (T_3/T_2) = 1950.7\,(1800 / 698.4) = 5027.6 \text{ kPa}$$

Expansion 3 to 4: $s_4 = s_3 \;\Rightarrow$ From Eq.8.33 and Eq.8.34

$$T_4 = T_3 \,(v_3/v_4)^{k-1} = 1800 \times (1/9)^{0.4} = 747.4 \text{ K}$$

$$P_4 = P_3(T_4/T_3)(v_3/v_4) = 5027.6\,(747.4/1800)\,(1/9) = \mathbf{232\ kPa}$$

Find now the net work

$$_1w_2 = u_1 - u_2 = C_v(T_1 - T_2) = 0.717(290 - 698.4) = -292.8 \text{ kJ/kg}$$

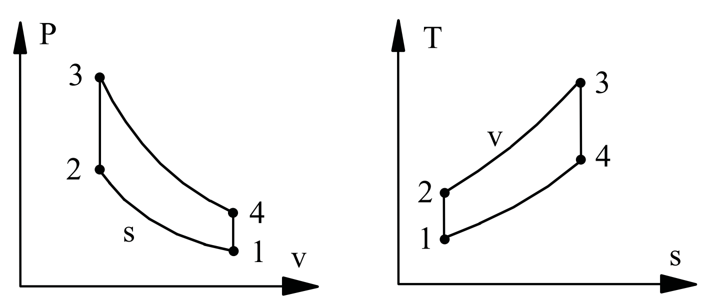
$$_3w_4 = u_3 - u_4 = C_v(T_3 - T_4) = 0.717(1800 - 747.4) = 754.7 \text{ kJ/kg}$$

Net work and overall efficiency

$$w_{NET} = {}_3w_4 + {}_1w_2 = 754.7 - 292.8 = \mathbf{461.9\ kJ/kg}$$

$$\eta = w_{NET}/q_H = 461.9/789.85 = \mathbf{0.585}$$

Comment: We could have found η from Eq.11.18 and then $w_{NET} = \eta q_H$.



11.109

At the beginning of compression in a diesel cycle T = 300 K, P = 200 kPa and after combustion (heat addition) is complete T = 1500 K and P = 7.0 MPa. Find the compression ratio, the thermal efficiency and the mean effective pressure.

Solution:

Standard Diesel cycle. See P-v and T-s diagrams for state numbers.

Compression process (isentropic) from Eqs.8.33-8.34

$P_2 = P_3 = 7000$ kPa => $v_1 / v_2 = (P_2/P_1)^{1/k} = (7000 / 200)^{0.7143} = \mathbf{12.67}$

$T_2 = T_1(P_2 / P_1)^{(k-1)/k} = 300(7000 / 200)^{0.2857} = 828.4$ K

Expansion process (isentropic) first get the volume ratios

$v_3 / v_2 = T_3 / T_2 = 1500 / 828.4 = 1.81$

$v_4 / v_3 = v_1 / v_3 = (v_1 / v_2)(v_2 / v_3) = 12.67 / 1.81 = 7$

The exhaust temperature follows from Eq.8.33

$T_4 = T_3(v_3 / v_4)^{k-1} = (1500 / 7)^{0.4} = 688.7$ K

$q_L = C_{vo}(T_4 - T_1) = 0.717(688.7 - 300) = 278.5$ kJ/kg

$q_H = h_3 - h_2 \approx C_{po}(T_3 - T_2) = 1.004(1500 - 828.4) = 674$ kJ/kg

Overall performance

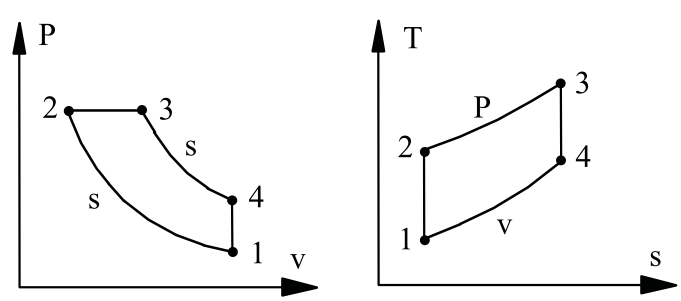
$\eta = 1 - q_L / q_H = 1 - 278.5 / 674 = \mathbf{0.587}$

$w_{net} = q_{net} = q_H - q_L = 674 - 278.5 = 395.5$ kJ/kg

$v_{max} = v_1 = R\,T_1 / P_1 = 0.287 \times 300 / 200 = 0.4305$ m^3/kg

$v_{min} = v_{max} / (v_1 / v_2) = 0.4305 / 12.67 = 0.034$ m^3/kg

$$P_{meff} = \frac{w_{net}}{v_{max} - v_{min}} = 395.5 / (0.4305 - 0.034) = \mathbf{997\ kPa}$$



Remark: This is a too low compression ratio for a practical diesel cycle.

11.163

The effect of a number of open feedwater heaters on the thermal efficiency of an ideal cycle is to be studied. Steam leaves the steam generator at 20 MPa, 600°C, and the cycle has a condenser pressure of 10 kPa. Determine the thermal efficiency for each of the following cases. **A:** No feedwater heater. **B**: One feedwater heater operating at 1 MPa. **C:** Two feedwater heaters, one operating at 3 MPa and the other at 0.2 MPa.

a) no feed water heater

$w_P = \int_1^2 vdP$

$\approx 0.00101(20000 - 10)$

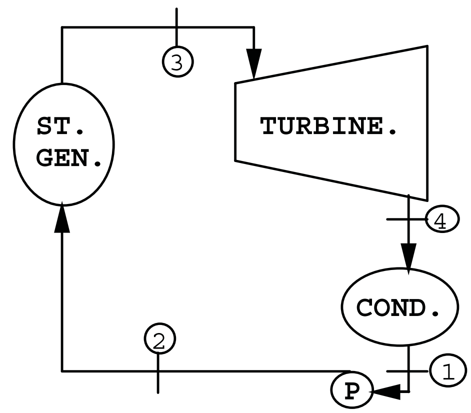
$= 20.2$ kJ/kg

$h_2 = h_1 + w_P = 191.8 + 20.2 = 212.0$

$s_4 = s_3 = 6.5048$

$= 0.6493 + x_4 \times 7.5009$

$x_4 = 0.78064$



$h_4 = 191.83 + 0.780\,64 \times 2392.8$

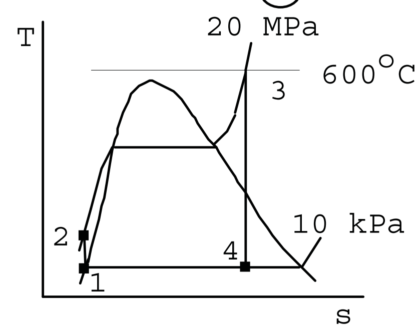
$= 2059.7$

$w_T = h_3 - h_4 = 3537.6 - 2059.7$

$= 1477.9$ kJ/kg

$w_N = w_T - w_P = 1477.9 - 20.2 = 1457.7$

$q_H = h_3 - h_2 = 3537.6 - 212.0 = 3325.6$



$$\eta_{TH} = \frac{w_N}{q_H} = \frac{1457.7}{3325.6} = \mathbf{0.438}$$

b) one feedwater heater

$w_{P12} = 0.00101(1000 - 10)$

$= 1.0$ kJ/kg

$h_2 = h_1 + w_{P12} = 191.8 + 1.0 = 192.8$

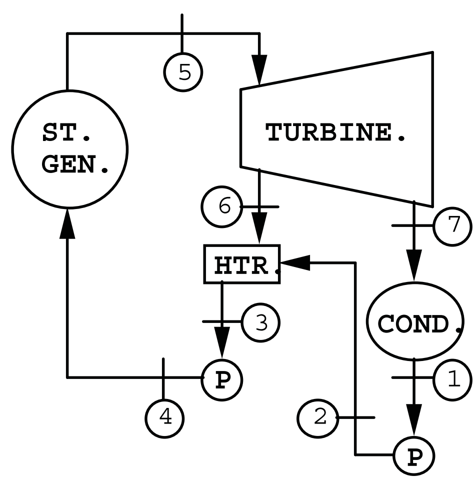
$w_{P34} = 0.001127\ (20000 - 1000)$

$= 21.4$ kJ/kg

$h_4 = h_3 + w_{P34} = 762.8 + 21.4 = 784.2$

$s_6 = s_5 = 6.5048$

$= 2.1387 + x_6 \times 4.4478$



$x_6 = 0.9816$

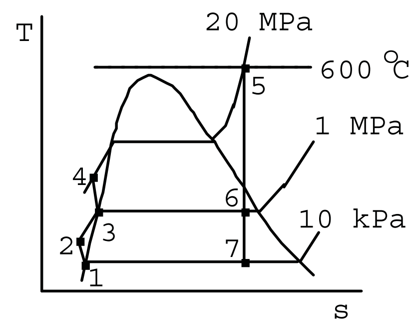
$h_6 = 762.8 + 0.9816 \times 2015.3 = 2741.1$

CV: heater

const: $m_3 = m_6 + m_2 = 1.0$ kg

1st law: $m_6 h_6 + m_2 h_2 = m_3 h_3$

$$m_6 = \frac{762.8 - 192.8}{2741.1 - 192.8} = 0.2237$$



$m_2 = 0.7763$, $h_7 = 2059.7$ ($= h_4$ of part a))

CV: turbine $w_T = (h_5 - h_6) + m_2(h_6 - h_7)$

$= (3537.6 - 2741.1) + 0.7763(2741.1 - 2059.7) = 1325.5$ kJ/kg

CV: pumps

$w_P = m_1 w_{P12} + m_3 w_{P34} = 0.7763(1.0) + 1(21.4) = 22.2$ kJ/kg

$w_N = 1325.5 - 22.2 = 1303.3$ kJ/kg

CV: steam generator

$q_H = h_5 - h_4 = 3537.6 - 784.2 = 2753.4$ kJ/kg

$\eta_{TH} = w_N/q_H = 1303.3/2753.4 =$ **0.473**

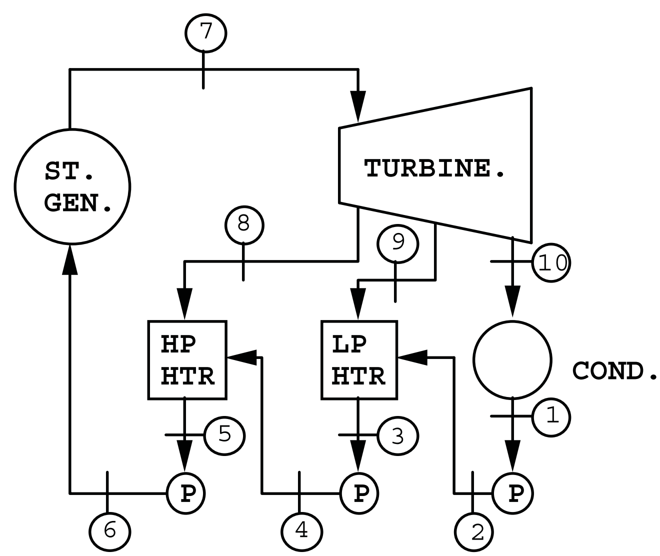
c) two feedwater heaters

$w_{P12} = 0.00101 \times (200 - 10) = 0.2$ kJ/kg

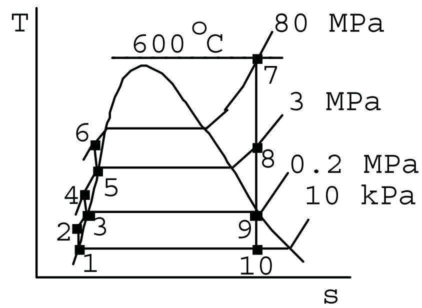
$h_2 = w_{P12} + h_1 = 191.8 + 0.2 = 192.0$

$w_{P34} = 0.001061 \times (3000 - 200) = 3.0$ kJ/kg

$h_4 = h_3 + w_{P34} = 504.7 + 3.0 = 507.7$



$w_{P56} = 0.001217(20000 - 3000)$
$= 20.7$ kJ/kg
$h_6 = h_5 + w_{P56} = 1008.4 + 20.7 = 1029.1$
$s_8 = s_7 = 6.5048$ at $P_8 = 3$ MPa $\Big\}$ $T_8 = 293.2\ ^oC$, $h_8 = 2974.8$
$s_9 = s_8 = 6.5048 = 1.5301 + x_9 \times 5.5970$



$x_9 = 0.8888 \;\Rightarrow\; h_9 = 504.7 + 0.888 \times 2201.9 = 2461.8$ kJ/kg

CV: high pressure heater

cont: $m_5 = m_4 + m_8 = 1.0$ kg ; 1st law: $m_5h_5 = m_4h_4 + m_8h_8$

$$m_8 = \frac{1008.4 - 507.7}{2974.8 - 507.7} = 0.2030 \qquad m_4 = 0.7970$$

CV: low pressure heater

cont: $m_9 + m_2 = m_3 = m_4$; 1st law: $m_9h_9 + m_2h_2 = m_3h_3$

$$m_9 = \frac{0.7970(504.7 - 192.0)}{2461.8 - 192.0} = 0.1098$$

$m_2 = 0.7970 - 0.1098 = 0.6872$

CV: turbine

$w_T = (h_7 - h_8) + (1 - m_8)(h_8 - h_9) + (1 - m_8 - m_9)(h_9 - h_{10})$
$= (3537.6 - 2974.8) + 0.797(2974.8 - 2461.8)$
$+ 0.6872(2461.8 - 2059.7) = 1248.0$ kJ/kg

CV: pumps

$w_P = m_1w_{P12} + m_3w_{P34} + m_5w_{P56}$
$= 0.6872(0.2) + 0.797(3.0) + 1(20.7) = 23.2$ kJ/kg
$w_N = 1248.0 - 23.2 = 1224.8$ kJ/kg

CV: steam generator

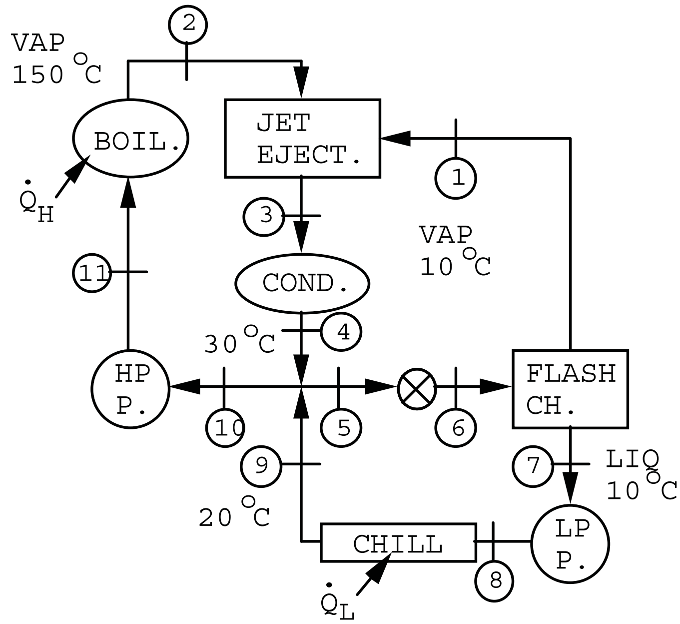
$q_H = h_7 - h_6 = 3537.6 - 1029.1 = 2508.5$ kJ/kg

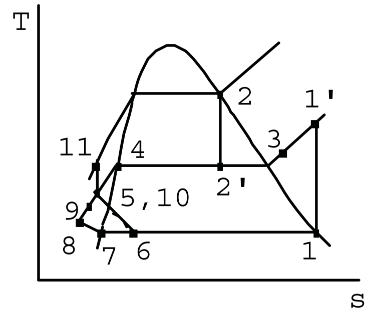
$\eta_{TH} = w_N/q_H = 1224.8/2508.5 = \mathbf{0.488}$

11.166

A jet ejector, a device with no moving parts, functions as the equivalent of a coupled turbine-compressor unit (see Problems 9.82 and 9.90). Thus, the turbine-compressor in the dual-loop cycle of Fig. P11.109 could be replaced by a jet ejector. The primary stream of the jet ejector enters from the boiler, the secondary stream enters from the evaporator, and the discharge flows to the condenser. Alternatively, a jet ejector may be used with water as the working fluid. The purpose of the device is to chill water, usually for an air-conditioning system. In this application the physical setup is as shown in Fig. P11.116. Using the data given on the diagram, evaluate the performance of this cycle in terms of the ratio Q_L/Q_H.

a. Assume an ideal cycle.
b. Assume an ejector efficiency of 20% (see Problem 9.90).





$T_1 = T_7 = 10\ ^oC$

$T_2 = 150\ ^oC$

$T_4 = 30\ ^oC$

$T_9 = 20\ ^oC$

Assume $T_5 = T_{10}$

(from mixing streams 4 & 9).

$P_3 = P_4 = P_5 = P_8 = P_9 = P_{10} = P_{G\ 30\ ^oC} = 4.246$ kPa

$P_{11} = P_2 = P_{G\ 150^oC} = 475.8$ kPa, $\quad P_1 = P_6 = P_7 = P_{G\ 10^oC} = 1.2276$ kPa

Cont: $\dot{m}_1 + \dot{m}_9 = \dot{m}_5 + \dot{m}_{10}$, $\quad \dot{m}_5 = \dot{m}_6 = \dot{m}_7 + \dot{m}_1$

$\dot{m}_7 = \dot{m}_8 = \dot{m}_9$, $\quad \dot{m}_{10} = \dot{m}_{11} = \dot{m}_2$, $\quad \dot{m}_3 = \dot{m}_4$

a) $\dot{m}_1 + \dot{m}_2 = \dot{m}_3$; ideal jet ejector

$s'_1 = s_1$ & $s'_2 = s_2$ (1' & 2' at $P_3 = P_4$)

then, $\dot{m}_1(h'_1 - h_1) = \dot{m}_2(h_2 - h'_2)$

From $s_2' = s_2 = 0.4369 + x_2' \times 8.0164; \quad x_2' = 0.7985$

$h_2' = 125.79 + 0.7985 \times 2430.5 = 2066.5$ kJ/kg

From $s_1' = s_1 = 8.9008 \Rightarrow T_1' = 112$ °C, $h_1' = 2710.4$ kJ/kg

$$\Rightarrow \dot{m}_1/\dot{m}_2 = \frac{2746.5 - 2066.5}{2710.4 - 2519.8} = 3.5677$$

Also $h_4 = 125.79$ kJ/kg, $h_7 = 42.01$ kJ/kg, $h_9 = 83.96$ kJ/kg

Mixing of streams 4 & 9 $\Rightarrow$ 5 & 10:

$$(\dot{m}_1 + \dot{m}_2)h_4 + \dot{m}_7 h_9 = (\dot{m}_7 + \dot{m}_1 + \dot{m}_2)h_{5\,=\,10}$$

Flash chamber (since $h_6 = h_5$) : $(\dot{m}_7 + \dot{m}_1)h_{5\,=\,10} = \dot{m}_1 h_1 + \dot{m}_7 h_1$

$\Rightarrow$ using the primary stream $\dot{m}_2 = 1$ kg/s:

$4.5677 \times 125.79 + \dot{m}_7 \times 83.96 = (\dot{m}_7 + 4.5677)h_5$

& $(\dot{m}_7 + 3.5677)h_5 = 3.5677 \times 2519.8 + \dot{m}_7 \times 42.01$

Solving, $\dot{m}_7 = 202.627$ & $h_5 = 84.88$ kJ/kg

LP pump: $-w_{LP\,P} = 0.0010(4.246 - 1.2276) = 0.003$ kJ/kg

$h_8 = h_7 - w_{LP\,P} = 42.01 + 0.003 = 42.01$ kJ/kg

Chiller: $\dot{Q}_L = \dot{m}_7(h_9 - h_8) = 202.627(83.96 - 42.01) = 8500$ kW (for $\dot{m}_2 = 1$)

HP pump: $-w_{HP\,P} = 0.001002(475.8 - 4.246) = 0.47$ kJ/kg

$h_{11} = h_{10} - w_{HP\,P} = 84.88 + 0.47 = 85.35$ kJ/kg

Boiler: $\dot{Q}_{11} = \dot{m}_{11}(h_2 - h_{11}) = 1(2746.5 - 85.35) = 2661.1$ kW

$\Rightarrow \dot{Q}_L/\dot{Q}_H = 8500/2661.1 =$ **3.194**

b) Jet eject. eff. $= (\dot{m}_1/\dot{m}_2)_{ACT}/(\dot{m}_1/\dot{m}_2)_{IDEAL} = 0.20$

$\Rightarrow (\dot{m}_1/\dot{m}_2)_{ACT} = 0.2 \times 3.5677 = 0.7135$

using $\dot{m}_2 = 1$ kg/s: $1.7135 \times 125.79 + \dot{m}_7 \times 83.96 = (\dot{m}_7 + 1.7135)h_5$

& $(\dot{m}_7 + 0.7135)h_5 = 0.7135 \times 2519.8 + \dot{m}_7 \times 42.01$

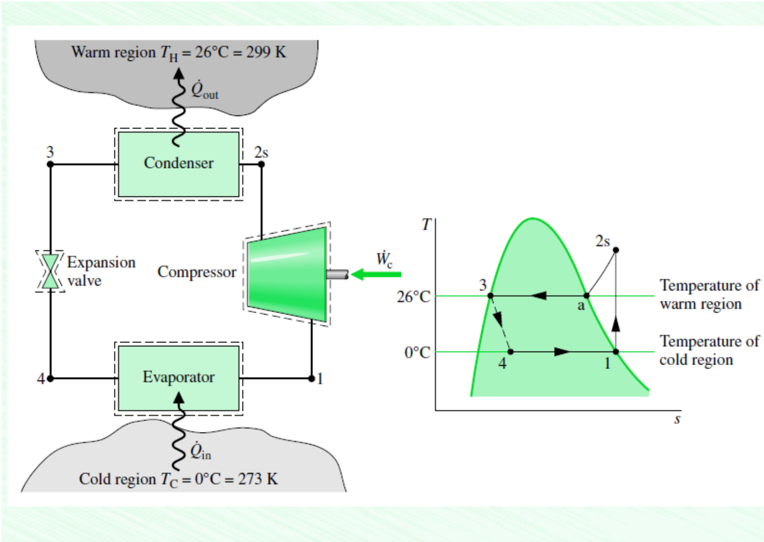
Solving, $\dot{m}_7 = 39.762$ & $h_5 = h_{10} = 85.69$ kJ/kg

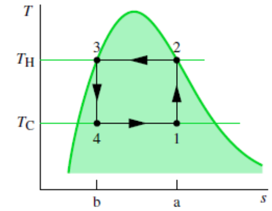
Then, $\dot{Q}_L = 39.762(83.96 - 42.01) = 1668 \text{ kW}$

$h_{11} = 85.69 + 0.47 = 86.16 \text{ kJ/kg}$

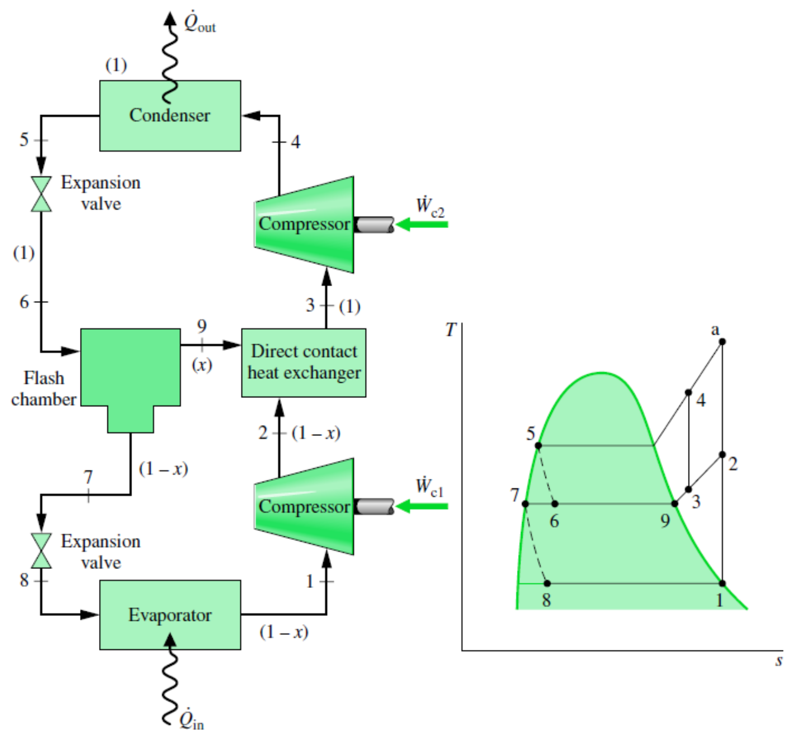
$\dot{Q}_H = 1(2746.5 - 86.16) = 2660.3 \text{ kW}$

& $\dot{Q}_L/\dot{Q}_H = 1668/2660.3 =$ **0.627**



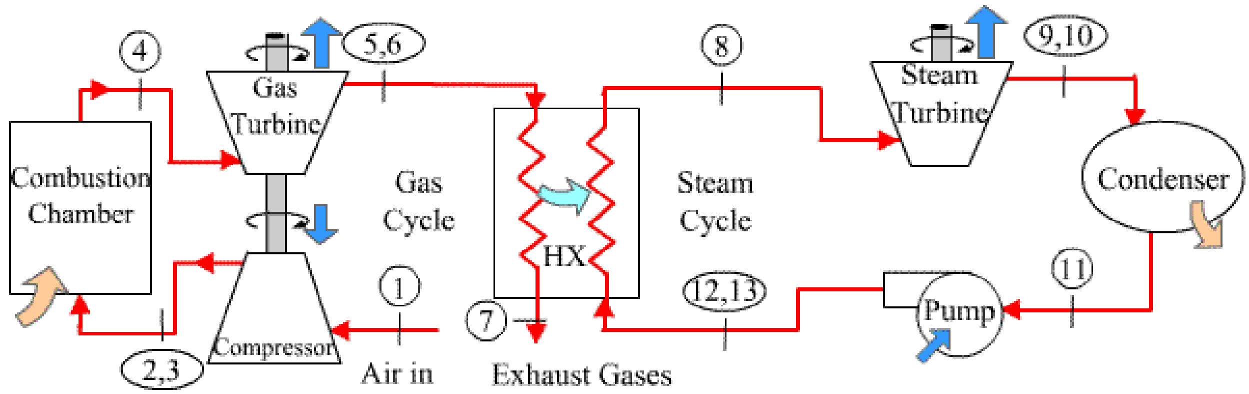


on cycle.



▲ **Figure 10.8** Refrigeration cycle with two stages of compression and flash intercooling.

EXAMPLE *E9-9* A combined gas turbine-steam power plant has a net power output of 500 MW. Air enters the compressor of the gas turbine at 100 kPa, 300 K, and has a compression ratio of 12 and an isentropic efficiency of 85%. The turbine has an isentropic efficiency of 90%, inlet conditions of 1200 kPa and 1400 K, and an exit pressure of 100 kPa. The air from the turbine exhaust passes through a heat exchanger and exits at 400 K. On the steam turbine side, steam at 8 MPa, 400°C enters the turbine, which has an isentropic efficiency of 85%, and expands to the condenser pressure of 8 kPa. Saturated water at 8 kPa is circulated back to the heat exchanger by a pump with an isentropic efficiency of 80%. Determine (a) the ratio of mass flow rates in the two cycles, (b) the mass flow rate of air, and (c) the thermal efficiency. (d) What-if-Scenario: What would the thermal efficiency be if the turbine inlet temperature increased to 1600 K? [Manual Solution] [TEST Solution]

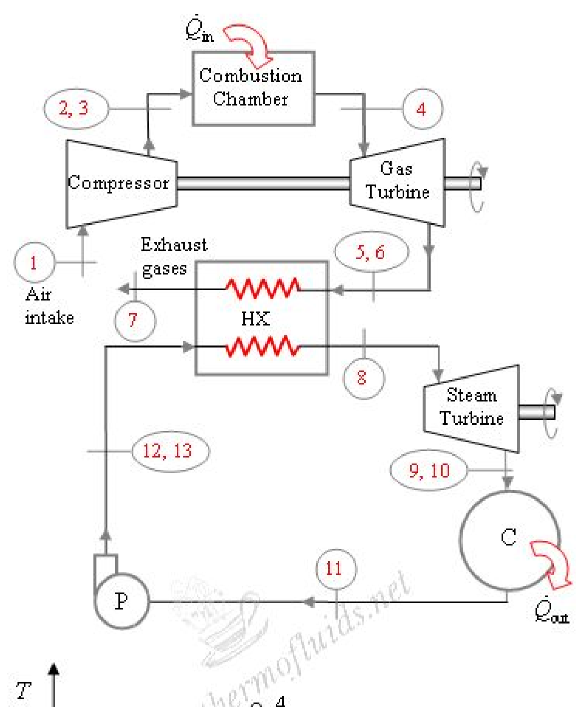


SOLUTION Use the IG and the PC models to evaluate the principal states of the two cycles depicted in Fig. 9.26 and perform energy analysis of each component for a complete solution of the problem.

Assumptions No frictional pressure drop in any component or piping. No stray heat losses from any device. Negligible changes in ke and pe across any device so that $j \cong h$. Air-standard Brayton cycle and Rankine cycle to model the two cycles. Use variable specific heat for air, i.e., the IG model for the Brayton cycle.

Analysis Use the manual approach or the PC/IG power cycle daemon to evaluate the enthalpies of each principal state. Steady-state energy balance for individual devices is employed to develop enthalpy relations as listed in the table below.

	Given	h (kJ/kg)		Given	h (kJ/kg)
1	p_1, T_1	1.9	**8**	p_8, T_8	3138.2
2	p_2, $s_2 = s_1$	311.7	**9**	p_9, $s_9 = s_8$	1990.0
3	$p_3 = p_2$, $h_3 = h_1 + (h_2 - h_1)/\eta_C$	366.5	**10**	$p_{10} = p_9$, $h_{10} = h_8 + (h_8 - h_9)\eta_T$	2162.2
4	$p_4 = p_2$, T_4	1217.8	**11**	$p_{11} = p_9$, $x_{11} = 0$	173.9



MANUAL SOLUTION OF EXAMPLE 9-9

5	$p_5 = p_1,\ s_5 = s_4$	471.1	**12**	$p_{12} = p_8,\ s_{12} = s_{11}$	181.9
6	$p_6 = p_5,$ $h_6 = h_4 + (h_4 - h_5)\eta_T$	545.8	**13**	$p_{13} = p_{12},$ $h_{13} = h_{11} + (h_{12} - h_{11})/\eta_P$	183.9
7	$p_7 = p_6,\ T_7$	103.0			

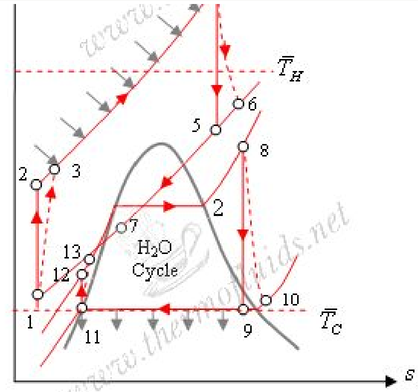


Fig. 9.26 A combined gas-vapor power cycle analyzed in Ex. 9-9 (see Anim. 9.A.*combinedCycle*).

An energy balance on the adiabatic heat exchanger produces

$$\dot{m}_1(h_6 - h_7) \cong \dot{m}_8(h_8 - h_{13});$$

$$\Rightarrow \quad \dot{m}_1 = \frac{h_8 - h_{13}}{h_6 - h_7}\dot{m}_8 = \frac{3138.2 - 183.9}{545.8 - 103.0}\dot{m}_8 = 6.673\dot{m}_8$$

With this relation, the net power output can be written as

$$\dot{W}_{net} = \dot{W}_{turb,I} + \dot{W}_{turb,II} - \dot{W}_{comp} - \dot{W}_{pump}$$
$$\cong \dot{m}_1(h_4 - h_6) + \dot{m}_8(h_8 - h_{10})$$
$$-\dot{m}_1(h_3 - h_1) - \dot{m}_8(h_{13} - h_{11})$$
$$= 452.2\dot{m}_1$$

Since the net output is 500 MW, the mass flow rate of air is

$$-\dot{m}_1(h_3 - h_1) - \dot{m}_8(h_{13} - h_{11})$$
$$= 452.2\dot{m}_1$$

Since the net output is 500 MW, the mass flow rate of air is

$$\dot{m}_1 = 500{,}000/452.2 = 1107 \text{ kg/s};$$

To obtain the thermal efficiency the external heat addition $\dot{Q}_{in}$ is evaluated first.

$$\dot{Q}_{in} \cong \dot{m}_1(h_4 - h_3) = 1107(1217.8 - 366.5) = 942.39 \text{ MW};$$

Therefore, $\eta_{th} = \dfrac{\dot{W}_{net}}{\dot{Q}_{in}} = \dfrac{500}{942.4} = 53.1\,\%$