Chapter 8

POWER AND REFRIGERATION CYCLES

Actual and Ideal Cycles, Carnot cycle, Air-Standard Assumptions

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8-1C The Carnot cycle is not suitable as an ideal cycle for all power producing devices because it cannot be approximated using the hardware of actual power producing devices.

8-2C It is less than the thermal efficiency of a Carnot cycle.

8-3C It represents the net work on both diagrams.

8-4C The cold air standard assumptions involves the additional assumption that air can be treated as an ideal gas with constant specific heats at room temperature.

8-5C Under the air standard assumptions, the combustion process is modeled as a heat addition process, and the exhaust process as a heat rejection process.

8-6C The air standard assumptions are: (1) the working fluid is air which behaves as an ideal gas, (2) all the processes are internally reversible, (3) the combustion process is replaced by the heat addition process, and (4) the exhaust process is replaced by the heat rejection process which returns the working fluid to its original state.

8-7C The clearance volume is the minimum volume formed in the cylinder whereas the displacement volume is the volume displaced by the piston as the piston moves between the top dead center and the bottom dead center.

8-8C It is the ratio of the maximum to minimum volumes in the cylinder.

8-9C The MEP is the fictitious pressure which, if acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle.

8-10C Yes.

8-11C Assuming no accumulation of carbon deposits on the piston face, the compression ratio will remain the same (otherwise it will increase). The mean effective pressure, on the other hand, will decrease as a car gets older as a result of wear and tear.

8-12C The SI and CI engines differ from each other in the way combustion is initiated; by a spark in SI engines, and by compressing the air above the self-ignition temperature of the fuel in CI engines.

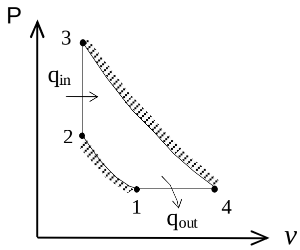
8-13C Stroke is the distance between the TDC and the BDC, bore is the diameter of the cylinder, TDC is the position of the piston when it forms the smallest volume in the cylinder, and clearance volume is the minimum volume formed in the cylinder.

8-14 The four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the net work output and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.

Analysis (*b*) The properties of air at various states are



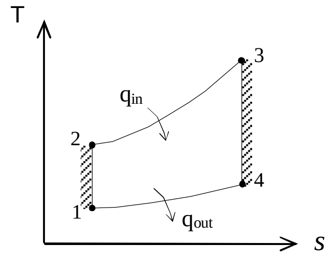
$$T_1 = 300\text{K} \longrightarrow \begin{array}{l} h_1 = 300.19\text{kJ/kg} \\ P_{r_1} = 1.386 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{800\text{kPa}}{100\text{kPa}} (1.386) = 11.088 \longrightarrow \begin{array}{l} u_2 = 389.22\text{kJ/kg} \\ T_2 = 539.8\text{K} \end{array}$$

$$T_3 = 1800\text{K} \longrightarrow \begin{array}{l} u_3 = 1487.2\text{kJ/kg} \\ P_{r_3} = 1310 \end{array}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2} P_2 = \frac{1800\text{K}}{539.8\text{K}} (800\text{kPa}) = 2668\text{kPa}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \frac{100\text{kPa}}{2668\text{kPa}} (1310) = 49.10 \longrightarrow h_4 = 828.1\text{kJ/kg}$$



From energy balances,

$$q_{in} = u_3 - u_2 = 1487.2 - 389.2 = 1098.0 \text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 = 828.1 - 300.19 = 527.9 \text{ kJ/kg}$$

$$w_{net,out} = q_{in} - q_{out} = 1098.0 - 527.9 = \mathbf{570.1 \text{ kJ/kg}}$$

(*c*) Then the thermal efficiency becomes

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{570.1 \text{ kJ/kg}}{1098.0 \text{ kJ/kg}} = \mathbf{51.9\%}$$

8-15 Problem 8-14 is reconsidered. The effect of varying the temperature after the constant volume heat addition from 1500 K to 2500 K is to be investigated. The net work output and thermal efficiency are to be plotted as a function of the maximum temperature of the cycle as well as the T-s and P-v diagrams for the cycle when the maximum temperature of the cycle is 1800 K.

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"We assume that this ideal gas cycle takes place in a piston-cylinder device;
therefore, we will use a closed system analysis."
"See the T-s diagram in Plot Window1 and the P-v diagram in Plot Window2"

"Input Data"
T[1]=300"K"
P[1]=100"kPa"
P[2] = 800"[kPa]"
T[3]=1800"K"
P[4] = 100 "[kPa]"

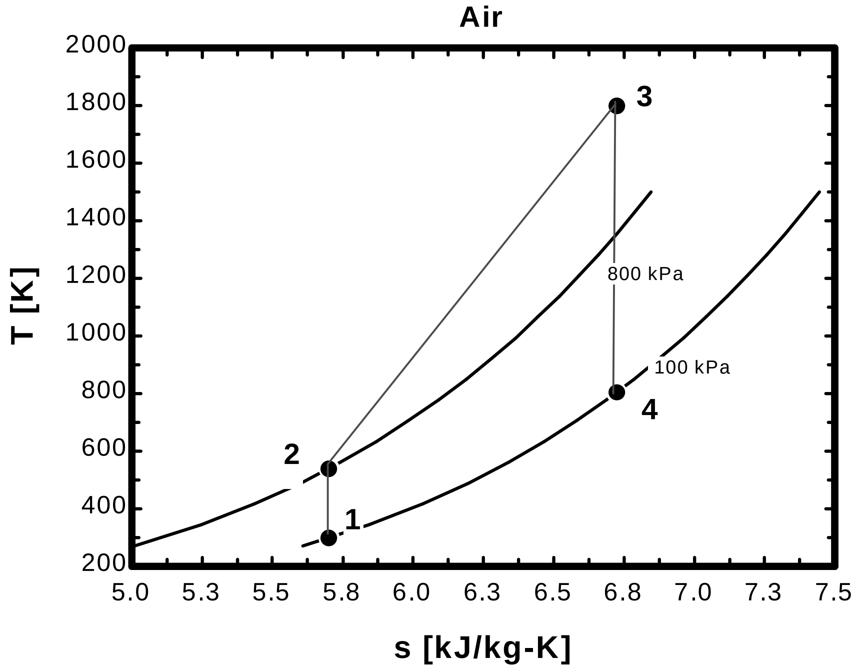
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=0.287*T[1]
"Conservation of energy for process 1 to 2"
q_12 -w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

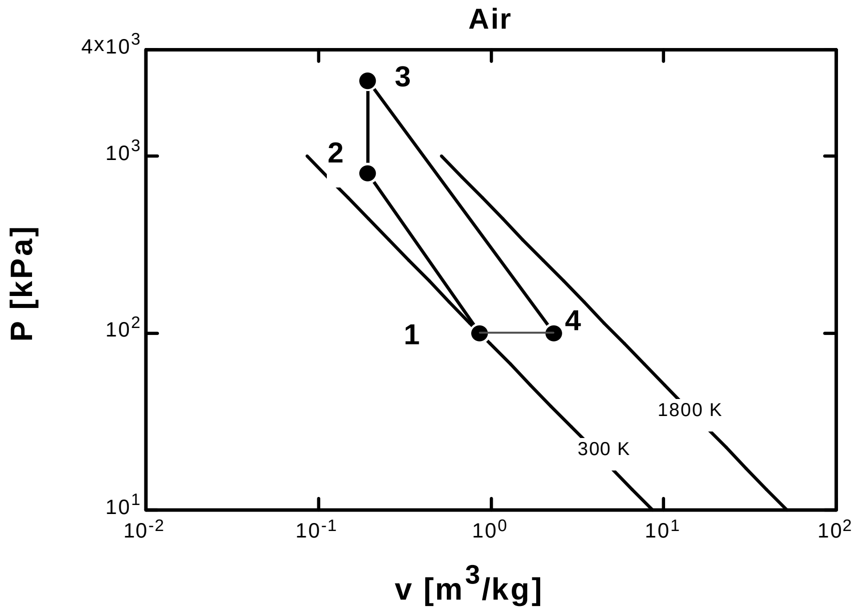
"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=0.287*T[3]
v[3]=v[2]
"Conservation of energy for process 2 to 3"
q_23 -w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])

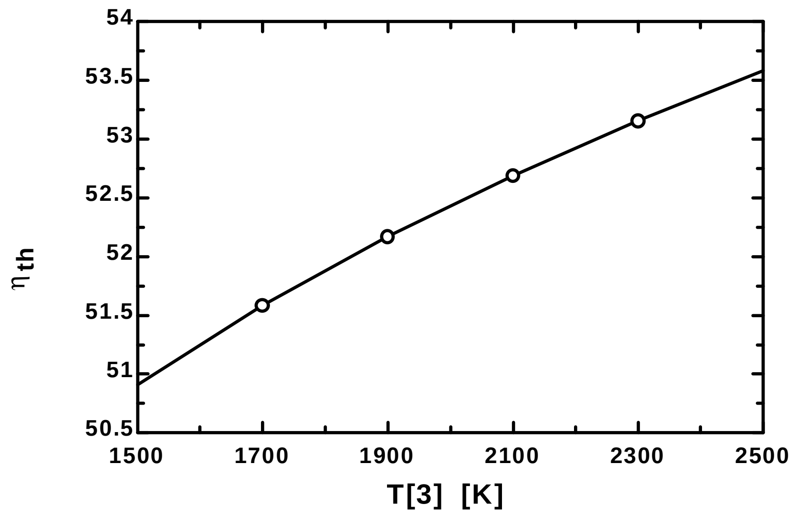
"Process 3-4 is isentropic expansion"
s[4]=entropy(air,T=T[4],P=P[4])
s[4]=s[3]
P[4]*v[4]/T[4]=P[3]*v[3]/T[3]
{P[4]*v[4]=0.287*T[4]}
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])

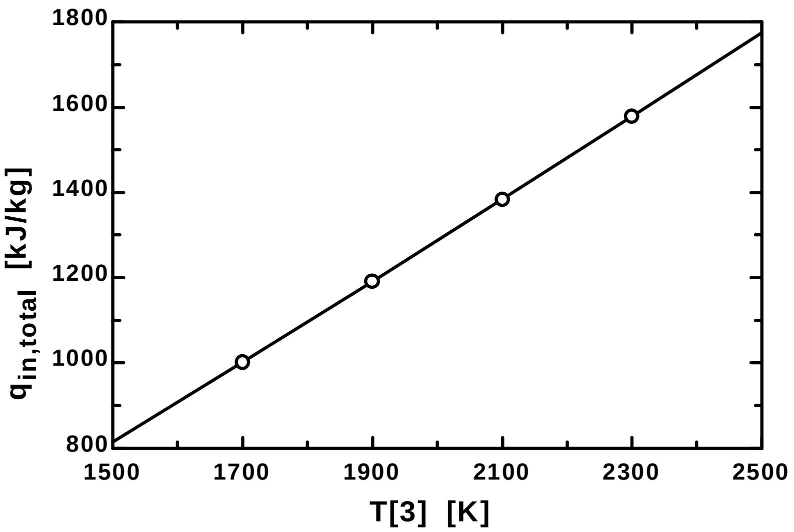
"Process 4-1 is constant pressure heat rejection"
{P[4]*v[4]/T[4]=P[1]*v[1]/T[1]}
"Conservation of energy for process 4 to 1"
q_41 -w_41 = DELTAu_41
w_41 =P[1]*(v[1]-v[4])        "constant pressure process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
q_in_total=q_23
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
```

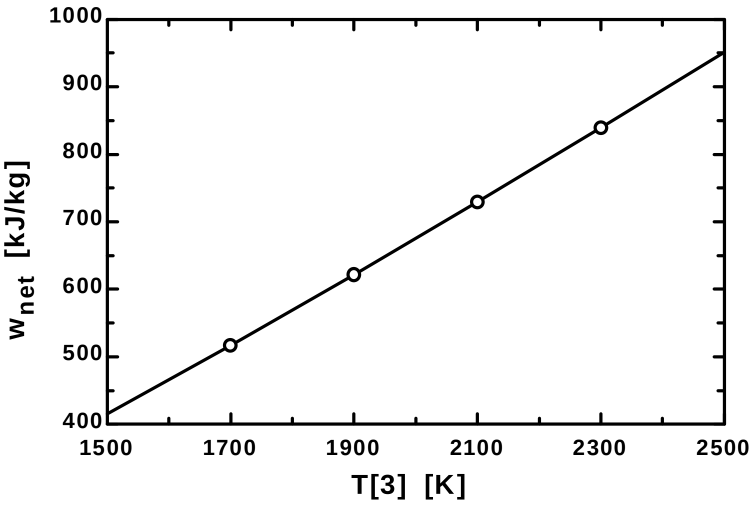
η_{th}	$q_{in,total}$ [kJ/kg]	w_{inet} [kJ/kg]	T_3 [K]
50.91	815.4	415.1	1500
51.58	1002	516.8	1700
52.17	1192	621.7	1900
52.69	1384	729.2	2100
53.16	1579	839.1	2300
53.58	1775	951.2	2500






54
53.5
53
52.5
52
51.5
51
50.5
ηth
1500
1700
1900
2100
2300
2500
T[3] [K]


1800
1600
1400
1200
1000
800
qin,total [kJ/kg]
1500
1700
1900
2100
2300
2500
T[3] [K]

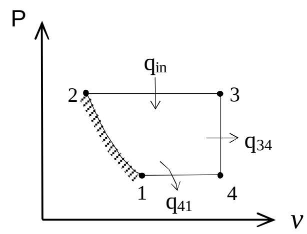

1000
900
800
700
600
500
400
wnet [kJ/kg]
1500
1700
1900
2100
2300
2500
T[3] [K]

8-16 The four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the maximum temperature in the cycle and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*b*) From the ideal gas isentropic relations and energy balance,



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{K})\left(\frac{1000\text{kPa}}{100\text{kPa}}\right)^{0.4/1.4} = 579.2\text{K}$$

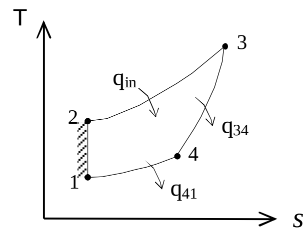
$$q_{in} = h_3 - h_2 = C_p(T_3 - T_2)$$

or,

$$2800 \text{ kJ/kg} = (1.005 \text{ kJ/kg}\cdot\text{K})(T_3 - 579.2)$$

$$T_{\max} = T_3 = \mathbf{3360\ \ K}$$

(*c*) $$\frac{P_3 v_3}{T_3} = \frac{P_4 v_4}{T_4} \longrightarrow T_4 = \frac{P_4}{P_3}T_3 = \frac{100 \text{ kPa}}{1000 \text{ kPa}}(3360 \text{ K}) = 336 \text{ K}$$



$$\begin{aligned} q_{out} &= q_{34,out} + q_{41,out} = (u_3 - u_4) + (h_4 - h_1) \\ &= C_v(T_3 - T_4) + C_p(T_4 - T_1) \\ &= (0.718 \text{ kJ/kg}\cdot\text{K})(3360 - 336)\text{K} + (1.005 \text{ kJ/kg}\cdot\text{K})(336 - 300)\text{K} \\ &= 2212 \text{ kJ/kg} \end{aligned}$$

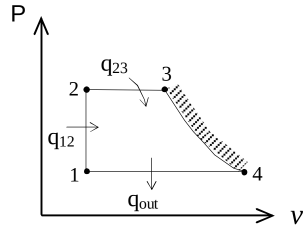
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{2212 \text{ kJ/kg}}{2800 \text{ kJ/kg}} = \mathbf{21.0\%}$$

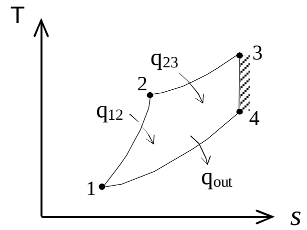
8-17E The four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the total heat input and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21E.

Analysis (*b*) The properties of air at various states are





$$T_1 = 540 \text{ R} \longrightarrow \begin{array}{l} u_1 = 92.04 \text{ Btu/lbm} \\ h_1 = 129.06 \text{ Btu/lbm} \end{array}$$

$$q_{\text{in},12} = u_2 - u_1 \longrightarrow \begin{array}{l} u_2 = u_1 + q_{\text{in},12} = 92.04 + 300 = 392.04 \text{Btu/lbm} \\ T_2 = 2116\text{R},\ h_2 = 537.1 \text{Btu/lbm} \end{array}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{2116\text{R}}{540\text{R}}(14.7\text{psia}) = 57.6\text{psia}$$

$$T_3 = 3200\text{R} \longrightarrow \begin{array}{l} h_3 = 849.48 \text{Btu/lbm} \\ P_{r_3} = 1242 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \frac{14.7\text{psia}}{57.6\text{psia}}(1242) = 317.0 \longrightarrow h_4 = 593.22 \text{Btu/lbm}$$

From energy balance,

$$q_{23,in} = h_3 - h_2 = 849.48 - 537.1 = 312.38 \text{ Btu/lbm}$$

$$q_{in} = q_{12,in} + q_{23,in} = 300 + 312.38 = \mathbf{612.38\ Btu/lbm}$$

$$q_{out} = h_4 - h_1 = 593.22 - 129.06 = 464.16 \text{ Btu/lbm}$$

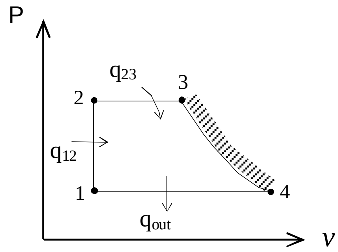
(*c*) Then the thermal efficiency becomes

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{464.16 \text{ Btu/lbm}}{612.38 \text{ Btu/lbm}} = \mathbf{24.2\%}$$

8-18E The four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the total heat input and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.171 Btu/lbm.R, and k = 1.4 (Table A-2E).

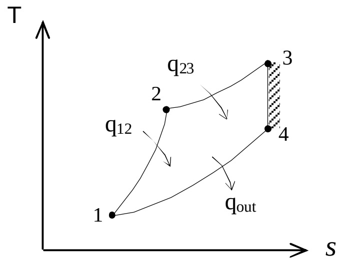


Analysis (*b*)

$$q_{in,12} = u_2 - u_1 = C_v(T_2 - T_1)$$
$$300\text{Btu/lbm} = (0.171\text{Btu/lbm.R})(T_2 - 540)\text{R}$$
$$T_2 = 2294\text{R}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{2294\text{R}}{540\text{R}}(14.7\text{psia}) = 62.46\text{psia}$$
$$q_{in,23} = h_3 - h_2 = C_P(T_3 - T_2) = (0.24\text{Btu/lbm}\cdot\text{R})(3200 - 2294)\text{R} = 217.4\text{Btu/lbm}$$

Process 3-4 is isentropic:



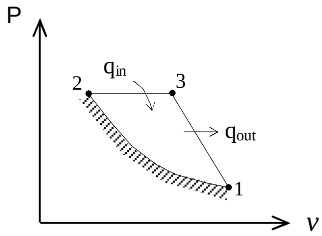
$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (3200\text{R})\left(\frac{14.7\text{psia}}{62.46\text{psia}}\right)^{0.4/1.4} = 2117\text{R}$$
$$q_{in} = q_{in,12} + q_{in,23} = 300 + 217.4 = \mathbf{517.4 Btu/lbm}$$
$$q_{out} = h_4 - h_1 = C_p(T_4 - T_1) = (0.240\text{Btu/lbm.R})(2117 - 540)$$
$$= 378.5\text{Btu/lbm}$$

(c) $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{378.5\ \text{Btu/lbm}}{517.4\ \text{Btu/lbm}} = \mathbf{26.8\%}$$

8-19 The three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the heat rejected and the thermal efficiency are to be determined. √

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).



Analysis (*b*) $$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{K})\left(\frac{1000\text{kPa}}{100\text{kPa}}\right)^{0.4/1.4} = 579.2\text{K}$$

$$Q_{in} = m(h_3 - h_2) = mC_p(T_3 - T_2)$$

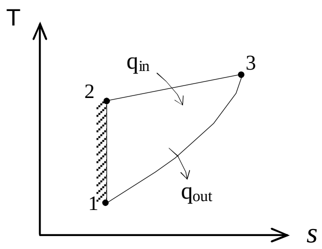
or,

$$2.76\text{ kJ} = (0.0015\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(T_3 - 579.2) \longrightarrow T_3 = 2410\text{ K}$$

Process 3-1 is a straight line on the P-*v* diagram, thus the w_{31} is simply the area under the process curve,

$$w_{31} = area = \frac{P_3 + P_1}{2}(v_1 - v_3) = \frac{P_3 + P_1}{2}\left(\frac{RT_1}{P_1} - \frac{RT_3}{P_3}\right)$$

$$= \left(\frac{1000 + 100\text{kPa}}{2}\right)\left(\frac{300\text{K}}{100\text{kPa}} - \frac{2410\text{K}}{1000\text{kPa}}\right)(0.287\text{kJ/kg}\cdot\text{K}) = 93.1\text{kJ/kg}$$



Energy balance for process 3-1 gives

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$-Q_{31,out} - W_{31,out} = m(u_1 - u_3)$$

$$Q_{31,out} = -mw_{31,out} - mC_v(T_1 - T_3) = -m\left[w_{31,out} + C_v(T_1 - T_3)\right]$$

$$= -(0.0015\text{ kg})\left[93.1 + (0.718\text{kJ/kg}\cdot\text{K})(300 - 2410)\text{K}\right] = \mathbf{2.133\ \ kJ}$$

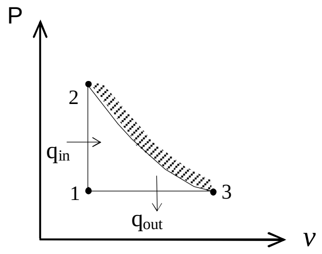
(*c*) $$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{2.133\text{ kJ}}{2.76\text{ kJ}} = \mathbf{22.7\%}$$

8-20 The three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the net work per cycle and the thermal efficiency are to be determined. √

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.

Analysis (*b*) The properties of air at various states are

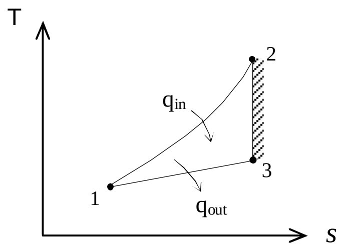


$$T_1 = 290K \longrightarrow \begin{array}{l} u_1 = 206.91\text{kJ/kg} \\ h_1 = 290.16\text{kJ/kg} \end{array}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = \frac{380\text{kPa}}{95\text{kPa}}(290\text{K}) = 1160\text{K}$$

$$\longrightarrow u_2 = 897.91\text{kJ/kg},\ P_{r_2} = 207.2$$

$$P_{r_3} = \frac{P_3}{P_2}P_{r_2} = \frac{95\text{kPa}}{380\text{kPa}}(207.2) = 51.8 \longrightarrow h_3 = 840.38\text{kJ/kg}$$



$$Q_{in} = m(u_2 - u_1) = (0.003\text{kg})(897.91 - 206.91)\text{kJ/kg} = 2.073\text{kJ}$$

$$Q_{out} = m(h_3 - h_1) = (0.003\text{kg})(840.38 - 290.16)\text{kJ/kg} = 1.651\text{kJ}$$

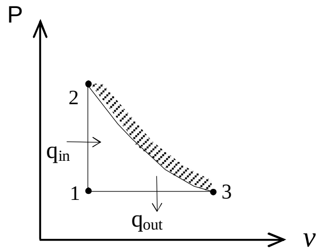
$$W_{net,out} = Q_{in} - Q_{out} = 2.073 - 1.651 = \mathbf{0.422kJ}$$

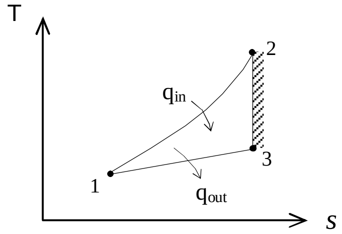
(c) $$\eta_{th} = \frac{W_{net,out}}{Q_{in}} = \frac{0.422\text{ kJ}}{2.073\text{ kJ}} = \mathbf{20.4\%}$$

8-21 The three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the net work per cycle and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).





Analysis (*b*) From the isentropic relations and energy balance,

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{380\text{kPa}}{95\text{kPa}}(290\text{K}) = 1160\text{K}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (1160\text{K})\left(\frac{95\text{kPa}}{380\text{kPa}}\right)^{0.4/1.4} = 780.6\text{K}$$

$$Q_{in} = m(u_2 - u_1) = mC_v(T_2 - T_1)$$

$$= (0.003\text{kg})(0.718\text{kJ/kg}\cdot\text{K})(1160 - 290)\text{K} = 1.87\text{kJ}$$

$$Q_{out} = m(h_3 - h_1) = mC_p(T_3 - T_1)$$

$$= (0.003\text{kg})(1.005\text{kJ/kg}\cdot\text{K})(780.6 - 290)\text{K} = 1.48\text{kJ}$$

$$W_{net,out} = Q_{in} - Q_{out} = 1.87 - 1.48 = \mathbf{0.39kJ}$$

(*c*) $$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{0.39\text{ kJ}}{1.87\text{ kJ}} = \mathbf{20.9\%}$$

8-22 A Carnot cycle with the specified temperature limits is considered. The net work output per cycle is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

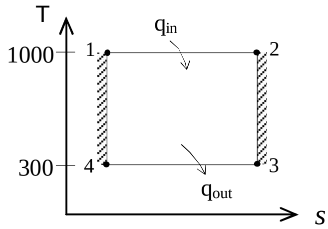
Properties The properties of air at room temperature are C_p = 1.005 kJ/kg.K, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis The minimum pressure in the cycle is P_3 and the maximum pressure is P_1. Then,

$$\frac{T_2}{T_3} = \left(\frac{P_2}{P_3}\right)^{(k-1)/k}$$

or,

$$P_2 = P_3\left(\frac{T_2}{T_3}\right)^{k/(k-1)} = (20\text{kPa})\left(\frac{1000\text{K}}{300\text{K}}\right)^{1.4/0.4} = 1352\text{kPa}$$



The heat input is determined from

$$s_2 - s_1 = C_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -(0.287\text{kJ/kg}\cdot\text{K})\ln\frac{1352\text{ kPa}}{1800\text{ kPa}} = 0.08205\text{ kJ/kg}\cdot\text{K}$$

$$Q_{in} = mT_H(s_2 - s_1) = (0.003\text{ kg})(1000\text{ K})(0.08205\text{ kJ/kg}\cdot\text{K}) = 0.246\text{ kJ}$$

Then,

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{1000\text{ K}} = 70.0\%$$

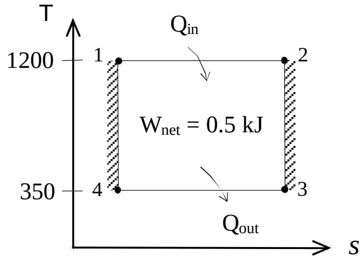
$$W_{net,out} = \eta_{th}Q_{in} = (0.70)(0.246\text{ kJ}) = \mathbf{0.172\ \ kJ}$$

8-23 A Carnot cycle with specified temperature limits is considered. The maximum pressure in the cycle, the heat transfer to the working fluid, and the mass of the working fluid are to be determined. √

Assumptions Air is an ideal gas with variable specific heats.

Analysis (*a*) In a Carnot cycle, the maximum pressure occurs at the beginning of the expansion process, which is state 1.

$$T_1 = 1200 \text{ K} \longrightarrow P_{r_1} = 238$$
$$T_4 = 350 \text{ K} \longrightarrow P_{r_4} = 2.379 \qquad \text{(Table A-21)}$$



$$P_1 = \frac{P_{r_1}}{P_{r_4}} P_4 = \frac{238}{2.379}(300\text{kPa}) = \mathbf{30{,}013\,kPa} = P_{\max}$$

(*b*) The heat input is determined from

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{350\text{K}}{1200\text{K}} = 70.83\%$$

$$Q_{in} = W_{net,out} / \eta_{th} = (0.5\text{kJ})/(0.7083) = \mathbf{0.706\,kJ}$$

(*c*) The mass of air is

$$s_4 - s_3 = \left(s_4^0 - s_3^0\right)^{\text{Ã}0} - R\ln\frac{P_4}{P_3} = -(0.287\text{kJ/kg}\cdot\text{K})\ln\frac{300\text{kPa}}{150\text{kPa}}$$
$$= -0.199\text{kJ/kg}\cdot K = s_1 - s_2$$

$$w_{net,out} = (s_2 - s_1)(T_H - T_L) = (0.199\text{kJ/kg}\cdot\text{K})(1200 - 350)K = 169.15\text{kJ/kg}$$

$$m = \frac{W_{net,out}}{w_{net,out}} = \frac{0.5\text{kJ}}{169.15\text{kJ/kg}} = \mathbf{0.00296\,kg}$$

8-24 A Carnot cycle with specified temperature limits is considered. The maximum pressure in the cycle, the heat transfer to the working fluid, and the mass of the working fluid are to be determined.

Assumptions Helium is an ideal gas with constant specific heats.

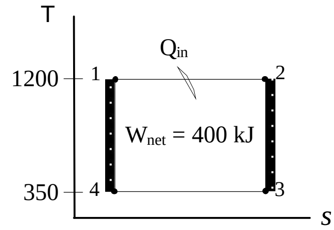
Properties The properties of helium at room temperature are R = 2.0769 kJ/kg.K and k = 1.667 (Table A-2).

Analysis (a) In a Carnot cycle, the maximum pressure occurs at the beginning of the expansion process, which is state 1.

$$\frac{T_1}{T_4} = \left(\frac{P_1}{P_4}\right)^{(k-1)/k}$$

or,

$$P_1 = P_4\left(\frac{T_1}{T_4}\right)^{k/(k-1)} = (300\text{kPa})\left(\frac{1200\text{K}}{350\text{K}}\right)^{1.667/0.667} = 6524\text{kPa}$$



(b) The heat input is determined from

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{350\text{K}}{1200\text{K}} = 70.83\%$$

$$Q_{in} = W_{net,out}/\eta_{th} = (0.5\text{kJ})/(0.7083) = \mathbf{0.706 kJ}$$

(c) The mass of helium is determined from

$$s_4 - s_3 = C_p \ln\frac{T_4}{T_3}^{\nearrow 0} - R\ln\frac{P_4}{P_3} = -(2.0769\text{ kJ/kg}\cdot\text{K})\ln\frac{300\text{ kPa}}{150\text{ kPa}}$$

$$= -1.4396\text{ kJ/kg}\cdot\text{K} = s_1 - s_2$$

$$w_{net,out} = (s_2 - s_1)(T_H - T_L) = (1.4396\text{ kJ/kg}\cdot\text{K})(1200-350)\text{K} = 1223.7\text{ kJ/kg}$$

$$m = \frac{W_{net,out}}{w_{net,out}} = \frac{0.5\text{ kJ}}{1223.7\text{ kJ/kg}} = \mathbf{0.000409\ kg}$$

Otto Cycle

8-25C The four processes that make up the Otto cycle are (1) isentropic compression, (2) v = constant heat addition, (3) isentropic expansion, and (4) v = constant heat rejection.

8-26C The ideal Otto cycle involves external irreversibilities, and thus it has a lower thermal efficiency.

8-27C For actual four-stroke engines, the rpm is twice the number of thermodynamic cycles; for two-stroke engines, it is equal to the number of thermodynamic cycles.

8-28C They are analyzed as closed system processes because no mass crosses the system boundaries during any of the processes.

8-29C It increases with both of them.

8-30C Because high compression ratios cause engine knock.

8-31C The thermal efficiency will be the highest for argon because it has the highest specific heat ratio, k = 1.667.

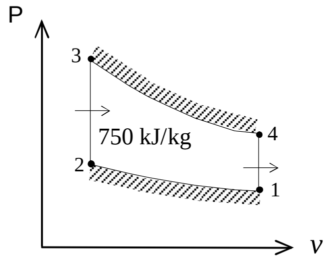
8-32C The fuel is injected into the cylinder in both engines, but it is ignited with a spark plug in gasoline engines.

8-33 An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The pressure and temperature at the end of the heat addition process, the net work output, the thermal efficiency, and the mean effective pressure for the cycle are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.

Analysis (*a*) Process 1-2: isentropic compression.



$$T_1 = 300\text{ K} \longrightarrow \begin{array}{l} u_1 = 214.07\text{ kJ/kg} \\ v_{r_1} = 621.2 \end{array}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{8}(621.2) = 77.65$$

$$\longrightarrow \begin{array}{l} T_2 = 673.1\text{K} \\ u_2 = 491.2\text{kJ/kg} \end{array}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1} P_1 = (8)\left(\frac{673.1\text{K}}{300\text{K}}\right)(95\text{kPa}) = 1705\text{kPa}$$

Process 2-3: v = constant heat addition.

$$q_{23,\text{in}} = u_3 - u_2 \longrightarrow u_3 = u_2 + q_{23,in} = 491.2 + 750 = 1241.2\text{ kJ/kg} \longrightarrow \begin{array}{l} T_3 = \mathbf{1539\ K} \\ v_{r_3} = 6.588 \end{array}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2} P_2 = \left(\frac{1539\text{K}}{673.1\text{K}}\right)(1705\text{kPa}) = \mathbf{3898kPa}$$

(*b*) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_1}{v_2} v_{r_3} = r v_{r_3} = (8)(6.588) = 52.70 \longrightarrow \begin{array}{l} T_4 = 774.5\text{K} \\ u_4 = 571.69\text{kJ/kg} \end{array}$$

Process 4-1: v = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 571.69 - 214.07 = 357.62\text{ kJ/kg}$$

$$w_{net,out} = q_{in} - q_{out} = 750 - 357.62 = \mathbf{392.38\ kJ/kg}$$

(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{392.38\text{kJ/kg}}{750\text{kJ/kg}} = \mathbf{52.3\%}$$

(*d*) $$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{95\text{kPa}} = 0.906\text{m}^3/\text{kg} = \text{v}_{\text{max}}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$MEP = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1 - 1/r)} = \frac{392.38\text{kJ/kg}}{(0.906\text{m}^3/\text{kg})(1 - 1/8)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{495.0kPa}$$

8-34 Problem 8-33 is reconsidered. The effect of varying the compression ratio from 5 to 10 is to be investigated. The net work output and thermal efficiency are to be plotted as a function of the compression ratio. Also, the T-s and P-v diagrams for the cycle are to be plotted when the compression ratio is 8.

```
"We assume that this ideal gas cycle takes place in a piston-cylinder device;
therefore, we will use a closed system analysis."
"See the T-s diagram in Plot Window1 and the P-v diagram in Plot Window2"

"Input Data"
T[1]=300"K"
P[1]=95"kPa"
q_23 = 750 "[kJ/kg]"
{r_comp = 8}

"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=0.287*T[1]
V[2] = V[1]/ r_comp

"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

"Process 2-3 is constant volume heat addition"
v[3]=v[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=0.287*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])

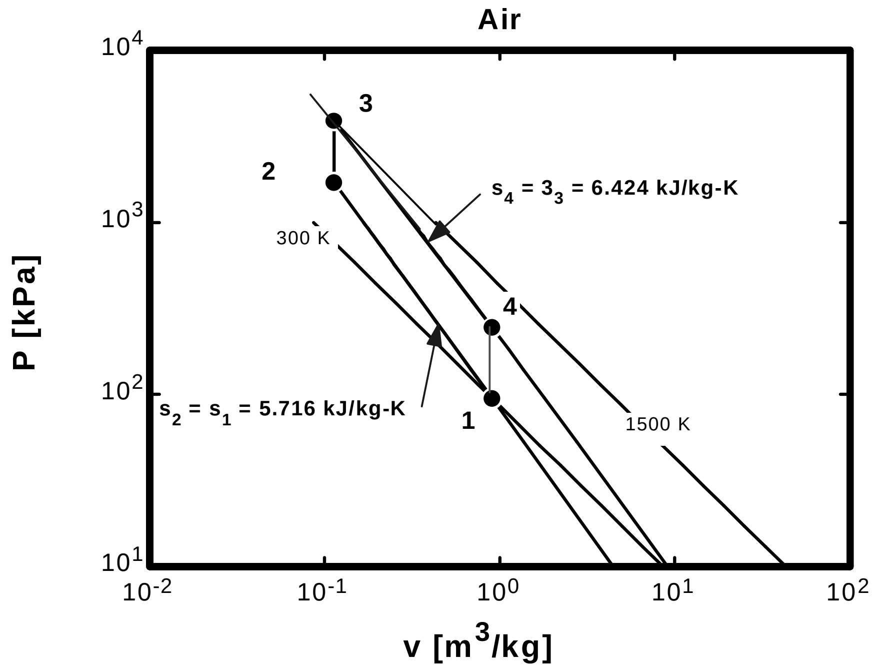
"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=0.287*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0      "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])

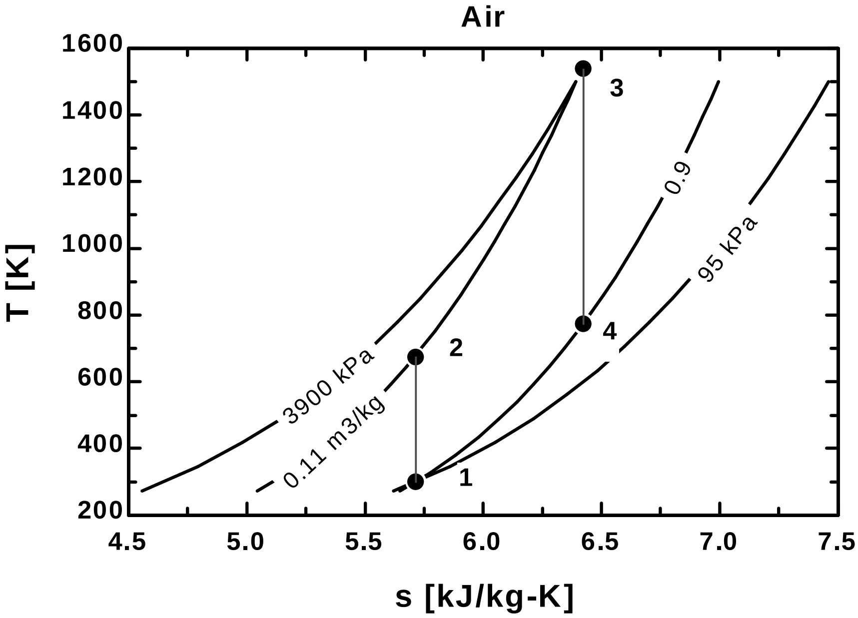
q_in_total=q_23
q_out_total = -q_41
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"

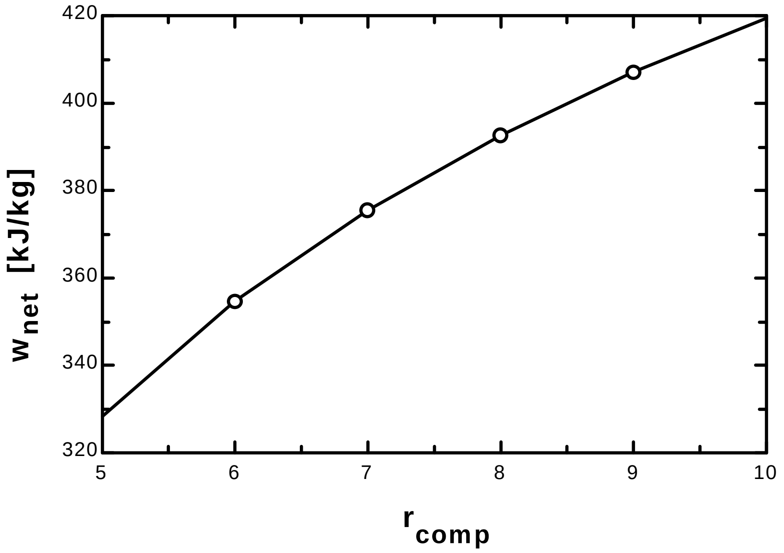
"The mean effective pressure is:"
```

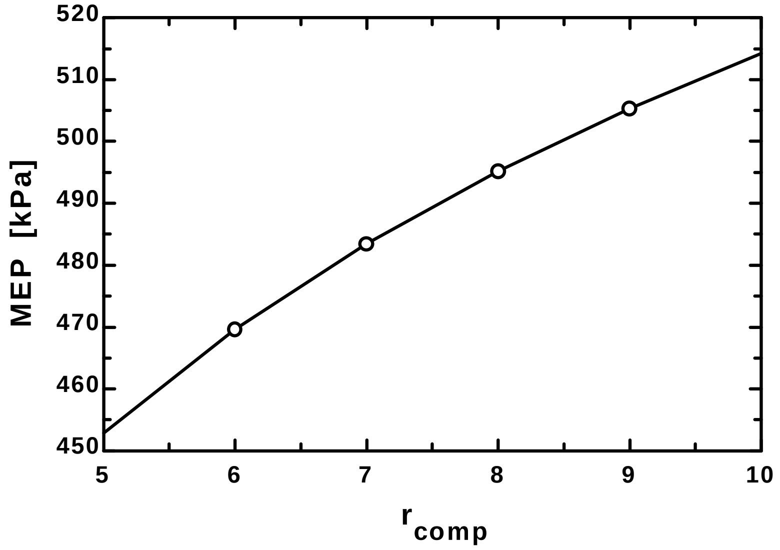
MEP = w_net/(V[1]-V[2])"[kPa]"

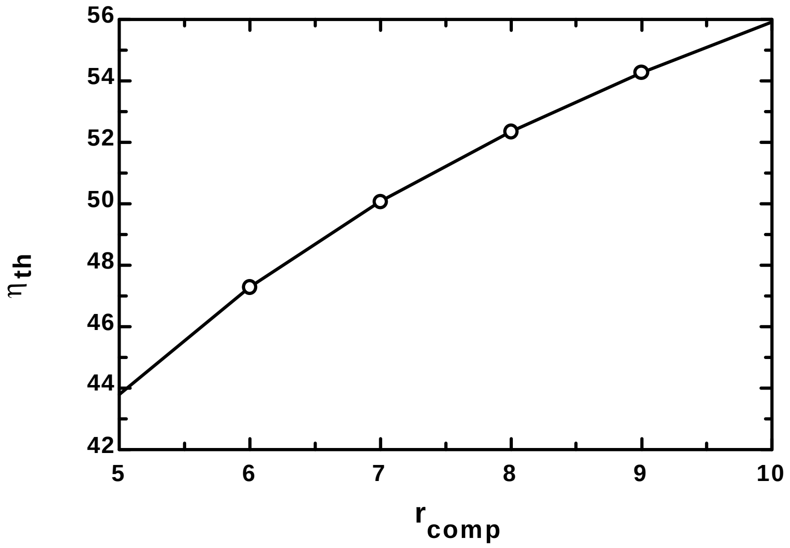
η_{th}	MEP [kPa]	r_{comp}	w_{net} [kJ/kg]
43.78	452.9	5	328.4
47.29	469.6	6	354.7
50.08	483.5	7	375.6
52.36	495.2	8	392.7
54.28	505.3	9	407.1
55.93	514.2	10	419.5






420
400
380
360
340
320
5
6
7
8
9
10
w net [kJ/kg]
r comp


520
510
500
490
480
470
460
450
5
6
7
8
9
10
MEP [kPa]
r comp

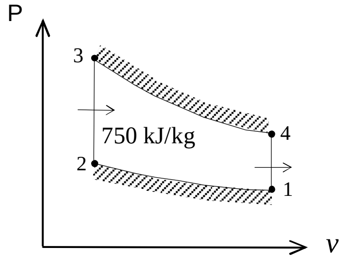

56
54
52
50
48
46
44
42
5
6
7
8
9
10
η th
r comp

8-35 An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The pressure and temperature at the end of the heat addition process, the net work output, the thermal efficiency, and the mean effective pressure for the cycle are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*a*) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\text{K})(8)^{0.4} = 689\text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (8)\left(\frac{689\text{K}}{300\text{K}}\right)(95\text{kPa}) = 1745\text{kPa}$$

Process 2-3: v = constant heat addition.

$$q_{23,\text{in}} = u_3 - u_2 = C_v(T_3 - T_2)$$

$$750\text{kJ/kg} = (0.718\text{kJ/kg}\cdot\text{K})(T_3 - 689)\text{K}$$

$$T_3 = \mathbf{1734K}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1734\text{K}}{689\text{K}}\right)(1745\text{kPa}) = \mathbf{4392kPa}$$

(*b*) Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1734\text{K})\left(\frac{1}{8}\right)^{0.4} = 755\text{K}$$

Process 4-1: v = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = C_v(T_4 - T_1) = (0.718\text{kJ/kg}\cdot\text{K})(755 - 300)\text{K} = 327\text{kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 750 - 327 = \mathbf{423kJ/kg}$$

(*c*)
$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{423\text{ kJ/kg}}{750\text{ kJ/kg}} = \mathbf{56.4\%}$$

(*d*)
$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{95\text{kPa}} = 0.906\text{m}^3/\text{kg} = \text{v}_{\text{max}}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

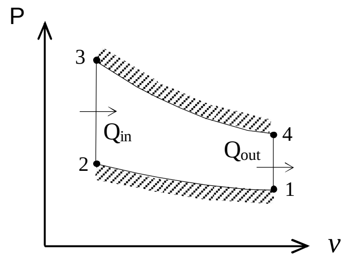
$$MEP = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1-1/r)} = \frac{423\text{kJ/kg}}{(0.906\text{m}^3/\text{kg})(1-1/8)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{534kPa}$$

8-36 An ideal Otto cycle with air as the working fluid has a compression ratio of 9.5. The highest pressure and temperature in the cycle, the amount of heat transferred, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*a*) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (290\text{K})(9.5)^{0.4} = 713.7\text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{713.7\text{K}}{290\text{K}}\right)(100\text{kPa}) = 2338\text{kPa}$$

Process 3-4: isentropic expansion.

$$T_3 = T_4\left(\frac{v_4}{v_3}\right)^{k-1} = (800\text{K})(9.5)^{0.4} = \mathbf{1969K}$$

Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1969\text{K}}{713.7\text{K}}\right)(2338\text{kPa}) = \mathbf{6449kPa}$$

(*b*) $$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{kPa})(0.0006\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\text{K})} = 7.21\times10^{-4}\,\text{kg}$$

$$Q_{\text{in}} = m(u_3 - u_2) = mC_v(T_3 - T_2) = (7.21\times10^{-4}\,\text{kg})(0.718\text{kJ/kg}\cdot\text{K})(1969 - 713.7)\text{K} = \mathbf{0.650kJ}$$

(c) Process 4-1: v = constant heat rejection.

$$Q_{\text{out}} = m(u_4 - u_1) = mC_v(T_4 - T_1) = -(7.21\times10^{-4}\,\text{kg})(0.718\text{kJ/kg}\cdot\text{K})(800 - 290)\text{K} = \mathbf{0.264kJ}$$

$$W_{\text{net}} = Q_{\text{in}} - Q_{\text{out}} = 0.650 - 0.264 = 0.386\text{ kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_{\text{in}}} = \frac{0.386\text{ kJ}}{0.650\text{ kJ}} = \mathbf{59.4\%}$$

(*d*) $$V_{\min} = V_2 = \frac{V_{\max}}{r}$$

$$MEP = \frac{W_{net,out}}{V_1 - V_2} = \frac{W_{net,out}}{V_1(1-1/r)} = \frac{0.386\text{kJ}}{(0.0006\text{m}^3)(1-1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{719kPa}$$

8-37 An Otto cycle with air as the working fluid has a compression ratio of 9.5. The highest pressure and temperature in the cycle, the amount of heat transferred, the thermal efficiency, and the mean effective pressure are to be determined.

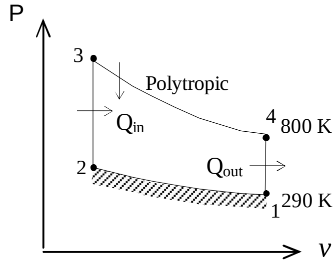
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are C_p = 1.005 kJ/kg·K, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*a*) Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (290\text{K})(9.5)^{0.4} = 713.7\text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{713.7\text{K}}{290\text{K}}\right)(100\text{kPa}) = 2338\text{kPa}$$



Process 3-4: polytropic expansion.

$$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(0.0006\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\text{ K})} = 7.209\times10^{-4}\text{ kg}$$

$$T_3 = T_4\left(\frac{v_4}{v_3}\right)^{n-1} = (800\text{K})(9.5)^{0.35} = \mathbf{1759\ K}$$

$$W_{34} = \frac{mR(T_4 - T_3)}{1-n} = \frac{(7.209\times10^{-4})(0.287\text{ kJ/kg}\cdot\text{K})(800-1759)\text{K}}{1-1.35} = 0.567\text{ kJ}$$

Then energy balance for process 3-4 gives

$$E_{in} - E_{out} = \Delta E_{system}$$

$$Q_{34,in} - W_{34,out} = m(u_4 - u_3)$$

$$Q_{34,in} = m(u_4 - u_3) + W_{34,out} = mC_v(T_4 - T_3) + W_{34,out}$$

$$Q_{34,in} = (7.209\times10^{-4}\text{ kg})(0.718\text{kJ/kg}\cdot\text{K})(800-1759)\text{K} + 0.567\text{kJ} = 0.071\text{kJ}$$

That is, 0.071 kJ of heat is added to the air during the expansion process (This is not realistic, and probably is due to assuming constant specific heats at room temperature).

(*b*) Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1759\text{K}}{713.7\text{K}}\right)(2338\text{kPa}) = \mathbf{5762kPa}$$

$$Q_{23,in} = m(u_3 - u_2) = mC_v(T_3 - T_2)$$

$$Q_{23,in} = (7.209\times10^{-4}\text{ kg})(0.718\text{kJ/kg}\cdot\text{K})(1759 - 713.7)K = \mathbf{0.541kJ}$$

Therefore,

$$Q_{in} = Q_{23,in} + Q_{34,in} = 0.541 + 0.071 = 0.612\text{ kJ}$$

(*c*) Process 4-1: v = constant heat rejection.

$$Q_{out} = m(u_4 - u_1) = mC_v(T_4 - T_1) = (7.209\times10^{-4}\text{ kg})(0.718\text{kJ/kg}\cdot\text{K})(800 - 290)K = \mathbf{0.264kJ}$$

$$W_{net,out} = Q_{in} - Q_{out} = 0.612 - 0.264 = 0.348\text{ kJ}$$

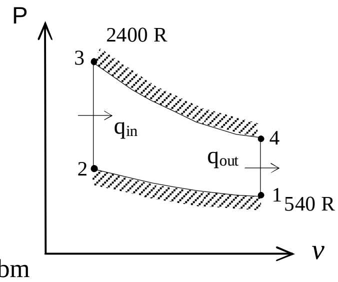
$$\eta_{th} = \frac{W_{net,out}}{Q_{in}} = \frac{0.348\text{ kJ}}{0.612\text{ kJ}} = \mathbf{56.9\%}$$

(*d*) $$V_{\min} = V_2 = \frac{V_{\max}}{r}$$

$$MEP = \frac{W_{net,out}}{V_1 - V_2} = \frac{W_{net,out}}{V_1(1-1/r)} = \frac{0.348\,\text{kJ}}{(0.0006\,\text{m}^3)(1-1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{648\,kPa}$$

8-38E An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The amount of heat transferred to the air during the heat addition process, the thermal efficiency, and the thermal efficiency of a Carnot cycle operating between the same temperature limits are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.



Properties The properties of air are given in Table A-21E.

Analysis (a) Process 1-2: isentropic compression.

$$T_1 = 540 \text{ R} \longrightarrow \begin{array}{l} u_1 = 92.04 \text{ Btu/lbm} \\ v_{r_1} = 144.32 \end{array}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_2} = \frac{1}{r} v_{r_2} = \frac{1}{8}(144.32) = 18.04 \longrightarrow u_2 = 211.28 \text{Btu/lbm}$$

Process 2-3: v = constant heat addition.

$$T_3 = 2400\text{R} \longrightarrow \begin{array}{l} u_3 = 452.70 \text{ Btu/lbm} \\ v_{r_3} = 2.419 \end{array}$$

$$q_{in} = u_3 - u_2 = 452.70 - 211.28 = 241.42 \textbf{ Btu/lbm}$$

(b) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (8)(2.419) = 19.35 \longrightarrow u_4 = 205.54 \text{ Btu/lbm}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = 205.54 - 92.04 = 113.50 \text{ Btu/lbm}$$

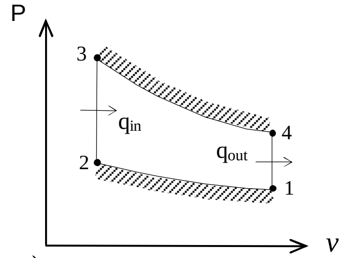
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{113.50 \text{ Btu/lbm}}{241.42 \text{ Btu/lbm}} = 47.0\%$$

(c) $$\eta_{th,C} = 1 - \frac{T_H}{T_L} = 1 - \frac{540 \text{ R}}{2400 \text{ R}} = \mathbf{77.5\%}$$

8-39E An ideal Otto cycle with argon as the working fluid has a compression ratio of 8. The amount of heat transferred to the argon during the heat addition process, the thermal efficiency, and the thermal efficiency of a Carnot cycle operating between the same temperature limits are to be determined. √

Assumptions **1** The air-standard assumptions are applicable with argon as the working fluid. **2** Kinetic and potential energy changes are negligible. **3** Argon is an ideal gas with constant specific heats.

Properties The properties of argon are C_p = 0.1253 Btu/lbm.R, C_v = 0.0756 Btu/lbm.R, and k = 1.667 (Table A-2E).



Analysis (*a*) Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (540\ \text{R})(8)^{0.667} = 2161\ \text{R}$$

Process 2-3: v = constant heat addition.

$$q_{in} = u_3 - u_2 = C_v(T_3 - T_2) = (0.0756\ \text{Btu/lbm.R})(2400 - 2161)\ \text{R} = \mathbf{18.07\ \ Btu/lbm.R}$$

(*b*) Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (2400\text{R})\left(\frac{1}{8}\right)^{0.667} = 600\text{R}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = C_v(T_4 - T_1) = (0.0756\ \text{Btu/lbm.R})(600 - 540)\text{R} = 4.536\ \text{Btu/lbm}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{4.536\ \text{Btu/lbm}}{18.07\ \text{Btu/lbm}} = \mathbf{74.9\%}$$

(*c*) $$\eta_{th,C} = 1 - \frac{T_H}{T_L} = 1 - \frac{540\ \text{R}}{2400\ \text{R}} = \mathbf{77.5\%}$$

Diesel Cycle

8-40C A diesel engine differs from the gasoline engine in the way combustion is initiated. In diesel engines combustion is initiated by compressing the air above the self-ignition temperature of the fuel whereas it is initiated by a spark plug in a gasoline engine.

8-41C The Diesel cycle differs from the Otto cycle in the heat addition process only; it takes place at constant volume in the Otto cycle, but at constant pressure in the Diesel cycle.

8-42C The gasoline engine.

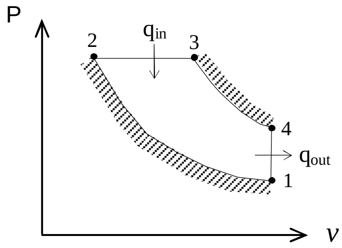
8-43C Diesel engines operate at high compression ratios because the diesel engines do not have the engine knock problem.

8-44C Cutoff ratio is the ratio of the cylinder volumes after and before the combustion process. As the cutoff ratio decreases, the efficiency of the diesel cycle increases.

8-45 An air-standard Diesel cycle with a compression ratio of 16 and a cutoff ratio of 2 is considered. The temperature after the heat addition process, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.



Analysis (*a*) Process 1-2: isentropic compression.

$$T_1 = 300\text{ K} \longrightarrow \begin{aligned} u_1 &= 214.07\text{ kJ/kg} \\ v_{r_1} &= 621.2 \end{aligned}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{16}(621.2) = 38.825 \longrightarrow \begin{aligned} T_2 &= 862.4\text{ K} \\ h_2 &= 890.9\text{ kJ/kg} \end{aligned}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2T_2 = (2)(862.4\text{ K}) = \mathbf{1724.8\ K}$$

$$\longrightarrow \begin{aligned} h_3 &= 1910.6\text{ kJ/kg} \\ v_{r_3} &= 4.546 \end{aligned}$$

(*b*) $$q_{in} = h_3 - h_2 = 1910.6 - 890.9 = 1019.7\text{ kJ/kg}$$

Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = \frac{v_4}{2v_2} v_{r_3} = \frac{r}{2} v_{r_3} = \frac{16}{2}(4.546) = 36.37 \longrightarrow u_4 = 659.7\text{kJ/kg}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = 659.7 - 214.07 = 445.63\text{ kJ/kg}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{445.63\text{ kJ/kg}}{1019.7\text{ kJ/kg}} = \mathbf{56.3\%}$$

(*c*) $$w_{net,out} = q_{in} - q_{out} = 1019.7 - 445.63 = 574.07\text{kJ/kg}$$

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{95\text{kPa}} = 0.906\text{m}^3/\text{kg} = \text{v}_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

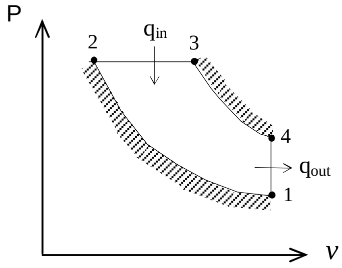
$$MEP = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1-1/r)} = \frac{574.07\text{kJ/kg}}{(0.906\text{m}^3/\text{kg})(1-1/16)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{675.9kPa}$$

8-46 An air-standard Diesel cycle with a compression ratio of 16 and a cutoff ratio of 2 is considered. The temperature after the heat addition process, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*a*) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\text{K})(16)^{0.4} = 909.4\text{K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2T_2 = (2)(909.4\text{K}) = \mathbf{1818.8K}$$

(*b*) $$q_{in} = h_3 - h_2 = C_p(T_3 - T_2) = (1.005\text{kJ/kg}\cdot\text{K})(1818.8 - 909.4)\text{K} = 913.9\text{kJ/kg}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2v_2}{v_4}\right)^{k-1} = (1818.8\text{K})\left(\frac{2}{16}\right)^{0.4} = 791.7\text{K}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = C_v(T_4 - T_1) = (0.718\text{kJ/kg}\cdot\text{K})(791.7 - 300)\text{K} = 353\text{kJ/kg}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{353\text{kJ/kg}}{913.9\text{kJ/kg}} = \mathbf{61.4\%}$$

(*c*) $$w_{net.out} = q_{in} - q_{out} = 913.9 - 353 = 560.9\text{kJ/kg}$$

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{95\text{kPa}} = 0.906\text{m}^3/\text{kg} = \text{v}_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$MEP = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1 - 1/r)} = \frac{560.9\text{kJ/kg}}{(0.906\text{m}^3/\text{kg})(1 - 1/16)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{660.4kPa}$$

8-47E An air-standard Diesel cycle with a compression ratio of 18.2 is considered. The cutoff ratio, the heat rejection per unit mass, and the thermal efficiency are to be determined. √

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

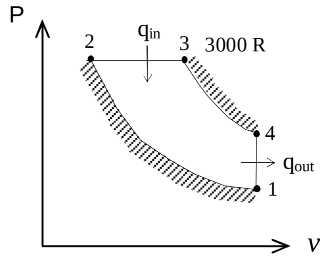
Properties The properties of air are given in Table A-21E.

Analysis (*a*) Process 1-2: isentropic compression.

$$T_1 = 540\text{ R} \longrightarrow \begin{array}{l} u_1 = 92.04\text{ Btu/lbm} \\ v_{r_1} = 144.32 \end{array}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{18.2}(144.32) = 7.93 \longrightarrow \begin{array}{l} T_2 = 1623.6\text{R} \\ h_2 = 402.05\text{Btu/lbm} \end{array}$$

Process 2-3: P = constant heat addition.



$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{3000\text{ R}}{1623.6\text{ R}} = 1.848$$

(*b*) $$T_3 = 3000\text{ R} \longrightarrow \begin{array}{l} h_3 = 790.68\text{ Btu/lbm} \\ v_{r_3} = 1.180 \end{array}$$

$$q_{in} = h_3 - h_2 = 790.68 - 402.05 = 388.63\text{ Btu/lbm}$$

Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = \frac{v_4}{1.848 v_2} v_{r_3} = \frac{r}{1.848} v_{r_3} = \frac{18.2}{1.848}(1.180) = 11.621 \longrightarrow u_4 = 250.91\text{Btu/lbm}$$

Process 4-1: *v* = constant heat rejection.

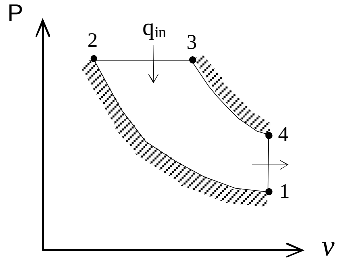
$$q_{out} = u_4 - u_1 = 250.91 - 92.04 = \mathbf{158.87\ \ Btu/lbm}$$

(*c*) $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{158.87\text{ Btu/lbm}}{388.63\text{ Btu/lbm}} = \mathbf{59.1\%}$$

8-48E An air-standard Diesel cycle with a compression ratio of 18.2 is considered. The cutoff ratio, the heat rejection per unit mass, and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.171 Btu/lbm.R, and k = 1.4 (Table A-2E).



Analysis (*a*) Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (540\text{R})(18.2)^{0.4} = 1724\text{R}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{3000\text{ R}}{1724\text{ R}} = \mathbf{1.741}$$

(*b*) $$q_{in} = h_3 - h_2 = C_p(T_3 - T_2) = (0.240\text{ Btu/lbm.R})(3000 - 1724)\text{R} = 306\text{ Btu/lbm}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1.741 v_2}{v_4}\right)^{k-1} = (3000\text{ R})\left(\frac{1.741}{18.2}\right)^{0.4} = 1173\text{ R}$$

Process 4-1: v = constant heat rejection.

$$\begin{aligned} q_{out} &= u_4 - u_1 = C_v(T_4 - T_1) \\ &= (0.171\text{ Btu/lbm.R})(1173 - 540)\text{R} = \mathbf{108\ Btu/lbm} \end{aligned}$$

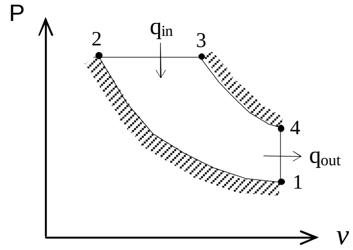
(*c*) $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{108\text{ Btu/lbm}}{306\text{ Btu/lbm}} = \mathbf{64.6\%}$$

8-49 An ideal diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (*a*) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (293\text{K})(20)^{0.4} = 971.1\text{K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3V_3}{T_3} = \frac{P_2V_2}{T_2} \longrightarrow \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{2200\text{ K}}{971.1\text{ K}} = 2.265$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{2.265V_2}{V_4}\right)^{k-1} = T_3\left(\frac{2.265}{r}\right)^{k-1} = (2200\text{ K})\left(\frac{2.265}{20}\right)^{0.4} = 920.6\text{ K}$$

$$q_{in} = h_3 - h_2 = C_p(T_3 - T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(2200 - 971.1)\text{K} = 1235\text{ kJ/kg}$$

$$q_{out} = u_4 - u_1 = C_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(920.6 - 293)\text{K} = 450.6\text{ kJ/kg}$$

$$w_{net,out} = q_{in} - q_{out} = 1235 - 450.6 = 784.4\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{784.4\text{ kJ/kg}}{1235\text{ kJ/kg}} = \mathbf{63.5\%}$$

(*b*)

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{K})}{95\text{kPa}} = 0.885\text{m}^3/\text{kg} = v_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

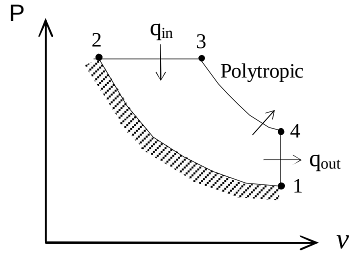
$$MEP = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1-1/r)} = \frac{784.4\text{kJ/kg}}{(0.885\text{m}^3/\text{kg})(1-1/20)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{933kPa}$$

8-50 A diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency and the mean effective pressure are to be determined. √

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (a) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (293\text{K})(20)^{0.4} = 971.1\text{ K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3V_3}{T_3} = \frac{P_2V_2}{T_2} \longrightarrow \frac{V_3}{V_2} = \frac{T_3}{T_2} = \frac{2200\text{ K}}{971.1\text{ K}} = 2.265$$

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{n-1} = \text{T}_3\left(\frac{2.265\text{V}_2}{\text{V}_4}\right)^{\text{n}-1} = \text{T}_3\left(\frac{2.265}{\text{r}}\right)^{\text{n}-1} = (2200\text{ K})\left(\frac{2.265}{20}\right)^{0.35} = 1026\text{ K}$$

$$q_{in} = h_3 - h_2 = C_p(T_3 - T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(2200 - 971.1)\text{ K} = 1235\text{ kJ/kg}$$

$$q_{out} = u_4 - u_1 = C_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(1026 - 293)\text{ K} = 526.3\text{ kJ/kg}$$

Note that q_{out} in this case does not represent the entire heat rejected since some heat is also rejected during the polytropic process, which is determined from an energy balance on process 3-4:

$$w_{34,out} = \frac{R(T_4 - T_3)}{1-n} = \frac{(0.287\text{kJ/kg}\cdot\text{K})(1026 - 2200)\text{ K}}{1-1.35} = 963\text{ kJ/kg}$$

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$\begin{aligned} q_{34,in} - w_{34,out} = u_4 - u_3 \longrightarrow q_{34,in} &= w_{34,out} + C_v(T_4 - T_3) \\ &= 963\text{ kJ/kg} + (0.718\text{ kJ/kg}\cdot\text{K})(1026 - 2200)\text{ K} \\ &= 120.1\text{ kJ/kg} \end{aligned}$$

which means that 120.1 kJ/kg of heat is transferred to the combustion gases during the expansion process. This is unrealistic since the gas is at a much higher temperature than the surroundings, and a hot gas loses heat during polytropic expansion. The cause of this unrealistic result is the constant specific heat assumption. If we were to use u data from the air table, we would obtain

$$q_{34,in} = w_{34,out} + (u_4 - u_3) = 963 + (781.3 - 1872.4) = -128.1\text{ kJ/kg}$$

which is a heat loss as expected. Then q_{out} becomes

$$q_{out} = q_{34,out} + q_{41,out} = 128.1 + 526.3 = 654.4\text{ kJ/kg}$$

and

$$w_{net,out} = q_{in} - q_{out} = 1235 - 654.4 = 580.6\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{580.6\text{ kJ/kg}}{1235\text{ kJ/kg}} = 47.0\%$$

(c)

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})}{95\ \text{kPa}} = 0.885\ \text{m}^3/\text{kg} = v_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$MEP = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1-1/r)} = \frac{580.6\ \text{kJ/kg}}{(0.885\ \text{m}^3/\text{kg})(1-1/20)}\left(\frac{1\,\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{691\ kPa}$$

8-51 Problem 8-50 is reconsidered. The effect of varying the compression ratio from 14 to 24 is to be investigated. The net work output, mean effective pressure and thermal efficiency as to be plotted as a function of the compression ratio. The T-s and P-v diagrams for the cycle are also to be plotted when the compression ratio is 20.

```
"Let's take advantage of the capabilities of EES and do this for variable specific heats."
"We assume that this ideal gas cycle takes place in a piston-cylinder device;
therefore, we will use a closed system analysis."

"See the T-s diagram in Plot Window1 and the P-v diagram in Plot Window2"
Procedure QTotal(q_12,q_23,q_34,q_41: q_in_total,q_out_total)
q_in_total = 0
q_out_total = 0
IF (q_12 > 0) THEN q_in_total = q_12 ELSE  q_out_total = - q_12
If q_23 > 0 then q_in_total = q_in_total + q_23 else q_out_total = q_out_total - q_23
If q_34 > 0 then q_in_total = q_in_total + q_34 else q_out_total = q_out_total - q_34
If q_41 > 0 then q_in_total = q_in_total + q_41 else q_out_total = q_out_total - q_41
END

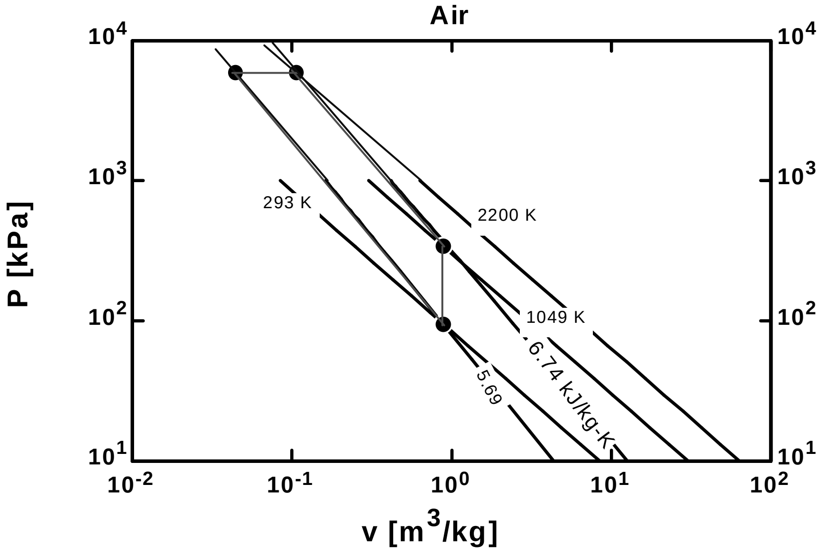
"Input Data"
T[1]=293"K"
P[1]=95"kPa"
T[3] = 2200"[K]"
n=1.35
{r_comp = 20}
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=0.287*T[1]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant pressure heat addition"
P[3]=P[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=0.287*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =P[2]*(V[3] - V[2])"constant pressure process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is polytropic expansion"
P[3]/P[4] =(V[4]/V[3])^n
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=0.287*T[4]
"Conservation of energy for process 3 to 4"
q_34 - w_34 = DELTAu_34    "q_34 is not 0 for the polytropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
P[3]*V[3]^n = Const
w_34=(P[4]*V[4]-P[3]*V[3])/(1-n)
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0       "constant volume process"
```

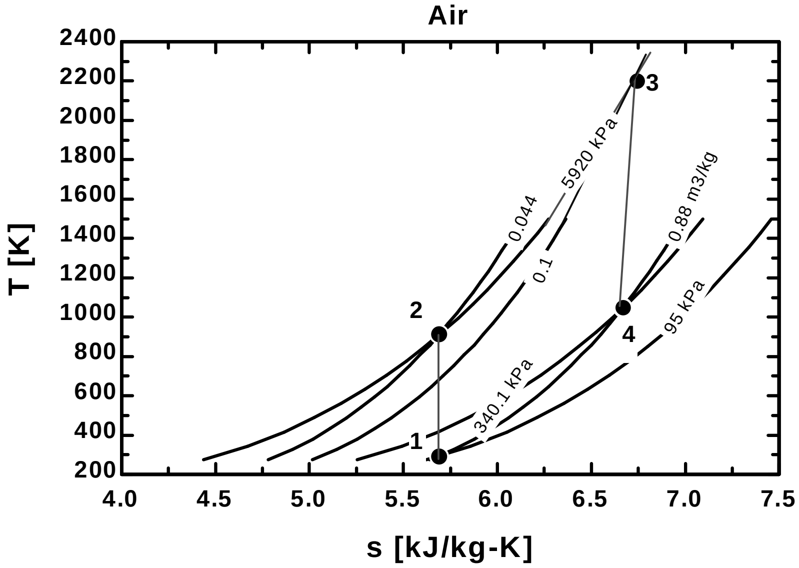
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])

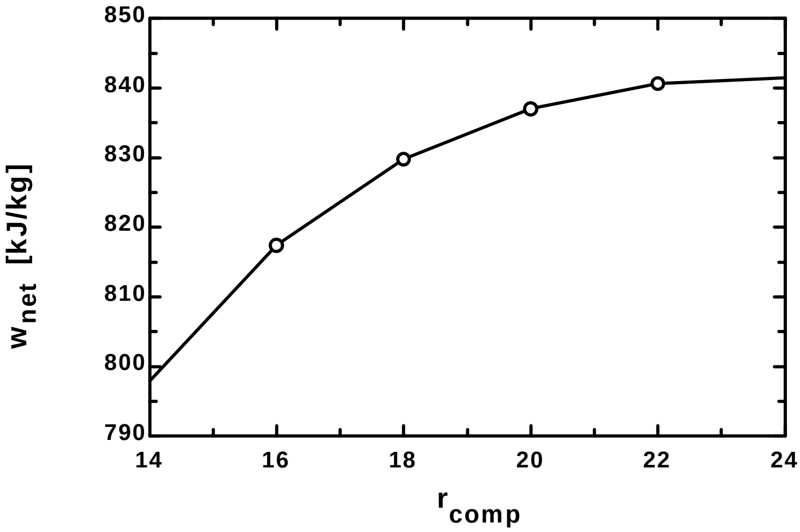
Call QTotal(q_12,q_23,q_34,q_41: q_in_total,q_out_total)

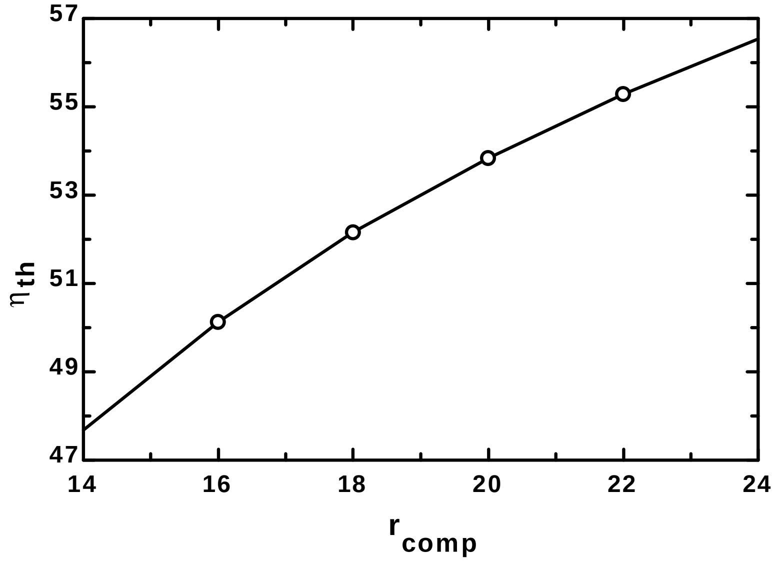
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
"The mean effective pressure is:"
MEP = w_net/(V[1]-V[2])"[kPa]"

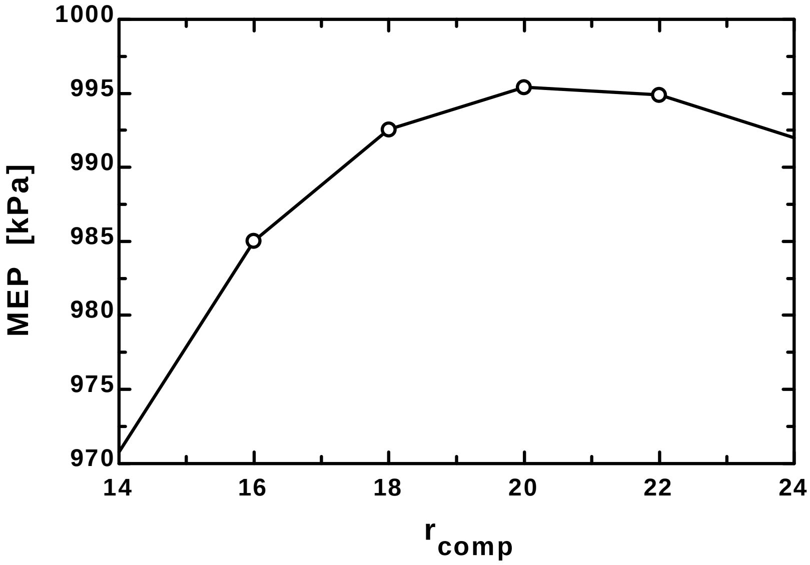
η_{th}	MEP [kPa]	r_{comp}	w_{net} [kJ/kg]
47.69	970.8	14	797.9
50.14	985	16	817.4
52.16	992.6	18	829.8
53.85	995.4	20	837.0
55.29	994.9	22	840.6
56.54	992	24	841.5






850
840
830
820
810
800
790
14
16
18
20
22
24
w_{net} [kJ/kg]
r_{comp}

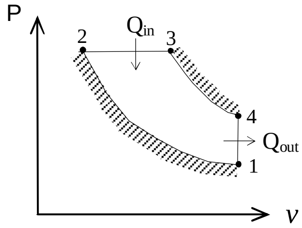

57
55
53
51
49
47
14
16
18
20
22
24
η_{th}
r_{comp}


1000
995
990
985
980
975
970
14
16
18
20
22
24
MEP [kPa]
r_{comp}

8-52 A four-cylinder ideal diesel engine with air as the working fluid has a compression ratio of 17 and a cutoff ratio of 2.2. The power the engine will deliver at 1500 rpm is to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).



Analysis Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (300\text{K})(17)^{0.4} = 931.8\text{K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2.2 T_2 = (2.2)(931.8\text{K}) = 2050\text{K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{n-1} = T_3\left(\frac{2.2V_2}{V_4}\right)^{n-1} = T_3\left(\frac{2.2}{r}\right)^{n-1} = (2050\text{K})\left(\frac{2.2}{17}\right)^{0.4} = 904.8\text{K}$$

$$V_1 = V_{disp} + V_2 = V_{disp} + V_1 / r \quad \rightarrow \quad V_1 = V_{disp}/(1-1/r) = 3/(1-1/17) = 3.188\,\text{L}$$

$$m = \frac{P_1 V_1}{RT_1} = \frac{(97\text{ kPa})(0.003188\,\text{m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})} = 3.592\times 10^{-3}\,\text{kg}$$

$$\begin{aligned} Q_{in} &= m(h_3 - h_2) = mC_p(T_3 - T_2) \\ &= (3.592\times10^{-3}\,\text{kg})(1.005\text{ kJ/kg}\cdot\text{K})(2050-931.8)\text{K} = 4.036\text{ kJ} \end{aligned}$$

$$\begin{aligned} Q_{out} &= m(u_4 - u_1) = mC_v(T_4 - T_1) \\ &= (3.592\times10^{-3}\,\text{kg})(0.718\text{ kJ/kg}\cdot\text{K})(904.8-300)\text{K} = 1.560\text{ kJ} \end{aligned}$$

$$W_{net,out} = Q_{in} - Q_{out} = 4.036 - 1.560 = 2.476\text{ kJ/rev}$$

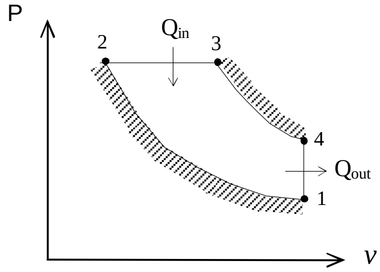
$$\dot{W}_{net,out} = \dot{n} W_{net,out} = (1500/60\text{ rev/s})(2.476\text{ kJ/rev}) = \mathbf{61.9\ \ kW}$$

Discussion Note that for 2-stroke engines, 1 thermodynamic cycle is equivalent to 1 mechanical cycle (and thus revolutions).

8-53 A four-cylinder ideal diesel engine with nitrogen as the working fluid has a compression ratio of 17 and a cutoff ratio of 2.2. The power the engine will deliver at 1500 rpm is to be determined.

Assumptions **1** The air-standard assumptions are applicable with nitrogen as the working fluid. **2** Kinetic and potential energy changes are negligible. **3** Nitrogen is an ideal gas with constant specific heats.

Properties The properties of nitrogen at room temperature are C_p = 1.039 kJ/kg·K, C_v = 0.743 kJ/kg·K, and $k = 1.4$ (Table A-2).



Analysis Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (300\text{K})(17)^{0.4} = 931.8\text{K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2}T_2 = 2.2T_2 = (2.2)(931.8\text{K}) = 2050\text{K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{n-1} = T_3\left(\frac{2.2V_2}{V_4}\right)^{n-1} = T_3\left(\frac{2.2}{r}\right)^{n-1} = (2050\text{K})\left(\frac{2.2}{17}\right)^{0.4} = 904.8\text{K}$$

$$V_1 = V_{disp} + V_2 = V_{disp} + V_1 / r \quad \to \quad V_1 = V_{disp} /(1-1/r) = 3/(1-1/17) = 3.188\,\text{L}$$

$$m = \frac{P_1 V_1}{RT_1} = \frac{(97\text{ kPa})(0.003188\,\text{m}^3)}{(0.2968\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})} = 3.473\times 10^{-3}\,\text{kg}$$

$$\begin{aligned} Q_{in} &= m(h_3 - h_2) = mC_p(T_3 - T_2) \\ &= (3.473\times 10^{-3}\,\text{kg})(1.039\text{ kJ/kg}\cdot\text{K})(2050-931.8)\text{K} = 4.035\text{ kJ} \end{aligned}$$

$$\begin{aligned} Q_{out} &= m(u_4 - u_1) = mC_v(T_4 - T_1) \\ &= (3.473\times 10^{-3}\,\text{kg})(0.743\,\text{kJ/kg}\cdot\text{K})(904.8-300)\text{K} = 1.561\,\text{kJ} \end{aligned}$$

$$W_{net,out} = Q_{in} - Q_{out} = 4.035 - 1.561 = 2.474\text{ kJ/rev}$$

$$\dot{W}_{net,out} = \dot{n}W_{net,out} = (1500/60\text{rev/s})(2.474\text{ kJ/rev}) = \mathbf{61.8\ \ kW}$$

Discussion Note that for 2-stroke engines, 1 thermodynamic cycle is equivalent to 1 mechanical cycle (and thus revolutions).

8-54 [*Also solved by EES on enclosed CD***]** An ideal dual cycle with air as the working fluid has a compression ratio of 14. The fraction of heat transferred at constant volume and the thermal efficiency of the cycle are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

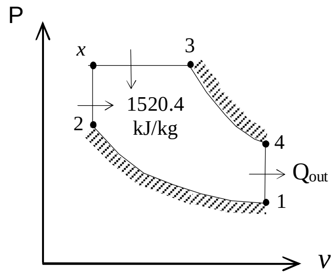
Properties The properties of air are given in Table A-21.

Analysis (a) Process 1-2: isentropic compression.

$$T_1 = 300 \text{ K} \longrightarrow \begin{array}{l} u_1 = 214.07 \text{ kJ/kg} \\ v_{r_1} = 621.2 \end{array}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{14}(621.2) = 44.37 \longrightarrow \begin{array}{l} T_2 = 823.1\text{K} \\ u_2 = 611.2\text{kJ/kg} \end{array}$$

Process 2-x, x-3: heat addition,



$$T_3 = 2200\text{K} \longrightarrow \begin{array}{l} h_3 = 2503.2\text{kJ/kg} \\ v_{r_3} = 2.012 \end{array}$$

$$q_{in} = q_{x-2,in} + q_{3-x,in} = (u_x - u_2) + (h_3 - h_x)$$
$$1520.4 = (u_x - 611.2) + (2503.2 - h_x)$$

By trial and error, we get T_x = 1300 K and h_x = 1395.97, u_x = 1022.82 kJ /kg.

Thus,

$$q_{2-x,in} = u_x - u_2 = 1022.82 - 611.2 = 411.62 \text{ kJ/kg}$$

and

$$ratio = \frac{q_{2-x,in}}{q_{in}} = \frac{411.62 \text{ kJ/kg}}{1520.4 \text{ kJ/kg}} = \mathbf{27.1\%}$$

(b) $$\frac{P_3 v_3}{T_3} = \frac{P_x v_x}{T_x} \longrightarrow \frac{v_3}{v_x} = \frac{T_3}{T_x} = \frac{2200 \text{ K}}{1300 \text{ K}} = 1.692 = \text{r}_\text{c}$$

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = \frac{v_4}{1.692 v_2} v_{r_3} = \frac{r}{1.692} v_{r_3} = \frac{14}{1.692}(2.012) = 16.648 \longrightarrow u_4 = 886.3\text{kJ/kg}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = 886.3 - 214.07 = 672.23 \text{ kJ/kg}$$
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{672.23 \text{ kJ/kg}}{1520.4 \text{ kJ/kg}} = \mathbf{55.8\%}$$

8-55 Problem 8-54 is reconsidered. The effect of varying the compression ratio from 10 to 18 is to be investigated. For a compression ratio of 14, the T-s and P-v diagrams for the cycle are to be plotted.

```
"We assume that this ideal dual cycle takes place in a piston-cylinder device;
therefore, we will use a closed system analysis."
"See Figure 8-23 for the P-v diagram for the cycle.  See the T-s diagram in
Plot Window1 and the P-v diagram in Plot Window2"

"Input Data"
T[1]=300"[K]"
P[1]=100"[kPa]"
T[4]=2200"[K]"
q_in_total=1520"[kJ/kg]"
r_v = 14
v[1]/v[2]=r_v "Compression ratio"

"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])"[kJ/kg-K]"
s[2]=s[1]"[kJ/kg-K]"
s[2]=entropy(air, T=T[2], v=v[2])"[kJ/kg-K]"
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=0.287*T[1]
"Conservation of energy for process 1 to 2"
q_12 -w_12 = DELTAu_12
q_12 =0"[kJ/kg]""isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])"[kJ/kg]"

"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])"[kJ/kg-K]"
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=0.287*T[3]
v[3]=v[2]"[m^3/kg]"
"Conservation of energy for process 2 to 3"
q_23 -w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])"[kJ/kg]"

"Process 3-4 is constant pressure heat addition"
s[4]=entropy(air, T=T[4], P=P[4])"[kJ/kg-K]"
{P[4]*v[4]/T[4]=P[3]*v[3]/T[3]}
P[4]*v[4]=0.287*T[4]
P[4]=P[3]"[kPa]"
"Conservation of energy for process 3 to4"
q_34 -w_34 = DELTAu_34
w_34 =P[3]*(v[4]-v[3])        "constant pressure process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
q_in_total=q_23+q_34

"Process 4-5 is isentropic expansion"
s[5]=entropy(air,T=T[5],P=P[5])"[kJ/kg-K]"
s[5]=s[4]"[kJ/kg-K]"
P[5]*v[5]/T[5]=P[4]*v[4]/T[4]
{P[5]*v[5]=0.287*T[5]}

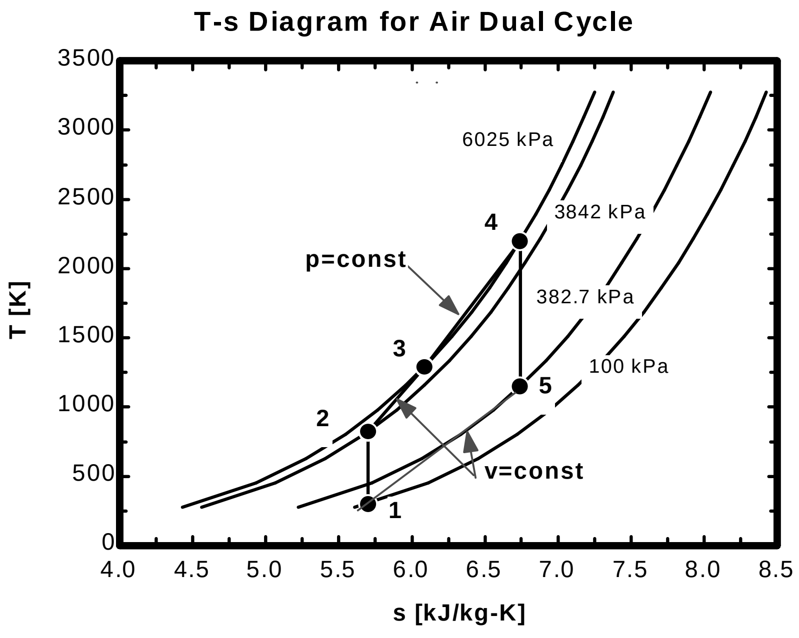
"Conservation of energy for process 4 to 5"
q_45 -w_45 = DELTAu_45
q_45 =0"[kJ/kg]""isentropic process"
```

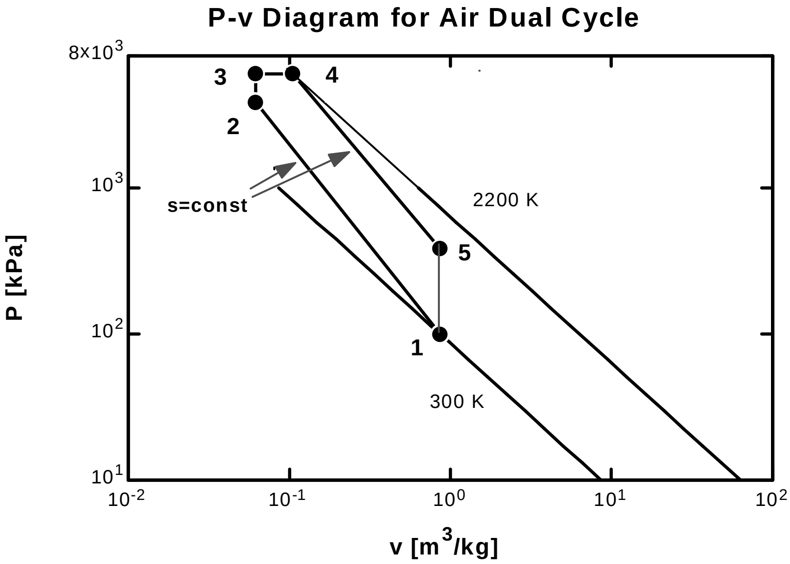
```
DELTAu_45=intenergy(air,T=T[5])-intenergy(air,T=T[4])"[kJ/kg]"

"Process 5-1 is constant volume heat rejection"
v[5]=v[1]"[m^3/kg]"
"Conservation of energy for process 2 to 3"
q_51 -w_51 = DELTAu_51
w_51 =0"[kJ/kg]""constant volume process"
DELTAu_51=intenergy(air,T=T[1])-intenergy(air,T=T[5])"[kJ/kg]"

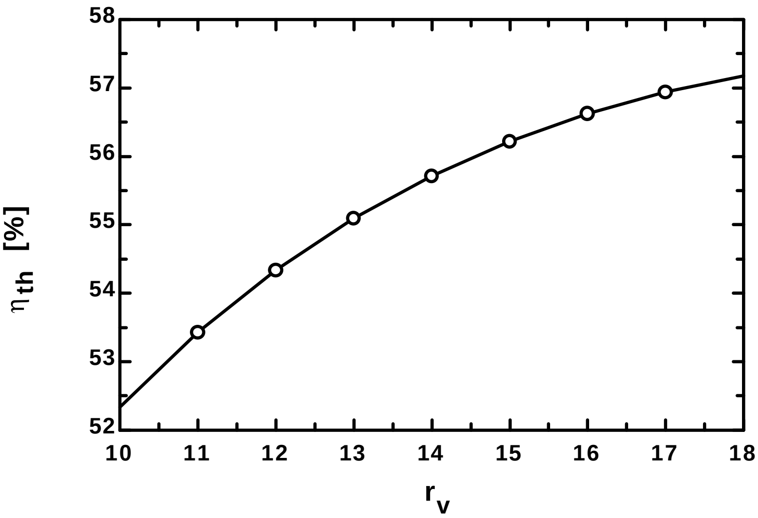
w_net = w_12+w_23+w_34+w_45+w_51
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
```

η_{th} [%]	r_v	w_{net} [kJ/kg]
52.33	10	795.4
53.43	11	812.1
54.34	12	826
55.09	13	837.4
55.72	14	846.9
56.22	15	854.6
56.63	16	860.7
56.94	17	865.5
57.17	18	869

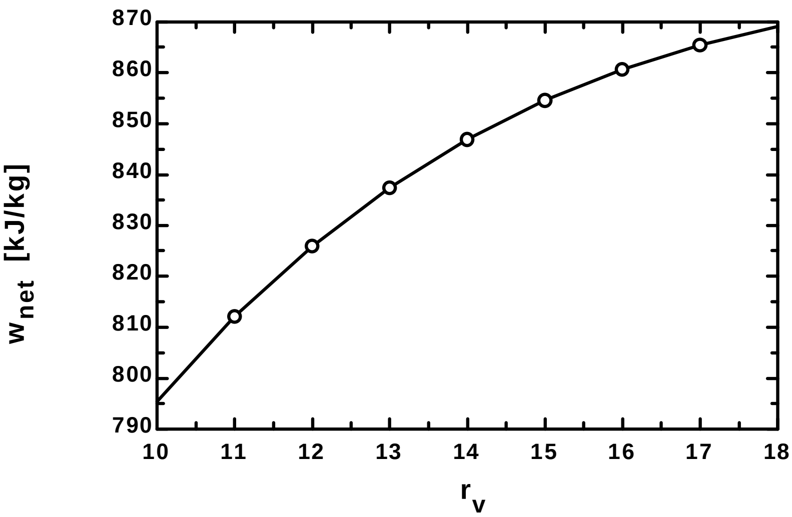




P-v Diagram for Air Dual Cycle
8x10^3
10^3
10^2
10^1
10^-2
10^-1
10^0
10^1
10^2
P [kPa]
v [m^3/kg]
1
2
3
4
5
s=const
2200 K
300 K



58
57
56
55
54
53
52
10
11
12
13
14
15
16
17
18
ηth [%]
rv



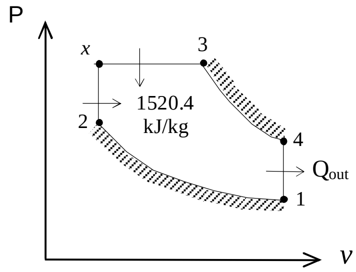
870
860
850
840
830
820
810
800
790
10
11
12
13
14
15
16
17
18
wnet [kJ/kg]
rv

8-56 An ideal dual cycle with air as the working fluid has a compression ratio of 14. The fraction of heat transferred at constant volume and the thermal efficiency of the cycle are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis (a) Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\text{K})(14)^{0.4} = 862\text{K}$$

Process 2-x, x-3: heat addition,

$$q_{in} = q_{2-x,in} + q_{3-x,in} = (u_x - u_2) + (h_3 - h_x)$$
$$= C_v(T_x - T_2) + C_p(T_3 - T_x)$$
$$1520.4\text{kJ/kg} = (0.718\text{kJ/kg}\cdot\text{K})(\text{T}_\text{x} - 862) + (1.005\text{kJ/kg}\cdot\text{K})(2200 - \text{T}_\text{x})$$

Solving for T_x we get T_x = 250 K which is impossible. Therefore, constant specific heats at room temperature turned out to be an unreasonable assumption in this case because of the very high temperatures involved.

Ideal and Actual Gas-Turbine (Brayton) Cycles

8-57C In gas turbine engines a gas is compressed, and thus the compression work requirements are very large since the steady-flow work is proportional to the specific volume.

8-58C They are (1) isentropic compression (in a compressor), (2) P = constant heat addition, (3) isentropic expansion (in a turbine), and (4) P = constant heat rejection.

8-59C For fixed maximum and minimum temperatures, (a) the thermal efficiency increases with pressure ratio, (b) the net work first increases with pressure ratio, reaches a maximum, and then decreases.

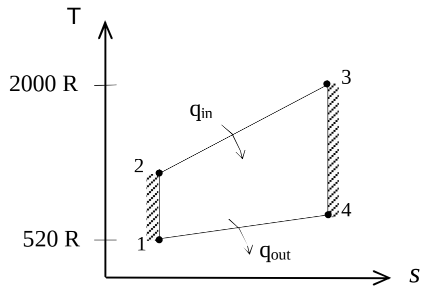
8-60C Back work ratio is the ratio of the compressor (or pump) work input to the turbine work output. It is usually between 0.40 and 0.6 for gas turbine engines.

8-61C As a result of turbine and compressor inefficiencies, (a) the back work ratio increases, and (b) the thermal efficiency decreases.

8-62E A simple ideal Brayton cycle with air as the working fluid has a pressure ratio of 10. The air temperature at the compressor exit, the back work ratio, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21E.



Analysis (*a*) Noting that process 1-2 is isentropic,

$$T_1 = 520 \text{ R} \longrightarrow \begin{array}{l} h_1 = 124.27 \text{ Btu/lbm} \\ P_{r_1} = 1.2147 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (10)(1.2147) = 12.147 \longrightarrow \begin{array}{l} T_2 = \mathbf{996.5 \text{ R}} \\ h_2 = 240.11 \text{ Btu/lbm} \end{array}$$

(*b*) Process 3-4 is isentropic, and thus

$$T_3 = 2000 \text{ R} \longrightarrow \begin{array}{l} h_3 = 504.71 \text{ Btu/lbm} \\ P_{r_3} = 174.0 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{10}\right)(174.0) = 17.4 \longrightarrow h_4 = 265.83 \text{ Btu/lbm}$$

$$w_{C,in} = h_2 - h_1 = 240.11 - 124.27 = 115.84 \text{ Btu/lbm}$$

$$w_{T,out} = h_3 - h_4 = 504.71 - 265.83 = 238.88 \text{ Btu/lbm}$$

Then the back-work ratio becomes

$$r_{bw} = \frac{w_{C,in}}{w_{T,out}} = \frac{115.84 \text{ Btu/lbm}}{238.88 \text{ Btu/lbm}} = \mathbf{48.5\%}$$

(*c*)

$$q_{in} = h_3 - h_2 = 504.71 - 240.11 = 264.60 \text{ Btu/lbm}$$

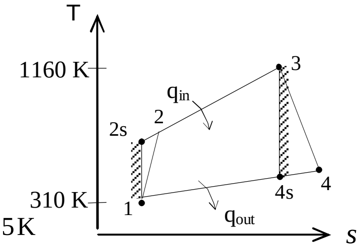
$$w_{net,out} = w_{T,out} - w_{C,in} = 238.88 - 115.84 = 123.04 \text{ Btu/lbm}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{123.04 \text{ Btu/lbm}}{264.60 \text{ Btu/lbm}} = \mathbf{46.5\%}$$

8-63 [*Also solved by EES on enclosed CD***]** A simple Brayton cycle with air as the working fluid has a pressure ratio of 8. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.



Analysis (*a*) Noting that process 1-2s is isentropic,

$$T_1 = 310 \text{ K} \longrightarrow \begin{aligned} h_1 &= 310.24 \text{ kJ/kg} \\ P_{r_1} &= 1.5546 \end{aligned}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.5546) = 12.44 \longrightarrow h_{2s} = 562.58 \text{ kJ/kg and } T_{2s} = 557.25 \text{K}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_C}$$

$$= 310.24 + \frac{562.58 - 310.24}{0.75} = 646.7 \text{ kJ/kg}$$

$$T_3 = 1160\text{K} \longrightarrow \begin{aligned} h_3 &= 1230.92 \text{kJ/kg} \\ P_{r_3} &= 207.2 \end{aligned}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(207.2) = 25.90 \longrightarrow h_{4s} = 692.19 \text{ kJ/kg and } T_{4s} = 680.3 \text{ K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow \begin{aligned} h_4 &= h_3 - \eta_T (h_3 - h_{4s}) \\ &= 1230.92 - (0.82)(1230.92 - 692.19) \\ &= 789.16 \text{kJ/kg} \end{aligned}$$

Thus, $T_4 = \mathbf{770.1\ K}$

(*b*) $q_{in} = h_3 - h_2 = 1230.92 - 646.7 = 584.2 \text{kJ/kg}$

$q_{out} = h_4 - h_1 = 789.16 - 310.24 = 478.92 \text{kJ/kg}$

$w_{net,out} = w_{in} - w_{out} = 584.2 - 478.92 = \mathbf{105.3 kJ/kg}$

(*c*) $\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{105.3 \text{kJ/kg}}{584.2 \text{kJ/kg}} = \mathbf{18.0\%}$

8-64 Problem 8-63 is reconsidered by allowing the mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic efficiencies of the turbine and compressor to vary. The compressor inlet pressure is to be taken 100 kPa. (A general solution for the problem is to be developed by taking advantage of the diagram window method for supplying data to EES).

```
"Input data - from diagram window"
{P_ratio = 8}
{T[1] = 310"K"
P[1]= 100"kPa"
T[3] = 1160"K"
m_dot = 20 "kg/s"
Eta_c = 75/100
Eta_t = 82/100}

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

"Compressor analysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])  "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

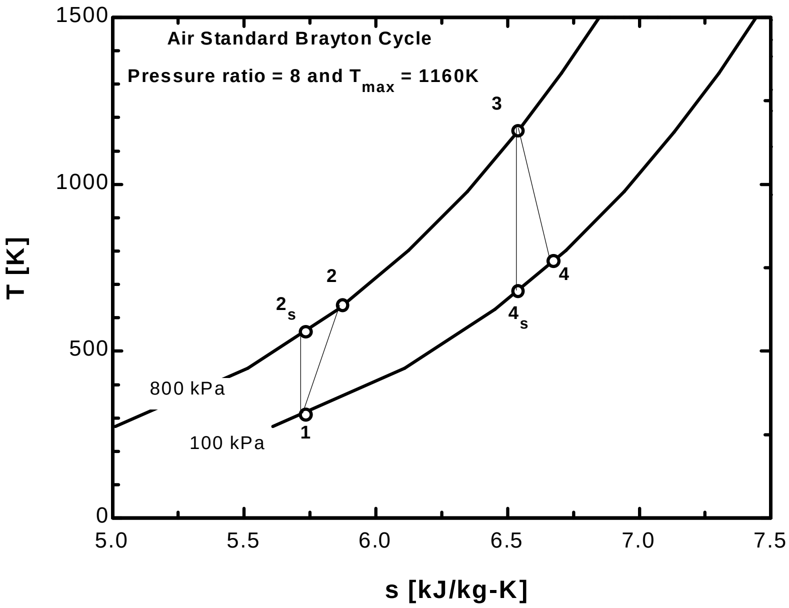
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming
W=0, ke=pe=0"

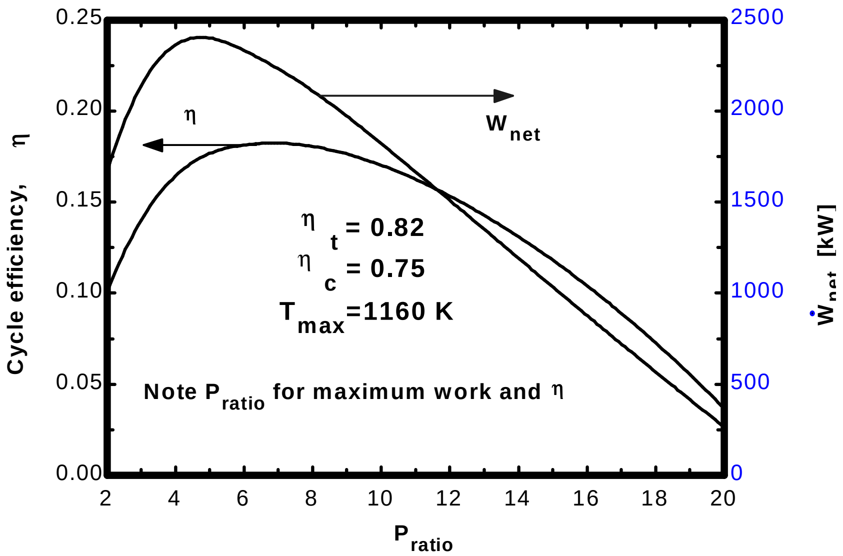
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine
exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c "Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
s[4]=entropy('air',T=T[4],P=P[4])
```

Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.5229	0.1	2	1818	1659	3477	16587
0.6305	0.1644	4	4033	2364	6396	14373
0.7038	0.1814	6	5543	2333	7876	12862
0.7611	0.1806	8	6723	2110	8833	11682
0.8088	0.1702	10	7705	1822	9527	10700
0.85	0.1533	12	8553	1510	10063	9852
0.8864	0.131	14	9304	1192	10496	9102
0.9192	0.1041	16	9980	877.2	10857	8426
0.9491	0.07272	18	10596	567.9	11164	7809
0.9767	0.03675	20	11165	266.1	11431	7241



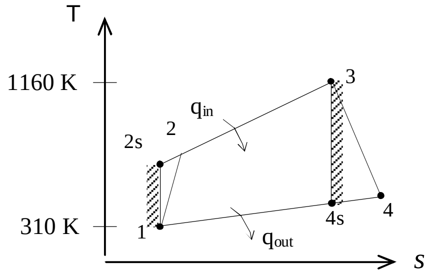


8-65 A simple Brayton cycle with air as the working fluid has a pressure ratio of 8. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **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-2).

Analysis (*a*) Using the compressor and turbine efficiency relations,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (310\text{K})(8)^{0.4/1.4} = 561.5\text{K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1160\text{K})\left(\frac{1}{8}\right)^{0.4/1.4} = 640.4\text{K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{C_p(T_{2s}-T_1)}{C_p(T_2-T_1)} \longrightarrow T_2 = T_1 + \frac{T_{2s}-T_1}{\eta_C}$$

$$= 310 + \frac{561.5-310}{0.75} = 645.3\text{K}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} = \frac{C_p(T_3-T_4)}{C_p(T_3-T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3-T_{4s})$$

$$= 1160 - (0.82)(1160-640.4)$$

$$= \mathbf{733.9K}$$

(*b*)

$$q_{in} = h_3 - h_2 = C_p(T_3 - T_2) = (1.005\text{kJ/kg}\cdot\text{K})(1160-645.3)\text{K} = 517.3\text{kJ/kg}$$

$$q_{out} = h_4 - h_1 = C_p(T_4 - T_1) = (1.005\text{kJ/kg}\cdot\text{K})(733.9-310)\text{K} = 426.0\text{kJ/kg}$$

$$w_{net,out} = w_{in} - w_{out} = 517.3 - 426.0 = \mathbf{91.3kJ/kg}$$

(*c*)

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{91.3\text{ kJ/kg}}{517.3\text{ kJ/kg}} = \mathbf{17.6\%}$$

8-66 A gas turbine power plant that operates on the simple Brayton cycle with air as the working fluid has a specified pressure ratio. The required mass flow rate of air is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **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-2).

Analysis (*a*) Using the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(12)^{0.4/1.4} = 610.2\text{ K}$$

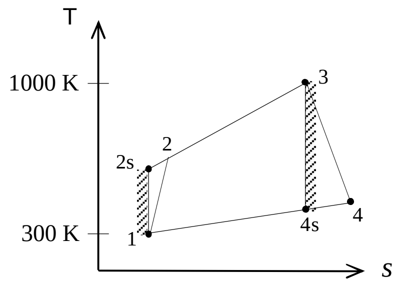
$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1000\text{ K})\left(\frac{1}{12}\right)^{0.4/1.4} = 491.7\text{ K}$$

$$w_{s,C,in} = h_{2s} - h_1 = C_p\left(T_{2s} - T_1\right) = (1.005\text{ kJ/kg}\cdot\text{K})(610.2 - 300)\text{K} = 311.75\text{ kJ/kg}$$

$$w_{s,T,out} = h_3 - h_{4s} = C_p\left(T_3 - T_{4s}\right) = (1.005\text{ kJ/kg}\cdot\text{K})(1000 - 491.7)\text{K} = 510.84\text{ kJ/kg}$$

$$w_{s,net,out} = w_{s,T,out} - w_{s,C,in} = 510.84 - 311.75 = 199.09\text{ kJ/kg}$$

$$\dot{m}_s = \frac{\dot{W}_{net,out}}{w_{s,net,out}} = \frac{90{,}000\text{ kJ/s}}{199.09\text{ kJ/kg}} = \mathbf{452.1\ \ kg/s}$$



(*b*) The net work output is determined to be

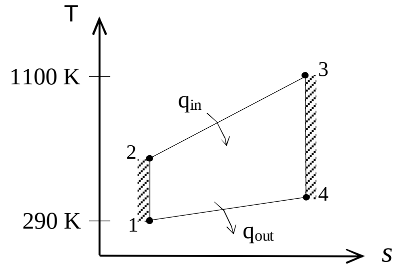
$$w_{a,net,out} = w_{a,T,out} - w_{a,C,in} = \eta_T w_{s,T,out} - w_{s,C,in}/\eta_C$$
$$= (0.80)(510.84) - 311.75/0.80 = 18.98\text{kJ/kg}$$

$$\dot{m}_a = \frac{\dot{W}_{net,out}}{w_{a,net,out}} = \frac{90{,}000\text{ kJ/s}}{18.98\text{ kJ/kg}} = \mathbf{4742\ \ kg/s}$$

8-67 A stationary gas-turbine power plant operates on a simple ideal Brayton cycle with air as the working fluid. The power delivered by this plant is to be determined assuming constant and variable specific heats.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas.

Analysis (*a*) Assuming constant specific heats,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (290\text{K})(8)^{0.4/1.4} = 525.3\text{K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1100\text{K})\left(\frac{1}{8}\right)^{0.4/1.4} = 607.2\text{ K}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p(T_4 - T_1)}{C_p(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{607.2 - 290}{1100 - 525.3} = 0.448$$

$$\dot{W}_{net,out} = \eta_T \dot{Q}_{in} = (0.448)(35{,}000\text{ kW}) = \mathbf{15{,}680\ kW}$$

(*b*) Assuming variable specific heats (Table A-21),

$$T_1 = 290\text{ K} \longrightarrow \begin{array}{l} h_1 = 290.16\text{ kJ/kg} \\ P_{r_1} = 1.2311 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.2311) = 9.8488 \longrightarrow h_2 = 526.12\text{ kJ/kg}$$

$$T_3 = 1100\text{ K} \longrightarrow \begin{array}{l} h_3 = 1161.07\text{ kJ/kg} \\ P_{r_3} = 167.1 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(167.1) = 20.89 \longrightarrow h_4 = 651.37\text{ kJ/kg}$$

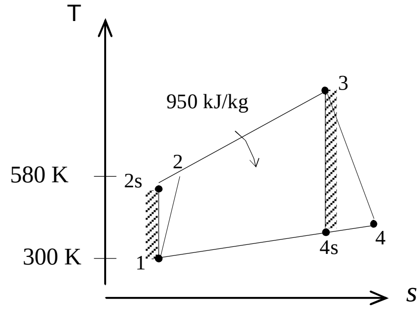
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} == 1 - \frac{h_4 - h_1}{h_3 - h_2} = 1 - \frac{651.37 - 290.16}{1161.07 - 526.11} = 0.431$$

$$\dot{W}_{net,out} = \eta_T \dot{Q}_{in} = (0.431)(35{,}000\text{ kW}) = \mathbf{15{,}085\ kW}$$

8-68 An actual gas-turbine power plant operates at specified conditions. The fraction of the turbine work output used to drive the compressor and the thermal efficiency are to be determined. √

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.



Analysis (a) Using the isentropic relations,

$$T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}$$
$$T_2 = 580 \text{ K} \longrightarrow h_2 = 586.04 \text{ kJ/kg}$$
$$r_p = \frac{P_2}{P_1} = \frac{700}{100} = 7$$
$$q_{in} = h_3 - h_2 \longrightarrow h_3 = 950 + 586.04 = 1536.04 \text{kJ/kg}$$
$$\longrightarrow P_{r_3} = 474.11$$
$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{7}\right)(474.11) = 67.73 \longrightarrow h_{4s} = 905.83 \text{kJ/kg}$$
$$w_{C,in} = h_2 - h_1 = 586.04 - 300.19 = 285.85 \text{kJ/kg}$$
$$w_{T,out} = \eta_T (h_3 - h_{4s}) = (0.86)(1536.04 - 905.83) = 542.0 \text{kJ/kg}$$

Thus, $$r_{bw} = \frac{w_{C,in}}{w_{T,out}} = \frac{285.85 \text{ kJ/kg}}{542.0 \text{ kJ/kg}} = \mathbf{52.7\%}$$

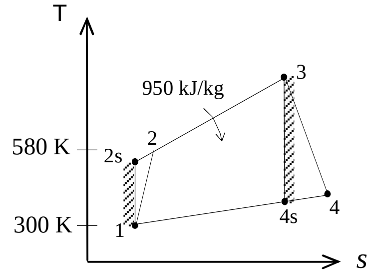
(b) $$w_{net.out} = w_{T,out} - w_{C,in} = 542.0 - 285.85 = 256.15 \text{ kJ/kg}$$
$$\eta_{th} = \frac{w_{net.out}}{q_{in}} = \frac{256.15 \text{ kJ/kg}}{950 \text{ kJ/kg}} = \mathbf{27.0\%}$$

8-69 A gas-turbine power plant operates at specified conditions. The fraction of the turbine work output used to drive the compressor and the thermal efficiency are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **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-2).

Analysis (*a*) Using constant specific heats,



$$r_p = \frac{P_2}{P_1} = \frac{700}{100} = 7$$

$$q_{in} = h_3 - h_2 = C_p\left(T_3 - T_2\right)$$

$$\longrightarrow T_3 = T_2 + q_{in}/C_p$$
$$= 580\text{K} + (950\text{kJ/kg})/(1.005\text{kJ/kg}\cdot\text{K})$$
$$= 1525.3\text{K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1525.3\text{K})\left(\frac{1}{7}\right)^{0.4/1.4} = 874.8\text{K}$$

$$w_{C,in} = h_2 - h_1 = C_p\left(T_2 - T_1\right) = (1.005\text{kJ/kg}\cdot\text{K})(580 - 300)\text{K} = 281.4\text{kJ/kg}$$

$$w_{T,out} = \eta_T\left(h_3 - h_{4s}\right) = \eta_T C_p\left(T_3 - T_{4s}\right) = (0.86)(1.005\text{kJ/kg}\cdot\text{K})(1525.3 - 874.8)\text{K} = 562.2\text{kJ/kg}$$

Thus, $r_{bw} = \dfrac{w_{C,in}}{w_{T,out}} = \dfrac{281.4\text{ kJ/kg}}{562.2\text{ kJ/kg}} = \mathbf{50.1\%}$

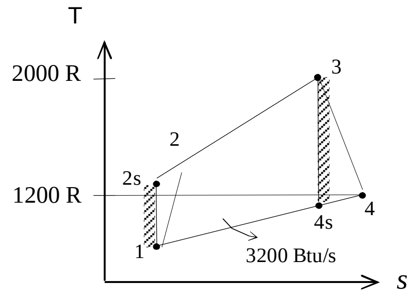
(b) $w_{net,out} = w_{T,out} - w_{C,in} = 562.2 - 281.4 = 280.8\text{ kJ/kg}$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{280.8\text{ kJ/kg}}{950\text{ kJ/kg}} = \mathbf{29.6\%}$$

8-70E A gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The net power output of the plant is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21E.



Analysis Using variable specific heats for air,

$$T_3 = 2000 \text{ R} \longrightarrow h_3 = 504.71 \text{ Btu/lbm}$$

$$T_4 = 1200 \text{ R} \longrightarrow h_4 = 291.30 \text{ Btu/lbm}$$

$$r_p = \frac{P_2}{P_1} = \frac{120}{15} = 8$$

$$\dot{Q}_{out} = \dot{m}(h_4 - h_1) \longrightarrow h_1 = 291.30 - 6400/40 = 131.30 \text{ Btu/lbm}$$

$$\longrightarrow P_{r_1} = 1.474$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.474) = 11.79 \longrightarrow h_{2s} = 238.07 \text{Btu/lbm}$$

$$\dot{W}_{C,in} = \dot{m}(h_{2s} - h_1)/\eta_C = (40 \text{lbm/s})(238.07 - 131.30)/(0.80) = 5339 \text{Btu/s}$$

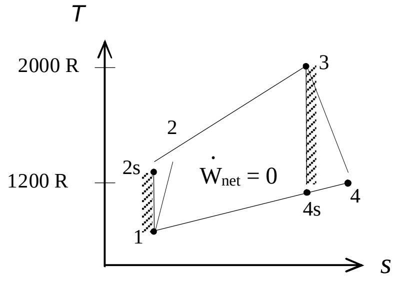
$$\dot{W}_{T,out} = \dot{m}(h_3 - h_4) = (40 \text{lbm/s})(504.71 - 291.30) = 8536 \text{Btu/s}$$

$$\dot{W}_{net,out} = \dot{W}_{T,out} - \dot{W}_{C,in} = 8536 - 5339 = 3197 \text{Btu/s} = \mathbf{3373 kW}$$

8-71E A gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The compressor efficiency for which the power plant produces zero net work is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21E.



Analysis Using variable specific heats,

$$T_3 = 2000 \text{ R} \longrightarrow h_3 = 504.71 \text{ Btu/lbm}$$

$$T_4 = 1200 \text{ R} \longrightarrow h_4 = 291.30 \text{ Btu/lbm}$$

$$r_p = \frac{P_2}{P_1} = \frac{120}{15} = 8$$

$$\dot{Q}_{out} = \dot{m}(h_4 - h_1) \longrightarrow h_1 = 291.30 - 6400/40 = 131.30 \text{Btu/lbm} \longrightarrow P_{r_1} = 1.474$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.474) = 11.79 \longrightarrow h_{2s} = 238.07 \text{Btu/lbm}$$

Then, $\dot{W}_{C,in} = \dot{W}_{T,out} \longrightarrow \dot{m}(h_{2s} - h_1)/\eta_C = \dot{m}(h_3 - h_4)$

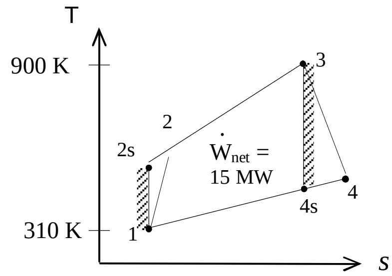
$$\eta_C = \frac{h_{2s} - h_1}{h_3 - h_4} = \frac{238.07 - 131.30}{504.71 - 291.30} = \mathbf{50.0\%}$$

8-72 A 15-MW gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The mass flow rate of air through the cycle is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.

Analysis Using variable specific heats,



$$T_1 = 310\text{ K} \longrightarrow \begin{array}{l} h_1 = 310.24\text{ kJ/kg} \\ P_{r_1} = 1.5546 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = (8)(1.5546) = 12.44 \longrightarrow h_{2s} = 562.26\text{kJ/kg}$$

$$T_3 = 900\text{K} \longrightarrow \begin{array}{l} h_3 = 932.93\text{kJ/kg} \\ P_{r_3} = 75.29 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3}P_{r_3} = \left(\frac{1}{8}\right)(75.29) = 9.411 \longrightarrow h_{4s} = 519.32\text{kJ/kg}$$

$$\begin{aligned} w_{net,out} &= w_{T,out} - w_{C,in} = \eta_T(h_3 - h_{4s}) - (h_{2s} - h_1)/\eta_C \\ &= (0.86)(932.93 - 519.32) - (562.26 - 310.24)/(0.80) = 40.68\text{kJ/kg} \end{aligned}$$

and

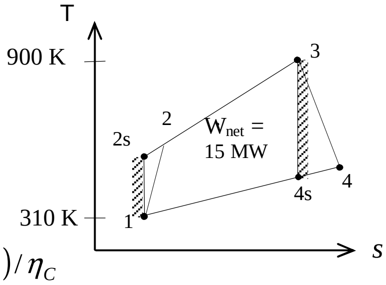
$$\dot{m} = \frac{\dot{W}_{net,out}}{w_{net,out}} = \frac{15{,}000\text{ kJ/s}}{40.68\text{ kJ/kg}} = \mathbf{368.7\ kg/s}$$

8-73 A 15-MW gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The mass flow rate of air through the cycle is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **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-2).

Analysis Using constant specific heats,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (310\text{ K})(8)^{0.4/1.4} = 561.5\text{K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (900\text{K})\left(\frac{1}{8}\right)^{0.4/1.4} = 496.8\text{K}$$

$$\begin{aligned} w_{net,out} &= w_{T,out} - w_{C,in} = \eta_T C_p(T_3 - T_{4s}) - C_p(T_{2s} - T_1)/\eta_C \\ &= (1.005\text{kJ/kg}\cdot\text{K})[(0.86)(900 - 496.8) - (561.5 - 310)/(0.80)]\text{K} \\ &= 32.5\text{kJ/kg} \end{aligned}$$

and

$$\dot{m} = \frac{\dot{W}_{net,out}}{w_{net,out}} = \frac{15{,}000\text{ kJ/s}}{32.5\text{ kJ/kg}} = \mathbf{461.5\ kg/s}$$

Brayton Cycle with Regeneration

8-74C Regeneration increases the thermal efficiency of a Brayton cycle by capturing some of the waste heat from the exhaust gases and preheating the air before it enters the combustion chamber.

8-75C Yes. At very high compression ratios, the gas temperature at the turbine exit may be lower than the temperature at the compressor exit. Therefore, if these two streams are brought into thermal contact in a regenerator, heat will flow to the exhaust gases instead of from the exhaust gases. As a result, the thermal efficiency will decrease.

8-76C The extent to which a regenerator approaches an ideal regenerator is called the effectiveness ε, and is defined as $\varepsilon = q_{regen,\ act} / q_{regen,\ max}$.

8-77C (b) turbine exit.

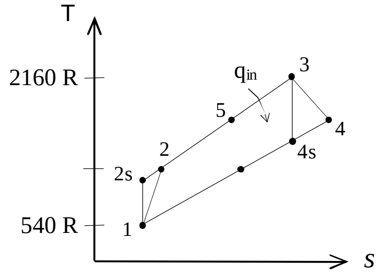
8-78C The steam injected increases the mass flow rate through the turbine and thus the power output. This, in turn, increases the thermal efficiency since $\eta = W / Q_{in}$ and W increases while $Q_{\rm in}$ remains constant. Steam can be obtained by utilizing the hot exhaust gases.

8-79E A car is powered by a gas turbine with a pressure ratio of 4. The thermal efficiency of the car and the mass flow rate of air for a net power output of 135 hp are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** The ambient air is 540 R and 14.5 psia. **4** The effectiveness of the regenerator is 0.9, and the isentropic efficiencies for both the compressor and the turbine are 80%. **5** The combustion gases can be treated as air.

Properties The properties of air at the compressor and turbine inlet temperatures can be obtained from Table A-21E.

Analysis The gas turbine cycle with regeneration can be analyzed as follows:



$$T_1 = 540\text{R} \longrightarrow \begin{array}{l} h_1 = 129.06 \text{Btu/lbm} \\ P_{r_1} = 1.386 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (4)(1.386) = 5.544 \longrightarrow h_{2s} = 192.0 \text{Btu/lbm}$$

$$T_3 = 2160\text{R} \longrightarrow \begin{array}{l} h_3 = 549.35 \text{Btu/lbm} \\ P_{r_3} = 230.12 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{4}\right)(230.12) = 57.53 \longrightarrow h_{4s} = 372.2 \text{Btu/lbm}$$

and

$$\eta_{\text{comp}} = \frac{h_{2s} - h_1}{h_2 - h_1} \rightarrow 0.80 = \frac{192.0 - 129.06}{h_2 - 129.06} \rightarrow h_2 = 207.74 \text{ kJ/kg}$$

$$\eta_{\text{turb}} = \frac{h_3 - h_4}{h_3 - h_{4s}} \rightarrow 0.80 = \frac{549.35 - h_4}{549.35 - 372.2} \rightarrow h_4 = 407.63 \text{ kJ/kg}$$

Then the thermal efficiency of the gas turbine cycle becomes

$$\dot{q}_{\text{regen}} = \varepsilon(h_4 - h_2) = 0.9(407.63 - 207.74) = 179.9 \text{ Btu/lbm}$$

$$q_{in} = (h_3 - h_2) - q_{regen} = (549.35 - 207.74) - 179.9 \text{ kJ/} = 161.7 \text{ Btu/s}$$

$$w_{net,out} = w_{T,out} - w_{C,in} = (h_3 - h_4) - (h_2 - h_1) = (549.35 - 407.63) - (207.74 - 129.06) = \mathbf{63.0\ Btu/lbm}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{63.0 \text{ Btu/lbm}}{161.7 \text{ Btu/lbm}} = \mathbf{0.39\%}$$

Finally, the mass flow rate of air through the turbine becomes

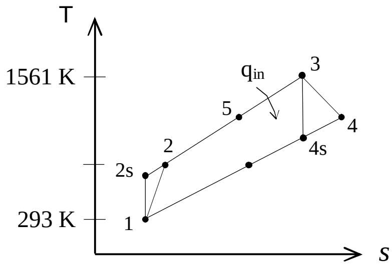
$$\dot{m}_{steam} = \frac{\dot{W}_{net}}{\dot{w}_{net}} = \frac{135 \text{ hp}}{63.0 \text{ Btu/lbm}} \left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}}\right) = \mathbf{1.51\ lbm/s}$$

8-80 [*Also solved by EES on enclosed CD***]** The thermal efficiency and power output of an actual gas turbine are given. The isentropic efficiency of the turbine and of the compressor, and the thermal efficiency of the gas turbine modified with a regenerator are to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible. **3** The mass flow rates of air and of the combustion gases are the same, and the properties of combustion gases are the same as those of air.

Properties The properties of air are given in Table A-21.

Analysis The properties at various states are



$$T_1 = 20°\text{C} = 293\text{K} \longrightarrow \begin{array}{l} h_1 = 293.2\text{kJ/kg} \\ P_{r_1} = 1.2765 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (14.7)(1.2765) = 18.765 \longrightarrow h_{2s} = 643.3\text{kJ/kg}$$

$$T_3 = 1288°\text{C} = 1561\text{K} \longrightarrow \begin{array}{l} h_3 = 1710.0\text{kJ/kg} \\ P_{r_3} = 712.5 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{14.7}\right)(712.5) = 48.47 \longrightarrow h_{4s} = 825.23\text{kJ/kg}$$

The net work output and the heat input per unit mass are

$$w_{net} = \frac{\dot{W}_{net}}{\dot{m}} = \frac{159{,}000\,\text{kW}}{1{,}536{,}000\,\text{kg/h}}\left(\frac{3600\,\text{s}}{1\,\text{h}}\right) = 372.66\text{ kJ/kg}$$

$$q_{in} = \frac{w_{net}}{\eta_{th}} = \frac{372.66\text{ kJ/kg}}{0.359} = 1038.0\text{ kJ/kg}$$

$$q_{in} = h_3 - h_2 \quad \rightarrow \quad h_2 = h_3 - q_{in} = 1710 - 1038 = 672.0\text{ kJ/kg}$$

$$q_{out} = q_{in} - w_{net} = 1038.0 - 372.66 = 665.34\text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 \quad \rightarrow \quad h_4 = q_{out} + h_1 = 665.34 + 293.2 = 958.54\text{ kJ/kg} \quad \rightarrow \quad T_4 = 650°\text{C}$$

Then the compressor and turbine efficiencies become

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{1710 - 958.54}{1710 - 825.23} = \mathbf{0.849}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{643.3 - 293.2}{672 - 293.2} = \mathbf{0.924}$$

When a regenerator is added, the new heat input and the thermal efficiency become

$$q_{regen} = \varepsilon(h_4 - h_2) = (0.80)(958.54 - 672.0) = 286.54\,\text{kJ/kg}$$

$$q_{in,new} = q_{in} - q_{regen} = 1038 - 286.54 = 751.46\,\text{kJ/kg}$$

$$\eta_{th,new} = \frac{w_{net}}{q_{in,new}} = \frac{372.66\,\text{kJ/kg}}{751.46\,\text{kJ/kg}} = \mathbf{0.496}$$

Discussion Note an 80% efficient regenerator would increase the thermal efficiency of this gas turbine from 35.9% to 49.6%.

8-81 Problem 8-80 is reconsidered. A solution is to be developed that allows different isentropic efficiencies for the compressor and turbine, and study the effect of the isentropic efficiencies on net work done and the heat supplied to the cycle. The T-s diagram for the cycle is to be plotted.

```
"For both the compressor and turbine we assume adiabatic, steady-flow,
and neglect KE and PE."
"This EES solution does not require that the isentropic efficiency of the compressor
and turbine be the same."
"See Plot Window1 for the T-s diagram and state notation for this problem.  Also see
Figures 8-38 and 8-39 in the text."

"Input data"
T[3] = 1288"C"
Pratio = 14.7
T[1] = 20"C"
P[1]= 100"kPa"
{T[4]=589"C"}
{W_dot_net=159"MW" }"We omit the information about the cycle net work"
m_dot = 1536000"kg/h"*convert(kg/h,kg/s)"[kg/s]"
{Eta_th_noreg=0.359} "We omit the information about the cycle efficiency."
Eta_reg = 0.80
Eta_c = 0.892"Compressor isentorpic efficiency"
Eta_t = 0.926"Turbien isentropic efficiency"

"Isentropic Compressor analysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = W_dot_compisen/W_dot_comp
"compressor adiabatic efficiency, W_dot_comp > W_dot_compisen"

"Conservation of energy for the compressor for the isentropic case:
E_dot_in - E_dot_out = DELTAE_dot=0 for steady-flow"
m_dot*h[1] + W_dot_compisen = m_dot*h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])
"Actual compressor analysis:"
m_dot*h[1] + W_dot_comp = m_dot*h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

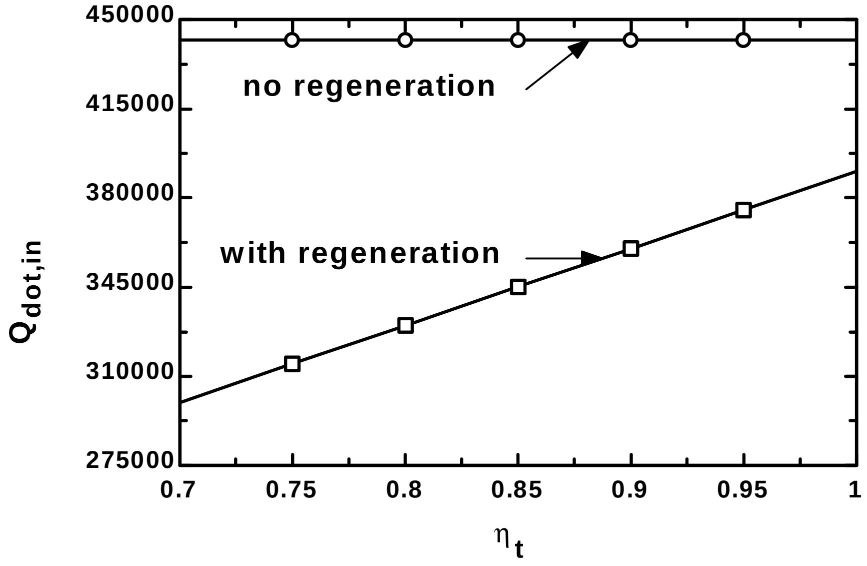
"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
E_dot_in - E_dot_out =DELTAE_dot_cv =0 for steady flow"
m_dot*h[2] + Q_dot_in_noreg = m_dot*h[3]
q_in_noreg=Q_dot_in_noreg/m_dot
h[3]=ENTHALPY(Air,T=T[3])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[3] /Pratio
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = W_dot_turb /W_dot_turbisen "turbine adiabatic efficiency, W_dot_turbisen >W_dot_turb"
```

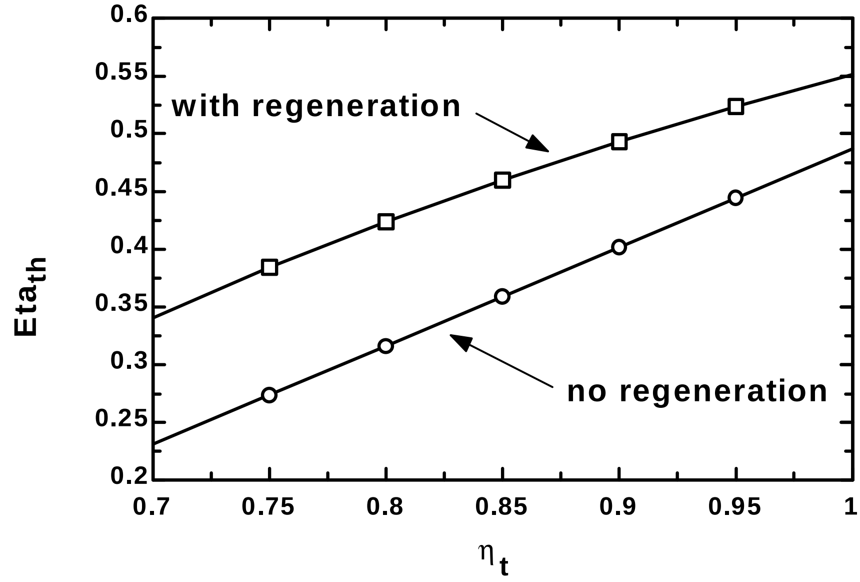
```
"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
E_dot_in -E_dot_out = DELTAE_dot_cv = 0 for steady-flow"
m_dot*h[3] = W_dot_turbisen + m_dot*h_s[4]
h_s[4]=ENTHALPY(Air,T=T_s[4])
"Actual Turbine analysis:"
m_dot*h[3] = W_dot_turb + m_dot*h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])
"Cycle analysis"
"Using the definition of the net cycle work and 1 MW = 1000 kW:"
W_dot_net*1000=W_dot_turb-W_dot_comp "kJ/s"
Eta_th_noreg=W_dot_net*1000/Q_dot_in_noreg"Cycle thermal efficiency"
Bwr=W_dot_comp/W_dot_turb"Back work ratio"
"With the regenerator the heat added in the external heat exchanger is"
m_dot*h[5] + Q_dot_in_withreg = m_dot*h[3]
q_in_withreg=Q_dot_in_withreg/m_dot
h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[2]
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[2])/(h[4]-h[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
m_dot*h[2] + m_dot*h[4]=m_dot*h[5]  + m_dot*h[6]
h[6]=ENTHALPY(Air, T=T[6])
s[6]=ENTROPY(Air,T=T[6], P=P[6])
P[6]=P[4]
"Cycle thermal efficiency with regenerator"
Eta_th_withreg=W_dot_net*1000/Q_dot_in_withreg

"The following data is used to complete the Array Table for plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=s[6]
```

$\eta_{th,noreg}$	$\eta_{th,withreg}$	$Q_{in,noreg}$ [kW]	$Q_{in,withreg}$ [kW]	W_{net} [kW]	η_c	η_t
0.2309	0.3405	442063	299766	102.1	0.892	0.7
0.2736	0.3841	442063	314863	120.9	0.892	0.75
0.3163	0.4237	442063	329960	139.8	0.892	0.8
0.359	0.4599	442063	345056	158.7	0.892	0.85
0.4016	0.493	442063	360153	177.6	0.892	0.9
0.4443	0.5234	442063	375250	196.4	0.892	0.95
0.487	0.5515	442063	390346	215.3	0.892	1



450000
415000
380000
345000
310000
275000
0.7
0.75
0.8
0.85
0.9
0.95
1
no regeneration
with regeneration
Qdot,in
ηt



0.6
0.55
0.5
0.45
0.4
0.35
0.3
0.25
0.2
0.7
0.75
0.8
0.85
0.9
0.95
1
with regeneration
no regeneration
Etath
ηt

8-82 An ideal Brayton cycle with regeneration is considered. The effectiveness of the regenerator is 100%. The net work output and the thermal efficiency of the cycle are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air are given in Table A-21.

Analysis Noting that this is an ideal cycle and thus the compression and expansion processes are isentropic, we have

$$T_1 = 300\text{K} \longrightarrow \begin{array}{l} h_1 = 300.19\text{kJ/kg} \\ P_{r_1} = 1.386 \end{array}$$

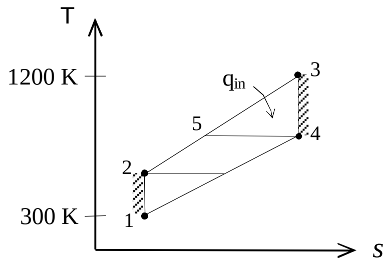
$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (10)(1.386) = 13.86 \longrightarrow h_2 = 579.87\text{kJ/kg}$$

$$T_3 = 1200\text{K} \longrightarrow \begin{array}{l} h_3 = 1277.79\text{kJ/kg} \\ P_{r_3} = 238 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{10}\right)(238) = 23.8 \longrightarrow h_4 = 675.85\text{kJ/kg}$$

$$w_{C,in} = h_2 - h_1 = 579.87 - 300.19 = 279.68\text{kJ/kg}$$

$$w_{T,out} = h_3 - h_4 = 1277.79 - 675.85 = 601.94\text{kJ/kg}$$



Thus,

$$w_{net} = w_{T,out} - w_{C,in} = 601.94 - 279.68 = \mathbf{322.26\,kJ/kg}$$

Also, $\varepsilon = 100\% \longrightarrow h_5 = h_4 = 675.85 \text{ kJ/kg}$

$$q_{in} = h_3 - h_5 = 1277.79 - 675.85 = 601.94 \text{ kJ/kg}$$

and

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{322.26 \text{ kJ/kg}}{601.94 \text{ kJ/kg}} = \mathbf{53.5\%}$$

8-83 Problem 8-82 is reconsidered. The effects of varying the isentropic efficiencies for the compressor and turbine and regenerator effectiveness on net work done and the heat supplied to the cycle for the variable specific heat case are to be investigated. The T-s diagram for the cycle is to be plotted.

```
"For both the compressor and turbine we assume adiabatic, steady-flow,
and neglect KE and PE."
"This analysis is done on a unit mass basis."
"This EES solution does not require that the isentropic efficiency of the compressor
and turbine be the same."
"See Plot Window1 for the T-s diagram and state notation for this problem."

"Input data"
T[3] = 1200"[K]"
Pratio = 10
T[1] = 300"[K]"
P[1]= 100"[kPa]"
Eta_reg = 1.0
Eta_c =0.8"Compressor isentorpic efficiency"
Eta_t =0.9"Turbien isentropic efficiency"

"Isentropic Compressor analysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[1] + w_compisen = h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])

"Actual compressor analysis:"
h[1] + w_comp = h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
h[2] + q_in_noreg = h[3]
h[3]=ENTHALPY(Air,T=T[3])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[3] /Pratio
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
h[3] = w_turbisen + h_s[4]
```

```
h_s[4]=ENTHALPY(Air,T=T_s[4])
"Actual Turbine analysis:"
h[3] = w_turb + h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])

"Cycle analysis"
w_net=w_turb-w_comp "[kJ/kg]"
Eta_th_noreg=w_net/q_in_noreg*100"[%]"    "Cycle thermal efficiency"
Bwr=w_comp/w_turb"Back work ratio"

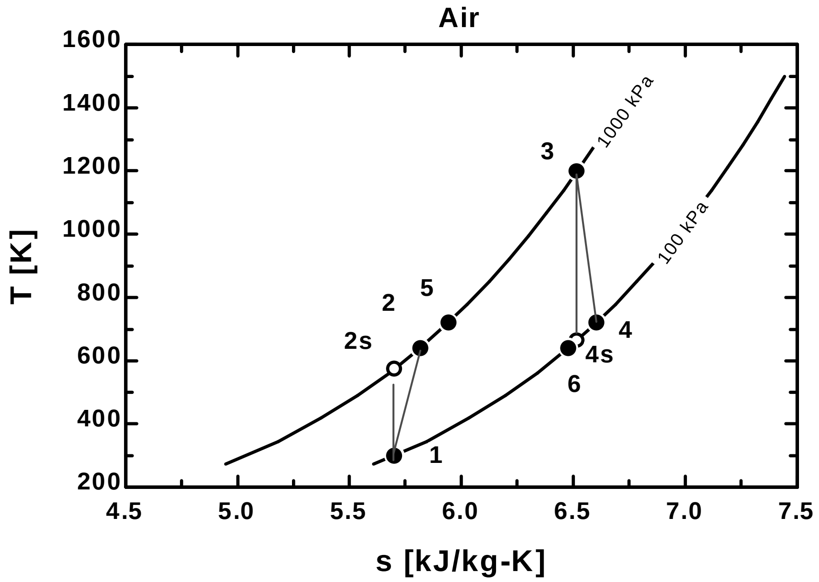
"With the regenerator the heat added in the external heat exchanger is"
h[5] + q_in_withreg = h[3]
h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[2]

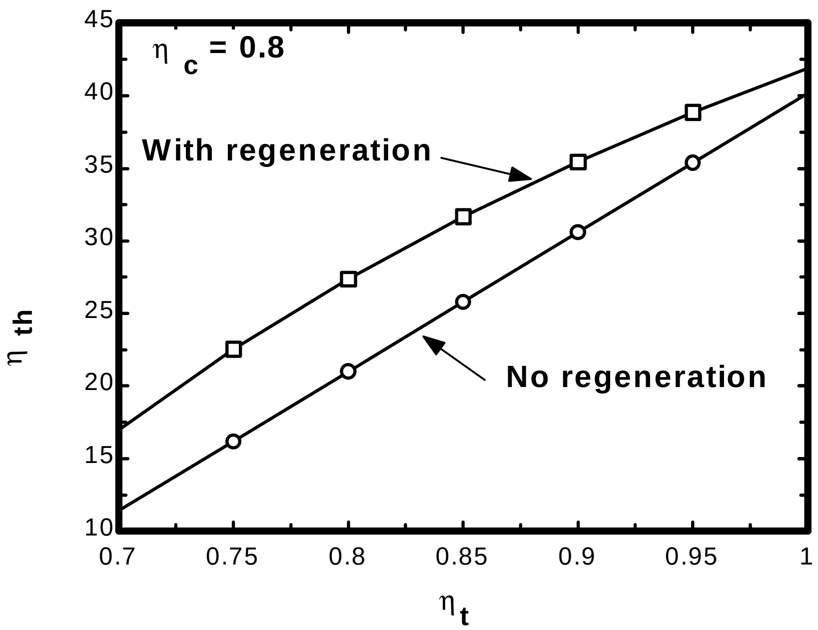
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[2])/(h[4]-h[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
h[2] + h[4]=h[5]  + h[6]
h[6]=ENTHALPY(Air, T=T[6])
s[6]=ENTROPY(Air,T=T[6], P=P[6])
P[6]=P[4]

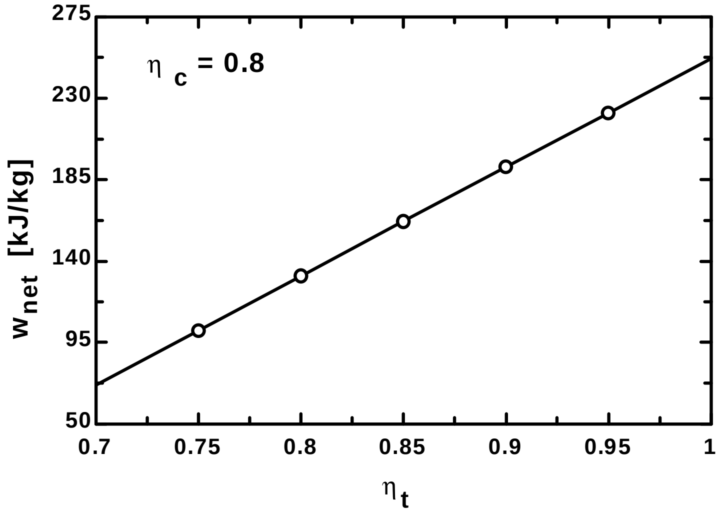
"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*100"[%]"

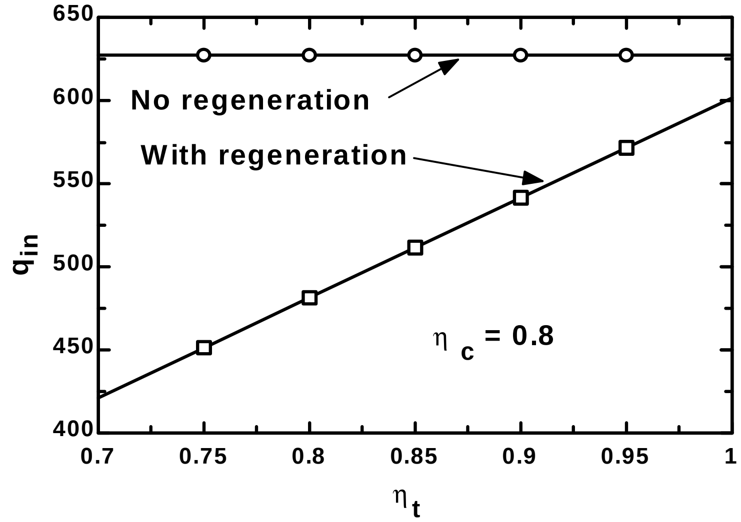
"The following data is used to complete the Array Table for plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=T[6]
```

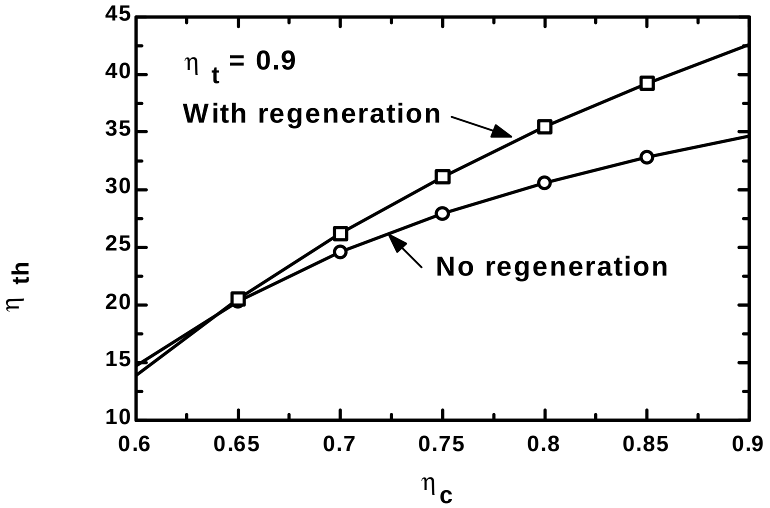
η_c	η_t	$\eta_{th,noreg}$ [%]	$\eta_{th,withreg}$ [%]	$q_{in,noreg}$ [kJ/kg]	$q_{in,withreg}$ [kJ/kg]	w_{net} [kJ/kg]
0.6	0.9	14.76	13.92	510.9	541.6	75.4
0.65	0.9	20.35	20.54	546.8	541.6	111.3
0.7	0.9	24.59	26.22	577.5	541.6	142
0.75	0.9	27.91	31.14	604.2	541.6	168.6
0.8	0.9	30.59	35.44	627.5	541.6	192
0.85	0.9	32.79	39.24	648	541.6	212.5
0.9	0.9	34.64	42.61	666.3	541.6	230.8

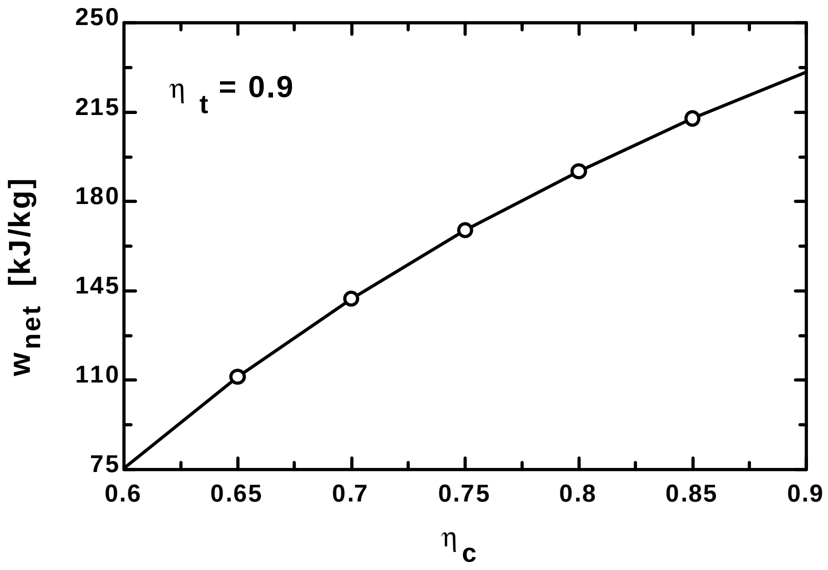

Air
T [K]
s [kJ/kg-K]
1000 kPa
100 kPa
1
2
2s
3
4
4s
5
6
1600
1400
1200
1000
800
600
400
200
4.5
5.0
5.5
6.0
6.5
7.0
7.5


η c = 0.8
With regeneration
No regeneration
η th
η t
45
40
35
30
25
20
15
10
0.7
0.75
0.8
0.85
0.9
0.95
1

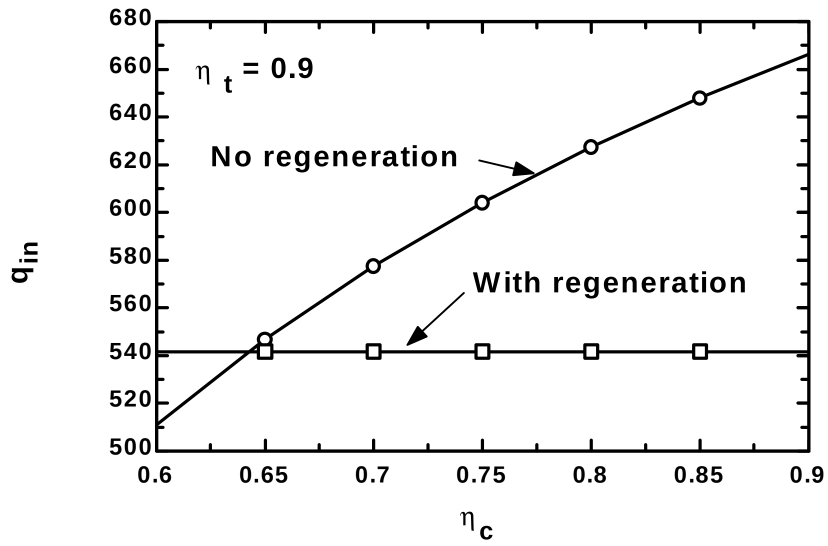

η c = 0.8
w net [kJ/kg]
η t
50
95
140
185
230
275
0.7
0.75
0.8
0.85
0.9
0.95
1


No regeneration
With regeneration
η c = 0.8
q in
η t
400
450
500
550
600
650
0.7
0.75
0.8
0.85
0.9
0.95
1


η t = 0.9
With regeneration
No regeneration
η th
η c
10
15
20
25
30
35
40
45
0.6
0.65
0.7
0.75
0.8
0.85
0.9



η t = 0.9
w_net [kJ/kg]
η_c
250
215
180
145
110
75
0.6
0.65
0.7
0.75
0.8
0.85
0.9



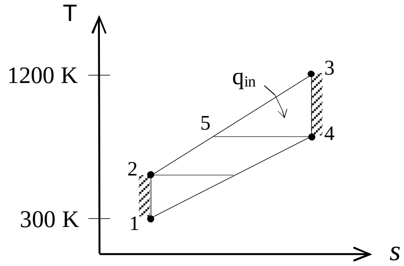
η t = 0.9
No regeneration
With regeneration
q_in
η_c
680
660
640
620
600
580
560
540
520
500
0.6
0.65
0.7
0.75
0.8
0.85
0.9

8-84 An ideal Brayton cycle with regeneration is considered. The effectiveness of the regenerator is 100%. The net work output and the thermal efficiency of the cycle are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are C_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2a).

Analysis Noting that this is an ideal cycle and thus the compression and expansion processes are isentropic, we have



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{K})(10)^{0.4/1.4} = 579.2\text{K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1200\text{K})\left(\frac{1}{10}\right)^{0.4/1.4} = 621.5\text{K}$$

$$\varepsilon = 100\% \longrightarrow T_5 = T_4 = 621.5\text{K and } T_6 = T_2 = 579.2\text{K}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p(T_6 - T_1)}{C_p(T_3 - T_5)} = 1 - \frac{T_6 - T_1}{T_3 - T_5} = 1 - \frac{579.2 - 300}{1200 - 621.5} = \mathbf{0.517}$$

$$\left(\text{or, } \eta_{th} = 1 - \left(\frac{T_1}{T_3}\right) r_p^{(k-1)/k} = 1 - \left(\frac{300}{1200}\right)(10)^{(1.4-1)/1.4} = 0.517\right)$$

Then,

$$\begin{aligned} w_{\text{net}} &= w_{\text{turb, out}} - w_{\text{comp, in}} = (h_3 - h_4) - (h_2 - h_1) \\ &= C_p[(T_3 - T_4) - (T_2 - T_1)] \\ &= (1.005\,\text{kJ/kg.K})[(1200 - 621.5) - (579.2 - 300)]\text{K} \\ &= \mathbf{300.8\ kJ/kg} \end{aligned}$$

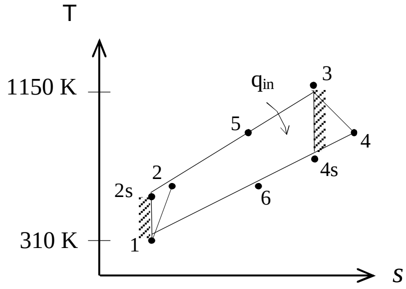
or,

$$\begin{aligned} w_{net} &= \eta_{th} q_{in} = \eta_{th}(h_3 - h_5) = \eta_{th} C_p (T_3 - T_5) \\ &= (0.517)(1.005\ \text{kJ/kg.K})(1200 - 621.5) \\ &= 300.6\ \text{kJ/kg} \end{aligned}$$

8-85 A Brayton cycle with regeneration using air as the working fluid is considered. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air are given in Table A-21.



Analysis (a) The properties of air at various states are

$$T_1 = 310\text{K} \longrightarrow \begin{array}{l} h_1 = 310.24\text{kJ/kg} \\ P_{r_1} = 1.5546 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (7)(1.5546) = 10.88 \longrightarrow h_{2s} = 541.26 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + (h_{2s} - h_1)/\eta_C = 310.24 + (541.26 - 310.24)/(0.75) = 618.26 \text{ kJ/kg}$$

$$T_3 = 1150\text{K} \longrightarrow \begin{array}{l} h_3 = 1219.25\text{kJ/kg} \\ P_{r_3} = 200.15 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{7}\right)(200.15) = 28.59 \longrightarrow h_{4s} = 711.80 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s}) = 1219.25 - (0.82)(1219.25 - 711.80) = 803.14 \text{ kJ/kg}$$

Thus, T_4 = **782.8 K**

(*b*)
$$\begin{aligned} w_{net} &= w_{T,out} - w_{C,in} = (h_3 - h_4) - (h_2 - h_1) \\ &= (1219.25 - 803.14) - (618.26 - 310.24) = \mathbf{108.09 \ kJ/kg} \end{aligned}$$

(*c*)
$$\varepsilon = \frac{h_5 - h_2}{h_4 - h_2} \longrightarrow \begin{aligned} h_5 &= h_2 + \varepsilon(h_4 - h_2) \\ &= 618.26 + (0.65)(803.14 - 618.26) \\ &= 738.43 \text{ kJ/kg} \end{aligned}$$

Then,

$$q_{in} = h_3 - h_5 = 1219.25 - 738.43 = 480.82 \text{ kJ/kg}$$

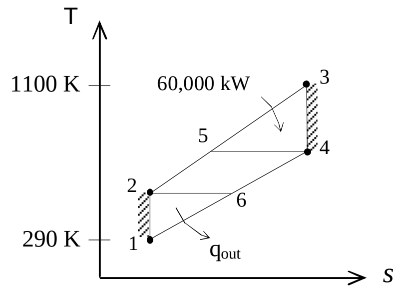
$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{108.09 \text{ kJ/kg}}{480.82 \text{ kJ/kg}} = \mathbf{22.5\%}$$

8-86 A stationary gas-turbine power plant operating on an ideal regenerative Brayton cycle with air as the working fluid is considered. The power delivered by this plant is to be determined for two cases.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas. **3** Kinetic and potential energy changes are negligible.

Properties When assuming constant specific heats, the properties of air at room temperature are C_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2a). When assuming variable specific heats, the properties of air are obtained from Table A-21.

Analysis (*a*) Assuming constant specific heats,



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (290\text{K})(8)^{0.4/1.4} = 525.3\text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1100\text{K})\left(\frac{1}{8}\right)^{0.4/1.4} = 607.2\text{ K}$$

$$\varepsilon = 100\% \longrightarrow T_5 = T_4 = 607.2\text{ K and } T_6 = T_2 = 525.3\text{ K}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p(T_6 - T_1)}{C_p(T_3 - T_5)} = 1 - \frac{T_6 - T_1}{T_3 - T_5} = 1 - \frac{525.3 - 290}{1100 - 607.2} = 0.5225$$

$$\dot{W}_{net} = \eta_T \dot{Q}_{in} = (0.5225)(90{,}000\text{ kW}) = \mathbf{47{,}027\text{ kW}}$$

(*b*) Assuming variable specific heats,

$$T_1 = 290\text{K} \longrightarrow \begin{array}{l} h_1 = 290.16\text{ kJ/kg} \\ P_{r_1} = 1.2311 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = (8)(1.2311) = 9.8488 \longrightarrow h_2 = 526.12\text{ kJ/kg}$$

$$T_3 = 1100\text{K} \longrightarrow \begin{array}{l} h_3 = 1161.07\text{ kJ/kg} \\ P_{r_3} = 167.1 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3}P_{r_3} = \left(\frac{1}{8}\right)(167.1) = 20.89 \longrightarrow h_4 = 651.37\text{ kJ/kg}$$

$$\varepsilon = 100\% \longrightarrow h_5 = h_4 = 651.37\text{ kJ/kg and } h_6 = h_2 = 526.12\text{ kJ/kg}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{h_6 - h_1}{h_3 - h_5} = 1 - \frac{526.12 - 290.16}{1161.07 - 651.37} = 0.5371$$

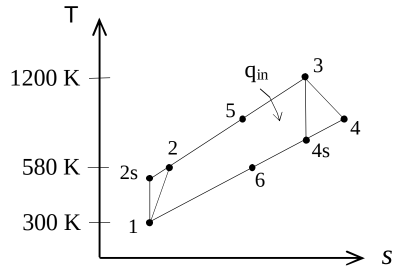
$$\dot{W}_{net} = \eta_T \dot{Q}_{in} = (0.5371)(90{,}000\text{kW}) = \mathbf{48{,}335\ \ kW}$$

8-87 A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air are given in Table A-21.

Analysis (*a*) The properties at various states are



$$r_p = P_2 / P_1 = 800/100 = 8$$

$$T_1 = 300\text{K} \longrightarrow h_1 = 300.19\text{kJ/kg}$$
$$T_2 = 580\text{K} \longrightarrow h_2 = 586.04\text{kJ/kg}$$
$$T_3 = 1200\text{K} \longrightarrow \text{h}_3 = 1277.79\text{kJ/kg}$$
$$\text{P}_{\text{r}_3} = 238.0$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(238.0) = 29.75 \longrightarrow h_{4s} = 719.75\text{kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 1277.79 - (0.86)(1277.79 - 719.75)$$
$$= 797.88 \text{ kJ/kg}$$

$$q_{regen} = \varepsilon(h_4 - h_2) = (0.72)(797.88 - 586.04) = \mathbf{152.5kJ/kg}$$

(*b*) $$w_{net} = w_{T,out} - w_{C,in} = (h_3 - h_4) - (h_2 - h_1)$$
$$= (1277.79 - 797.88) - (586.04 - 300.19) = 194.06 \text{ kJ/kg}$$

$$q_{in} = (h_3 - h_2) - q_{regen} = (1277.79 - 586.04) - 152.52 = 539.23 \text{ kJ/kg}$$

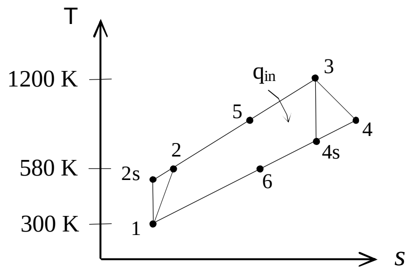
$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{194.06 \text{ kJ/kg}}{539.23 \text{ kJ/kg}} = \mathbf{36.0\%}$$

8-88 A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined. √

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are C_p = 1.005 kJ/kg.K and k = 1.4 (Table A-2a).

Analysis (a) Using the isentropic relations and turbine efficiency,



$$r_p = P_2 / P_1 = 800/100 = 8$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1200\text{K})\left(\frac{1}{8}\right)^{0.4/1.4} = 662.5\text{K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{C_p(T_3 - T_4)}{C_p(T_3 - T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3 - T_{4s})$$
$$= 1200 - (0.86)(1200 - 662.5)$$
$$= 737.8\text{K}$$

$$q_{regen} = \varepsilon(h_4 - h_2) = \varepsilon C_p(T_4 - T_2)$$
$$= (0.72)(1.005\text{kJ/kg}\cdot\text{K})(737.8 - 580)\text{K} = \mathbf{114.2kJ/kg}$$

(b)
$$w_{net} = w_{T,out} - w_{C,in} = C_p(T_3 - T_s) - C_p(T_2 - T_1)$$
$$= (1.005\text{kJ/kg}\cdot\text{K})[(1200 - 737.8) - (580 - 300)]\text{K} = 183.1\text{kJ/kg}$$

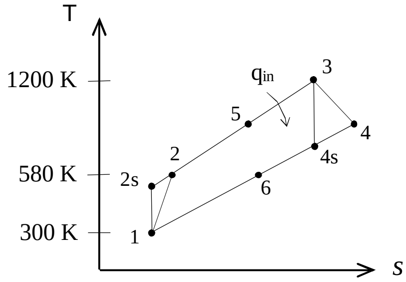
$$q_{in} = (h_3 - h_2) - q_{regen} = C_p(T_3 - T_2) - q_{regen}$$
$$= (1.005\text{kJ/kg}\cdot\text{K})(1200 - 580)\text{K} - 114.2 = 508.9\text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{183.1\text{kJ/kg}}{508.9\text{kJ/kg}} = \mathbf{36.0\%}$$

8-89 A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air are given in Table A-21.



Analysis (*a*) The properties of air at various states are

$$r_p = P_2 / P_1 = 800/100 = 8$$

$$T_1 = 300\text{K} \longrightarrow h_1 = 300.19\text{kJ/kg}$$
$$T_2 = 580\text{K} \longrightarrow h_2 = 586.04\text{kJ/kg}$$
$$T_3 = 1200\text{K} \longrightarrow \text{h}_3 = 1277.79\text{kJ/kg}$$
$$\text{P}_{\text{r}_3} = 238.0$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(238.0) = 29.75 \longrightarrow h_{4s} = 719.75\text{kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s})$$
$$= 1277.79 - (0.86)(1277.79 - 719.75)$$
$$= 797.88 \text{ kJ/kg}$$

$$q_{regen} = \varepsilon(h_3 - h_2) = (0.70)(797.88 - 586.04) = \mathbf{148.3 kJ/kg}$$

(*b*)
$$w_{net} = w_{T,out} - w_{C,in} = (h_3 - h_4) - (h_2 - h_1)$$
$$= (1277.79 - 797.88) - (586.04 - 300.19) = 194.06 \text{ kJ/kg}$$

$$q_{in} = (h_3 - h_2) - q_{regen} = (1277.79 - 586.04) - 148.3 = 543.5 \text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{194.06 \text{ kJ/kg}}{543.5 \text{ kJ/kg}} = \mathbf{35.7\%}$$

Carnot Vapor Cycle

8-90C Because excessive moisture in steam causes erosion on the turbine blades. The highest moisture content allowed is about 10%.

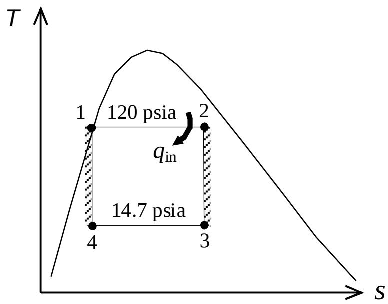
8-91C The Carnot cycle is not a realistic model for steam power plants because (1) limiting the heat transfer processes to two-phase systems to maintain isothermal conditions severely limits the maximum temperature that can be used in the cycle, (2) the turbine will have to handle steam with a high moisture content which causes erosion, and (3) it is not practical to design a compressor that will handle two phases.

8-92E 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

$$T_H = T_{\text{sat @120 psia}} = 341.3°\text{F} = 801.3 \text{ R}$$
$$T_L = T_{\text{sat @14.7 psia}} = 212°\text{F} = 672 \text{ R}$$



and

$$\eta_{\text{th, C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{672 \text{ R}}{801.3 \text{ R}} = \mathbf{16.1\%}$$

(*b*) Noting that $s_4 = s_1 = s_{f\text{@ 120 psia}} = 0.49201$ Btu/lbm·R,

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{0.49201 - 0.31212}{1.4446} = \mathbf{0.1245}$$

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

$$h_1 = h_{f\text{@ 120 psia}} = 312.67 \text{ Btu/lbm}$$
$$h_2 = h_f + x_2 h_{fg} = 312.67 + (0.95)(878.5) = 1147.25 \text{ Btu/lbm}$$

Thus,

$$q_{\text{in}} = h_2 - h_1 = 1147.25 - 312.67 = 834.58 \text{ Btu/lbm}$$

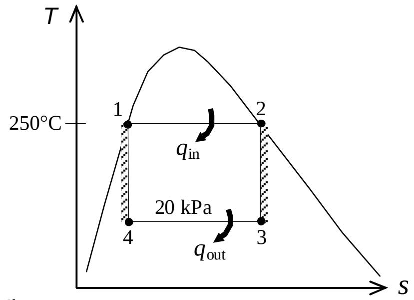
and,

$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.161)(834.58 \text{ Btu/lbm}) = \mathbf{134.4 \text{ Btu/lbm}}$$

8-93 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



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

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

$$q_{\text{in}} = h_{fg@\,250°C} = 1716.2\text{ kJ/kg}$$

Thus,

$$q_{\text{out}} = q_L = \frac{T_L}{T_H} q_{\text{in}} = \left(\frac{333.1\text{ K}}{523\text{ K}}\right)(1716.2\text{ kJ/kg}) = \mathbf{1093.1\ kJ/kg}$$

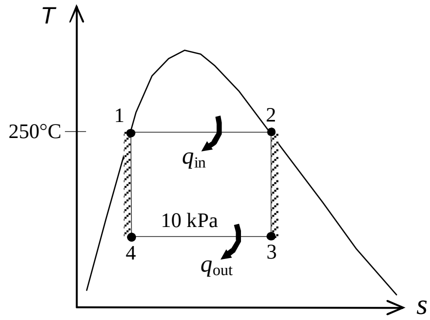
(*c*) The net work output of this cycle is

$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.363)(1716.2\text{ kJ/kg}) = \mathbf{623.0\ kJ/kg}$$

8-94 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



$$\eta_{\text{th, C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{318.8\text{ K}}{523\text{ K}} = \mathbf{39.0\%}$$

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

$$q_{\text{in}} = h_{fg@\,250°\text{C}} = 1716.2\text{ kJ/kg}$$

Thus,

$$q_{\text{out}} = q_L = \frac{T_L}{T_H} q_{\text{in}} = \left(\frac{318.8\text{K}}{523\text{ K}}\right)(1716.2\text{ kJ/kg}) = \mathbf{1046.1\ kJ/kg}$$

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

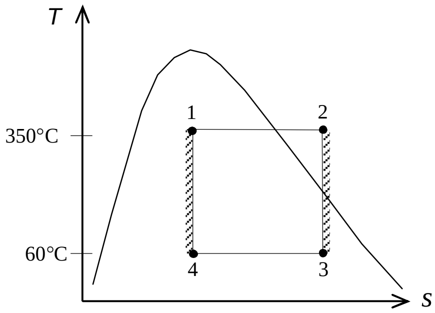
$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.390)(1716.2\text{ kJ/kg}) = \mathbf{669.3\ kJ/kg}$$

8-95 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 $s_2 = s_3 = s_f + x_3 s_{fg}$

$$= 0.8312 + 0.891 \times 7.0784 = 7.138 \text{ kJ/kg·K}$$

Thus ,

$$\left.\begin{array}{l} T_2 = 350\,°\text{C} \\ s_2 = 7.138 \text{ kJ/kg}\cdot\text{K} \end{array}\right\} P_2 = 1.40 \text{ MPa}$$

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

$$s_4 = s_f + x_4 s_{fg} = 0.8312 + (0.1)(7.0784) = 1.539 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$w_{\text{net}} = \text{Area} = (T_H - T_L)(s_3 - s_4) = (350 - 60)(7.138 - 1.539) = \mathbf{1624 \text{ kJ/kg}}$$

The Simple Rankine Cycle

8-96C 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.

8-97C Heat rejected decreases; everything else increases.

8-98C Heat rejected and heat supplied decrease; everything else increases.

8-99C The pump work remains the same, the moisture content decreases, everything else increases.

8-100C 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.

8-101C 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.

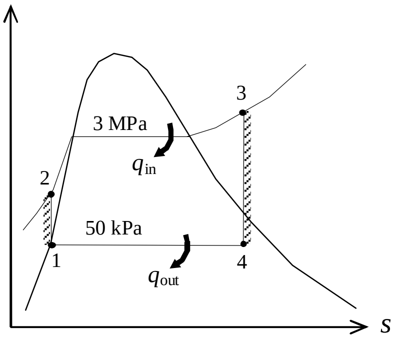
8-102C We would reject this proposal because $w_{turb} = h_1 - h_2 - q_{out}$, and any heat loss from the steam will adversely affect the turbine work output.

8-103C 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.

8-104 A steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The thermal efficiency of the cycle and the net 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 steam tables (Tables A-4, A-5, and A-6),



$$h_1 = h_{f\,@\,50\text{ kPa}} = 340.49\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,50\text{ kPa}} = 0.001030\text{ m}^3\text{/kg}$$

$$\begin{aligned}w_{p,\text{in}} &= v_1(P_2 - P_1)\\ &= (0.001030\text{ m}^3\text{/kg})(3000-50)\text{ kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)\\ &= 3.04\text{ kJ/kg}\end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 340.49 + 3.04 = 343.53\text{ kJ/kg}$$

$$\left.\begin{array}{l}P_3 = 3\text{ MPa}\\ T_3 = 400^\circ\text{C}\end{array}\right\}\begin{array}{l}h_3 = 3230.9\text{ kJ/kg}\\ s_3 = 6.9212\text{ kJ/kg}\cdot\text{K}\end{array}$$

$$\left.\begin{array}{l}P_4 = 50\text{ kPa}\\ s_4 = s_3\end{array}\right\}x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.9212-1.0910}{6.5029} = 0.8966$$
$$\begin{aligned}h_4 &= h_f + x_4 h_{fg} = 340.49 + (0.8966)(2305.4)\\ &= 2407.5\text{ kJ/kg}\end{aligned}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3230.9 - 343.53 = 2887.37\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2407.5 - 340.49 = 2067.01\text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2887.37 - 2067.01 = 820.36\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2067.01}{2887.73} = \mathbf{28.4\%}$$

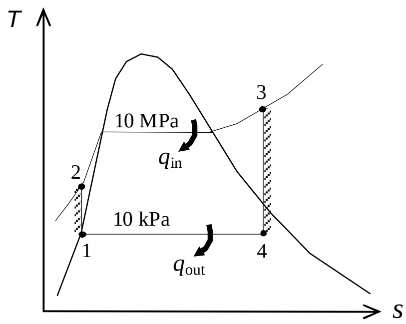
(*b*) $$\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} = (60\text{ kg/s})(820.36\text{ kJ/kg}) = \mathbf{49.2\ MW}$$

8-105 A steam power plant that operates on a simple ideal Rankine cycle is considered. The quality of the steam at the turbine exit, 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),

$$h_1 = h_{f @ 10\text{ kPa}} = 191.83\text{ kJ/kg}$$
$$v_1 = v_{f @ 10\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$



$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.00101\text{ m}^3\text{/kg})(10{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.09\text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 10.09 = 201.92\text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 10\text{ MPa} \\ T_3 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3373.7\text{ kJ/kg} \\ s_3 &= 6.5966\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 10\text{ kPa} \\ s_4 &= s_3 \end{aligned}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.5966 - 0.6493}{7.5009} = 0.793$$
$$h_4 = h_f + x_4 h_{fg} = 191.83 + (0.793)(2392.8) = 2089.3\text{ kJ/kg}$$

(*b*)
$$q_{\text{in}} = h_3 - h_2 = 3373.7 - 201.92 = 3171.78\text{ kJ / kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2089.3 - 191.83 = 1897.47\text{ kJ / kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3171.78 - 1897.47 = 1274.31\text{ kJ / kg}$$

and

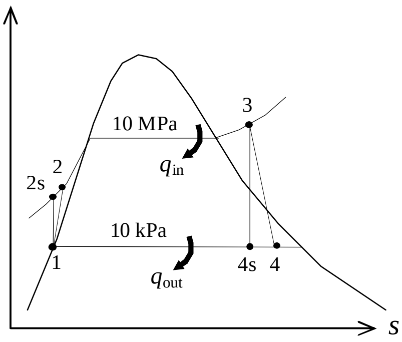
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1274.31\text{kJ/kg}}{3171.78\text{kJ/kg}} = \mathbf{40.2\%}$$

(*c*)
$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{210{,}000\text{kJ/s}}{1274.31\text{kJ/kg}} = \mathbf{165\ kg/s}$$

8-106 A steam power plant that operates on a simple nonideal Rankine cycle is considered. The quality of the steam at the turbine exit, 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),



$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.83\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1)/\eta_p \\ &= \left(0.00101\text{ m}^3\text{/kg}\right)\left(10{,}000 - 10\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/(0.85) \\ &= 11.87\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 11.87 = 203.70\text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 10\text{ MPa} \\ T_3 &= 500\,^\circ\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3373.7\text{ kJ/kg} \\ s_3 &= 6.5966\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_{4s} &= 10\text{ kPa} \\ s_{4s} &= s_3 \end{aligned}\right\} x_{4s} = \frac{s_{4s} - s_f}{s_{fg}} = \frac{6.5966 - 0.6493}{7.5009} = 0.793$$

$$h_{4s} = h_f + x_4 h_{fg} = 191.83 + (0.793)(2392.8) = 2089.3\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T\left(h_3 - h_{4s}\right)$$
$$= 3373.7 - (0.85)(3373.7 - 2089.3) = 2281.96\text{ kJ/kg}$$

$$\left.\begin{aligned} P_4 &= 10\text{ kPa} \\ h_4 &= 2281.96\text{ kJ/kg} \end{aligned}\right\} x_4 = \mathbf{0.874}$$

(*b*)

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 3373.7 - 203.70 = 3170.0\text{ kJ/kg} \\ q_{\text{out}} &= h_4 - h_1 = 2281.96 - 191.83 = 2090.13\text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 3170.0 - 2090.13 = 1079.87\text{ kJ/kg} \end{aligned}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1079.87\text{kJ/kg}}{3170.0\text{kJ/kg}} = \mathbf{34.1\%}$$

(*c*)

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{210{,}000\text{kJ/s}}{1079.87\text{kJ/kg}} = \mathbf{194.5kg/s}$$

8-107E A steam power plant that operates on a simple ideal Rankine cycle between the specified pressure limits is considered. The minimum turbine inlet temperature, the rate of heat input in the boiler, and the thermal efficiency of the cycle are to be determined.

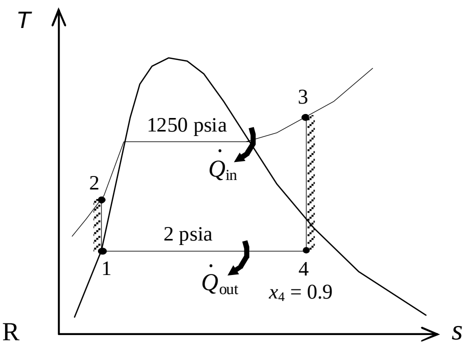
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

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

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

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.01623\text{ ft}^3\text{/lbm})(1250 - 2\text{ psia})\left(\frac{1\text{ Btu}}{5.4039\text{ psia}\cdot\text{ft}^3}\right) \\ &= 3.75\text{ Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 94.02 + 3.75 = 97.77\text{ Btu/lbm}$$



$$h_4 = h_f + x_4 h_{fg} = 94.02 + (0.9)(1022.1) = 1013.91\text{ Btu/lbm}$$
$$s_4 = s_f + x_4 s_{fg} = 0.17499 + (0.9)(1.7448) = 1.7453\text{ Btu/lbm}\cdot\text{R}$$

$$\left.\begin{array}{l} P_3 = 1250\text{ psia} \\ s_3 = s_4 \end{array}\right\} \begin{array}{l} h_3 = 1695.74\text{ Btu/lbm} \\ T_3 = \mathbf{1340.7°F} \end{array}$$

(*b*) $$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (75\text{ lbm/s})(1695.74 - 97.77) = \mathbf{119{,}848\text{ Btu/s}}$$

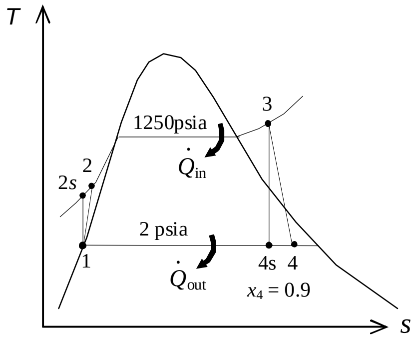
(*c*) $$\dot{Q}_{\text{out}} = \dot{m}(h_4 - h_1) = (75\text{ lbm/s})(1013.91 - 94.02) = 68{,}992\text{ Btu/s}$$

$$\eta_{th} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{68{,}992\text{ Btu/s}}{119{,}848\text{ Btu/s}} = \mathbf{42.4\%}$$

8-108E A steam power plant operates on a simple nonideal Rankine cycle between the specified pressure limits. The minimum turbine inlet temperature, the 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-4E, A-5E, and A-6E),



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

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1)/\eta_P \\ &= (0.01623 \text{ ft}^3\text{/lbm})(1250 - 2 \text{ psia})\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right)/0.85 \\ &= 4.41 \text{ Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 94.02 + 4.41 = 98.43 \text{ Btu/lbm}$$

$$h_4 = h_f + x_4 h_{fg} = 94.02 + (0.9)(1022.1) = 1013.91 \text{ Btu/lbm}$$
$$s_4 = s_f + x_4 s_{fg} = 0.17499 + (0.9)(1.7448) = 1.7453 \text{ Btu/lbm}\cdot\text{R}$$

The turbine inlet temperature is determined by trial and error ,

Try 1:
$$\left.\begin{array}{l} P_3 = 1250 \text{ psia} \\ T_3 = 900°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1438.4 \text{ Btu/lbm} \\ s_3 = 1.582 \text{ Btu/lbm.R} \end{array}$$

$$x_{4s} = \frac{s_{4s} - s_f}{s_{fg}} = \frac{s_3 - s_f}{s_{fg}} = \frac{1.582 - 0.17499}{1.7448} = 0.8064$$
$$h_{4s} = h_f + x_{4s} h_{fg} = 94.02 + (0.8064)(1022.1) = 918.2 \text{ Btu/lbm}$$
$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{1438.4 - 1013.91}{1438.4 - 918.2} = 0.8160$$

Try 2:
$$\left.\begin{array}{l} P_3 = 1250 \text{ psia} \\ T_3 = 1000°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1498.2 \text{ Btu/lbm} \\ s_3 = 1.6244 \text{ Btu/lbm.R} \end{array}$$

$$x_{4s} = \frac{s_{4s} - s_f}{s_{fg}} = \frac{s_3 - s_f}{s_{fg}} = \frac{1.6244 - 0.17499}{1.7448} = 0.8307$$
$$h_{4s} = h_f + x_{4s} h_{fg} = 94.02 + (0.8307)(1022.1) = 943.1 \text{ Btu/lbm}$$
$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{1498.2 - 1013.91}{1498.2 - 943.1} = 0.8724$$

By linear interpolation, at η_T = 0.85 we obtain T_3 = **960.3°F**. Also, h_3 = 1474.4 Btu/lbm.

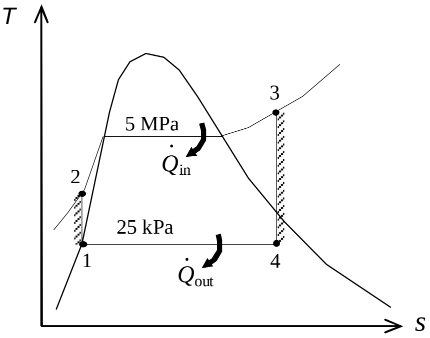
(*b*) $$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (75 \text{ lbm/s})(1474.4 - 98.43) = \mathbf{103{,}200 \text{ Btu/s}}$$

(*c*) $$\dot{Q}_{\text{out}} = \dot{m}(h_4 - h_1) = (75 \text{ lbm/s})(1013.91 - 94.02) = 69{,}000 \text{ Btu/s}$$
$$\eta_{th} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{69{,}000 \text{ Btu/s}}{103{,}200 \text{ Btu/s}} = \mathbf{33.1\%}$$

8-109 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),



$$h_1 = h_{f\,@\,25\text{ kPa}} = 271.93\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,25\text{ kPa}} = 0.00102\text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.00102\text{ m}^3\text{/kg})(5{,}000 - 25\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 5.07\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 271.93 + 5.07 = 277.0\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 5\text{ MPa} \\ T_3 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3316.2\text{ kJ/kg} \\ s_3 = 6.8186\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 25\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8186 - 0.8931}{6.9383} = 0.8540$$
$$h_4 = h_f + x_4 h_{fg} = 271.93 + (0.8540)(2346.3) = 2275.7\text{ kJ/kg}$$

The thermal efficiency is determined from

$$q_{\text{in}} = h_3 - h_2 = 3316.2 - 277.0 = 3039.2\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2275.7 - 271.93 = 2003.8\text{ kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2003.8}{3039.2} = 34.1\%$$

Thus,

$$\eta_{\text{overall}} = \eta_{th} \times \eta_{\text{comb}} \times \eta_{\text{gen}} = (0.341)(0.75)(0.96) = \mathbf{24.6\%}$$

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

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{overall}}} = \frac{300{,}000\text{ kJ/s}}{0.246} = 1{,}219{,}500\text{ kJ/s}$$

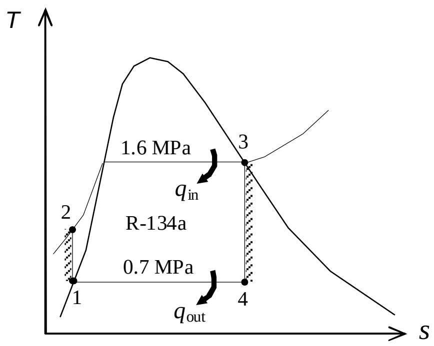
and

$$\dot{m}_{\text{coal}} = \frac{\dot{Q}_{\text{in}}}{C_{\text{coal}}} = \frac{1{,}219{,}500\text{ kJ/s}}{29{,}300\text{ kJ/kg}}\left(\frac{1\text{ ton}}{1000\text{ kg}}\right) = 0.04162\text{ tons/s} = \mathbf{149.8\ tons/h}$$

8-110 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),



$$
\begin{aligned}
h_1 &= h_{f\,@\,0.7\text{ MPa}} = 86.78 \text{ kJ/kg} \\
v_1 &= v_{f\,@\,0.7\text{ MPa}} = 0.0008328 \text{ m}^3\text{/kg} \\
w_{p,\text{in}} &= v_1(P_2 - P_1) \\
&= (0.0008328 \text{ m}^3\text{/kg})(1600 - 700 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\
&= 0.75 \text{ kJ/kg}
\end{aligned}
$$

$$h_2 = h_1 + w_{p,\text{in}} = 86.78 + 0.75 = 87.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.6 \text{ MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_3 = h_{g\,@\,1.6\text{ MPa}} = 275.33 \text{ kJ/kg} \\ s_3 = s_{g\,@\,1.6\text{ MPa}} = 0.8982 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 0.7 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{0.8982 - 0.3242}{0.9080 - 0.3242} = 0.983$$

$$h_4 = h_f + x_4 h_{fg} = 86.78 + (0.983)(175.07) = 258.87 \text{ kJ/kg}$$

Thus ,

$$
\begin{aligned}
q_{\text{in}} &= h_3 - h_2 = 275.33 - 87.53 = 187.80 \text{ kJ/kg} \\
q_{\text{out}} &= h_4 - h_1 = 258.87 - 86.78 = 172.09 \text{ kJ/kg} \\
w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 187.80 - 172.09 = 15.71 \text{ kJ/kg}
\end{aligned}
$$

and

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{15.71 \text{ kJ/kg}}{187.80 \text{ kJ/kg}} = \mathbf{8.4\%}$$

(*b*) $\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (6 \text{ kg/s})(15.71 \text{ kJ/kg}) = \mathbf{94.26\ kW}$

8-111 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),

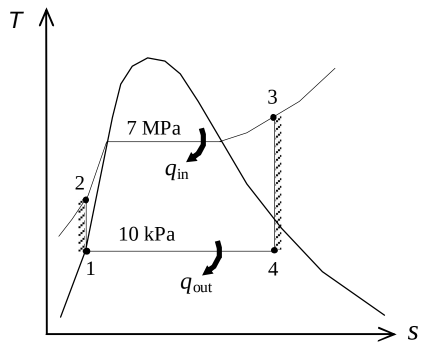
$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.83\text{ kJ/kg}$$

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

$$w_{p,\text{in}} = v_1(P_2 - P_1)$$

$$= (0.00101\ \text{m}^3\text{/kg})(7{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 7.06\text{ kJ/kg}$$



$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 7.06 = 198.89\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 7\text{ MPa} \\ T_3 = 500^\circ\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3410.5\text{ kJ/kg} \\ s_3 = 6.7975\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7975 - 0.6493}{7.5009} = 0.820$$

$$h_4 = h_f + x_4 h_{fg} = 191.83 + (0.820)(2392.8) = 2153.93\text{ kJ/kg}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 3410.5 - 198.89 = 3211.61\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 2153.93 - 191.83 = 1962.10\text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3211.61 - 1962.10 = 1249.51\text{ kJ/kg}$$

and

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1249.51\text{kJ/kg}}{3211.61\text{kJ/kg}} = \mathbf{38.9\%}$$

(*b*)

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000\text{kJ/s}}{1249.51\text{kJ/kg}} = \mathbf{36.0 kg/s}$$

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

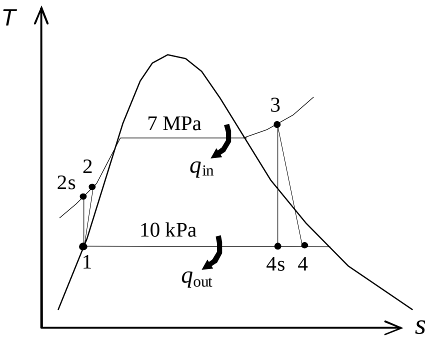
$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (36.0\ \text{kg/s})(1962.1\ \text{kJ/kg}) = 70{,}636\ \text{kJ/s}$$

$$\Delta T_{\text{cooling water}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}C)_{\text{cooling water}}} = \frac{70{,}636\ \text{kJ/s}}{(2000\ \text{kg/s})(4.18\text{ kJ/kg}\cdot{}^\circ\text{C})} = \mathbf{8.45^\circ C}$$

8-112 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),



$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.83\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1)/\eta_p \\ &= \left(0.00101\text{ m}^3\text{/kg}\right)\left(7{,}000 - 10\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/(0.87) \\ &= 8.11\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 8.11 = 199.94\text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 7\text{ MPa} \\ T_3 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3410.5\text{ kJ/kg} \\ s_3 &= 6.7975\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 10\text{ kPa} \\ s_4 &= s_3 \end{aligned}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.7975 - 0.6493}{7.5009} = 0.820$$

$$h_{4s} = h_f + x_4 h_{fg} = 191.83 + (0.820)(2392.8) = 2153.93\text{ kJ/kg}$$

$$\begin{aligned} \eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 &= h_3 - \eta_T(h_3 - h_{4s}) \\ &= 3410.5 - (0.87)(3410.5 - 2153.93) = 2317.28\text{ kJ/kg} \end{aligned}$$

Thus,

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 3410.5 - 199.94 = 3210.56\text{ kJ/kg} \\ q_{\text{out}} &= h_4 - h_1 = 2317.28 - 191.83 = 2125.45\text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 3210.56 - 2125.45 = 1085.11\text{ kJ/kg} \end{aligned}$$

and

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1085.11\text{kJ/kg}}{3210.56\text{kJ/kg}} = \mathbf{33.8\%}$$

(*b*)

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000\text{kJ/s}}{1085.11\text{kJ/kg}} = \mathbf{41.5kg/s}$$

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

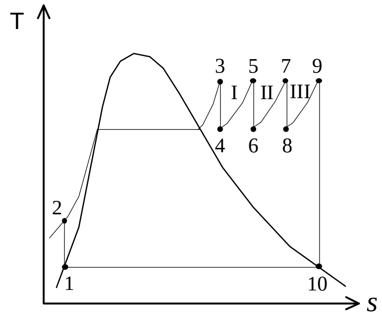
$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (41.5\text{ kg/s})(2125.45\text{ kJ/kg}) = 88{,}143\text{ kJ/s}$$

$$\Delta T_{\text{coolingwater}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}C)_{\text{coolingwater}}} = \frac{88{,}143\text{ kJ/s}}{(2000\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})} = \mathbf{10.54°C}$$

The Reheat Rankine Cycle

8-113C The pump work remains the same, the moisture content decreases, everything else increases.

8-114C The T-s diagram of the ideal Rankine cycle with 3 stages of reheat is shown on the side. The cycle efficiency will increase as the number of reheating stages increases.

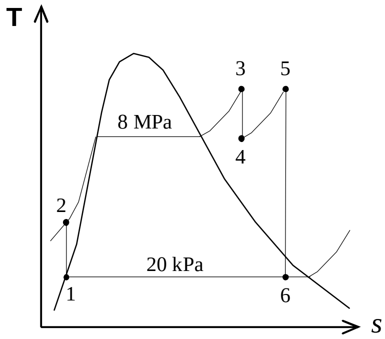


8-115C The thermal efficiency of the simple ideal Rankine cycle will probably be higher since the average temperature at which heat is added will be higher in this case.

8-116 [*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),



$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.40\text{ kJ/kg}$$

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

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= \left(0.001017\ \text{m}^3/\text{kg}\right)\left(8{,}000 - 20\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 8.12\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 251.40 + 8.12 = 259.52\text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 8\text{ MPa} \\ T_3 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3398.3\text{ kJ/kg} \\ s_3 &= 6.7240\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 3\text{ MPa} \\ s_4 &= s_3 \end{aligned}\right\} h_4 = 3104.1\text{ kJ/kg}$$

$$\left.\begin{aligned} P_5 &= 3\text{ MPa} \\ T_5 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_5 &= 3456.5\text{ kJ/kg} \\ s_5 &= 7.2338\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 20\text{ kPa} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} x_6 &= \frac{s_6 - s_f}{s_{fg}} = \frac{7.2338 - 0.8320}{7.0766} = 0.9046 \\ h_6 &= h_f + x_6 h_{fg} = 251.40 + (0.9046)(2358.3) = 2384.7\text{ kJ/kg} \end{aligned}$$

The turbine work output and the thermal efficiency are determined from

$$w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3398.3 - 3104.1 + 3456.5 - 2384.7 = \mathbf{1366\ \ kJ/kg}$$

and

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3398.3 - 259.52 + 3456.5 - 3104.1 = 3491.2\text{ kJ/kg}$$

$$w_{\text{net}} = w_{T,out} - w_{p,\text{in}} = 1366 - 8.12 = 1357.88\text{ kJ/kg}$$

Thus,

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1357.88\text{ kJ/kg}}{3491.2\text{ kJ/kg}} = \mathbf{38.9\%}$$

8-117 Problem 8-116 is reconsidered. The problem is to be solved using the "diagram window data entry feature of EES". The effects of the turbine and pump efficiencies are to be considered, and the effects of reheat on the steam quality at the low-pressure turbine exit is to be plotted. Also, the cycle is to be shown on a T-s diagram with respect to the saturation lines.

```
"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"}

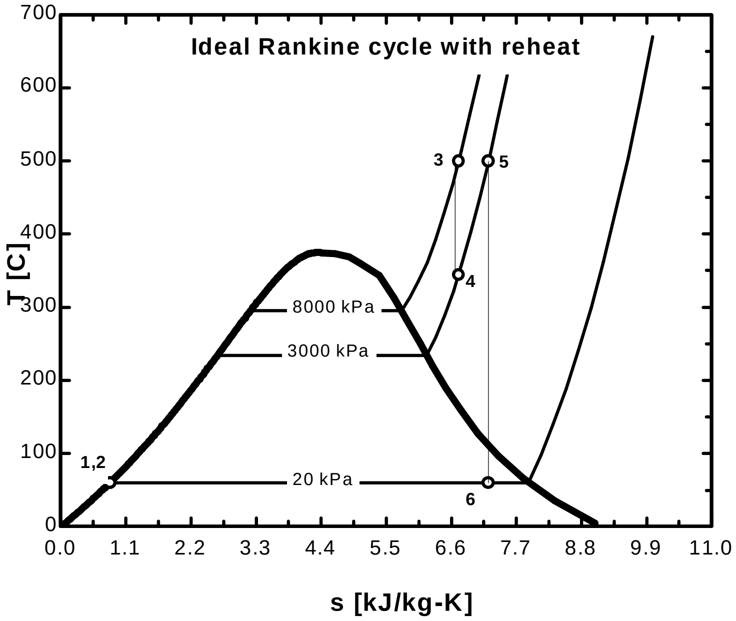
"Pump analysis"
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
    x6$=''
    if (x6>1) then x6$='(superheated)'
    if (x6<0) then x6$='(subcooled)'
end

P[1] = P[6]
P[2]=P[3]
x[1]=0      "Sat'd liquid"
h[1]=enthalpy(STEAM,P=P[1],x=x[1])
v[1]=volume(STEAM,P=P[1],x=x[1])
s[1]=entropy(STEAM,P=P[1],x=x[1])
T[1]=temperature(STEAM,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(STEAM,P=P[2],h=h[2])
s[2]=entropy(STEAM,P=P[2],h=h[2])
T[2]=temperature(STEAM,P=P[2],h=h[2])

"High Pressure Turbine analysis"
h[3]=enthalpy(STEAM,T=T[3],P=P[3])
s[3]=entropy(STEAM,T=T[3],P=P[3])
v[3]=volume(STEAM,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(STEAM,s=s_s[4],P=P[4])
Ts[4]=temperature(STEAM,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(STEAM,P=P[4],h=h[4])
s[4]=entropy(STEAM,T=T[4],P=P[4])
v[4]=volume(STEAM,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(STEAM,T=T[5],P=P[5])
h[5]=enthalpy(STEAM,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(STEAM,s=s_s[6],P=P[6])
Ts[6]=temperature(STEAM,s=s_s[6],P=P[6])
vs[6]=volume(STEAM,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(STEAM,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('steam',h=h[6],P=P[6])
s[6]=entropy('steam',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

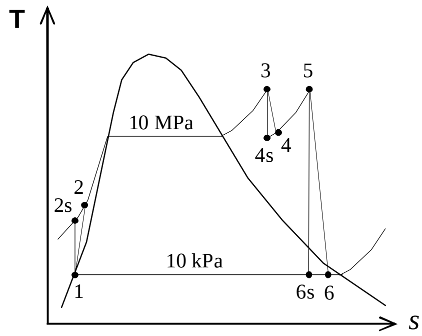
Variables in Main

Eff=0.3891	Eta_p=1
Eta_t=1	h[1]=251.3 [kJ/kg]
h[2]=259.5 [kJ/kg]	h[3]=3399 [kJ/kg]
h[4]=3104 [kJ/kg]	h[5]=3457 [kJ/kg]
h[6]=2385 [kJ/kg]	hs[4]=3104 [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=3492 [kJ/kg]
Q_out=2133 [kJ/kg]	s[1]=0.8318 [kJ/kg-K]
s[2]=0.8318 [kJ/kg-K]	s[3]=6.724 [kJ/kg-K]
s[4]=6.724 [kJ/kg-K]	s[5]=7.234 [kJ/kg-K]
s[6]=7.234 [kJ/kg-K]	s_s[4]=6.724 [kJ/kg-K]
s_s[6]=7.234 [kJ/kg-K]	T[1]=60.05 [C]
T[2]=60.39 [C]	T[3]=500 [C]
T[4]=345.4 [C]	T[5]=500 [C]
T[6]=60.05 [C]	Ts[4]=345.4 [C]
Ts[6]=60.05 [C]	v[1]=0.001017 [m^3/kg]
v[2]=0.001014 [m^3/kg]	v[3]=0.04174 [m^3/kg]
v[4]=0.08968 [m^3/kg]	vs[6]=6.93 [m^3/kg]
W_net=1359 [kJ/kg]	W_p=8.117 [kJ/kg]
W_p_s=8.117 [kJ/kg]	W_t_hp=294.7 [kJ/kg]
W_t_lp=1072 [kJ/kg]	x6s$=''
x[1]=0	x[6]=0.9048

8-118 A steam power plant that operates on a reheat Rankine cycle is considered. The quality (or temperature, if superheated) of the steam at the turbine exit, 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),



$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.83 \text{ kJ/kg}$$

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

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1)/\eta_p \\ &= \left(0.00101 \text{ m}^3\text{/kg}\right)\left(10{,}000 - 10 \text{ kPa}\right)\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 10.62 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 10.62 = 202.45 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3373.7 \text{ kJ/kg} \\ s_3 = 6.5966 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{4s} = 1 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} h_{4s} = 2782.5 \text{ kJ/kg}$$

$$\begin{aligned} \eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 &= h_3 - \eta_T\left(h_3 - h_{4s}\right) \\ &= 3373.7 - (0.80)(3373.7 - 2782.5) = 2900.74 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_5 = 1 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3478.5 \text{ kJ/kg} \\ s_5 = 7.7622 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{6s} = 10 \text{ kPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{7.7622 - 0.6493}{7.5009} = 0.948 \ \text{(at turbine exit)} \\ h_{6s} = h_f + x_{6s} h_{fg} = 191.83 + (0.948)(2392.8) = 2460.2 \text{ kJ/kg} \end{array}$$

$$\begin{aligned} \eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 &= h_5 - \eta_T\left(h_5 - h_{6s}\right) \\ &= 3478.5 - (0.80)(3478.5 - 2460.2) \\ &= 2663.86 \text{ kJ/kg} > h_g \ \text{(superheated vapor)} \end{aligned}$$

From steam tables at 10 kPa we read $T_6 = \mathbf{87.5°C}$.

(*b*) $$w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3373.7 - 2900.74 + 3478.5 - 2663.86 = 1287.6 \text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3373.7 - 202.45 + 3478.5 - 2900.74 = 3749.0 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{p,in}} = 1287.6 - 10.62 = 1277.0 \text{ kJ/kg}$$

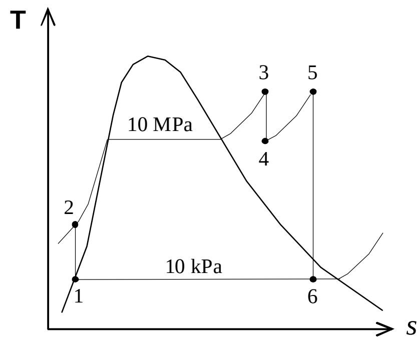
Thus the thermal efficiency is $$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1277.0 \text{ kJ/kg}}{3749.0 \text{ kJ/kg}} = \mathbf{34.1\%}$$

(*c*) The mass flow rate of the steam is $$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{80{,}000 \text{ kJ/s}}{1277.0 \text{ kJ/kg}} = \mathbf{62.6 \text{ kg/s}}$$

8-119 A steam power plant that operates on the ideal reheat Rankine cycle is considered. The quality (or temperature, if superheated) of the steam at the turbine exit, 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),



$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.83 \text{ kJ/kg}$$
$$v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101 \text{ m}^3\text{/kg}$$
$$w_{p,\text{in}} = v_1(P_2 - P_1)$$
$$= (0.00101 \text{ m}^3\text{/kg})(10{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 10.09 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 10.09 = 201.92 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3373.7 \text{ kJ/kg} \\ s_3 = 6.5966 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 1 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 2782.5 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 1 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3478.5 \text{ kJ/kg} \\ s_5 = 7.7622 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{7.7622 - 0.6493}{7.5009} = \mathbf{0.948} \ \ (\text{at turbine exit}) \\ h_6 = h_f + x_6 h_{fg} = 191.83 + (0.948)(2392.8) = 2460.2 \text{ kJ/kg} \end{array}$$

(*b*)
$$w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3373.7 - 2782.5 + 3478.5 - 2460.2 = 1609.5 \text{ kJ/kg}$$
$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3373.7 - 201.92 + 3478.5 - 2782.5 = 3867.78 \text{ kJ/kg}$$
$$w_{\text{net}} = w_{\text{T,out}} - w_{p,\text{in}} = 1609.5 - 10.09 = 1599.41 \text{ kJ/kg}$$

Thus the thermal efficiency is

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1599.41 \text{ kJ/kg}}{3867.78 \text{ kJ/kg}} = \mathbf{41.4\%}$$

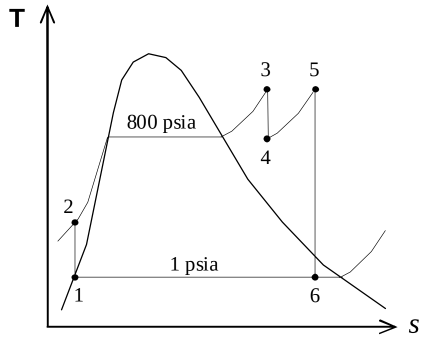
(*c*) The mass flow rate of the steam is

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{80{,}000 \text{ kJ/s}}{1599.41 \text{ kJ/kg}} = \mathbf{50.0 \text{ kg/s}}$$

8-120E A steam power plant that operates on the ideal reheat Rankine cycle is considered. The pressure at which reheating takes place, the net power output, the thermal efficiency, and the minimum mass 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 (*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),



$$h_1 = h_{\text{sat@ 1 psia}} = 69.74 \text{ Btu/lbm}$$
$$v_1 = v_{\text{sat@ 1 psia}} = 0.016136 \text{ ft}^3\text{/lbm}$$
$$T_1 = T_{\text{sat@ 1 psia}} = 101.70°\text{F}$$
$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= \left(0.016136 \text{ ft}^3\text{/lbm}\right)(800 - 1 \text{ psia})\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right) \\ &= 2.39 \text{ Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 69.74 + 2.39 = 72.13 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 800 \text{ psia} \\ T_3 = 900°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1455.6 \text{ Btu/lbm} \\ s_3 = 1.6408 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} s_4 = s_3 \\ (\text{sat.vapor}) \end{array}\right\} \begin{array}{l} h_4 = h_{g@s_g=s_4} = 1178.9 \text{ Btu/lbm} \\ P_4 = P_{\text{sat@}s_g=s_4} = \mathbf{62.81\ psia} \quad \text{(the reheat pressure)} \end{array}$$

$$\left.\begin{array}{l} P_5 = 62.81 \text{ psia} \\ T_5 = 800°\text{F} \end{array}\right\} \begin{array}{l} h_5 = 1431.1 \text{ Btu/lbm} \\ s_5 = 1.8977 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_6 = 1 \text{ psia} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{1.8977 - 0.13266}{1.8453} = 0.9565 \\ h_6 = h_f + x_6 h_{fg} = 69.74 + (0.9565)(1036) = 1060.7 \text{ Btu/lbm} \end{array}$$

(*b*)
$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 1455.6 - 72.13 + 1431.1 - 1178.9 = 1635.7 \text{ Btu/lbm}$$
$$q_{\text{out}} = h_6 - h_1 = 1060.7 - 69.74 = 991.0 \text{ Btu/lbm}$$

Thus,

$$\eta_{th} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{991.0 \text{ Btu/lbm}}{1635.7 \text{ Btu/lbm}} = \mathbf{39.4\%}$$

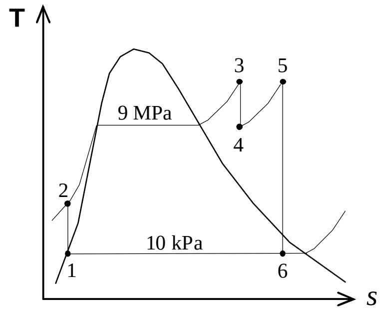
(*c*) The mass flow rate of the cooling water will be minimum when it is heated to the temperature of the steam in the condenser, which is 101.7°F,

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net}} = (1 - \eta_{th})\dot{Q}_{\text{in}} = (1 - 0.394)\left(6\times10^4 \text{ Btu/s}\right) = 3.636\times10^4 \text{ Btu/s}$$
$$\dot{m}_{\text{cool}} = \frac{\dot{Q}_{\text{out}}}{C\Delta T} = \frac{3.636\times10^4 \text{ Btu/s}}{(1.0 \text{ Btu/lbm}\cdot°\text{F})(101.7 - 45)°\text{F}} = \mathbf{641.3\ lbm/s}$$

8-121 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),



$$h_1 = h_{\text{sat@ 10 kPa}} = 191.83 \text{ kJ/kg}$$

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

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.00101 \text{ m}^3\text{/kg})(9{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 9.08 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.83 + 9.08 = 200.91 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 9 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3386.1 \text{ kJ/kg} \\ s_3 = 6.6576 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.83 + (0.90)(2392.8) = 2345.4 \text{ kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6493 + (0.90)(7.5009) = 7.4001 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_5 = 500°\text{C} \\ s_5 = s_6 \end{array}\right\} \begin{array}{l} P_5 = \mathbf{2.146 \text{ MPa}} \text{ (the reheat pressure)} \\ h_5 = 3466.0 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_4 = 2.146 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 2979.5 \text{ kJ/kg}$$

(*b*) The rate of heat supply is

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{m}[(h_3 - h_2) + (h_5 - h_4)] \\ &= (25 \text{ kJ/s})(3386.1 - 200.91 + 3466 - 2979.5) \text{kJ/kg} = \mathbf{91{,}792 \text{ kJ/s}} \end{aligned}$$

(*c*) The thermal efficiency is determined from

$$\dot{Q}_{\text{out}} = \dot{m}(h_6 - h_1) = (25 \text{ kJ/s})(2345.4 - 191.83) \text{kJ/kg} = 53{,}839 \text{ kJ/s}$$

Thus,

$$\eta_{th} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{53{,}839 \text{ kJ/s}}{91{,}792 \text{ kJ/s}} = \mathbf{41.3\%}$$

The Reversed Carnot Cycle

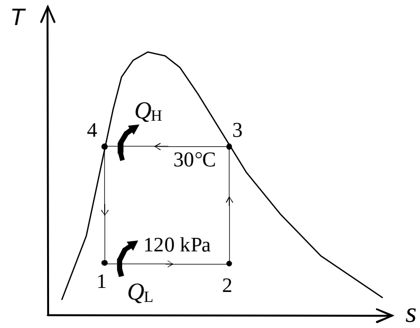
8-122C Because the compression process involves the compression of a liquid-vapor mixture which requires a compressor that will handle two phases, and the expansion process involves the expansion of high-moisture content refrigerant.

8-123 A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The coefficient of performance, the amount of heat absorbed from the refrigerated space, and the net work input are to be determined.

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

Analysis (*a*) Noting that T_H = 30°C = 303 K and $T_L = T_{\text{sat @ 120 kPa}}$ = -22.36°C = 250.6 K, the COP of this Carnot refrigerator is determined from

$$\text{COP}_{\text{R,C}} = \frac{1}{T_H / T_L - 1} = \frac{1}{(303\text{ K})/(250.6\text{ K}) - 1} = \mathbf{4.78}$$



(*b*) From the refrigerant tables (Table A-11),

$$h_3 = h_{g @ 30°\text{C}} = 263.50 \text{ kJ/kg}$$
$$h_4 = h_{f @ 30°\text{C}} = 91.49 \text{ kJ/kg}$$

Thus,

$$q_H = h_3 - h_4 = 263.50 - 91.49 = 172.01 \text{ kJ/kg}$$

and

$$\frac{q_H}{q_L} = \frac{T_H}{T_L} \longrightarrow q_L = \frac{T_L}{T_H} q_H = \left(\frac{250.6\text{ K}}{303\text{ K}}\right)(172.01\text{ kJ/kg}) = \mathbf{142.3\ \ kJ/kg}$$

(*c*) The net work input is determined from

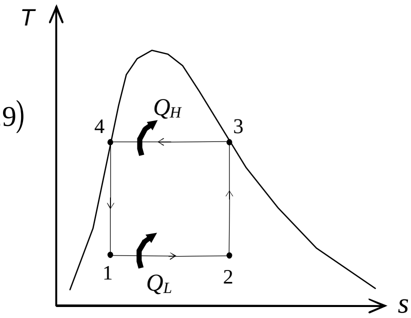
$$w_{\text{net}} = q_H - q_L = 172.01 - 142.3 = \mathbf{29.71\ \ kJ/kg}$$

8-124E A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The coefficient of performance, the quality at the beginning of the heat-absorption process, and the net work input are to be determined.

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

Analysis (*a*) Noting that $T_H = T_{\text{sat @ 90 psia}} = 72.83°\text{F} = 532.8\text{ R}$ and $T_L = T_{\text{sat @ 30 psia}} = 15.38°\text{F} = 475.4\text{ R}$.

$$\text{COP}_{\text{R,C}} = \frac{1}{T_H/T_L - 1} = \frac{1}{(532.8\text{ R})/(475.4\text{ R}) - 1} = \mathbf{8.28}$$



(*b*) Process 4-1 is isentropic, and thus

$$\begin{aligned} s_1 = s_4 &= (s_f + x_4 s_{fg})_{\text{@ 90 psia}} = 0.0729 + (0.05)(0.2172 - 0.0729) \\ &= 0.0801\text{ Btu/lbm}\cdot\text{R} \end{aligned}$$

$$x_1 = \left(\frac{s_1 - s_f}{s_{fg}}\right)_{\text{@ 30 psia}} = \frac{0.0801 - 0.0364}{0.2209 - 0.0364} = \mathbf{0.237}$$

(*c*) Remembering that on a *T-s* diagram the area enclosed represents the net work, and $s_3 = s_{g\text{ @ 90 psia}} = 0.2172\text{ Btu/lbm}\cdot\text{R}$,

$$\begin{aligned} w_{\text{net,in}} &= (T_H - T_L)(s_3 - s_4) \\ &= [72.83 - (-15.38)\text{ R}](0.2172 - 0.0801)\text{ Btu/lbm}\cdot\text{R} \\ &= \mathbf{7.88\ \ Btu/lbm} \end{aligned}$$

Ideal and Actual Vapor-Compression Cycles

8-125C Yes; the throttling process is an internally irreversible process.

8-126C To make the ideal vapor-compression refrigeration cycle more closely approximate the actual cycle.

8-127C No. Assuming the water is maintained at 10°C in the evaporator, the evaporator pressure will be the saturation pressure corresponding to this pressure, which is 1.2 kPa. It is not practical to design refrigeration or air-conditioning devices that involve such extremely low pressures.

8-128C Allowing a temperature difference of 10°C for effective heat transfer, the condensation temperature of the refrigerant should be 25°C. The saturation pressure corresponding to 25°C is 0.67 MPa. Therefore, the recommended pressure would be 0.7 MPa.

8-129C The area enclosed by the cyclic curve on a T-s diagram represents the net work input for the reversed Carnot cycle, but not so for the ideal vapor-compression refrigeration cycle. This is because the latter cycle involves an irreversible process for which the process path is not known.

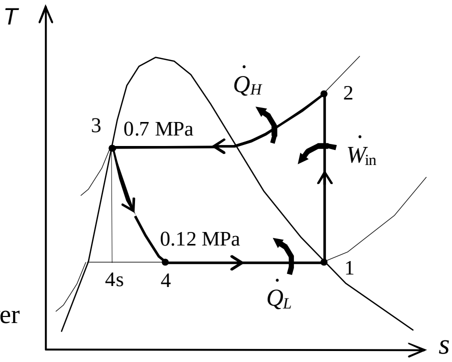
8-130C The cycle that involves saturated liquid at 30°C will have a higher COP because, judging from the T-s diagram, it will require a smaller work input for the same refrigeration capacity.

8-131C The minimum temperature that the refrigerant can be cooled to before throttling is the temperature of the sink (the cooling medium) since heat is transferred from the refrigerant to the cooling medium.

8-132 An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the rate of heat rejection to the environment, and the COP are to be determined.

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

Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),



$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 120\text{ kPa}} = 233.86\text{ kJ/kg} \\ s_1 = s_{g\ @\ 120\text{ kPa}} = 0.9354\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 270.22\text{ kJ/kg}\ (T_2 = 34.6°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7\text{ MPa} \\ \text{sat.liquid} \end{array}\right\} h_3 = h_{f\ @\ 0.7\text{ MPa}} = 86.78\text{ kJ/kg}$$

$$h_4 \cong h_3 = 86.78\text{ kJ/kg}\ (\text{throttling})$$

Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05\text{ kg/s})(233.86 - 86.78)\text{ kJ/kg} = \mathbf{7.35\ \ kW}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05\text{ kg/s})(270.22 - 233.86)\text{ kJ/kg} = \mathbf{1.82\ \ kW}$$

(*b*) The rate of heat rejection to the environment is determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.35 + 1.82 = \mathbf{9.17\ kW}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.35\text{ kW}}{1.82\text{ kW}} = \mathbf{4.04}$$

8-133 An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the rate of heat rejection to the environment, and the COP are to be determined.

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

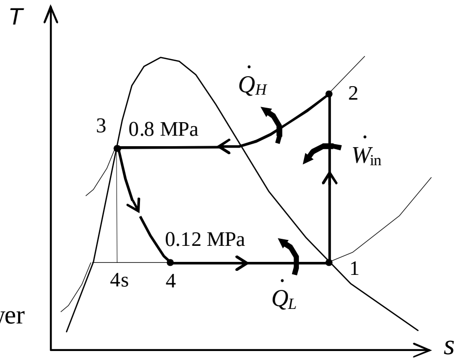
Analysis (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 120\text{ kPa}} = 233.86 \text{ kJ/kg} \\ s_1 = s_{g\ @\ 120\text{ kPa}} = 0.9354 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 273.04 \text{ kJ/kg } (T_2 = 39.4°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.8 \text{ MPa} \\ \text{sat.liquid} \end{array}\right\} h_3 = h_{f\ @\ 0.7\text{ MPa}} = 93.42 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 93.42 \text{ kJ/kg (throttling)}$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(233.86 - 93.42) \text{ kJ/kg} = \mathbf{7.02 \text{ kW}}$$

and

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(273.04 - 233.86) \text{ kJ/kg} = \mathbf{1.96 \text{ kW}}$$

(b) The rate of heat rejection to the environment is determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{in} = 7.02 + 1.96 = \mathbf{8.98 kW}$$

(c) The COP of the refrigerator is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{7.02\text{kW}}{1.96\text{kW}} = \mathbf{3.58}$$

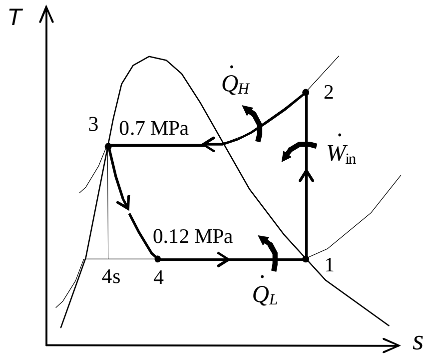
8-134 An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The throttling valve in the cycle is replaced by an isentropic turbine. The percentage increase in the COP and in the rate of heat removal from the refrigerated space due to this replacement are to be determined.

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

Analysis If the throttling valve in the previous problem is replaced by an isentropic turbine, we would have $s_{4s} = s_3 = s_{f\text{ @ 0.7 MPa}}$ = 0.3242 kJ/kg·K, and the enthalpy at the turbine exit would be

$$x_{4s} = \left(\frac{s_3 - s_f}{s_{fg}}\right)_{\text{@ 120 kPa}} = \frac{0.3242 - 0.0879}{0.9354 - 0.0879} = 0.279$$

$$h_{4s} = \left(h_f + x_{4s} h_{fg}\right)_{\text{@ 120 kPa}} = 21.32 + (0.279)(212.54) = 80.62 \text{ kJ/kg}$$



Then,

$$\dot{Q}_L = \dot{m}(h_1 - h_{4s}) = (0.05 \text{ kg/s})(233.86 - 80.62) \text{ kJ/kg} = 7.66 \text{ kW}$$

and

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_\text{in}} = \frac{7.66 \text{ kW}}{1.82 \text{ kW}} = 4.21$$

Then the percentage increase in $\dot{Q}$ and COP becomes

$$\text{Increase in } \dot{Q}_L = \frac{\Delta \dot{Q}_L}{\dot{Q}_L} = \frac{7.66 - 7.35}{7.35} = \mathbf{4.2\%}$$

$$\text{Increase in COP}_\text{R} = \frac{\Delta \text{COP}_\text{R}}{\text{COP}_\text{R}} = \frac{4.21 - 4.04}{4.04} = \mathbf{4.2\%}$$

8-135 [*Also solved by EES on enclosed CD***]** An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the end of the throttling process, the COP, and the power input to the compressor are to be determined.

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

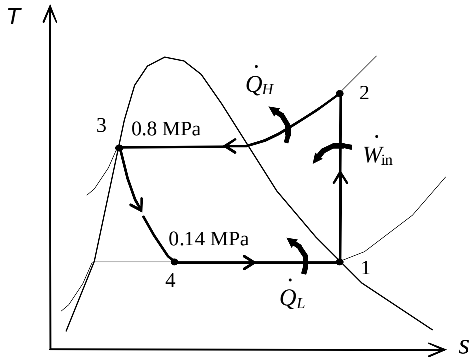
Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,140\text{ kPa}} = 236.04 \text{ kJ/kg} \\ s_1 = s_{g\,@\,140\text{ kPa}} = 0.9322 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 272.05 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.8 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.8\text{ MPa}} = 93.42 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 93.42 \text{ kJ/kg (throttling)}$$



The quality of the refrigerant at the end of the throttling process is

$$x_4 = \left(\frac{h_4 - h_f}{h_{fg}}\right)_{@\,140\text{ kPa}} = \frac{93.42 - 25.77}{210.27} = \mathbf{0.322}$$

(*b*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{q_L}{w_\text{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{236.04 - 93.42}{272.05 - 236.04} = \mathbf{3.96}$$

(*c*) The power input to the compressor is determined from

$$\dot{W}_\text{in} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{5 \text{ kW}}{3.96} = \mathbf{1.26\ kW}$$

8-136 Problem 8-135 is reconsidered. The effect of evaporator pressure on the COP and the power input as the evaporator pressure varies from 100 kPa to 400 kPa is to be investigated. The COP and the power input are to be plotted as functions of evaporator pressure.

```
"Input Data"
 {P[1]=140"kPa"}
  {P[2] = 800"kPa"
Fluid$='R134a'
Eta_c=1.0 "Compressor isentropic efficiency"
Q_dot_in=300/60"kJ/s"}

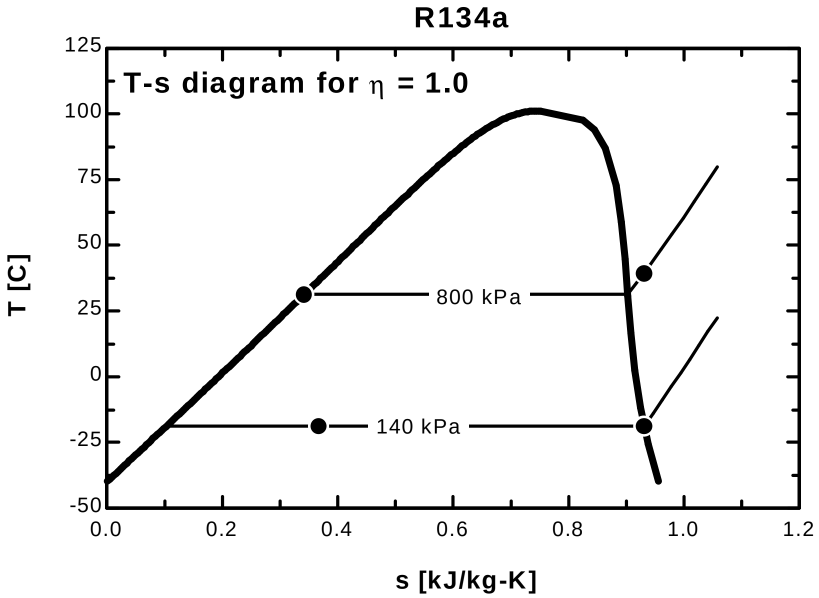
"Compressor"
x[1]=1    "assume inlet to be saturated vapor"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,h=h[1],P=P[1]) "properties for state 1"
s[1]=entropy(Fluid$,T=T[1],x=x[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
Wc=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+Wc=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_c=m_dot*Wc

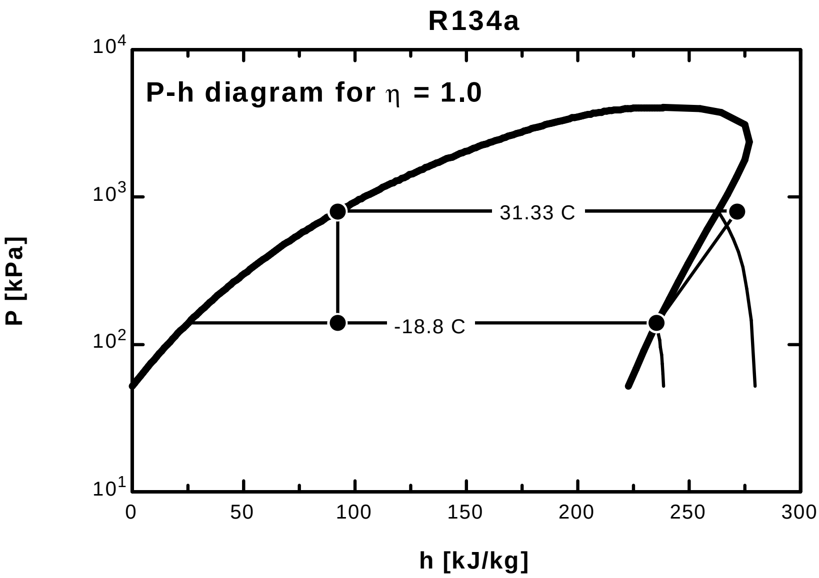
"Condenser"
P[3]=P[2] "neglect pressure drops across condenser"
T[3]=temperature(Fluid$,h=h[3],P=P[3]) "properties for state 3"
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,T=T[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
Q_dot_out=m_dot*Qout

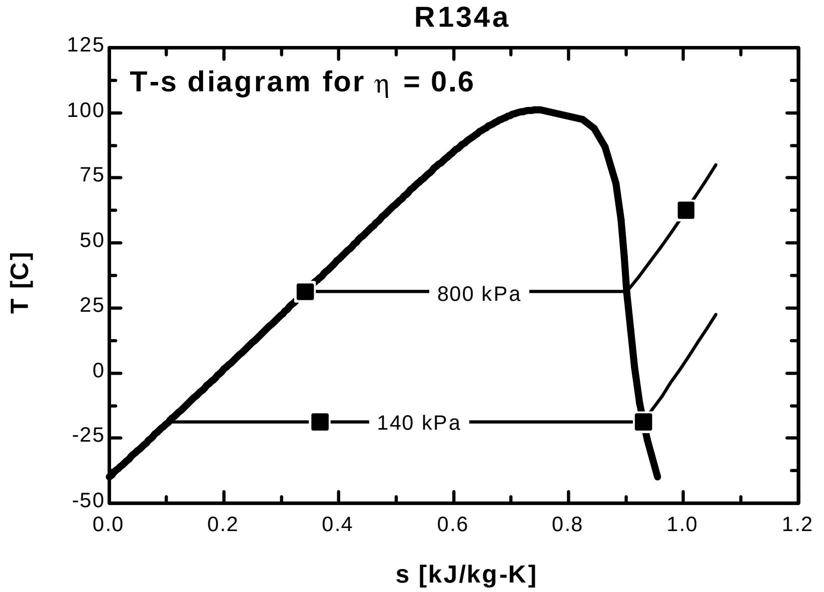
"Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])

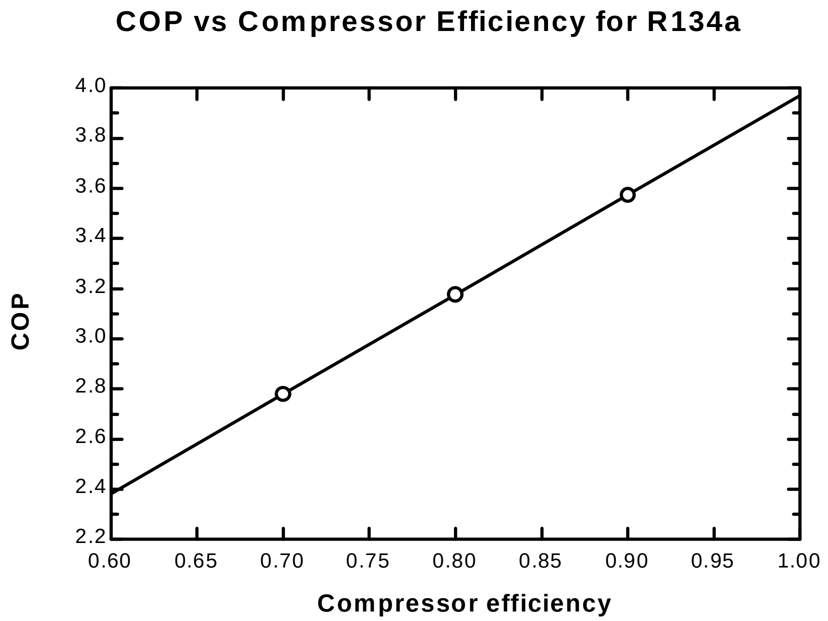
"Evaporator"
P[4]=P[1] "neglect pressure drop across evaporator"
Q_in + h[4]=h[1] "energy balance on evaporator"
Q_dot_in=m_dot*Q_in
COP=Q_dot_in/W_dot_c "definition of COP"
COP_plot = COP
W_dot_in = W_dot_c
```

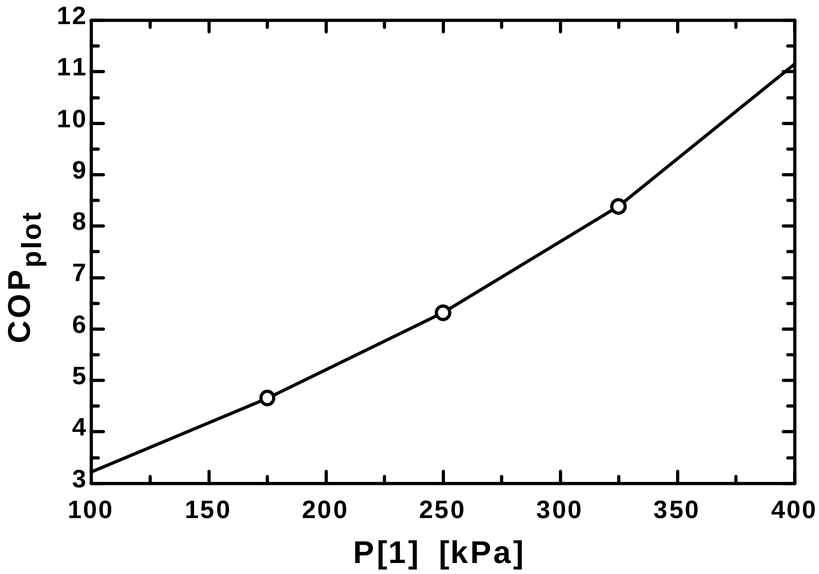
P_1 [kPa]	COP_{plot}	W_{in} [kW]
100	3.22	1.553
175	4.658	1.073
250	6.316	0.7916
325	8.387	0.5961
400	11.15	0.4484

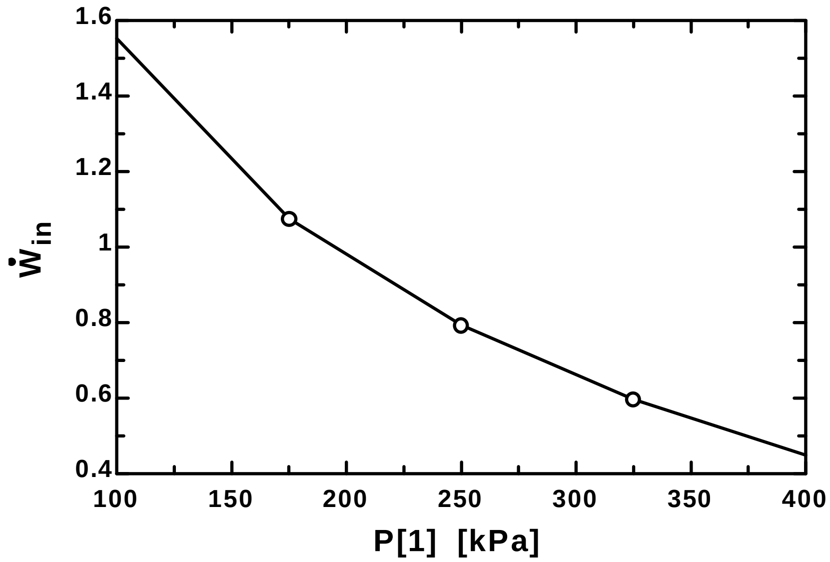

R134a
T-s diagram for η = 1.0
800 kPa
140 kPa
T [C]
s [kJ/kg-K]
125
100
75
50
25
0
-25
-50
0.0
0.2
0.4
0.6
0.8
1.0
1.2


R134a
P-h diagram for η = 1.0
31.33 C
-18.8 C
P [kPa]
h [kJ/kg]
10^4
10^3
10^2
10^1
0
50
100
150
200
250
300


R134a
T-s diagram for η = 0.6
800 kPa
140 kPa
T [C]
s [kJ/kg-K]


COP vs Compressor Efficiency for R134a
COP
Compressor efficiency


COP_plot
P[1] [kPa]
12
11
10
9
8
7
6
5
4
3
100
150
200
250
300
350
400


Ẇin
P[1] [kPa]
1.6
1.4
1.2
1
0.8
0.6
0.4
100
150
200
250
300
350
400

8-137 A nonideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the end of the throttling process, the COP, the power input to the compressor, and the irreversibility rate associated with the compression process are to be determined.

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

Analysis (*a*) The refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

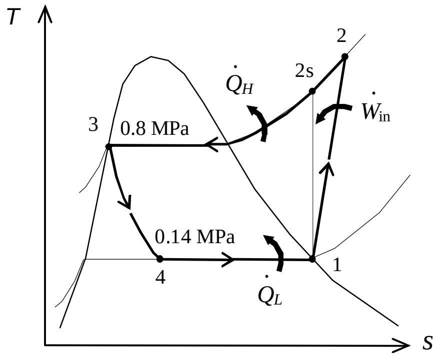
$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,140\text{ kPa}} = 236.04 \text{ kJ/kg} \\ s_1 = s_{g\,@\,140\text{ kPa}} = 0.9322 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 272.05 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + (h_{2s} - h_1)/\eta_C$$
$$= 236.04 + (272.05 - 236.04)/(0.85)$$
$$= 278.40 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.8 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.8\text{ MPa}} = 93.42 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 93.42 \text{ kJ/kg (throttling)}$$



The quality of the refrigerant at the end of the throttling process is

$$x_4 = \left(\frac{h_4 - h_f}{h_{fg}}\right)_{@\,140\text{ kPa}} = \frac{93.42 - 25.77}{210.27} = \mathbf{0.322}$$

(*b*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{q_L}{w_\text{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{236.04 - 93.42}{278.40 - 236.04} = \mathbf{3.37}$$

(*c*) The power input to the compressor is determined from

$$\dot{W}_\text{in} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{5 \text{ kW}}{3.37} = \mathbf{1.48\ kW}$$

The exergy destruction associated with the compression process is determined from

$$\dot{X}_{destroyed} = T_0 \dot{S}_\text{gen} = T_0 \dot{m}\left(s_2 - s_1 + \cancelto{0}{\frac{q_\text{surr}}{T_0}}\right) = T_0 \dot{m}(s_2 - s_1)$$

where

$$\dot{m} = \frac{\dot{Q}_L}{q_L} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{5 \text{ kJ/s}}{(236.04 - 93.42) \text{ kJ/kg}} = 0.03506 \text{ kg/s}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ h_2 = 278.40 \text{ kJ/kg} \end{array}\right\} s_2 = 0.9522 \text{ kJ/kg}\cdot\text{K}$$

Thus, $\dot{X}_{destroyed} = (298 \text{ K})(0.03506 \text{ kg/s})(0.9522 - 0.9322) \text{ kJ/kg}\cdot\text{K} = \mathbf{0.209\ kW}$

8-138 A refrigerator with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the isentropic efficiency of the compressor, and the COP of the refrigerator 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-12 and A-13),

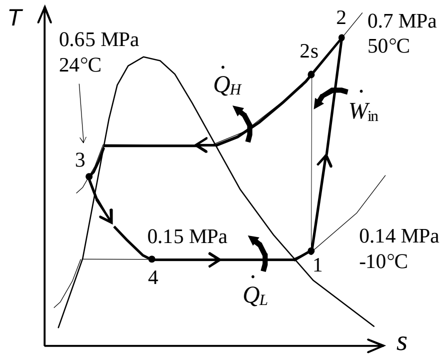
$$\left.\begin{array}{l} P_1 = 0.14 \text{ MPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 243.40 \text{ kJ/kg} \\ s_1 = 0.9606 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ T_2 = 50°\text{C} \end{array}\right\} h_2 = 286.35 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{2s} = 0.7 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 278.06 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.65 \text{ MPa} \\ T_3 = 24°\text{C} \end{array}\right\} h_3 = h_{f\,@\,24°\text{C}} = 82.90 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 82.90 \text{ kJ/kg (throttling)}$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.12 \text{ kg/s})(243.40 - 82.90) \text{ kJ/kg} = \mathbf{19.3 \text{ kW}}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.12 \text{ kg/s})(286.35 - 243.40) \text{ kJ/kg} = \mathbf{5.15 \text{ kW}}$$

(*b*) The adiabatic efficiency of the compressor is determined from

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{278.06 - 243.40}{286.35 - 243.40} = \mathbf{80.7\%}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{19.3 \text{kW}}{5.15 \text{kW}} = \mathbf{3.75}$$

8-139E An ice-making machine operates on the ideal vapor-compression refrigeration cycle, using refrigerant-134a as the working fluid. The power input to the ice machine is to be determined.

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

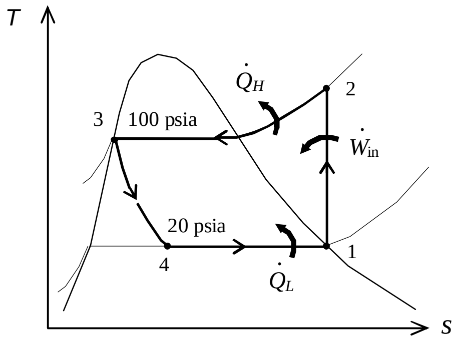
Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 20 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 20\text{ psia}} = 101.39 \text{ Btu/lbm} \\ s_1 = s_{g\ @\ 20\text{ psia}} = 0.2227 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ s_2 = s_1 \end{array}\right\} h_2 = 115.64 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 100 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\ @\ 100\text{ psia}} = 36.99 \text{ Btu/lbm}$$

$$h_4 \cong h_3 = 36.99 \text{ Btu/lbm (throttling)}$$



The cooling load of this refrigerator is

$$\dot{Q}_L = \dot{m}_{\text{ice}}(\Delta h)_{\text{ice}} = (20/3600 \text{ lbm/s})(169 \text{ Btu/lbm}) = 0.9389 \text{ Btu/s}$$

Then the mass flow rate of the refrigerant and the power input become

$$\dot{m}_R = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{0.9389 \text{ Btu/s}}{(101.39 - 36.99) \text{ Btu/lbm}} = 0.0146 \text{ lbm/s}$$

and

$$\dot{W}_{\text{in}} = \dot{m}_R (h_2 - h_1) = (0.0146 \text{ lbm/s})(115.64 - 101.39) \text{ Btu/lbm}\left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}}\right) = \mathbf{0.294\ hp}$$

8-140 A refrigerator with refrigerant-134a as the working fluid is considered. The power input to the compressor, the rate of heat removal from the refrigerated space, and the pressure drop and the rate of heat gain in the line between the evaporator and the compressor 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-12 and A-13),

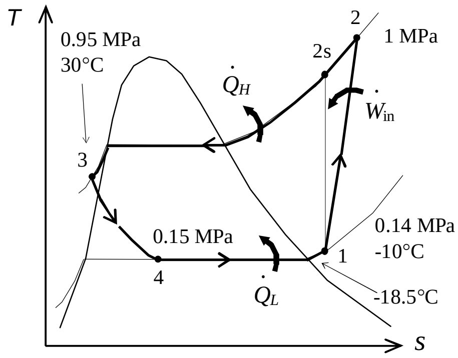
$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 243.40 \text{ kJ/kg} \\ s_1 = 0.9606 \text{ kJ/kg} \cdot \text{K} \\ v_1 = 0.14549 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.0 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 286.04 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.95 \text{ MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\ @\ 30\ °\text{C}} = 91.49 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 91.49 \text{ kJ/kg (throttling)}$$

$$\left.\begin{array}{l} T_5 = -18.5°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} P_5 = 0.14187 \text{ MPa} \\ h_5 = 236.23 \text{ kJ/kg} \end{array}$$



Then the mass flow rate of the refrigerant and the power input becomes

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.3/60 \text{ m}^3\text{/s}}{0.14549 \text{ m}^3\text{/kg}} = 0.0344 \text{ kg/s}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_{2s} - h_1)/\eta_C = (0.0344 \text{ kg/s})[(286.04 - 243.40) \text{ kJ/kg}]/(0.78) = \mathbf{1.88\ kW}$$

(*b*) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}(h_5 - h_4) = (0.0344 \text{ kg/s})(236.23 - 91.49) \text{ kJ/kg} = \mathbf{4.98\ kW}$$

(*c*) The pressure drop and the heat gain in the line between the evaporator and the compressor are

$$\Delta P = P_5 - P_1 = 141.87 - 140 = \mathbf{1.87}$$

and

$$\dot{Q}_{\text{gain}} = \dot{m}(h_1 - h_5) = (0.0344 \text{ kg/s})(243.40 - 236.23) \text{ kJ/kg} = \mathbf{0.247\ \ kW}$$

8-141 Problem 8-140 is reconsidered. The effects of varying the compressor isentropic efficiency over the range 60 percent to 100 percent and the compressor inlet volume flow rate from 0.1 m3/min to 1.0 m3/min on the power input and the rate of refrigeration are to be investigated. The rate of refrigeration and the power input to the compressor are to be plotted as functions of compressor efficiency for compressor inlet volume flow rates of 0.1, 0.5, and 1.0 m3/min.

```
"Input Data"
{T[5]=-18.5"[C]""T at evaporator exit"
 P[1]=140"[kPa]"
T[1] = -10"[C]"}
{V_dot[1]=0.1"[m^3/min]"}
  {P[2] = 1000"[kPa]"
P[3]=950"[kPa]"
T[3] = 30"[C]"}

{Fluid$='R134a'}
{Eta_c=0.78 }"Compressor isentropic efficiency"

"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],T=T[1])  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],T=T[1])
v[1]=volume(Fluid$,P=P[1],T=T[1])"[m^3/kg]"
m_dot=V_dot[1]/v[1]*convert(m^3/min,m^3/s)"[kg/s]"
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"

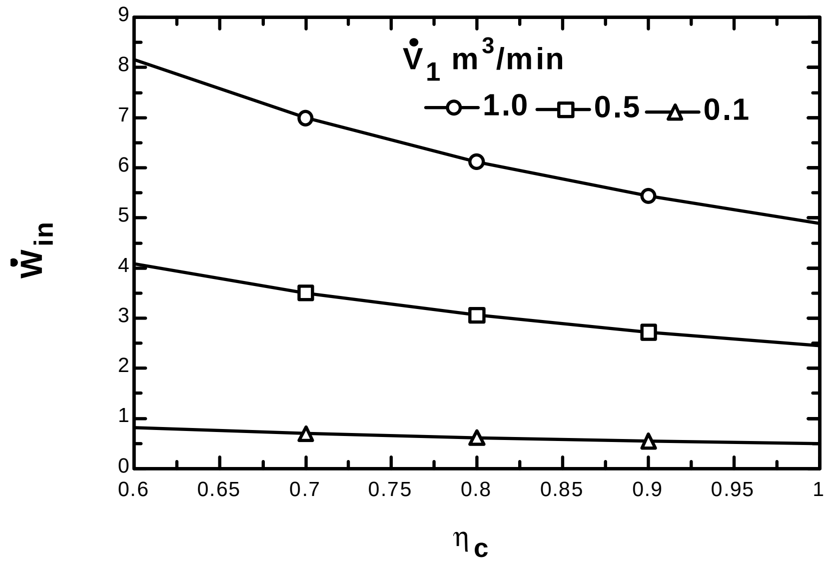
Wc=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+Wc=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_c=m_dot*Wc

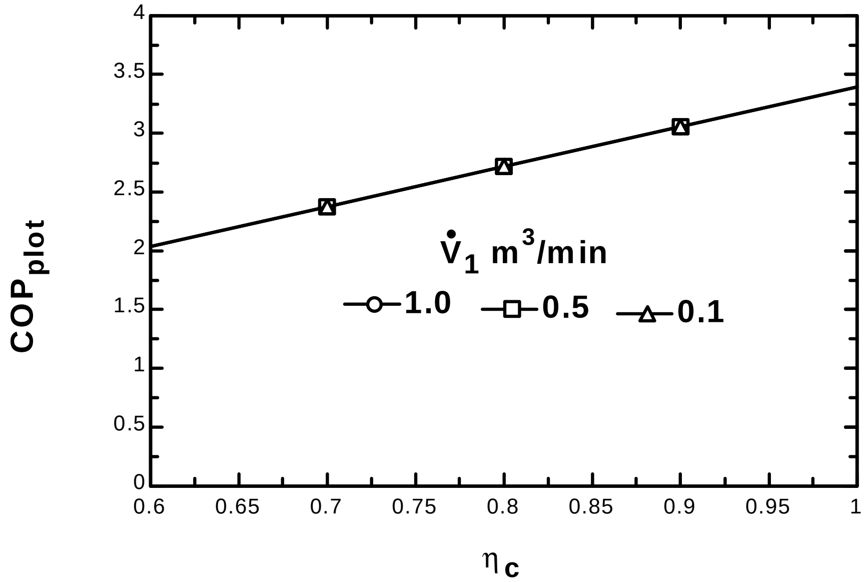
"Condenser"
h[3]=enthalpy(Fluid$,P=P[3],T=T[3]) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],T=T[3])
h[2]=Qout+h[3] "energy balance on condenser"
Q_dot_out=m_dot*Qout

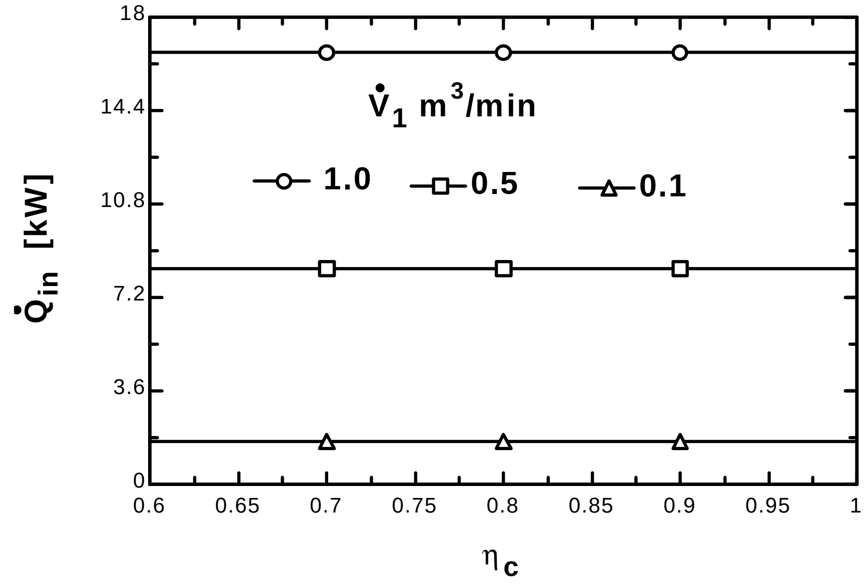
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])

"Evaporator"
P[4]=pressure(Fluid$,T=T[5],x=0)"pressure=Psat at evaporator exit temp."
P[5] = P[4]
h[5]=enthalpy(Fluid$,T=T[5],x=1) "properties for state 5"
Q_in + h[4]=h[5] "energy balance on evaporator"
Q_dot_in=m_dot*Q_in
COP=Q_dot_in/W_dot_c "definition of COP"
COP_plot = COP
W_dot_in = W_dot_c
Q_dot_line5to1=m_dot*(h[1]-h[5])"[kW]"
```

COP_{plot}	W_{in} [kW]	Q_{in} [kW]	η_c [kW]
2.039	0.8162	1.664	0.6
2.378	0.6996	1.664	0.7
2.718	0.6121	1.664	0.8
3.058	0.5441	1.664	0.9
3.398	0.4897	1.664	1







18
14.4
10.8
7.2
3.6
0
0.6
0.65
0.7
0.75
0.8
0.85
0.9
0.95
1
$\dot{V}_1$ m³/min
1.0
0.5
0.1
$\dot{Q}_{in}$ [kW]
η_c

Heat Pump Systems

8-142C A heat pump system is more cost effective in Miami because of the low heating loads and high cooling loads at that location.

8-143C A water-source heat pump extracts heat from water instead of air. Water-source heat pumps have higher COPs than the air-source systems because the temperature of water is higher than the temperature of air in winter.

8-144E A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. The power input to the heat pump and the electric power saved by using a heat pump instead of a resistance heater are to be determined.

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

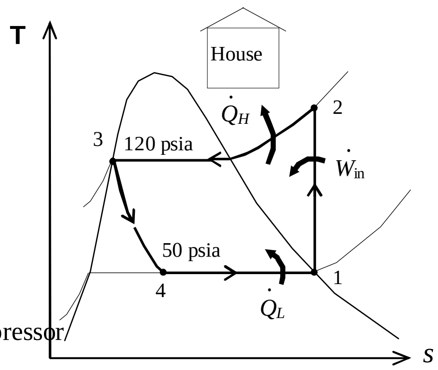
Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12E and A-13E),

$$\left.\begin{array}{l} P_1 = 50 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,50\text{ psia}} = 107.43 \text{ Btu/lbm} \\ s_1 = s_{g\,@\,50\text{ psia}} = 0.2189 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 120 \text{ psia} \\ s_2 = s_1 \end{array}\right\} h_2 = 115.15 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 120 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,120\text{ psia}} = 40.91 \text{ Btu/lbm}$$

$$h_4 \cong h_3 = 40.91 \text{ Btu/lbm (throttling)}$$



The mass flow rate of the refrigerant and the power input to the compressor are determined from

$$\dot{m} = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{60{,}000/3600 \text{ Btu/s}}{(115.15 - 40.91) \text{ Btu/lbm}} = 0.225 \text{ lbm/s}$$

and

$$\begin{aligned} \dot{W}_{in} &= \dot{m}(h_2 - h_1) = (0.225 \text{ kg/s})(115.15 - 107.43) \text{ Btu/lbm} = 1.73 \text{ Btu/s} \\ &= \mathbf{2.45 \text{ hp}} \text{ since } 1 \text{ hp} = 0.7068 \text{ Btu/s} \end{aligned}$$

The electrical power required without the heat pump is

$$\dot{W}_e = \dot{Q}_H = (60{,}000/3600 \text{ Btu/s})\left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}}\right) = 23.58 \text{ hp}$$

Thus,

$$\begin{aligned} \dot{W}_{saved} &= \dot{W}_e - \dot{W}_{in} = 23.58 - 2.45 = \mathbf{21.13 \text{ hp}} \\ &= 15.76 \text{ kW since } 1 \text{ hp} = 0.7457 \text{ kW} \end{aligned}$$

8-145 A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. The power input to the heat pump is to be determined.

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

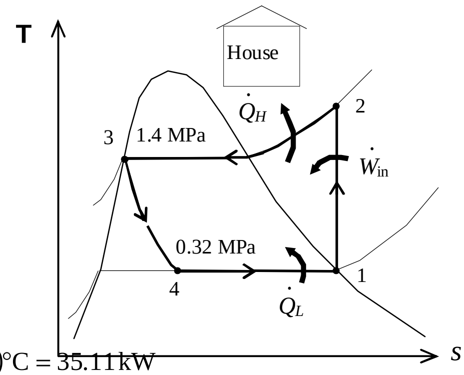
Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l}P_1 = 320\text{ kPa}\\ \text{sat. vapor}\end{array}\right\}\begin{array}{l}h_1 = h_{g\ @320\text{ kPa}} = 248.66\text{ kJ/kg}\\ s_1 = s_{g\ @\ 320\text{ kPa}} = 0.9177\text{ kJ/kg}\cdot\text{K}\end{array}$$

$$\left.\begin{array}{l}P_2 = 1.4\text{ MPa}\\ s_2 = s_1\end{array}\right\}h_2 = 279.14\text{ kJ/kg}$$

$$\left.\begin{array}{l}P_3 = 1.4\text{ MPa}\\ \text{sat. liquid}\end{array}\right\}h_3 = h_{f\ @1.4\text{ MPa}} = 125.26\text{ kJ/kg}$$

$$h_4 \cong h_3 = 125.26\text{ kJ/kg}\quad(\text{throttling})$$



The heating load of this heat pump is determined from

$$\dot{Q}_H = [\dot{m}C(T_2 - T_1)]_{\text{water}} = (0.24\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})(50-15)°\text{C} = 35.11\text{ kW}$$

and

$$\dot{m}_R = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{35.11\text{ kJ/s}}{(279.14-125.26)\text{ kJ/kg}} = 0.2282\text{ kg/s}$$

Then,

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) = (0.2282\text{ kg/s})(279.14-248.66)\text{ kJ/kg} = \mathbf{6.96\ kW}$$

8-146 A heat pump with refrigerant-134a as the working fluid heats a house by using underground water as the heat source. The power input to the heat pump, the rate of heat absorption from the water, and the increase in electric power input if an electric resistance heater is used instead of a heat pump are to be determined.

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

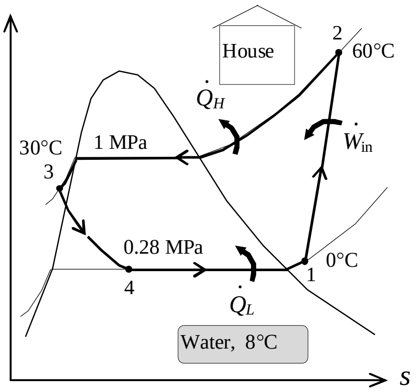
Analysis (*a*) From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 280\text{ kPa} \\ T_1 = 0°\text{C} \end{array}\right\} h_1 = 247.64\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1.0\text{ MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 291.36\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.0\text{ MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\,@\,30°\text{C}} = 91.49\text{ kJ/kg}$$

$$h_4 \cong h_3 = 91.49\text{ kJ/kg}\ \ (\text{throttling})$$



The mass flow rate of the refrigerant is

$$\dot{m}_R = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{60{,}000/3{,}600\text{ kJ/s}}{(291.36 - 91.49)\text{ kJ/kg}} = 0.0834\text{ kg/s}$$

Then the power input to the compressor becomes

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.0834\text{ kg/s})(291.36 - 247.64)\text{ kJ/kg} = \mathbf{3.65\ kW}$$

(*b*) The rate of hat absorption from the water is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.0834\text{ kg/s})(247.64 - 91.49)\text{ kJ/kg} = \mathbf{13.02\ kW}$$

(*c*) The electrical power required without the heat pump is

$$\dot{W}_e = \dot{Q}_H = 60{,}000 / 3600\text{ kJ/s} = 16.67\text{ kW}$$

Thus,

$$\dot{W}_{\text{increase}} = \dot{W}_e - \dot{W}_{\text{in}} = 16.67 - 3.65 = \mathbf{13.02kW}$$

8-147 Problem 8-146 is reconsidered. The effects of varying the compressor isentropic efficiency over the range 60 percent to 100 percent is to be investigated. The power input to the compressor and the electric power saved by using a heat pump rather than electric resistance heating are to be plotted as functions of compressor efficiency.

```
"Input Data"
"Input Data is supplied in the diagram window"
 {P[1]=280"[kPa]"
T[1] = 0"[C]"
P[2] = 1000"[kPa]"
T[3] = 30"[C]"
Q_dot_out = 60000"[kJ/h]"
Fluid$='R134a'
"Use ETA_c = 0.623 to obtain T[2] = 60C"
Eta_c=1.0 "Compressor isentropic efficiency"}

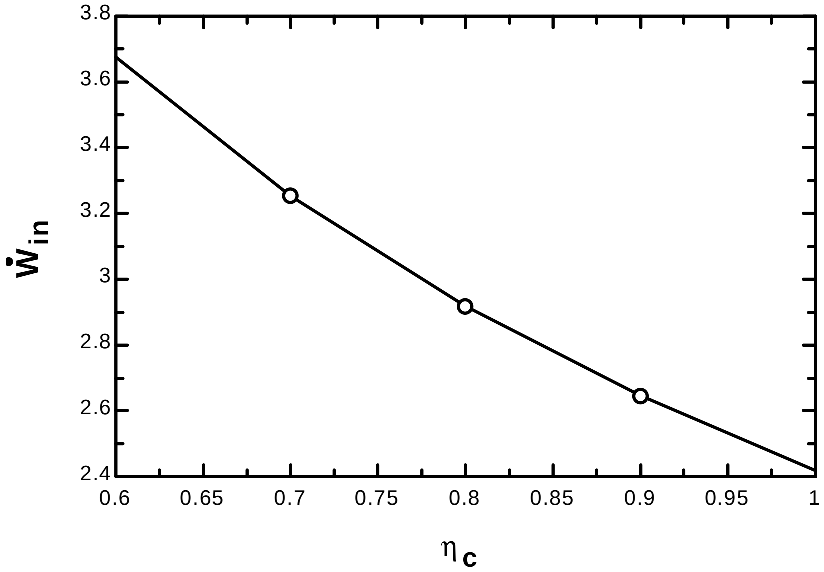
"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],T=T[1])  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],T=T[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
Wc=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+Wc=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
{h[2]=enthalpy(Fluid$,P=P[2],T=T[2]) }
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_c=m_dot*Wc

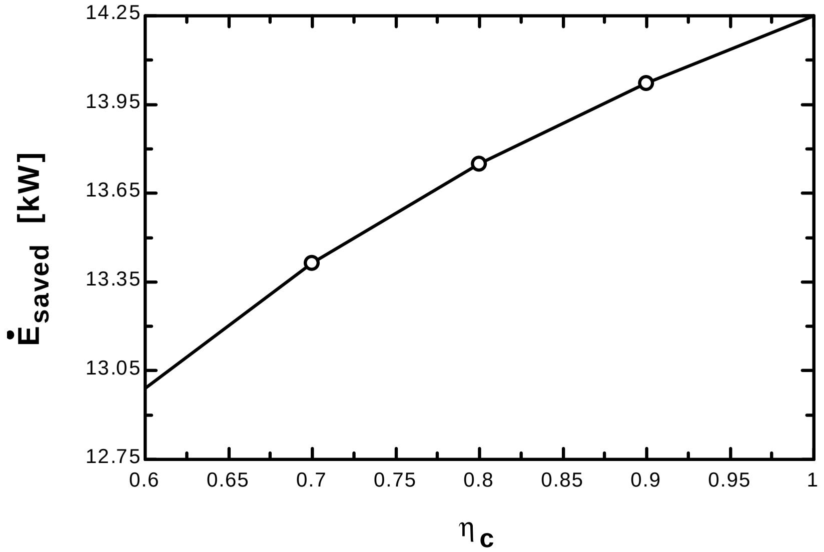
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],T=T[3]) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],T=T[3])
h[2]=Qout+h[3] "energy balance on condenser"
Q_dot_out*convert(kJ/h,kJ/s)=m_dot*Qout

"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])

"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
Q_dot_in=m_dot*Q_in
COP=Q_dot_out*convert(kJ/h,kJ/s)/W_dot_c "definition of COP"
COP_plot = COP
W_dot_in = W_dot_c
E_dot_saved = Q_dot_out*convert(kJ/h,kJ/s) - W_dot_c"[kW]"
```

W_{in} [kW]	η_c	E_{saved}
3.675	0.6	12.99
3.253	0.7	13.41
2.917	0.8	13.75
2.645	0.9	14.02
2.418	1	14.25


3.8
3.6
3.4
3.2
3
2.8
2.6
2.4
$\dot{W}_{in}$
0.6
0.65
0.7
0.75
0.8
0.85
0.9
0.95
1
η_c


14.25
13.95
13.65
13.35
13.05
12.75
$\dot{E}_{saved}$ [kW]
0.6
0.65
0.7
0.75
0.8
0.85
0.9
0.95
1
η_c

Review Problems

8-148 A turbocharged four-stroke V-16 diesel engine produces 4000 hp at 1050 rpm. The amount of power produced per cylinder per mechanical and per thermodynamic cycle is to be determined.

Analysis Noting that there are 16 cylinders and each thermodynamic cycle corresponds to 2 mechanical cycles, we have

(*a*)

$$
\begin{aligned}
w_{\text{mechanical}} &= \frac{\text{Total power produced}}{(\text{No. of cylinders})(\text{No. of mechanical cycles})} \\
&= \frac{4000\,\text{hp}}{(16\,\text{cylinders})(1050\,\text{rev/min})}\left(\frac{42.41\,\text{Btu/min}}{1\,\text{hp}}\right) \\
&= \mathbf{10.1\,Btu/cyl \cdot mech\ cycle} \quad (= 10.7\,\text{kJ/cyl} \cdot \text{mech cycle})
\end{aligned}
$$

(*b*)

$$
\begin{aligned}
w_{\text{thermodynamic}} &= \frac{\text{Total power produced}}{(\text{No. of cylinders})(\text{No. of thermodynamic cycles})} \\
&= \frac{4000\,\text{hp}}{(16\,\text{cylinders})(1050/2\,\text{rev/min})}\left(\frac{42.41\,\text{Btu/min}}{1\,\text{hp}}\right) \\
&= \mathbf{20.2\,Btu/cyl \cdot therm\ cycle} \quad (= 21.3\,\text{kJ/cyl} \cdot \text{therm cycle})
\end{aligned}
$$

8-149 A simple ideal Brayton cycle operating between the specified temperature limits is considered. The pressure ratio for which the compressor and the turbine exit temperature of air are equal is to be determined.

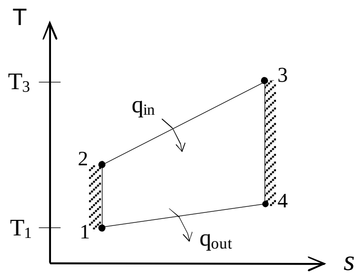
Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

Properties The specific heat ratio of air is k =1.4 (Table A-2).

Analysis We treat air as an ideal gas with constant specific heats. Using the isentropic relations, the temperatures at the compressor and turbine exit can be expressed as

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1(r_p)^{(k-1)/k}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k}$$



Setting $T_2 = T_4$ and solving for r_p gives

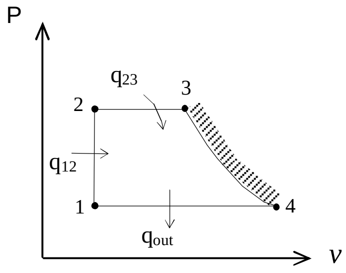
$$r_p = \left(\frac{T_3}{T_1}\right)^{k/2(k-1)} = \left(\frac{1800\text{ K}}{300\text{ K}}\right)^{1.4/0.8} = \mathbf{23.0}$$

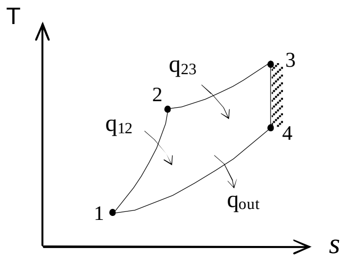
Therefore, the compressor and turbine exit temperatures will be equal when the compression ratio is 23.

8-150 The four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the net work output and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.





Analysis (b) We treat air as an ideal gas with variable specific heats,

$$T_1 = 300\text{K} \longrightarrow u_1 = 214.07\text{kJ/kg}$$
$$h_1 = 300.19\text{kJ/kg}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \left(\frac{300\text{kPa}}{100\text{kPa}}\right)(300\text{K})$$
$$= 900\text{K} \longrightarrow u_2 = 674.58\text{kJ/kg}$$
$$h_2 = 932.93\text{kJ/kg}$$

$$T_3 = 1300K \longrightarrow u_3 = 1022.82\text{kJ/kg}$$
$$h_3 = 1395.97\text{kJ/kg},\ P_{r3} = 330.9$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{100\text{kPa}}{300\text{kPa}}\right)(330.9) = 110.3$$
$$\longrightarrow h_4 = 1036.46\text{kJ/kg}$$

$$q_{in} = q_{12,in} + q_{23,in} = (u_2 - u_1) + (h_3 - h_2)$$
$$= (674.58 - 214.07) + (1395.97 - 932.93)$$
$$= 923.55\text{kJ/kg}$$

$$q_{out} = h_4 - h_1 = 1036.46 - 300.19 = 736.27\text{kJ/kg}$$

$$w_{net} = q_{in} - q_{out} = 923.55 - 736.27 = \mathbf{187.28kJ/kg}$$

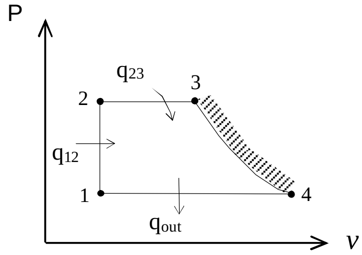
(c) $$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{187.28\text{ kJ/kg}}{923.55\text{ kJ/kg}} = \mathbf{20.3\%}$$

8-151 All four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the net work output and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

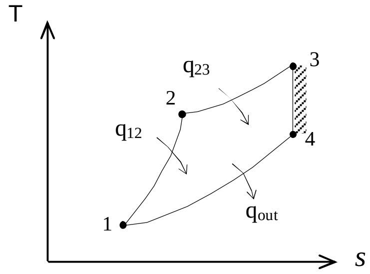
Analysis (*b*) Process 3-4 is isentropic:



$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{K})\left(\frac{1}{3}\right)^{0.4/1.4} = 949.8\text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = \left(\frac{300\text{kPa}}{100\text{kPa}}\right)(300\text{K}) = 900\text{K}$$

$$\begin{aligned} q_{in} &= q_{12,in} + q_{23,in} = (u_2 - u_1) + (h_3 - h_2) = C_v(T_2 - T_1) + C_p(T_3 - T_2) \\ &= (0.718\text{kJ/kg}\cdot\text{K})(900-300)\text{K} + (1.005\text{kJ/kg}\cdot\text{K})(1300-900)\text{K} \\ &= 832.8\text{kJ/kg} \end{aligned}$$



$$\begin{aligned} q_{out} &= h_4 - h_1 = C_p(T_4 - T_1) = (1.005\text{kJ/kg}\cdot\text{K})(949.8-300)\text{K} \\ &= 653\text{kJ/kg} \end{aligned}$$

$$w_{net} = q_{in} - q_{out} = 832.8 - 653 = \mathbf{179.8 kJ/kg}$$

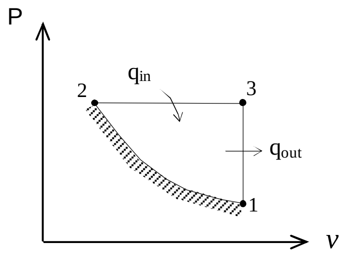
(*c*) $$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{179.8\text{ kJ/kg}}{832.8\text{ kJ/kg}} = \mathbf{21.6\%}$$

8-152 The three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the maximum temperature in the cycle and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.

Analysis (*b*) We treat air as an ideal gas with variable specific heats,

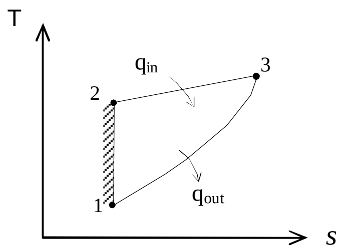


$$T_1 = 300\text{K} \longrightarrow \begin{array}{l} u_1 = 214.07\text{kJ/kg} \\ P_{r_1} = 1.386 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = \left(\frac{700\text{kPa}}{100\text{kPa}}\right)(1.386) = 9.702 \longrightarrow h_2 = 523.90\text{kJ/kg}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_1 v_1}{T_1} \longrightarrow T_{\max} = T_3 = \frac{P_3}{P_1}T_1 = \left(\frac{700\text{kPa}}{100\text{kPa}}\right)(300\text{K}) = \mathbf{2100K}$$

$$T_3 = 2100\text{K} \longrightarrow \begin{array}{l} u_3 = 1775.3\text{kJ/kg} \\ h_3 = 2377.7\text{ kJ/kg} \end{array}$$



(*c*) $$q_{in} = h_3 - h_2 = 2377.7 - 523.9 = 1853.8\text{ kJ/kg}$$

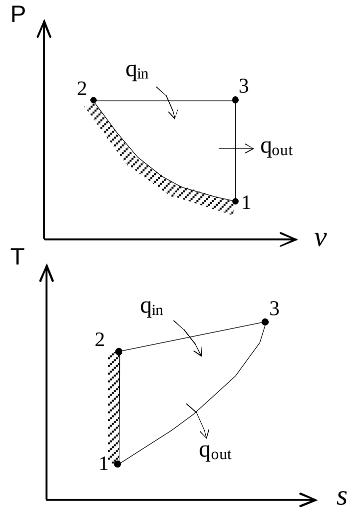
$$q_{out} = u_3 - u_1 = 1775.3 - 214.07 = 1561.23\text{kJ/kg}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1561.23\text{kJ/kg}}{1853.8\text{kJ/kg}} = \mathbf{15.8\%}$$

8-153 All three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the maximum temperature in the cycle and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).



Analysis (*b*) We treat air as an ideal gas with constant specific heats.

Process 1-2 is isentropic:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{K})\left(\frac{700\text{kPa}}{100\text{kPa}}\right)^{0.4/1.4} = 523.1\text{K}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_1 v_1}{T_1} \longrightarrow T_{\max} = T_3 = \frac{P_3}{P_1}T_1 = \left(\frac{700\text{kPa}}{100\text{kPa}}\right)(300\text{K}) = \mathbf{2100K}$$

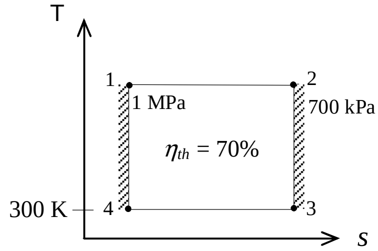
(*c*) $q_{in} = h_3 - h_2 = C_p(T_3 - T_2) = (1.005\text{kJ/kg}\cdot\text{K})(2100 - 523.1)\text{K} = 1584.8\text{kJ/kg}$

$$q_{out} = u_3 - u_1 = C_v(T_3 - T_1) = (0.718\text{kJ/kg}\cdot\text{K})(2100 - 300)\text{K} = 1292.4\text{kJ/kg}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1292.4\text{kJ/kg}}{1584.8\text{kJ/kg}} = \mathbf{18.5\%}$$

8-154 A Carnot cycle executed in a closed system uses air as the working fluid. The net work output per cycle is to be determined.

Assumptions **1** Air is an ideal gas with variable specific heats.



Analysis (a) The maximum temperature is determined from

$$\eta_{th} = 1 - \frac{T_L}{T_H} \longrightarrow 0.70 = 1 - \frac{300\text{ K}}{T_H} \longrightarrow T_H = 1000\text{K}$$

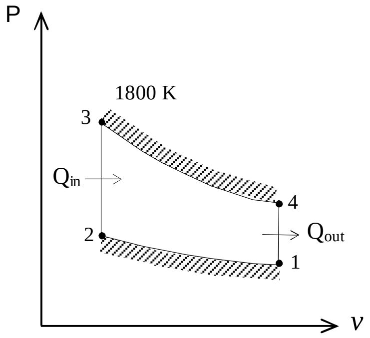
$$s_2 - s_1 = \overset{\text{Ã}0}{s_2^o - s_1^o} - R\ln\frac{P_2}{P_1} = -(0.287\text{kJ/kg}\cdot\text{K})\ln\frac{700\text{kPa}}{1000\text{kPa}}$$
$$= 0.1204\text{kJ/kg}\cdot\text{K}$$

$$W_{net} = m(s_2 - s_1)(T_H - T_L)$$
$$= (0.0015\text{ kg})(0.1024\text{kJ/kg}\cdot\text{K})(1000 - 300)\text{K}$$
$$= \mathbf{0.107\ \ kJ}$$

8-155 [*Also solved by EES on enclosed CD*] A four-cylinder spark-ignition engine with a compression ratio of 8 is considered. The amount of heat supplied per cylinder, the thermal efficiency, and the rpm for a net power output of 60 kW are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.



Analysis (*a*) Process 1-2: isentropic compression.

$$T_1 = 290K \longrightarrow u_1 = 206.91\text{kJ/kg}$$
$$v_{r_1} = 676.1$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{8}(676.1) = 84.51$$
$$\longrightarrow u_2 = 475.11\text{kJ/kg}$$

Process 2-3: v = constant heat addition.

$$T_3 = 1800\text{K} \longrightarrow u_3 = 1487.2\text{kJ/kg}$$
$$v_{r_3} = 3.994$$

$$m = \frac{P_1 V_1}{RT_1} = \frac{(98\text{kPa})(0.0006\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\text{K})} = 7.065\times 10^{-4}\text{ kg}$$

$$Q_{in} = m(u_3 - u_2) = (7.065\times 10^{-4}\text{ kg})(1487.2 - 475.11)\text{kJ/kg} = \mathbf{0.715 kJ}$$

(*b*) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (8)(3.994) = 31.95 \longrightarrow u_4 = 693.23\text{ kJ/kg}$$

Process 4-1: v = constant heat rejection.

$$Q_{out} = m(u_4 - u_1) = (7.065\times 10^{-4}\text{ kg})(693.23 - 206.91)\text{kJ/kg} = \mathbf{0.344\ kJ}$$

$$W_{net} = Q_{in} - Q_{out} = 0.715 - 0.344 = 0.371\text{ kJ}$$

$$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{0.371\text{kJ}}{0.715\text{kJ}} = \mathbf{51.9\%}$$

(*c*) $$\dot{n} = \frac{\dot{W}_{net}}{n_{cyl} W_{net,cyl}} = \frac{60\text{kJ/s}}{4\times 0.371\text{ kJ}}\left(\frac{60\text{s}}{1\text{min}}\right) = \mathbf{2426\ rpm}$$

Discussion Note for the ideal Otto cycle, a thermodynamic cycle is equivalent to a mechanical cycle (a revolution) (In actual 4-storke engines, 2 revolutions correspond to 1 thermodynamic cycle).

8-156 Problem 8-155 is reconsidered. The effect of varying the compression ratio from 5 to 11 on the net work done and the efficiency of the cycle is to be investigated. Also, the P-v and T-s diagrams for the cycle are to be plotted.

```
"Input Data"
T[1]=(17+273)"K"
P[1]=98"kPa"
T[3]=1800"K"
V_cyl=0.6"L"*1E-3"m^3/L"
r_v=8 "Compression ratio"
W_dot_net = 60"kW"
N_cyl=4"number of cylinders"
v[1]/v[2]=r_v

"The first part of the solution is done per unit mass."
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
s[2]=entropy(air, T=T[2], v=v[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=0.287*T[1]
"Conservation of energy for process 1 to 2: no heat transfer (s=const.) with work input"
 w_in = DELTAu_12
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

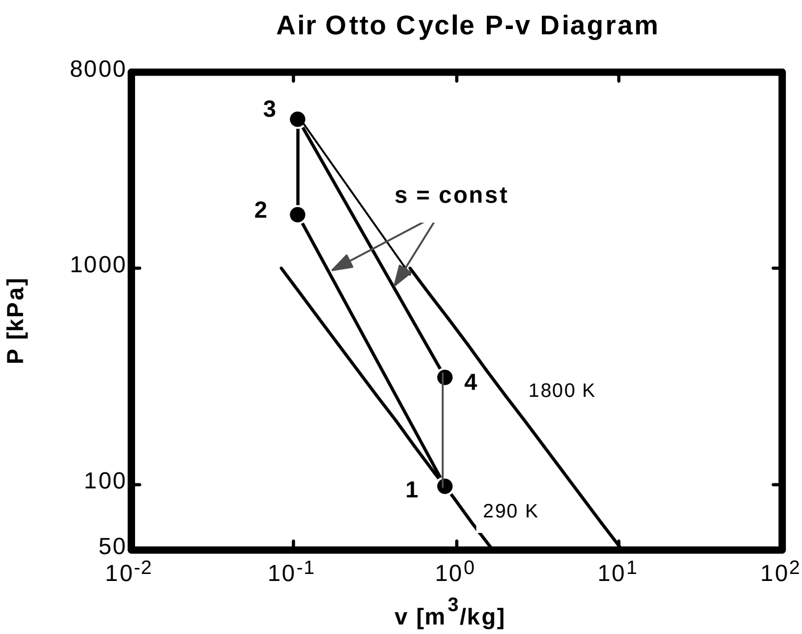
"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=0.287*T[3]
v[3]=v[2]
"Conservation of energy for process 2 to 3: the work is zero for v=const, heat is added"
q_in  = DELTAu_23
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])

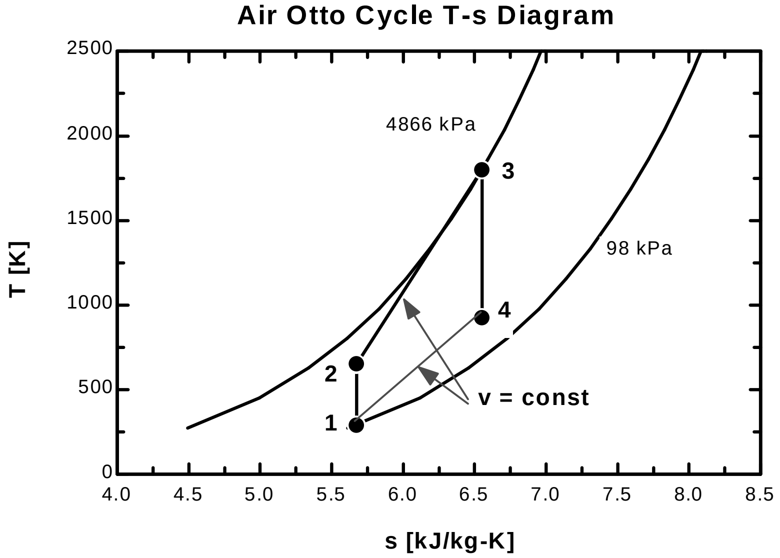
"Process 3-4 is isentropic expansion"
s[4]=entropy(air,T=T[4],P=P[4])
s[4]=s[3]
P[4]*v[4]/T[4]=P[3]*v[3]/T[3]
{P[4]*v[4]=0.287*T[4]}
"Conservation of energy for process 3 to 4: no heat transfer (s=const) with work output"
 - w_out = DELTAu_34
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])

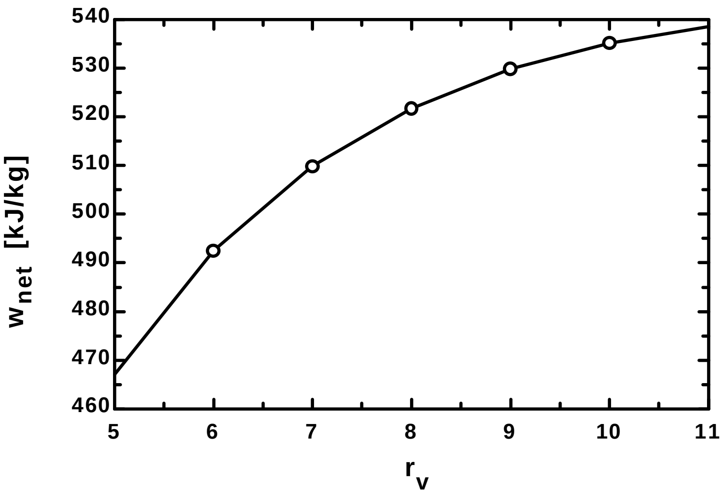
"Process 4-1 is constant volume heat rejection"
v[4]=v[1]
"Conservation of energy for process 2 to 3: the work is zero for v=const; heat is rejected"
- q_out = DELTAu_41
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
w_net = w_out - w_in
Eta_th=w_net/q_in*100 "Thermal efficiency, in percent"

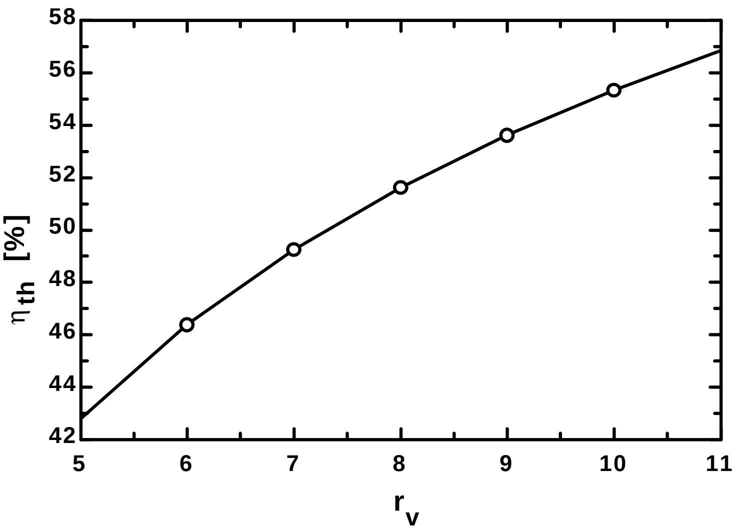
"The mass contained in each cylinder is found from the volume of the cylinder:"
V_cyl=m*v[1]
"The net work done per cycle is:"
W_dot_net=m*w_net"kJ/cyl"*N_cyl*N_dot"mechanical cycles/min"*1"min"/60"s"*1"thermal cycle"/2"mechanical cycles"
```

η_{th} [%]	r_v	w_{net} [kJ/kg]
42.81	5	467.1
46.39	6	492.5
49.26	7	509.8
51.63	8	521.7
53.63	9	529.8
55.35	10	535.2
56.85	11	538.5




Air Otto Cycle T-s Diagram
4866 kPa
98 kPa
1
2
3
4
v = const
T [K]
s [kJ/kg-K]
0
500
1000
1500
2000
2500
4.0
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5


wnet [kJ/kg]
rv
460
470
480
490
500
510
520
530
540
5
6
7
8
9
10
11


ηth [%]
rv
42
44
46
48
50
52
54
56
58
5
6
7
8
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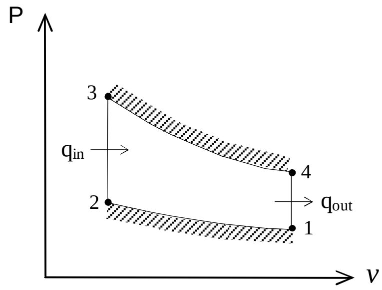
10
11

8-157 An ideal Otto cycle with air as the working fluid with a compression ratio of 9.2 is considered. The amount of heat transferred to the air, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.



Analysis (*a*) Process 1-2: isentropic compression.

$$T_1 = 300\text{K} \longrightarrow u_1 = 214.07\text{kJ/kg}$$
$$v_{r_1} = 621.2$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{9.2}(621.2) = 67.52 \longrightarrow T_2 = 708.3K$$
$$u_2 = 518.9 kJ/kg$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1} P_1 = (9.2)\left(\frac{708.3\text{K}}{300\text{K}}\right)(98\text{kPa}) = 2129\text{kPa}$$

Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{P_3}{P_2} T_2 = 2T_2 = (2)(708.3) = 1416.6\text{K} \longrightarrow u_3 = 1128.7\text{kJ/kg}$$
$$v_{r_3} = 8.593$$

$$q_{in} = u_3 - u_2 = 1128.7 - 518.9 = \mathbf{609.8 kJ/kg}$$

(*b*) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (9.2)(8.593) = 79.06 \longrightarrow u_4 = 487.75\text{kJ/kg}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = 487.75 - 214.07 = 273.7 \text{ kJ/kg}$$
$$w_{net} = q_{in} - q_{out} = 609.8 - 273.7 = \mathbf{336.1 \text{ kJ/kg}}$$

(*c*) $$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{336.1 \text{ kJ/kg}}{609.8 \text{ kJ/kg}} = \mathbf{55.1\%}$$

(*d*) $$v_{\max} = v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{98\text{kPa}} = 0.879\text{m}^3/\text{kg}$$

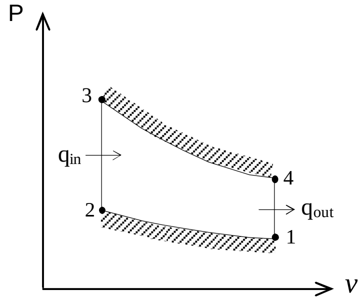
$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$MEP = \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1(1-1/r)} = \frac{336.1\text{kJ/kg}}{(0.879\text{m}^3/\text{kg})(1-1/9.2)}\left(\frac{1\text{kPa}\cdot\text{m}^3}{1\text{kJ}}\right) = \mathbf{429kPa}$$

8-158 An ideal Otto cycle with air as the working fluid with a compression ratio of 9.2 is considered. The amount of heat transferred to the air, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).



Analysis (*a*) Process 1-2 is isentropic compression:

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\text{K})(9.2)^{0.4} = 728.8\text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.2)\left(\frac{728.8\text{K}}{300\text{K}}\right)(98\text{kPa}) = 2190\text{kPa}$$

Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{P_3}{P_2}T_2 = 2T_2 = (2)(728.8) = 1457.6\text{K}$$

$$q_{in} = u_3 - u_2 = C_v(T_3 - T_2) = (0.718\text{kJ/kg}\cdot\text{K})(1457.6 - 728.8)\text{K} = \mathbf{523.3kJ/kg}$$

(*b*) Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1457.6\text{K})\left(\frac{1}{9.2}\right)^{0.4} = 600.0\text{K}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = C_v(T_4 - T_1) = (0.718\text{kJ/kg}\cdot\text{K})(600 - 300)\text{K} = 215.4\text{kJ/kg}$$

$$w_{net} = q_{in} - q_{out} = 523.3 - 215.4 = \mathbf{307.9kJ/kg}$$

(*c*) $$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{307.9\text{ kJ/kg}}{523.3\text{ kJ/kg}} = \mathbf{58.8\%}$$

(*d*) $$v_{\max} = v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{98\text{kPa}} = 0.879\text{m}^3/\text{kg}$$

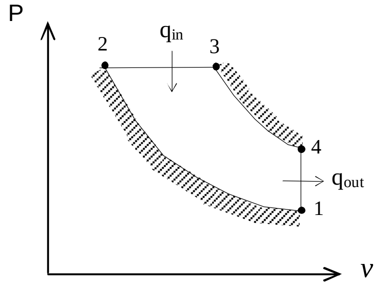
$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$MEP = \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1(1 - 1/r)} = \frac{307.9\text{kJ/kg}}{(0.879\text{m}^3/\text{kg})(1 - 1/9.2)}\left(\frac{1\text{kPa}\cdot\text{m}^3}{1\text{kJ}}\right) = \mathbf{393kPa}$$

8-159 An engine operating on the ideal diesel cycle with air as the working fluid is considered. The pressure at the beginning of the heat-rejection process, the net work per cycle, and the mean effective pressure are to be determined. √

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.



Analysis (a) The compression and the cutoff ratios are

$$r = \frac{V_1}{V_2} = \frac{1200\ \text{cm}^3}{75\ \text{cm}^3} = 16 \qquad r_c = \frac{V_3}{V_2} = \frac{150\ \text{cm}^3}{75\ \text{cm}^3} = 2$$

Process 1-2: isentropic compression.

$$T_1 = 290K \longrightarrow \begin{aligned} u_1 &= 206.91\text{kJ/kg} \\ v_{r_1} &= 676.1 \end{aligned}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{16}(676.1) = 42.256 \longrightarrow \begin{aligned} T_2 &= 837.3\text{K} \\ h_2 &= 863.03\text{kJ/kg} \end{aligned}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2T_2 = (2)(837.3) = 1674.6\ \text{K}$$

$$\longrightarrow \begin{aligned} h_3 &= 1848.9\text{kJ/kg} \\ v_{r_3} &= 5.002 \end{aligned}$$

Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = \frac{v_4}{2v_2} v_{r_3} = \frac{r}{2} v_{r_3} = \left(\frac{16}{2}\right)(5.002) = 40.016 \longrightarrow \begin{aligned} T_4 &= 853.4\text{K} \\ u_4 &= 636.00\text{kJ/kg} \end{aligned}$$

Process 4-1: v = constant heat rejection.

$$\frac{P_4 v_4}{T_4} = \frac{P_1 v_1}{T_1} \longrightarrow P_4 = \frac{T_4}{T_1} P_1 = \left(\frac{853.4\ \text{K}}{290\ \text{K}}\right)(100\ \text{kPa}) = \mathbf{294.3\ \ kPa}$$

(*b*)
$$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{kPa})(0.0012\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\text{K})} = 1.442 \times 10^{-3}\ \text{kg}$$

$$Q_{in} = m(h_3 - h_2) = (1.442 \times 10^{-3}\ \text{kg})(1848.9 - 863.08) = 1.422\text{kJ}$$

$$Q_{out} = m(u_4 - u_1) = (1.442 \times 10^{-3}\ \text{kg})(636.00 - 206.91)\text{kJ/kg} = 0.619\text{kJ}$$

$$W_{net} = Q_{in} - Q_{out} = 1.422 - 0.619 = \mathbf{0.803\ \ kJ}$$

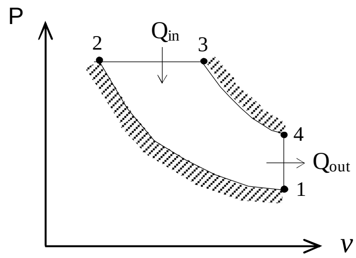
(*c*)
$$MEP = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1(1 - 1/r)} = \frac{0.803\ \text{kJ}}{(0.0012\text{m}^3)(1 - 1/16)}\left(\frac{1\ \text{kPa}\cdot\text{m}^3}{1\ \text{kJ}}\right) = \mathbf{714\ \ kPa}$$

8-160 An engine operating on the ideal diesel cycle with argon as the working fluid is considered. The pressure at the beginning of the heat-rejection process, the net work per cycle, and the mean effective pressure are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Argon is an ideal gas with constant specific heats.

Properties The properties of argon at room temperature are C_p = 0.5203 kJ/kg.K, C_v = 0.3122 kJ/kg·K, and k = 1.667 (Table A-2).

Analysis (*a*) The compression and the cutoff ratios are



$$r = \frac{V_1}{V_2} = \frac{1200 \text{ cm}^3}{75 \text{ cm}^3} = 16 \qquad r_c = \frac{V_3}{V_2} = \frac{150 \text{ cm}^3}{75 \text{ cm}^3} = 2$$

Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{V_2}{V_1}\right)^{k-1} = (290\text{K})(16)^{0.667} = 1843\text{K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2T_2 = (2)(1843) = 3686\text{K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{2V_2}{V_4}\right)^{k-1} = T_3\left(\frac{2}{r}\right)^{k-1} = (3686\text{K})\left(\frac{2}{16}\right)^{0.667} = 920.9\text{K}$$

Process 4-1: v = constant heat rejection.

$$\frac{P_4 v_4}{T_4} = \frac{P_1 v_1}{T_1} \longrightarrow P_4 = \frac{T_4}{T_1} P_1 = \left(\frac{920.9 \text{ K}}{290 \text{ K}}\right)(100 \text{ kPa}) = \mathbf{317.6 \ kPa}$$

(*b*) $$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{kPa})(0.0012\text{m}^3)}{(0.2081\text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290\text{K})} = 1.988\times10^{-3}\text{kg}$$

$$Q_{in} = m(h_3 - h_2) = mC_p(T_3 - T_2) = (1.988\times10^{-3}\text{kg})(0.5203\text{ kJ/kg}\cdot\text{K})(3686-1843)\text{K} = 1.906\text{ kJ}$$

$$Q_{out} = m(u_4 - u_1) = mC_v(T_4 - T_1) = (1.988\times10^{-3}\text{kg})(0.3122\text{ kJ/kg}\cdot\text{K})(920.9-290)\text{K} = 0.392\text{ kJ}$$

$$W_{net} = Q_{in} - Q_{out} = 1.906 - 0.392 = \mathbf{1.514 \ kJ}$$

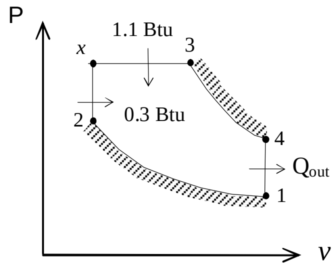
(*c*) $$MEP = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1(1-1/r)} = \frac{1.514\text{ kJ}}{(0.0012\text{ m}^3)(1-1/16)}\left(\frac{1\text{ kPa}\cdot\text{m}^3}{1\text{ kJ}}\right) = \mathbf{1346 \ kPa}$$

8-161E An ideal dual cycle with air as the working fluid with a compression ratio of 12 is considered. The thermal efficiency of the cycle is to be determined. √

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** 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, C_v = 0.171 Btu/lbm.R, and k = 1.4 (Table A-2E).

Analysis (*a*) The mass of air is



$$m = \frac{P_1 V_1}{RT_1} = \frac{(14.7\,\text{psia})(75/1728\,\text{ft}^3)}{(0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550\,\text{R})} = 3.132\times10^{-3}\,\text{lbm}$$

Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (550\,\text{R})(12)^{0.4} = 1486\,\text{R}$$

Process 2-x: v = *constant* heat addition,

$$Q_{2-x,in} = m(u_x - u_2) = mC_v(T_x - T_2)$$
$$0.3\,\text{Btu} = (3.132\times10^{-3}\,\text{lbm})(0.171\,\text{Btu/lbm}\cdot\text{R})(T_x - 1486)\text{R} \longrightarrow T_x = 2046\,\text{R}$$

Process x-3: P = *constant* heat addition.

$$Q_{x-3,in} = m(h_3 - h_x) = mC_p(T_3 - T_x)$$
$$1.1\,\text{Btu} = (3.132\times10^{-3}\,\text{lbm})(0.240\,\text{Btu/lbm}\cdot\text{R})(T_3 - 2046)\text{R} \longrightarrow T_3 = 3509\,\text{R}$$

$$\frac{P_3 V_3}{T_3} = \frac{P_x V_x}{T_x} \longrightarrow r_c = \frac{V_3}{V_x} = \frac{T_3}{T_x} = \frac{3509\,\text{R}}{2046\,\text{R}} = 1.715$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{1.715V_1}{V_4}\right)^{k-1} = T_3\left(\frac{1.715}{r}\right)^{k-1} = (3509\,\text{R})\left(\frac{1.715}{12}\right)^{0.4} = 1611\,\text{R}$$

Process 4-1: v = constant heat rejection.

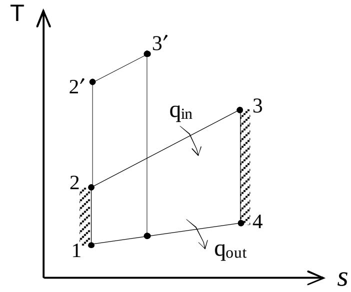
$$Q_{out} = m(u_4 - u_1) = mC_v(T_4 - T_1)$$
$$= (3.132\times10^{-3}\,\text{lbm})(0.171\,\text{Btu/lbm}\cdot\text{R})(1611 - 550)\text{R} = 0.568\,\text{Btu}$$

$$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{0.568\,\text{Btu}}{1.4\,\text{Btu}} = \mathbf{59.4\%}$$

8-162 A simple ideal Brayton cycle with air as the working fluid is considered. The changes in the net work output per unit mass and the thermal efficiency are to be determined. √

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The properties of air are given in Table A-21.



Analysis (*a*) The properties at various states are

$$T_1 = 300 \text{ K} \longrightarrow \begin{array}{l} h_1 = 300.19 \text{ kJ/kg} \\ P_{r_1} = 1.386 \end{array}$$

$$T_3 = 1300 \text{ K} \longrightarrow \begin{array}{l} h_3 = 1395.97 \text{ kJ/kg} \\ P_{r_3} = 330.9 \end{array}$$

For $r_p = 6$,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (6)(1.386) = 8.316 \longrightarrow h_2 = 501.40 \text{kJ/kg}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{6}\right)(330.9) = 55.15 \longrightarrow h_4 = 855.3 \text{kJ/kg}$$

$$q_{in} = h_3 - h_2 = 1395.97 - 501.40 = 894.57 \text{kJ/kg}$$
$$q_{out} = h_4 - h_1 = 855.3 - 300.19 = 555.11 \text{kJ/kg}$$
$$w_{net} = q_{in} - q_{out} = 894.57 - 555.11 = 339.46 \text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{339.46 \text{kJ/kg}}{894.57 \text{kJ/kg}} = 37.9\%$$

For $r_p = 12$,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (12)(1.386) = 16.63 \longrightarrow h_2 = 610.6 \text{kJ/kg}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{12}\right)(330.9) = 27.58 \longrightarrow h_4 = 704.6 \text{kJ/kg}$$

$$q_{in} = h_3 - h_2 = 1395.97 - 610.60 = 785.37 \text{kJ/kg}$$
$$q_{out} = h_4 - h_1 = 704.6 - 300.19 = 404.41 \text{kJ/kg}$$
$$w_{net} = q_{in} - q_{out} = 785.37 - 404.41 = 380.96 \text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{380.96 \text{kJ/kg}}{785.37 \text{kJ/kg}} = 48.5\%$$

Thus,

(*a*) $\Delta w_{net} = 380.96 - 339.46 = \mathbf{41.5\ kJ/kg}$ (increase)

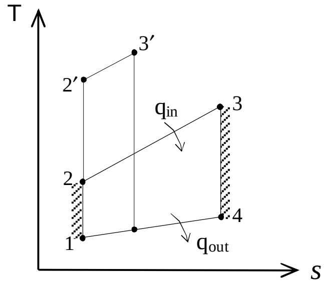
(*b*) $\Delta \eta_{th} = 48.5\% - 37.9\% = \mathbf{10.6\%}$ (increase)

8-163 A simple ideal Brayton cycle with air as the working fluid is considered. The changes in the net work output per unit mass and the thermal efficiency are to be determined.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.287 kJ/kg.K, C_p = 1.005 kJ/kg.K, C_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis Processes 1-2 and 3-4 are isentropic. Therefore, For r_p = 6,



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{K})(6)^{0.4/1.4} = 500.6\text{K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{K})\left(\frac{1}{6}\right)^{0.4/1.4} = 779.1\text{K}$$

$$\begin{aligned} q_{in} &= h_3 - h_2 = C_p(T_3 - T_2) \\ &= (1.005\text{kJ/kg}\cdot\text{K})(1300 - 500.6)\text{K} = 803.4\text{kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{out} &= h_4 - h_1 = C_p(T_4 - T_1) \\ &= (1.005\text{kJ/kg}\cdot\text{K})(779.1 - 300)\text{K} = 481.5\text{kJ/kg} \end{aligned}$$

$$w_{net} = q_{in} - q_{out} = 803.4 - 481.5 = 321.9\text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{321.9\text{kJ/kg}}{803.4\text{kJ/kg}} = 40.1\%$$

For r_p = 12,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{K})(12)^{0.4/1.4} = 610.2\text{K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{K})\left(\frac{1}{12}\right)^{0.4/1.4} = 639.2\text{K}$$

$$\begin{aligned} q_{in} &= h_3 - h_2 = C_p(T_3 - T_2) \\ &= (1.005\text{kJ/kg}\cdot\text{K})(1300 - 610.2)\text{K} = 693.2\text{kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{out} &= h_4 - h_1 = C_p(T_4 - T_1) \\ &= (1.005\text{kJ/kg}\cdot\text{K})(639.2 - 300)\text{K} = 340.9\text{kJ/kg} \end{aligned}$$

$$w_{net} = q_{in} - q_{out} = 693.2 - 340.9 = 352.3\text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{352.3\text{kJ/kg}}{693.2\text{kJ/kg}} = 50.8\%$$

Thus,

(a) $\Delta w_{net} = 352.3 - 321.9 = \mathbf{30.4\ kJ/kg}$ (increase)

(b) $\Delta \eta_{th} = 50.8\% - 40.1\% = \mathbf{10.7\%}$ (increase)

8-164 A regenerative Brayton cycle with helium as the working fluid is considered. The thermal efficiency and the required mass flow rate of helium are to be determined for 100 percent and 80 percent isentropic efficiencies for both the compressor and the turbine.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

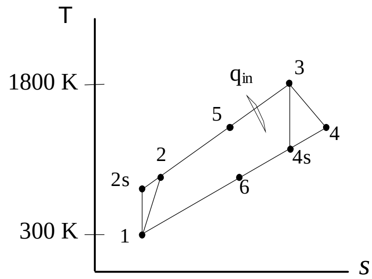
Properties The properties of helium at room temperature are C_p = 5.1926 kJ/kg.K and k = 1.667 (Table A-2).

Analysis (a) Assuming $\eta_T = \eta_C = 100\%$,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(8)^{0.667/1.667} = 689.4\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1800\text{ K})\left(\frac{1}{8}\right)^{0.667/1.667} = 783.3\text{ K}$$

$$\varepsilon = \frac{h_5 - h_2}{h_4 - h_2} = \frac{C_p(T_5 - T_2)}{C_p(T_4 - T_2)} \longrightarrow T_5 = T_2 + \varepsilon(T_4 - T_2)$$
$$= 689.4 + (0.75)(783.3 - 689.4)$$
$$= 759.8\text{ K}$$



$$w_{net} = w_{T,out} - w_{C,in} = (h_3 - h_4) - (h_2 - h_1) = C_p[(T_3 - T_4) - (T_2 - T_1)]$$
$$= (5.1926\text{ kJ/kg}\cdot\text{K})[(1800 - 783.3) - (689.4 - 300)]\text{K} = 3257.3\text{kJ/kg}$$

$$\dot{m} = \frac{\dot{W}_{net}}{w_{net}} = \frac{450{,}000\text{ kJ/s}}{3257.3\text{ kJ/kg}} = \mathbf{13.82\ kg/s}$$

$$q_{in} = h_3 - h_5 = C_p(T_3 - T_5) = (5.1926\text{ kJ/kg}\cdot\text{K})(1800 - 759.8)\text{K} = 5401.3\text{ kJ/kg}$$

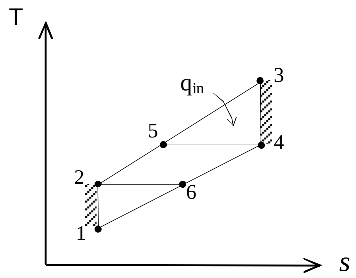
$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{3257.3\text{ kJ/kg}}{5401.3\text{ kJ/kg}} = \mathbf{60.3\%}$$

8-165 An ideal regenerative Brayton cycle is considered. The pressure ratio that maximizes the thermal efficiency of the cycle is to be determined, and to be compared with the pressure ratio that maximizes the cycle net work.

Analysis Using the isentropic relations, the temperatures at the compressor and turbine exit can be expressed as

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1\left(r_p\right)^{(k-1)/k}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = T_3 r_p^{(1-k)/k}$$



Then,

$$q_{in} = h_3 - h_5 = C_p\left(T_3 - T_5\right) = C_p\left(T_3 - T_4\right) = C_p T_3\left(1 - r_p^{(1-k)/k}\right)$$

$$q_{out} = h_6 - h_1 = C_p\left(T_6 - T_1\right) = C_p\left(T_2 - T_1\right) = C_p T_1\left(r_p^{(k-1)/k} - 1\right)$$

$$w_{net} = q_{in} - q_{out} = C_p\left(T_3 - T_3 r_p^{(1-k)/k} - T_1 r_p^{(k-1)/k} + T_1\right)$$

To maximize the net work, we must have

$$\frac{\partial w_{net}}{\partial r_p} = C_p\left(-\frac{1-k}{k}T_3 r_p^{\,(1-k)/k\,-1} - \frac{k-1}{k}T_1 r_p^{\,(k-1)/k\,-1}\right) = 0$$

Solving for r_p gives

$$r_p = \left(\frac{T_1}{T_3}\right)^{k/2(1-k)}$$

Similarly,

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p T_1\left(r_p^{\,(k-1)/k} - 1\right)}{C_p T_3\left(1 - r_p^{\,(1-k)/k}\right)}$$

which simplifies to

$$\eta_{th} = 1 - \frac{T_1}{T_3} r_p^{\,(k-1)/k}$$

When $r_p = 1$, the thermal efficiency becomes $\eta_{th} = 1 - T_1/T_3$, which is the Carnot efficiency. Therefore, the efficiency is a maximum when $r_p = 1$, and must decrease as r_p increases for the fixed values of T_1 and T_3. Note that the compression ratio cannot be less than 1, and the factor

$$r_p^{\,(k-1)/k}$$

is always greater than 1 for $r_p > 1$. Also note that the net work $w_{net} = 0$ for $r_p = 1$. This being the case, the pressure ratio for maximum thermal efficiency, which is $r_p = 1$, is always less than the pressure ratio for maximum work.

8-166 Using EES (or other) software, the effect of variable specific heats on the thermal efficiency of the ideal Otto cycle using air as the working fluid is to be investigated. The percentage of error involved in using constant specific heat values at room temperature is to be determined for the following combinations of compression ratios and maximum cycle temperatures: r = 6, 8, 10, 12 and Tmax = 1000, 1500, 2000, 2500 K.

```
"We assume that this ideal gas cycle takes place in a piston-cylinder device;
therefore, we will use a closed system analysis."
"See the T-s diagram in Plot Window1 and the P-v diagram in Plot Window2"

Procedure ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
"For Air:"
C_V = 0.718"[kJ/kg-K]"
k = 1.4
T2 = T[1]*r_comp^(k-1)
P2 = P[1]*r_comp^k
q_in_23 = C_V*(T[3]-T2)
T4 = T[3]*(1/r_comp)^(k-1)
q_out_41 = C_V*(T4-T[1])
Eta_th_ConstProp = (1-q_out_41/q_in_23)*100"[%]"

"The Easy Way to calculate the constant property Otto cycle efficiency is:"
Eta_th_easy = (1 - 1/r_comp^(k-1))*100"[%]"

END
"Input Data"

T[1]=300"K"
P[1]=100"kPa"
{T[3] = 1000"[k]"}

r_comp = 12

"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]

T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=0.287*T[1]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

"Process 2-3 is constant volume heat addition"
v[3]=v[2]

s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=0.287*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])

"Process 3-4 is isentropic expansion"
s[4]=s[3]
```

```
s[4]=entropy(air,T=T[4],P=P[4])

P[4]*v[4]=0.287*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])

"Process 4-1 is constant volume heat rejection"
V[4] = V[1]

"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0       "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])

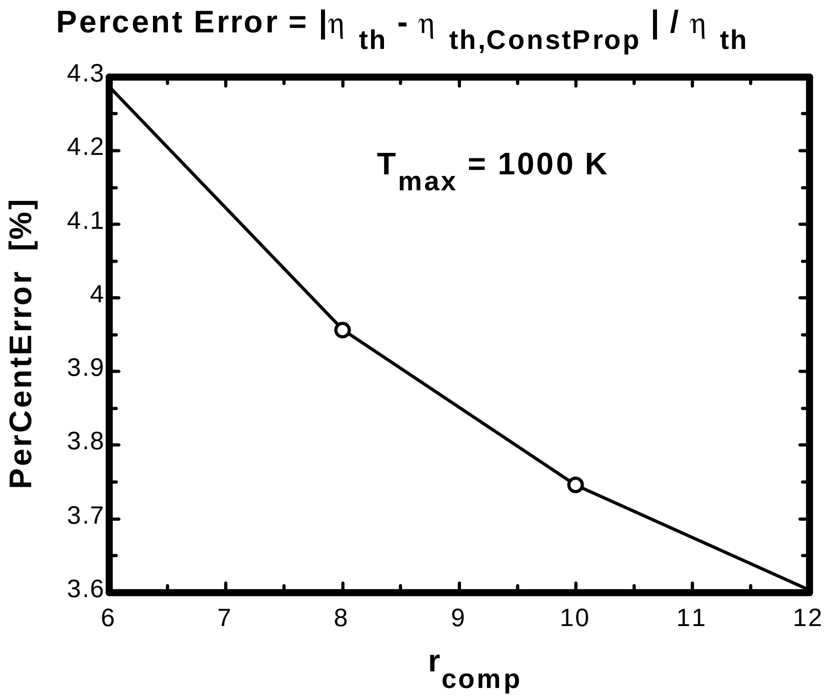
q_in_total=q_23
q_out_total = -q_41

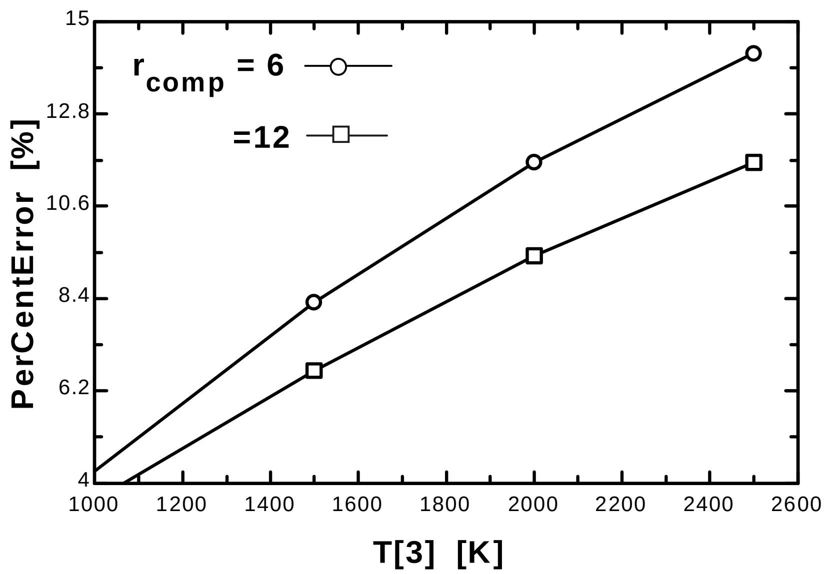
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"

Call ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)

PerCentError = ABS(Eta_th - Eta_th_ConstProp)/Eta_th*100"[%]"
```

PerCentError [%]	r_{comp}	η_{th} [%]	$\eta_{th,ConstProp}$ [%]	$\eta_{th,easy}$ [%]	T_3 [K
3.604	12	60.8	62.99	62.99	1000
6.681	12	59.04	62.99	62.99	1500
9.421	12	57.57	62.99	62.99	2000
11.64	12	56.42	62.99	62.99	2500


Percent Error = |η th - η th,ConstProp | / η th
Tmax = 1000 K
PerCentError [%]
rcomp
4.3
4.2
4.1
4
3.9
3.8
3.7
3.6
6
7
8
9
10
11
12


rcomp = 6
=12
PerCentError [%]
T[3] [K]
15
12.8
10.6
8.4
6.2
4
1000
1200
1400
1600
1800
2000
2200
2400
2600

8-167 Using EES (or other) software, the effects of compression ratio on the net work output and the thermal efficiency of the Otto cycle for a maximum cycle temperature of 2000 K are to be investigated for the case of variable specific heats. The compression ratio is to be varied from 6 to 15 with an increment of 1. The results are to be tabulated and plotted against the compression ratio.

```
"See the T-s diagram in Plot Window1 and the P-v diagram in Plot Window2"

"Input Data"
T[1]=300"K"
P[1]=100"kPa"
T[3] = 2000"[k]"
r_comp = 12

"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]

T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=0.287*T[1]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

"Process 2-3 is constant volume heat addition"
v[3]=v[2]

s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=0.287*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])

"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])

P[4]*v[4]=0.287*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])

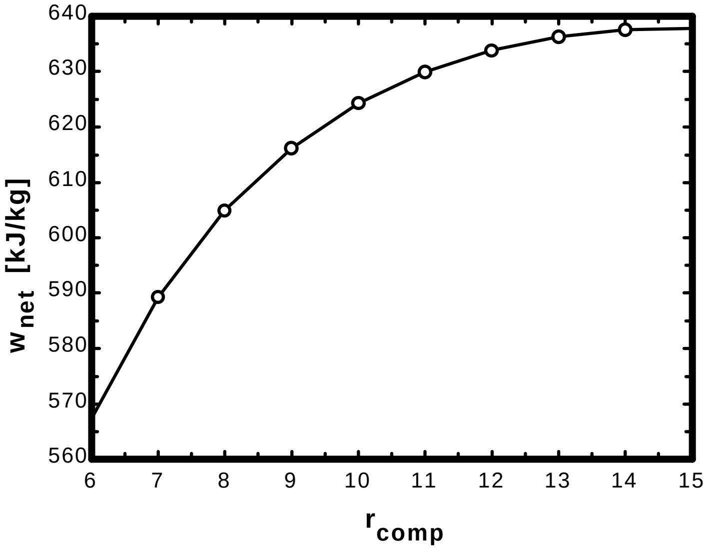
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]

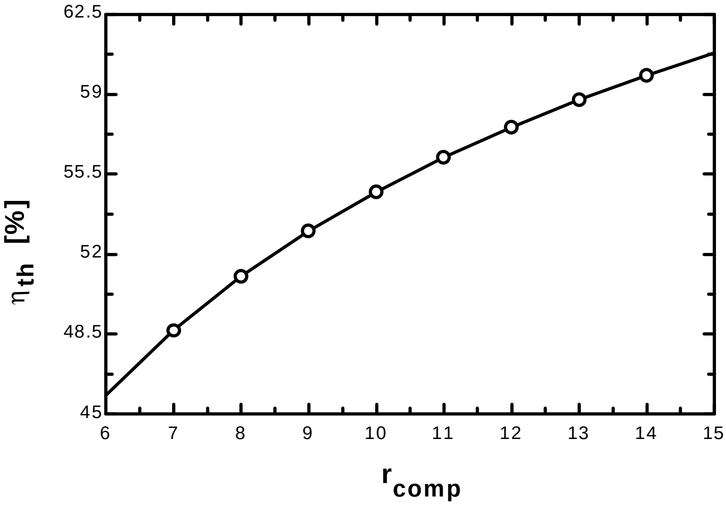
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0        "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])

q_in_total=q_23
q_out_total = -q_41
```

w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"

η_{th} [%]	r_{comp}	w_{net} [kJ/kg]
45.83	6	567.4
48.67	7	589.3
51.03	8	604.9
53.02	9	616.2
54.74	10	624.3
56.24	11	630
57.57	12	633.8
58.75	13	636.3
59.83	14	637.5
60.8	15	637.9





8-168 Using EES (or other) software, the effects of pressure ratio on the net work output and the thermal efficiency of a simple Brayton cycle are to be investigated for a maximum cycle temperature of 1800 K and variable specific heats. The pressure ratio is to be varied from 5 to 24 with an increment of 1. The results are to be tabulated and plotted against the pressure ratio. The pressure ratio at which the net work output becomes a maximum, and the pressure ratio at which the thermal efficiency becomes a maximum are to be determined.

```
“Let's allow the mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic
efficiencies of the turbine and compressor to vary.   Choose an initial mass flow rate of 1
kg/s.”
P_ratio = 8
T[1] = 300"K"
P[1]= 100"kPa"
T[3] = 1800"K"
m_dot = 1 "kg/s"
Eta_c = 100/100
Eta_t = 100/100

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

"Compressor analysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])  "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

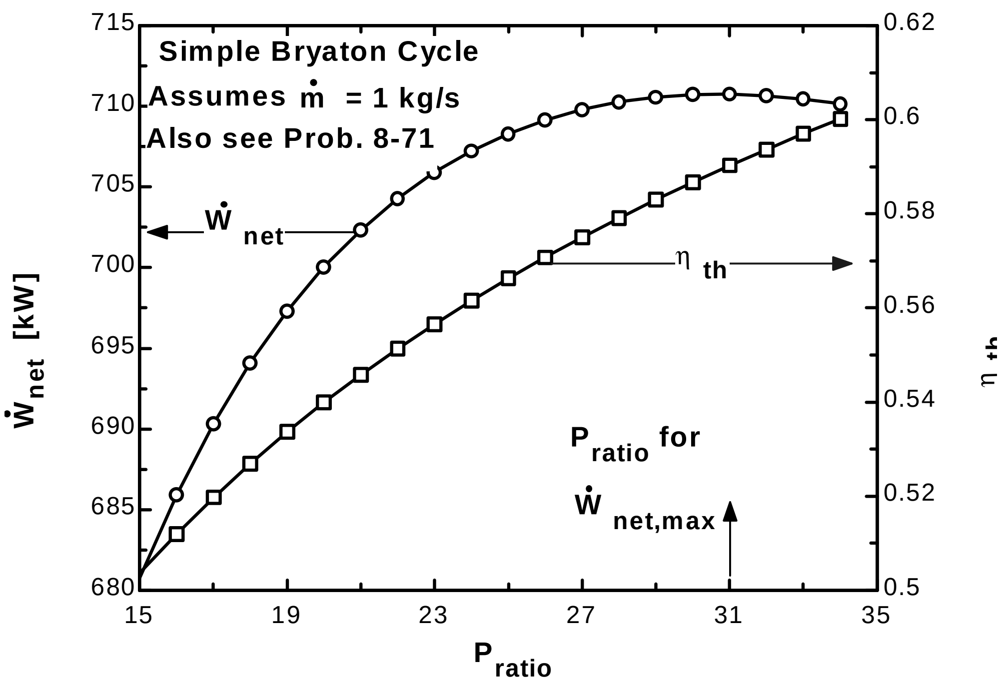
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming
W=0, ke=pe=0"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine
exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"
"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
```

```
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
s[4]=entropy('air',T=T[4],P=P[4])
```

Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.3398	0.5036	15	350.4	680.8	1031	1352
0.3457	0.512	16	362.4	685.9	1048	1340
0.3513	0.5197	17	373.9	690.3	1064	1328
0.3567	0.5269	18	384.8	694.1	1079	1317
0.3618	0.5336	19	395.4	697.3	1093	1307
0.3668	0.5399	20	405.5	700	1106	1297
0.3716	0.5458	21	415.3	702.3	1118	1287
0.3762	0.5513	22	424.7	704.3	1129	1277
0.3806	0.5566	23	433.8	705.9	1140	1268
0.385	0.5615	24	442.7	707.2	1150	1259
0.3892	0.5663	25	451.2	708.3	1160	1251
0.3932	0.5707	26	459.6	709.2	1169	1243
0.3972	0.575	27	467.7	709.8	1177	1234
0.401	0.5791	28	475.5	710.3	1186	1227
0.4048	0.583	29	483.2	710.6	1194	1219
0.4084	0.5867	30	490.7	710.7	1201	1211
0.412	0.5903	31	498	710.8	1209	1204
0.4155	0.5937	32	505.1	710.7	1216	1197
0.4189	0.597	33	512.1	710.4	1223	1190
0.4222	0.6002	34	518.9	710.1	1229	1183



8-169 Problem 8-168 is reconsidered. The effects of adiabatic efficiencies on Brayton Cycle are to be investigated.

```
Let's allow the mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic
efficiencies of the turbine and compressor to vary.   Choose an initial mass flow rate of 1
kg/s
P_ratio = 8
T[1] = 300"K"
P[1]= 100"kPa"
T[3] = 1800"K"
m_dot = 1 "kg/s"
Eta_c = 85/100
Eta_t = 85/100

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

"Compressor analysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

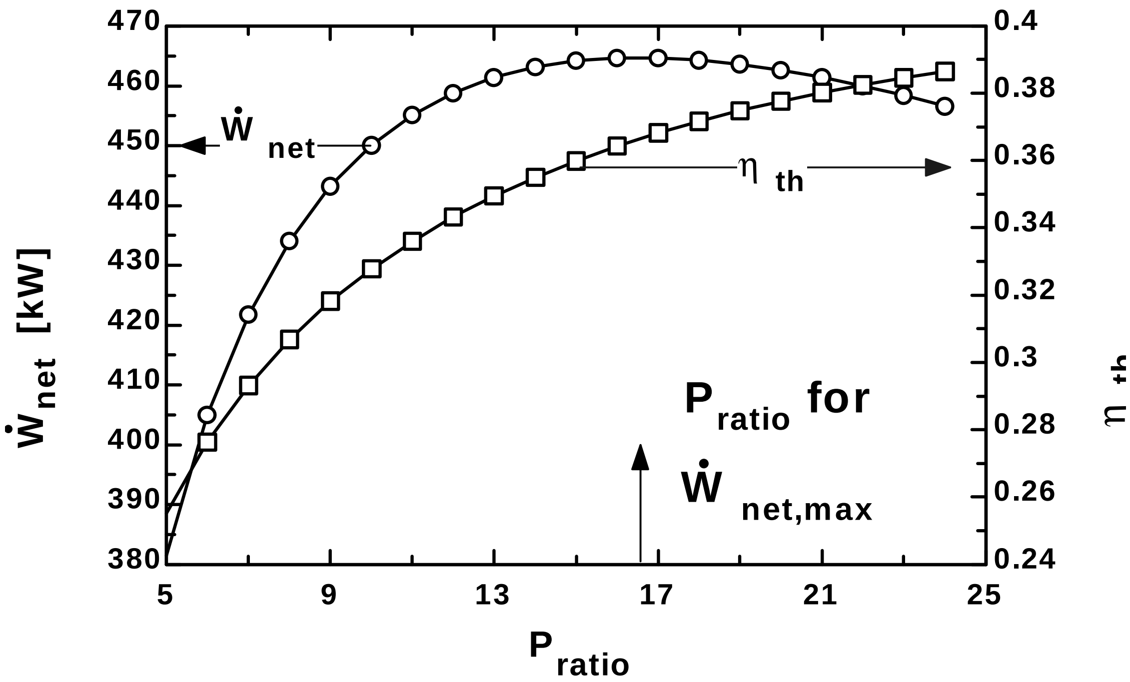
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming
W=0, ke=pe=0"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine
exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
s[4]=entropy('air',T=T[4],P=P[4])
```

Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.3515	0.2551	5	206.8	381.5	588.3	1495
0.3689	0.2764	6	236.7	405	641.7	1465
0.3843	0.2931	7	263.2	421.8	685	1439
0.3981	0.3068	8	287.1	434.1	721.3	1415
0.4107	0.3182	9	309	443.3	752.2	1393
0.4224	0.3278	10	329.1	450.1	779.2	1373
0.4332	0.3361	11	347.8	455.1	803	1354
0.4433	0.3432	12	365.4	458.8	824.2	1337
0.4528	0.3495	13	381.9	461.4	843.3	1320
0.4618	0.355	14	397.5	463.2	860.6	1305
0.4704	0.3599	15	412.3	464.2	876.5	1290
0.4785	0.3643	16	426.4	464.7	891.1	1276
0.4862	0.3682	17	439.8	464.7	904.6	1262
0.4937	0.3717	18	452.7	464.4	917.1	1249
0.5008	0.3748	19	465.1	463.6	928.8	1237
0.5077	0.3777	20	477.1	462.6	939.7	1225
0.5143	0.3802	21	488.6	461.4	950	1214
0.5207	0.3825	22	499.7	460	959.6	1202
0.5268	0.3846	23	510.4	458.4	968.8	1192
0.5328	0.3865	24	520.8	456.6	977.4	1181



8-170 Using EES (or other) software, the effects of pressure ratio, maximum cycle temperature, and compressor and turbine inefficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with air as the working fluid are to be investigated for the case of constant specific heats for air at room temperature. The net work output and the thermal efficiency are to be determined for all combinations of the following parameters:.
Pressure ratio: 5, 8, 14
Maximum cycle temperature: 800, 1200, 1600 K
Compressor adiabatic efficiency: 80, 100 percent
Turbine adiabatic efficiency: 80, 100 percent

Let's allow the mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic efficiencies of the turbine and compressor to vary. Choose an initial mass flow rate of 1 kg/s "

```
"Input data - from diagram window"
{P_ratio = 8}
{T[1] = 300"K"
P[1]= 100"kPa"
T[3] = 800"K"
m_dot = 1 "kg/s"
Eta_c = 75/100
Eta_t = 82/100}

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

"Compressor analysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])  "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming
W=0, ke=pe=0"

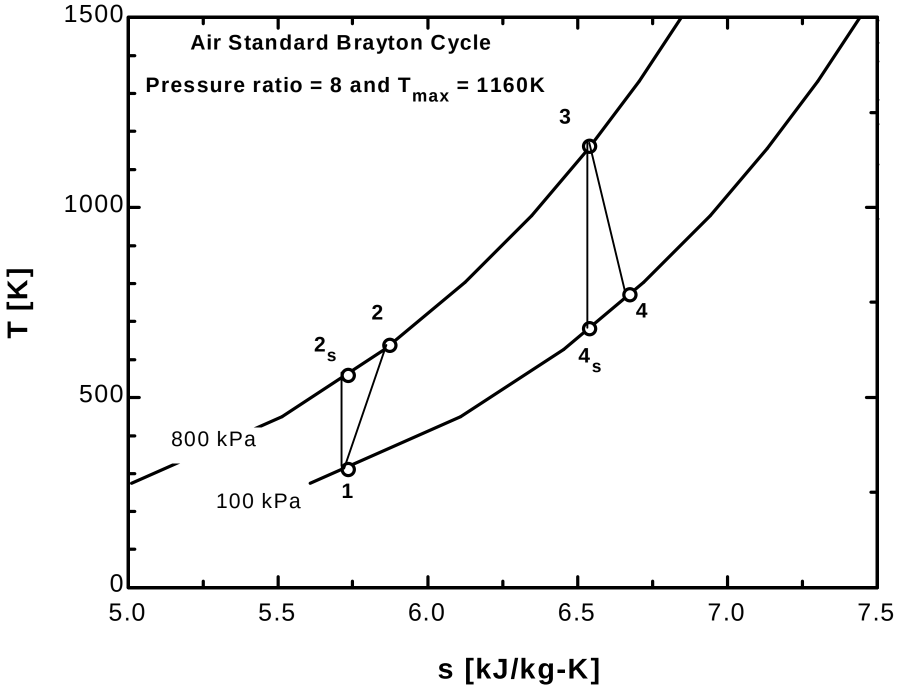

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine
exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

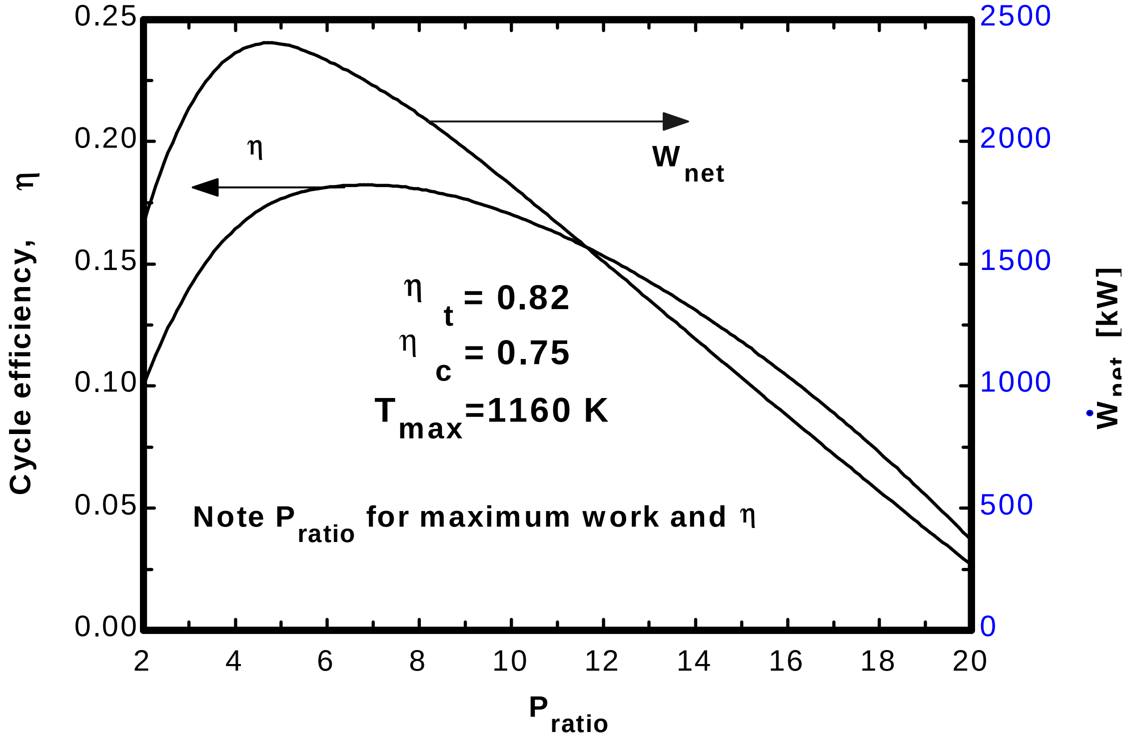
"Cycle analysis"
```

```
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
s[4]=entropy('air',T=T[4],P=P[4])
```

Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.5229	0.1	2	1818	1659	3477	16587
0.6305	0.1644	4	4033	2364	6396	14373
0.7038	0.1814	6	5543	2333	7876	12862
0.7611	0.1806	8	6723	2110	8833	11682
0.8088	0.1702	10	7705	1822	9527	10700
0.85	0.1533	12	8553	1510	10063	9852
0.8864	0.131	14	9304	1192	10496	9102
0.9192	0.1041	16	9980	877.2	10857	8426
0.9491	0.07272	18	10596	567.9	11164	7809
0.9767	0.03675	20	11165	266.1	11431	7241




0.25
0.20
0.15
0.10
0.05
0.00
2500
2000
1500
1000
500
0
2
4
6
8
10
12
14
16
18
20
η
W net
η t = 0.82
η c = 0.75
T max =1160 K
Note P ratio for maximum work and η
Cycle efficiency, η
Ẇ net [kW]
P ratio

8-171 Problem 8-170 is to be repeated by considering the variation of specific heats of air with temperature.

```
Procedure ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
"For Air:"
C_V = 0.718"[kJ/kg-K]"
k = 1.4
T2 = T[1]*r_comp^(k-1)
P2 = P[1]*r_comp^k
q_in_23 = C_V*(T[3]-T2)
T4 = T[3]*(1/r_comp)^(k-1)
q_out_41 = C_V*(T4-T[1])
Eta_th_ConstProp = (1-q_out_41/q_in_23)*100"[%]"

"The Easy Way to calculate the constant property Otto cycle efficiency is:"
Eta_th_easy = (1 - 1/r_comp^(k-1))*100"[%]"

END

"Input Data"

T[1]=300"K"
P[1]=100"kPa"
{T[3] = 1000"[k]"}
r_comp = 12

"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=0.287*T[1]
V[2] = V[1]/ r_comp

"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

"Process 2-3 is constant volume heat addition"
v[3]=v[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=0.287*T[3]

"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])

"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=0.287*T[4]

"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
```

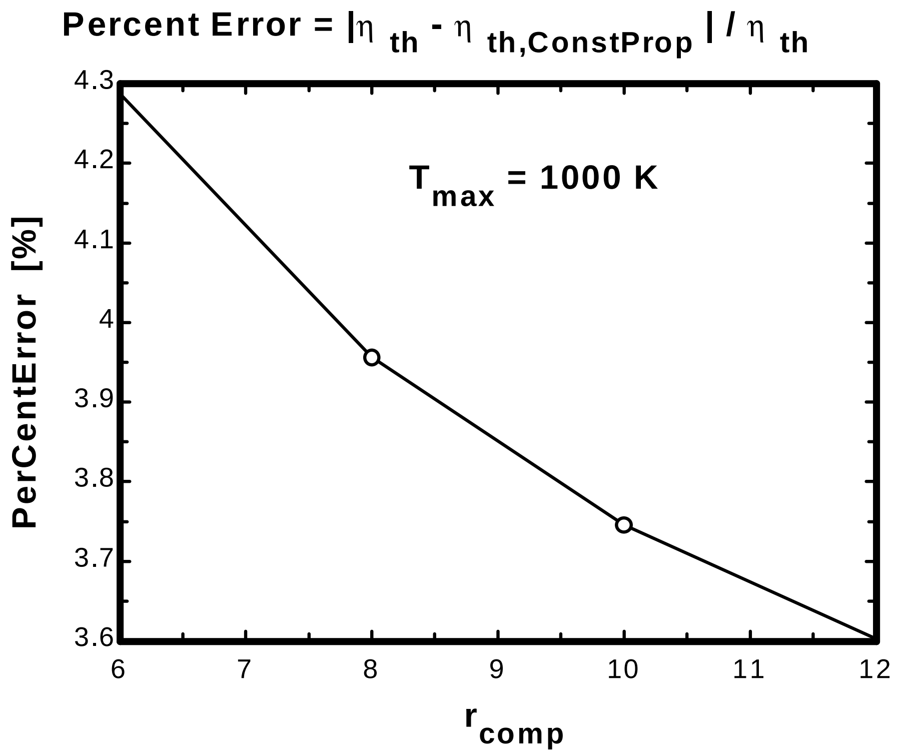
```
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]

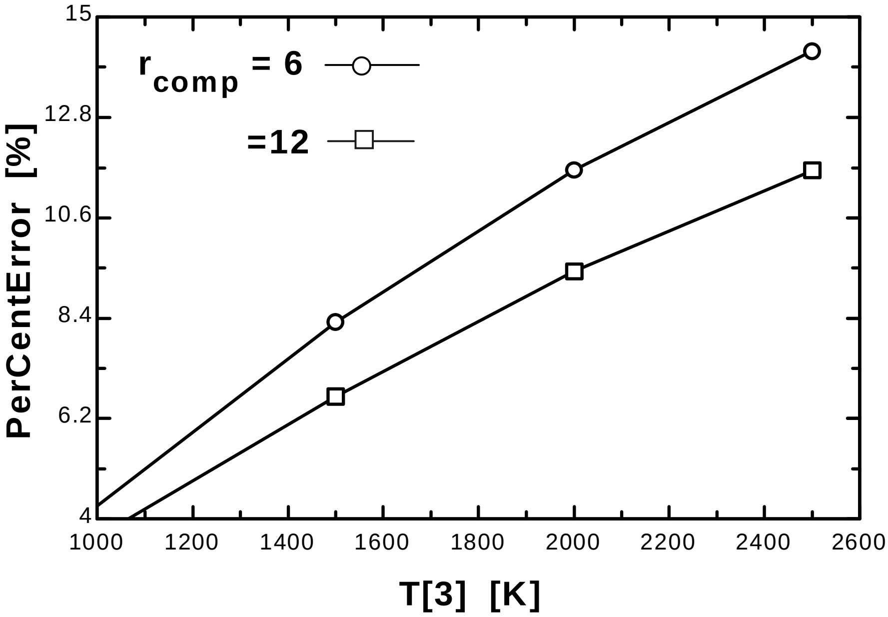
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0       "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])

q_in_total=q_23
q_out_total = -q_41
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"

Call ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
PerCentError = ABS(Eta_th - Eta_th_ConstProp)/Eta_th*100"[%]"
```

PerCentError [%]	r_{comp}	η_{th} [%]	$\eta_{th,ConstProp}$ [%]	$\eta_{th,easy}$ [%]	T_3 [K]
3.604	12	60.8	62.99	62.99	1000
6.681	12	59.04	62.99	62.99	1500
9.421	12	57.57	62.99	62.99	2000
11.64	12	56.42	62.99	62.99	2500





15
12.8
10.6
8.4
6.2
4
1000
1200
1400
1600
1800
2000
2200
2400
2600
PerCentError [%]
T[3] [K]
r comp = 6
=12

8-172 Problem 8-170 is to be repeated using helium as the working fluid.

```
Function hFunc(WorkFluid$,T,P)
"The EES functions teat helium as a real gas; thus, T and P are needed for helium's
enthalpy."
IF WorkFluid$ = 'Air' then hFunc:=enthalpy(Air,T=T) ELSE
    hFunc: = enthalpy(Helium,T=T,P=P)
endif
END

Procedure EtaCheck(Eta_th:EtaError$)
If Eta_th < 0 then EtaError$ = 'Why are the net work done and efficiency < 0?' Else
EtaError$ = ''
END

"Input data - from diagram window"
{P_ratio = 8}
{T[1] = 300"K"
P[1]= 100"kPa"
T[3] = 800"K"
m_dot = 1 "kg/s"
Eta_c = 0.8
Eta_t = 0.8
WorkFluid$ = 'Helium'}

"Inlet conditions"
h[1]=hFunc(WorkFluid$,T[1],P[1])
s[1]=ENTROPY(WorkFluid$,T=T[1],P=P[1])

"Compressor analysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(WorkFluid$,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2]
at compressor exit"
h_s[2]=hFunc(WorkFluid$,T_s[2],P[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])  "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]   "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

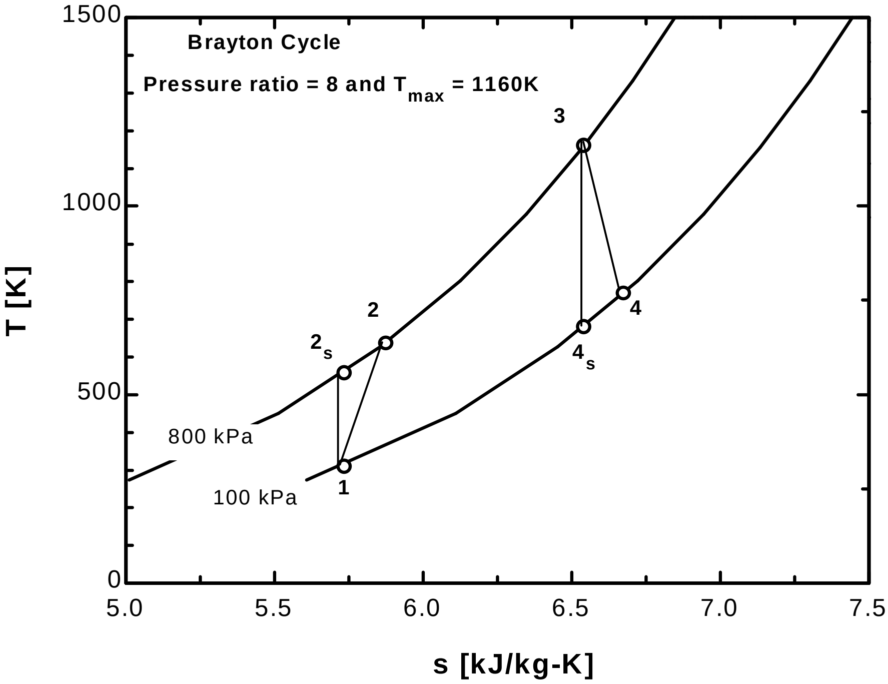
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=hFunc(WorkFluid$,T[3],P[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming
W=0, ke=pe=0"

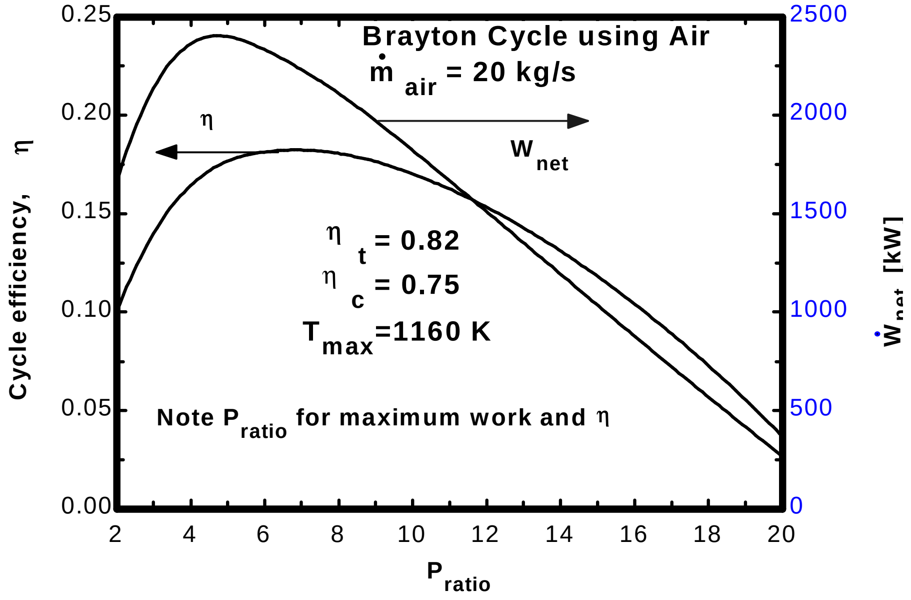
"Turbine analysis"
s[3]=ENTROPY(WorkFluid$,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(WorkFluid$,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4]
at turbine exit"
h_s[4]=hFunc(WorkFluid$,T_s[4],P[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic
efficiency, Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"
```

```
"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta_th=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Call EtaCheck(Eta_th:EtaError$)
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
s[4]=entropy('air',T=T[4],P=P[4])
```

Bwr	η	P_{ratio}	W_c [kW]	W_{net} [kW]	W_t [kW]	Q_{in} [kW]
0.5229	0.1	2	1818	1659	3477	16587
0.6305	0.1644	4	4033	2364	6396	14373
0.7038	0.1814	6	5543	2333	7876	12862
0.7611	0.1806	8	6723	2110	8833	11682
0.8088	0.1702	10	7705	1822	9527	10700
0.85	0.1533	12	8553	1510	10063	9852
0.8864	0.131	14	9304	1192	10496	9102
0.9192	0.1041	16	9980	877.2	10857	8426
0.9491	0.07272	18	10596	567.9	11164	7809
0.9767	0.03675	20	11165	266.1	11431	7241




Brayton Cycle using Air
ṁ air = 20 kg/s
η
Wnet
ηt = 0.82
ηc = 0.75
Tmax=1160 K
Note Pratio for maximum work and η
Cycle efficiency, η
Ẇnet [kW]
Pratio
0.00
0.05
0.10
0.15
0.20
0.25
0
500
1000
1500
2000
2500
2
4
6
8
10
12
14
16
18
20

8-173 Using EES (or other) software, the effects of pressure ratio, maximum cycle temperature, regenerator effectiveness, and compressor and turbine efficiencies on the net work output per unit mass and on the thermal efficiency of a regenerative Brayton cycle with air as the working fluid are to be investigated for the case of constant specific heats for air at room temperature. The net work output and the thermal efficiency are to be determined for all combinations of the following parameters:
Pressure ratio: 6, 10
Maximum cycle temperature: 1500, 2000 K
Compressor adiabatic efficiency: 80, 100 percent
Turbine adiabatic efficiency: 80, 100 percent
Regenerator effectiveness: 70, 90 percent"

```
"For both the compressor and turbine we assume adiabatic, steady-flow,
and neglect KE and PE."
"Input data for air"
C_P = 1.005"[kJ/kg-K]"
k = 1.4

"Other Input data from the diagram window"
{T[3] = 1200"[K]"
Pratio = 10
T[1] = 300"[K]"
P[1]= 100"[kPa]"
Eta_reg = 1.0
Eta_c =0.8"Compressor isentorpic efficiency"
Eta_t =0.9"Turbien isentropic efficiency"}

"Isentropic Compressor analysis"
T_s[2] = T[1]*Pratio^((k-1)/k)
P[2] = Pratio*P[1]
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"
"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
 w_compisen = C_P*(T_s[2]-T[1])
"Actual compressor analysis:"
 w_comp = C_P*(T[2]-T[1])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
q_in_noreg = C_P*(T[3] - T[2])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
P[4] = P[3] /Pratio
T_s[4] = T[3]*(1/Pratio)^((k-1)/k)
"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen = C_P*(T[3] -  T_s[4])

"Actual Turbine analysis:"
w_turb = C_P*(T[3] -  T[4])
```

```
"Cycle analysis"
w_net=w_turb-w_comp "[kJ/kg]"
Eta_th_noreg=w_net/q_in_noreg*100"[%]"    "Cycle thermal efficiency"
Bwr=w_comp/w_turb"Back work ratio"

"With the regenerator the heat added in the external heat exchanger is"
q_in_withreg = C_P*(T[3] - T[5])
P[5]=P[2]
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (T[5]-T[2])/(T[4]-T[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
T[2] + T[4]=T[5]  + T[6]
P[6]=P[4]
"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*100"[%]"
```

η_c	η_t	$\eta_{th,noreg}$ [%]	$\eta_{th,withreg}$ [%]	$q_{in,noreg}$ [kJ/kg]	$q_{in,withreg}$ [kJ/kg]	w_{net} [kJ/kg]
0.6	0.9	14.76	13.92	510.9	541.6	75.4
0.65	0.9	20.35	20.54	546.8	541.6	111.3
0.7	0.9	24.59	26.22	577.5	541.6	142
0.75	0.9	27.91	31.14	604.2	541.6	168.6
0.8	0.9	30.59	35.44	627.5	541.6	192
0.85	0.9	32.79	39.24	648	541.6	212.5
0.9	0.9	34.64	42.61	666.3	541.6	230.8

SOLUTION

Variables in Main
Bwr=0.5214
C_P=1.005 [kJ/kg-K]
Eta_c=0.8
Eta_reg=0.7
Eta_t=0.8
Eta_th_noreg=24.24 [%]
Eta_th_withreg=37.03 [%]
k=1.4
Pratio=6
P[1]=100 [kPa]
P[2]=600 [kPa]
P[3]=600 [kPa]
P[4]=100 [kPa]
P[5]=600 [kPa]
P[6]=100 [kPa]
q_in_noreg=954.1 [kJ/kg]
q_in_withreg=624.5 [kJ/kg]
T[1]=300 [K]
T[2]=550.7 [K]
T[3]=1500 [K]
T[4]=1019 [K]
T[5]=878.7 [K]
T[6]=691.2 [K]
T_s[2]=500.6 [K]
T_s[4]=899 [K]
w_comp=251.9 [kJ/kg]
w_compisen=201.6 [kJ/kg]
w_net=231.3 [kJ/kg]

w_turb=483.2 [kJ/kg]
w_turbisen=604 [kJ/kg]

8-174 Problem 8-173 is to be repeated by considering the variation of specific heats of air with temperature.

```
"Input data"
"Input data from the diagram window"
{T[3] = 1200"[K]"
Pratio = 10
T[1] = 300"[K]"
P[1]= 100"[kPa]"
Eta_reg = 1.0
Eta_c =0.8"Compressor isentorpic efficiency"
Eta_t =0.9"Turbien isentropic efficiency"}

"Isentropic Compressor analysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[1] + w_compisen = h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])

"Actual compressor analysis:"
h[1] + w_comp = h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
h[2] + q_in_noreg = h[3]
h[3]=ENTHALPY(Air,T=T[3])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[3] /Pratio
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
h[3] = w_turbisen + h_s[4]
h_s[4]=ENTHALPY(Air,T=T_s[4])

"Actual Turbine analysis:"
h[3] = w_turb + h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])

"Cycle analysis"
```

```
w_net=w_turb-w_comp "[kJ/kg]"
Eta_th_noreg=w_net/q_in_noreg*100"[%]"    "Cycle thermal efficiency"
Bwr=w_comp/w_turb"Back work ratio"

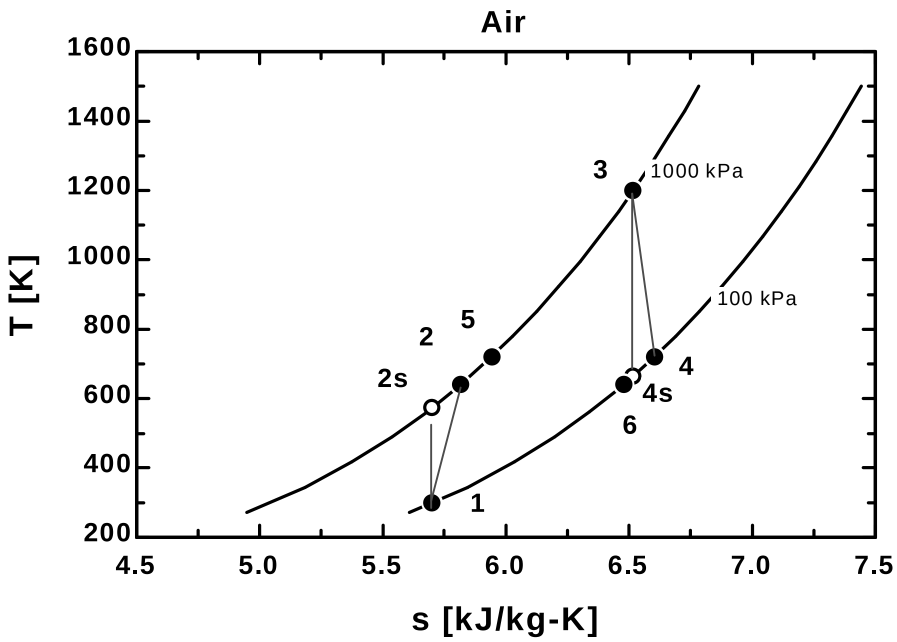
"With the regenerator the heat added in the external heat exchanger is"
h[5] + q_in_withreg = h[3]
h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[2]

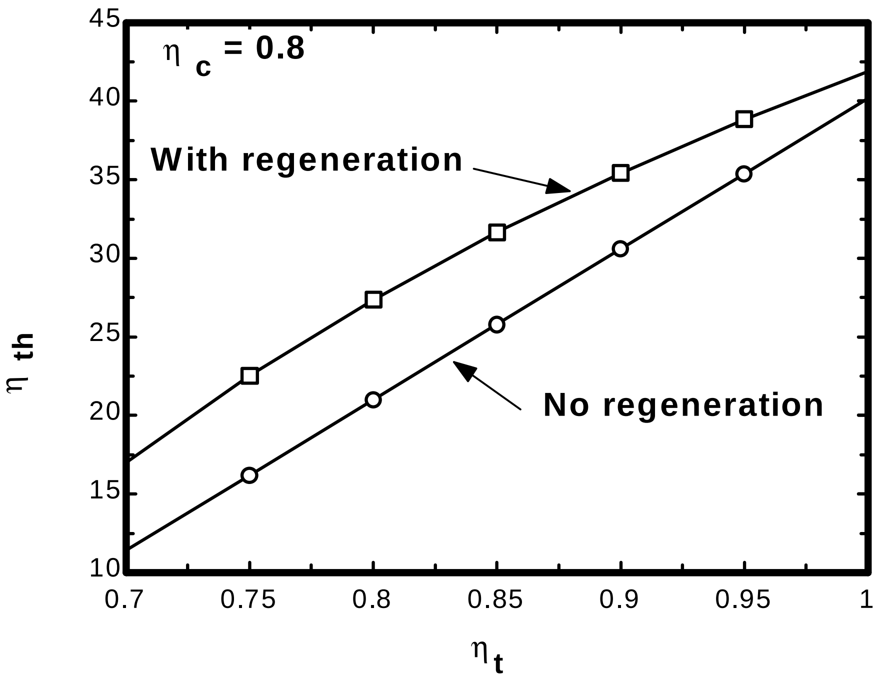
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[2])/(h[4]-h[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
h[2] + h[4]=h[5]  + h[6]
h[6]=ENTHALPY(Air, T=T[6])
s[6]=ENTROPY(Air,T=T[6], P=P[6])
P[6]=P[4]

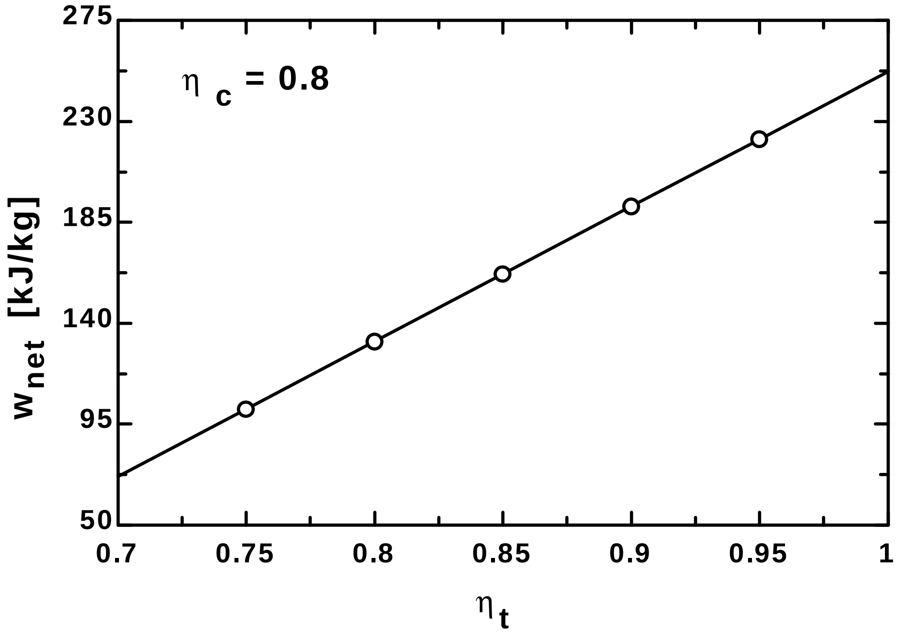
"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*100"[%]"

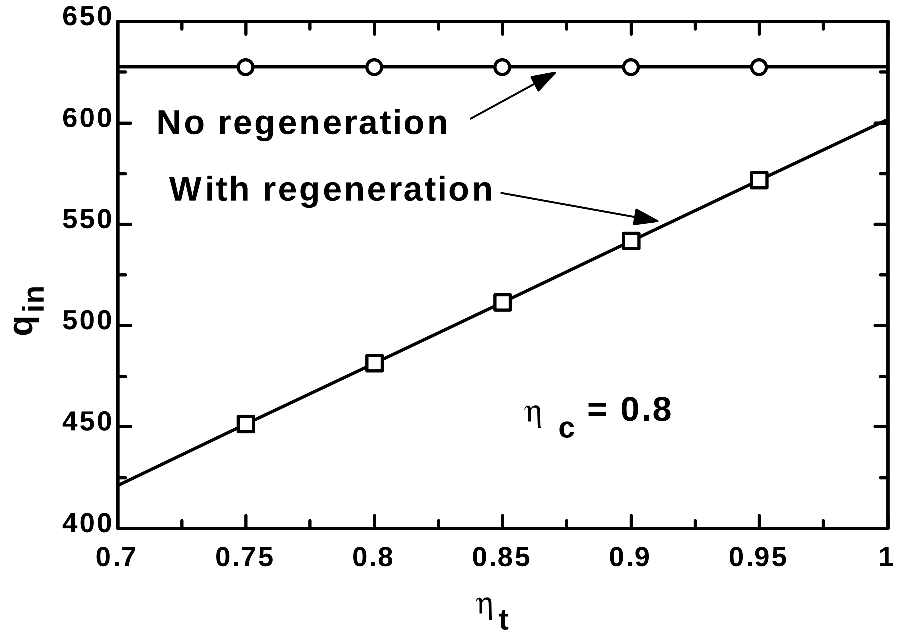
"The following data is used to complete the Array Table for plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=T[6]
```

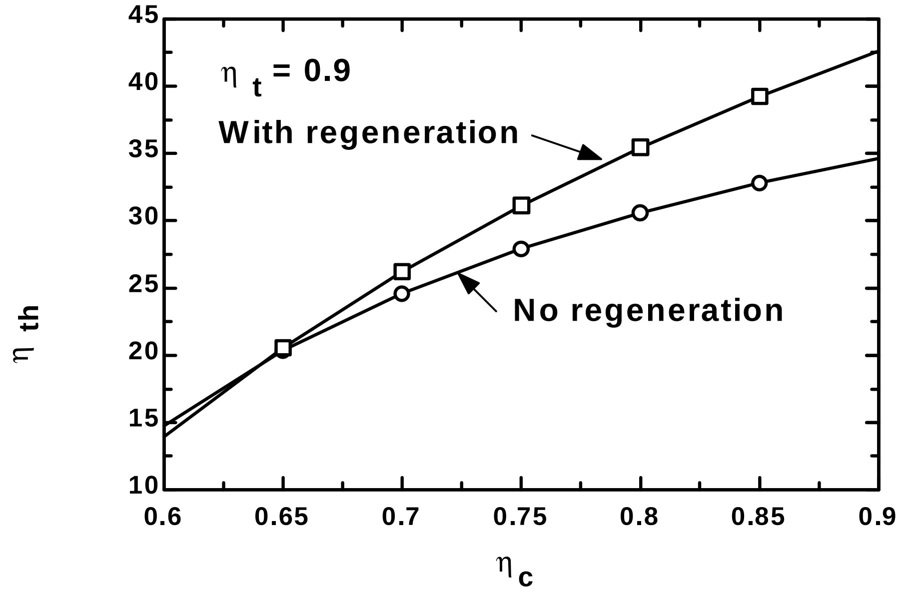
η_c	η_t	$\eta_{th,noreg}$ [%]	$\eta_{th,withreg}$ [%]	$q_{in,noreg}$ [kJ/kg]	$q_{in,withreg}$ [kJ/kg]	w_{net} [kJ/kg]
0.6	0.9	14.76	13.92	510.9	541.6	75.4
0.65	0.9	20.35	20.54	546.8	541.6	111.3
0.7	0.9	24.59	26.22	577.5	541.6	142
0.75	0.9	27.91	31.14	604.2	541.6	168.6
0.8	0.9	30.59	35.44	627.5	541.6	192
0.85	0.9	32.79	39.24	648	541.6	212.5
0.9	0.9	34.64	42.61	666.3	541.6	230.8

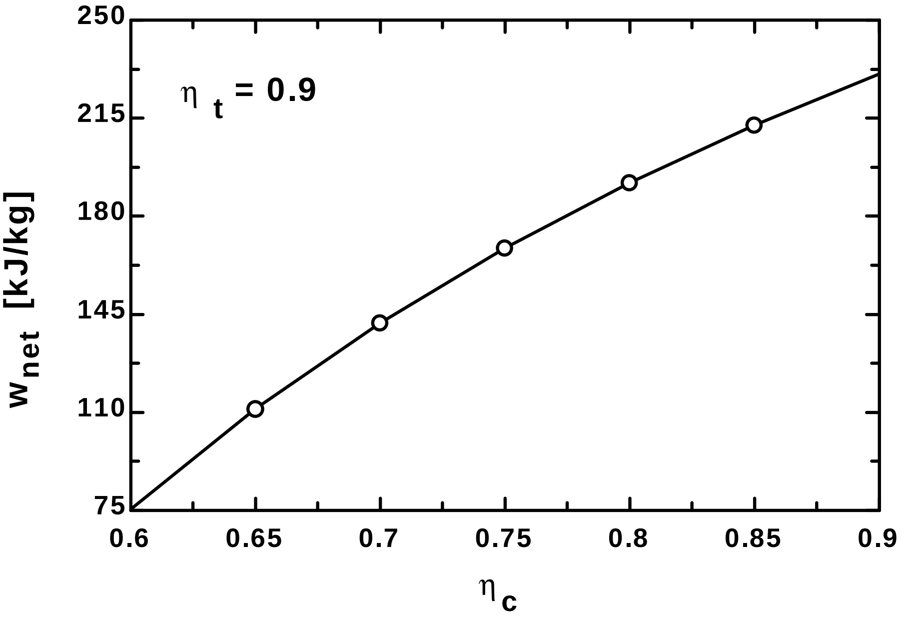

Air
1600
1400
1200
1000
800
600
400
200
4.5
5.0
5.5
6.0
6.5
7.0
7.5
T [K]
s [kJ/kg-K]
1000 kPa
100 kPa
1
2
2s
3
4
4s
5
6

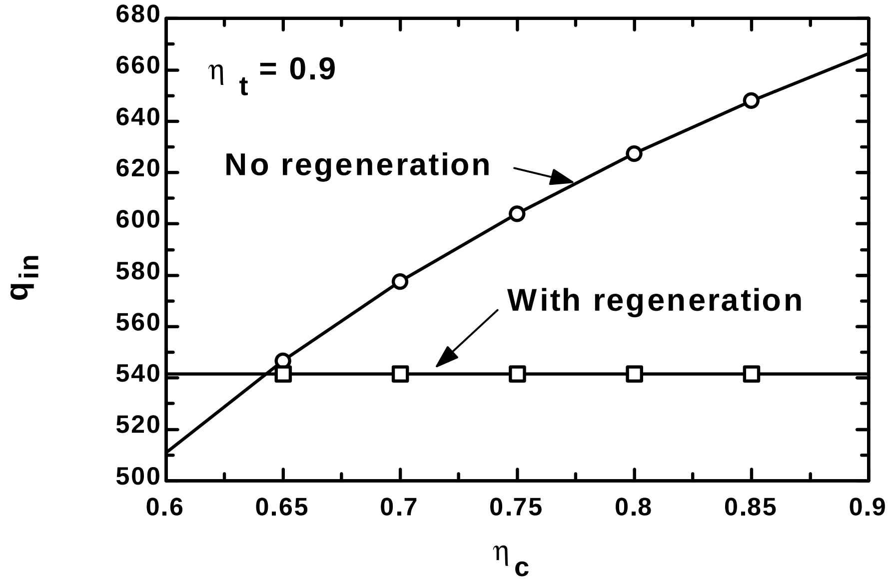

$\eta_c = 0.8$
With regeneration
No regeneration
45
40
35
30
25
20
15
10
0.7
0.75
0.8
0.85
0.9
0.95
1
η_{th}
η_t


275
230
185
140
95
50
0.7
0.75
0.8
0.85
0.9
0.95
1
η c = 0.8
wnet [kJ/kg]
ηt


650
600
550
500
450
400
0.7
0.75
0.8
0.85
0.9
0.95
1
No regeneration
With regeneration
η c = 0.8
qin
ηt


η t = 0.9
With regeneration
No regeneration
η th
45
40
35
30
25
20
15
10
0.6
0.65
0.7
0.75
0.8
0.85
0.9
η c


η t = 0.9
w net [kJ/kg]
250
215
180
145
110
75
0.6
0.65
0.7
0.75
0.8
0.85
0.9
η c



680
660
640
620
600
580
560
540
520
500
η t = 0.9
No regeneration
With regeneration
q_in
0.6
0.65
0.7
0.75
0.8
0.85
0.9
η c

8-175 Problem 8-173 is to be repeated using helium as the working fluid.

```
"Input data for helium"
C_P = 5.1926"[kJ/kg-K]"
k = 1.667

"Other Input data from the diagram window"
{T[3] = 1200"[K]"
Pratio = 10
T[1] = 300"[K]"
P[1]= 100"[kPa]"
Eta_reg = 1.0
Eta_c =0.8"Compressor isentorpic efficiency"
Eta_t =0.9"Turbien isentropic efficiency"}

"Isentropic Compressor analysis"
T_s[2] = T[1]*Pratio^((k-1)/k)
P[2] = Pratio*P[1]
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
 w_compisen = C_P*(T_s[2]-T[1])

"Actual compressor analysis:"
 w_comp = C_P*(T[2]-T[1])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
q_in_noreg = C_P*(T[3] - T[2])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
P[4] = P[3] /Pratio
T_s[4] = T[3]*(1/Pratio)^((k-1)/k)
"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen = C_P*(T[3] -  T_s[4])

"Actual Turbine analysis:"
w_turb = C_P*(T[3] -  T[4])

"Cycle analysis"
w_net=w_turb-w_comp "[kJ/kg]"
Eta_th_noreg=w_net/q_in_noreg*100"[%]"    "Cycle thermal efficiency"
Bwr=w_comp/w_turb"Back work ratio"

"With the regenerator the heat added in the external heat exchanger is"
q_in_withreg = C_P*(T[3] - T[5])
P[5]=P[2]

"The regenerator effectiveness gives h[5] and thus T[5] as:"
```

Eta_reg = (T[5]-T[2])/(T[4]-T[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
T[2] + T[4]=T[5] + T[6]
P[6]=P[4]

"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*100"[%]"

η_c	η_t	$\eta_{th,noreg}$ [%]	$\eta_{th,withreg}$ [%]	$q_{in,noreg}$ [kJ/kg]	$q_{in,withreg}$ [kJ/kg]	w_{net} [kJ/kg]
0.6	0.9	14.76	13.92	510.9	541.6	75.4
0.65	0.9	20.35	20.54	546.8	541.6	111.3
0.7	0.9	24.59	26.22	577.5	541.6	142
0.75	0.9	27.91	31.14	604.2	541.6	168.6
0.8	0.9	30.59	35.44	627.5	541.6	192
0.85	0.9	32.79	39.24	648	541.6	212.5
0.9	0.9	34.64	42.61	666.3	541.6	230.8

SOLUTION
Variables in Main
Bwr=0.8
C_P=5.193 [kJ/kg-K]
Eta_c=0.8
Eta_reg=0.7
Eta_t=0.8
Eta_th_noreg=19.38 [%]
Eta_th_withreg=19.81 [%]
k=1.667
Pratio=6
P[1]=100 [kPa]
P[2]=600 [kPa]
P[3]=600 [kPa]
P[4]=100 [kPa]
P[5]=600 [kPa]
P[6]=100 [kPa]
q_in_noreg=2632 [kJ/kg]
q_in_withreg=2575 [kJ/kg]
T[1]=300 [K]
T[2]=693 [K]
T[3]=1200 [K]
T[4]=708.7 [K]
T[5]=704 [K]
T[6]=697.7 [K]
T_s[2]=614.4 [K]
T_s[4]=585.9 [K]
w_comp=2041 [kJ/kg]
w_compisen=1633 [kJ/kg]
w_net=510.1 [kJ/kg]
w_turb=2551 [kJ/kg]
w_turbisen=3189 [kJ/kg]

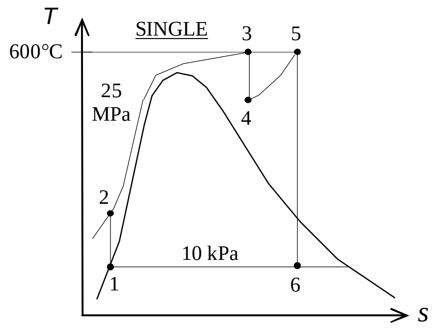
8-176 A steam power plant operating on the ideal Rankine cycle with reheating is considered. The reheat pressures of the cycle are to be determined for the cases of single and double reheat.

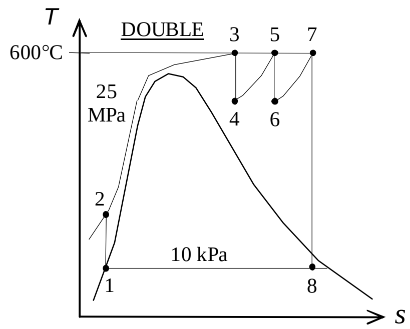
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

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

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ x_6 = 0.88 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.83 + (0.88)(2392.8) = 2297.5 \text{ kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6493 + (0.88)(7.5009) = 7.250 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_5 = 600°\text{C} \\ s_5 = s_6 \end{array}\right\} P_5 = \mathbf{5.098\ MPa}$$





(*b*) Double Reheat :

$$\left.\begin{array}{l} P_3 = 25 \text{ MPa} \\ T_3 = 600°\text{C} \end{array}\right\} s_3 = 6.3602 \text{ kJ/kg}\cdot\text{K}$$

$$\begin{array}{l} P_4 = P_x \\ s_4 = s_3 \end{array} \quad \text{and} \quad \begin{array}{l} P_5 = P_x \\ T_5 = 600°\text{C} \end{array}$$

Any pressure P_x selected between the limits of 25 MPa and 5.098 MPa will satisfy the requirements, and can be used for the double reheat pressure.

8-177E 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),

$$\begin{aligned}\dot{Q}_{\text{brine}} &= \dot{m}_{\text{brine}} C_p (T_2 - T_1) \\ -22{,}790{,}000 \text{ Btu/h} &= (384{,}286 \text{ lbm/h})(1.03 \text{ Btu/lbm}\cdot{}^\circ\text{F})(T_2 - 325^\circ\text{F}) \\ T_2 &= \mathbf{267.4^\circ F}\end{aligned}$$

(*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,

$$\begin{aligned}\dot{Q}_{\text{air}} &= \dot{m}_{\text{air}} C_p (T_9 - T_8) \\ &= (4{,}195{,}100 \text{ lbm/h})(0.24 \text{ Btu/lbm}\cdot{}^\circ\text{F})(84.5 - 55^\circ\text{F}) \\ &= \mathbf{29.7\ MBtu/h}\end{aligned}$$

(*c*) The mass flow rate of geothermal water at the preheater is determined from the steady-flow energy balance on the geothermal water,

$$\begin{aligned}\dot{Q}_{\text{geo}} &= \dot{m}_{\text{geo}} C_p (T_{\text{out}} - T_{\text{in}}) \\ -11{,}140{,}000 \text{ Btu/h} &= \dot{m}_{\text{geo}} (1.03 \text{ Btu/lbm}\cdot{}^\circ\text{F})(154.0 - 221.8^\circ\text{F}) \\ \dot{m}_{\text{geo}} &= \mathbf{187{,}120\ lbm/h}\end{aligned}$$

(*d*) The rate of heat input is

$$\begin{aligned}\dot{Q}_{\text{in}} &= \dot{Q}_{\text{vaporizer}} + \dot{Q}_{\text{reheater}} = 22{,}790{,}000 + 11{,}140{,}000 \\ &= 33{,}930{,}000 \text{ Btu/h}\end{aligned}$$

and

$$\dot{W}_{\text{net}} = 1271 - 200 = 1071 \text{ kW}$$

Then,

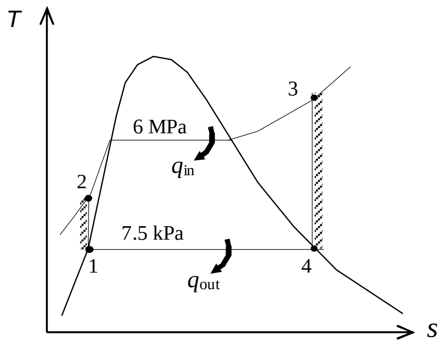
$$\eta_{th} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1071 \text{ kW}}{33{,}930{,}000 \text{ Btu/h}}\left(\frac{3412.14 \text{ Btu}}{1 \text{ kWh}}\right) = \mathbf{10.8\%}$$

8-178 A steam power plant operates on the simple ideal Rankine cycle. The turbine inlet temperature, the net power output, the thermal efficiency, and the minimum mass 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 (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,7.5\text{ kPa}} = 168.79\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,7.5\text{ kPa}} = 0.001008\text{ m}^3\text{/kg}$$
$$T_1 = T_{\text{sat}\,@\,7.5\text{ kPa}} = 40.29°\text{C}$$



$$\begin{aligned} w_{\text{p,in}} &= v_1(P_2 - P_1) \\ &= \left(0.001008\text{ m}^3\text{/kg}\right)\left(6{,}000 - 7.5\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 6.04\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 168.79 + 6.04 = 174.8\text{ kJ/kg}$$

$$h_4 = h_{g\,@\,7.5\text{ kPa}} = 2574.8\text{ kJ/kg}$$
$$s_4 = s_{g\,@\,7.5\text{ kPa}} = 8.2515\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 6\text{ MPa} \\ s_3 = s_4 \end{array}\right\} \begin{array}{l} h_3 = 4856.1\text{ kJ/kg} \\ T_3 = \mathbf{1092.4°C} \end{array}$$

(*b*)

$$q_{\text{in}} = h_3 - h_2 = 4856.1 - 174.8 = 4681.3\text{ kJ / kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2574.8 - 168.79 = 2406\text{ kJ / kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 4681.3 - 2406.0 = 2275.3\text{ kJ / kg}$$

and

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{2275.3\text{ kJ / kg}}{4681.3\text{ kJ / kg}} = \mathbf{48.6\%}$$

Thus,

$$\dot{W}_{\text{net}} = \eta_{th}\dot{Q}_{\text{in}} = (0.486)(60{,}000\text{kJ/s}) = \mathbf{29{,}160\text{ kJ/s}}$$

(*c*) The mass flow rate of the cooling water will be minimum when it is heated to the temperature of the steam in the condenser, which is 40.29°C,

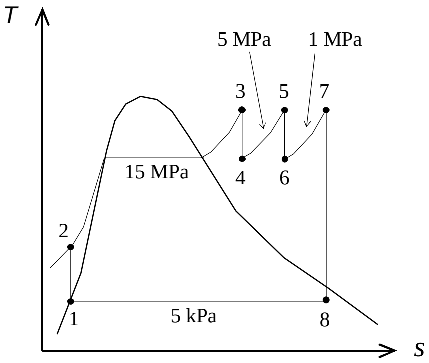
$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net}} = 60{,}000 - 29{,}160 = 30{,}840\text{ kJ/s}$$

$$\dot{m}_{\text{cool}} = \frac{\dot{Q}_{\text{out}}}{C\Delta T} = \frac{30{,}840\text{ kJ/s}}{(4.18\text{ kJ/kg}\cdot°\text{C})(40.29 - 18\,°\text{C})} = \mathbf{331.0\ \ kg/s}$$

8-179 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),



$$h_1 = h_{f\ @\ 5\ \text{kPa}} = 137.82\ \text{kJ/kg}$$
$$v_1 = v_{f\ @\ 5\ \text{kPa}} = 0.001005\ \text{m}^3/\text{kg}$$
$$w_{\text{p,in}} = v_1(P_2 - P_1) = (0.001005\ \text{m}^3/\text{kg})(15{,}000 - 5\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 15.07\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 137.82 + 15.07 = 152.89\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 15\ \text{MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3308.6\ \text{kJ/kg} \\ s_3 = 6.3443\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 5\ \text{MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 3005.7\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 5\ \text{MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3433.8\ \text{kJ/kg} \\ s_5 = 6.9759\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 1\ \text{MPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 2970.7\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 1\ \text{MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3478.5\ \text{kJ/kg} \\ s_7 = 7.7622\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 5\ \text{kPa} \\ s_8 = s_7 \end{array}\right\} \begin{array}{l} x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{7.7622 - 0.4764}{7.9187} = 0.920 \\ h_8 = h_f + x_8 h_{fg} = 137.82 + (0.920)(2423.7) = 2367.8\ \text{kJ/kg} \end{array}$$

Then,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) + (h_7 - h_6) = 3308.6 - 152.89 + 3433.8 - 3005.7 + 3478.5 - 2970.7 = 4091.6\ \text{kJ/kg}$$
$$q_{\text{out}} = h_8 - h_1 = 2367.8 - 137.82 = 2230.0\ \text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 4091.6 - 2230.0 = 1861.6\ \text{kJ/kg}$$

Thus,

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1861.6\text{kJ/kg}}{4091.6\text{kJ/kg}} = \mathbf{45.5\%}$$

(*b*) $$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{120{,}000\text{kJ/s}}{1861.6\text{kJ/kg}} = \mathbf{64.5\ kg/s}$$

8-180 Using EES (or other) software, the effect of the condenser pressure on the performance a simple ideal Rankine cycle is to be investigated. Turbine inlet conditions of steam are maintained constant at 5 MPa and 500°C while the condenser pressure is varied from 5 kPa to 100 kPa. The thermal efficiency of the cycle is to be determined and plotted against the condenser pressure.

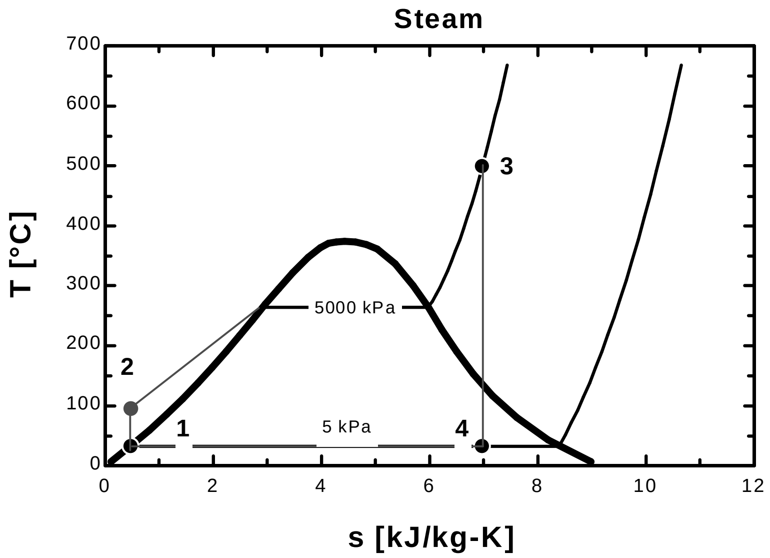
"Let's modify this problem to include the effects of the turbine and pump efficiencies and also show the effects of reheat on the steam quality at the low pressure turbine exit."

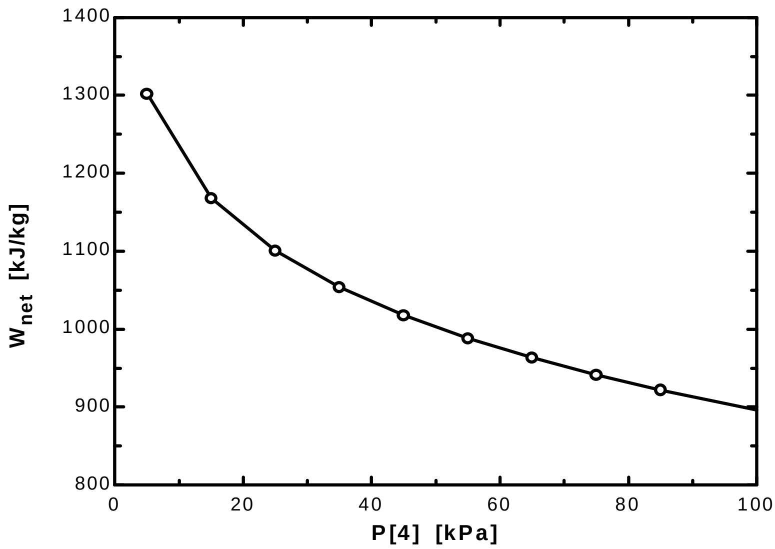
```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
   x4$=''
   if (x4>1) then x4$='(superheated)'
   if (x4<0) then x4$='(compressed)'
end

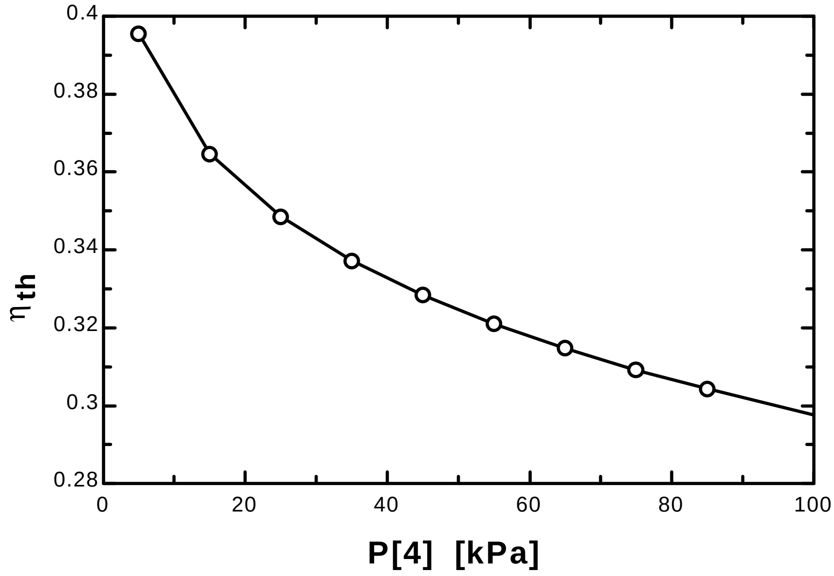
P[3] = 5000"[kPa]"
T[3] = 500"[C]"
P[4] = 5"[kPa]"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"

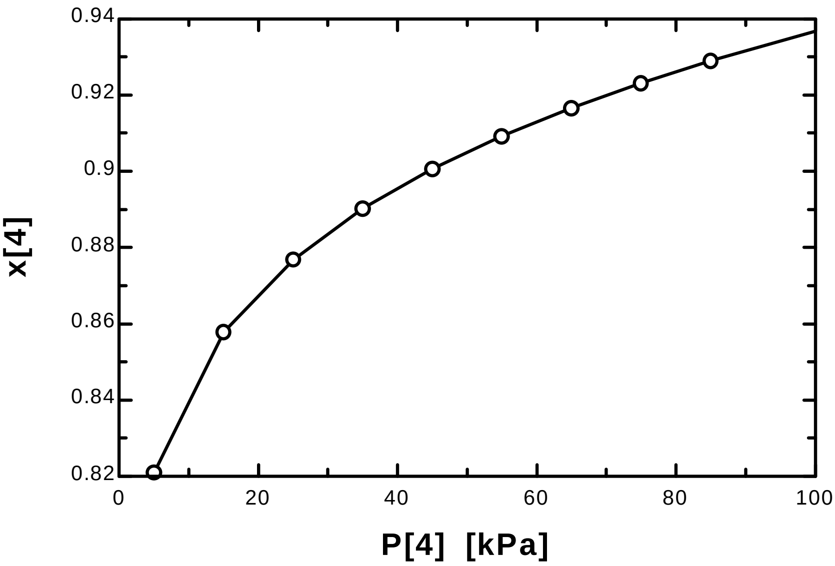
"Pump analysis"
P[1] = P[4]
P[2]=P[3]
x[1]=0      "Sat'd liquid"
h[1]=enthalpy(STEAM,P=P[1],x=x[1])
v[1]=volume(STEAM,P=P[1],x=x[1])
s[1]=entropy(STEAM,P=P[1],x=x[1])
T[1]=temperature(STEAM,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
s[2]=entropy(STEAM,P=P[2],h=h[2])
T[2]=temperature(STEAM,P=P[2],h=h[2])
"Turbine analysis"
h[3]=enthalpy(STEAM,T=T[3],P=P[3])
s[3]=entropy(STEAM,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(STEAM,s=s_s[4],P=P[4])
Ts[4]=temperature(STEAM,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(STEAM,P=P[4],h=h[4])
s[4]=entropy(STEAM,h=h[4],P=P[4])
x[4]=quality(STEAM,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	0.821	3291	1989
0.3646	15	1168	0.8578	3203	2035
0.3485	25	1100	0.8769	3157	2057
0.3372	35	1054	0.8902	3125	2071
0.3284	45	1018	0.9006	3099	2082
0.321	55	988.3	0.9092	3078	2090
0.3148	65	963.3	0.9166	3060	2097
0.3092	75	941.5	0.9231	3044	2103
0.3043	85	922.1	0.929	3030	2108
0.2977	100	896.5	0.9367	3011	2115






0.4
0.38
0.36
0.34
0.32
0.3
0.28
0
20
40
60
80
100
η_{th}
P[4] [kPa]


0.94
0.92
0.9
0.88
0.86
0.84
0.82
0
20
40
60
80
100
x[4]
P[4] [kPa]

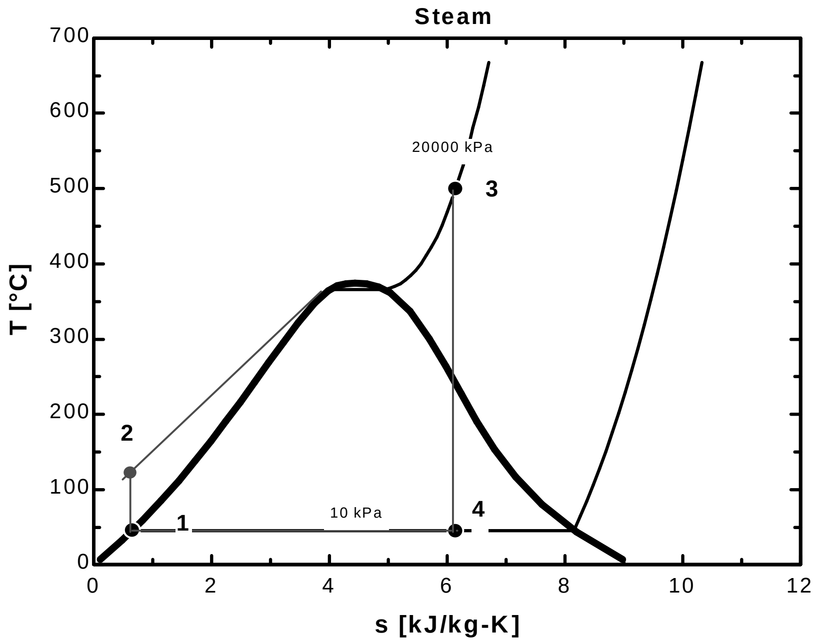
8-181 Using EES (or other) software, the effect of the boiler pressure on the performance of a simple ideal Rankine cycle is to be investigated. Steam enters the turbine at 500°C and exits at 10 kPa, and the boiler pressure is varied from 0.5 MPa to 20 MPa. The thermal efficiency of the cycle is to be determined and plotted against the boiler pressure.

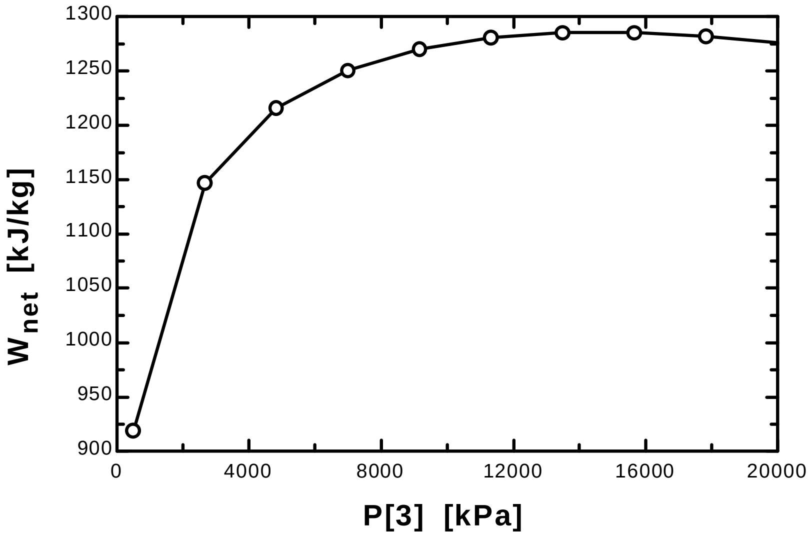
```
Let's modify this problem to include the effects of the turbine and pump efficiencies and also
show the effects of reheat on the steam quality at the low pressure turbine exit."

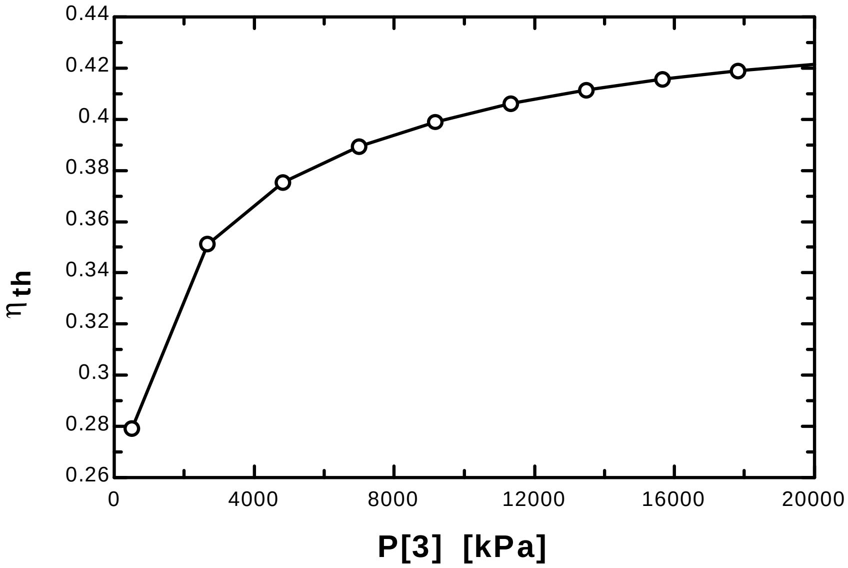
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
    x4$=''
    if (x4>1) then x4$='(superheated)'
    if (x4<0) then x4$='(compressed)'
end

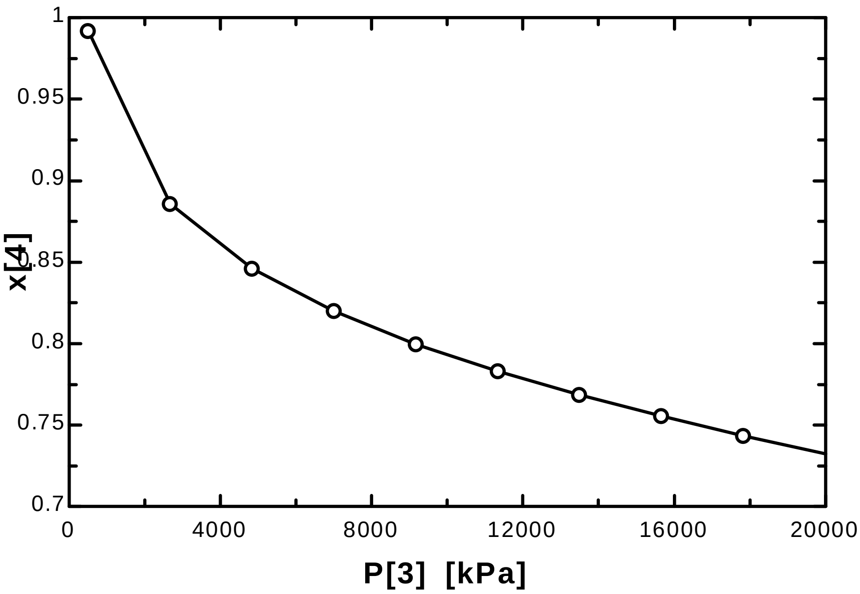
{P[3] = 20000"[kPa]"}
T[3] = 500"[C]"
P[4] = 10"[kPa]"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"
"Pump analysis"
P[1] = P[4]
P[2]=P[3]
x[1]=0      "Sat'd liquid"
h[1]=enthalpy(STEAM,P=P[1],x=x[1])
v[1]=volume(STEAM,P=P[1],x=x[1])
s[1]=entropy(STEAM,P=P[1],x=x[1])
T[1]=temperature(STEAM,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(STEAM,P=P[2],h=h[2])
T[2]=temperature(STEAM,P=P[2],h=h[2])
"Turbine analysis"
h[3]=enthalpy(STEAM,T=T[3],P=P[3])
s[3]=entropy(STEAM,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(STEAM,s=s_s[4],P=P[4])
Ts[4]=temperature(STEAM,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(STEAM,P=P[4],h=h[4])
s[4]=entropy(STEAM,h=h[4],P=P[4])
x[4]=quality(STEAM,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
```

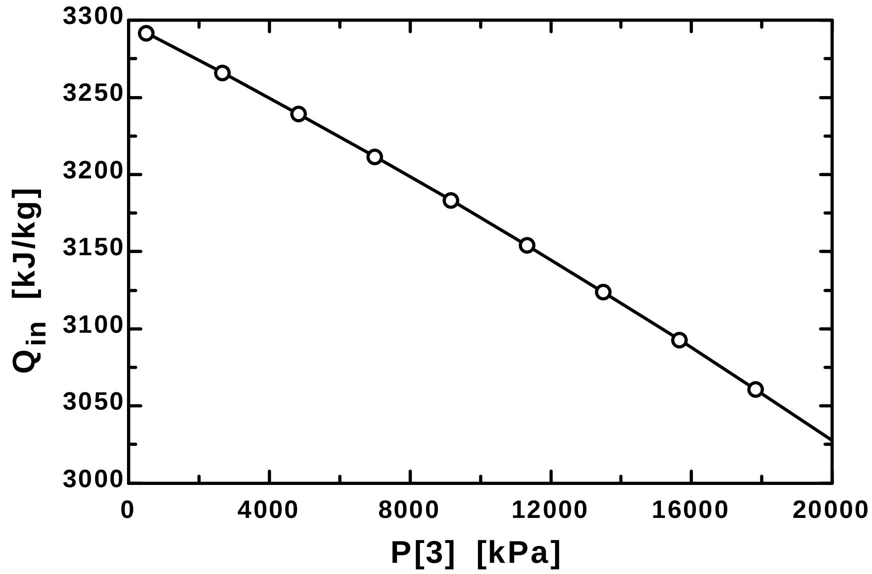
η_{th}	W_{net} [kJ/kg]	x_4	P_3 [kPa]	Q_{in} [kJ/kg]	Q_{out} [kJ/kg]	W_p [kJ/kg]	W_t [kJ/kg]
0.2793	919.3	0.9918	500	3292	2372	0.495	919.8
0.3513	1147	0.8858	2667	3266	2119	2.684	1150
0.3753	1216	0.846	4833	3239	2024	4.873	1221
0.3894	1251	0.8199	7000	3212	1961	7.062	1258
0.399	1270	0.7998	9167	3183	1913	9.251	1279
0.406	1281	0.7832	11333	3154	1873	11.44	1292
0.4114	1285	0.7687	13500	3124	1839	13.63	1299
0.4156	1285	0.7556	15667	3093	1807	15.82	1301
0.4188	1282	0.7436	17833	3061	1779	18.01	1300
0.4213	1276	0.7324	20000	3028	1752	20.2	1296

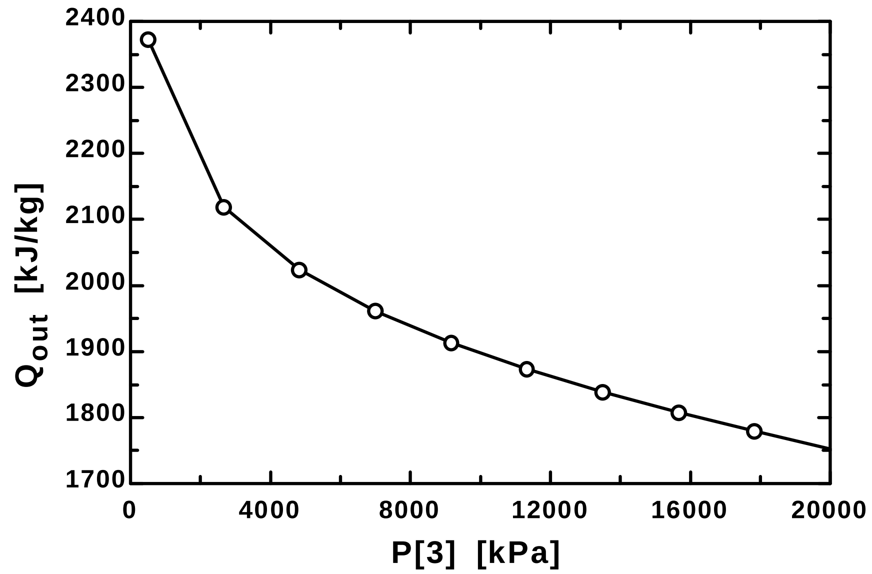


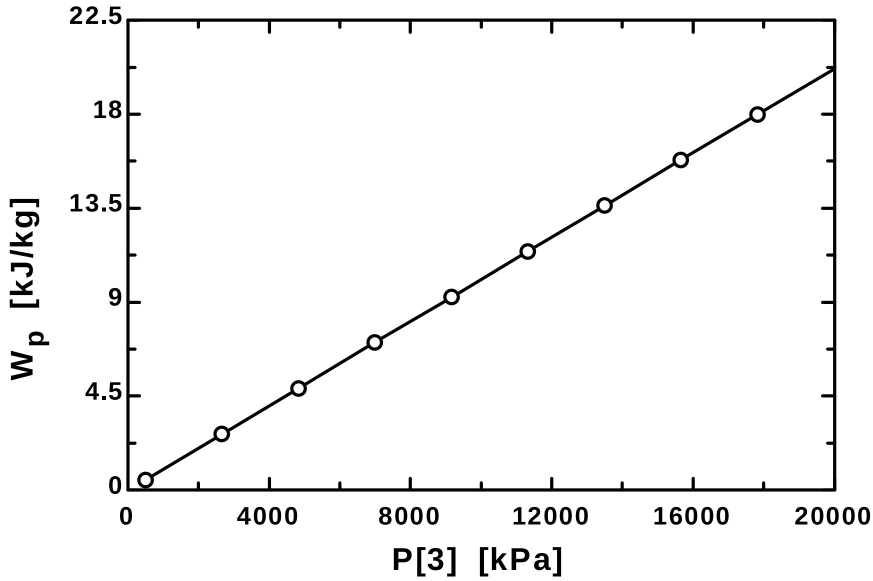


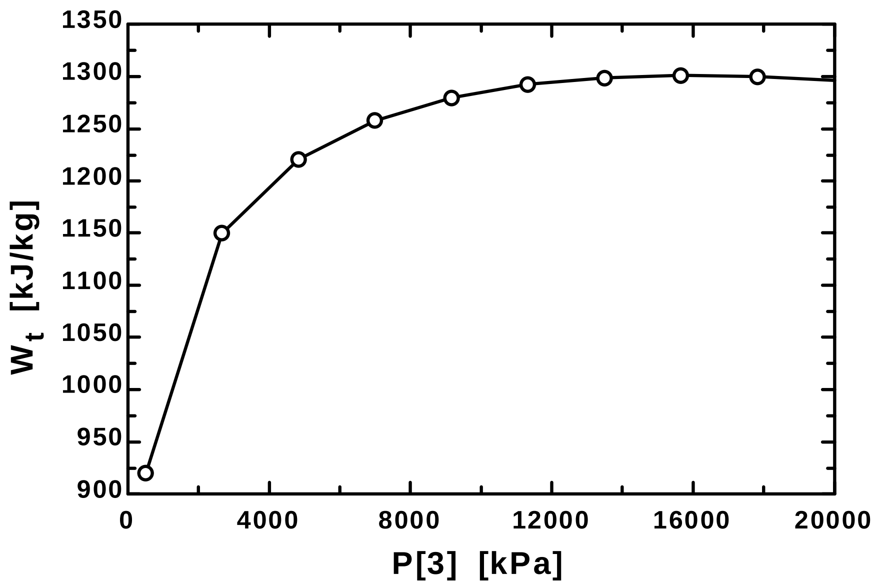

0.44
0.42
0.4
0.38
0.36
0.34
0.32
0.3
0.28
0.26
η th
0
4000
8000
12000
16000
20000
P[3] [kPa]


1
0.95
0.9
0.85
0.8
0.75
0.7
x[4]
0
4000
8000
12000
16000
20000
P[3] [kPa]


3300
3250
3200
3150
3100
3050
3000
0
4000
8000
12000
16000
20000
Q in [kJ/kg]
P[3] [kPa]


2400
2300
2200
2100
2000
1900
1800
1700
0
4000
8000
12000
16000
20000
Q out [kJ/kg]
P[3] [kPa]


22.5
18
13.5
9
4.5
0
0
4000
8000
12000
16000
20000
P[3] [kPa]
W_p [kJ/kg]


1350
1300
1250
1200
1150
1100
1050
1000
950
900
0
4000
8000
12000
16000
20000
P[3] [kPa]
W_t [kJ/kg]

8-182 Using EES (or other) software, the effect of superheating the steam on the performance of a simple ideal Rankine cycle is to be investigated. Steam enters the turbine at 3 MPa and exits at 10 kPa, and the turbine inlet temperature is varied from 250°C to 1100°C. The thermal efficiency of the cycle is to be determined and plotted against the turbine inlet temperature.

```
Let's modify this problem to include the effects of the turbine and pump efficiencies and
also
show the effects of reheat on the steam quality at the low pressure turbine exit."

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"
P[1] = P[4]
P[2]=P[3]
x[1]=0      "Sat'd liquid"
h[1]=enthalpy(STEAM,P=P[1],x=x[1])
v[1]=volume(STEAM,P=P[1],x=x[1])
s[1]=entropy(STEAM,P=P[1],x=x[1])
T[1]=temperature(STEAM,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(STEAM,P=P[2],h=h[2])
T[2]=temperature(STEAM,P=P[2],h=h[2])

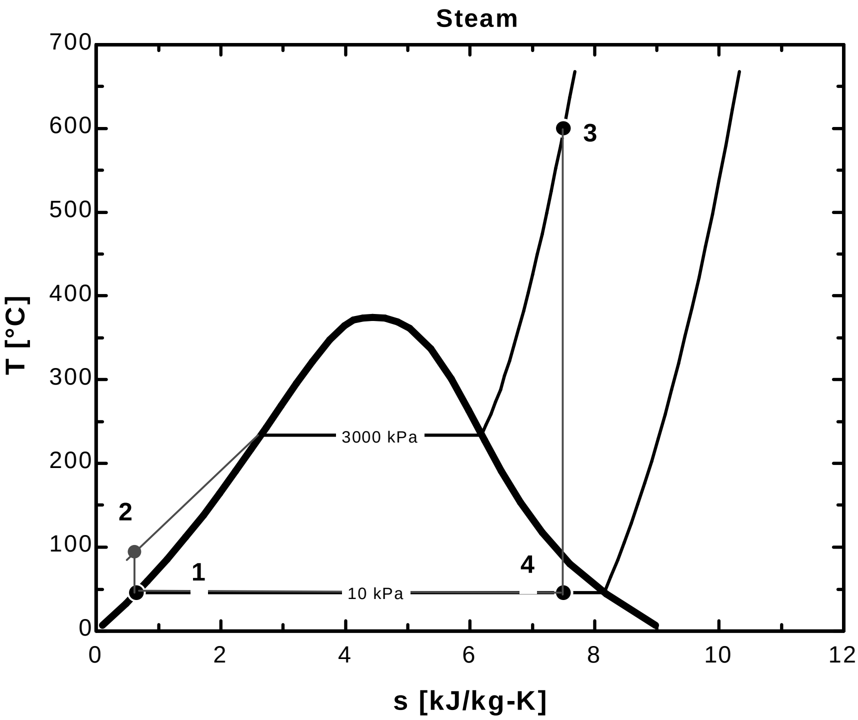
"Turbine analysis"
h[3]=enthalpy(STEAM,T=T[3],P=P[3])
s[3]=entropy(STEAM,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(STEAM,s=s_s[4],P=P[4])
Ts[4]=temperature(STEAM,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(STEAM,P=P[4],h=h[4])
s[4]=entropy(STEAM,h=h[4],P=P[4])
x[4]=quality(STEAM,h=h[4],P=P[4])
h[3] =W_t+h[4]"SSSF First Law for the turbine"
x4s$=x4$(x[4])

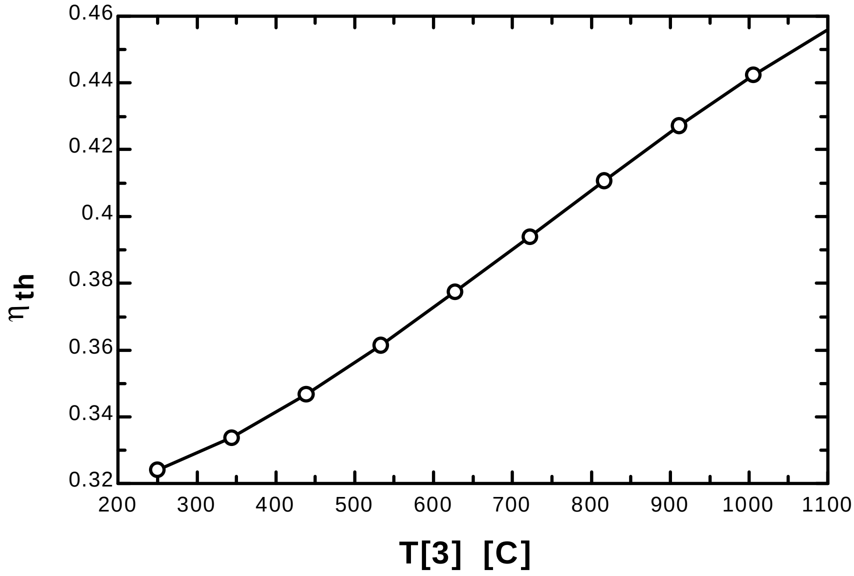
"Boiler analysis"
Q_in + h[2]=h[3]"SSSF First Law for the Boiler"

"Condenser analysis"
h[4]=Q_out+h[1]"SSSF First Law for the Condenser"

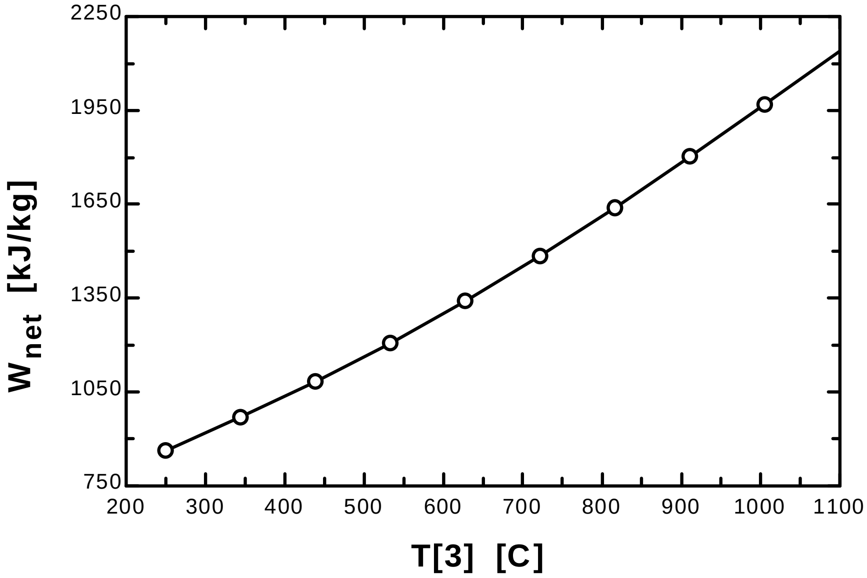
"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

η_{th}	W_{net} [kJ/kg]	x_4	T_3 [C]
0.3242	862.3	0.7516	250
0.3338	970.3	0.8096	344.4
0.3467	1083	0.8533	438.9
0.3615	1206	0.8907	533.3
0.3775	1340	0.9241	627.8
0.394	1485	0.9548	722.2
0.4107	1639	0.9833	816.7
0.4272	1803	100	911.1
0.4424	1970	100	1006
0.456	2139	100	1100





0.46
0.44
0.42
0.4
0.38
0.36
0.34
0.32
200
300
400
500
600
700
800
900
1000
1100
η_{th}
T[3] [C]



2250
1950
1650
1350
1050
750
200
300
400
500
600
700
800
900
1000
1100
W_{net} [kJ/kg]
T[3] [C]

8-183 Using EES (or other) software, the effect of reheat pressure on the performance of an ideal reheat Rankine cycle is to be investigated. The maximum and minimum pressures in the cycle are 15 MPa and 10 kPa, respectively, and steam enters both stages of the turbine at 500°C. The reheat pressure is varied from 12.5 MPa to 0.5 MPa. The the thermal efficiency of the cycle is to be calculated and plotted against the reheat pressure.

```
“Let's modify this problem to include the effects of the turbine and pump efficiencies and
also
 show the effects of reheat on the steam quality at the low pressure turbine exit."

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

P[6] = 10"kPa"
P[3] = 15000"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"
P[1] = P[6]
P[2]=P[3]
x[1]=0   "Sat'd liquid"
h[1]=enthalpy(STEAM,P=P[1],x=x[1])
v[1]=volume(STEAM,P=P[1],x=x[1])
s[1]=entropy(STEAM,P=P[1],x=x[1])
T[1]=temperature(STEAM,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(STEAM,P=P[2],h=h[2])
s[2]=entropy(STEAM,P=P[2],h=h[2])
T[2]=temperature(STEAM,P=P[2],h=h[2])

"High Pressure Turbine analysis"
h[3]=enthalpy(STEAM,T=T[3],P=P[3])
s[3]=entropy(STEAM,T=T[3],P=P[3])
v[3]=volume(STEAM,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(STEAM,s=s_s[4],P=P[4])
Ts[4]=temperature(STEAM,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(STEAM,P=P[4],h=h[4])
s[4]=entropy(STEAM,T=T[4],P=P[4])
v[4]=volume(STEAM,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(STEAM,T=T[5],P=P[5])
h[5]=enthalpy(STEAM,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(STEAM,s=s_s[6],P=P[6])
```

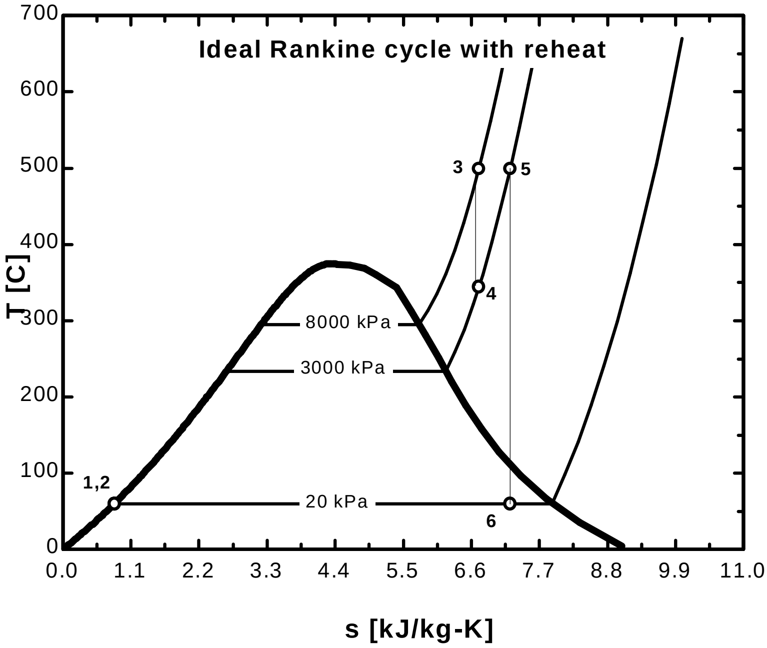
```
Ts[6]=temperature(STEAM,s=s_s[6],P=P[6])
vs[6]=volume(STEAM,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(STEAM,h=h[6],P=P[6])

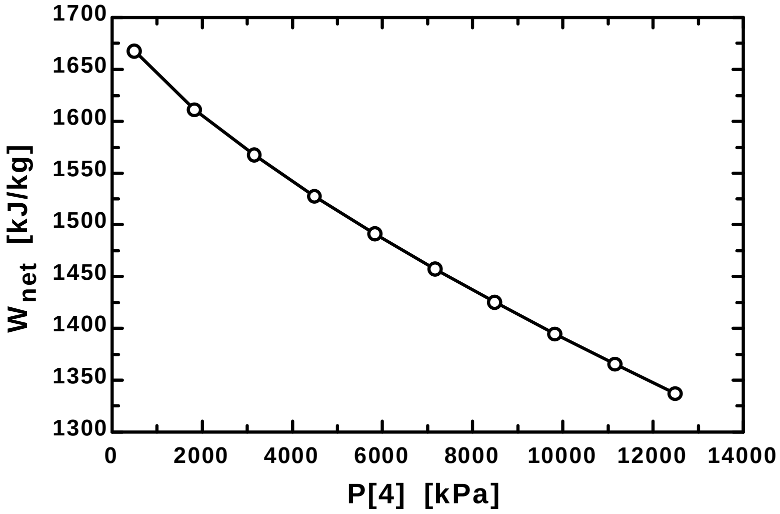
"Boiler analysis"
Q_in + h[2]+h[4]=h[3]+h[5]"SSSF First Law for the Boiler"

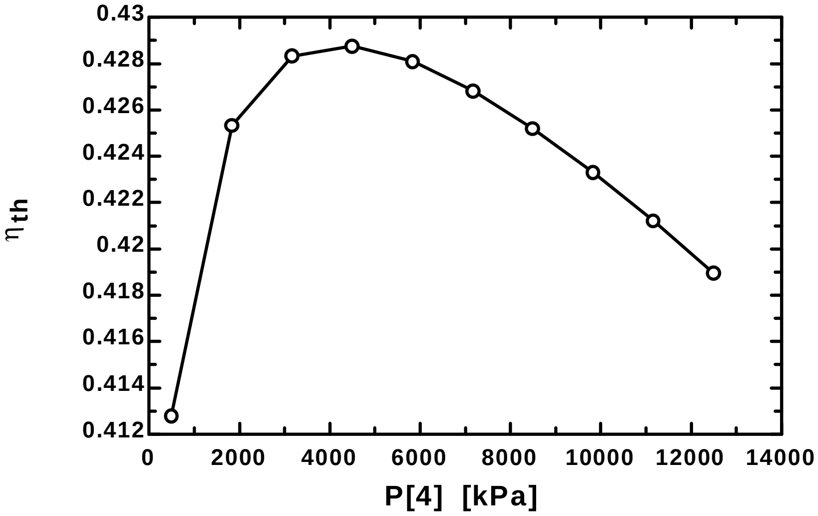
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature('steam',h=h[6],P=P[6])
s[6]=entropy('steam',h=h[6],P=P[6])
x6s$=x6$(x[6])

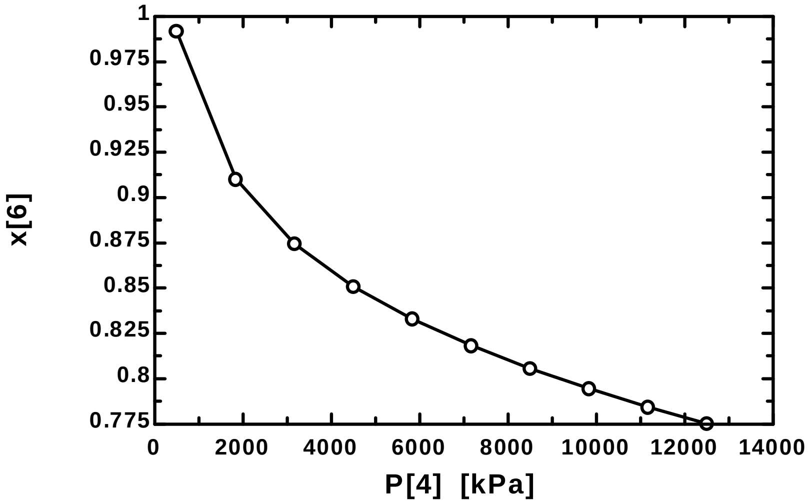
"Cycle Statistics"
W_net=W_t_hp+W_t_lp-W_p
Eta_th=W_net/Q_in
```

η_{th}	P_4 [kPa]	w_{net} [kJ/kg]	x_6
0.4128	500	1668	0.9918
0.4253	1833	1611	0.91
0.4283	3167	1567	0.8745
0.4288	4500	1528	0.8509
0.4281	5833	1491	0.8329
0.4268	7167	1457	0.8182
0.4252	8500	1425	0.8055
0.4233	9833	1395	0.7944
0.4212	11167	1365	0.7844
0.419	12500	1337	0.7752




1700
1650
1600
1550
1500
1450
1400
1350
1300
Wnet [kJ/kg]
0
2000
4000
6000
8000
10000
12000
14000
P[4] [kPa]


0.43
0.428
0.426
0.424
0.422
0.42
0.418
0.416
0.414
0.412
ηth
0
2000
4000
6000
8000
10000
12000
14000
P[4] [kPa]


x[6]
P[4] [kPa]
1
0.975
0.95
0.925
0.9
0.875
0.85
0.825
0.8
0.775
0
2000
4000
6000
8000
10000
12000
14000

8-184 Using EES (or other) software, the effect of number of reheat stages on the performance of an ideal reheat Rankine cycle is to be investigated. The thermal efficiency of the cycle is to be determined, and it is to be plotted against the number of reheat stages of 1, 2, 4, and 8.

```
Let's modify this problem to include the effects of the turbine and pump efficiencies and
also
show the effects of reheat on the steam quality at the low pressure turbine exit."

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"kPa"

P5=P4
P6=P5*R_P
s5=entropy(STEAM,T=T5,P=P5)
h5=enthalpy(STEAM,T=T5,P=P5)
s_s6=s5
hs6=enthalpy(STEAM,s=s_s6,P=P6)
Ts6=temperature(STEAM,s=s_s6,P=P6)
vs6=volume(STEAM,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(STEAM,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"

"Pump analysis"
P[1] = P[6]
P[2]=P[3]
x[1]=0      "Sat'd liquid"
h[1]=enthalpy(STEAM,P=P[1],x=x[1])
v[1]=volume(STEAM,P=P[1],x=x[1])
s[1]=entropy(STEAM,P=P[1],x=x[1])
T[1]=temperature(STEAM,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(STEAM,P=P[2],h=h[2])
s[2]=entropy(STEAM,P=P[2],h=h[2])
T[2]=temperature(STEAM,P=P[2],h=h[2])

"High Pressure Turbine analysis"
h[3]=enthalpy(STEAM,T=T[3],P=P[3])
s[3]=entropy(STEAM,T=T[3],P=P[3])
v[3]=volume(STEAM,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(STEAM,s=s_s[4],P=P[4])
Ts[4]=temperature(STEAM,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(STEAM,P=P[4],h=h[4])
s[4]=entropy(STEAM,T=T[4],P=P[4])
v[4]=volume(STEAM,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"

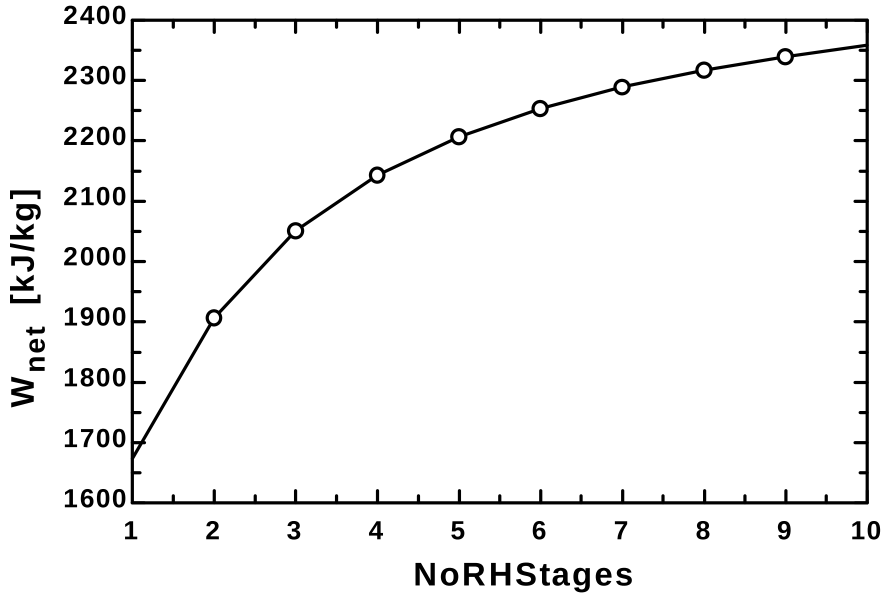
"Low Pressure Turbine analysis"
Call Reheat(P[3],T[3],T[5],h[4],NoRHStages,Pratio,Eta_t:Q_in_reheat,W_t_lp,h6)
h[6]=h6

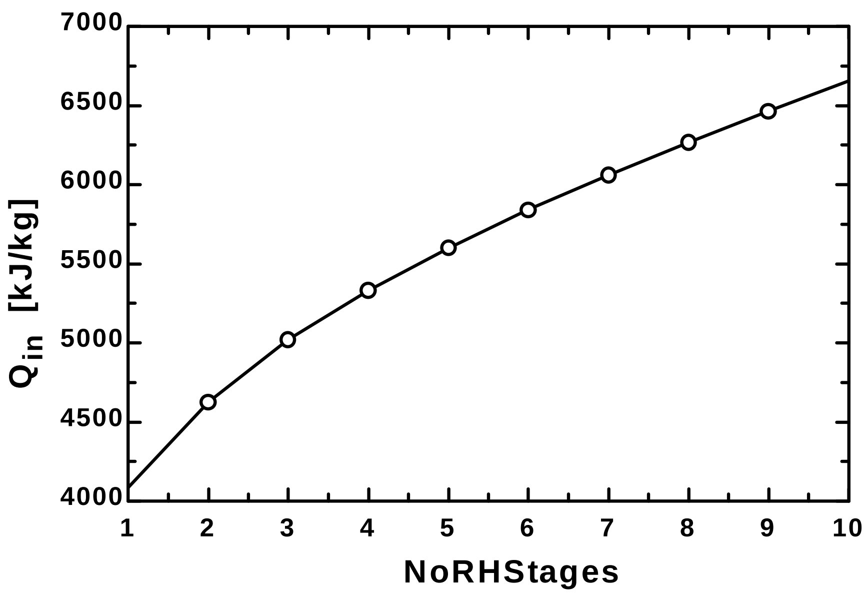
{P[5]=P[4]
s[5]=entropy(STEAM,T=T[5],P=P[5])
h[5]=enthalpy(STEAM,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(STEAM,s=s_s[6],P=P[6])
Ts[6]=temperature(STEAM,s=s_s[6],P=P[6])
vs[6]=volume(STEAM,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(STEAM,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('steam',h=h[6],P=P[6])
s[6]=entropy('steam',h=h[6],P=P[6])
x[6]=QUALITY(STEAM,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}	NoRHStages	Q_{in} [kJ/kg]	W_{net} [kJ/kg]
0.4097	1	4084	1673
0.4122	2	4627	1907
0.4084	3	5021	2050
0.4017	4	5335	2143
0.3939	5	5602	2206
0.3858	6	5841	2253
0.3776	7	6061	2289
0.3696	8	6268	2317
0.3618	9	6466	2339
0.3543	10	6656	2358

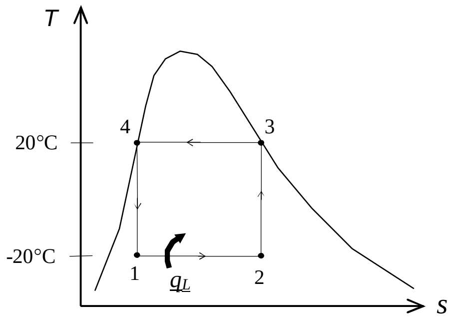




8-185 A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The COP, the condenser and evaporator pressures, and the net work input are to be determined.

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

Analysis (*a*) The COP of this refrigeration cycle is determined from



$$\mathrm{COP}_{\mathrm{R,C}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(293\ \mathrm{K})/(253\ \mathrm{K}) - 1} = \mathbf{6.33}$$

(*b*) The condenser and evaporative pressures are (Table A-11)

$$P_{\mathrm{evap}} = P_{\mathrm{sat\ @\ -20°C}} = \mathbf{0.13299\ kPa}$$
$$P_{\mathrm{cond}} = P_{\mathrm{sat\ @\ 20°C}} = \mathbf{0.57160\ kPa}$$

(*c*) The net work input is determined from

$$h_1 = (h_f + x_1 h_{fg})_{@-20°\mathrm{C}} = 24.26 + (0.2)(211.05) = 66.5\ \mathrm{kJ/kg}$$
$$h_2 = (h_f + x_2 h_{fg})_{@-20°\mathrm{C}} = 24.26 + (0.85)(211.05) = 203.7\ \mathrm{kJ/kg}$$

$$q_L = h_2 - h_1 = 203.7 - 66.5 = 137.2\ \mathrm{kJ/kg}$$

and

$$w_{\mathrm{net,in}} = \frac{q_L}{\mathrm{COP_R}} = \frac{137.2\ \mathrm{kJ/kg}}{6.33} = \mathbf{21.7\ kJ/kg}$$

8-186 A large refrigeration plant that operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid is considered. The mass flow rate of the refrigerant, the power input to the compressor, and the mass flow rate of the cooling water are to be determined.

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

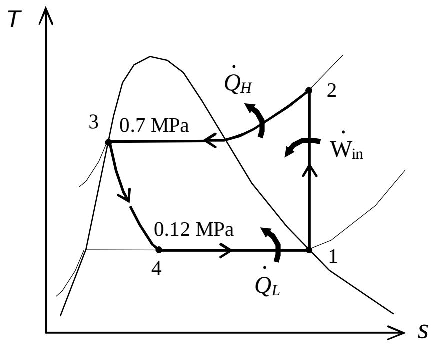
Analysis In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,120\text{ kPa}} = 233.86\text{ kJ/kg} \\ s_1 = s_{g\,@\,120\text{ kPa}} = 0.9354\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 270.22\text{ kJ/kg} \quad (T_2 = 34.6°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.7\text{ MPa}} = 86.78\text{ kJ/kg}$$

$$h_4 \cong h_3 = 86.78\text{ kJ/kg} \quad (\text{throttling})$$



The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{100\text{ kJ/s}}{(233.86 - 86.78)\text{ kJ/kg}} = \mathbf{0.68\ kg/s}$$

(*b*) The power input to the compressor is

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.68\text{ kg/s})(270.22 - 233.86)\text{ kJ/kg} = \mathbf{24.7\ kW}$$

(*c*) The mass flow rate of the cooling water is determined from

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.68\text{ kg/s})(270.22 - 86.78)\text{ kJ/kg} = 124.7\text{ kW}$$

and

$$\dot{m}_{\text{cooling}} = \frac{\dot{Q}_H}{(C_p \Delta T)_{\text{water}}} = \frac{124.7\text{ kJ/s}}{(4.18\text{ kJ/kg}\cdot°\text{C})(8°\text{C})} = \mathbf{3.73\ kg/s}$$

8-187 Problem 8-186 is reconsidered. The effect of evaporator pressure on the COP and the power input as the evaporator pressure varies from 120 kPa to 380 kPa is to be investigated. The COP and the power input are to be plotted as functions of evaporator pressure.

```
"Input Data"
P[1]=120"[kPa]"
P[2] = 700"[kPa]"
Q_dot_in= 100"[kW]"
DELTAT_cw = 8"[C]"
C_P_cw = 4.18"[kJ/kg-K]"
Fluid$='R134a'
Eta_c=1.0 "Compressor isentropic efficiency"

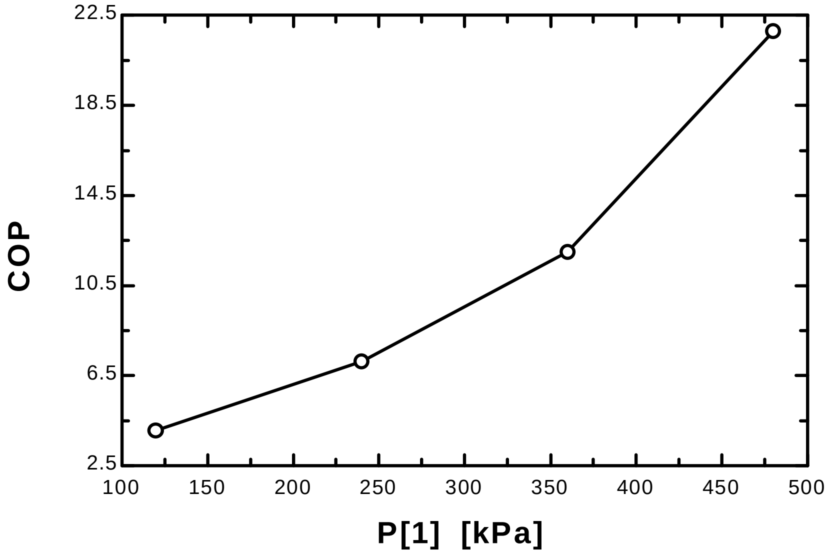
"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],x=1)  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],x=1)
T[1]=temperature(Fluid$,h=h[1],P=P[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
Wc=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+Wc=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
{h[2]=enthalpy(Fluid$,P=P[2],T=T[2]) }
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_c=m_dot*Wc

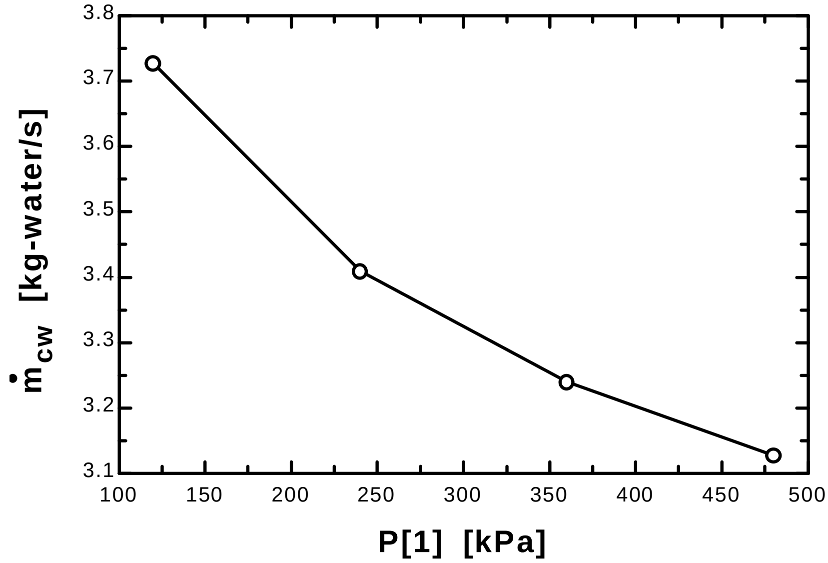
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
Q_dot_out=m_dot*Qout

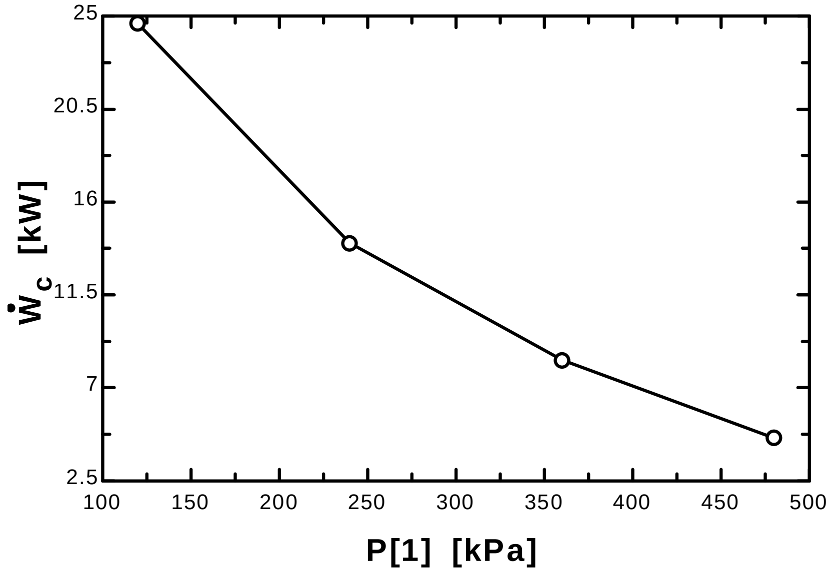
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])

"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
Q_dot_in=m_dot*Q_in
COP=Q_dot_in/W_dot_c "definition of COP"
COP_plot = COP
W_dot_in = W_dot_c
m_dot_cw*C_P_cw*DELTAT_cw = Q_dot_out
```

COP	**m [kg/s]**	**m_{cw} [kg/s]**	**W_c [kW]**	**P_1 [kPa]**
4.059	0.6766	3.727	24.63	120
7.15	0.6338	3.409	13.99	240
11.99	0.6094	3.24	8.339	360
21.79	0.5926	3.128	4.59	480


22.5
18.5
14.5
10.5
6.5
2.5
COP
100
150
200
250
300
350
400
450
500
P[1] [kPa]

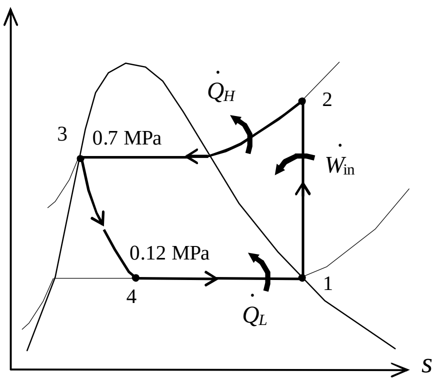

3.8
3.7
3.6
3.5
3.4
3.3
3.2
3.1
ṁcw [kg-water/s]
100
150
200
250
300
350
400
450
500
P[1] [kPa]


25
20.5
16
11.5
7
2.5
Ẇc [kW]
100
150
200
250
300
350
400
450
500
P[1] [kPa]

8-188 A large refrigeration plant operates on the vapor-compression cycle with refrigerant-134a as the working fluid. The mass flow rate of the refrigerant, the power input to the compressor, the mass flow rate of the cooling water, and the rate of exergy destruction associated with the compression process are to be determined.

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

Analysis (*a*) The refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),



$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 120\text{ kPa}} = 233.86\text{ kJ/kg} \\ s_1 = s_{g\ @\ 120\text{ kPa}} = 0.9354\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 270.22\text{ kJ/kg} \quad (T_{2s} = 34.6°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\ @\ 0.7\text{ MPa}} = 86.78\text{ kJ/kg}$$

$$h_4 \cong h_3 = 86.78\text{ kJ/kg} \quad (\text{throttling})$$

The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{100\text{ kJ/s}}{(233.86 - 86.78)\text{ kJ/kg}} = \mathbf{0.68\ kg/s}$$

(*b*) The actual enthalpy at the compressor exit is

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + (h_{2s} - h_1)/\eta_C = 233.86 + (270.22 - 233.86)/(0.75)$$
$$= 282.34\text{ kJ/kg}$$

Thus,

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.68\text{ kg/s})(282.34 - 233.86)\text{ kJ/kg} = \mathbf{33.0\ kW}$$

(*c*) The mass flow rate of the cooling water is determined from

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.68\text{ kg/s})(282.34 - 86.78)\text{ kJ/kg} = 133.0\text{ kW}$$

and

$$\dot{m}_{\text{cooling}} = \frac{\dot{Q}_H}{(C_p \Delta T)_{\text{water}}} = \frac{133.0\text{ kJ/s}}{(4.18\text{ kJ/kg}\cdot°\text{C})(8°\text{C})} = \mathbf{3.98\ \ kg/s}$$

The exergy destruction associated with this adiabatic compression process is determined from

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \dot{m}(s_2 - s_1)$$

where

$$\left.\begin{array}{l} P_2 = 0.7\text{ MPa} \\ h_2 = 282.34\text{ kJ/kg} \end{array}\right\} s_2 = 0.9741\text{ kJ/kg}\cdot\text{K}$$

Thus, $\dot{X}_{\text{destroyed}} = (298\text{ K})(0.68\text{ kg/s})(0.9741 - 0.9354)\text{ kJ/kg}\cdot\text{K} = \mathbf{7.84\ kW}$

8-189 A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid is used to heat a house. The rate of heat supply to the house, the volume flow rate of the refrigerant at the compressor inlet, and the COP of this heat pump are to be determined.

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

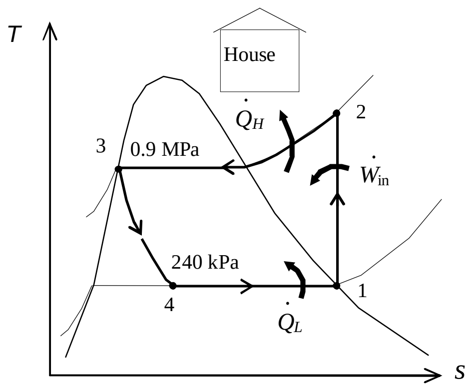
Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 240\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 240\text{ kPa}} = 244.09\text{ kJ/kg} \\ s_1 = s_{g\ @\ 240\text{ kPa}} = 0.9222\text{ kJ/kg}\cdot\text{K} \\ v_1 = v_{g\ @\ 240\text{ kPa}} = 0.0834\text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.9\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 271.41\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.9\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\ @\ 0.9\text{ MPa}} = 99.56\text{ kJ/kg}$$

$$h_4 \cong h_3 = 99.56\text{ kJ/kg}\ \ (\text{throttling})$$



The rate of heat supply to the house is determined from

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.24\text{ kg/s})(271.41 - 99.56)\text{ kJ/kg} = \mathbf{41.24\ kW}$$

(*b*) The volume flow rate of the refrigerant at the compressor inlet is

$$\dot{V}_1 = \dot{m}\text{v}_1 = (0.24\text{ kg/s})(0.0834\text{ m}^3\text{/kg}) = \mathbf{0.02002\ m^3/s}$$

(*c*) The COP of t his heat pump is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_{in}} = \frac{h_2 - h_3}{h_2 - h_1} = \frac{271.41 - 99.56}{271.41 - 244.09} = \mathbf{6.29}$$

8-190 A house is cooled adequately by a 3.5 ton air-conditioning unit. The rate of heat gain of the house when the air-conditioner is running continuously is to be determined.

Assumptions **1** The heat gain includes heat transfer through the walls and the roof, infiltration heat gain, solar heat gain, internal heat gain, etc. **2** Steady operating conditions exist.

Analysis Noting that 1 ton of refrigeration is equivalent to a cooling rate of 211 kJ/min, the rate of heat gain of the house in steady operation is simply equal to the cooling rate of the air-conditioning system,

$$\dot{Q}_{\text{heat gain}} = \dot{Q}_{\text{cooling}} = (3.5 \text{ ton})(211 \text{ kJ/min}) = 738.5 \text{ kJ/min} = \mathbf{44{,}310 \text{ kJ/h}}$$

8-191 A room is cooled adequately by a 5000 Btu/h window air-conditioning unit. The rate of heat gain of the room when the air-conditioner is running continuously is to be determined.

Assumptions **1** The heat gain includes heat transfer through the walls and the roof, infiltration heat gain, solar heat gain, internal heat gain, etc. **2** Steady operating conditions exist.

Analysis The rate of heat gain of the room in steady operation is simply equal to the cooling rate of the air-conditioning system,

$$\dot{Q}_{\text{heat gain}} = \dot{Q}_{\text{cooling}} = \mathbf{5{,}000 \text{ Btu/h}}$$

8-192 A heat pump water heater has a COP of 2.2 and consumes 2 kW when running. It is to be determined if this heat pump can be used to meet the cooling needs of a room by absorbing heat from it.

Assumptions The COP of the heat pump remains constant whether heat is absorbed from the outdoor air or room air.

Analysis The COP of the heat pump is given to be 2.2. Then the COP of the air-conditioning system becomes

$$\text{COP}_{\text{air-cond}} = \text{COP}_{\text{heat pump}} - 1 = 2.2 - 1 = 1.2$$

Then the rate of cooling (heat absorption from the air) becomes

$$\dot{Q}_{\text{cooling}} = \text{COP}_{\text{air-cond}} \dot{W}_{in} = (1.2)(2 \text{ kW}) = 2.4 \text{ kW} = 8640 \text{ kJ/h}$$

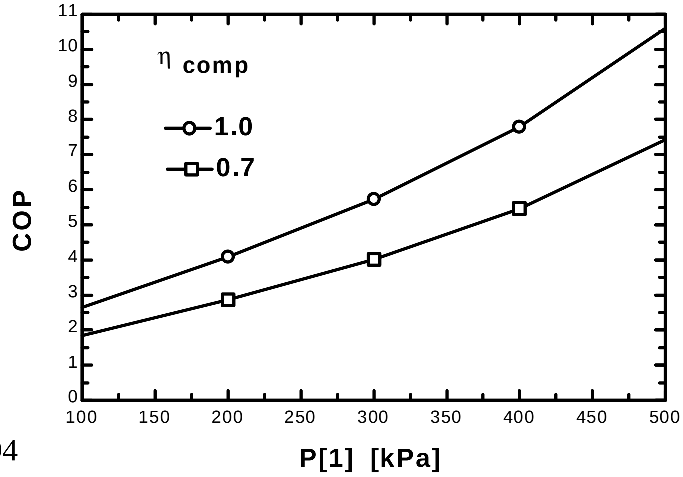
since 1 kW = 3600 kJ/h. We conclude that this heat pump **can meet** the cooling needs of the room since its cooling rate is greater than the rate of heat gain of the room.

8-193 Using EES (or other) software, the effect of the evaporator pressure on the COP of an ideal vapor-compression refrigeration cycle with R-134a as the working fluid is to be investigated. The condenser pressure is kept constant at 1 MPa while the evaporator pressure is varied from 100 kPa to 500 kPa. The COP of the refrigeration cycle is to plotted against the evaporator pressure.

```
"Input Data"
P[1]=100"[kPa]"
P[2] = 1000"[kPa]"
Fluid$='R134a'
Eta_c=1.0 "Compressor isentropic efficiency"

"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],x=1)  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],x=1)
T[1]=temperature(Fluid$,h=h[1],P=P[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
W_c=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+W_c=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])
"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
"Coefficient of Performance:"
COP=Q_in/W_c "definition of COP"
```

COP	η_c	P_1 [kPa]
1.853	0.7	100
2.864	0.7	200
4.013	0.7	300
5.458	0.7	400
7.417	0.7	500

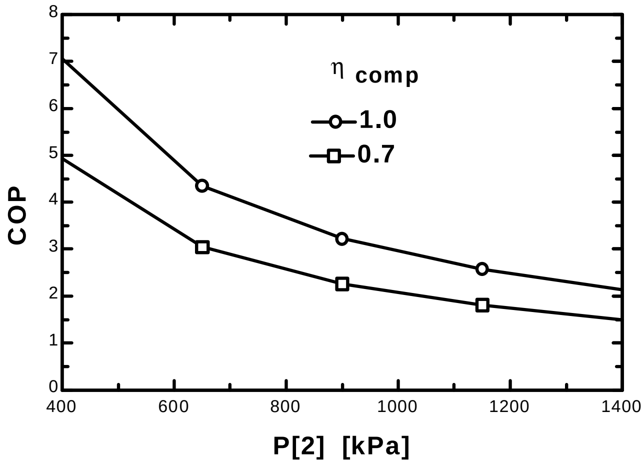


8-194 Using EES (or other) software, the effect of the condenser pressure on the COP of an ideal vapor-compression refrigeration cycle with R-134a as the working fluid is to be investigated. The evaporator pressure is kept constant at 120 kPa while the condenser pressure is varied from 400 kPa to 1400 kPa. The COP of the refrigeration cycle is to be plotted against the condenser pressure.

```
"Input Data"
P[1]=120"[kPa]"
P[2] = 400"[kPa]"
Fluid$='R134a'
Eta_c=1.0"Compressor isentropic efficiency"

"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],x=1)  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],x=1)
T[1]=temperature(Fluid$,h=h[1],P=P[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
W_c=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+W_c=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])
"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
"Coefficient of Performance:"
COP=Q_in/W_c "definition of COP"
```

COP	η_c	P_2 [kPa]
4.938	0.7	400
3.043	0.7	650
2.26	0.7	900
1.804	0.7	1150
1.494	0.7	1400



8-195 ⋯ 8-215 Design and Essay Problems

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