Solutions Manual for

Thermodynamics: An Engineering Approach

Seventh Edition

Yunus A. Cengel, Michael A. Boles

McGraw-Hill, 2011

Chapter 9

GAS POWER CYCLES

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

9-1C It is less than the thermal efficiency of a Carnot cycle.

9-2C It represents the net work on both diagrams.

9-3C 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.

9-4C 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.

9-5C 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.

9-6C It is the ratio of the maximum to minimum volumes in the cylinder.

9-7C 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.

9-8C Yes.

9-9C 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.

9-10C 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.

9-11C 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.

9-12E The maximum possible thermal efficiency of a gas power cycle with specified reservoirs is to be determined.

Analysis The maximum efficiency this cycle can have is

$$\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(40+460)\text{ R}}{(940+460)\text{ R}} = \mathbf{0.643}$$

9-13 An air-standard cycle executed in a piston-cylinder system is composed of three specified processes. The cycle is to be sketcehed on the *P*-v and *T*-*s* diagrams and the back work ratio 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.

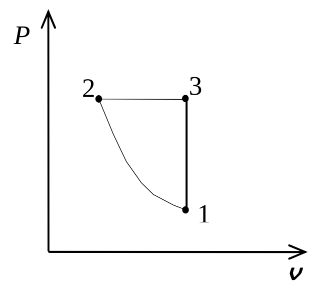
Properties The properties of air are given as R = 0.287 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4.

Analysis (a) The *P*-v and *T*-*s* diagrams of the cycle are shown in the figures.

(b) Process 1-2: Isentropic compression

$$w_{1-2,in} = mc_v(T_2 - T_1)$$

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1}$$

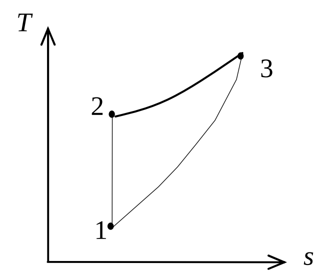


Process 2-3: Constant pressure heat addition

$$w_{2-3,out} = \int_2^3 P\,dv = P_2(v_3 - v_2) = mR(T_3 - T_2)$$

The back work ratio is

$$r_{bw} = \frac{w_{1-2,in}}{w_{2-3,out}} = \frac{mc_v(T_2 - T_1)}{mR(T_3 - T_2)}$$



Noting that

$$R = c_p - c_v \text{ and } k = \frac{c_p}{c_v} \text{ and thus, } c_v = \frac{R}{k-1}$$

From ideal gas relation,

$$\frac{T_3}{T_2} = \frac{v_3}{v_2} = \frac{v_1}{v_2} = r$$

Substituting these into back work relation,

$$r_{bw} = \frac{R}{k-1}\frac{1}{R}\frac{T_2}{T_2}\frac{(1 - T_1/T_2)}{(T_3/T_2 - 1)}$$

$$= \frac{1}{k-1}\frac{\left(1 - \dfrac{1}{r^{k-1}}\right)}{r-1} = \frac{1}{k-1}\frac{\left(1 - r^{1-k}\right)}{r-1}$$

$$= \frac{1}{1.4-1}\frac{\left(1 - 6^{-0.4}\right)}{6-1}$$

$$= \mathbf{0.256}$$

9-14 The three processes of an air-standard cycle are described. The cycle is to be shown on the *P*-v and *T*-*s* diagrams, and the back work ratio 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 are given as R = 0.287 kJ/kg.K, c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, and k = 1.4.

Analysis (a) The *P*-v and *T*-*s* diagrams of the cycle are shown in the figures.

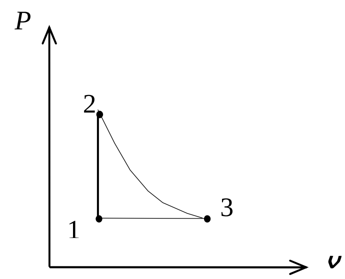
(b) The temperature at state 2 is

$$T_2 = T_1 \frac{P_2}{P_1} = (300\text{ K})\frac{700\text{ kPa}}{100\text{ kPa}} = 2100\text{ K}$$

$$T_3 = T_2 = 2100\text{ K}$$

During process 1-3, we have

$$w_{3-1,in} = -\int_3^1 P d\boldsymbol{v} = -P_1(\boldsymbol{v}_1 - \boldsymbol{v}_3) = -R(T_1 - T_3)$$
$$= -(0.287\text{ kJ/kg}\cdot\text{K})(300-2100)\text{K} = 516.6\text{ kJ/kg}$$

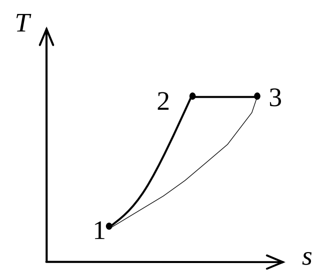


During process 2-3, we have

$$w_{2-3,out} = \int_2^3 P d\boldsymbol{v} = \int_2^3 \frac{RT}{\boldsymbol{v}} d\boldsymbol{v} = RT\ln\frac{\boldsymbol{v}_3}{\boldsymbol{v}_2} = RT\ln\frac{7\boldsymbol{v}_2}{\boldsymbol{v}_2} = RT\ln 7$$
$$= (0.287\text{ kJ/kg}\cdot\text{K})(2100)\text{K}\ln 7 = 1172.8\text{ kJ/kg}$$

The back work ratio is then

$$r_{bw} = \frac{w_{3-1,in}}{w_{2-3,out}} = \frac{516.6\text{ kJ/kg}}{1172.8\text{ kJ/kg}} = \mathbf{0.440}$$



Heat input is determined from an energy balance on the cycle during process 1-3,

$$q_{1-3,in} - w_{2-3,out} = \Delta u_{1-3}$$
$$q_{1-3,in} - = \Delta u_{1-3} + w_{2-3,out}$$
$$= c_v(T_3 - T_1) + w_{2-3,out}$$
$$= (0.718\text{ kJ/kg}\cdot\text{K})(2100-300) + 1172.8\text{ kJ/kg}$$
$$= 2465\text{ kJ/kg}$$

The net work output is

$$w_{net} = w_{2-3,out} - w_{3-1,in} = 1172.8 - 516.6 = 656.2\text{ kJ/kg}$$

(c) The thermal efficiency is then

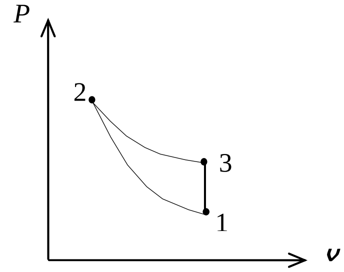
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{656.2\text{ kJ}}{2465\text{ kJ}} = 0.266 = \mathbf{26.6\%}$$

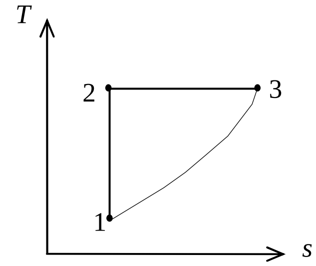
9-15 The three processes of an ideal gas power cycle are described. The cycle is to be shown on the *P*-*v* and *T*-*s* diagrams, and the maximum temperature, expansion and compression works, and thermal efficiency are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** The ideal gas has constant specific heats.

Properties The properties of ideal gas are given as R = 0.3 kJ/kg.K, c_p = 0.9 kJ/kg.K, c_v = 0.6 kJ/kg·K, and k = 1.5.

Analysis (a) The *P*-*v* and *T*-*s* diagrams of the cycle are shown in the figures.





(b) The maximum temperature is determined from

$$T_{\max} = T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (27+273\text{ K})(6)^{1.5-1} = \mathbf{734.8\ K}$$

(c) An energy balance during process 2-3 gives

$$q_{2-3,in} - w_{2-3,out} = \Delta u_{2-3} = c_v(T_3 - T_2) = 0 \quad \text{since} \quad T_3 = T_2$$

$$q_{2-3,in} = w_{2-3,out}$$

Then, the work of compression is

$$q_{2-3,in} = w_{2-3,out} = \int_2^3 P\,dv = \int_2^3 \frac{RT}{v}\,dv = RT_2 \ln\frac{v_3}{v_2} = RT_2 \ln r$$

$$= (0.3\text{ kJ/kg}\cdot\text{K})(734.8\text{ K})\ln 6 = \mathbf{395.0\ kJ/kg}$$

(d) The work during isentropic compression is determined from an energy balance during process 1-2:

$$w_{1-2,in} = \Delta u_{1-2} = c_v(T_2 - T_1)$$

$$= (0.6\text{ kJ/kg}\cdot\text{K})(734.8 - 300)$$

$$= \mathbf{260.9\ kJ/kg}$$

(e) Net work output is

$$w_{net} = w_{2-3,out} - w_{1-2,in} = 395.0 - 260.9 = 134.1\text{ kJ/kg}$$

The thermal efficiency is then

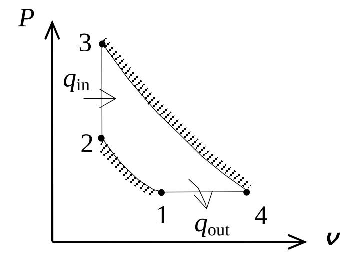
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{134.1\text{ kJ}}{395.0\text{ kJ}} = 0.339 = \mathbf{33.9\%}$$

9-16 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 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-17.

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



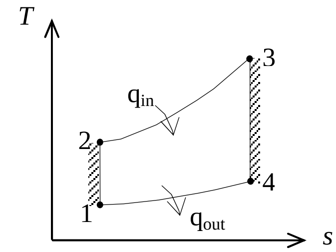
$$T_1 = 295\text{ K} \longrightarrow \begin{array}{l} h_1 = 295.17\text{ kJ/kg} \\ P_{r_1} = 1.3068 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{600\text{ kPa}}{100\text{ kPa}}(1.3068) = 7.841 \longrightarrow \begin{array}{l} u_2 = 352.29\text{ kJ/kg} \\ T_2 = 490.3\text{ K} \end{array}$$

$$T_3 = 1500\text{ K} \longrightarrow \begin{array}{l} u_3 = 1205.41\text{ kJ/kg} \\ P_{r_3} = 601.9 \end{array}$$

$$\frac{P_3 \boldsymbol{v}_3}{T_3} = \frac{P_2 \boldsymbol{v}_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2} P_2 = \frac{1500\text{ K}}{490.3\text{ K}}(600\text{ kPa}) = 1835.6\text{ kPa}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \frac{100\text{ kPa}}{1835.6\text{ kPa}}(601.9) = 32.79 \longrightarrow h_4 = 739.71\text{ kJ/kg}$$



From energy balances,

$$q_{\text{in}} = u_3 - u_2 = 1205.41 - 352.29 = 853.1\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 739.71 - 295.17 = 444.5\text{ kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 853.1 - 444.5 = \mathbf{408.6\text{ kJ/kg}}$$

(*c*) Then the thermal efficiency becomes

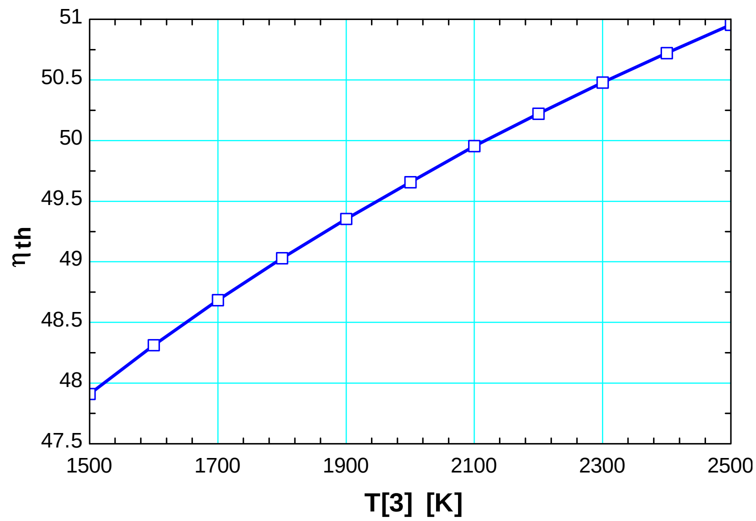
$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{408.6\text{ kJ/kg}}{853.1\text{ kJ/kg}} = 0.479 = \mathbf{47.9\%}$$

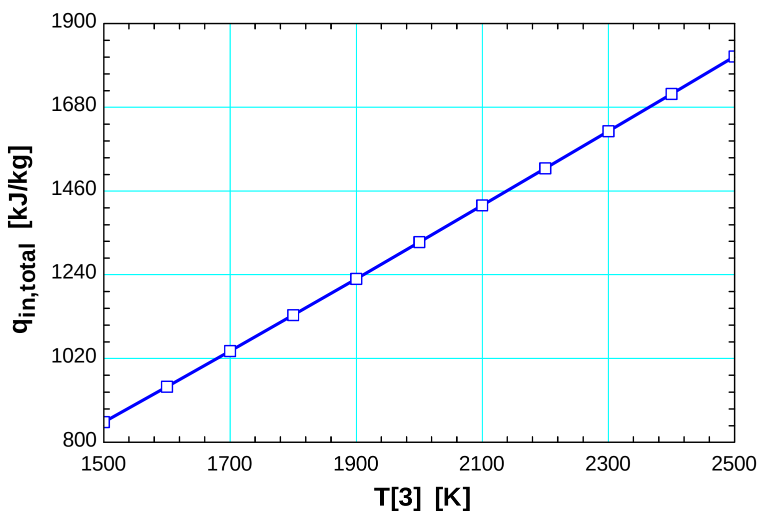
9-17 Problem 9-16 is reconsidered. The effect of the maximum temperature of the cycle on the net work output and thermal efficiency is to be investigated. Also, *T*-*s* and *P*-v diagrams for the cycle are to be plotted.

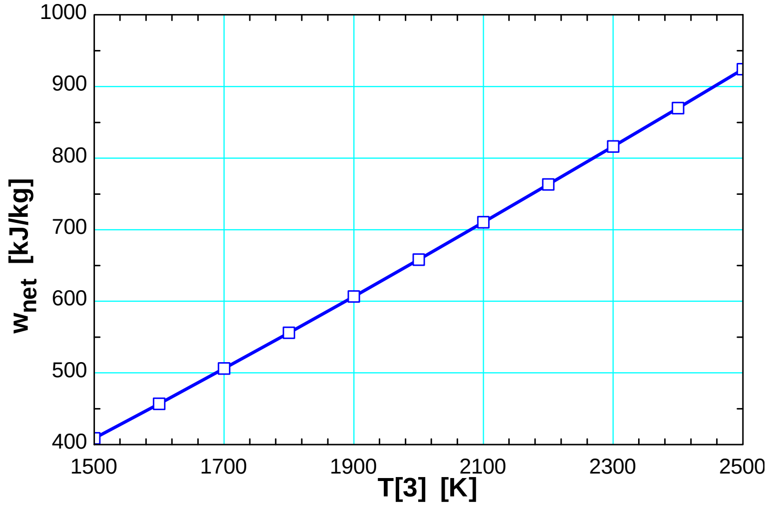
Analysis Using EES, the problem is solved as follows:

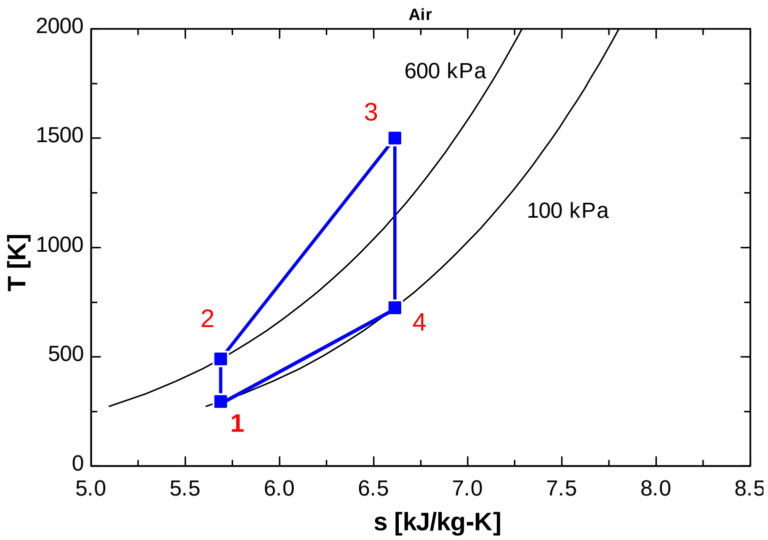
```
"Input Data"
T[1]=295 [K]
P[1]=100 [kPa]
P[2] = 600 [kPa]
T[3]=1500 [K]
P[4] = 100 [kPa]

"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]=R*T[1]
R=0.287 [kJ/kg-K]
"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]=R*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"
```

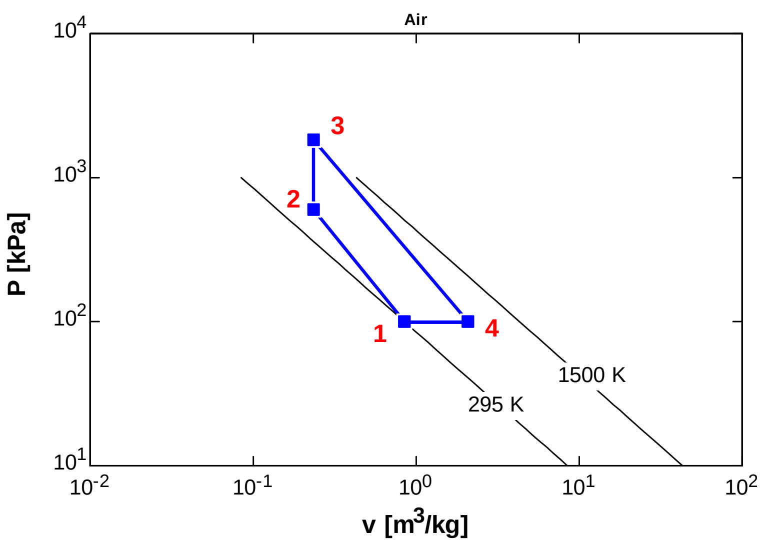
T_3 [K]	η_{th}	$q_{in,total}$ [kJ/kg]	W_{net} [kJ/kg]
1500	47.91	852.9	408.6
1600	48.31	945.7	456.9
1700	48.68	1040	506.1
1800	49.03	1134	556
1900	49.35	1229	606.7
2000	49.66	1325	658.1
2100	49.95	1422	710.5
2200	50.22	1519	763
2300	50.48	1617	816.1
2400	50.72	1715	869.8
2500	50.95	1813	924






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


Air
2000
1500
1000
500
0
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5
s [kJ/kg-K]
T [K]
600 kPa
100 kPa
1
2
3
4

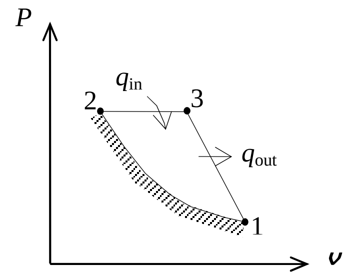

Air
10^4
10^3
10^2
10^1
10^-2
10^-1
10^0
10^1
10^2
v [m^3/kg]
P [kPa]
1500 K
295 K
1
2
3
4

9-18 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*) The temperature at state 2 and the heat input are

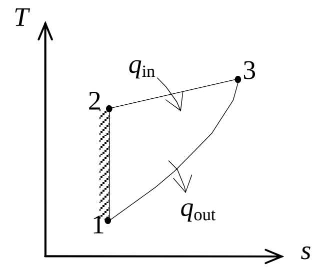


$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{1000\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} = 579.2\text{ K}$$

$$Q_{\text{in}} = m(h_3 - h_2) = mc_p(T_3 - T_2)$$

$$2.76\text{ kJ} = (0.004\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(T_3 - 579.2) \longrightarrow T_3 = 1266\text{ K}$$

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



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

$$= \left(\frac{1000 + 100\text{ kPa}}{2}\right)\left(\frac{300\text{ K}}{100\text{ kPa}} - \frac{1266\text{ K}}{1000\text{ kPa}}\right)(0.287\text{ kJ/kg}\cdot\text{K})$$

$$= 273.7\text{ kJ/kg}$$

Energy balance for process 3-1 gives

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} \longrightarrow -Q_{31,\text{out}} - W_{31,\text{out}} = m(u_1 - u_3)$$

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

$$= -(0.004\text{ kg})\left[273.7 + (0.718\text{ kJ/kg}\cdot\text{K})(300 - 1266)\text{K}\right] = \mathbf{1.679\text{ kJ}}$$

(*c*) The thermal efficiency is then

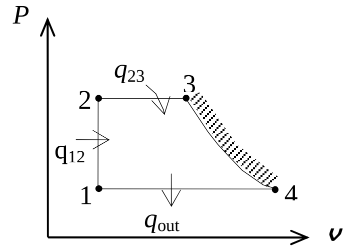
$$\eta_{\text{th}} = 1 - \frac{Q_{\text{out}}}{Q_{\text{in}}} = 1 - \frac{1.679\text{ kJ}}{2.76\text{ kJ}} = \mathbf{39.2\%}$$

9-19E 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-17E.

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



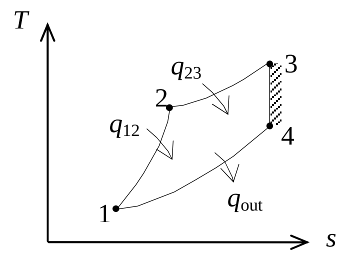
$$T_1 = 540 \text{ R} \longrightarrow u_1 = 92.04 \text{ Btu/lbm}, \quad h_1 = 129.06 \text{ Btu/lbm}$$

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

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

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

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



From energy balance,

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

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

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

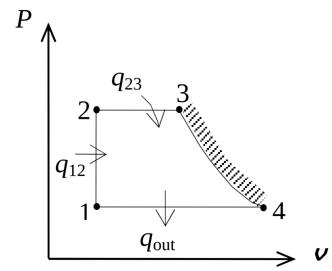
(*c*) Then the thermal efficiency becomes

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

9-20E 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*) The temperature at state 2 and the heat input are

$$q_{\text{in},12} = u_2 - u_1 = c_v(T_2 - T_1)$$

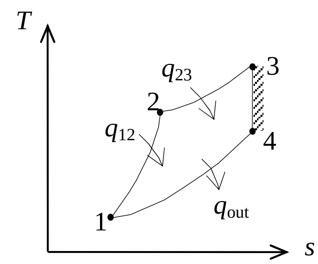
$$300\text{ Btu/lbm} = (0.171\text{ Btu/lbm.R})(T_2 - 540)\text{R}$$

$$T_2 = 2294\text{ R}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{2294\text{ R}}{540\text{ R}}(14.7\text{ psia}) = 62.46\text{ psia}$$

$$q_{\text{in},23} = h_3 - h_2 = c_P(T_3 - T_2) = (0.24\text{ Btu/lbm}\cdot\text{R})(3200 - 2294)\text{R} = 217.4\text{ Btu/lbm}$$

Process 3-4 is isentropic:



$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (3200\text{ R})\left(\frac{14.7\text{ psia}}{62.46\text{ psia}}\right)^{0.4/1.4} = 2117\text{ R}$$

$$q_{\text{in}} = q_{\text{in},12} + q_{\text{in},23} = 300 + 217.4 = \mathbf{517.4\ Btu/lbm}$$

$$q_{\text{out}} = h_4 - h_1 = c_p(T_4 - T_1) = (0.240\text{ Btu/lbm.R})(2117 - 540) = 378.5\text{ Btu/lbm}$$

(c) The thermal efficiency is then

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

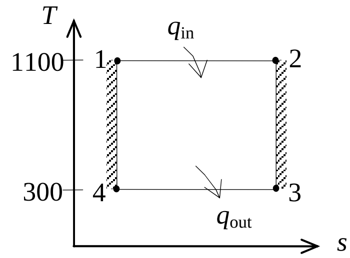
9-21 A Carnot cycle with the specified temperature limits is considered. The net work output per cycle is to be determined.

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, R = 0.287 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

$$P_2 = P_3\left(\frac{T_2}{T_3}\right)^{k/(k-1)} = (20\text{ kPa})\left(\frac{1100\text{ K}}{300\text{ K}}\right)^{1.4/0.4} = 1888\text{ kPa}$$

The heat input is determined from

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -(0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{1888\text{ kPa}}{3000\text{ kPa}} = 0.1329\text{ kJ/kg}\cdot\text{K}$$

$$Q_{\text{in}} = mT_H(s_2 - s_1) = (0.6\text{ kg})(1100\text{ K})(0.1329\text{ kJ/kg}\cdot\text{K}) = 87.73\text{ kJ}$$

Then,

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{1100\text{ K}} = 0.7273 = 72.7\%$$

$$W_{\text{net,out}} = \eta_{\text{th}} Q_{\text{in}} = (0.7273)(87.73\text{ kJ}) = \mathbf{63.8\ kJ}$$

9-22 A Carnot cycle executed in a closed system with air as the working fluid is considered. The minimum pressure in the cycle, the heat rejection from the cycle, the thermal efficiency of the cycle, and the second-law efficiency of an actual cycle operating between the same temperature limits are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

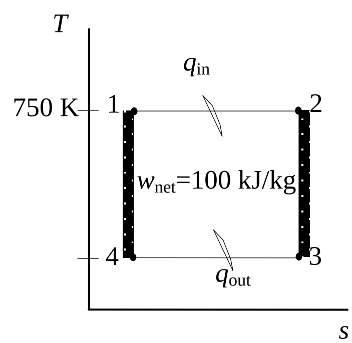
Properties The properties of air at room temperatures are R = 0.287 kJ/kg.K and k = 1.4 (Table A-2).

Analysis (a) The minimum temperature is determined from

$$w_{\text{net}} = (s_2 - s_1)(T_H - T_L) \longrightarrow 100 \text{ kJ/kg} = (0.25 \text{ kJ/kg}\cdot\text{K})(750 - T_L)\text{K} \longrightarrow T_L = 350 \text{ K}$$

The pressure at state 4 is determined from

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



or

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

$$800 \text{ kPa} = P_4\left(\frac{750 \text{ K}}{350 \text{ K}}\right)^{1.4/0.4} \longrightarrow P_4 = 110.1 \text{ kPa}$$

The minimum pressure in the cycle is determined from

$$\Delta s_{12} = -\Delta s_{34} = c_p \ln\frac{T_4}{T_3}^{\nearrow 0} - R\ln\frac{P_4}{P_3}$$

$$-0.25 \text{ kJ/kg}\cdot\text{K} = -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{110.1 \text{ kPa}}{P_3} \longrightarrow P_3 = \mathbf{46.1 \text{ kPa}}$$

(b) The heat rejection from the cycle is

$$q_{\text{out}} = T_L \Delta s_{12} = (350 \text{ K})(0.25 \text{ kJ/kg.K}) = \mathbf{87.5 \text{ kJ/kg}}$$

(c) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{350 \text{ K}}{750 \text{ K}} = \mathbf{0.533}$$

(d) The power output for the Carnot cycle is

$$\dot{W}_{\text{Carnot}} = \dot{m}w_{\text{net}} = (90 \text{ kg/s})(100 \text{ kJ/kg}) = 9000 \text{ kW}$$

Then, the second-law efficiency of the actual cycle becomes

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{actual}}}{\dot{W}_{\text{Carnot}}} = \frac{5200 \text{ kW}}{9000 \text{ kW}} = \mathbf{0.578}$$

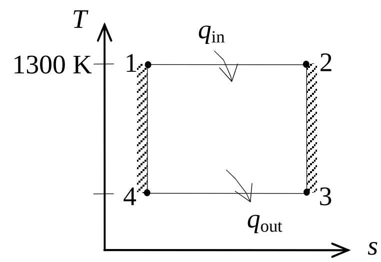
9-23 An ideal gas Carnot cycle with air as the working fluid is considered. The maximum temperature of the low-temperature energy reservoir, the cycle's thermal efficiency, and the amount of heat that must be supplied per cycle are to be determined.

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-2a).

Analysis The temperature of the low-temperature reservoir can be found by applying the isentropic expansion process relation

$$T_1 = T_2\left(\frac{v_2}{v_1}\right)^{k-1} = (1027+273\text{ K})\left(\frac{1}{12}\right)^{1.4-1} = \mathbf{481.1K}$$



Since the Carnot engine is completely reversible, its efficiency is

$$\eta_{\text{th,Carnot}} = 1-\frac{T_L}{T_H} = 1-\frac{481.1\text{ K}}{(1027+273)\text{ K}} = \mathbf{0.630}$$

The work output per cycle is

$$W_{\text{net}} = \frac{\dot{W}_{\text{net}}}{\dot{n}} = \frac{500\text{ kJ/s}}{1500\text{ cycle/min}}\left(\frac{60\text{ s}}{1\text{ min}}\right) = 20\text{ kJ/cycle}$$

According to the definition of the cycle efficiency,

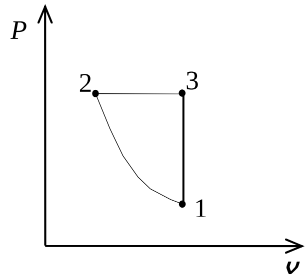
$$\eta_{\text{th,Carnot}} = \frac{W_{\text{net}}}{Q_{\text{in}}} \longrightarrow Q_{\text{in}} = \frac{W_{\text{net}}}{\eta_{\text{th,Carnot}}} = \frac{20\text{ kJ/cycle}}{0.63} = \mathbf{31.75\ kJ/cycle}$$

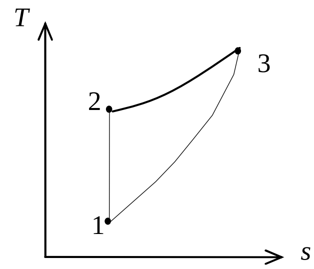
9-24 An air-standard cycle executed in a piston-cylinder system is composed of three specified processes. The cycle is to be sketcehed on the *P*-v and *T*-*s* diagrams; the heat and work interactions and the thermal efficiency of the cycle are to be determined; and an expression for thermal efficiency as functions of compression ratio and specific heat ratio is to be obtained.

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 are given as R = 0.3 kJ/kg·K and c_v = 0.3 kJ/kg·K.

Analysis (a) The *P*-v and *T*-*s* diagrams of the cycle are shown in the figures.





(b) Noting that

$$c_p = c_v + R = 0.7 + 0.3 = 1.0 \text{ kJ/kg}\cdot\text{K}$$

$$k = \frac{c_p}{c_v} = \frac{1.0}{0.7} = 1.429$$

Process 1-2: Isentropic compression

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (293\text{ K})(5)^{0.429} = 584.4\text{ K}$$

$$w_{1-2,\text{in}} = c_v(T_2 - T_1) = (0.7\text{ kJ/kg}\cdot\text{K})(584.4 - 293)\text{ K} = \mathbf{204.0\ kJ/kg}$$

$$q_{1-2} = \mathbf{0}$$

From ideal gas relation,

$$\frac{T_3}{T_2} = \frac{v_3}{v_2} = \frac{v_1}{v_2} = r \longrightarrow T_3 = (584.4)(5) = 2922$$

Process 2-3: Constant pressure heat addition

$$w_{2-3,\text{out}} = \int_2^3 P dv = P_2(v_3 - v_2) = R(T_3 - T_2)$$
$$= (0.3\text{ kJ/kg}\cdot\text{K})(2922 - 584.4)\text{ K} = \mathbf{701.3\ kJ/kg}$$

$$q_{2-3,\text{in}} = w_{2-3,out} + \Delta u_{2-3} = \Delta h_{2-3}$$
$$= c_p(T_3 - T_2) = (1\text{ kJ/kg}\cdot\text{K})(2922 - 584.4)\text{ K} = \mathbf{2338\ kJ/kg}$$

Process 3-1: Constant volume heat rejection

$$q_{3-1,\text{out}} = \Delta u_{1-3} = c_v(T_3 - T_1) = (0.7\text{ kJ/kg}\cdot\text{K})(2922 - 293)\text{ K} = \mathbf{1840.3\ kJ/kg}$$

$$w_{3-1} = \mathbf{0}$$

(c) Net work is

$$w_{\text{net}} = w_{2-3,\text{out}} - w_{1-2,\text{in}} = 701.3 - 204.0 = 497.3\text{ kJ/kg}\cdot\text{K}$$

The thermal efficiency is then

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{497.3\text{ kJ}}{2338\text{ kJ}} = 0.213 = \mathbf{21.3\%}$$

(d) The expression for the cycle thermal efficiency is obtained as follows:

$$
\begin{aligned}
\eta_{\text{th}} &= \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{w_{2-3,\text{out}} - w_{1-2,\text{in}}}{q_{\text{in}}} \\
&= \frac{R(T_3 - T_2) - c_v(T_2 - T_1)}{c_p(T_3 - T_2)} \\
&= \frac{R}{c_p} - \frac{c_v(T_1 r^{k-1} - T_1)}{c_p(rT_1 r^{k-1} - T_1 r^{k-1})} \\
&= \frac{R}{c_p} - \frac{c_v T_1 r^{k-1}\left(1 - \dfrac{T_1}{T_1 r^{k-1}}\right)}{c_p T_1 r^{k-1}(r-1)} \\
&= \frac{R}{c_p} - \frac{1}{k(r-1)}\left(1 - \frac{T_1}{T_1 r^{k-1}}\right) \\
&= \frac{R}{c_p} - \frac{1}{k(r-1)}\left(1 - \frac{1}{r^{k-1}}\right) \\
&= \left(1 - \frac{1}{k}\right) - \frac{1}{k(r-1)}\left(1 - \frac{1}{r^{k-1}}\right)
\end{aligned}
$$

since

$$
\frac{R}{c_p} = \frac{c_p - c_v}{c_p} = 1 - \frac{c_v}{c_p} = 1 - \frac{1}{k}
$$

Otto Cycle

9-25C 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.

9-26C The ideal Otto cycle involves external irreversibilities, and thus it has a lower thermal efficiency.

9-27C 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.

9-28C They are analyzed as closed system processes because no mass crosses the system boundaries during any of the processes.

9-29C It increases with both of them.

9-30C Because high compression ratios cause engine knock.

9-31C The thermal efficiency will be the highest for argon because it has the highest specific heat ratio, $k = 1.667$.

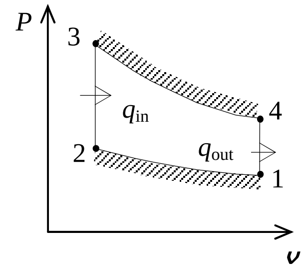
9-32C The fuel is injected into the cylinder in both engines, but it is ignited with a spark plug in gasoline engines.

9-33 An ideal Otto cycle is considered. The thermal efficiency and the rate of heat input 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-2a).

Analysis The definition of cycle thermal efficiency reduces to



$$\eta_{th} = 1 - \frac{1}{r^{k-1}} = 1 - \frac{1}{10.5^{1.4-1}} = 0.6096 = \mathbf{61.0\%}$$

The rate of heat addition is then

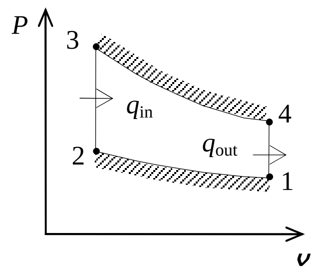
$$\dot{Q}_{in} = \frac{\dot{W}_{net}}{\eta_{th}} = \frac{90 \text{ kW}}{0.6096} = \mathbf{148 \text{ kW}}$$

9-34 An ideal Otto cycle is considered. The thermal efficiency and the rate of heat input 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-2a).

Analysis The definition of cycle thermal efficiency reduces to



$$\eta_{th} = 1 - \frac{1}{r^{k-1}} = 1 - \frac{1}{8.5^{1.4-1}} = 0.5752 = \mathbf{57.5\%}$$

The rate of heat addition is then

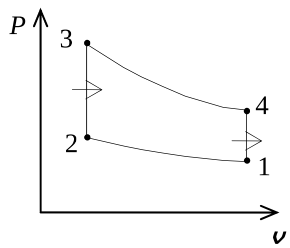
$$\dot{Q}_{in} = \frac{\dot{W}_{net}}{\eta_{th}} = \frac{90 \text{ kW}}{0.5752} = \mathbf{157 \text{ kW}}$$

9-35 The two isentropic processes in an Otto cycle are replaced with polytropic processes. The heat added to and rejected from this cycle, and the cycle's 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 $R = 0.287$ kPa·m^3/kg·K, $c_p = 1.005$ kJ/kg·K, $c_v = 0.718$ kJ/kg·K, and $k = 1.4$ (Table A-2a).

Analysis The temperature at the end of the compression is



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = T_1 r^{n-1} = (288\text{ K})(8)^{1.3-1} = 537.4\text{ K}$$

And the temperature at the end of the expansion is

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = T_3\left(\frac{1}{r}\right)^{n-1} = (1473\text{ K})\left(\frac{1}{8}\right)^{1.3-1} = 789.4\text{ K}$$

The integral of the work expression for the polytropic compression gives

$$w_{1-2} = \frac{RT_1}{n-1}\left[\left(\frac{v_1}{v_2}\right)^{n-1} - 1\right] = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(288\text{ K})}{1.3-1}(8^{1.3-1} - 1) = 238.6\text{ kJ/kg}$$

Similarly, the work produced during the expansion is

$$w_{3-4} = -\frac{RT_3}{n-1}\left[\left(\frac{v_3}{v_4}\right)^{n-1} - 1\right] = -\frac{(0.287\text{ kJ/kg}\cdot\text{K})(1473\text{ K})}{1.3-1}\left[\left(\frac{1}{8}\right)^{1.3-1} - 1\right] = 654.0\text{ kJ/kg}$$

Application of the first law to each of the four processes gives

$$q_{1-2} = w_{1-2} - c_v(T_2 - T_1) = 238.6\text{ kJ/kg} - (0.718\text{ kJ/kg}\cdot\text{K})(537.4 - 288)\text{K} = 59.53\text{ kJ/kg}$$

$$q_{2-3} = c_v(T_3 - T_2) = (0.718\text{ kJ/kg}\cdot\text{K})(1473 - 537.4)\text{K} = 671.8\text{ kJ/kg}$$

$$q_{3-4} = w_{3-4} - c_v(T_3 - T_4) = 654.0\text{ kJ/kg} - (0.718\text{ kJ/kg}\cdot\text{K})(1473 - 789.4)\text{K} = 163.2\text{ kJ/kg}$$

$$q_{4-1} = c_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(789.4 - 288)\text{K} = 360.0\text{ kJ/kg}$$

The head added and rejected from the cycle are

$$q_{in} = q_{2-3} + q_{3-4} = 671.8 + 163.2 = \mathbf{835.0\text{ kJ/kg}}$$
$$q_{out} = q_{1-2} + q_{4-1} = 59.53 + 360.0 = \mathbf{419.5\text{ kJ/kg}}$$

The thermal efficiency of this cycle is then

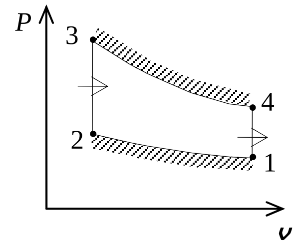
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{419.5}{835.0} = \mathbf{0.498}$$

9-36 An ideal Otto cycle is considered. The heat added to and rejected from this cycle, and the cycle's 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 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis The temperature at the end of the compression is



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (288\ \text{K})(8)^{1.4-1} = 661.7\ \text{K}$$

and the temperature at the end of the expansion is

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1} = (1473\ \text{K})\left(\frac{1}{8}\right)^{1.4-1} = 641.2\ \text{K}$$

Application of the first law to the heat addition process gives

$$q_{\text{in}} = c_v(T_3 - T_2) = (0.718\ \text{kJ/kg}\cdot\text{K})(1473 - 661.7)\text{K} = \mathbf{582.5\ kJ/kg}$$

Similarly, the heat rejected is

$$q_{\text{out}} = c_v(T_4 - T_1) = (0.718\ \text{kJ/kg}\cdot\text{K})(641.2 - 288)\text{K} = \mathbf{253.6\ kJ/kg}$$

The thermal efficiency of this cycle is then

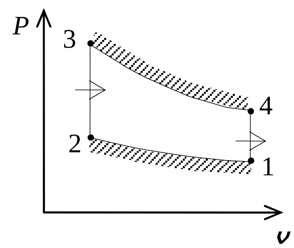
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{253.6}{582.5} = \mathbf{0.565}$$

9-37E A six-cylinder, four-stroke, spark-ignition engine operating on the ideal Otto cycle is considered. The power produced by the engine 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 R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis From the data specified in the problem statement,



$$r = \frac{v_1}{v_2} = \frac{v_1}{0.14 v_1} = 7.143$$

Since the compression and expansion processes are isentropic,

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (525\text{ R})(7.143)^{1.4-1} = 1153\text{ R}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1} = (2060\text{ R})\left(\frac{1}{7.143}\right)^{1.4-1} = 938.2\text{ R}$$

Application of the first law to the compression and expansion processes gives

$$\begin{aligned} w_{\text{net}} &= c_v(T_3 - T_4) - c_v(T_2 - T_1) \\ &= (0.171\text{ Btu/lbm}\cdot\text{R})(2060 - 938.2)\text{R} - (0.171\text{ Btu/lbm}\cdot\text{R})(1153 - 525)\text{R} \\ &= 84.44\text{ Btu/lbm} \end{aligned}$$

When each cylinder is charged with the air-fuel mixture,

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(525\text{ R})}{14\text{ psia}} = 13.89\text{ ft}^3/\text{lbm}$$

The total air mass taken by all 6 cylinders when they are charged is

$$m = N_{\text{cyl}}\frac{\Delta V}{v_1} = N_{\text{cyl}}\frac{\pi B^2 S/4}{v_1} = (6)\frac{\pi(3.5/12\text{ ft})^2(3.9/12\text{ ft})/4}{13.89\text{ ft}^3/\text{lbm}} = 0.009380\text{ lbm}$$

The net work produced per cycle is

$$W_{\text{net}} = m w_{\text{net}} = (0.009380\text{ lbm})(84.44\text{ Btu/lbm}) = 0.7920\text{ Btu/cycle}$$

The power produced is determined from

$$\dot{W}_{\text{net}} = \frac{W_{\text{net}}\dot{n}}{N_{\text{rev}}} = \frac{(0.7920\text{ Btu/cycle})(2500/60\text{ rev/s})}{2\text{ rev/cycle}}\left(\frac{1\text{ hp}}{0.7068\text{ Btu/s}}\right) = \mathbf{23.3\text{ hp}}$$

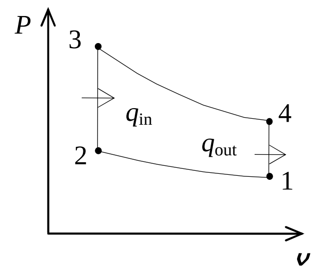
since there are two revolutions per cycle in a four-stroke engine.

9-38E An Otto cycle with non-isentropic compression and expansion processes is considered. The thermal efficiency, the heat addition, 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 R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis We begin by determining the temperatures of the cycle states using the process equations and component efficiencies. The ideal temperature at the end of the compression is then



$$T_{2s} = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (520\text{ R})(8)^{1.4-1} = 1195\text{ R}$$

With the isentropic compression efficiency, the actual temperature at the end of the compression is

$$\eta = \frac{T_{2s}-T_1}{T_2-T_1} \longrightarrow T_2 = T_1 + \frac{T_{2s}-T_1}{\eta} = (520\text{ R}) + \frac{(1195-520)\text{ R}}{0.85} = 1314\text{ R}$$

Similarly for the expansion,

$$T_{4s} = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1}{r}\right)^{k-1} = (2300+460\text{ R})\left(\frac{1}{8}\right)^{1.4-1} = 1201\text{ R}$$

$$\eta = \frac{T_3-T_4}{T_3-T_{4s}} \longrightarrow T_4 = T_3 - \eta(T_3-T_{4s}) = (2760\text{ R}) - (0.95)(2760-1201)\text{ R} = 1279\text{ R}$$

The specific heat addition is that of process 2-3,

$$q_{in} = c_v(T_3-T_2) = (0.171\text{ Btu/lbm}\cdot\text{R})(2760-1314)\text{R} = \mathbf{247.3\ Btu/lbm}$$

The net work production is the difference between the work produced by the expansion and that used by the compression,

$$\begin{aligned} w_{net} &= c_v(T_3-T_4) - c_v(T_2-T_1) \\ &= (0.171\text{ Btu/lbm}\cdot\text{R})(2760-1279)\text{R} - (0.171\text{ Btu/lbm}\cdot\text{R})(1314-520)\text{R} \\ &= 117.5\text{ Btu/lbm} \end{aligned}$$

The thermal efficiency of this cycle is then

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{117.5\text{ Btu/lbm}}{247.3\text{ Btu/lbm}} = \mathbf{0.475}$$

At the beginning of compression, the maximum specific volume of this cycle is

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520\text{ R})}{13\text{ psia}} = 14.82\text{ ft}^3/\text{lbm}$$

while the minimum specific volume of the cycle occurs at the end of the compression

$$v_2 = \frac{v_1}{r} = \frac{14.82\text{ ft}^3/\text{lbm}}{8} = 1.852\text{ ft}^3/\text{lbm}$$

The engine's mean effective pressure is then

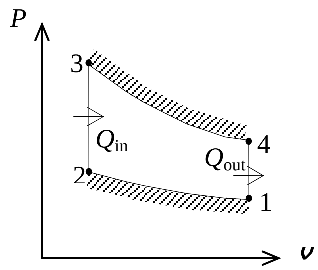
$$\text{MEP} = \frac{w_{net}}{v_1 - v_2} = \frac{117.5\text{ Btu/lbm}}{(14.82-1.852)\text{ ft}^3/\text{lbm}}\left(\frac{5.404\text{ psia}\cdot\text{ft}^3}{1\text{ Btu}}\right) = \mathbf{49.0\ psia}$$

9-39 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, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

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



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (308\text{ K})(9.5)^{0.4} = 757.9\text{ K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{757.9\text{ K}}{308\text{ K}}\right)(100\text{ kPa}) = 2338\text{ kPa}$$

Process 3-4: isentropic expansion.

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

Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1969\text{ K}}{757.9\text{ K}}\right)(2338\text{ kPa}) = \mathbf{6072\ kPa}$$

(*b*) $$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(0.0006\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})} = 6.788\times10^{-4}\text{ kg}$$

$$Q_{in} = m(u_3 - u_2) = mc_v(T_3 - T_2) = (6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(1969 - 757.9)\text{K} = \mathbf{0.590\ kJ}$$

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

$$Q_{out} = m(u_4 - u_1) = mc_v(T_4 - T_1) = -(6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(800 - 308)\text{K} = 0.240\text{ kJ}$$

$$W_{net} = Q_{in} - Q_{out} = 0.590 - 0.240 = 0.350\text{ kJ}$$

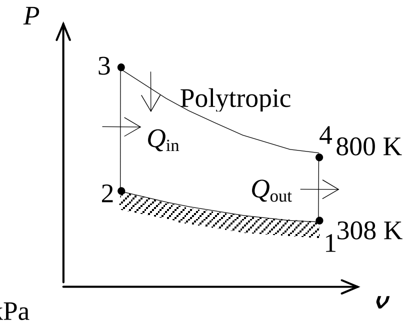
$$\eta_{th} = \frac{W_{net,out}}{Q_{in}} = \frac{0.350\text{ kJ}}{0.590\text{ kJ}} = \mathbf{59.4\%}$$

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

$$\text{MEP} = \frac{W_{net,out}}{V_1 - V_2} = \frac{W_{net,out}}{V_1(1-1/r)} = \frac{0.350\text{ kJ}}{(0.0006\text{ m}^3)(1-1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{652\ kPa}$$

9-40 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, $R = 0.287$ kJ/kg·K, and $k = 1.4$ (Table A-2).

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

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (308\text{ K})(9.5)^{0.4} = 757.9\text{ K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{757.9\text{ K}}{308\text{ K}}\right)(100\text{ kPa}) = 2338\text{ kPa}$$

Process 3-4: polytropic expansion.

$$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(0.0006\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})} = 6.788\times10^{-4}\text{ kg}$$

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

$$W_{34} = \frac{mR(T_4 - T_3)}{1-n} = \frac{(6.788\times10^{-4})(0.287\text{ kJ/kg}\cdot\text{K})(800-1759)\text{K}}{1-1.35} = 0.5338\text{ kJ}$$

Then energy balance for process 3-4 gives

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

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

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

$$Q_{34,in} = (6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(800-1759)\text{K} + 0.5338\text{ kJ} = 0.0664\text{ kJ}$$

That is, 0.066 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.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1759\text{ K}}{757.9\text{ K}}\right)(2338\text{ kPa}) = \mathbf{5426\text{ kPa}}$$

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

$$Q_{23,in} = (6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(1759-757.9)\text{K} = 0.4879\text{ kJ}$$

Therefore, $Q_{in} = Q_{23,in} + Q_{34,in} = 0.4879 + 0.0664 = \mathbf{0.5543\text{ kJ}}$

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

$$Q_{out} = m(u_4 - u_1) = mc_v(T_4 - T_1) = (6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(800-308)\text{K} = 0.2398\text{ kJ}$$

$$W_{net,out} = Q_{in} - Q_{out} = 0.5543 - 0.2398 = 0.3145\text{ kJ}$$

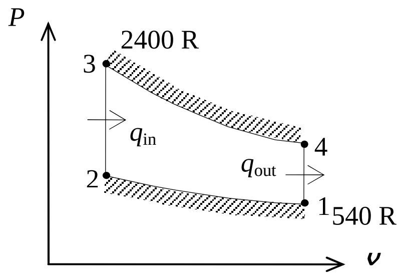
$$\eta_{th} = \frac{W_{net,out}}{Q_{in}} = \frac{0.3145\text{ kJ}}{0.5543\text{ kJ}} = \mathbf{56.7\%}$$

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

$$\text{MEP} = \frac{W_{net,out}}{V_1 - V_2} = \frac{W_{net,out}}{V_1(1-1/r)} = \frac{0.3145\text{ kJ}}{(0.0006\text{ m}^3)(1-1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{586\text{ kPa}}$$

9-41E 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-17E.

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.

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

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

(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.

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

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{113.50 \text{ Btu/lbm}}{241.42 \text{ Btu/lbm}} = \mathbf{53.0\%}$$

(c) The thermal efficiency of a Carnot cycle operating between the same temperature limits is

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

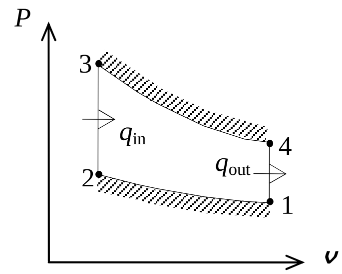
9-42E An ideal Otto cycle with argon as the working fluid has a compression ratio of 8. The amount of heat transferred to the argon during the heat addition process, the thermal efficiency, and the thermal efficiency of a Carnot cycle operating between the same temperature limits are to be determined.

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.

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

(*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*) The thermal efficiency of a Carnot cycle operating between the same temperature limits is

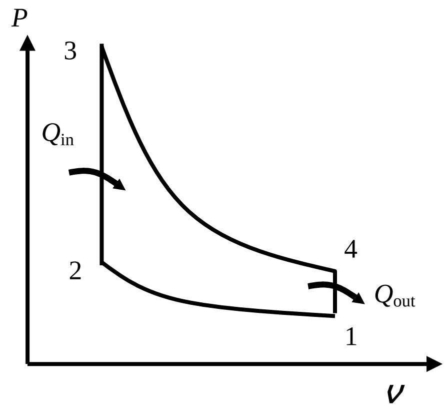
$$\eta_{th,C} = 1 - \frac{T_L}{T_H} = 1 - \frac{540\text{ R}}{2400\text{ R}} = \mathbf{77.5\%}$$

9-43 A gasoline engine operates on an Otto cycle. The compression and expansion processes are modeled as polytropic. The temperature at the end of expansion process, the net work output, the thermal efficiency, the mean effective pressure, the engine speed for a given net power, and the specific fuel consumption 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 850 K are c_p = 1.110 kJ/kg·K, c_v = 0.823 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.349 (Table A-2b).

Analysis (*a*) Process 1-2: polytropic compression



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = (310\text{ K})(11)^{1.3\text{-}1} = 636.5\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{n} = (100\text{ kPa})(11)^{1.3} = 2258\text{ kPa}$$

$$w_{12} = \frac{R(T_2 - T_1)}{1-n} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(636.5-310)\text{K}}{1-1.3} = -312.3\text{ kJ/kg}$$

Process 2-3: constant volume heat addition

$$T_3 = T_2\left(\frac{P_3}{P_2}\right) = (636.5\text{ K})\left(\frac{8000\text{ kPa}}{2258\text{ kPa}}\right) = 2255\text{ K}$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$$
$$= (0.823\text{ kJ/kg}\cdot\text{K})(2255-636.5)\text{K} = 1332\text{ kJ/kg}$$

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = (2255\text{ K})\left(\frac{1}{11}\right)^{1.3\text{-}1} = \mathbf{1098\ K}$$

$$P_4 = P_3\left(\frac{v_2}{v_1}\right)^{n} = (8000\text{ kPa})\left(\frac{1}{11}\right)^{1.3} = 354.2\text{ kPa}$$

$$w_{34} = \frac{R(T_4 - T_3)}{1-n} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(1098-2255)\text{K}}{1-1.3} = 1106\text{ kJ/kg}$$

Process 4-1: constant volume heat rejection.

(*b*) The net work output and the thermal efficiency are

$$w_{net,out} = w_{34} - w_{12} = 1106 - 312.3 = \mathbf{794\ kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{794\text{ kJ/kg}}{1332\text{ kJ/kg}} = 0.596 = \mathbf{59.6\%}$$

(*c*) The mean effective pressure is determined as follows

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K})(310\text{ K})}{100\text{ kPa}} = 0.8897\text{ m}^3\text{/kg} = v_{max}$$

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

$$\text{MEP} = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1-1/r)} = \frac{794\text{ kJ/kg}}{(0.8897\text{ m}^3\text{/kg})(1-1/11)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{982\ kPa}$$

(*d*) The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 11 = \frac{V_c + 0.0016\,\text{m}^3}{V_c} \longrightarrow V_c = 0.00016\,\text{m}^3$$

$$V_1 = V_c + V_d = 0.00016 + 0.0016 = 0.00176\,\text{m}^3$$

The total mass contained in the cylinder is

$$m_t = \frac{P_1 V_1}{RT_1} = \frac{(100\,\text{kPa})/0.00176\,\text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(310\ \text{K})} = 0.001978\,\text{kg}$$

The engine speed for a net power output of 50 kW is

$$\dot{n} = 2\frac{\dot{W}_{\text{net}}}{m_t w_{\text{net}}} = (2\ \text{rev/cycle})\frac{50\ \text{kJ/s}}{(0.001978\ \text{kg})(794\ \text{kJ/kg}\cdot\text{cycle})}\left(\frac{60\ \text{s}}{1\ \text{min}}\right) = \mathbf{3820\ rev/min}$$

Note that there are two revolutions in one cycle in four-stroke engines.

(*e*) The mass of fuel burned during one cycle is

$$\text{AF} = \frac{m_a}{m_f} = \frac{m_t - m_f}{m_f} \longrightarrow 16 = \frac{(0.001978\,\text{kg}) - m_f}{m_f} \longrightarrow m_f = 0.0001164\,\text{kg}$$

Finally, the specific fuel consumption is

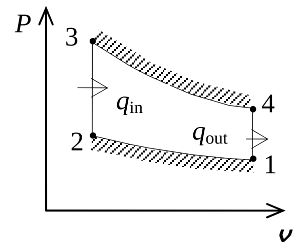
$$\text{sfc} = \frac{m_f}{m_t w_{\text{net}}} = \frac{0.0001164\,\text{kg}}{(0.001978\,\text{kg})(794\,\text{kJ/kg})}\left(\frac{1000\ \text{g}}{1\,\text{kg}}\right)\left(\frac{3600\ \text{kJ}}{1\,\text{kWh}}\right) = \mathbf{267\ g/kWh}$$

9-44 The expressions for the maximum gas temperature and pressure of an ideal Otto cycle are to be determined when the compression ratio is doubled.

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.

Analysis The temperature at the end of the compression varies with the compression ratio as

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1}$$



since T_1 is fixed. The temperature rise during the combustion remains constant since the amount of heat addition is fixed. Then, the maximum cycle temperature is given by

$$T_3 = q_{in}/c_v + T_2 = q_{in}/c_v + T_1 r^{k-1}$$

The smallest gas specific volume during the cycle is

$$v_3 = \frac{v_1}{r}$$

When this is combined with the maximum temperature, the maximum pressure is given by

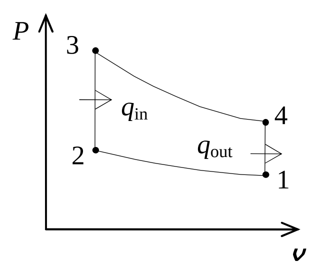
$$P_3 = \frac{RT_3}{v_3} = \frac{Rr}{v_1}\left(q_{in}/c_v + T_1 r^{k-1}\right)$$

9-45 It is to be determined if the polytropic exponent to be used in an Otto cycle model will be greater than or less than the isentropic exponent.

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.

Analysis During a polytropic process,

$$Pv^n = \text{constant}$$
$$TP^{(n-1)/n} = \text{constant}$$



and for an isentropic process,

$$Pv^k = \text{constant}$$
$$TP^{(k-1)/k} = \text{constant}$$

If heat is lost during the expansion of the gas,

$$T_4 > T_{4s}$$

where T_{4s} is the temperature that would occur if the expansion were reversible and adiabatic (n=k). This can only occur when

$$n \le k$$

Diesel Cycle

9-46C 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.

9-47C 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.

9-48C The gasoline engine.

9-49C Diesel engines operate at high compression ratios because the diesel engines do not have the engine knock problem.

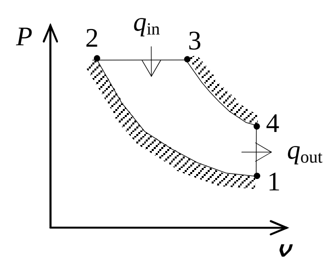
9-50C Cutoff ratio is the ratio of the cylinder volumes after and before the combustion process. As the cutoff ratio decreases, the efficiency of the diesel cycle increases.

9-51 An ideal diesel cycle has a compression ratio of 20 and a cutoff ratio of 1.3. The maximum temperature of the air and the rate of heat addition 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, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis We begin by using the process types to fix the temperatures of the states.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (288\text{ K})(20)^{1.4-1} = 954.6\text{ K}$$

$$T_3 = T_2\left(\frac{v_3}{v_2}\right) = T_2 r_c = (954.6\text{ K})(1.3) = \mathbf{1241\ K}$$

Combining the first law as applied to the various processes with the process equations gives

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\frac{r_c^k - 1}{k(r_c - 1)} = 1 - \frac{1}{20^{1.4-1}}\frac{1.3^{1.4} - 1}{1.4(1.3-1)} = 0.6812$$

According to the definition of the thermal efficiency,

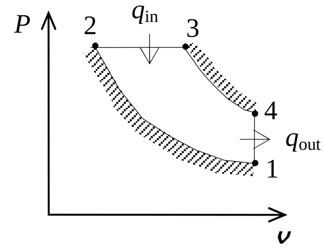
$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{th}}} = \frac{250\text{ kW}}{0.6812} = \mathbf{367\ kW}$$

9-52E An ideal diesel cycle has a a cutoff ratio of 1.4. The power produced 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 R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis The specific volume of the air at the start of the compression is



$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(510\,\text{R})}{14.4\,\text{psia}} = 13.12\,\text{ft}^3/\text{lbm}$$

The total air mass taken by all 8 cylinders when they are charged is

$$m = N_{\text{cyl}}\frac{\Delta V}{v_1} = N_{\text{cyl}}\frac{\pi B^2 S/4}{v_1} = (8)\frac{\pi(4/12\,\text{ft})^2(4/12\,\text{ft})/4}{13.12\,\text{ft}^3/\text{lbm}} = 0.01774\,\text{lbm}$$

The rate at which air is processed by the engine is determined from

$$\dot{m} = \frac{m\dot{n}}{N_{\text{rev}}} = \frac{(0.01774\,\text{lbm/cycle})(1800/60\,\text{rev/s})}{2\,\text{rev/cycle}} = 0.2661\,\text{lbm/s} = 958.0\,\text{lbm/h}$$

since there are two revolutions per cycle in a four-stroke engine. The compression ratio is

$$r = \frac{1}{0.045} = 22.22$$

At the end of the compression, the air temperature is

$$T_2 = T_1 r^{k-1} = (510\,\text{R})(22.22)^{1.4-1} = 1763\,\text{R}$$

Application of the first law and work integral to the constant pressure heat addition gives

$$q_{\text{in}} = c_p(T_3 - T_2) = (0.240\,\text{Btu/lbm}\cdot\text{R})(2760 - 1763)\text{R} = 239.3\,\text{Btu/lbm}$$

while the thermal efficiency is

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\frac{r_c^k - 1}{k(r_c - 1)} = 1 - \frac{1}{22.22^{1.4-1}}\frac{1.4^{1.4} - 1}{1.4(1.4 - 1)} = 0.6892$$

The power produced by this engine is then

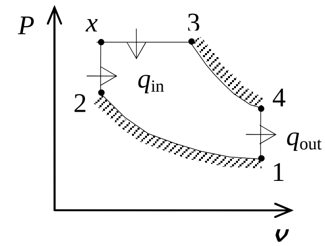
$$\begin{aligned}\dot{W}_{\text{net}} &= \dot{m}w_{\text{net}} = \dot{m}\eta_{\text{th}}q_{\text{in}} \\ &= (958.0\,\text{lbm/h})(0.6892)(239.3\,\text{Btu/lbm})\left(\frac{1\,\text{hp}}{2544.5\,\text{Btu/h}}\right) \\ &= \mathbf{62.1\,hp}\end{aligned}$$

9-53 An ideal dual cycle has a compression ratio of 14 and cutoff ratio of 1.2. The thermal efficiency, amount of heat added, and the maximum gas pressure and temperature 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, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis The specific volume of the air at the start of the compression is



$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}{80\,\text{kPa}} = 1.051\,\text{m}^3/\text{kg}$$

and the specific volume at the end of the compression is

$$v_2 = \frac{v_1}{r} = \frac{1.051\,\text{m}^3/\text{kg}}{14} = 0.07508\,\text{m}^3/\text{kg}$$

The pressure at the end of the compression is

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^k = P_1 r^k = (80\,\text{kPa})(14)^{1.4} = 3219\,\text{kPa}$$

and the maximum pressure is

$$P_x = P_3 = r_p P_2 = (1.5)(3219\,\text{kPa}) = \mathbf{4829\,kPa}$$

The temperature at the end of the compression is

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (293\,\text{K})(14)^{1.4-1} = 842.0\,\text{K}$$

and $$T_x = T_2\left(\frac{P_3}{P_2}\right) = (842.0\,\text{K})\left(\frac{4829\,\text{kPa}}{3219\,\text{kPa}}\right) = 1263\,\text{K}$$

From the definition of cutoff ratio

$$v_3 = r_c v_x = r_c v_2 = (1.2)(0.07508\,\text{m}^3/\text{kg}) = 0.09010\,\text{m}^3/\text{kg}$$

The remaining state temperatures are then

$$T_3 = T_x\left(\frac{v_3}{v_x}\right) = (1263\,\text{K})\left(\frac{0.09010}{0.07508}\right) = \mathbf{1516\,K}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1516\,\text{K})\left(\frac{0.09010}{1.051}\right)^{1.4-1} = 567.5\,\text{K}$$

Applying the first law and work expression to the heat addition processes gives

$$\begin{aligned} q_{\text{in}} &= c_v(T_x - T_2) + c_p(T_3 - T_x) \\ &= (0.718\,\text{kJ/kg}\cdot\text{K})(1263 - 842.0)\text{K} + (1.005\,\text{kJ/kg}\cdot\text{K})(1516 - 1263)\text{K} \\ &= \mathbf{556.5\,kJ/kg} \end{aligned}$$

The heat rejected is

$$q_{\text{out}} = c_v(T_4 - T_1) = (0.718\,\text{kJ/kg}\cdot\text{K})(567.5 - 293)\text{K} = 197.1\,\text{kJ/kg}$$

Then, $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{197.1\,\text{kJ/kg}}{556.5\,\text{kJ/kg}} = \mathbf{0.646}$$

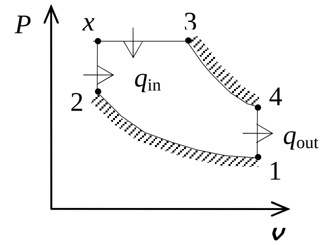
9-54 An ideal dual cycle has a compression ratio of 14 and cutoff ratio of 1.2. The thermal efficiency, amount of heat added, and the maximum gas pressure and temperature are to be determined.

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, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis The specific volume of the air at the start of the compression is

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(253\ \text{K})}{80\ \text{kPa}} = 0.9076\ \text{m}^3/\text{kg}$$



and the specific volume at the end of the compression is

$$v_2 = \frac{v_1}{r} = \frac{0.9076\ \text{m}^3/\text{kg}}{14} = 0.06483\ \text{m}^3/\text{kg}$$

The pressure at the end of the compression is

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^k = P_1 r^k = (80\ \text{kPa})(14)^{1.4} = 3219\ \text{kPa}$$

and the maximum pressure is

$$P_x = P_3 = r_p P_2 = (1.5)(3219\ \text{kPa}) = \mathbf{4829\ kPa}$$

The temperature at the end of the compression is

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (253\ \text{K})(14)^{1.4-1} = 727.1\ \text{K}$$

and $$T_x = T_2\left(\frac{P_3}{P_2}\right) = (727.1\ \text{K})\left(\frac{4829\ \text{kPa}}{3219\ \text{kPa}}\right) = 1091\ \text{K}$$

From the definition of cutoff ratio

$$v_3 = r_c v_x = r_c v_2 = (1.2)(0.06483\ \text{m}^3/\text{kg}) = 0.07780\ \text{m}^3/\text{kg}$$

The remaining state temperatures are then

$$T_3 = T_x\left(\frac{v_3}{v_x}\right) = (1091\ \text{K})\left(\frac{0.07780}{0.06483}\right) = \mathbf{1309\ K}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1309\ \text{K})\left(\frac{0.07780}{0.9076}\right)^{1.4-1} = 490.0\ \text{K}$$

Applying the first law and work expression to the heat addition processes gives

$$\begin{aligned} q_{in} &= c_v(T_x - T_2) + c_p(T_3 - T_x) \\ &= (0.718\ \text{kJ/kg}\cdot\text{K})(1091-727.1)\text{K} + (1.005\ \text{kJ/kg}\cdot\text{K})(1309-1091)\text{K} \\ &= \mathbf{480.4\ kJ/kg} \end{aligned}$$

The heat rejected is

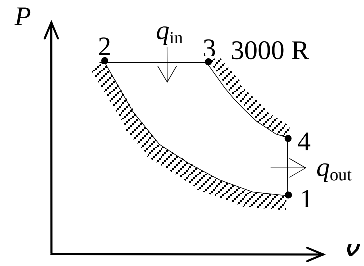
$$q_{out} = c_v(T_4 - T_1) = (0.718\ \text{kJ/kg}\cdot\text{K})(490.0-253)\text{K} = 170.2\ \text{kJ/kg}$$

Then, $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{170.2\ \text{kJ/kg}}{480.4\ \text{kJ/kg}} = \mathbf{0.646}$$

9-55E 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-17E.



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.

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

(*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.

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

(*c*)

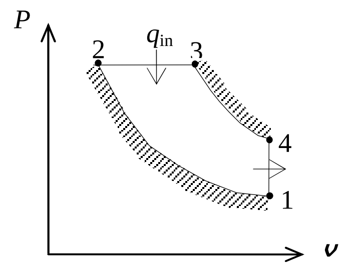
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{158.87 \text{ Btu/lbm}}{388.63 \text{ Btu/lbm}} = \mathbf{59.1\%}$$

9-56E 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.

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

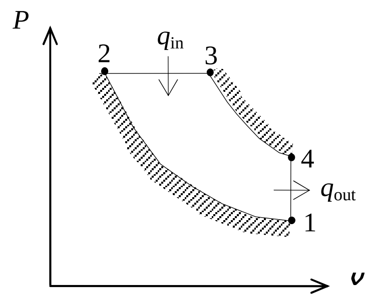
(*c*) $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{108 \text{ Btu/lbm}}{306 \text{ Btu/lbm}} = \mathbf{64.6\%}$$

9-57 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, R = 0.287 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.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2200\text{K}}{971.1\text{K}} = 2.265$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.265 v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.265}{r}\right)^{k-1} = (2200\text{ K})\left(\frac{2.265}{20}\right)^{0.4} = 920.6\text{ K}$$

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

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

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

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

(*b*)

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

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

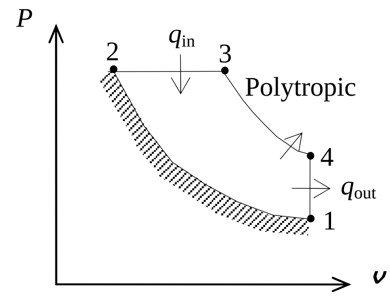
$$\text{MEP} = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1 - 1/r)} = \frac{784.4\text{ kJ/kg}}{(0.885\text{ m}^3/\text{kg})(1 - 1/20)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{933\text{ kPa}}$$

9-58 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, R = 0.287 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.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2200\text{ K}}{971.1\text{ K}} = 2.265$$

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = T_3\left(\frac{2.265 v_2}{v_4}\right)^{n-1} = T_3\left(\frac{2.265}{r}\right)^{n-1} = (2200\text{ K})\left(\frac{2.265}{20}\right)^{0.35} = 1026\text{ K}$$

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

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

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:

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

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

$$q_{34,in} - w_{34,out} = u_4 - u_3 \longrightarrow q_{34,in} = w_{34,out} + c_v(T_4 - T_3)$$

$$= 963\text{ kJ/kg} + (0.718\text{ kJ/kg}\cdot\text{K})(1026 - 2200)\text{ K}$$

$$= 120.1\text{ kJ/kg}$$

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

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

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

(b)

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

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

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

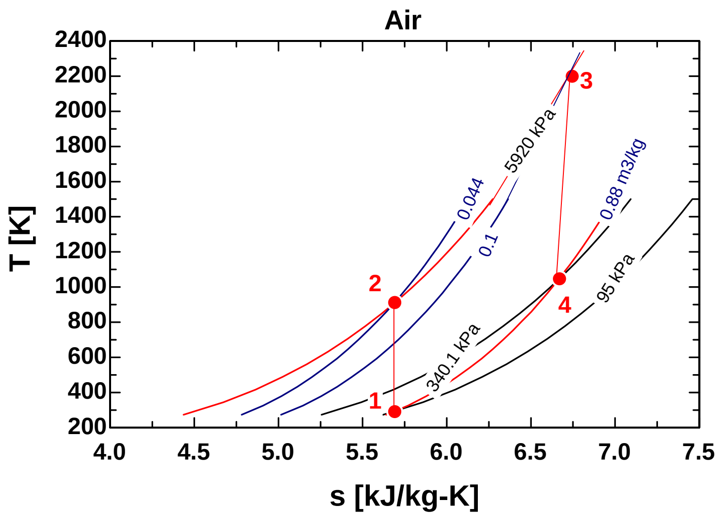
9-59 Problem 9-58 is reconsidered. The effect of the compression ratio on the net work output, mean effective pressure, and thermal efficiency is to be investigated. Also, *T-s* and *P-v* diagrams for the cycle are to be plotted.

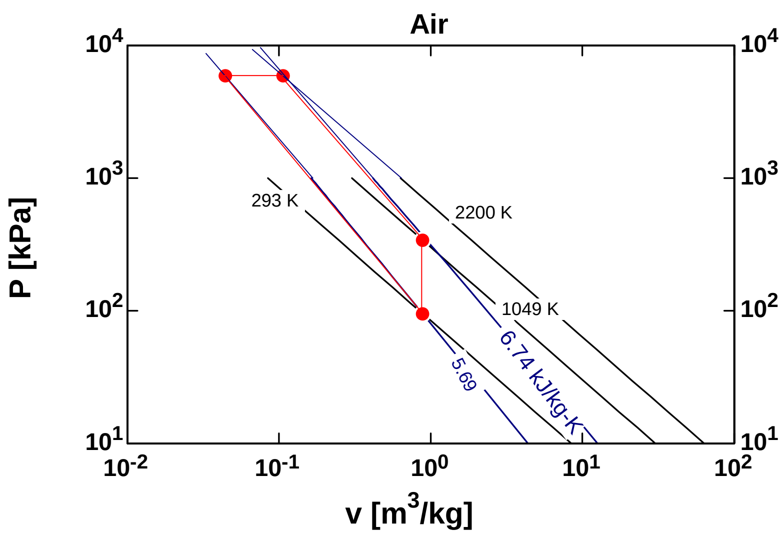
Analysis Using EES, the problem is solved as follows:

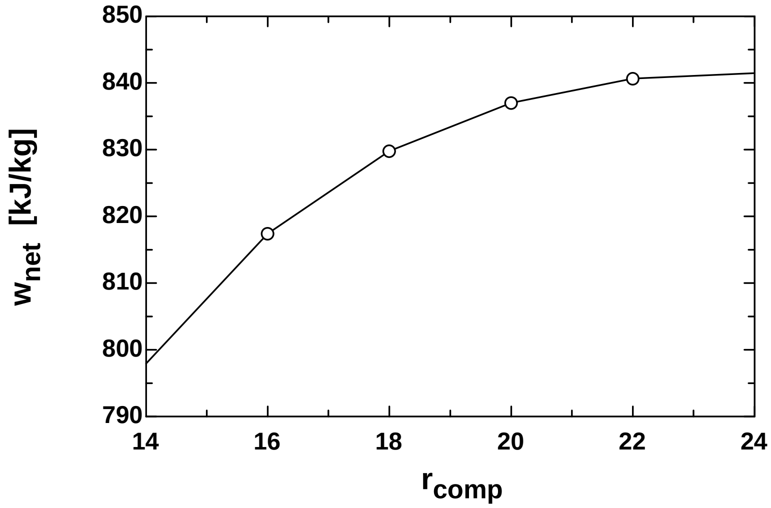
```
Procedure QTotal(q_12,q_23,q_34,q_41: q_in_total,q_out_total)
q_in_total = 0
q_out_total = 0
IF (q_12 > 0) THEN q_in_total = q_12 ELSE  q_out_total = - q_12
If q_23 > 0 then q_in_total = q_in_total + q_23 else q_out_total = q_out_total - q_23
If q_34 > 0 then q_in_total = q_in_total + q_34 else q_out_total = 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]=R*T[1]
R=0.287 [kJ/kg-K]
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]=R*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]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 - w_34 = DELTAu_34    "q_34 is not 0 for the ploytropic 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])

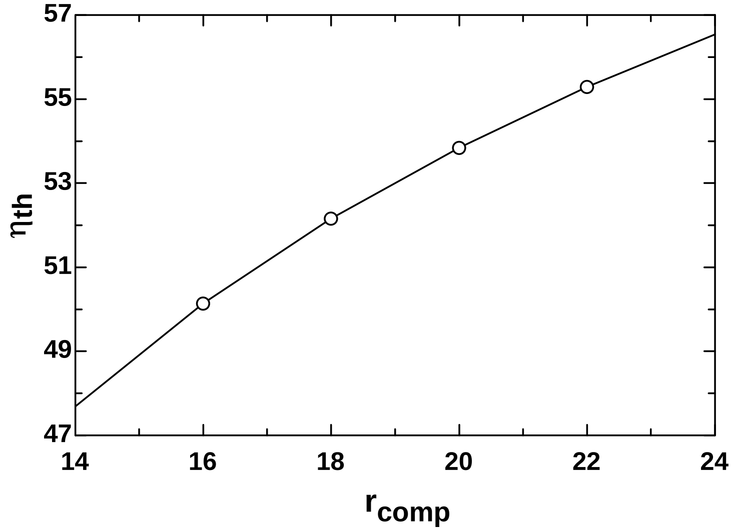
r_{comp}	η_{th}	MEP [kPa]	w_{net} [kJ/kg]
14	47.69	970.8	797.9
16	50.14	985	817.4
18	52.16	992.6	829.8
20	53.85	995.4	837.0
22	55.29	994.9	840.6
24	56.54	992	841.5



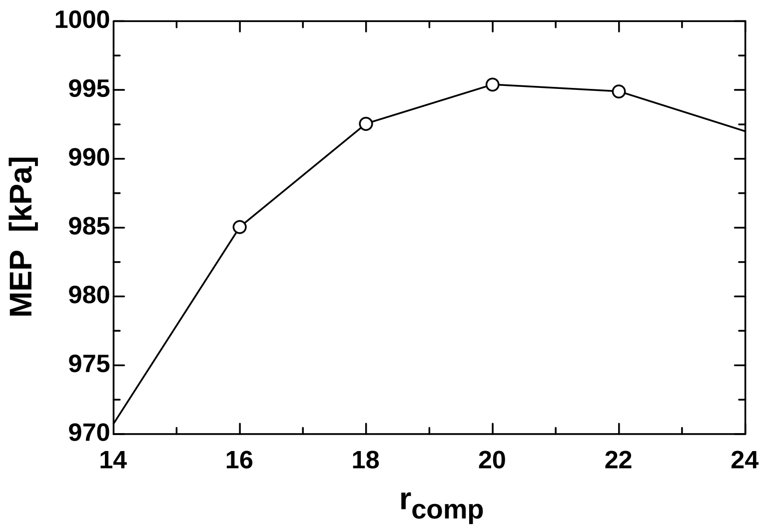




850
840
830
820
810
800
790
14
16
18
20
22
24
r_{comp}
w_{net} [kJ/kg]



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



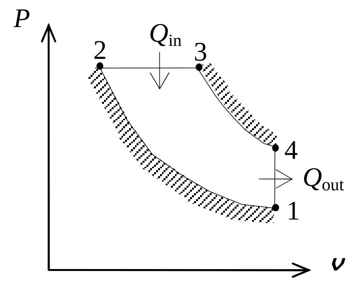
1000
995
990
985
980
975
970
14
16
18
20
22
24
r_{comp}
MEP [kPa]

9-60 A four-cylinder ideal diesel engine with air as the working fluid has a compression ratio of 22 and a cutoff ratio of 1.8. The power the engine will deliver at 2300 rpm is to be determined.

Assumptions **1** The cold 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, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (343\text{ K})(22)^{0.4} = 1181\text{ K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2}T_2 = 1.8T_2 = (1.8)(1181\text{ K}) = 2126\text{ K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.2v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.2}{r}\right)^{k-1} = (2216\text{ K})\left(\frac{1.8}{22}\right)^{0.4} = 781\text{ K}$$

$$m = \frac{P_1 v_1}{RT_1} = \frac{(97\text{ kPa})(0.0020\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(343\text{ K})} = 0.001971\text{ kg}$$

$$\begin{aligned} Q_{\text{in}} &= m(h_3 - h_2) = mc_p(T_3 - T_2) \\ &= (0.001971\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(2216 - 1181)\text{K} = 1.871\text{ kJ} \end{aligned}$$

$$\begin{aligned} Q_{\text{out}} &= m(u_4 - u_1) = mc_v(T_4 - T_1) \\ &= (0.001971\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(781 - 343)\text{K} = 0.6198\text{ kJ} \end{aligned}$$

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 1.871 - 0.6198 = 1.251\text{ kJ/rev}$$

$$\dot{W}_{\text{net,out}} = \dot{n}W_{\text{net,out}} = (2300/60\text{ rev/s})(1.251\text{ kJ/rev}) = \mathbf{48.0\ kW}$$

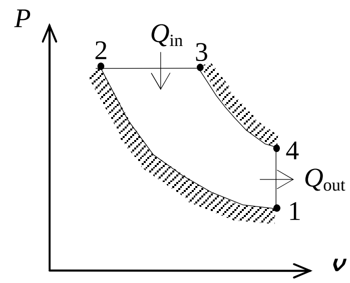
Discussion Note that for 2-stroke engines, 1 thermodynamic cycle is equivalent to 1 mechanical cycle (and thus revolutions).

9-61 A four-cylinder ideal diesel engine with nitrogen as the working fluid has a compression ratio of 22 and a cutoff ratio of 1.8. The power the engine will deliver at 2300 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, R = 0.2968 kJ/kg·K, and k = 1.4 (Table A-2).

Analysis Process 1-2: isentropic compression.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (343\ \text{K})(22)^{0.4} = 1181\ \text{K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2}T_2 = 1.8T_2 = (1.8)(1181\ \text{K}) = 2126\ \text{K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.2v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.2}{r}\right)^{k-1} = (2216\ \text{K})\left(\frac{1.8}{22}\right)^{0.4} = 781\ \text{K}$$

$$m = \frac{P_1 v_1}{RT_1} = \frac{(97\ \text{kPa})(0.0020\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(343\ \text{K})} = 0.001906\ \text{kg}$$

$$\begin{aligned} Q_{\text{in}} &= m(h_3 - h_2) = mc_p(T_3 - T_2) \\ &= (0.001906\ \text{kg})(1.039\ \text{kJ/kg}\cdot\text{K})(2216 - 1181)\text{K} = 1.871\ \text{kJ} \end{aligned}$$

$$\begin{aligned} Q_{\text{out}} &= m(u_4 - u_1) = mc_v(T_4 - T_1) \\ &= (0.001906\ \text{kg})(0.743\ \text{kJ/kg}\cdot\text{K})(781 - 343)\text{K} = 0.6202\ \text{kJ} \end{aligned}$$

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 1.871 - 0.6202 = 1.251\ \text{kJ/rev}$$

$$\dot{W}_{\text{net,out}} = \dot{n}W_{\text{net,out}} = (2300/60\ \text{rev/s})(1.251\ \text{kJ/rev}) = \mathbf{47.9\ kW}$$

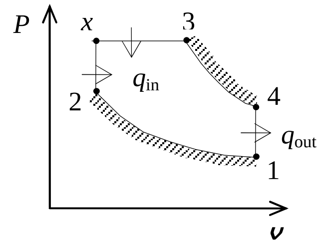
Discussion Note that for 2-stroke engines, 1 thermodynamic cycle is equivalent to 1 mechanical cycle (and thus revolutions).

9-62 An ideal dual cycle has a compression ratio of 18 and cutoff ratio of 1.1. The power produced by 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 = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis We begin by fixing the temperatures at all states.



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (291\text{ K})(18)^{1.4-1} = 924.7\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1 r^{k} = (90\text{ kPa})(18)^{1.4} = 5148\text{ kPa}$$

$$P_x = P_3 = r_p P_2 = (1.1)(5148\text{ kPa}) = 5663\text{ kPa}$$

$$T_x = T_2\left(\frac{P_x}{P_2}\right) = (924.7\text{ K})\left(\frac{5663\text{ kPa}}{5148\text{ kPa}}\right) = 1017\text{ K}$$

$$T_3 = r_c T_x = (1.1)(1017\text{ K}) = 1119\text{ K}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{r_c}{r}\right)^{k-1} = (1119\text{ K})\left(\frac{1.1}{18}\right)^{1.4-1} = 365.8\text{ K}$$

Applying the first law to each of the processes gives

$$w_{1-2} = c_v(T_2 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(924.7 - 291)\text{K} = 455.0\text{ kJ/kg}$$

$$q_{x-3} = c_p(T_3 - T_x) = (1.005\text{ kJ/kg}\cdot\text{K})(1119 - 1017)\text{K} = 102.5\text{ kJ/kg}$$

$$w_{x-3} = q_{x-3} - c_v(T_3 - T_x) = 102.5 - (0.718\text{ kJ/kg}\cdot\text{K})(1119 - 1017)\text{K} = 29.26\text{ kJ/kg}$$

$$w_{3-4} = c_v(T_3 - T_4) = (0.718\text{ kJ/kg}\cdot\text{K})(1119 - 365.8)\text{K} = 540.8\text{ kJ/kg}$$

The net work of the cycle is

$$w_{\text{net}} = w_{3-4} + w_{x-3} - w_{1-2} = 540.8 + 29.26 - 455.0 = 115.1\text{ kJ/kg}$$

The mass in the device is given by

$$m = \frac{P_1 V_1}{RT_1} = \frac{(90\text{ kPa})(0.003\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(291\text{ K})} = 0.003233\text{ kg}$$

The net power produced by this engine is then

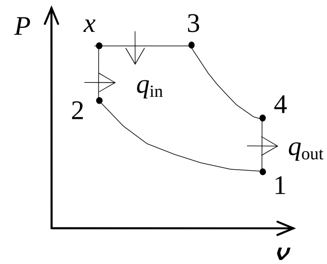
$$\dot{W}_{\text{net}} = m w_{\text{net}}\dot{n} = (0.003233\text{ kg/cycle})(115.1\text{ kJ/kg})(4000/60\text{ cycle/s}) = \mathbf{24.8\ kW}$$

9-63 A dual cycle with non-isentropic compression and expansion processes is considered. The power produced by 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 = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis We begin by fixing the temperatures at all states.



$$T_{2s} = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (291\text{ K})(18)^{1.4-1} = 924.7\text{ K}$$

$$\eta = \frac{T_{2s}-T_1}{T_2-T_1} \longrightarrow T_2 = T_1 + \frac{T_{2s}-T_1}{\eta} = (291\text{ K}) + \frac{(924.7-291)\text{ K}}{0.85} = 1037\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1 r^k = (90\text{ kPa})(18)^{1.4} = 5148\text{ kPa}$$

$$P_x = P_3 = r_p P_2 = (1.1)(5148\text{ kPa}) = 5663\text{ kPa}$$

$$T_x = T_2\left(\frac{P_x}{P_2}\right) = (1037\text{ K})\left(\frac{5663\text{ kPa}}{5148\text{ kPa}}\right) = 1141\text{ K}$$

$$T_3 = r_c T_x = (1.1)(1141\text{ K}) = 1255\text{ K}$$

$$T_{4s} = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{r_c}{r}\right)^{k-1} = (1255\text{ K})\left(\frac{1.1}{18}\right)^{1.4-1} = 410.3\text{ K}$$

$$\eta = \frac{T_3-T_4}{T_3-T_{4s}} \longrightarrow T_4 = T_3 - \eta(T_3 - T_{4s}) = (1255\text{ K}) - (0.90)(1255-410.3)\text{ K} = 494.8\text{ K}$$

Applying the first law to each of the processes gives

$$w_{1-2} = c_v(T_2 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(1037-291)\text{K} = 535.6\text{ kJ/kg}$$

$$q_{x-3} = c_p(T_3 - T_x) = (1.005\text{ kJ/kg}\cdot\text{K})(1255-1141)\text{K} = 114.6\text{ kJ/kg}$$

$$w_{x-3} = q_{x-3} - c_v(T_3 - T_x) = 114.6 - (0.718\text{ kJ/kg}\cdot\text{K})(1255-1141)\text{K} = 32.75\text{ kJ/kg}$$

$$w_{3-4} = c_v(T_3 - T_4) = (0.718\text{ kJ/kg}\cdot\text{K})(1255-494.8)\text{K} = 545.8\text{ kJ/kg}$$

The net work of the cycle is

$$w_{\text{net}} = w_{3-4} + w_{x-3} - w_{1-2} = 545.8 + 32.75 - 535.6 = 42.95\text{ kJ/kg}$$

The mass in the device is given by

$$m = \frac{P_1 V_1}{RT_1} = \frac{(90\text{ kPa})(0.003\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(291\text{ K})} = 0.003233\text{ kg}$$

The net power produced by this engine is then

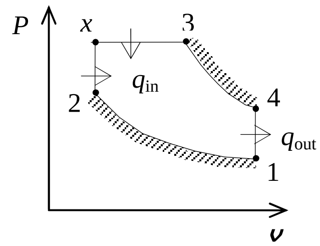
$$\dot{W}_{\text{net}} = m w_{\text{net}} \dot{n} = (0.003233\text{ kg/cycle})(42.95\text{ kJ/kg})(4000/60\text{ cycle/s}) = \mathbf{9.26\text{ kW}}$$

9-64E An ideal dual cycle has a compression ratio of 15 and cutoff ratio of 1.4. The net work, heat addition, 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 = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis Working around the cycle, the germane properties at the various states are



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = T_1 r^{k-1} = (535\text{ R})(15)^{1.4-1} = 1580\text{ R}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = P_1 r^{k} = (14.2\text{ psia})(15)^{1.4} = 629.2\text{ psia}$$

$$P_x = P_3 = r_p P_2 = (1.1)(629.2\text{ psia}) = 692.1\text{ psia}$$

$$T_x = T_2\left(\frac{P_x}{P_2}\right) = (1580\text{ R})\left(\frac{692.1\text{ psia}}{629.2\text{ psia}}\right) = 1738\text{ R}$$

$$T_3 = T_x\left(\frac{v_3}{v_x}\right) = T_x r_c = (1738\text{ R})(1.4) = 2433\text{ R}$$

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{r_c}{r}\right)^{k-1} = (2433\text{ R})\left(\frac{1.4}{15}\right)^{1.4-1} = 942.2\text{ R}$$

Applying the first law to each of the processes gives

$$w_{1-2} = c_v(T_2 - T_1) = (0.171\text{ Btu/lbm}\cdot\text{R})(1580 - 535)\text{R} = 178.7\text{ Btu/lbm}$$

$$q_{2-x} = c_v(T_x - T_2) = (0.171\text{ Btu/lbm}\cdot\text{R})(1738 - 1580)\text{R} = 27.02\text{ Btu/lbm}$$

$$q_{x-3} = c_p(T_3 - T_x) = (0.240\text{ Btu/lbm}\cdot\text{R})(2433 - 1738)\text{R} = 166.8\text{ Btu/lbm}$$

$$w_{x-3} = q_{x-3} - c_v(T_3 - T_x) = 166.8\text{ Btu/lbm} - (0.171\text{ Btu/lbm}\cdot\text{R})(2433 - 1738)\text{R} = 47.96\text{ Btu/lbm}$$

$$w_{3-4} = c_v(T_3 - T_4) = (0.171\text{ Btu/lbm}\cdot\text{R})(2433 - 942.2)\text{R} = 254.9\text{ Btu/lbm}$$

The net work of the cycle is

$$w_{\text{net}} = w_{3-4} + w_{x-3} - w_{1-2} = 254.9 + 47.96 - 178.7 = \mathbf{124.2\ Btu/lbm}$$

and the net heat addition is

$$q_{\text{in}} = q_{2-x} + q_{x-3} = 27.02 + 166.8 = \mathbf{193.8\ Btu/lbm}$$

Hence, the thermal efficiency is

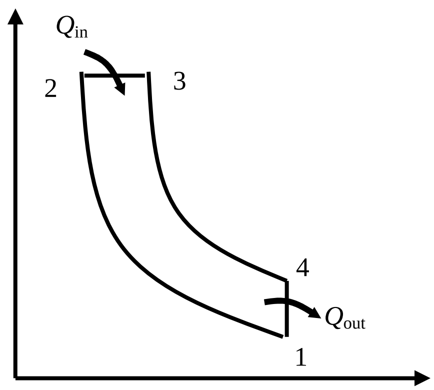
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{124.2\text{ Btu/lbm}}{193.8\text{ Btu/lbm}} = \mathbf{0.641}$$

9-65 A six-cylinder compression ignition engine operates on the ideal Diesel cycle. The maximum temperature in the cycle, the cutoff ratio, the net work output per cycle, the thermal efficiency, the mean effective pressure, the net power output, and the specific fuel consumption 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 850 K are c_p = 1.110 kJ/kg·K, c_v = 0.823 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.349 (Table A-2b).

Analysis (*a*) Process 1-2: Isentropic compression



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (340\text{ K})(19)^{1.349-1} = 950.1\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = (95\text{ kPa})(19)^{1.349} = 5044\text{ kPa}$$

The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 19 = \frac{V_c + 0.0045\text{ m}^3}{V_c}$$

$$V_c = 0.0001778\text{ m}^3$$

$$V_1 = V_c + V_d = 0.0001778 + 0.0032 = 0.003378\text{ m}^3$$

The total mass contained in the cylinder is

$$m = \frac{P_1V_1}{RT_1} = \frac{(95\text{ kPa})(0.003378\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(340\text{ K})} = 0.003288\text{ kg}$$

The mass of fuel burned during one cycle is

$$AF = \frac{m_a}{m_f} = \frac{m - m_f}{m_f} \longrightarrow 28 = \frac{(0.003288\text{ kg}) - m_f}{m_f} \longrightarrow m_f = 0.0001134\text{ kg}$$

Process 2-3: constant pressure heat addition

$$Q_{in} = m_f q_{HV}\eta_c = (0.0001134\text{ kg})(42{,}500\text{ kJ/kg})(0.98) = 4.723\text{ kJ}$$

$$Q_{in} = mc_v(T_3 - T_2) \longrightarrow 4.723\text{ kJ} = (0.003288\text{ kg})(0.823\text{ kJ/kg.K})(T_3 - 950.1)\text{K} \longrightarrow T_3 = \mathbf{2244\ K}$$

The cutoff ratio is

$$\beta = \frac{T_3}{T_2} = \frac{2244\text{ K}}{950.1\text{ K}} = \mathbf{2.362}$$

(*b*) $$V_2 = \frac{V_1}{r} = \frac{0.003378\text{ m}^3}{19} = 0.0001778\text{ m}^3$$

$$V_3 = \beta V_2 = (2.362)(0.0001778\text{ m}^3) = 0.0004199\text{ m}^3$$

$$V_4 = V_1$$

$$P_3 = P_2$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{\boldsymbol{v}_3}{\boldsymbol{v}_4}\right)^{k-1} = (2244\text{ K})\left(\frac{0.0004199\text{ m}^3}{0.003378\text{ m}^3}\right)^{1.349\text{-}1} = 1084\text{ K}$$

$$P_4 = P_3\left(\frac{\boldsymbol{v}_3}{\boldsymbol{v}_4}\right)^{k} = (5044\text{ kPa})\left(\frac{0.0004199\text{ m}^3}{0.003378\text{ m}^3}\right)^{1.349} = 302.9\text{ kPa}$$

Process 4-1: constant voume heat rejection.

$$Q_{\text{out}} = mc_{\boldsymbol{v}}(T_4 - T_1) = (0.003288\text{ kg})(0.823\text{ kJ/kg}\cdot\text{K})(1084 - 340)\text{K} = 2.013\text{ kJ}$$

The net work output and the thermal efficiency are

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 4.723 - 2.013 = \mathbf{2.710\ kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_{\text{in}}} = \frac{2.710\text{ kJ}}{4.723\text{ kJ}} = 0.5737 = \mathbf{57.4\%}$$

(*c*) The mean effective pressure is determined to be

$$\text{MEP} = \frac{W_{\text{net,out}}}{\boldsymbol{v}_1 - \boldsymbol{v}_2} = \frac{2.710\text{ kJ}}{(0.003378 - 0.0001778)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{847\ kPa}$$

(*d*) The power for engine speed of 1750 rpm is

$$\dot{W}_{\text{net}} = W_{\text{net}}\frac{\dot{n}}{2} = (2.710\text{ kJ/cycle})\frac{1750\text{ (rev/min)}}{(2\text{ rev/cycle})}\left(\frac{1\text{ min}}{60\text{ s}}\right) = \mathbf{39.5\ kW}$$

Note that there are two revolutions in one cycle in four-stroke engines.

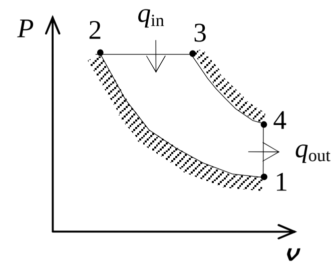
(*e*) Finally, the specific fuel consumption is

$$\text{sfc} = \frac{m_f}{W_{\text{net}}} = \frac{0.0001134\text{ kg}}{2.710\text{ kJ/kg}}\left(\frac{1000\text{ g}}{1\text{ kg}}\right)\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = \mathbf{151\ g/kWh}$$

9-66 An expression for cutoff ratio of an ideal diesel cycle is to be developed.

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.

Analysis Employing the isentropic process equations,



$$T_2 = T_1 r^{k-1}$$

while the ideal gas law gives

$$T_3 = T_2 r_c = r_c r^{k-1} T_1$$

When the first law and the closed system work integral is applied to the constant pressure heat addition, the result is

$$q_{\text{in}} = c_p (T_3 - T_2) = c_p (r_c r^{k-1} T_1 - r^{k-1} T_1)$$

When this is solved for cutoff ratio, the result is

$$r_c = 1 + \frac{q_{\text{in}}}{c_p r^{k-1} T_1}$$

9-67 An expression for the thermal efficiency of a dual cycle is to be developed and the thermal efficiency for a given case is to be calculated.

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.

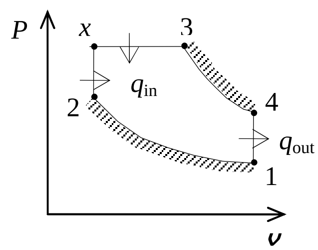
Properties The properties of air at room temperature are c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2)

Analysis The thermal efficiency of a dual cycle may be expressed as

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_x - T_2) + c_p(T_3 - T_x)}$$

By applying the isentropic process relations for ideal gases with constant specific heats to the processes 1-2 and 3-4, as well as the ideal gas equation of state, the temperatures may be eliminated from the thermal efficiency expression. This yields the result

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_p r_c^k - 1}{k r_p (r_c - 1) + r_p - 1}\right]$$



where

$$r_p = \frac{P_x}{P_2} \quad \text{and} \quad r_c = \frac{v_3}{v_x}$$

When $r_c = r_p$, we obtain

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}}\left(\frac{r_p^{k+1} - 1}{k(r_p^2 - r_p) + r_p - 1}\right)$$

For the case r = 20 and r_p = 2,

$$\eta_{\text{th}} = 1 - \frac{1}{20^{1.4-1}}\left(\frac{2^{1.4+1} - 1}{1.4(2^2 - 2) + 2 - 1}\right) = \mathbf{0.660}$$

9-68 An expression regarding the thermal efficiency of a dual cycle for a special case is to be obtained.

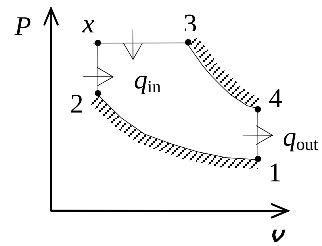
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Analysis The thermal efficiency of a dual cycle may be expressed as

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_v (T_4 - T_1)}{c_v (T_x - T_2) + c_p (T_3 - T_x)}$$

By applying the isentropic process relations for ideal gases with constant specific heats to the processes 1-2 and 3-4, as well as the ideal gas equation of state, the temperatures may be eliminated from the thermal efficiency expression. This yields the result

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}} \left[\frac{r_p r_c^k - 1}{k r_p (r_c - 1) + r_p - 1} \right]$$



where

$$r_p = \frac{P_x}{P_2} \quad \text{and} \quad r_c = \frac{v_3}{v_x}$$

When $r_c = r_p$, we obtain

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}} \left(\frac{r_p^{k+1} - 1}{k(r_p^2 - r_p) + r_p - 1} \right)$$

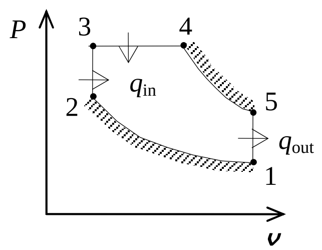
Rearrangement of this result gives

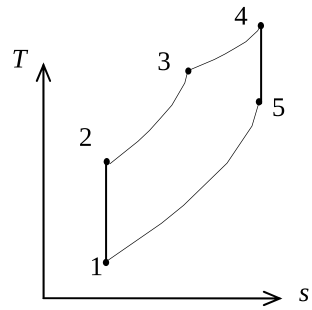
$$\frac{r_p^{k+1} - 1}{k(r_p^2 - r_p) + r_p - 1} = (1 - \eta_{\text{th}}) r^{k-1}$$

9-69 The five processes of the dual cycle is described. The *P-v* and *T-s* diagrams for this cycle is to be sketched. An expression for the cycle thermal efficiency is to be obtained and the limit of the efficiency is to be evaluated for certain cases.

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.

Analysis (a) The *P-v* and *T-s* diagrams for this cycle are as shown.





(b) Apply first law to the closed system for processes 2-3, 3-4, and 5-1 to show:

$$q_{in} = C_v\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right)$$

$$q_{out} = C_v\left(T_5 - T_1\right)$$

The cycle thermal efficiency is given by

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_v\left(T_5 - T_1\right)}{C_v\left(T_3 - T_2\right) + C_p\left(T_4 - T_3\right)} = 1 - \frac{T_1\left(T_5/T_1 - 1\right)}{T_2\left(T_3/T_2 - 1\right) + kT_3\left(T_4/T_3 - 1\right)}$$

$$\eta_{th} = 1 - \frac{\left(T_5/T_1 - 1\right)}{\frac{T_2}{T_1}\left(T_3/T_2 - 1\right) + k\frac{T_3}{T_1}\left(T_4/T_3 - 1\right)}$$

Process 1-2 is isentropic; therefore,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = r^{k-1}$$

Process 2-3 is constant volume; therefore,

$$\frac{T_3}{T_2} = \frac{P_3 V_3}{P_2 V_2} = \frac{P_3}{P_2} = r_p$$

Process 3-4 is constant pressure; therefore,

$$\frac{P_4 V_4}{T_4} = \frac{P_3 V_3}{T_3} \Rightarrow \frac{T_4}{T_3} = \frac{V_4}{V_3} = r_c$$

Process 4-5 is isentropic; therefore,

$$\frac{T_5}{T_4} = \left(\frac{V_4}{V_5}\right)^{k-1} = \left(\frac{V_4}{V_1}\right)^{k-1} = \left(\frac{r_c V_3}{V_1}\right)^{k-1} = \left(\frac{r_c V_2}{V_1}\right)^{k-1} = \left(\frac{r_c}{r}\right)^{k-1}$$

Process 5-1 is constant volume; however T_5/T_1 is found from the following.

$$\frac{T_5}{T_1} = \frac{T_5}{T_4}\frac{T_4}{T_3}\frac{T_3}{T_2}\frac{T_2}{T_1} = \left(\frac{r_c}{r}\right)^{k-1} r_c r_p r^{k-1} = r_c^k r_p$$

The ratio T_3/T_1 is found from the following.

$$\frac{T_3}{T_1} = \frac{T_3}{T_2}\frac{T_2}{T_1} = r_p r^{k-1}$$

The efficiency becomes

$$\eta_{th} = 1 - \frac{r_c^k r_p - 1}{r^{k-1}\left(r_p - 1\right) + k r_p r^{k-1}\left(r_c - 1\right)}$$

(c) In the limit as r_p approaches unity, the cycle thermal efficiency becomes

$$\lim_{r_p \to 1} \eta_{th} = 1 - \left\{ \lim_{r_p \to 1} \frac{r_c^k r_p - 1}{r^{k-1}\left(r_p - 1\right) + k r_p r^{k-1}\left(r_c - 1\right)} \right\}$$

$$\lim_{r_p \to 1} \eta_{th} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k\left(r_c - 1\right)}\right] = \eta_{th\ Diesel}$$

(d) In the limit as r_c approaches unity, the cycle thermal efficiency becomes

$$\lim_{r_c \to 1} \eta_{th} = 1 - \left\{ \lim_{r_p \to 1} \frac{r_c^k r_p - 1}{r^{k-1}\left(r_p - 1\right) + k r_p r^{k-1}\left(r_c - 1\right)} \right\} = 1 - \left\{ \frac{r_p - 1}{r^{k-1}\left(r_p - 1\right)} \right\}$$

$$\lim_{r_c \to 1} \eta_{th} = 1 - \frac{1}{r^{k-1}} = \eta_{th\ Otto}$$

Stirling and Ericsson Cycles

9-70C The Stirling cycle.

9-71C The two isentropic processes of the Carnot cycle are replaced by two constant pressure regeneration processes in the Ericsson cycle.

9-72C The efficiencies of the Carnot and the Stirling cycles would be the same, the efficiency of the Otto cycle would be less.

9-73C The efficiencies of the Carnot and the Ericsson cycles would be the same, the efficiency of the Diesel cycle would be less.

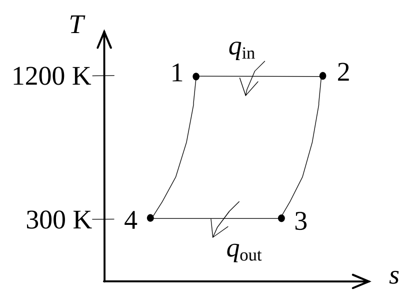
9-74 An ideal steady-flow Ericsson engine with air as the working fluid is considered. The maximum pressure in the cycle, the net work output, and the thermal efficiency of the cycle are to be determined.

Assumptions Air is an ideal gas.

Properties The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

Analysis (*a*) The entropy change during process 3-4 is

$$s_4 - s_3 = -\frac{q_{34,\text{out}}}{T_0} = -\frac{150 \text{ kJ/kg}}{300 \text{ K}} = -0.5 \text{ kJ/kg}\cdot\text{K}$$



and

$$s_4 - s_3 = c_p \ln\frac{T_4}{T_3}^{\nearrow 0} - R\ln\frac{P_4}{P_3}$$

$$= -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{P_4}{120 \text{ kPa}} = -0.5 \text{ kJ/kg}\cdot\text{K}$$

It yields $P_4 =$ **685.2 kPa**

(*b*) For reversible cycles,

$$\frac{q_{\text{out}}}{q_{\text{in}}} = \frac{T_L}{T_H} \longrightarrow q_{\text{in}} = \frac{T_H}{T_L}q_{\text{out}} = \frac{1200 \text{ K}}{300 \text{ K}}(150 \text{ kJ/kg}) = 600 \text{ kJ/kg}$$

Thus,

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 600 - 150 = \mathbf{450 \text{ kJ/kg}}$$

(*c*) The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{K}}{1200\text{K}} = \mathbf{75.0\%}$$

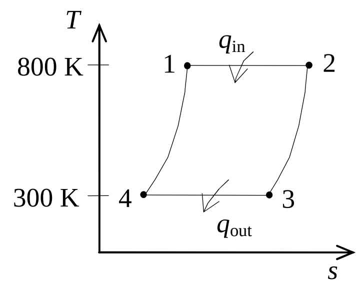
9-75 An ideal Stirling engine with air as the working fluid operates between the specified temperature and pressure limits. The net work produced per cycle and the thermal efficiency of the cycle are to be determined.

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, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis Since the specific volume is constant during process 2-3,

$$P_2 = P_3 \frac{T_2}{T_3} = (100\text{ kPa})\left(\frac{800\text{ K}}{300\text{ K}}\right) = 266.7\text{ kPa}$$



Heat is only added to the system during reversible process 1-2. Then,

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1} \\ &= 0 - (0.287\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{266.7\text{ kPa}}{2000\text{ kPa}}\right) \\ &= 0.5782\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$q_{in} = T_1(s_2 - s_1) = (800\text{ K})(0.5782\text{ kJ/kg}\cdot\text{K}) = 462.6\text{ kJ/kg}$$

The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{800\text{ K}} = \mathbf{0.625}$$

Then,

$$W_{net} = \eta_{th} m q_{in} = (0.625)(1\text{ kg})(462.6\text{ kJ/kg}) = \mathbf{289.1\ kJ}$$

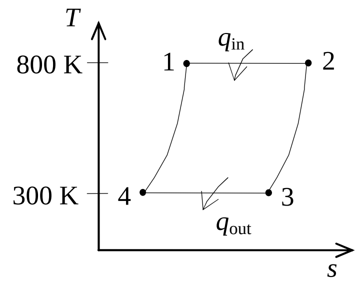
9-76 An ideal Stirling engine with air as the working fluid operates between the specified temperature and pressure limits. The power produced and the rate of heat input are to be determined.

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, R = 0.287 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis Since the specific volume is constant during process 2-3,

$$P_2 = P_3 \frac{T_2}{T_3} = (100\text{ kPa})\left(\frac{800\text{ K}}{300\text{ K}}\right) = 266.7\text{ kPa}$$



Heat is only added to the system during reversible process 1-2. Then,

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1}$$

$$= 0 - (0.287\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{266.7\text{ kPa}}{2000\text{ kPa}}\right)$$

$$= 0.5782\text{ kJ/kg}\cdot\text{K}$$

$$q_{in} = T_1(s_2 - s_1) = (800\text{ K})(0.5782\text{ kJ/kg}\cdot\text{K}) = 462.6\text{ kJ/kg}$$

The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{800\text{ K}} = 0.625$$

Then,

$$W_{net} = \eta_{th} m q_{in} = (0.625)(1\text{ kg})(462.6\text{ kJ/kg}) = 289.1\text{ kJ}$$

The rate at which heat is added to this engine is

$$\dot{Q}_{in} = m q_{in}\dot{n} = (1\text{ kg/cycle})(462.6\text{ kJ/kg})(1300/60\text{ cycle/s}) = \mathbf{10{,}020\text{ kW}}$$

while the power produced by the engine is

$$\dot{W}_{net} = W_{net}\dot{n} =)(289.1\text{ kJ/cycle})(1300/60\text{ cycle/s}) = \mathbf{6264\text{ kW}}$$

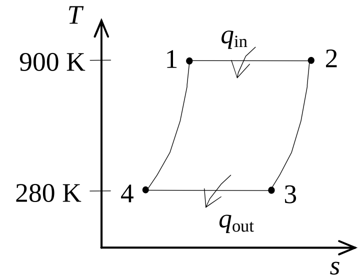
9-77 An ideal Ericsson cycle operates between the specified temperature limits. The rate of heat addition is to be determined.

Analysis The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{280 \text{ K}}{900 \text{ K}} = 0.6889$$

According to the general definition of the thermal efficiency, the rate of heat addition is

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{th}}} = \frac{500 \text{ kW}}{0.6889} = \mathbf{726\ kW}$$



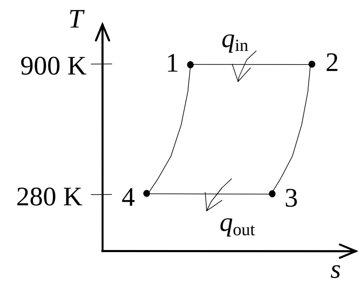
9-78 An ideal Ericsson cycle operates between the specified temperature limits. The power produced by the cycle is to be determined.

Analysis The power output is 500 kW when the cycle is repeated 2000 times per minute. Then the work per cycle is

$$W_{\text{net}} = \frac{\dot{W}_{\text{net}}}{\dot{n}} = \frac{500 \text{ kJ/s}}{(2000/60) \text{ cycle/s}} = 15 \text{ kJ/cycle}$$

When the cycle is repeated 3000 times per minute, the power output will be

$$\dot{W}_{\text{net}} = \dot{n} W_{\text{net}} = (3000/60 \text{ cycle/s})(15 \text{ kJ/cycle}) = \mathbf{750\ kW}$$



9-79E An ideal Stirling engine with air as the working fluid is considered. The temperature of the source-energy reservoir, the amount of air contained in the engine, and the maximum air pressure during the cycle are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

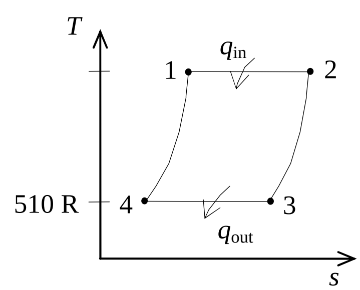
Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm.R (Table A-1E), c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2Ea).

Analysis From the thermal efficiency relation,

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = 1 - \frac{T_L}{T_H} \longrightarrow \frac{2 \text{ Btu}}{6 \text{ Btu}} = 1 - \frac{510 \text{ R}}{T_H} \longrightarrow T_H = \mathbf{765\ R}$$

State 3 may be used to determine the mass of air in the system,

$$m = \frac{P_3 V_3}{RT_3} = \frac{(10 \text{ psia})(0.5 \text{ ft}^3)}{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(510 \text{ R})} = \mathbf{0.02647\ lbm}$$



The maximum pressure occurs at state 1,

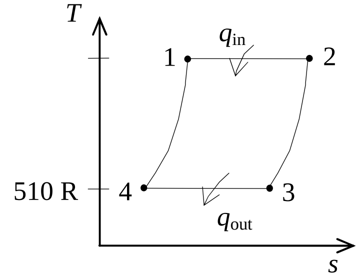
$$P_1 = \frac{mRT_1}{V_1} = \frac{(0.02647 \text{ lbm})(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(765 \text{ R})}{0.06 \text{ ft}^3} = \mathbf{125\ psia}$$

9-80E An ideal Stirling engine with air as the working fluid is considered. The temperature of the source-energy reservoir, the amount of air contained in the engine, and the maximum air pressure during the cycle are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm.R, c_p = 0.240 Btu/lbm·R, c_v = 0.171 Btu/lbm·R, and k = 1.4 (Table A-2E).

Analysis From the thermal efficiency relation,



$$\eta_{th} = \frac{W_{net}}{Q_{in}} = 1 - \frac{T_L}{T_H} \longrightarrow \frac{2.5\text{ Btu}}{6\text{ Btu}} = 1 - \frac{510\text{ R}}{T_H} \longrightarrow T_H = \mathbf{874\ R}$$

State 3 may be used to determine the mass of air in the system,

$$m = \frac{P_3 V_3}{RT_3} = \frac{(10\text{ psia})(0.5\text{ ft}^3)}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(510\text{ R})} = \mathbf{0.02647\ lbm}$$

The maximum pressure occurs at state 1,

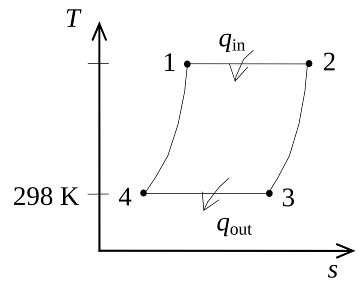
$$P_1 = \frac{mRT_1}{V_1} = \frac{(0.02647\text{ lbm})(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(874\text{ R})}{0.06\text{ ft}^3} = \mathbf{143\ psia}$$

9-81 An ideal Stirling engine with air as the working fluid operates between specified pressure limits. The heat added to the cycle and the net work produced by the cycle are to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg.K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis Applying the ideal gas equation to the isothermal process 3-4 gives



$$P_4 = P_3 \frac{v_3}{v_4} = (50\text{ kPa})(12) = 600\text{ kPa}$$

Since process 4-1 is one of constant volume,

$$T_1 = T_4\left(\frac{P_1}{P_4}\right) = (298\text{ K})\left(\frac{3600\text{ kPa}}{600\text{ kPa}}\right) = 1788\text{ K}$$

Adapting the first law and work integral to the heat addition process gives

$$q_{in} = w_{1-2} = RT_1 \ln\frac{v_2}{v_1} = (0.287\text{ kJ/kg}\cdot\text{K})(1788\text{ K})\ln(12) = \mathbf{1275\ kJ/kg}$$

Similarly,

$$q_{out} = w_{3-4} = RT_3 \ln\frac{v_4}{v_3} = (0.287\text{ kJ/kg}\cdot\text{K})(298\text{ K})\ln\left(\frac{1}{12}\right) = \mathbf{212.5\ kJ/kg}$$

The net work is then

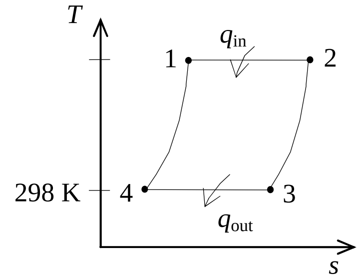
$$w_{net} = q_{in} - q_{out} = 1275 - 212.5 = \mathbf{1063\ kJ/kg}$$

9-82 An ideal Stirling engine with air as the working fluid operates between specified pressure limits. The heat transfer in the regenerator is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg.K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis Applying the ideal gas equation to the isothermal process 3-4 gives



$$P_4 = P_3 \frac{v_3}{v_4} = (50\text{ kPa})(12) = 600\text{ kPa}$$

Since process 4-1 is one of constant volume,

$$T_1 = T_4\left(\frac{P_1}{P_4}\right) = (298\text{ K})\left(\frac{3600\text{ kPa}}{600\text{ kPa}}\right) = 1788\text{ K}$$

Application of the first law to process 4-1 gives

$$q_{\text{regen}} = c_v(T_1 - T_4) = (0.718\text{ kJ/kg}\cdot\text{K})(1788 - 298)\text{K} = \mathbf{1070\text{ kJ/kg}}$$

Ideal and Actual Gas-Turbine (Brayton) Cycles

9-83C 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.

9-84C 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.

9-85C 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.

9-86C 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.

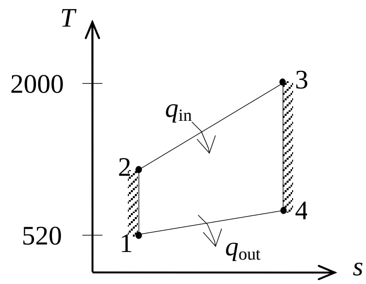
9-87C As a result of turbine and compressor inefficiencies, (a) the back work ratio increases, and (b) the thermal efficiency decreases.

9-88E 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-17E.

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



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

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

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

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

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

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

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

Then the back-work ratio becomes

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

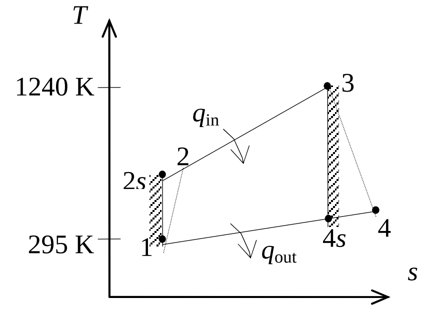
(*c*)

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

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

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

9-89 A simple Brayton cycle with air as the working fluid has a pressure ratio of 10. 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-17.

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

$$T_1 = 295\text{ K} \longrightarrow \begin{array}{l} h_1 = 295.17\text{ kJ/kg} \\ P_{r_1} = 1.3068 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (10)(1.3068) = 13.07 \longrightarrow h_{2s} = 570.26\text{ kJ/kg} \text{ and } T_{2s} = 564.9\text{ K}$$

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

$$= 295.17 + \frac{570.26 - 295.17}{0.83} = 626.60\text{ kJ/kg}$$

$$T_3 = 1240\text{ K} \longrightarrow \begin{array}{l} h_3 = 1324.93\text{ kJ/kg} \\ P_{r_3} = 272.3 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{10}\right)(272.3) = 27.23 \longrightarrow h_{4s} = 702.07\text{ kJ/kg} \text{ and } T_{4s} = 689.6\text{ K}$$

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

$$= 1324.93 - (0.87)(1324.93 - 702.07)$$

$$= 783.04\text{ kJ/kg}$$

Thus,

$$T_4 = \mathbf{764.4\ K}$$

(*b*) $$q_{in} = h_3 - h_2 = 1324.93 - 626.60 = 698.3\text{ kJ/kg}$$

$$q_{out} = h_4 - h_1 = 783.04 - 295.17 = 487.9\text{ kJ/kg}$$

$$w_{net,out} = q_{in} - q_{out} = 698.3 - 487.9 = \mathbf{210.4\ kJ/kg}$$

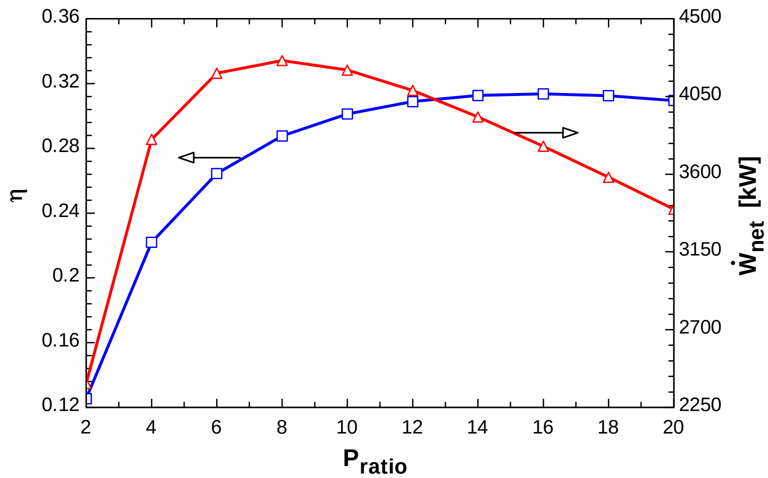
(*c*) $$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{210.4\text{ kJ/kg}}{698.3\text{ kJ/kg}} = 0.3013 = \mathbf{30.1\%}$$

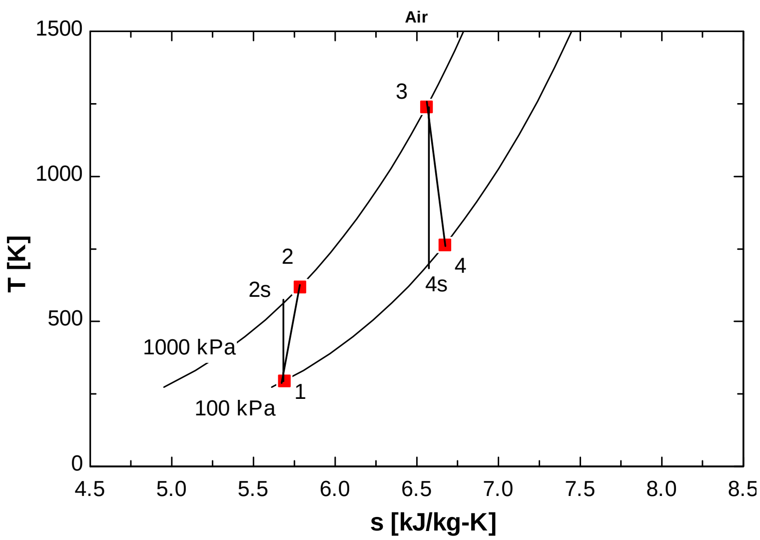
9-90 Problem 9-89 is reconsidered. The mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic efficiencies of the turbine and compressor are to be varied and a general solution for the problem by taking advantage of the diagram window method for supplying data to EES is to be developed.

Analysis Using EES, the problem is solved as follows:

```
"Input data - from diagram window"
{P_ratio = 10}
{T[1] = 295 [K]
P[1]= 100 [kPa]
T[3] = 1240 [K]
m_dot = 20 [kg/s]
Eta_c = 83/100
Eta_t = 87/100}

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

"Compressor anaysis"
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



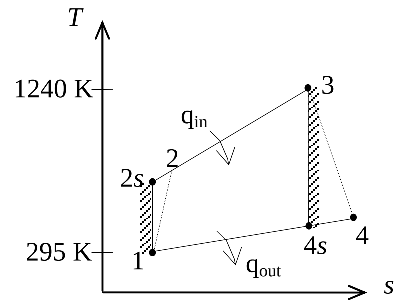


9-91 A simple Brayton cycle with air as the working fluid has a pressure ratio of 10. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

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,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (295\text{ K})(10)^{0.4/1.4} = 569.6\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1240\text{ K})\left(\frac{1}{10}\right)^{0.4/1.4} = 642.3\text{ K}$$

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

$$= 295 + \frac{569.6-295}{0.83} = 625.8\text{ K}$$

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

$$= 1240 - (0.87)(1240-642.3)$$

$$= \mathbf{720\ K}$$

(*b*) $$q_{\text{in}} = h_3 - h_2 = c_p(T_3-T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(1240-625.8)\text{K} = 617.3\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = c_p(T_4-T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(720-295)\text{K} = 427.1\text{ kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 617.3 - 427.1 = \mathbf{190.2\ kJ/kg}$$

(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{190.2\text{ kJ/kg}}{617.3\text{ kJ/kg}} = 0.3081 = \mathbf{30.8\%}$$

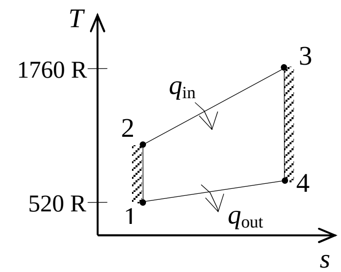
9-92E A simple ideal Brayton cycle with helium has a pressure ratio of 14. The power output is to be determined.

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

Properties The properties of helium are c_p = 1.25 Btu/lbm·R and k = 1.667 (Table A-2Ea).

Analysis Using the isentropic relations for an ideal gas,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1 r_p^{(k-1)/k} = (520\ \text{R})(14)^{0.667/1.667} = 1495\ \text{R}$$



Similarly,

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1760\ \text{R})\left(\frac{1}{14}\right)^{0.667/1.667} = 612.2\ \text{K}$$

Applying the first law to the constant-pressure heat addition process 2-3 produces

$$q_{\text{in}} = c_p(T_3 - T_2) = (1.25\ \text{Btu/lbm}\cdot\text{R})(1760 - 1495)\text{R} = 331.3\ \text{Btu/lbm}$$

Similarly,

$$q_{\text{out}} = c_p(T_4 - T_1) = (1.25\ \text{Btu/lbm}\cdot\text{R})(612.2 - 520)\text{R} = 115.3\ \text{Btu/lbm}$$

The net work production is then

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 331.3 - 115.3 = 216.0\ \text{Btu/lbm}$$

and

$$\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} = (100\ \text{lbm/min})(216.0\ \text{Btu/lbm})\left(\frac{1\ \text{hp}}{42.41\ \text{Btu/min}}\right) = \mathbf{509.3\ hp}$$

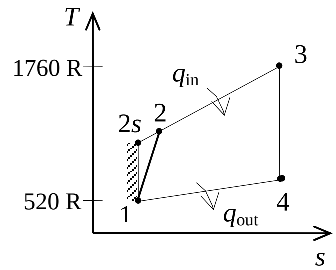
9-93E A simple Brayton cycle with helium has a pressure ratio of 14. The power output is to be determined.

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

Properties The properties of helium at room temperature are c_p = 1.25 Btu/lbm·R and k = 1.667 (Table A-2Ea).

Analysis For the compression process,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1 r_p^{(k-1)/k} = (520\text{ R})(14)^{0.667/1.667} = 1495\text{ R}$$



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

$$= 520 + \frac{1495-520}{0.95} = 1546\text{ R}$$

For the isentropic expansion process,

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1760\text{ R})\left(\frac{1}{14}\right)^{0.667/1.667} = 612.2\text{ R}$$

Applying the first law to the constant-pressure heat addition process 2-3 produces

$$q_{\text{in}} = c_p(T_3 - T_2) = (1.25\text{ Btu/lbm}\cdot\text{R})(1760-1546)\text{R} = 267.5\text{ Btu/lbm}$$

Similarly,

$$q_{\text{out}} = c_p(T_4 - T_1) = (1.25\text{ Btu/lbm}\cdot\text{R})(612.2-520)\text{R} = 115.3\text{ Btu/lbm}$$

The net work production is then

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 267.5 - 115.3 = 152.2\text{ Btu/lbm}$$

and

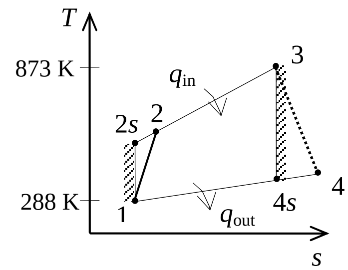
$$\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} = (100\text{ lbm/min})(152.2\text{ Btu/lbm})\left(\frac{1\text{ hp}}{42.41\text{ Btu/min}}\right) = \mathbf{358.9\ hp}$$

9-94 A simple Brayton cycle with air as the working fluid operates between the specified temperature and pressure limits. The effects of non-isentropic compressor and turbine on the back-work ratio is to be compared.

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-2a).

Analysis For the compression process,



$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (288\text{ K})(12)^{0.4/1.4} = 585.8\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_2 = T_1 + \frac{T_{2s}-T_1}{\eta_C}$$
$$= 288 + \frac{585.8-288}{0.90} = 618.9\text{ K}$$

For the expansion process,

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (873\text{ K})\left(\frac{1}{12}\right)^{0.4/1.4} = 429.2\text{ K}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} = \frac{c_p(T_3-T_4)}{c_p(T_3-T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3-T_{4s})$$
$$= 873-(0.90)(873-429.2)$$
$$= 473.6\text{ K}$$

The isentropic and actual work of compressor and turbine are

$$W_{\text{Comp},s} = c_p(T_{2s}-T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(585.8-288)\text{K} = 299.3\text{ kJ/kg}$$

$$W_{\text{Comp}} = c_p(T_2-T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(618.9-288)\text{K} = 332.6\text{ kJ/kg}$$

$$W_{\text{Turb},s} = c_p(T_3-T_{4s}) = (1.005\text{ kJ/kg}\cdot\text{K})(873-429.2)\text{K} = 446.0\text{ kJ/kg}$$

$$W_{\text{Turb}} = c_p(T_3-T_4) = (1.005\text{ kJ/kg}\cdot\text{K})(873-473.6)\text{K} = 401.4\text{ kJ/kg}$$

The back work ratio for 90% efficient compressor and isentropic turbine case is

$$r_{\text{bw}} = \frac{W_{\text{Comp}}}{W_{\text{Turb},s}} = \frac{332.6\text{ kJ/kg}}{446.0\text{ kJ/kg}} = \mathbf{0.7457}$$

The back work ratio for 90% efficient turbine and isentropic compressor case is

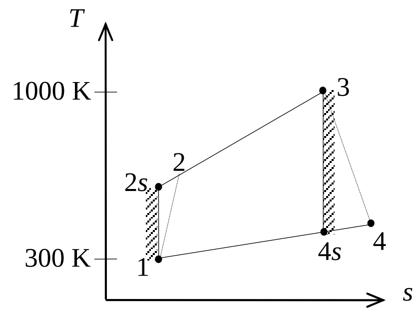
$$r_{\text{bw}} = \frac{W_{\text{Comp},s}}{W_{\text{Turb}}} = \frac{299.3\text{ kJ/kg}}{401.4\text{ kJ/kg}} = \mathbf{0.7456}$$

The two results are almost identical.

9-95 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,

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

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

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

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

$$w_{\text{s,net,out}} = w_{\text{s,T,out}} - w_{\text{s,C,in}} = 510.84 - 311.75 = 199.1\text{ kJ/kg}$$

$$\dot{m}_s = \frac{\dot{W}_{\text{net,out}}}{w_{\text{s,net,out}}} = \frac{70{,}000\text{ kJ/s}}{199.1\text{ kJ/kg}} = \mathbf{352\ \ kg/s}$$

(b) The net work output is determined to be

$$w_{\text{a,net,out}} = w_{\text{a,T,out}} - w_{\text{a,C,in}} = \eta_T w_{\text{s,T,out}} - w_{\text{s,C,in}}/\eta_C$$
$$= (0.85)(510.84) - 311.75/0.85 = 67.5\text{ kJ/kg}$$

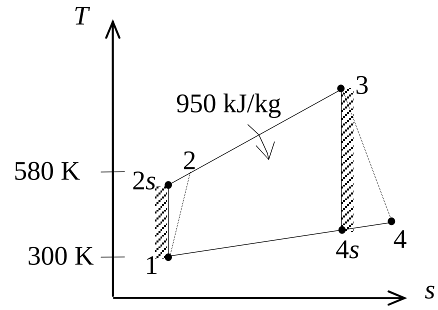
$$\dot{m}_a = \frac{\dot{W}_{\text{net,out}}}{w_{\text{a,net,out}}} = \frac{70{,}000\text{ kJ/s}}{67.5\text{ kJ/kg}} = \mathbf{1037\ \ kg/s}$$

9-96 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-17.

Analysis (a) Using the isentropic relations,



$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.19\text{ kJ/kg}$$

$$T_2 = 580\text{ K} \longrightarrow h_2 = 586.04\text{ kJ/kg}$$

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

$$q_{in} = h_3 - h_2 \longrightarrow h_3 = 950 + 586.04 = 1536.04\text{kJ/kg}$$

$$\to P_{r_3} = 474.11$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{7}\right)(474.11) = 67.73 \longrightarrow h_{4s} = 905.83\text{ kJ/kg}$$

$$w_{\text{C,in}} = h_2 - h_1 = 586.04 - 300.19 = 285.85\text{ kJ/kg}$$

$$w_{\text{T,out}} = \eta_T (h_3 - h_{4s}) = (0.86)(1536.04 - 905.83) = 542.0\text{ kJ/kg}$$

Thus,

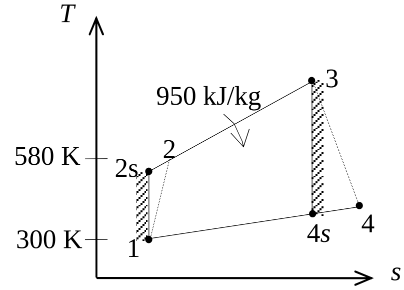
$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{285.85\text{ kJ/kg}}{542.0\text{ kJ/kg}} = \mathbf{52.7\%}$$

(b) $$w_{\text{net.out}} = w_{\text{T,out}} - w_{\text{C,in}} = 542.0 - 285.85 = 256.15\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{256.15\text{ kJ/kg}}{950\text{ kJ/kg}} = \mathbf{27.0\%}$$

9-97 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,

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

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) \longrightarrow T_3 = T_2 + q_{in}/c_p$$
$$= 580\text{ K} + (950\text{ kJ/kg})/(1.005\text{ kJ/kg}\cdot\text{K})$$
$$= 1525.3\text{ K}$$

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

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

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

Thus,

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

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

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

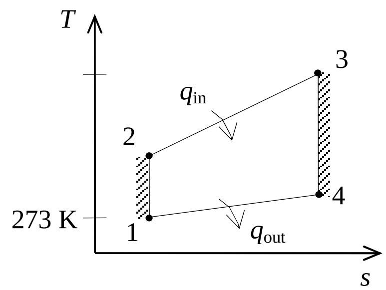
9-98 An aircraft engine operates as a simple ideal Brayton cycle with air as the working fluid. The pressure ratio and the rate of heat input are given. The net power and the thermal efficiency are to be determined.

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-2a).

Analysis For the isentropic compression process,

$$T_2 = T_1 r_p^{(k-1)/k} = (273\text{ K})(10)^{0.4/1.4} = 527.1\text{ K}$$



The heat addition is

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = \frac{500\text{ kW}}{1\text{ kg/s}} = 500\text{ kJ/kg}$$

Applying the first law to the heat addition process,

$$q_{in} = c_p(T_3 - T_2)$$

$$T_3 = T_2 + \frac{q_{in}}{c_p} = 527.1\text{ K} + \frac{500\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 1025\text{ K}$$

The temperature at the exit of the turbine is

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1025\text{ K})\left(\frac{1}{10}\right)^{0.4/1.4} = 530.9\text{ K}$$

Applying the first law to the adiabatic turbine and the compressor produce

$$w_T = c_p(T_3 - T_4) = (1.005\text{ kJ/kg}\cdot\text{K})(1025 - 530.9)\text{K} = 496.6\text{ kJ/kg}$$

$$w_C = c_p(T_2 - T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(527.1 - 273)\text{K} = 255.4\text{ kJ/kg}$$

The net power produced by the engine is then

$$\dot{W}_{net} = \dot{m}(w_T - w_C) = (1\text{ kg/s})(496.6 - 255.4)\text{kJ/kg} = \mathbf{241.2\text{ kW}}$$

Finally the thermal efficiency is

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{241.2\text{ kW}}{500\text{ kW}} = \mathbf{0.482}$$

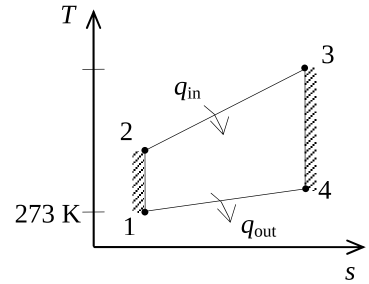
9-99 An aircraft engine operates as a simple ideal Brayton cycle with air as the working fluid. The pressure ratio and the rate of heat input are given. The net power and the thermal efficiency are to be determined.

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-2a).

Analysis For the isentropic compression process,

$$T_2 = T_1 r_p^{(k-1)/k} = (273\text{ K})(15)^{0.4/1.4} = 591.8\text{ K}$$



The heat addition is

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = \frac{500\text{ kW}}{1\text{ kg/s}} = 500\text{ kJ/kg}$$

Applying the first law to the heat addition process,

$$q_{in} = c_p(T_3 - T_2)$$

$$T_3 = T_2 + \frac{q_{in}}{c_p} = 591.8\text{ K} + \frac{500\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 1089\text{ K}$$

The temperature at the exit of the turbine is

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1089\text{ K})\left(\frac{1}{15}\right)^{0.4/1.4} = 502.3\text{ K}$$

Applying the first law to the adiabatic turbine and the compressor produce

$$w_T = c_p(T_3 - T_4) = (1.005\text{ kJ/kg}\cdot\text{K})(1089 - 502.3)\text{K} = 589.6\text{ kJ/kg}$$

$$w_C = c_p(T_2 - T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(591.8 - 273)\text{K} = 320.4\text{ kJ/kg}$$

The net power produced by the engine is then

$$\dot{W}_{net} = \dot{m}(w_T - w_C) = (1\text{ kg/s})(589.6 - 320.4)\text{kJ/kg} = \mathbf{269.2\text{ kW}}$$

Finally the thermal efficiency is

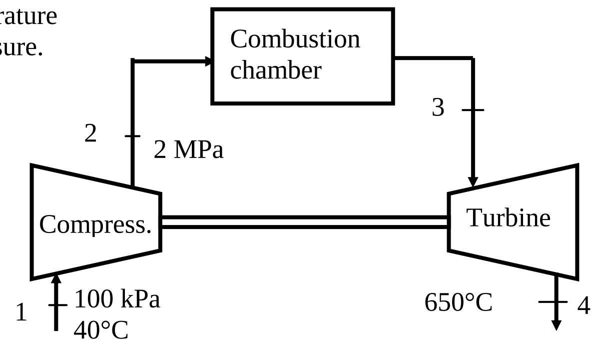
$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{269.2\text{ kW}}{500\text{ kW}} = \mathbf{0.538}$$

9-100 A gas-turbine plant operates on the simple Brayton cycle. The net power output, the back work ratio, 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 gas constant of air is R = 0.287 kJ/kg·K (Table A-1).

Analysis (a) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.



Process 1-2: Compression

$$T_1 = 40°\text{C} \longrightarrow h_1 = 313.6 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 40°\text{C} \\ P_1 = 100 \text{ kPa} \end{array}\right\} s_1 = 5.749 \text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 2000 \text{ kPa} \\ s_2 = s_1 = 5.749 \text{ kJ/kg.K} \end{array}\right\} h_{2s} = 736.7 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.85 = \frac{736.7 - 313.6}{h_2 - 313.6} \longrightarrow h_2 = 811.4 \text{ kJ/kg}$$

Process 3-4: Expansion

$$T_4 = 650°\text{C} \longrightarrow h_4 = 959.2 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.88 = \frac{h_3 - 959.2}{h_3 - h_{4s}}$$

We cannot find the enthalpy at state 3 directly. However, using the following lines in EES together with the isentropic efficiency relation, we find h_3 = 1873 kJ/kg, T_3 = 1421ºC, s_3 = 6.736 kJ/kg.K. The solution by hand would require a trial-error approach.

```
h_3=enthalpy(Air, T=T_3)
s_3=entropy(Air, T=T_3, P=P_2)
h_4s=enthalpy(Air, P=P_1, s=s_3)
```

The mass flow rate is determined from

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(100 \text{ kPa})(700/60 \text{ m}^3/\text{s})}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(40 + 273 \text{ K})} = 12.99 \text{ kg/s}$$

The net power output is

$$\dot{W}_{\text{C,in}} = \dot{m}(h_2 - h_1) = (12.99 \text{ kg/s})(811.4 - 313.6)\text{kJ/kg} = 6464 \text{ kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}(h_3 - h_4) = (12.99 \text{ kg/s})(1873 - 959.2)\text{kJ/kg} = 11{,}868 \text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{C,in}} = 11{,}868 - 6464 = \mathbf{5404\ kW}$$

(b) The back work ratio is

$$r_{\text{bw}} = \frac{\dot{W}_{\text{C,in}}}{\dot{W}_{\text{T,out}}} = \frac{6464 \text{ kW}}{11{,}868 \text{ kW}} = \mathbf{0.545}$$

(c) The rate of heat input and the thermal efficiency are

$$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (12.99 \text{ kg/s})(1873 - 811.4)\text{kJ/kg} = 13{,}788 \text{ kW}$$

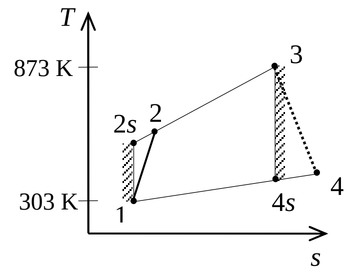
$$\eta_{th} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{5404 \text{ kW}}{13{,}788 \text{ kW}} = 0.392 = \mathbf{39.2\%}$$

9-101 A simple Brayton cycle with air as the working fluid operates between the specified temperature and pressure limits. The cycle is to be sketched on the *T*-*s* cycle and the isentropic efficiency of the turbine and the cycle 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 are given as c_v = 0.718 kJ/kg·K, c_p = 1.005 kJ/kg·K, R = 0.287 kJ/kg·K, k = 1.4.

Analysis (b) For the compression process,



$$\begin{aligned}\dot{W}_{\text{Comp}} &= \dot{m}c_p(T_2 - T_1)\\ &= (200\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(330 - 30)\text{K}\\ &= 60{,}300\text{ kW}\end{aligned}$$

For the turbine during the isentropic process,

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{100\text{ kPa}}{800\text{ kPa}}\right)^{0.4/1.4} = 772.9\text{ K}$$

$$\dot{W}_{\text{Turb,s}} = \dot{m}c_p(T_3 - T_{4s}) = (200\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400 - 772.9)\text{K} = 126{,}050\text{ kW}$$

The actual power output from the turbine is

$$\dot{W}_{\text{net}} = \dot{W}_{\text{Turb}} - \dot{W}_{\text{Comp}}$$

$$\dot{W}_{\text{Turb}} = \dot{W}_{\text{net}} + \dot{W}_{\text{Turb}} = 60{,}000 + 60{,}300 = 120{,}300\text{ kW}$$

The isentropic efficiency of the turbine is then

$$\eta_{\text{Turb}} = \frac{\dot{W}_{\text{Turb}}}{\dot{W}_{\text{Turb,s}}} = \frac{120{,}300\text{ kW}}{126{,}050\text{ kW}} = 0.954 = \mathbf{95.4\%}$$

(c) The rate of heat input is

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_3 - T_2) = (200\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})[(1400 - (330 + 273)]\text{K} = 160{,}200\text{ kW}$$

The thermal efficiency is then

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{60{,}000\text{ kW}}{160{,}200\text{ kW}} = 0.375 = \mathbf{37.5\%}$$

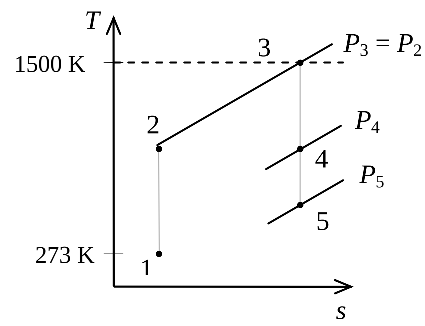
9-102 A modified Brayton cycle with air as the working fluid operates at a specified pressure ratio. The *T*-*s* diagram is to be sketched and the temperature and pressure at the exit of the high-pressure turbine and the mass flow rate of air are to be determined.

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 are given as c_v = 0.718 kJ/kg·K, c_p = 1.005 kJ/kg·K, R = 0.287 kJ/kg·K, k = 1.4.

Analysis (b) For the compression process,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (273\text{ K})(8)^{0.4/1.4} = 494.5\text{ K}$$



The power input to the compressor is equal to the power output from the high-pressure turbine. Then,

$$\dot{W}_{\text{Comp,in}} = \dot{W}_{\text{HP Turb,out}}$$
$$\dot{m}c_p(T_2 - T_1) = \dot{m}c_p(T_3 - T_4)$$
$$T_2 - T_1 = T_3 - T_4$$
$$T_4 = T_3 + T_1 - T_2 = 1500 + 273 - 494.5 = \mathbf{1278.5\ K}$$

The pressure at this state is

$$\frac{P_4}{P_3} = \left(\frac{T_4}{T_3}\right)^{k/(k-1)} \longrightarrow P_4 = rP_1\left(\frac{T_4}{T_3}\right)^{k/(k-1)} = 8(100\text{ kPa})\left(\frac{1278.5\text{ K}}{1500\text{ K}}\right)^{1.4/0.4} = \mathbf{457.3\ kPa}$$

(c) The temperature at state 5 is determined from

$$T_5 = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = (1278.5\text{ K})\left(\frac{100\text{ kPa}}{457.3\text{ kPa}}\right)^{0.4/1.4} = 828.1\text{ K}$$

The net power is that generated by the low-pressure turbine since the power output from the high-pressure turbine is equal to the power input to the compressor. Then,

$$\dot{W}_{\text{LP Turb}} = \dot{m}c_p(T_4 - T_5)$$
$$\dot{m} = \frac{\dot{W}_{\text{LP Turb}}}{c_p(T_4 - T_5)} = \frac{200{,}000\text{ kW}}{(1.005\text{ kJ/kg}\cdot\text{K})(1278.5 - 828.1)\text{K}} = \mathbf{441.8\ kg/s}$$

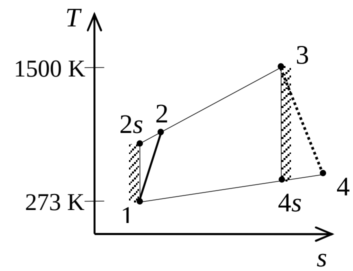
9-103 A simple Brayton cycle with air as the working fluid operates at a specified pressure ratio and between the specified temperature and pressure limits. The cycle is to be sketched on the *T-s* cycle and the volume flow rate of the air into the compressor is to be determined. Also, the effect of compressor inlet temperature on the mass flow rate and the net power output are to be investigated.

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 are given as c_v = 0.718 kJ/kg·K, c_p = 1.005 kJ/kg·K, R = 0.287 kJ/kg·K, k = 1.4.

Analysis (b) For the compression process,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (273\text{ K})(7)^{0.4/1.4} = 476.0\text{ K}$$



$$\eta_{\text{Comp}} = \frac{\dot{W}_{\text{Comp,s}}}{\dot{W}_{\text{Comp}}} = \frac{\dot{m}c_p(T_{2s}-T_1)}{\dot{m}c_p(T_2-T_1)} = \frac{T_{2s}-T_1}{T_2-T_1}$$

$$0.80 = \frac{476.0-273}{T_2-273} \longrightarrow T_2 = 526.8\text{ K}$$

For the expansion process,

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1500\text{ K})\left(\frac{1}{7}\right)^{0.4/1.4} = 860.3\text{ K}$$

$$\eta_{\text{Turb}} = \frac{\dot{W}_{\text{Turb}}}{\dot{W}_{\text{Turb,s}}} = \frac{\dot{m}c_p(T_3-T_4)}{\dot{m}c_p(T_3-T_{4s})} = \frac{T_3-T_4}{T_3-T_{4s}}$$

$$0.90 = \frac{1500-T_4}{1500-860.3} \longrightarrow T_4 = 924.3\text{ K}$$

Given the net power, the mass flow rate is determined from

$$\dot{W}_{\text{net}} = \dot{W}_{\text{Turb}} - \dot{W}_{\text{Comp}} = \dot{m}c_p(T_3-T_4) - \dot{m}c_p(T_2-T_1)$$

$$\dot{W}_{\text{net}} = \dot{m}c_p\left[(T_3-T_4)-(T_2-T_1)\right]$$

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{c_p\left[(T_3-T_4)-(T_2-T_1)\right]}$$

$$= \frac{150{,}000\text{ kW}}{(1.005\text{ kJ/kg}\cdot\text{K})\left[(1500-924.3)-(526.8-273)\right]}$$

$$= 463.7\text{ kg/s}$$

The specific volume and the volume flow rate at the inlet of the compressor are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(273\text{ K})}{100\text{ kPa}} = 0.7835\text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}\boldsymbol{v}_1 = (463.7\text{ kg/s})(0.7835\text{ m}^3/\text{kg}) = \mathbf{363.2\ m^3/s}$$

(c) For a fixed compressor inlet velocity and flow area, when the compressor inlet temperature increases, the specific volume increases since $\boldsymbol{v} = \frac{RT}{P}$. When specific volume increases, the mass flow rate decreases since $\dot{m} = \frac{\dot{V}}{\boldsymbol{v}}$. Note that volume flow rate is the same since inlet velocity and flow area are fixed ($\dot{V} = AV$). When mass flow rate decreases, the net power decreases since $\dot{W}_{\text{net}} = \dot{m}(w_{\text{Turb}} - w_{\text{Comp}})$. Therefore, when the inlet temperature increases, both mass flow rate and the net power decrease.

Brayton Cycle with Regeneration

9-104C 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.

9-105C 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.

9-106C The extent to which a regenerator approaches an ideal regenerator is called the effectiveness ε, and is defined as $\varepsilon = q_{\text{regen, act}} / q_{\text{regen, max}}$.

9-107C (b) turbine exit.

9-108C The steam injected increases the mass flow rate through the turbine and thus the power output. This, in turn, increases the thermal efficiency since $\eta = W / Q_{\text{in}}$ and W increases while Q_{in} remains constant. Steam can be obtained by utilizing the hot exhaust gases.

9-109 A Brayton cycle with regeneration produces 150 kW power. The rates of heat addition and rejection are to be determined.

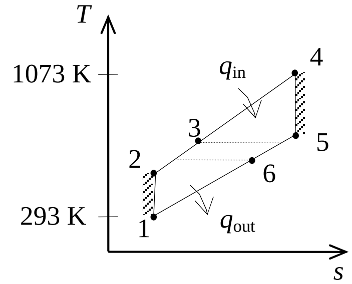
Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

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

Analysis According to the isentropic process expressions for an ideal gas,

$$T_2 = T_1 r_p^{(k-1)/k} = (293\text{ K})(8)^{0.4/1.4} = 530.8\text{ K}$$

$$T_5 = T_4\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1073\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 592.3\text{ K}$$



When the first law is applied to the heat exchanger, the result is

$$T_3 - T_2 = T_5 - T_6$$

while the regenerator temperature specification gives

$$T_3 = T_5 - 10 = 592.3 - 10 = 582.3\text{ K}$$

The simultaneous solution of these two results gives

$$T_6 = T_5 - (T_3 - T_2) = 592.3 - (582.3 - 530.8) = 540.8\text{ K}$$

Application of the first law to the turbine and compressor gives

$$\begin{aligned} w_{\text{net}} &= c_p(T_4 - T_5) - c_p(T_2 - T_1) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(1073 - 592.3)\text{ K} - (1.005\text{ kJ/kg}\cdot\text{K})(530.8 - 293)\text{ K} \\ &= 244.1\text{ kJ/kg} \end{aligned}$$

Then,

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150\text{ kW}}{244.1\text{ kJ/kg}} = 0.6145\text{ kg/s}$$

Applying the first law to the combustion chamber produces

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_4 - T_3) = (0.6145\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1073 - 582.3)\text{K} = \mathbf{303.0\text{ kW}}$$

Similarly,

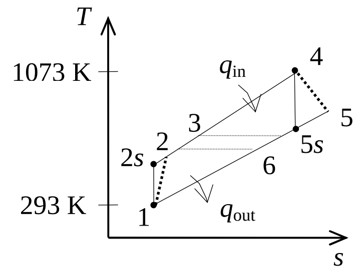
$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_6 - T_1) = (0.6145\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(540.8 - 293)\text{K} = \mathbf{153.0\text{ kW}}$$

9-110 A Brayton cycle with regeneration produces 150 kW power. The rates of heat addition and rejection 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 For the compression and expansion processes we have



$$T_{2s} = T_1 r_p^{(k-1)/k} = (293\text{ K})(8)^{0.4/1.4} = 530.8\text{ K}$$

$$\eta_C = \frac{c_p(T_{2s} - T_1)}{c_p(T_2 - T_1)} \longrightarrow T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_C}$$
$$= 293 + \frac{530.8 - 293}{0.87} = 566.3\text{ K}$$

$$T_{5s} = T_4\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1073\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 592.3\text{ K}$$

$$\eta_T = \frac{c_p(T_4 - T_5)}{c_p(T_4 - T_{5s})} \longrightarrow T_5 = T_4 - \eta_T(T_4 - T_{5s})$$
$$= 1073 - (0.93)(1073 - 592.3)$$
$$= 625.9\text{ K}$$

When the first law is applied to the heat exchanger, the result is

$$T_3 - T_2 = T_5 - T_6$$

while the regenerator temperature specification gives

$$T_3 = T_5 - 10 = 625.9 - 10 = 615.9\text{ K}$$

The simultaneous solution of these two results gives

$$T_6 = T_5 - (T_3 - T_2) = 625.9 - (615.9 - 566.3) = 576.3\text{ K}$$

Application of the first law to the turbine and compressor gives

$$w_{\text{net}} = c_p(T_4 - T_5) - c_p(T_2 - T_1)$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})(1073 - 625.9)\text{ K} - (1.005\text{ kJ/kg}\cdot\text{K})(566.3 - 293)\text{ K}$$
$$= 174.7\text{ kJ/kg}$$

Then,

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150\text{ kW}}{174.7\text{ kJ/kg}} = 0.8586\text{ kg/s}$$

Applying the first law to the combustion chamber produces

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_4 - T_3) = (0.8586\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1073 - 615.9)\text{K} = \mathbf{394.4\text{ kW}}$$

Similarly,

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_6 - T_1) = (0.8586\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(576.3 - 293)\text{K} = \mathbf{244.5\text{ kW}}$$

9-111 A Brayton cycle with regeneration is considered. The thermal efficiencies of the cycle for parallel-flow and counter-flow arrangements of the regenerator are to be compared.

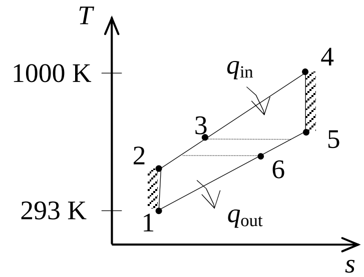
Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

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

Analysis According to the isentropic process expressions for an ideal gas,

$$T_2 = T_1 r_p^{(k-1)/k} = (293\text{ K})(7)^{0.4/1.4} = 510.9\text{ K}$$

$$T_5 = T_4 \left(\frac{1}{r_p}\right)^{(k-1)/k} = (1000\text{ K})\left(\frac{1}{7}\right)^{0.4/1.4} = 573.5\text{ K}$$



When the first law is applied to the heat exchanger as originally arranged, the result is

$$T_3 - T_2 = T_5 - T_6$$

while the regenerator temperature specification gives

$$T_3 = T_5 - 6 = 573.5 - 6 = 567.5\text{ K}$$

The simultaneous solution of these two results gives

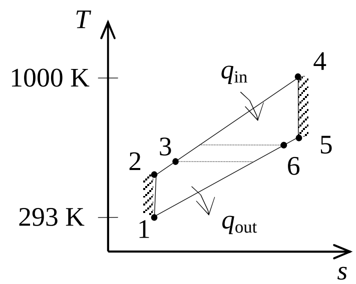
$$T_6 = T_5 - T_3 + T_2 = 573.5 - 567.5 + 510.9 = 516.9\text{ K}$$

The thermal efficiency of the cycle is then

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_6 - T_1}{T_4 - T_3} = 1 - \frac{516.9 - 293}{1000 - 567.5} = \mathbf{0.482}$$

For the rearranged version of this cycle,

$$T_3 = T_6 - 6$$



An energy balance on the heat exchanger gives

$$T_3 - T_2 = T_5 - T_6$$

The solution of these two equations is

$$T_3 = 539.2\text{ K}$$
$$T_6 = 545.2\text{ K}$$

The thermal efficiency of the cycle is then

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_6 - T_1}{T_4 - T_3} = 1 - \frac{545.2 - 293}{1000 - 539.2} = \mathbf{0.453}$$

9-112E An ideal Brayton cycle with regeneration has a pressure ratio of 11. The thermal efficiency of the cycle is to be determined with and without regenerator cases.

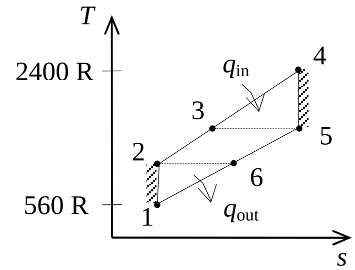
Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis According to the isentropic process expressions for an ideal gas,

$$T_2 = T_1 r_p^{(k-1)/k} = (560\text{ R})(11)^{0.4/1.4} = 1111\text{ R}$$

$$T_5 = T_4\left(\frac{1}{r_p}\right)^{(k-1)/k} = (2400\text{ R})\left(\frac{1}{11}\right)^{0.4/1.4} = 1210\text{ R}$$



The regenerator is ideal (i.e., the effectiveness is 100%) and thus,

$$T_3 = T_5 = 1210\text{ R}$$
$$T_6 = T_2 = 1111\text{ R}$$

The thermal efficiency of the cycle is then

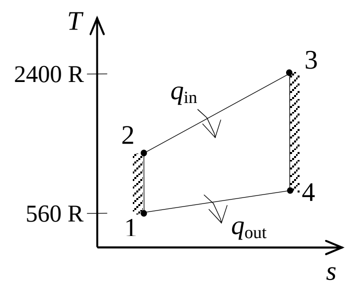
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_6 - T_1}{T_4 - T_3} = 1 - \frac{1111 - 560}{2400 - 1210} = 0.537 = \mathbf{53.7\%}$$

The solution without a regenerator is as follows:

$$T_2 = T_1 r_p^{(k-1)/k} = (560\text{ R})(11)^{0.4/1.4} = 1111\text{ R}$$

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (2400\text{ R})\left(\frac{1}{11}\right)^{0.4/1.4} = 1210\text{ R}$$

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{1210 - 560}{2400 - 1111} = 0.496 = \mathbf{49.6\%}$$



9-113E 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 95 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-17E.

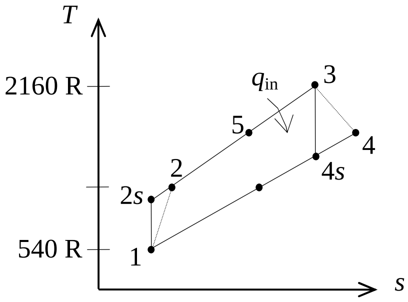
Analysis The gas turbine cycle with regeneration can be analyzed as follows:

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

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

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

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



and

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

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

Then the thermal efficiency of the gas turbine cycle becomes

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

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

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

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{63.0 \text{ Btu/lbm}}{161.7 \text{ Btu/lbm}} = 0.39 = \mathbf{39\%}$$

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

$$\dot{m}_{\text{air}} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{95 \text{ hp}}{63.0 \text{ Btu/lbm}} \left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}}\right) = \mathbf{1.07 \text{ lbm/s}}$$

9-114 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-17.

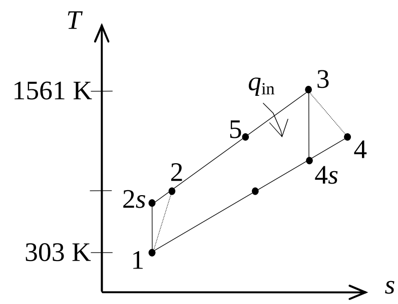
Analysis The properties at various states are

$$T_1 = 30°\text{C} = 303\text{ K} \longrightarrow \begin{array}{l} h_1 = 303.21\text{ kJ/kg} \\ P_{r_1} = 1.4356 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (14.7)(1.4356) = 21.10 \longrightarrow h_{2s} = 653.25\text{ kJ/kg}$$

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

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



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

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

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

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

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

$$q_{\text{out}} = h_4 - h_1 \rightarrow h_4 = q_{\text{out}} + h_1 = 665.34 + 303.21 = 968.55\text{ kJ/kg} \rightarrow T_4 = 931.7\text{ K} = 658.7°\text{C}$$

Then the compressor and turbine efficiencies become

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{1710 - 968.55}{1710 - 825.23} = 0.838 = \mathbf{83.8\%}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{653.25 - 303.21}{672 - 303.21} = 0.949 = \mathbf{94.9\%}$$

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

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = (0.65)(968.55 - 672.0) = 192.8\text{ kJ/kg}$$

$$q_{\text{in,new}} = q_{\text{in}} - q_{\text{regen}} = 1038 - 192.8 = 845.2\text{ kJ/kg}$$

$$\eta_{\text{th,new}} = \frac{w_{\text{net}}}{q_{\text{in,new}}} = \frac{372.66\text{ kJ/kg}}{845.2\text{ kJ/kg}} = 0.441 = \mathbf{44.1\%}$$

Discussion Note a 65% efficient regenerator would increase the thermal efficiency of this gas turbine from 35.9% to 44.1%.

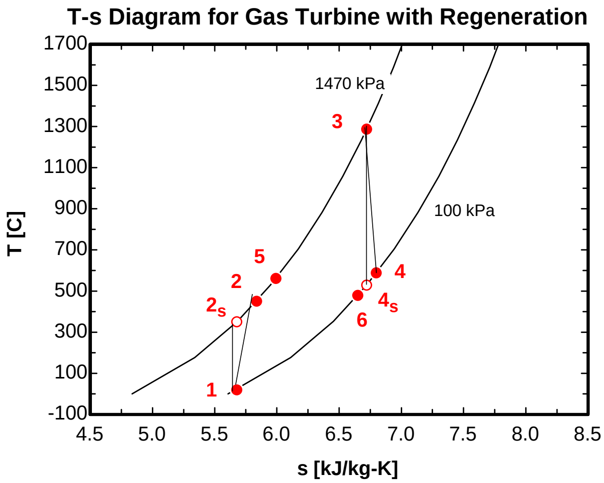
9-115 Problem 9-114 is reconsidered. A solution that allows different isentropic efficiencies for the compressor and turbine is to be developed and the effect of the isentropic efficiencies on net work done and the heat supplied to the cycle is to be studied. Also, the *T-s* diagram for the cycle is to be plotted.

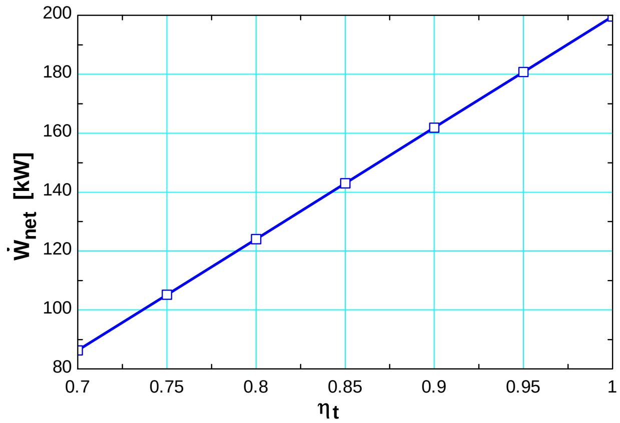
Analysis Using EES, the problem is solved as follows:

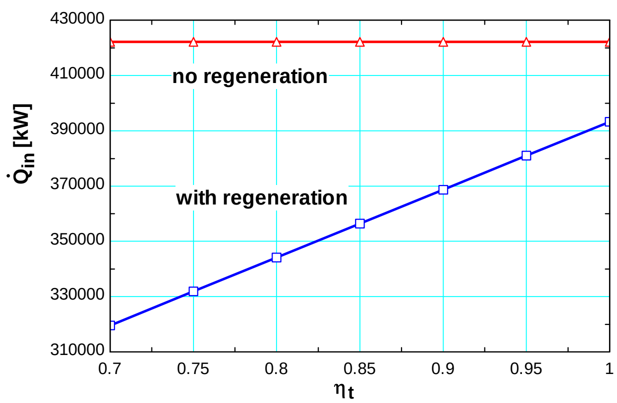
```
"Input data"
T[3] = 1288 [C]
Pratio = 14.7
T[1] = 30 [C]
P[1]= 100 [kPa]
{T[4]=659 [C]}
{W_dot_net=159 [MW] }"We omit the information about the cycle net work"
m_dot = 1536000 [kg/h]*Convert(kg/h,kg/s)
{Eta_th_noreg=0.359} "We omit the information about the cycle efficiency."
Eta_reg = 0.65
Eta_c = 0.84 "Compressor isentropic efficiency"
Eta_t = 0.95 "Turbien isentropic efficiency"

"Isentropic Compressor anaysis"
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])
```

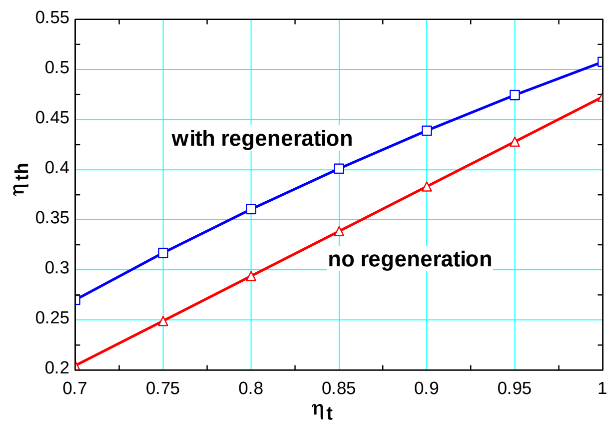
```
"Actual Turbine analysis:"
m_dot*h[3] = W_dot_turb + m_dot*h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])

"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]=T[6]
```

η_t	η_c	$\eta_{th,noreg}$	$\eta_{th,withreg}$	$Q_{innoreg}$ [kW]	$Q_{inwithreg}$ [kW]	W_{net} [kW]
0.7	0.84	0.2044	0.27	422152	319582	86.3
0.75	0.84	0.2491	0.3169	422152	331856	105.2
0.8	0.84	0.2939	0.3605	422152	344129	124.1
0.85	0.84	0.3386	0.4011	422152	356403	142.9
0.9	0.84	0.3833	0.4389	422152	368676	161.8
0.95	0.84	0.4281	0.4744	422152	380950	180.7
1	0.84	0.4728	0.5076	422152	393223	199.6



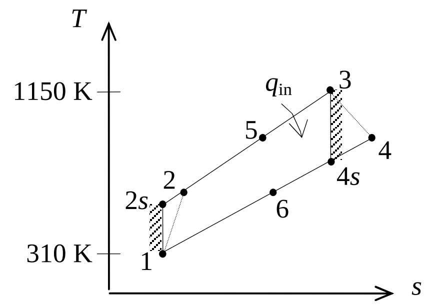






9-116 A Brayton cycle with regeneration using air as the working fluid is considered. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

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-17.

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,

$$T_4 = \mathbf{782.8\text{ K}}$$

(*b*)

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

(*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,

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

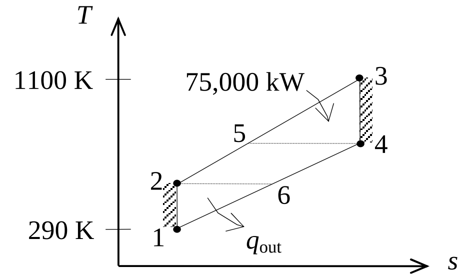
$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{108.09\text{ kJ/kg}}{480.82\text{ kJ/kg}} = \mathbf{22.5\%}$$

9-117 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-17.

Analysis (*a*) Assuming constant specific heats,



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

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

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

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

$$\dot{W}_{\text{net}} = \eta_T \dot{Q}_{\text{in}} = (0.5225)(75{,}000\text{ kW}) = \mathbf{39{,}188\ \ kW}$$

(*b*) Assuming variable specific heats,

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

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

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

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

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

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

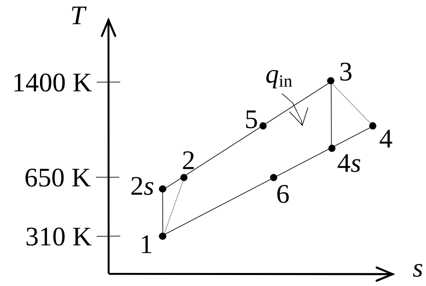
$$\dot{W}_{\text{net}} = \eta_T \dot{Q}_{\text{in}} = (0.5371)(75{,}000\text{ kW}) = \mathbf{40{,}283\ \ kW}$$

9-118 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-17.

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



$$r_p = P_2 / P_1 = 900/100 = 9$$

$$T_1 = 310 \text{ K} \longrightarrow h_1 = 310.24 \text{ kJ/kg}$$
$$T_2 = 650 \text{ K} \longrightarrow h_2 = 659.84 \text{ kJ/kg}$$
$$T_3 = 1400 \text{ K} \longrightarrow h_3 = 1515.42 \text{ kJ/kg}$$
$$P_{r_3} = 450.5$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{9}\right)(450.5) = 50.06 \longrightarrow h_{4s} = 832.44 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 1515.42 - (0.90)(1515.42 - 832.44)$$
$$= 900.74 \text{ kJ/kg}$$

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = (0.80)(900.74 - 659.84) = \mathbf{192.7\ kJ/kg}$$

(*b*) $$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1)$$
$$= (1515.42 - 900.74) - (659.84 - 310.24) = 265.08 \text{ kJ/kg}$$

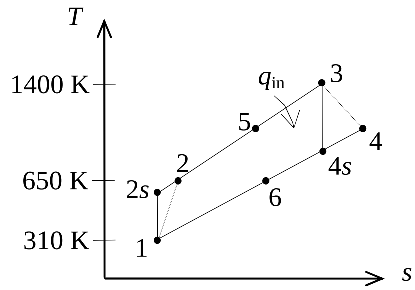
$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = (1515.42 - 659.84) - 192.7 = 662.88 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{265.08 \text{ kJ/kg}}{662.88 \text{ kJ/kg}} = 0.400 = \mathbf{40.0\%}$$

9-119 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,

$$r_p = P_2 / P_1 = 900/100 = 9$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{1}{9}\right)^{0.4/1.4} = 747.3\text{ K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{c_p(T_3 - T_4)}{c_p(T_3 - T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3 - T_{4s})$$
$$= 1400 - (0.90)(1400 - 747.3)$$
$$= 812.6\text{ K}$$

$$q_{regen} = \varepsilon(h_4 - h_2) = \varepsilon c_p(T_4 - T_2) = (0.80)(1.005\text{ kJ/kg}\cdot\text{K})(812.6 - 650)\text{K} = \mathbf{130.7\ kJ/kg}$$

(b)

$$w_{net} = w_{T,out} - w_{C,in} = c_p(T_3 - T_4) - c_p(T_2 - T_1)$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})[(1400 - 812.6) - (650 - 310)]\text{K} = 248.7\text{ kJ/kg}$$

$$q_{in} = (h_3 - h_2) - q_{regen} = c_p(T_3 - T_2) - q_{regen}$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})(1400 - 650)\text{K} - 130.7 = 623.1\text{ kJ/kg}$$

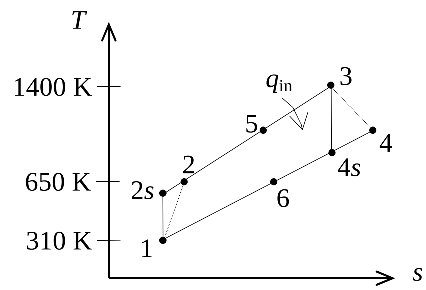
$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{248.7\text{ kJ/kg}}{623.1\text{ kJ/kg}} = 0.399 = \mathbf{39.9\%}$$

9-120 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-17.

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



$$r_p = P_2 / P_1 = 900/100 = 9$$

$$T_1 = 310 \text{ K} \longrightarrow h_1 = 310.24 \text{ kJ/kg}$$

$$T_2 = 650 \text{ K} \longrightarrow h_2 = 659.84 \text{ kJ/kg}$$

$$T_3 = 1400 \text{ K} \longrightarrow h_3 = 1515.42 \text{ kJ/kg}$$

$$P_{r_3} = 450.5$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{9}\right)(450.5) = 50.06 \longrightarrow h_{4s} = 832.44 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 1515.42 - (0.90)(1515.42 - 832.44)$$
$$= 900.74 \text{ kJ/kg}$$

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = (0.70)(900.74 - 659.84) = \mathbf{168.6\ kJ/kg}$$

(*b*)

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1)$$
$$= (1515.42 - 900.74) - (659.84 - 310.24) = 265.08 \text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = (1515.42 - 659.84) - 168.6 = 687.18 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{265.08 \text{ kJ/kg}}{687.18 \text{ kJ/kg}} = 0.386 = \mathbf{38.6\%}$$

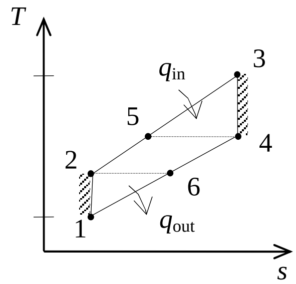
9-121 An expression for the thermal efficiency of an ideal Brayton cycle with an ideal regenerator is to be developed.

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.

Analysis The expressions for the isentropic compression and expansion processes are

$$T_2 = T_1 r_p^{(k-1)/k}$$

$$T_4 = T_3 \left(\frac{1}{r_p} \right)^{(k-1)/k}$$



For an ideal regenerator,

$$T_5 = T_4$$
$$T_6 = T_2$$

The thermal efficiency of the cycle is

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_6 - T_1}{T_3 - T_5} = 1 - \frac{T_1}{T_3} \frac{(T_6 / T_1) - 1}{1 - (T_5 / T_3)}$$

$$= 1 - \frac{T_1}{T_3} \frac{(T_2 / T_1) - 1}{1 - (T_4 / T_3)}$$

$$= 1 - \frac{T_1}{T_3} \frac{r_p^{(k-1)/k} - 1}{1 - r_p^{-(k-1)/k}}$$

$$= 1 - \frac{T_1}{T_3} r_p^{(k-1)/k}$$

Brayton Cycle with Intercooling, Reheating, and Regeneration

9-122C As the number of compression and expansion stages are increased and regeneration is employed, the ideal Brayton cycle will approach the Ericsson cycle.

9-123C Because the steady-flow work is proportional to the specific volume of the gas. Intercooling decreases the average specific volume of the gas during compression, and thus the compressor work. Reheating increases the average specific volume of the gas, and thus the turbine work output.

9-124C (a) decrease, (b) decrease, and (c) decrease.

9-125C (a) increase, (b) decrease, and (c) decrease.

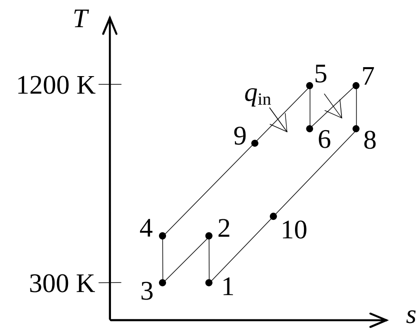
9-126C (a) increase, (b) decrease, (c) decrease, and (d) increase.

9-127C (a) increase, (b) decrease, (c) increase, and (d) decrease.

9-128C (c) The Carnot (or Ericsson) cycle efficiency.

9-129 An ideal gas-turbine cycle with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency of the cycle are to be determined for the cases of with and without a regenerator.

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-17.

Analysis (*a*) The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine since this is an ideal cycle. Then,

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

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_2 = h_4 = 411.26\text{ kJ/kg}$$

$$T_5 = 1200\text{ K} \longrightarrow \begin{aligned} h_5 &= h_7 = 1277.79\text{ kJ/kg} \\ P_{r_5} &= 238 \end{aligned}$$

$$P_{r_6} = \frac{P_6}{P_5} P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_6 = h_8 = 946.36\text{ kJ/kg}$$

$$w_{C,in} = 2(h_2 - h_1) = 2(411.26 - 300.19) = 222.14\text{ kJ/kg}$$

$$w_{T,out} = 2(h_5 - h_6) = 2(1277.79 - 946.36) = 662.86\text{ kJ/kg}$$

Thus,

$$r_{bw} = \frac{w_{C,in}}{w_{T,out}} = \frac{222.14\text{ kJ/kg}}{662.86\text{ kJ/kg}} = \mathbf{33.5\%}$$

$$q_{in} = (h_5 - h_4) + (h_7 - h_6) = (1277.79 - 411.26) + (1277.79 - 946.36) = 1197.96\text{ kJ/kg}$$

$$w_{net} = w_{T,out} - w_{C,in} = 662.86 - 222.14 = 440.72\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{440.72\text{ kJ/kg}}{1197.96\text{ kJ/kg}} = \mathbf{36.8\%}$$

(*b*) When a regenerator is used, r_{bw} remains the same. The thermal efficiency in this case becomes

$$q_{regen} = \varepsilon(h_8 - h_4) = (0.75)(946.36 - 411.26) = 401.33\text{ kJ/kg}$$

$$q_{in} = q_{in,old} - q_{regen} = 1197.96 - 401.33 = 796.63\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{440.72\text{ kJ/kg}}{796.63\text{ kJ/kg}} = \mathbf{55.3\%}$$

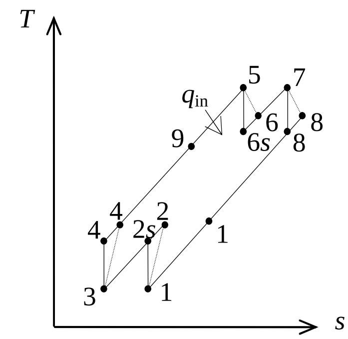
9-130 A gas-turbine cycle with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency of the cycle are to be determined for the cases of with and without a regenerator.

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-17.

Analysis (*a*) The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine. Then,

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



$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_{2s} = h_{4s} = 411.26\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} \longrightarrow h_2 = h_4 = h_1 + (h_{2s}-h_1)/\eta_C$$
$$= 300.19 + (411.26-300.19)/(0.84)$$
$$= 432.42\text{ kJ/kg}$$

$$T_5 = 1200\text{ K} \longrightarrow h_5 = h_7 = 1277.79\text{ kJ/kg}$$
$$P_{r_5} = 238$$

$$P_{r_6} = \frac{P_6}{P_5}P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_{6s} = h_{8s} = 946.36\text{ kJ/kg}$$

$$\eta_T = \frac{h_5-h_6}{h_5-h_{6s}} \longrightarrow h_6 = h_8 = h_5 - \eta_T(h_5-h_{6s})$$
$$= 1277.79-(0.88)(1277.79-946.36)$$
$$= 986.13\text{ kJ/kg}$$

$$w_{\text{C,in}} = 2(h_2-h_1) = 2(432.42-300.19) = 264.46\text{ kJ/kg}$$
$$w_{\text{T,out}} = 2(h_5-h_6) = 2(1277.79-986.13) = 583.32\text{ kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{264.46\text{ kJ/kg}}{583.32\text{ kJ/kg}} = 0.453 = \mathbf{45.3\%}$$

$$q_{\text{in}} = (h_5-h_4)+(h_7-h_6) = (1277.79-432.42)+(1277.79-986.13) = 1137.03\text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 583.32-264.46 = 318.86\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{318.86\text{ kJ/kg}}{1137.03\text{ kJ/kg}} = 0.280 = \mathbf{28.0\%}$$

(*b*) When a regenerator is used, r_{bw} remains the same. The thermal efficiency in this case becomes

$$q_{\text{regen}} = \varepsilon(h_8-h_4) = (0.75)(986.13-432.42) = 415.28\text{ kJ/kg}$$
$$q_{\text{in}} = q_{\text{in,old}} - q_{\text{regen}} = 1137.03-415.28 = 721.75\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{318.86\text{ kJ/kg}}{721.75\text{ kJ/kg}} = 0.442 = \mathbf{44.2\%}$$

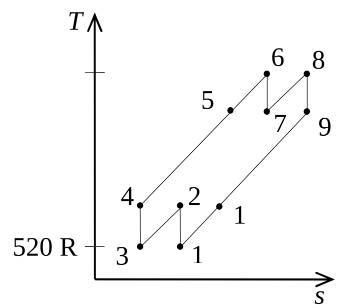
9-131E An ideal regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The power produced and consumed by each compression and expansion stage, and the rate of heat rejected are to be determined.

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 = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis The pressure ratio for each stage is

$$r_p = \sqrt{12} = 3.464$$



According to the isentropic process expressions for an ideal gas,

$$T_2 = T_4 = T_1 r_p^{(k-1)/k} = (520\text{ R})(3.464)^{0.4/1.4} = 741.6\text{ R}$$

Since this is an ideal cycle,

$$T_5 = T_7 = T_9 = T_4 + 50 = 741.6 + 50 = 791.6\text{ R}$$

For the isentropic expansion processes,

$$T_6 = T_8 = T_7 r_p^{(k-1)/k} = (791.6\text{ R})(3.464)^{0.4/1.4} = 1129\text{ R}$$

The heat input is

$$q_{\text{in}} = 2c_p(T_6 - T_5) = 2(0.24\text{ Btu/lbm}\cdot\text{R})(1129 - 791.6)\text{ R} = 162.0\text{ Btu/lbm}$$

The mass flow rate is then

$$\dot{m} = \frac{\dot{Q}_{\text{in}}}{q_{\text{in}}} = \frac{500\text{ Btu/s}}{162.0\text{ Btu/lbm}} = 3.086\text{ lbm/s}$$

Application of the first law to the expansion process 6-7 gives

$$\begin{aligned}\dot{W}_{\text{6-7,out}} &= \dot{m}c_p(T_6 - T_7)\\ &= (3.086\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(1129 - 791.6)\text{ R}\left(\frac{1\text{ kW}}{0.94782\text{ Btu/s}}\right)\\ &= \mathbf{263.6\ kW}\end{aligned}$$

The same amount of power is produced in process 8-9. When the first law is adapted to the compression process 1-2 it becomes

$$\begin{aligned}\dot{W}_{\text{12,in}} &= \dot{m}c_p(T_2 - T_1)\\ &= (3.086\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(741.6 - 520)\text{ R}\left(\frac{1\text{ kW}}{0.94782\text{ Btu/s}}\right)\\ &= \mathbf{173.2\ kW}\end{aligned}$$

Compression process 3-4 uses the same amount of power. The rate of heat rejection from the cycle is

$$\begin{aligned}\dot{Q}_{\text{out}} &= 2\dot{m}c_p(T_2 - T_3)\\ &= 2(3.086\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(741.6 - 520)\text{ R}\\ &= \mathbf{328.3\ Btu/s}\end{aligned}$$

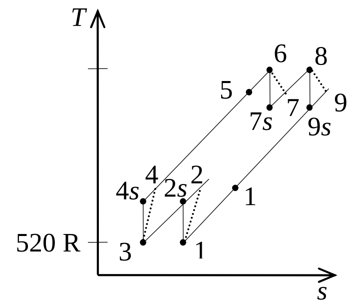
9-132E An ideal regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The power produced and consumed by each compression and expansion stage, and the rate of heat rejected are to be determined.

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 = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis The pressure ratio for each stage is

$$r_p = \sqrt{12} = 3.464$$



For the compression processes,

$$T_{2s} = T_{4s} = T_1 r_p^{(k-1)/k} = (520\text{ R})(3.464)^{0.4/1.4} = 741.6\text{ R}$$

$$\eta_C = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_2 = T_4 = T_1 + \frac{T_{2s}-T_1}{\eta_C}$$

$$= 520 + \frac{741.6-520}{0.85} = 780.7\text{ R}$$

Since the regenerator is ideal,

$$T_5 = T_7 = T_9 = T_4 + 50 = 780.7 + 50 = 830.7\text{ R}$$

For the expansion processes,

$$T_{6s} = T_{8s} = T_7 r_p^{(k-1)/k} = (830.7\text{ R})(3.464)^{0.4/1.4} = 1185\text{ R}$$

$$\eta_T = \frac{c_p(T_6-T_7)}{c_p(T_{6s}-T_7)} \longrightarrow T_6 = T_8 = T_7 + \eta_T(T_{6s}-T_7) = 830.7 + (0.90)(1185-830.7) = 1150\text{ R}$$

The heat input is

$$q_{\text{in}} = 2c_p(T_6 - T_5) = 2(0.24\text{ Btu/lbm}\cdot\text{R})(1150-830.7)\text{ R} = 153.3\text{ Btu/lbm}$$

The mass flow rate is then

$$\dot{m} = \frac{\dot{Q}_{\text{in}}}{q_{\text{in}}} = \frac{500\text{ Btu/s}}{153.3\text{ Btu/lbm}} = 3.262\text{ lbm/s}$$

Application of the first law to the expansion process 6-7 gives

$$\dot{W}_{\text{6-7,out}} = \dot{m}c_p(T_6 - T_7)$$

$$= (3.262\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(1150-830.7)\text{ R}\left(\frac{1\text{ kW}}{0.94782\text{ Btu/s}}\right) = \mathbf{263.7\ kW}$$

The same amount of power is produced in process 8-9. When the first law is adapted to the compression process 1-2 it becomes

$$\dot{W}_{\text{12,in}} = \dot{m}c_p(T_2 - T_1)$$

$$= (3.262\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(780.7-520)\text{ R}\left(\frac{1\text{ kW}}{0.94782\text{ Btu/s}}\right) = \mathbf{215.3\ kW}$$

Compression process 3-4 uses the same amount of power. The rate of heat rejection from the cycle is

$$\dot{Q}_{\text{out}} = 2\dot{m}c_p(T_2 - T_3)$$

$$= 2(3.262\text{ lbm/s})(0.24\text{ Btu/lbm}\cdot\text{R})(780.7-520)\text{ R} = \mathbf{408.2\ Btu/s}$$

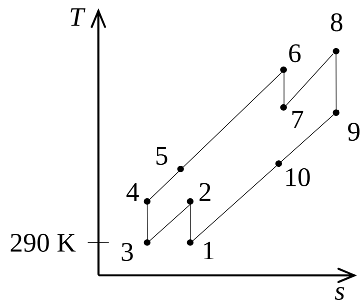
9-133 A regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The thermal efficiency of the cycle is to be determined.

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 The temperatures at various states are obtained as follows

$$T_2 = T_4 = T_1 r_p^{(k-1)/k} = (290\text{ K})(4)^{0.4/1.4} = 430.9\text{ K}$$



$$T_5 = T_4 + 20 = 430.9 + 20 = 450.9\text{ K}$$

$$q_{\text{in}} = c_p(T_6 - T_5)$$

$$T_6 = T_5 + \frac{q_{\text{in}}}{c_p} = 450.9\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 749.4\text{ K}$$

$$T_7 = T_6\left(\frac{1}{r_p}\right)^{(k-1)/k} = (749.4\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 504.3\text{ K}$$

$$T_8 = T_7 + \frac{q_{\text{in}}}{c_p} = 504.3\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 802.8\text{ K}$$

$$T_9 = T_8\left(\frac{1}{r_p}\right)^{(k-1)/k} = (802.8\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 540.2\text{ K}$$

$$T_{10} = T_9 - 20 = 540.2 - 20 = 520.2\text{ K}$$

The heat input is

$$q_{\text{in}} = 300 + 300 = 600\text{ kJ/kg}$$

The heat rejected is

$$\begin{aligned} q_{\text{out}} &= c_p(T_{10} - T_1) + c_p(T_2 - T_3) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(520.2 - 290 + 430.9 - 290)\text{ R} \\ &= 373.0\text{ kJ/kg} \end{aligned}$$

The thermal efficiency of the cycle is then

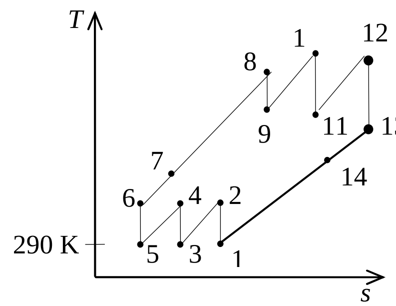
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{373.0}{600} = \mathbf{0.378}$$

9-134 A regenerative gas-turbine cycle with three stages of compression and three stages of expansion is considered. The thermal efficiency of the cycle is 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 The temperatures at various states are obtained as follows



$$T_2 = T_4 = T_6 = T_1 r_p^{(k-1)/k} = (290\text{ K})(4)^{0.4/1.4} = 430.9\text{ K}$$

$$T_7 = T_6 + 20 = 430.9 + 20 = 450.9\text{ K}$$

$$q_{\text{in}} = c_p (T_8 - T_7)$$

$$T_8 = T_7 + \frac{q_{\text{in}}}{c_p} = 450.9\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 749.4\text{ K}$$

$$T_9 = T_8 \left(\frac{1}{r_p}\right)^{(k-1)/k} = (749.4\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 504.3\text{ K}$$

$$T_{10} = T_9 + \frac{q_{\text{in}}}{c_p} = 504.3\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 802.8\text{ K}$$

$$T_{11} = T_{10} \left(\frac{1}{r_p}\right)^{(k-1)/k} = (802.8\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 540.2\text{ K}$$

$$T_{12} = T_{11} + \frac{q_{\text{in}}}{c_p} = 540.2\text{ K} + \frac{300\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 838.7\text{ K}$$

$$T_{13} = T_{12} \left(\frac{1}{r_p}\right)^{(k-1)/k} = (838.7\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 564.4\text{ K}$$

$$T_{14} = T_{13} - 20 = 564.4 - 20 = 544.4\text{ K}$$

The heat input is

$$q_{\text{in}} = 300 + 300 + 300 = 900\text{ kJ/kg}$$

The heat rejected is

$$\begin{aligned} q_{\text{out}} &= c_p (T_{14} - T_1) + c_p (T_2 - T_3) + c_p (T_4 - T_5) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(544.4 - 290 + 430.9 - 290 + 430.9 - 290)\text{ R} \\ &= 538.9\text{ kJ/kg} \end{aligned}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{538.9}{900} = 0.401 = \mathbf{40.1\%}$$

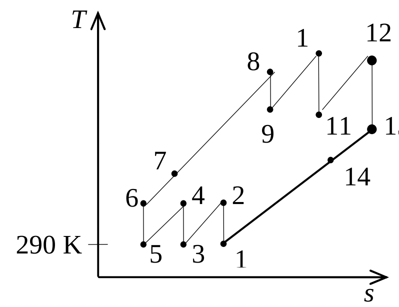
9-135 A regenerative gas-turbine cycle with three stages of compression and three stages of expansion is considered. The thermal efficiency of the cycle is to be determined.

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 Since all compressors share the same compression ratio and begin at the same temperature,

$$T_2 = T_4 = T_6 = T_1 r_p^{(k-1)/k} = (290\text{ K})(4)^{0.4/1.4} = 430.9\text{ K}$$



From the problem statement,

$$T_7 = T_{13} - 65$$

The relations for heat input and expansion processes are

$$q_{in} = c_p (T_8 - T_7) \longrightarrow T_8 = T_7 + \frac{q_{in}}{c_p}$$

$$T_9 = T_8 \left(\frac{1}{r_p}\right)^{(k-1)/k}$$

$$T_{10} = T_9 + \frac{q_{in}}{c_p}, \qquad T_{11} = T_{10} \left(\frac{1}{r_p}\right)^{(k-1)/k}$$

$$T_{12} = T_{11} + \frac{q_{in}}{c_p}, \qquad T_{13} = T_{12} \left(\frac{1}{r_p}\right)^{(k-1)/k}$$

The simultaneous solution of above equations using EES software gives the following results

$$T_7 = 520.7\text{ K}, \quad T_8 = 819.2\text{ K}, \quad T_9 = 551.3\text{ K}$$
$$T_{10} = 849.8\text{ K}, \quad T_{11} = 571.9\text{ K}, \quad T_{12} = 870.4\text{ K}, \quad T_{13} = 585.7\text{ K}$$

From an energy balance on the regenerator,

$$T_7 - T_6 = T_{13} - T_{14}$$
$$(T_{13} - 65) - T_6 = T_{13} - T_{14} \longrightarrow T_{14} = T_6 + 65 = 430.9 + 65 = 495.9\text{ K}$$

The heat input is

$$q_{in} = 300 + 300 + 300 = 900\text{ kJ/kg}$$

The heat rejected is

$$\begin{aligned} q_{out} &= c_p (T_{14} - T_1) + c_p (T_2 - T_3) + c_p (T_4 - T_5) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(495.9 - 290 + 430.9 - 290 + 430.9 - 290)\text{ R} \\ &= 490.1\text{ kJ/kg} \end{aligned}$$

The thermal efficiency of the cycle is then

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{490.1}{900} = 0.455 = \mathbf{45.5\%}$$

Jet-Propulsion Cycles

9-136C The power developed from the thrust of the engine is called the propulsive power. It is equal to thrust times the aircraft velocity.

9-137C The ratio of the propulsive power developed and the rate of heat input is called the propulsive efficiency. It is determined by calculating these two quantities separately, and taking their ratio.

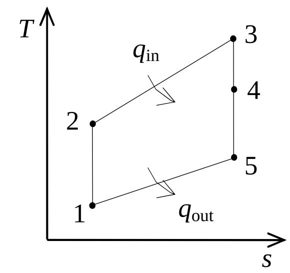
9-138C It reduces the exit velocity, and thus the thrust.

9-139E A turboprop engine operating on an ideal cycle is considered. The thrust force generated is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm·R (Table A-1E), c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis Working across the two isentropic processes of the cycle yields



$$T_2 = T_1 r_p^{(k-1)/k} = (450\text{ R})(10)^{0.4/1.4} = 868.8\text{ R}$$

$$T_5 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1400\text{ R})\left(\frac{1}{10}\right)^{0.4/1.4} = 725.1\text{ R}$$

Since the work produced by expansion 3-4 equals that used by compression 1-2, an energy balance gives

$$T_4 = T_3 - (T_2 - T_1) = 1400 - (868.8 - 450) = 981.2\text{ R}$$

The excess enthalpy generated by expansion 4-5 is used to increase the kinetic energy of the flow through the propeller,

$$\dot{m}_e c_p (T_4 - T_5) = \dot{m}_p \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2}$$

which when solved for the velocity at which the air leaves the propeller gives

$$V_{\text{exit}} = \left[2\frac{\dot{m}_e}{\dot{m}_p} c_p (T_4 - T_5) + V_{\text{inlet}}^2\right]^{1/2}$$

$$= \left[2\frac{1}{20}(0.24\text{ Btu/lbm}\cdot\text{R})(981.2 - 725.1)\text{R}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right) + (600\text{ ft/s})^2\right]^{1/2}$$

$$= 716.9\text{ ft/s}$$

The mass flow rate through the propeller is

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3)(450\text{ R})}{8\text{ psia}} = 20.84\text{ ft}^3/\text{lbm}$$

$$\dot{m}_p = \frac{AV_1}{\boldsymbol{v}_1} = \frac{\pi D^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi(10\text{ ft})^2}{4}\frac{600\text{ ft/s}}{20.84\text{ ft}^3/\text{lbm}} = 2261\text{ lbm/s}$$

The thrust force generated by this propeller is then

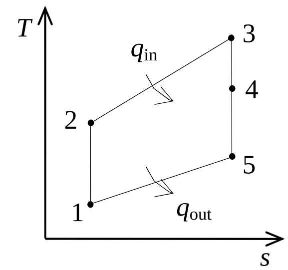
$$F = \dot{m}_p (V_{\text{exit}} - V_{\text{inlet}}) = (2261\text{ lbm/s})(716.9 - 600)\text{ft/s}\left(\frac{1\text{ lbf}}{32.174\text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{8215\ lbf}$$

9-140E A turboprop engine operating on an ideal cycle is considered. The thrust force generated is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm·R (Table A-1E), c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis Working across the two isentropic processes of the cycle yields



$$T_2 = T_1 r_p^{(k-1)/k} = (450\text{ R})(10)^{0.4/1.4} = 868.8\text{ R}$$

$$T_5 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1400\text{ R})\left(\frac{1}{10}\right)^{0.4/1.4} = 725.1\text{ R}$$

Since the work produced by expansion 3-4 equals that used by compression 1-2, an energy balance gives

$$T_4 = T_3 - (T_2 - T_1) = 1400 - (868.8 - 450) = 981.2\text{ R}$$

The mass flow rate through the propeller is

$$\boldsymbol{v}_1 = \frac{RT}{P} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3)(450\text{ R})}{8\text{ psia}} = 20.84\text{ ft}^3/\text{lbm}$$

$$\dot{m}_p = \frac{AV_1}{\boldsymbol{v}_1} = \frac{\pi D^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi(8\text{ ft})^2}{4}\frac{600\text{ ft/s}}{20.84\text{ ft}^3/\text{lbm}} = 1447\text{ lbm/s}$$

According to the previous problem,

$$\dot{m}_e = \frac{\dot{m}_p}{20} = \frac{2261\text{ lbm/s}}{20} = 113.1\text{ lbm/s}$$

The excess enthalpy generated by expansion 4-5 is used to increase the kinetic energy of the flow through the propeller,

$$\dot{m}_e c_p (T_4 - T_5) = \dot{m}_p \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2}$$

which when solved for the velocity at which the air leaves the propeller gives

$$\begin{aligned} V_{\text{exit}} &= \left[2\frac{\dot{m}_e}{\dot{m}_p}c_p(T_4 - T_5) + V_{\text{inlet}}^2\right]^{1/2} \\ &= \left[2\frac{113.1\text{ lbm/s}}{1447\text{ lbm/s}}(0.24\text{ Btu/lbm}\cdot\text{R})(981.2 - 725.1)\text{R}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right) + (600\text{ ft/s})^2\right]^{1/2} \\ &= 775.0\text{ ft/s} \end{aligned}$$

The thrust force generated by this propeller is then

$$F = \dot{m}_p(V_{\text{exit}} - V_{\text{inlet}}) = (1447\text{ lbm/s})(775 - 600)\text{ft/s}\left(\frac{1\text{ lbf}}{32.174\text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{7870\ lbf}$$

9-141 A turbofan engine operating on an ideal cycle produces 50,000 N of thrust. The air temperature at the fan outlet needed to produce this thrust is to be determined.

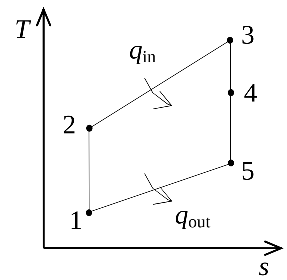
Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis The total mass flow rate is

$$\boldsymbol{v}_1 = \frac{RT}{P} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3)(253\ \text{K})}{50\ \text{kPa}} = 1.452\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{AV_1}{\boldsymbol{v}_1} = \frac{\pi D^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi(2.5\ \text{m})^2}{4}\frac{200\ \text{m/s}}{1.452\ \text{m}^3/\text{kg}} = 676.1\ \text{kg/s}$$



Now,

$$\dot{m}_e = \frac{\dot{m}}{8} = \frac{676.1\ \text{kg/s}}{8} = 84.51\ \text{kg/s}$$

The mass flow rate through the fan is

$$\dot{m}_f = \dot{m} - \dot{m}_e = 676.1 - 84.51 = 591.6\ \text{kg/s}$$

In order to produce the specified thrust force, the velocity at the fan exit will be

$$F = \dot{m}_f (V_{\text{exit}} - V_{\text{inlet}})$$

$$V_{\text{exit}} = V_{\text{inlet}} + \frac{F}{\dot{m}_f} = (200\ \text{m/s}) + \frac{50{,}000\ \text{N}}{591.6\ \text{kg/s}}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 284.5\ \text{m/s}$$

An energy balance on the stream passing through the fan gives

$$c_p(T_4 - T_5) = \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2}$$

$$T_5 = T_4 - \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2c_p}$$

$$= 253\ \text{K} - \frac{(284.5\ \text{m/s})^2 - (200\ \text{m/s})^2}{2(1.005\ \text{kJ/kg}\cdot\text{K})}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

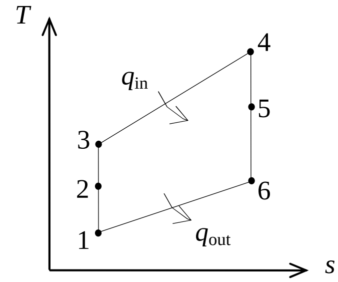
$$= \mathbf{232.6\ K}$$

9-142 A pure jet engine operating on an ideal cycle is considered. The velocity at the nozzle exit and the thrust produced are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K and k = 1.4 (Table A-2a).

Analysis (*a*) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 240 m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).



Diffuser:

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

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^{2\nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 260\text{ K} + \frac{(240\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 288.7\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (45\text{ kPa})\left(\frac{288.7\text{ K}}{260\text{ K}}\right)^{1.4/0.4} = 64.88\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (13)(64.88\text{ kPa}) = 843.5\text{ kPa}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (288.7\text{ K})(13)^{0.4/1.4} = 600.7\text{ K}$$

Turbine:

$$w_{comp,in} = w_{turb,out} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or $\quad T_5 = T_4 - T_3 + T_2 = 830 - 600.7 + 288.7 = 518.0\text{ K}$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (830\text{ K})\left(\frac{45\text{ kPa}}{843.5\text{ kPa}}\right)^{0.4/1.4} = 359.3\text{ K}$$

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

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\nearrow 0}}{2} \longrightarrow 0 = c_p(T_6 - T_5) + V_6^2/2$$

or $\quad V_6 = V_{exit} = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(518.0 - 359.3)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{564.8\ m/s}$

The mass flow rate through the engine is

$$\boldsymbol{v}_1 = \frac{RT}{P} = \frac{(0.287\text{ kPa}\cdot\text{m}^3)(260\text{ K})}{45\text{ kPa}} = 1.658\text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{AV_1}{\boldsymbol{v}_1} = \frac{\pi D^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi(1.6\text{ m})^2}{4}\frac{240\text{ m/s}}{1.658\text{ m}^3/\text{kg}} = 291.0\text{ kg/s}$$

The thrust force generated is then

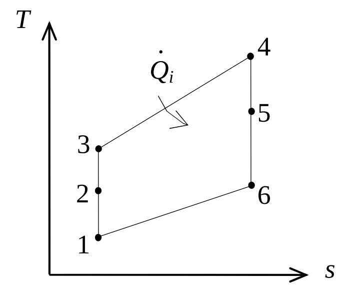
$$F = \dot{m}(V_{exit} - V_{inlet}) = (291.0\text{ kg/s})(564.8 - 240)\text{m/s}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{94{,}520\ N}$$

9-143 A turbojet aircraft flying at an altitude of 9150 m is operating on the ideal jet propulsion cycle. The velocity of exhaust gases, the propulsive power developed, and the rate of fuel consumption are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit. **5** The turbine work output is equal to the compressor work input.

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*) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 320 m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).



Diffuser:

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

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^{2\ \nearrow 0} - V_1^2}{2}$$

$$0 = c_p\left(T_2 - T_1\right) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 241\text{ K} + \frac{(320\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 291.9\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (32\text{ kPa})\left(\frac{291.9\text{ K}}{241\text{ K}}\right)^{1.4/0.4} = 62.6\text{ kPa}$$

Compressor:

$$P_3 = P_4 = \left(r_p\right)\left(P_2\right) = (12)(62.6\text{ kPa}) = 751.2\text{ kPa}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (291.9\text{ K})(12)^{0.4/1.4} = 593.7\text{ K}$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p\left(T_3 - T_2\right) = c_p\left(T_4 - T_5\right)$$

or $\quad T_5 = T_4 - T_3 + T_2 = 1400 - 593.7 + 291.9 = 1098.2\text{K}$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{32\text{ kPa}}{751.2\text{ kPa}}\right)^{0.4/1.4} = 568.2\text{ K}$$

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

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\ \nearrow 0}}{2} \longrightarrow 0 = c_p\left(T_6 - T_5\right) + V_6^2/2$$

or $\quad V_6 = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(1098.2 - 568.2)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{1032\text{ m/s}}$

(*b*) $\quad \dot{W}_p = \dot{m}\left(V_{\text{exit}} - V_{\text{inlet}}\right)V_{\text{aircraft}} = (60\text{ kg/s})(1032 - 320)\text{m/s}(320\text{ m/s})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{13{,}670\text{ kW}}$

(*c*) $\quad \dot{Q}_{\text{in}} = \dot{m}\left(h_4 - h_3\right) = \dot{m}c_p\left(T_4 - T_3\right) = (60\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400 - 593.7)\text{K} = 48{,}620\text{ kJ/s}$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}_{\text{in}}}{\text{HV}} = \frac{48{,}620\text{ kJ/s}}{42{,}700\text{ kJ/kg}} = \mathbf{1.14\text{ kg/s}}$$

9-144 A turbojet aircraft is flying at an altitude of 9150 m. The velocity of exhaust gases, the propulsive power developed, and the rate of fuel consumption are to be determined.

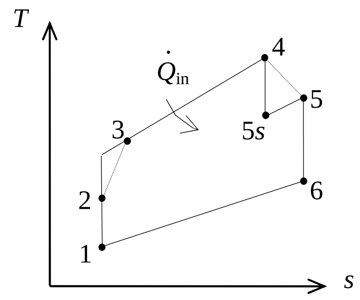
Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit.

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*) For convenience, we assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 320 m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).

Diffuser:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\;\nearrow 0 \text{ (steady)}}$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$
$$0 = h_2 - h_1 + \frac{V_2^{2\;\nearrow 0} - V_1^2}{2}$$
$$0 = c_p(T_2 - T_1) - V_1^2/2$$



$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 241\text{ K} + \frac{(320\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 291.9\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (32\text{ kPa})\left(\frac{291.9\text{ K}}{241\text{ K}}\right)^{1.4/0.4} = 62.6\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (12)(62.6\text{ kPa}) = 751.2\text{ kPa}$$

$$T_{3s} = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (291.9\text{ K})(12)^{0.4/1.4} = 593.7\text{ K}$$

$$\eta_C = \frac{h_{3s} - h_2}{h_3 - h_2} = \frac{c_p(T_{3s} - T_2)}{c_p(T_3 - T_2)}$$
$$T_3 = T_2 + (T_{3s} - T_2)/\eta_C = 291.9 + (593.7 - 291.9)/(0.80) = 669.2\text{ K}$$

Turbine:

$$w_{comp,in} = w_{turb,out} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or,

$$T_5 = T_4 - T_3 + T_2 = 1400 - 669.2 + 291.9 = 1022.7\text{ K}$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} = \frac{c_p(T_4 - T_5)}{c_p(T_4 - T_{5s})}$$
$$T_{5s} = T_4 - (T_4 - T_5)/\eta_T = 1400 - (1400 - 1022.7)/0.85 = 956.1\text{ K}$$

$$P_5 = P_4\left(\frac{T_{5s}}{T_4}\right)^{k/(k-1)} = (751.2\text{ kPa})\left(\frac{956.1\text{ K}}{1400\text{ K}}\right)^{1.4/0.4} = 197.7\text{ kPa}$$

Nozzle:

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (1022.7\text{ K})\left(\frac{32\text{ kPa}}{197.7\text{ kPa}}\right)^{0.4/1.4} = 607.8\text{ K}$$

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

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

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^2{}^{\nearrow 0}}{2}$$

$$0 = c_p(T_6 - T_5) + V_6^2/2$$

or,

$$V_6 = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(1022.7 - 607.8)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{913.2\ m/s}$$

(*b*)

$$\dot{W}_p = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}}$$
$$= (60\text{ kg/s})(913.2 - 320)\text{m/s}(320\text{ m/s})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$
$$= \mathbf{11{,}390\ kW}$$

(*c*)

$$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3) = (60\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400 - 669.2)\text{K} = 44{,}067\text{ kJ/s}$$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}_{\text{in}}}{\text{HV}} = \frac{44{,}067\text{ kJ/s}}{42{,}700\text{ kJ/kg}} = \mathbf{1.03\ kg/s}$$

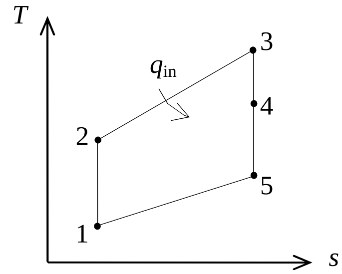
9-145 A turbojet aircraft that has a pressure rate of 9 is stationary on the ground. The force that must be applied on the brakes to hold the plane stationary is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the nozzle exit.

Properties The properties of air are given in Table A-17.

Analysis (*a*) Using variable specific heats for air,

Compressor:



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

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (9)(1.2311) = 11.08 \longrightarrow h_2 = 544.07 \text{ kJ/kg}$$

$$\dot{Q}_{in} = \dot{m}_{fuel} \times \text{HV} = (0.5 \text{ kg/s})(42{,}700 \text{ kJ/kg}) = 21{,}350 \text{ kJ/s}$$

$$q_{in} = \frac{\dot{Q}_{in}}{\dot{m}} = \frac{21{,}350 \text{ kJ/s}}{20 \text{ kg/s}} = 1067.5 \text{ kJ/kg}$$

$$q_{in} = h_3 - h_2 \longrightarrow h_3 = h_2 + q_{in} = 544.07 + 1067.5 = 1611.6 \text{ kJ/kg} \longrightarrow P_{r_3} = 568.5$$

Turbine:

$$w_{comp,in} = w_{turb,out} \longrightarrow h_2 - h_1 = h_3 - h_4$$

or

$$h_4 = h_3 - h_2 + h_1 = 1611.6 - 544.07 + 290.16 = 1357.7 \text{ kJ/kg}$$

Nozzle:

$$P_{r_5} = P_{r_3}\left(\frac{P_5}{P_3}\right) = (568.5)\left(\frac{1}{9}\right) = 63.17 \longrightarrow h_5 = 888.56 \text{ kJ/kg}$$

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

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

$$h_4 + V_4^2/2 = h_5 + V_5^2/2$$

$$0 = h_5 - h_4 + \frac{V_5^2 - V_4^{2\nearrow 0}}{2}$$

or

$$V_5 = \sqrt{2(h_4 - h_5)} = \sqrt{(2)(1357.7 - 888.56)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 968.6 \text{ m/s}$$

$$\text{Brake force} = \text{Thrust} = \dot{m}(V_{exit} - V_{inlet}) = (20 \text{ kg/s})(968.6 - 0)\text{m/s}\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{19{,}370 \text{ N}}$$

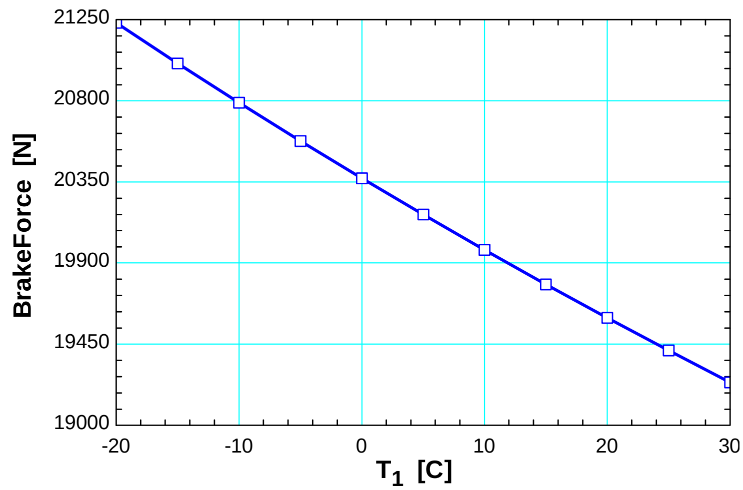
9-146 Problem 9-145 is reconsidered. The effect of compressor inlet temperature on the force that must be applied to the brakes to hold the plane stationary is to be investigated.

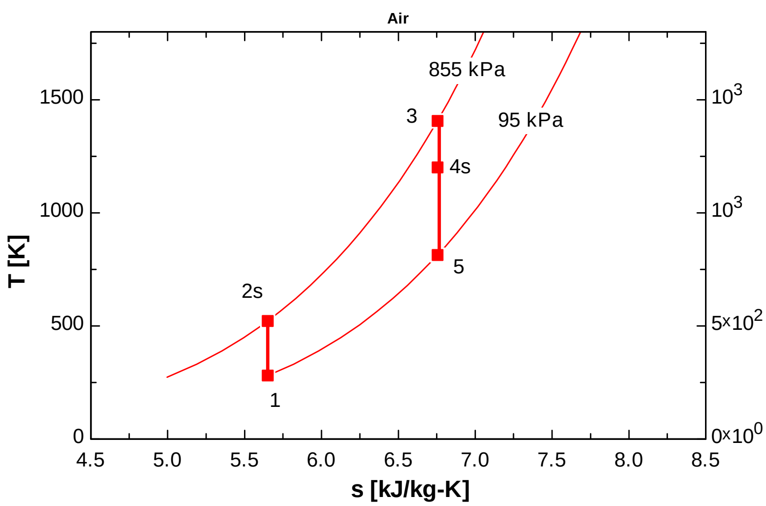
Analysis Using EES, the problem is solved as follows:

```
P_ratio =9
T_1 = 7 [C]
T[1] = T_1+273 "[K]"
P[1]= 95 [kPa]
P[5]=P[1]
Vel[1]=0 [m/s]
V_dot[1] = 18.1 [m^3/s]
HV_fuel = 42700 [kJ/kg]
m_dot_fuel = 0.5 [kg/s]
Eta_c = 1.0
Eta_t = 1.0
Eta_N = 1.0

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])
v[1]=volume(Air,T=T[1],P=P[1])
m_dot = V_dot[1]/v[1]
"Compressor anaysis"
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])
Q_dot_in = m_dot_fuel*HV_fuel
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,h=h_s[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"
T[4]=TEMPERATURE(Air,h=h[4])
P[4]=pressure(Air,s=s_s[4],h=h_s[4])
"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
W_dot_net = 0 [kW]

"Exit nozzle analysis:"
s[4]=entropy('air',T=T[4],P=P[4])
s_s[5]=s[4] "For the ideal case the entropies are constant across the nozzle"

T_s[5]=TEMPERATURE(Air,s=s_s[5], P=P[5]) "T_s[5] is the isentropic value of T[5] at nozzle exit"
h_s[5]=ENTHALPY(Air,T=T_s[5])
Eta_N=(h[4]-h[5])/(h[4]-h_s[5])
m_dot*h[4] =  m_dot*(h_s[5] + Vel_s[5]^2/2*convert(m^2/s^2,kJ/kg))
```

```
m_dot*h[4] =  m_dot*(h[5] + Vel[5]^2/2*convert(m^2/s^2,kJ/kg))
T[5]=TEMPERATURE(Air,h=h[5])
s[5]=entropy('air',T=T[5],P=P[5])

"Brake Force to hold the aircraft:"
Thrust = m_dot*(Vel[5] - Vel[1]) "[N]"
BrakeForce = Thrust "[N]"
"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
s[2]=entropy('air',T=T[2],P=P[2])
```

Brake Force [N]	m [kg/s]	T_3 [K]	T_1 [C]
21232	23.68	1284	-20
21007	23.22	1307	-15
20788	22.78	1330	-10
20576	22.35	1352	-5
20369	21.94	1375	0
20168	21.55	1398	5
19972	21.17	1420	10
19782	20.8	1443	15
19596	20.45	1466	20
19415	20.1	1488	25
19238	19.77	1510	30





9-147 Air enters a turbojet engine. The thrust produced by this turbojet engine is to be determined.

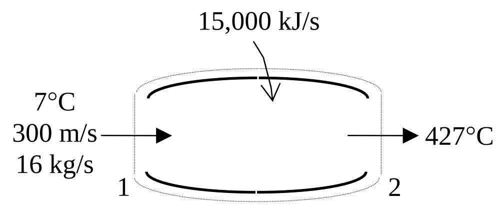
Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit.

Properties The properties of air are given in Table A-17.

Analysis We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 300 m/s. Taking the entire engine as our control volume and writing the steady-flow energy balance yield

$$T_1 = 280 \text{ K} \longrightarrow h_1 = 280.13 \text{ kJ/kg}$$

$$T_2 = 700 \text{ K} \longrightarrow h_2 = 713.27 \text{ kJ/kg}$$



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

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

$$\dot{Q}_{\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$\dot{Q}_{\text{in}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

$$15{,}000 \text{ kJ/s} = (16 \text{ kg/s})\left[713.27 - 280.13 + \frac{V_2^2 - (300 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right]$$

It gives

$$V_2 = 1048 \text{ m/s}$$

Thus,

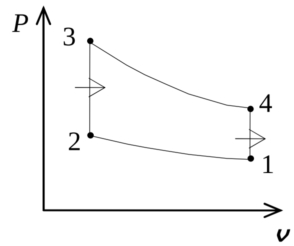
$$F_p = \dot{m}(V_2 - V_1) = (16 \text{ kg/s})(1048 - 300)\text{m/s} = \mathbf{11{,}968 \text{ N}}$$

Second-Law Analysis of Gas Power Cycles

9-148 The process with the highest exergy destruction for an ideal Otto cycle described in Prob. 9-36 is to be determined.

Analysis From Prob. 9-36, q_{in} = 582.5 kJ/kg, q_{out} = 253.6 kJ/kg, T_1 = 288 K, T_2 = 661.7 K, T_3 = 1473 K, and T_4 = 641.2 K. The exergy destruction during a process of the cycle is

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

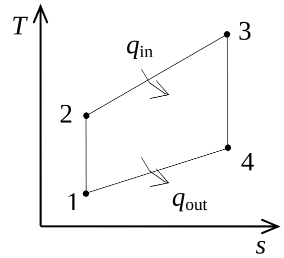


Application of this equation for each process of the cycle gives

$$x_{dest,1\text{-}2} = \mathbf{0} \quad \text{(isentropic process)}$$

$$s_3 - s_2 = s_4 - s_1 = c_v \ln\frac{T_3}{T_2} + R\ln\frac{v_3}{v_2}$$

$$= (0.718\ \text{kJ/kg}\cdot\text{K})\ln\frac{1473\ \text{K}}{661.7\ \text{K}} + 0 = 0.5746\ \text{kJ/kg}\cdot\text{K}$$



$$x_{dest,2\text{-}3} = T_0\left(s_3 - s_2 - \frac{q_{in}}{T_{source}}\right)$$

$$= (288\ \text{K})\left(0.5746\ \text{kJ/kg}\cdot\text{K} - \frac{582.5\ \text{kJ/kg}}{1473\ \text{K}}\right)$$

$$= \mathbf{51.59\ kJ/kg}$$

$$x_{dest,3\text{-}4} = \mathbf{0} \quad \text{(isentropic process)}$$

$$x_{dest,4\text{-}1} = T_0\left(s_1 - s_4 + \frac{q_{out}}{T_{sink}}\right)$$

$$= (288\ \text{K})\left(-0.5746\ \text{kJ/kg}\cdot\text{K} + \frac{253.6\ \text{kJ/kg}}{288\ \text{K}}\right)$$

$$= \mathbf{88.12\ kJ/kg}$$

The largest exergy destruction in the cycle occurs during the heat-rejection process.

9-149E The exergy destruction associated with the heat rejection process of the Diesel cycle described in Prob. 9-55E and the exergy at the end of the expansion stroke are to be determined.

Analysis From Prob. 9-55E, q_{out} = 158.9 Btu/lbm, T_1 = 540 R, P_1 = 14.7 psia, T_4 = 1420.6 R, P_4 = 38.62 psia and $\boldsymbol{v}_4 = \boldsymbol{v}_1$.

The entropy change during process 4-1 is

$$\begin{aligned} s_1 - s_4 &= s^o_{1\,@540\text{R}} - s^o_{4\,@1420.6\,\text{R}} - R\ln(P_1/P_4) \\ &= 0.60078 - 0.83984 - (0.06855)\ln(14.7/38.62) \\ &= -0.1728 \text{ Btu/lbm}\cdot\text{R} \end{aligned}$$

Thus,

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = (540\text{R})\left(-0.1728 \text{ Btu/lbm}\cdot\text{R} + \frac{158.9 \text{ Btu/lbm}}{540 \text{ R}}\right) = \mathbf{65.6\ \ Btu/lbm}$$

Noting that state 4 is identical to the state of the surroundings, the exergy at the end of the power stroke (state 4) is determined from

$$\phi_4 = (u_4 - u_0) - T_0(s_4 - s_0) + P_0(\boldsymbol{v}_4 - \boldsymbol{v}_0)$$

where

$$\begin{aligned} u_4 - u_0 &= u_4 - u_1 = q_{\text{out}} = 158.9 \text{ Btu/lbm}\cdot\text{R} \\ \boldsymbol{v}_4 - \boldsymbol{v}_0 &= \boldsymbol{v}_4 - \boldsymbol{v}_1 = 0 \\ s_4 - s_0 &= s_4 - s_1 = 0.1741 \text{ Btu/lbm}\cdot\text{R} \end{aligned}$$

Thus,

$$\phi_4 = (158.9 \text{ Btu/lbm}) - (540\text{R})(0.1728 \text{ Btu/lbm}\cdot\text{R}) + 0 = \mathbf{65.6\,Btu/lbm}$$

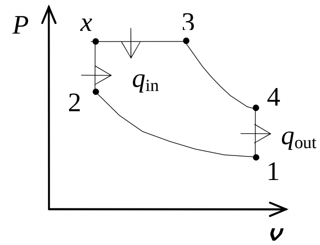
Discussion Note that the exergy at state 4 is identical to the exergy destruction for the process 4-1 since state 1 is identical to the dead state, and the entire exergy at state 4 is wasted during process 4-1.

9-150 The exergy loss of each process for an ideal dual cycle described in Prob. 9-63 is to be determined.

Analysis From Prob. 9-63, $q_{in,x\text{-}3}$ = 114.6 kJ/kg, T_1 = 291 K, T_2 = 1037 K, T_x = 1141 K, T_3 = 1255 K, and T_4 = 494.8 K. Also,

$$q_{in,2-x} = c_v(T_x - T_2) = (0.718 \text{ kJ/kg}\cdot\text{K})(1141-1037)\text{K} = 74.67 \text{ kJ/kg}$$

$$q_{out} = c_v(T_4 - T_1) = (0.718 \text{ kJ/kg}\cdot\text{K})(494.8-291)\text{K} = 146.3 \text{ kJ/kg}$$



The exergy destruction during a process of the cycle is

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

Application of this equation for each process of the cycle gives

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.005 \text{ kJ/kg}\cdot\text{K})\ln\frac{1037 \text{ K}}{291 \text{ K}} - (0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{5148 \text{ kPa}}{90 \text{ kPa}} \\ &= 0.1158 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$x_{dest,1\text{-}2} = T_0(s_2 - s_1) = (291 \text{ K})(0.1158 \text{ kJ/kg}\cdot\text{K}) = \mathbf{33.7 \text{ kJ/kg}}$$

$$\begin{aligned} s_x - s_2 &= c_v \ln\frac{T_x}{T_2} + R\ln\frac{v_x}{v_2} \\ &= (0.718 \text{ kJ/kg}\cdot\text{K})\ln\frac{1141 \text{ K}}{1037 \text{ K}} + 0 = 0.06862 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$x_{dest,\,2\text{-}x} = T_0\left(s_x - s_2 - \frac{q_{in,\,2\text{-}x}}{T_{source}}\right) = (291 \text{ K})\left(0.06862 \text{ kJ/kg}\cdot\text{K} - \frac{74.67 \text{ kJ/kg}}{1255 \text{ K}}\right) = \mathbf{2.65 \text{ kJ/kg}}$$

$$s_3 - s_x = c_p \ln\frac{T_3}{T_x} - R\ln\frac{P_3}{P_x} = (1.005 \text{ kJ/kg}\cdot\text{K})\ln\frac{1255 \text{ K}}{1141 \text{ K}} - 0 = 0.09571 \text{ kJ/kg}\cdot\text{K}$$

$$x_{dest,\,x\text{-}3} = T_0\left(s_3 - s_x - \frac{q_{in,\,x\text{-}3}}{T_{source}}\right) = (291 \text{ K})\left(0.09571 \text{ kJ/kg}\cdot\text{K} - \frac{114.6 \text{ kJ/kg}}{1255 \text{ K}}\right) = \mathbf{1.28 \text{ kJ/kg}}$$

$$\begin{aligned} s_4 - s_3 &= c_v \ln\frac{T_4}{T_3} + R\ln\frac{v_4}{v_3} = c_v \ln\frac{T_4}{T_3} + R\ln\frac{r}{r_c} \\ &= (0.718 \text{ kJ/kg}\cdot\text{K})\ln\frac{494.8 \text{ K}}{1255 \text{ K}} + (0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{18}{1.1} = 0.1339 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$x_{dest,\,3\text{-}4} = T_0(s_4 - s_3) = (291 \text{ K})(0.1339 \text{ kJ/kg}\cdot\text{K}) = \mathbf{39.0 \text{ kJ/kg}}$$

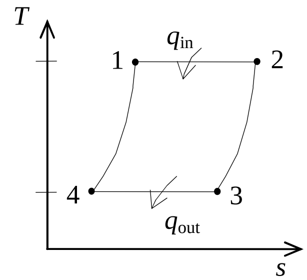
$$s_1 - s_4 = c_v \ln\frac{T_1}{T_4} + R\ln\frac{v_1}{v_4} = (0.718 \text{ kJ/kg}\cdot\text{K})\ln\frac{291 \text{ K}}{494.8 \text{ K}} + 0 = -0.3811 \text{ kJ/kg}\cdot\text{K}$$

$$x_{dest,4\text{-}1} = T_0\left(s_1 - s_4 + \frac{q_{out}}{T_{sink}}\right) = (291 \text{ K})\left(-0.3811 \text{ kJ/kg}\cdot\text{K} + \frac{146.3 \text{ kJ/kg}}{291 \text{ K}}\right) = \mathbf{35.4 \text{ kJ/kg}}$$

9-151 The exergy loss of each process for an air-standard Stirling cycle described in Prob. 9-81 is to be determined.

Analysis From Prob. 9-81, q_{in} = 1275 kJ/kg, q_{out} = 212.5 kJ/kg, $T_1 = T_2$ = 1788 K, $T_3 = T_4$ = 298 K. The exergy destruction during a process of the cycle is

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



Application of this equation for each process of the cycle gives

$$s_2 - s_1 = c_v \ln\frac{T_2}{T_1} + R\ln\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}$$
$$= 0 + (0.287\ \text{kJ/kg}\cdot\text{K})\ln(12) = 0.7132\ \text{kJ/kg}\cdot\text{K}$$

$$x_{dest,1\text{-}2} = T_0\left(s_2 - s_1 - \frac{q_{in}}{T_{source}}\right)$$
$$= (298\ \text{K})\left(0.7132\ \text{kJ/kg}\cdot\text{K} - \frac{1275\ \text{kJ/kg}}{1788\ \text{K}}\right) = \mathbf{0.034\ kJ/kg \approx 0}$$

$$s_4 - s_3 = c_v \ln\frac{T_4}{T_3} + R\ln\frac{\boldsymbol{v}_4}{\boldsymbol{v}_3}$$
$$= 0 + (0.287\ \text{kJ/kg}\cdot\text{K})\ln\left(\frac{1}{12}\right) = -0.7132\ \text{kJ/kg}\cdot\text{K}$$

$$x_{dest,3\text{-}4} = T_0\left(s_4 - s_3 + \frac{q_{out}}{T_{sink}}\right)$$
$$= (298\ \text{K})\left(-0.7132\ \text{kJ/kg}\cdot\text{K} + \frac{212.5\ \text{kJ/kg}}{298\ \text{K}}\right) = \mathbf{-0.034\ kJ/kg \approx 0}$$

These results are not surprising since Stirling cycle is totally reversible. Exergy destructions are not calculated for processes 2-3 and 4-1 because there is no interaction with the surroundings during these processes to alter the exergy destruction.

9-152 The exergy destruction associated with each of the processes of the Brayton cycle described in Prob. 9-89 is to be determined.

Analysis From Prob. 9-89, q_{in} = 698.3 kJ/kg, q_{out} = 487.9 kJ/kg, and

$$T_1 = 295 \text{ K} \longrightarrow s_1^\circ = 1.68515 \text{ kJ/kg}\cdot\text{K}$$
$$h_2 = 626.60 \text{ kJ/kg} \longrightarrow s_2^\circ = 2.44117 \text{ kJ/kg}\cdot\text{K}$$
$$T_3 = 1240 \text{ K} \longrightarrow s_3^\circ = 3.21751 \text{ kJ/kg}\cdot\text{K}$$
$$h_4 = 783.04 \text{ kJ/kg} \longrightarrow s_4^\circ = 2.66807 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$x_{\text{destroyed},12} = T_0 s_{\text{gen},12} = T_0\left(s_2 - s_1\right) = T_0\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right) =$$
$$= (310 \text{ K})(2.44117 - 1.68515 - (0.287 \text{ kJ/kg}\cdot\text{K})\ln(10)) = \textbf{29.51 kJ/kg}$$

$$x_{\text{destroyed},23} = T_0 s_{\text{gen},23} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = T_0\left(s_3^\circ - s_2^\circ - R\ln\frac{P_3}{P_2}^{\nearrow 0} + \frac{-q_{in}}{T_H}\right)$$
$$= (310 \text{ K})\left(3.21751 - 2.44117 - \frac{698.3 \text{ kJ/kg}}{1600 \text{ K}}\right) = \textbf{105.4 kJ/kg}$$

$$x_{\text{destroyed},34} = T_0 s_{\text{gen},34} = T_0\left(s_4 - s_3\right) = T_0\left(s_4^\circ - s_3^\circ - R\ln\frac{P_4}{P_3}\right) =$$
$$= (310 \text{ K})(2.66807 - 3.21751 - (0.287 \text{ kJ/kg}\cdot\text{K})\ln(1/10)) = \textbf{34.53 kJ/kg}$$

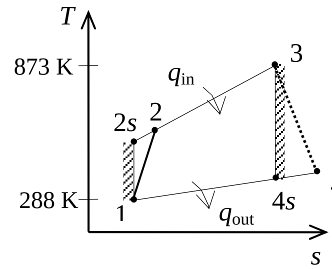
$$x_{\text{destroyed},41} = T_0 s_{\text{gen},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = T_0\left(s_1^\circ - s_4^\circ - R\ln\frac{P_1}{P_4}^{\nearrow 0} + \frac{q_{out}}{T_L}\right)$$
$$= (310 \text{ K})\left(1.68515 - 2.66807 + \frac{487.9 \text{ kJ/kg}}{310 \text{ K}}\right) = \textbf{183.2 kJ/kg}$$

9-153 Exergy analysis is to be used to answer the question in Prob. 9-94.

Analysis From Prob. 9-94, T_1 = 288 K, T_{2s} = 585.8 K, T_2 = 618.9 K, T_3 = 873 K, T_{4s} = 429.2 K, T_4 = 473.6 K, r_p = 12. The exergy change of a flow stream between an inlet and exit state is given by

$$\Delta\psi = h_e - h_i - T_0(s_e - s_i)$$

This is also the expression for reversible work. Application of this equation for isentropic and actual compression processes gives



$$s_{2s} - s_1 = c_p \ln\frac{T_{2s}}{T_1} - R\ln\frac{P_2}{P_1}$$
$$= (1.005 \text{ kJ/kg}\cdot\text{K})\ln\frac{585.8\text{ K}}{288\text{ K}} - (0.287 \text{ kJ/kg}\cdot\text{K})\ln(12)$$
$$= 0.0003998 \text{ kJ/kg}\cdot\text{K}$$

$$w_{\text{rev},1-2s} = c_p(T_{2s} - T_1) - T_0(s_{2s} - s_1)$$
$$= (1.005 \text{ kJ/kg}\cdot\text{K})(585.8 - 288)\text{K} - (288\text{ K})(0.0003998 \text{ kJ/kg}\cdot\text{K}) = 299.2 \text{ kJ/kg}$$

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$
$$= (1.005 \text{ kJ/kg}\cdot\text{K})\ln\frac{618.9\text{ K}}{288\text{ K}} - (0.287 \text{ kJ/kg}\cdot\text{K})\ln(12) = 0.05564 \text{ kJ/kg}\cdot\text{K}$$

$$w_{\text{rev},1-2} = c_p(T_2 - T_1) - T_0(s_2 - s_1)$$
$$= (1.005 \text{ kJ/kg}\cdot\text{K})(618.9 - 288)\text{K} - (288\text{ K})(0.05564 \text{ kJ/kg}\cdot\text{K}) = 316.5 \text{ kJ/kg}$$

The irreversibilities therefore increase the minimum work that must be supplied to the compressor by

$$\Delta w_{\text{rev,C}} = w_{\text{rev},1-2} - w_{\text{rev},1-2s} = 316.5 - 299.2 = \mathbf{17.3\ kJ/kg}$$

Repeating the calculations for the turbine,

$$s_3 - s_{4s} = c_p \ln\frac{T_3}{T_{4s}} - R\ln\frac{P_3}{P_4}$$
$$= (1.005 \text{ kJ/kg}\cdot\text{K})\ln\frac{873\text{ K}}{429.2\text{K}} - (0.287 \text{ kJ/kg}\cdot\text{K})\ln(12) = 0.0003944 \text{ kJ/kg}\cdot\text{K}$$

$$w_{\text{rev},3-4s} = c_p(T_3 - T_{4s}) - T_0(s_3 - s_{4s})$$
$$= (1.005 \text{ kJ/kg}\cdot\text{K})(873 - 429.2)\text{K} - (288\text{ K})(0.0003944 \text{ kJ/kg}\cdot\text{K}) = 445.9 \text{ kJ/kg}$$

$$s_3 - s_4 = c_p \ln\frac{T_3}{T_4} - R\ln\frac{P_3}{P_4}$$
$$= (1.005 \text{ kJ/kg}\cdot\text{K})\ln\frac{873\text{ K}}{473.6\text{K}} - (0.287 \text{ kJ/kg}\cdot\text{K})\ln(12) = -0.09854 \text{ kJ/kg}\cdot\text{K}$$

$$w_{\text{rev},3-4} = c_p(T_3 - T_{4s}) - T_0(s_3 - s_{4s})$$
$$= (1.005 \text{ kJ/kg}\cdot\text{K})(873 - 473.6)\text{K} - (288\text{ K})(-0.09854 \text{ kJ/kg}\cdot\text{K}) = 429.8 \text{ kJ/kg}$$

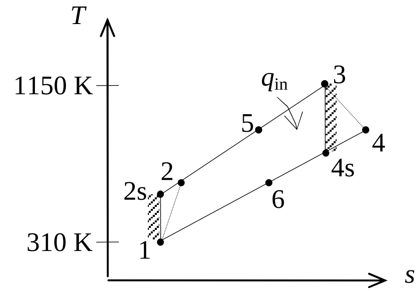
$$\Delta w_{\text{rev,T}} = w_{\text{rev},3-4s} - w_{\text{rev},3-4} = 445.9 - 429.8 = \mathbf{16.1\,kJ/kg}$$

Hence, it is clear that the compressor is a little more sensitive to the irreversibilities than the turbine.

9-154 The total exergy destruction associated with the Brayton cycle described in Prob. 9-116 and the exergy at the exhaust gases at the turbine exit are to be determined.

Properties The gas constant of air is R = 0.287 kJ/kg·K (Table A-1).

Analysis From Prob. 9-116, q_{in} = 480.82, q_{out} = 372.73 kJ/kg, and



$$T_1 = 310\text{ K} \longrightarrow s_1^\circ = 1.73498\text{ kJ/kg}\cdot\text{K}$$
$$h_2 = 618.26\text{ kJ/kg} \longrightarrow s_2^\circ = 2.42763\text{ kJ/kg}\cdot\text{K}$$
$$T_3 = 1150\text{ K} \longrightarrow s_3^\circ = 3.12900\text{ kJ/kg}\cdot\text{K}$$
$$h_4 = 803.14\text{ kJ/kg} \longrightarrow s_4^\circ = 2.69407\text{ kJ/kg}\cdot\text{K}$$
$$h_5 = 738.43\text{ kJ/kg} \longrightarrow s_5^\circ = 2.60815\text{ kJ/kg}\cdot\text{K}$$

and, from an energy balance on the heat exchanger,

$$h_5 - h_2 = h_4 - h_6 \longrightarrow h_6 = 803.14 - (738.43 - 618.26) = 682.97\text{ kJ/kg}$$
$$\longrightarrow s_6^\circ = 2.52861\text{ kJ/kg}\cdot\text{K}$$

Thus,

$$x_{\text{destroyed},12} = T_0 s_{\text{gen},12} = T_0\left(s_2 - s_1\right) = T_0\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right)$$
$$= (290\text{ K})(2.42763 - 1.73498 - (0.287\text{ kJ/kg}\cdot\text{K})\ln(7)) = \mathbf{38.91\ kJ/kg}$$
$$x_{\text{destroyed},34} = T_0 s_{\text{gen},34} = T_0\left(s_4 - s_3\right) = T_0\left(s_4^\circ - s_3^\circ - R\ln\frac{P_4}{P_3}\right)$$
$$= (290\text{ K})(2.69407 - 3.12900 - (0.287\text{kJ/kg}\cdot\text{K})\ln(1/7)) = \mathbf{35.83\ \ kJ/kg}$$
$$x_{\text{destroyed, regen}} = T_0 s_{\text{gen,regen}} = T_0\left[(s_5 - s_2) + (s_6 - s_4)\right] = T_0\left[\left(s_5^\circ - s_2^\circ\right) + \left(s_6^\circ - s_4^\circ\right)\right]$$
$$= (290\text{ K})(2.60815 - 2.42763 + 2.52861 - 2.69407) = \mathbf{4.37\ kJ/kg}$$
$$x_{\text{destroyed},53} = T_0 s_{\text{gen},53} = T_0\left(s_3 - s_5 - \frac{q_{R,53}}{T_R}\right) = T_0\left(s_3^\circ - s_5^\circ - R\ln\frac{P_3}{P_5}^{\nearrow 0} - \frac{q_{in}}{T_H}\right)$$
$$= (290\text{ K})\left(3.12900 - 2.60815 - \frac{480.82\text{ kJ/kg}}{1500\text{ K}}\right) = \mathbf{58.09\ \ kJ/kg}$$
$$x_{\text{destroyed},61} = T_0 s_{\text{gen},61} = T_0\left(s_1 - s_6 + \frac{q_{R,61}}{T_R}\right) = T_0\left(s_1^\circ - s_6^\circ - R\ln\frac{P_1}{P_6}^{\nearrow 0} + \frac{q_{out}}{T_L}\right)$$
$$= (290\text{ K})\left(1.73498 - 2.52861 + \frac{372.73\text{ kJ/kg}}{290\text{ K}}\right) = \mathbf{142.6\ \ kJ/kg}$$

Noting that $h_0 = h_{@\ 290\text{ K}}$ = 290.16 kJ/kg and $T_0 = 290\text{ K} \longrightarrow s_1^\circ = 1.66802\text{ kJ/kg}\cdot\text{K}$, the stream exergy at the exit of the regenerator (state 6) is determined from

$$\phi_6 = (h_6 - h_0) - T_0(s_6 - s_0) + \frac{V_6^2}{2}^{\nearrow 0} + gz_6^{\nearrow 0}$$

where

$$s_6 - s_0 = s_6 - s_1 = s_6^\circ - s_1^\circ - R\ln\frac{P_6}{P_1}^{\nearrow 0} = 2.52861 - 1.66802 = 0.86059\text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\phi_6 = 682.97 - 290.16 - (290\text{ K})(0.86059\text{ kJ/kg}\cdot\text{K}) = \mathbf{143.2\ kJ/kg}$$

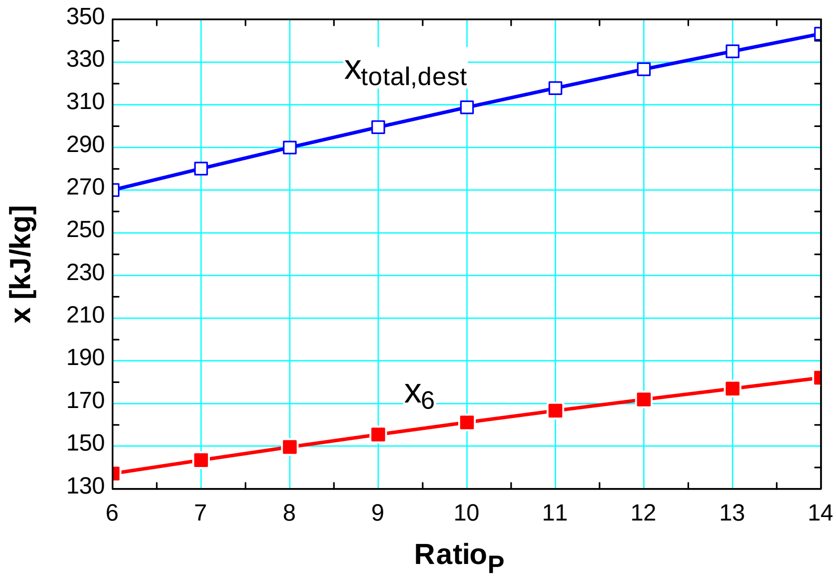
9-155 Prob. 9-154 is reconsidered. The effect of the cycle pressure on the total irreversibility for the cycle and the exergy of the exhaust gas leaving the regenerator is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
"Given"
T[1]=310 [K]
P[1]=100 [kPa]
Ratio_P=7
P[2]=Ratio_P*P[1]
T[3]=1150 [K]
eta_C=0.75
eta_T=0.82
epsilon=0.65
T_H=1500 [K]
T0=290 [K]
P0=100 [kPa]

"Analysis for Problem 9-154"
q_in=h[3]-h[5]
q_out=h[6]-h[1]
h[5]-h[2]=h[4]-h[6]
s[2]=entropy(Fluid$, P=P[2], h=h[2])
s[4]=entropy(Fluid$, h=h[4], P=P[4])
s[5]=entropy(Fluid$, h=h[5], P=P[5])
P[5]=P[2]
s[6]=entropy(Fluid$, h=h[6], P=P[6])
P[6]=P[1]
h[0]=enthalpy(Fluid$, T=T0)
s[0]=entropy(Fluid$, T=T0, P=P0)
x_destroyed_12=T0*(s[2]-s[1])
x_destroyed_34=T0*(s[4]-s[3])
x_destroyed_regen=T0*(s[5]-s[2]+s[6]-s[4])
x_destroyed_53=T0*(s[3]-s[5]-q_in/T_H)
x_destroyed_61=T0*(s[1]-s[6]+q_out/T0)
x_total=x_destroyed_12+x_destroyed_34+x_destroyed_regen+x_destroyed_53+x_destroyed_61
x6=h[6]-h[0]-T0*(s[6]-s[0]) "since state 0 and state 1 are identical"

"Analysis for Problem 9-116"
Fluid$='air'
"(a)"
h[1]=enthalpy(Fluid$, T=T[1])
s[1]=entropy(Fluid$, T=T[1], P=P[1])
s_s[2]=s[1] "isentropic compression"
h_s[2]=enthalpy(Fluid$, P=P[2], s=s_s[2])
eta_C=(h_s[2]-h[1])/(h[2]-h[1])
h[3]=enthalpy(Fluid$, T=T[3])
s[3]=entropy(Fluid$, T=T[3], P=P[3])
P[3]=P[2]
s_s[4]=s[3] "isentropic expansion"
h_s[4]=enthalpy(Fluid$, P=P[4], s=s_s[4])
P[4]=P[1]
eta_T=(h[3]-h[4])/(h[3]-h_s[4])
q_regen=epsilon*(h[4]-h[2])

"(b)"
w_C_in=(h[2]-h[1])
w_T_out=h[3]-h[4]
```

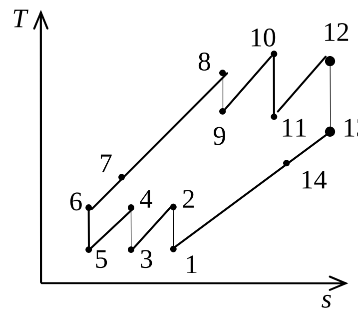
```
w_net_out=w_T_out-w_C_in
q_in=(h[3]-h[2])-q_regen
eta_th=w_net_out/q_in
```

Ratio_P	x_{total} [kJ/kg]	x6 [kJ/kg]
6	270.1	137.2
7	280	143.5
8	289.9	149.6
9	299.5	155.5
10	308.8	161.1
11	317.8	166.6
12	326.6	171.9
13	335.1	177.1
14	343.3	182.1



9-156 The exergy loss of each process for a regenerative Brayton cycle with three stages of reheating and intercooling described in Prob. 9-135 is to be determined.

Analysis From Prob. 9-135,



$r_p = 4$, $q_{in,7\text{-}8} = q_{in,9\text{-}10} = q_{in,11\text{-}12} = 300$ kJ/kg,

$q_{out,14\text{-}1} = 206.9$ kJ/kg, $q_{out,2\text{-}3} = q_{out,4\text{-}5} = 141.6$ kJ/kg,

$T_1 = T_3 = T_5 = 290$ K , $T_2 = T_4 = T_6 = 430.9$ K

$T_7 = 520.7\text{ K}, \quad T_8 = 819.2\text{ K}, \quad T_9 = 551.3\text{ K}$

$T_{10} = 849.8\text{ K}, \quad T_{11} = 571.9\text{ K}, \quad T_{12} = 870.4\text{ K},$

$T_{13} = 585.7\text{ K}, \quad T_{14} = 495.9\text{ K}$

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

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

Application of this equation for each process of the cycle gives

$$x_{dest,1\text{-}2} = x_{dest,3\text{-}4} = x_{dest,5\text{-}6} = T_0\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)$$
$$= (290)\left[(1.005)\ln\frac{430.9}{290} - (0.287)\ln(4)\right] = \mathbf{0.03\ kJ/kg \approx 0}$$

$$x_{dest,7\text{-}8} = T_0\left(c_p \ln\frac{T_8}{T_7} - R\ln\frac{P_8}{P_7} - \frac{q_{in,7\text{-}8}}{T_{source}}\right) = (290)\left[(1.005)\ln\frac{819.2}{520.7} - 0 - \frac{300}{870.4}\right] = \mathbf{32.1\ kJ/kg}$$

$$x_{dest,9\text{-}10} = T_0\left(c_p \ln\frac{T_{10}}{T_9} - R\ln\frac{P_8}{P_7} - \frac{q_{in,9\text{-}10}}{T_{source}}\right) = (290)\left[(1.005)\ln\frac{849.8}{551.3} - 0 - \frac{300}{870.4}\right] = \mathbf{26.2\ kJ/kg}$$

$$x_{dest,11\text{-}12} = T_0\left(c_p \ln\frac{T_{12}}{T_{11}} - R\ln\frac{P_{12}}{P_{11}} - \frac{q_{in,11\text{-}12}}{T_{source}}\right) = (290)\left[(1.005)\ln\frac{870.4}{571.9} - 0 - \frac{300}{870.4}\right] = \mathbf{22.5\ kJ/kg}$$

$$x_{dest,8\text{-}9} = T_0\left(c_p \ln\frac{T_9}{T_8} - R\ln\frac{P_9}{P_8}\right) = (290)\left[(1.005)\ln\frac{551.3}{819.2} - (0.287)\ln\left(\frac{1}{4}\right)\right] = \mathbf{-0.05\ kJ/kg \approx 0}$$

$$x_{dest,10\text{-}11} = T_0\left(c_p \ln\frac{T_{11}}{T_{10}} - R\ln\frac{P_{11}}{P_{10}}\right) = (290)\left[(1.005)\ln\frac{571.9}{849.8} - (0.287)\ln\left(\frac{1}{4}\right)\right] = \mathbf{-0.04\ kJ/kg \approx 0}$$

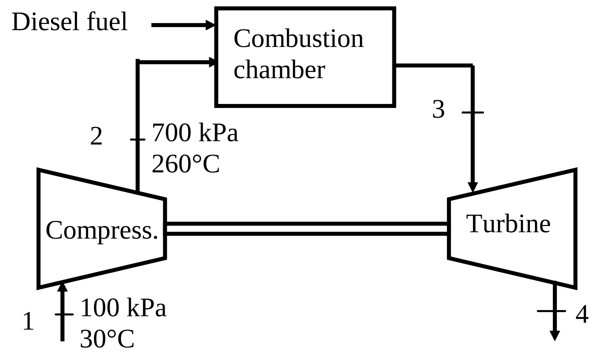
$$x_{dest,12\text{-}13} = T_0\left(c_p \ln\frac{T_{13}}{T_{12}} - R\ln\frac{P_{13}}{P_{12}}\right) = (290)\left[(1.005)\ln\frac{585.7}{870.4} - (0.287)\ln\left(\frac{1}{4}\right)\right] = \mathbf{-0.08\ kJ/kg \approx 0}$$

$$x_{dest,14\text{-}1} = T_0\left(c_p \ln\frac{T_1}{T_{14}} - R\ln\frac{P_1}{P_{14}} + \frac{q_{out,14\text{-}1}}{T_{sink}}\right) = (290)\left[(1.005)\ln\frac{290}{495.9} - 0 + \frac{206.9}{290}\right] = \mathbf{50.6\ kJ/kg}$$

$$x_{dest,2\text{-}3} = x_{dest,4\text{-}5} = T_0\left(c_p \ln\frac{T_3}{T_2} - R\ln\frac{P_3}{P_2} + \frac{q_{out,2\text{-}3}}{T_{sink}}\right) = (290)\left[(1.005)\ln\frac{290}{430.9} - 0 + \frac{141.6}{290}\right] = \mathbf{26.2\ kJ/kg}$$

$$x_{dest,regen} = T_0(\Delta s_{6-7} + \Delta s_{13-14}) = T_0\left(c_p \ln\frac{T_7}{T_6} + c_p \ln\frac{T_{14}}{T_{13}}\right)$$
$$= (290)\left[(1.005)\ln\frac{520.7}{430.9} + (1.005)\ln\frac{495.9}{585.7}\right] = \mathbf{6.66\ kJ/kg}$$

9-157 A gas-turbine plant uses diesel fuel and operates on simple Brayton cycle. The isentropic efficiency of the compressor, the net power output, the back work ratio, the thermal efficiency, and the second-law 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 500ºC = 773 K are c_p = 1.093 kJ/kg·K, c_v = 0.806 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.357 (Table A-2b).

Analysis (*a*) The isentropic efficiency of the compressor may be determined if we first calculate the exit temperature for the isentropic case

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (303\text{ K})\left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)^{(1.357\text{-}1)/1.357} = 505.6\text{ K}$$

$$\eta_C = \frac{T_{2s}-T_1}{T_2-T_1} = \frac{(505.6-303)\text{K}}{(533-303)\text{K}} = \mathbf{0.881}$$

(*b*) The total mass flowing through the turbine and the rate of heat input are

$$\dot{m}_t = \dot{m}_a + \dot{m}_f = \dot{m}_a + \frac{\dot{m}_a}{\text{AF}} = 12.6\text{ kg/s} + \frac{12.6\text{ kg/s}}{60} = 12.6\text{ kg/s} + 0.21\text{ kg/s} = 12.81\text{ kg/s}$$

$$\dot{Q}_{\text{in}} = \dot{m}_f q_{\text{HV}}\eta_c = (0.21\text{ kg/s})(42{,}000\text{ kJ/kg})(0.97) = 8555\text{ kW}$$

The temperature at the exit of combustion chamber is

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_3-T_2) \longrightarrow 8555\text{ kJ/s} = (12.81\text{ kg/s})(1.093\text{ kJ/kg.K})(T_3-533)\text{K} \longrightarrow T_3 = 1144\text{ K}$$

The temperature at the turbine exit is determined using isentropic efficiency relation

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1144\text{ K})\left(\frac{100\text{ kPa}}{700\text{ kPa}}\right)^{(1.357\text{-}1)/1.357} = 685.7\text{ K}$$

$$\eta_T = \frac{T_3-T_4}{T_3-T_{4s}} \longrightarrow 0.85 = \frac{(1144-T_4)\text{K}}{(1144-685.7)\text{K}} \longrightarrow T_4 = 754.4\text{ K}$$

The net power and the back work ratio are

$$\dot{W}_{\text{C,in}} = \dot{m}_a c_p(T_2-T_1) = (12.6\text{ kg/s})(1.093\text{ kJ/kg.K})(533-303)\text{K} = 3168\text{ kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}c_p(T_3-T_4) = (12.81\text{ kg/s})(1.093\text{ kJ/kg.K})(1144-754.4)\text{K} = 5455\text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{C,in}} = 5455-3168 = \mathbf{2287\text{ kW}}$$

$$r_{\text{bw}} = \frac{\dot{W}_{\text{C,in}}}{\dot{W}_{\text{T,out}}} = \frac{3168\text{ kW}}{5455\text{ kW}} = \mathbf{0.581}$$

(*c*) The thermal efficiency is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{2287\text{ kW}}{8555\text{ kW}} = \mathbf{0.267}$$

The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). The maximum temperature for the cycle can be taken to be the turbine inlet temperature. That is,

$$\eta_{\max} = 1-\frac{T_1}{T_3} = 1-\frac{303\text{ K}}{1144\text{ K}} = 0.735$$

and $$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\max}} = \frac{0.267}{0.735} = \mathbf{0.364}$$

9-158 A modern compression ignition engine operates on the ideal dual cycle. The maximum temperature in the cycle, the net work output, the thermal efficiency, the mean effective pressure, the net power output, the second-law efficiency of the cycle, and the rate of exergy of the exhaust gases are to be determined.

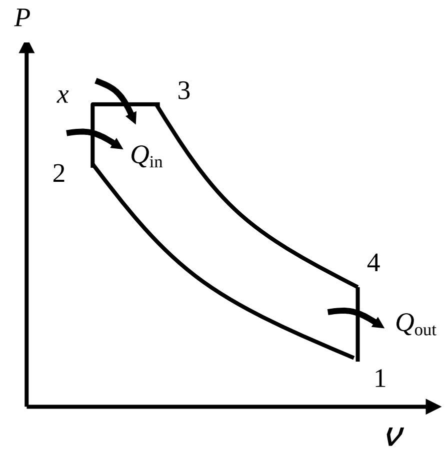
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at 1000 K are c_p = 1.142 kJ/kg·K, c_v = 0.855 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.336 (Table A-2b).

Analysis (*a*) The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

$$r = \frac{\boldsymbol{V}_c + \boldsymbol{V}_d}{\boldsymbol{V}_c} \longrightarrow 16 = \frac{\boldsymbol{V}_c + 0.0018\,\text{m}^3}{\boldsymbol{V}_c} \longrightarrow \boldsymbol{V}_c = 0.00012\,\text{m}^3 = \boldsymbol{V}_2 = \boldsymbol{V}_x$$

$$\boldsymbol{V}_1 = \boldsymbol{V}_c + \boldsymbol{V}_d = 0.00012 + 0.0018 = 0.00192\,\text{m}^3 = \boldsymbol{V}_4$$



Process 1-2: Isentropic compression

$$T_2 = T_1\left(\frac{\boldsymbol{v}_1}{\boldsymbol{v}_2}\right)^{k-1} = (343\text{ K})(16)^{1.336\text{-}1} = 870.7\text{ K}$$

$$P_2 = P_1\left(\frac{\boldsymbol{v}_1}{\boldsymbol{v}_2}\right)^{k} = (95\text{ kPa})(16)^{1.336} = 3859\text{ kPa}$$

Process 2-x and x-3: Constant-volume and constant pressure heat addition processes:

$$T_x = T_2\frac{P_x}{P_2} = (870.7\text{ K})\frac{7500\text{ kPa}}{3859\text{ kPa}} = 1692\text{ K}$$

$$q_{2\text{-}x} = c_v(T_x - T_2) = (0.855\text{ kJ/kg.K})(1692 - 870.7)\text{K} = 702.6\text{ kJ/kg}$$

$$q_{2-x} = q_{x\text{-}3} = c_p(T_3 - T_x) \longrightarrow 702.6\text{ kJ/kg} = (0.855\text{ kJ/kg.K})(T_3 - 1692)\text{K} \longrightarrow T_3 = \mathbf{2308\ K}$$

(*b*) $$q_{\text{in}} = q_{2-x} + q_{x\text{-}3} = 702.6 + 702.6 = 1405\text{ kJ/kg}$$

$$\boldsymbol{V}_3 = \boldsymbol{V}_x\frac{T_3}{T_x} = (0.00012\,\text{m}^3)\frac{2308\text{ K}}{1692\text{ K}} = 0.0001636\,\text{m}^3$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{\boldsymbol{V}_3}{\boldsymbol{V}_4}\right)^{k-1} = (2308\text{ K})\left(\frac{0.0001636\,\text{m}^3}{0.00192\,\text{m}^3}\right)^{1.336\text{-}1} = 1009\text{ K}$$

$$P_4 = P_3\left(\frac{\boldsymbol{V}_3}{\boldsymbol{V}_4}\right)^{k} = (7500\text{ kPa})\left(\frac{0.0001636\,\text{m}^3}{0.00192\,\text{m}^3}\right)^{1.336} = 279.4\text{ kPa}$$

Process 4-1: constant voume heat rejection.

$$q_{\text{out}} = c_v(T_4 - T_1) = (0.855\text{ kJ/kg}\cdot\text{K})(1009 - 343)\text{K} = 569.3\text{ kJ/kg}$$

The net work output and the thermal efficiency are

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 1405 - 569.3 = \mathbf{835.8\ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{835.8\text{ kJ/kg}}{1405\text{ kJ/kg}} = 0.5948 = \mathbf{59.5\%}$$

(*c*) The mean effective pressure is determined to be

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95\,\text{kPa})(0.00192\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(343\,\text{K})} = 0.001853\,\text{kg}$$

$$\text{MEP} = \frac{m w_{\text{net,out}}}{V_1 - V_2} = \frac{(0.001853\,\text{kg})(835.8\text{kJ/kg})}{(0.00192 - 0.00012)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{860.4\ kPa}$$

(*d*) The power for engine speed of 3500 rpm is

$$\dot{W}_{\text{net}} = m w_{\text{net}} \frac{\dot{n}}{2} = (0.001853\,\text{kg})(835.8\,\text{kJ/kg})\frac{2200\,(\text{rev/min})}{(2\,\text{rev/cycle})}\left(\frac{1\,\text{min}}{60\,\text{s}}\right) = \mathbf{28.39\ kW}$$

Note that there are two revolutions in one cycle in four-stroke engines.

(*e*) The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). We take the dead state temperature and pressure to be 25ºC and 100 kPa.

$$\eta_{\max} = 1 - \frac{T_0}{T_3} = 1 - \frac{(25+273)\,\text{K}}{2308\,\text{K}} = 0.8709$$

and

$$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\max}} = \frac{0.5948}{0.8709} = 0.683 = \mathbf{68.3\%}$$

The rate of exergy of the exhaust gases is determined as follows

$$x_4 = u_4 - u_0 - T_0(s_4 - s_0) = c_v(T_4 - T_0) - T_0\left[c_p \ln\frac{T_4}{T_0} - R\ln\frac{P_4}{P_0}\right]$$

$$= (0.855)(1009 - 298) - (298)\left[(1.142\,\text{kJ/kg.K})\ln\frac{1009}{298} - (0.287\,\text{kJ/kg.K})\ln\frac{279.4}{100}\right] = 285.0\,\text{kJ/kg}$$

$$\dot{X}_4 = m x_4 \frac{\dot{n}}{2} = (0.001853\,\text{kg})(285.0\,\text{kJ/kg})\frac{2200\,(\text{rev/min})}{(2\,\text{rev/cycle})}\left(\frac{1\,\text{min}}{60\,\text{s}}\right) = \mathbf{9.683\ kW}$$

Review Problems

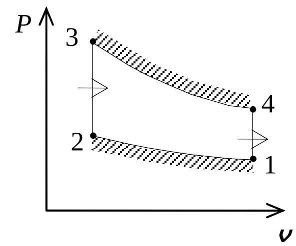
9-159 An Otto cycle with a compression ratio of 7 is considered. The thermal efficiency is to be determined using constant and variable specific heats.

Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are R = 0.287 kPa·m^3/kg·K, c_p = 1.005 kJ/kg·K, c_v = 0.718 kJ/kg·K, and k = 1.4 (Table A-2a).

Analysis (*a*) Constant specific heats:

$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}} = 1 - \frac{1}{7^{1.4-1}} = 0.5408 = \mathbf{54.1\%}$$



(b) Variable specific heats: (using air properties from Table A-17)

Process 1-2: isentropic compression.

$$T_1 = 288\text{ K} \longrightarrow \begin{array}{l} u_1 = 205.48\text{ kJ/kg} \\ v_{r1} = 688.1 \end{array}$$

$$v_{r2} = \frac{v_2}{v_1} v_{r2} = \frac{1}{r} v_{r2} = \frac{1}{7}(688.1) = 98.3 \longrightarrow u_2 = 447.62\text{ kJ/kg}$$

Process 2-3: v = constant heat addition.

$$T_3 = 1273\text{ K} \longrightarrow \begin{array}{l} u_3 = 998.51\text{ kJ/kg} \\ v_{r3} = 12.045 \end{array}$$

$$q_{in} = u_3 - u_2 = 998.51 - 447.62 = 550.89\text{ kJ/kg}$$

Process 3-4: isentropic expansion.

$$v_{r4} = \frac{v_4}{v_3} v_{r3} = r v_{r3} = (7)(12.045) = 84.32 \longrightarrow u_4 = 475.54\text{ kJ/kg}$$

Process 4-1: v = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 475.54 - 205.48 = 270.06\text{ kJ/kg}$$

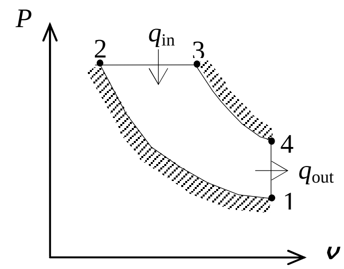
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{270.06\text{ kJ/kg}}{550.89\text{ kJ/kg}} = 0.5098 = \mathbf{51.0\%}$$

9-160E An ideal diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency is to be determined using constant and variable specific heats.

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, R = 0.06855 Btu/lbm·R, and $k = 1.4$ (Table A-2Ea).

Analysis (a) Constant specific heats:



Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (505\ \mathrm{R})(20)^{0.4} = 1673.8\ \mathrm{R}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2260\ \mathrm{R}}{1673.8\ \mathrm{R}} = 1.350$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1.350 v_2}{v_4}\right)^{k-1} = T_3\left(\frac{1.350}{r}\right)^{k-1} = (2260\ \mathrm{R})\left(\frac{1.350}{20}\right)^{0.4} = 768.8\ \mathrm{R}$$

$$q_{in} = h_3 - h_2 = c_p(T_3 - T_2) = (0.240\ \mathrm{Btu/lbm \cdot R})(2260 - 1673.8)\mathrm{R} = 140.7\ \mathrm{Btu/lbm}$$

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) = (0.171\ \mathrm{Btu/lbm \cdot R})(768.8 - 505)\mathrm{R} = 45.11\ \mathrm{Btu/lbm}$$

$$w_{net,out} = q_{in} - q_{out} = 140.7 - 45.11 = 95.59\ \mathrm{Btu/lbm}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{95.59\ \mathrm{Btu/lbm}}{140.7\ \mathrm{Btu/lbm}} = 0.6794 = \mathbf{67.9\%}$$

(b) Variable specific heats: (using air properties from Table A-17)

Process 1-2: isentropic compression.

$$T_1 = 505\ \mathrm{R} \longrightarrow \begin{array}{l} u_1 = 86.06\ \mathrm{Btu/lbm} \\ v_{r1} = 170.82 \end{array}$$

$$v_{r2} = \frac{v_2}{v_1} v_{r1} = \frac{1}{r} v_{r1} = \frac{1}{20}(170.82) = 8.541 \longrightarrow \begin{array}{l} T_2 = 1582.3\ \mathrm{R} \\ h_2 = 391.01\ \mathrm{Btu/lbm} \end{array}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2260\ \mathrm{R}}{1582.3\ \mathrm{R}} = 1.428$$

$$T_3 = 2260\ \mathrm{R} \longrightarrow \begin{array}{l} h_3 = 577.52\ \mathrm{Btu/lbm} \\ v_{r3} = 2.922 \end{array}$$

$$q_{in} = h_3 - h_2 = 577.52 - 391.01 = 186.51\ \mathrm{Btu/lbm}$$

Process 3-4: isentropic expansion.

$$v_{r4} = \frac{v_4}{v_3} v_{r3} = \frac{v_4}{1.428 v_2} v_{r3} = \frac{r}{1.428} v_{r3} = \frac{20}{1.428}(2.922) = 40.92 \longrightarrow u_4 = 152.65\ \mathrm{Btu/lbm}$$

Process 4-1: v = constant heat rejection.

$$q_{out} = u_4 - u_1 = 152.65 - 86.06 = 66.59\ \mathrm{Btu/lbm}$$

Then $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{66.59\ \mathrm{Btu/lbm}}{186.51\ \mathrm{Btu/lbm}} = 0.6430 = \mathbf{64.3\%}$$

9-161E A simple ideal Brayton cycle with air as the working fluid operates between the specified temperature limits. The net work is to be determined using constant and variable specific heats.

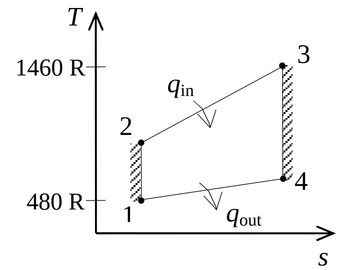
Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible.

Properties The properties of air at room temperature are c_p = 0.240 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis (*a*) Constant specific heats:

$$T_2 = T_1 r_p^{(k-1)/k} = (480\ \text{R})(12)^{0.4/1.4} = 976.3\ \text{R}$$

$$T_4 = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = (1460\ \text{R})\left(\frac{1}{12}\right)^{0.4/1.4} = 717.8\ \text{R}$$



$$\begin{aligned} w_{\text{net}} &= w_{\text{turb}} - w_{\text{comp}} \\ &= c_p(T_3 - T_4) - c_p(T_2 - T_1) \\ &= c_p(T_3 - T_4 + T_1 - T_2) \\ &= (0.240\ \text{Btu/lbm}\cdot\text{R})(1460 - 717.8 + 480 - 976.3)\text{R} \\ &= \mathbf{59.0\ Btu/lbm} \end{aligned}$$

(*b*) Variable specific heats: (using air properties from Table A-17E)

$$T_1 = 480\ \text{R} \longrightarrow \begin{array}{l} h_1 = 114.69\ \text{Btu/lbm} \\ P_{r1} = 0.9182 \end{array}$$

$$P_{r2} = \frac{P_2}{P_1} P_{r1} = (12)(0.9182) = 11.02 \longrightarrow h_2 = 233.63\ \text{Btu/lbm}$$

$$T_3 = 1460\ \text{R} \longrightarrow \begin{array}{l} h_3 = 358.63\ \text{Btu/lbm} \\ P_{r3} = 50.40 \end{array}$$

$$P_{r4} = \frac{P_4}{P_3} P_{r3} = \left(\frac{1}{12}\right)(50.40) = 4.12 \longrightarrow h_4 = 176.32\ \text{Btu/lbm}$$

$$\begin{aligned} w_{\text{net}} &= w_{\text{turb}} - w_{\text{comp}} \\ &= (h_3 - h_4) - (h_2 - h_1) \\ &= (358.63 - 176.32) - (233.63 - 114.69) \\ &= \mathbf{63.4\ Btu/lbm} \end{aligned}$$

9-162 A turbocharged four-stroke V-16 diesel engine produces 3500 hp at 1200 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 (revolutions), we have

(*a*)

$$w_{\text{mechanical}} = \frac{\text{Total power produced}}{(\text{No. of cylinders})(\text{No. of mechanical cycles})} = \frac{3500\ \text{hp}}{(16\ \text{cylinders})(1200\ \text{rev/min})}\left(\frac{42.41\ \text{Btu/min}}{1\ \text{hp}}\right) = \mathbf{7.73\ Btu/cyl \cdot mech\ cycle} \quad (= 8.16\ \text{kJ/cyl} \cdot \text{mech cycle})$$

(*b*)

$$w_{\text{thermodynamic}} = \frac{\text{Total power produced}}{(\text{No. of cylinders})(\text{No. of thermodynamic cycles})} = \frac{3500\ \text{hp}}{(16\ \text{cylinders})(1200/2\ \text{rev/min})}\left(\frac{42.41\ \text{Btu/min}}{1\ \text{hp}}\right) = \mathbf{15.46\ Btu/cyl \cdot therm\ cycle} \quad (= 16.31\ \text{kJ/cyl} \cdot \text{therm cycle})$$

9-163 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.

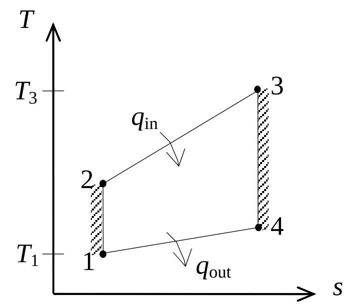
Assumptions **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

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

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1\left(r_p\right)^{(k-1)/k}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k}$$



Setting $T_2 = T_4$ and solving for r_p gives

$$r_p = \left(\frac{T_3}{T_1}\right)^{k/2(k-1)} = \left(\frac{1500\ \text{K}}{300\ \text{K}}\right)^{1.4/0.8} = \mathbf{16.7}$$

Therefore, the compressor and turbine exit temperatures will be equal when the compression ratio is 16.7.

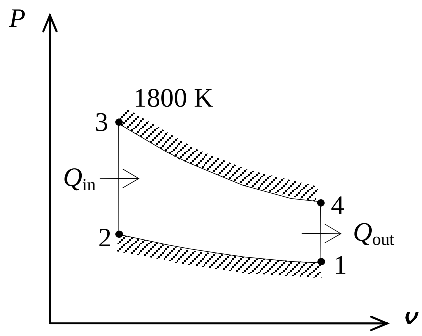
9-164 A four-cylinder spark-ignition engine with a compression ratio of 8 is considered. The amount of heat supplied per cylinder, the thermal efficiency, and the rpm for a net power output of 60 kW are to be determined.

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 gas constant of air is R = 0.287 kJ/kg·K (Table A-1). The properties of air are given in Table A-17.

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

$$T_1 = 310\text{ K} \longrightarrow u_1 = 221.25\text{ kJ/kg}$$
$$\boldsymbol{v}_{r_1} = 572.3$$



$$\boldsymbol{v}_{r_2} = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}\boldsymbol{v}_{r_1} = \frac{1}{r}\boldsymbol{v}_{r_1} = \frac{1}{10.5}(572.3) = 54.50$$
$$\longrightarrow u_2 = 564.29\text{ kJ/kg}$$

Process 2-3: $\boldsymbol{v}$ = constant heat addition.

$$T_3 = 2100\text{ K} \longrightarrow u_3 = 1775.3\text{ kJ/kg}$$
$$\boldsymbol{v}_{r_3} = 2.356$$

$$m = \frac{P_1\boldsymbol{v}_1}{RT_1} = \frac{(98\text{ kPa})(0.0004\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(310\text{ K})} = 4.406\times10^{-4}\text{ kg}$$

$$Q_{\text{in}} = m(u_3 - u_2) = (4.406\times10^{-4}\text{ kg})(1775.3 - 564.29)\text{kJ/kg} = \mathbf{0.5336\text{ kJ}}$$

(b) Process 3-4: isentropic expansion.

$$\boldsymbol{v}_{r_4} = \frac{\boldsymbol{v}_4}{\boldsymbol{v}_3}\boldsymbol{v}_{r_3} = r\boldsymbol{v}_{r_3} = (10.5)(2.356) = 24.74 \longrightarrow u_4 = 764.05\text{ kJ/kg}$$

Process 4-1: $\boldsymbol{v}$ = constant heat rejection.

$$Q_{\text{out}} = m(u_4 - u_1) = (4.406\times10^{-4}\text{ kg})(764.05 - 221.25)\text{kJ/kg} = 0.2392\text{ kJ}$$

$$W_{\text{net}} = Q_{\text{in}} - Q_{\text{out}} = 0.5336 - 0.2392 = 0.2944\text{ kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{0.2944\text{ kJ}}{0.5336\text{ kJ}} = 0.5517 = \mathbf{55.2\%}$$

(c) $$\dot{n} = 2\frac{\dot{W}_{\text{net}}}{n_{\text{cyl}}W_{\text{net,cyl}}} = (2\text{ rev/cycle})\frac{45\text{ kJ/s}}{4\times(0.2944\text{ kJ/cycle})}\left(\frac{60\text{ s}}{1\text{ min}}\right) = \mathbf{4586\text{ rpm}}$$

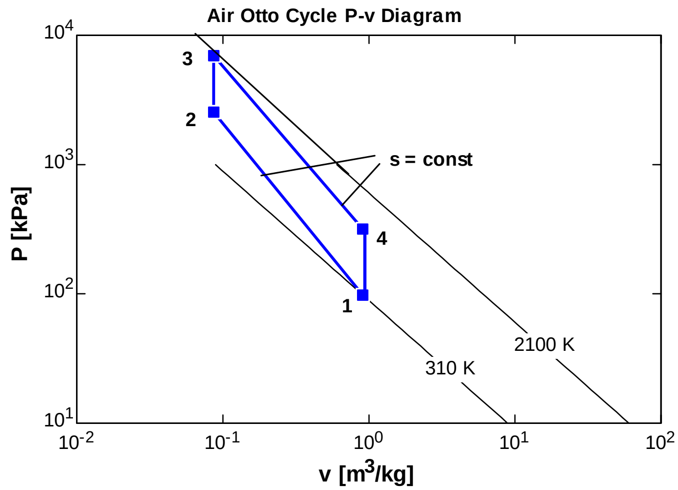
Note that for four-stroke cycles, there are two revolutions per cycle.

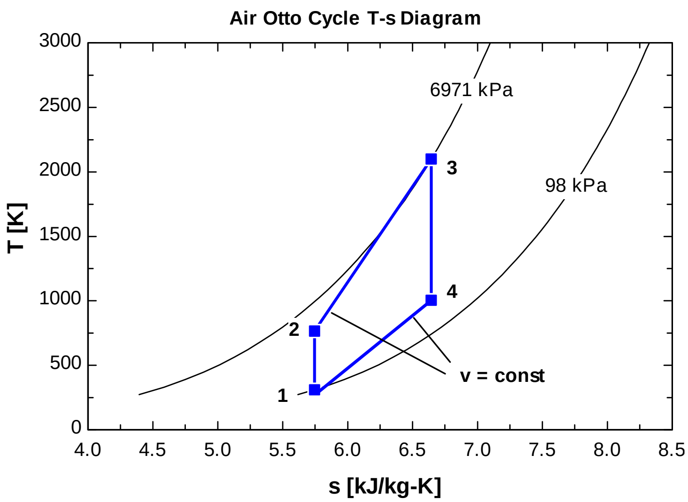
9-165 Problem 9-164 is reconsidered. The effect of the compression ratio net work done and the efficiency of the cycle is to be investigated. Also, the *T-s* and *P-v* diagrams for the cycle are to be plotted.

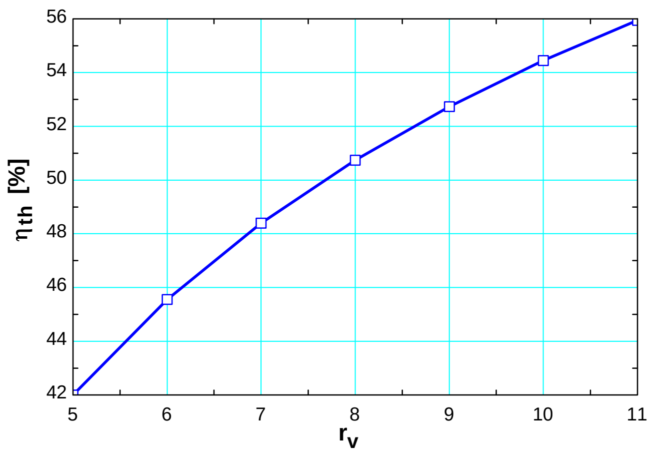
Analysis Using EES, the problem is solved as follows:

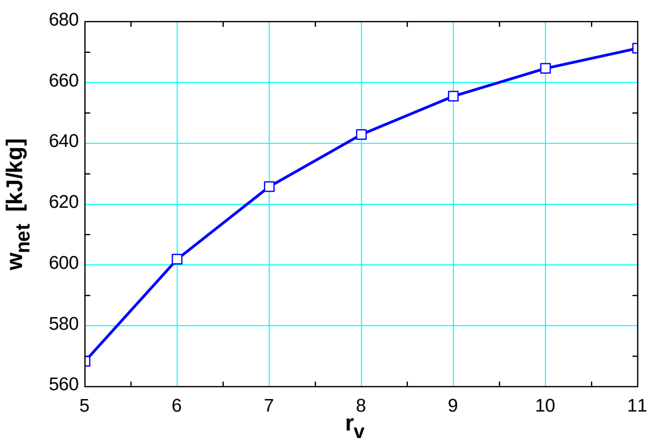
```
"Input Data"
T[1]=(37+273) [K]
P[1]=98 [kPa]
T[3]= 2100 [K]
V_cyl=0.4 [L]*Convert(L, m^3)
r_v=10.5 "Compression ratio"
W_dot_net = 45 [kW]
N_cyl=4 "number of cyclinders"
v[1]/v[2]=r_v

"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]=R*T[1]
R=0.287 [kJ/kg-K]
"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]=R*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]=R*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*Convert(, %) "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"
```

r_v	η_{th} [%]	w_{net} [kJ/kg]
5	42	568.3
6	45.55	601.9
7	48.39	625.7
8	50.74	642.9
9	52.73	655.5
10	54.44	664.6
11	55.94	671.2







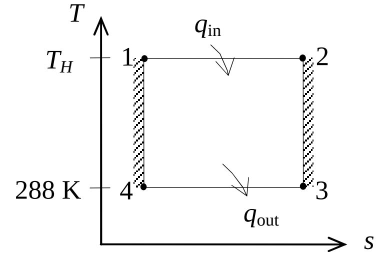


9-166 An ideal gas Carnot cycle with helium as the working fluid is considered. The pressure ratio, compression ratio, and minimum temperature of the energy source are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Helium is an ideal gas with constant specific heats.

Properties The specific heat ratio of helium is k = 1.667 (Table A-2a).

Analysis From the definition of the thermal efficiency of a Carnot heat engine,



$$\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} \longrightarrow T_H = \frac{T_L}{1-\eta_{\text{th,Carnot}}} = \frac{(15+273)\text{ K}}{1-0.50} = \mathbf{576\ K}$$

An isentropic process for an ideal gas is one in which Pv^k remains constant. Then, the pressure ratio is

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{k/(k-1)} = \left(\frac{576\text{ K}}{288\text{ K}}\right)^{1.667/(1.667-1)} = \mathbf{5.65}$$

Based on the process equation, the compression ratio is

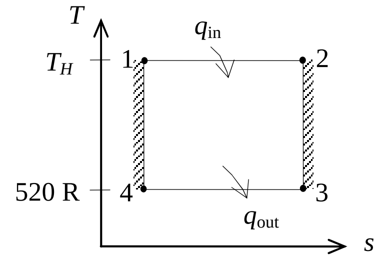
$$\frac{v_1}{v_2} = \left(\frac{P_2}{P_1}\right)^{1/k} = (5.65)^{1/1.667} = \mathbf{2.83}$$

9-167E An ideal gas Carnot cycle with helium as the working fluid is considered. The pressure ratio, compression ratio, and minimum temperature of the energy-source reservoir are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Helium is an ideal gas with constant specific heats.

Properties The specific heat ratio of helium is k = 1.667 (Table A-2Ea).

Analysis From the definition of the thermal efficiency of a Carnot heat engine,



$$\eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} \longrightarrow T_H = \frac{T_L}{1-\eta_{\text{th,Carnot}}} = \frac{(60+460)\text{ R}}{1-0.60} = \mathbf{1300\ R}$$

An isentropic process for an ideal gas is one in which Pv^k remains constant. Then, the pressure ratio is

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{k/(k-1)} = \left(\frac{1300\text{ R}}{520\text{ R}}\right)^{1.667/(1.667-1)} = \mathbf{9.88}$$

Based on the process equation, the compression ratio is

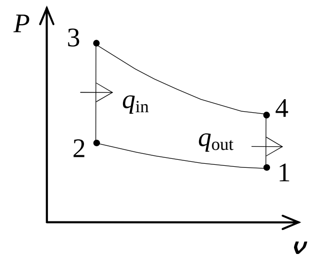
$$\frac{v_1}{v_2} = \left(\frac{P_2}{P_1}\right)^{1/k} = (9.88)^{1/1.667} = \mathbf{3.95}$$

9-168 The compression ratio required for an ideal Otto cycle to produce certain amount of work when consuming a given amount of fuel 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. **4** The combustion efficiency is 100 percent.

Properties The properties of air at room temperature are k = 1.4 (Table A-2).

Analysis The heat input to the cycle for 0.043 grams of fuel consumption is



$$Q_{\text{in}} = m_{\text{fuel}} q_{\text{HV}} = (0.035 \times 10^{-3}\ \text{kg})(43{,}000\ \text{kJ/kg}) = 1.505\ \text{kJ}$$

The thermal efficiency is then

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{1\ \text{kJ}}{1.505\ \text{kJ}} = 0.6645$$

From the definition of thermal efficiency, we obtain the required compression ratio to be

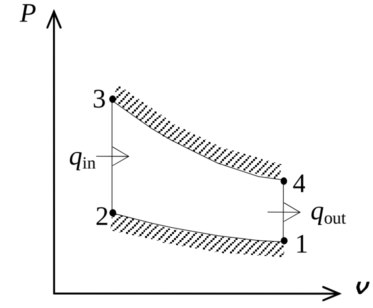
$$\eta_{\text{th}} = 1 - \frac{1}{r^{k-1}} \longrightarrow r = \frac{1}{(1-\eta_{\text{th}})^{1/(k-1)}} = \frac{1}{(1-0.6645)^{1/(1.4-1)}} = \mathbf{15.3}$$

9-169 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 gas constant of air is R = 0.287 kJ/kg·K (Table A-1). The properties of air are given in Table A-17.

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



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

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{9.2}(621.2) = 67.52 \longrightarrow \begin{array}{l} T_2 = 708.3 \text{ K} \\ u_2 = 518.9 \text{ kJ/kg} \end{array}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1} P_1 = (9.2)\left(\frac{708.3 \text{ K}}{300 \text{ K}}\right)(98 \text{ kPa}) = 2129 \text{ kPa}$$

Process 2-3: v = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{P_3}{P_2} T_2 = 2T_2 = (2)(708.3) = 1416.6 \text{ K} \longrightarrow \begin{array}{l} u_3 = 1128.7 \text{ kJ/kg} \\ v_{r_3} = 8.593 \end{array}$$

$$q_{in} = u_3 - u_2 = 1128.7 - 518.9 = \mathbf{609.8 \text{ kJ/kg}}$$

(*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*)
$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{336.1 \text{ kJ/kg}}{609.8 \text{ kJ/kg}} = \mathbf{55.1\%}$$

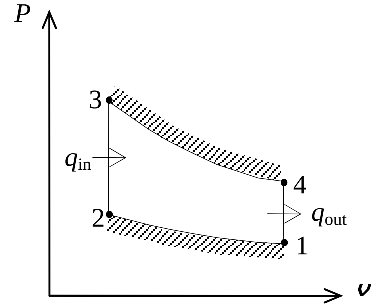
(*d*)
$$v_{max} = v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})}{98 \text{ kPa}} = 0.879 \text{ m}^3/\text{kg}$$

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

$$\text{MEP} = \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1(1-1/r)} = \frac{336.1 \text{ kJ/kg}}{(0.879 \text{ m}^3/\text{kg})(1-1/9.2)}\left(\frac{1 \text{ kPa}\cdot\text{m}^3}{1 \text{ kJ}}\right) = \mathbf{429 \text{ kPa}}$$

9-170 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-2a).

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.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{P_3}{P_2}T_2 = 2T_2 = (2)(728.8) = 1457.6\text{ K}$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2) = (0.718\text{ kJ/kg}\cdot\text{K})(1457.6 - 728.8)\text{K} = \mathbf{523.3\ kJ/kg}$$

(*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.

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(600 - 300)\text{K} = 215.4\text{ kJ/kg}$$

$$w_{net} = q_{in} - q_{out} = 523.3 - 215.4 = \mathbf{307.9\ kJ/kg}$$

(*c*) $$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{307.9\text{ kJ/kg}}{523.3\text{ kJ/kg}} = \mathbf{58.8\%}$$

(*d*) $$v_{max} = v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{98\text{ kPa}} = 0.879\text{ m}^3/\text{kg}$$

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

$$\text{MEP} = \frac{w_{net}}{v_1 - v_2} = \frac{w_{net}}{v_1(1-1/r)} = \frac{307.9\text{ kJ/kg}}{(0.879\text{ m}^3/\text{kg})(1-1/9.2)}\left(\frac{1\text{ kPa}\cdot\text{m}^3}{1\text{ kJ}}\right) = \mathbf{393\ kPa}$$

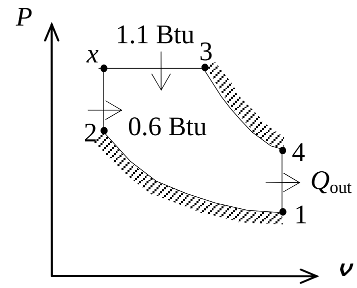
9-171E An ideal dual cycle with air as the working fluid with a compression ratio of 12 is considered. The thermal efficiency of the cycle is to be determined.

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 The mass of air is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(14.7 \text{ psia})(98/1728 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(580 \text{ R})} = 0.003881 \text{ lbm}$$



Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (580 \text{ R})(14)^{0.4} = 1667 \text{ R}$$

Process 2-x: v = *constant* heat addition,

$$Q_{2-x,\text{in}} = m(u_x - u_2) = mc_v(T_x - T_2)$$

$$0.6 \text{ Btu} = (0.003881 \text{ lbm})(0.171 \text{ Btu/lbm}\cdot\text{R})(T_x - 1667)\text{R} \longrightarrow T_x = 2571 \text{ R}$$

Process x-3: P = *constant* heat addition.

$$Q_{x-3,\text{in}} = m(h_3 - h_x) = mc_p(T_3 - T_x)$$

$$1.1 \text{ Btu} = (0.003881 \text{ lbm})(0.240 \text{ Btu/lbm}\cdot\text{R})(T_3 - 2571)\text{R} \longrightarrow T_3 = 3752 \text{ R}$$

$$\frac{P_3 V_3}{T_3} = \frac{P_x V_x}{T_x} \longrightarrow r_c = \frac{V_3}{V_x} = \frac{T_3}{T_x} = \frac{3752 \text{ R}}{2571 \text{ R}} = 1.459$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{1.459V_1}{V_4}\right)^{k-1} = T_3\left(\frac{1.459}{r}\right)^{k-1} = (3752 \text{ R})\left(\frac{1.459}{14}\right)^{0.4} = 1519 \text{ R}$$

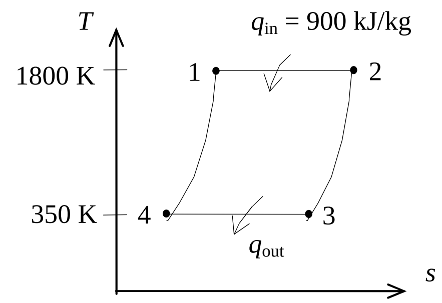
Process 4-1: v = constant heat rejection.

$$Q_\text{out} = m(u_4 - u_1) = mc_v(T_4 - T_1)$$
$$= (0.003881 \text{ lbm})(0.171 \text{ Btu/lbm}\cdot\text{R})(1519 - 580)\text{R} = 0.6229 \text{ Btu}$$

$$\eta_\text{th} = 1 - \frac{Q_\text{out}}{Q_\text{in}} = 1 - \frac{0.6229 \text{ Btu}}{1.7 \text{ Btu}} = 0.6336 = \mathbf{63.4\%}$$

9-172 An ideal Stirling cycle with air as the working fluid is considered. The maximum pressure in the cycle and the net work output 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 (*a*) The entropy change during process 1-2 is

$$s_2 - s_1 = \frac{q_{12}}{T_H} = \frac{900 \text{ kJ/kg}}{1800 \text{ K}} = 0.5 \text{ kJ/kg} \cdot \text{K}$$

and

$$s_2 - s_1 = c_v \ln \frac{T_2}{T_1}^{\nearrow 0} + R \ln \frac{v_2}{v_1} \longrightarrow 0.5 \text{ kJ/kg} \cdot \text{K} = (0.287 \text{ kJ/kg} \cdot \text{K}) \ln \frac{v_2}{v_1} \longrightarrow \frac{v_2}{v_1} = 5.710$$

$$\frac{P_3 v_3}{T_3} = \frac{P_1 v_1}{T_1} \longrightarrow P_1 = P_3 \frac{v_3}{v_1}\frac{T_1}{T_3} = P_3 \frac{v_2}{v_1}\frac{T_1}{T_3} = (200 \text{ kPa})(5.710)\left(\frac{1800 \text{ K}}{350 \text{ K}}\right) = \mathbf{5873 \text{ kPa}}$$

(*b*) The net work output is

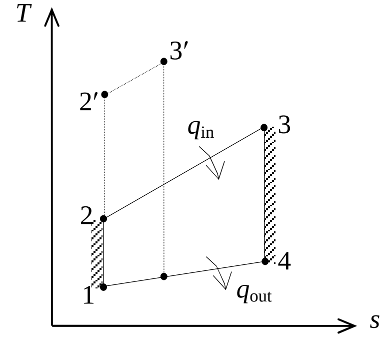
$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = \left(1 - \frac{T_L}{T_H}\right) q_{\text{in}} = \left(1 - \frac{350 \text{ K}}{1800 \text{ K}}\right)(900 \text{ kJ/kg}) = \mathbf{725 \text{ kJ/kg}}$$

9-173 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-17.

Analysis 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*) $\Delta w_{net} = 380.96 - 339.46 = \mathbf{41.5 \text{ kJ/kg}}$ (increase)

(*b*) $\Delta \eta_{th} = 48.5\% - 37.9\% = \mathbf{10.6\%}$ (increase)

9-174 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.

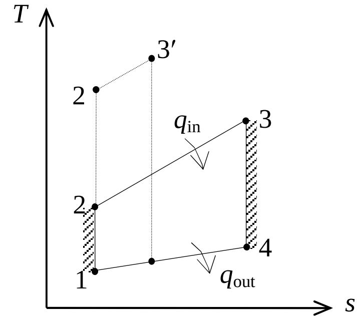
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

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,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(12)^{0.4/1.4} = 610.2\text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{ K})\left(\frac{1}{12}\right)^{0.4/1.4} = 639.2\text{ K}$$

$$\begin{aligned} q_{in} &= h_3 - h_2 = c_p(T_3 - T_2) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(1300 - 610.2)\text{K} = 693.2\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{out} &= h_4 - h_1 = c_p(T_4 - T_1) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(639.2 - 300)\text{K} = 340.9\text{ kJ/kg} \end{aligned}$$

$$w_{net} = q_{in} - q_{out} = 693.2 - 340.9 = 352.3\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{352.3\text{ kJ/kg}}{693.2\text{ kJ/kg}} = 50.8\%$$

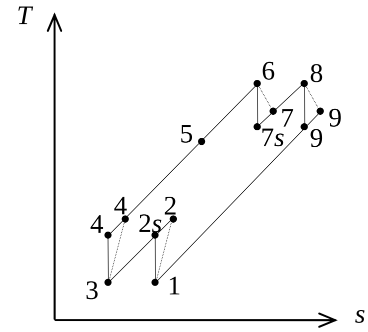
Thus,

(*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)

9-175 A regenerative gas-turbine engine operating with two stages of compression and two stages of expansion is considered. The back work ratio 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 The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.

$$T_{4s} = T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(4)^{0.4/1.4} = 445.8\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_4 = T_2 = T_1 + (T_{2s}-T_1)/\eta_C$$
$$= 300 + (445.8-300)/(0.78)$$
$$= 486.9\text{ K}$$

$$T_{9s} = T_{7s} = T_6\left(\frac{P_7}{P_6}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 942.1\text{ K}$$

$$\eta_T = \frac{h_6-h_7}{h_6-h_{7s}} = \frac{c_p(T_6-T_7)}{c_p(T_6-T_{7s})} \longrightarrow T_9 = T_7 = T_6 - \eta_T(T_6-T_{7s})$$
$$= 1400 - (0.86)(1400-942.1)$$
$$= 1006\text{ K}$$

$$\varepsilon = \frac{h_5-h_4}{h_9-h_4} = \frac{c_p(T_5-T_4)}{c_p(T_9-T_4)} \longrightarrow T_5 = T_4 + \varepsilon(T_9-T_4)$$
$$= 486.9 + (0.75)(1006-486.9)$$
$$= 876.4\text{ K}$$

$$w_{\text{C,in}} = 2(h_2-h_1) = 2c_p(T_2-T_1) = 2(1.005\text{ kJ/kg}\cdot\text{K})(486.9-300)\text{K} = 375.7\text{ kJ/kg}$$

$$w_{\text{T,out}} = 2(h_6-h_7) = 2c_p(T_6-T_7) = 2(1.005\text{ kJ/kg}\cdot\text{K})(1400-1006)\text{K} = 791.5\text{ kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{375.7\text{ kJ/kg}}{791.5\text{ kJ/kg}} = \mathbf{0.475}$$

$$q_{\text{in}} = (h_6-h_5)+(h_8-h_7) = c_p[(T_6-T_5)+(T_8-T_7)]$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})[(1400-876.4)+(1400-1006)]\text{K} = 922.0\text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 791.5 - 375.7 = 415.8\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{415.8\text{ kJ/kg}}{922.0\text{ kJ/kg}} = 0.451 = \mathbf{45.1\%}$$

9-176 Problem 9-175 is reconsidered. The effect of the isentropic efficiencies for the compressor and turbine and regenerator effectiveness on net work done and the heat supplied to the cycle is to be investigated. Also, the *T-s* diagram for the cycle is to be plotted.

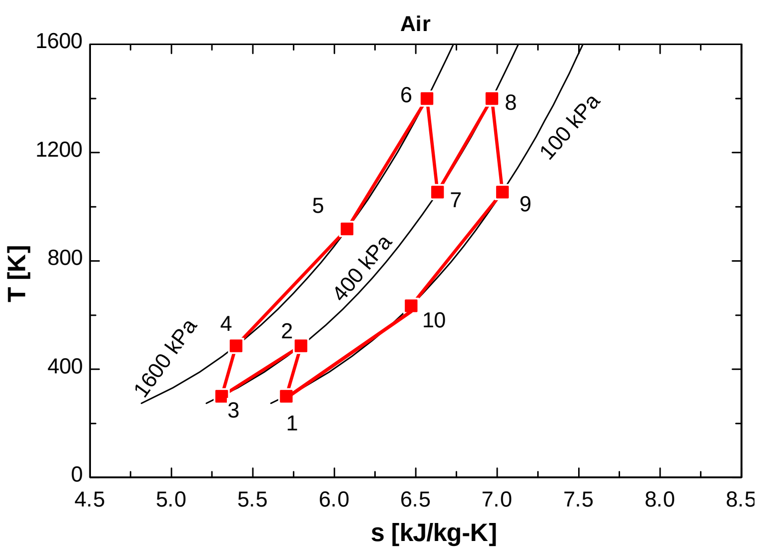
Analysis Using EES, the problem is solved as follows:

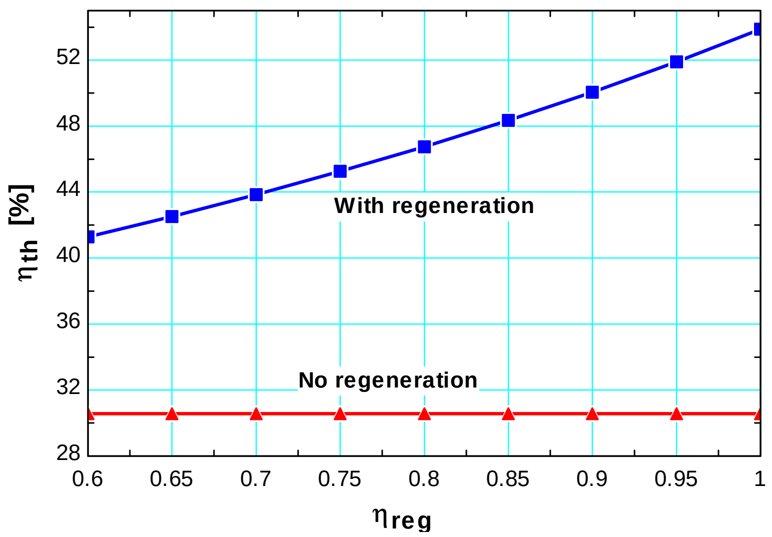
```
"Input data"
T[6] = 1400 [K]
T[8] = T[6]
Pratio = 4
T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = T[1]
Eta_reg = 0.75 "Regenerator effectiveness"
Eta_c =0.78 "Compressor isentorpic efficiency"
Eta_t =0.86 "Turbine isentropic efficiency"

"LP Compressor:"
"Isentropic Compressor anaysis"
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_LP/w_comp_LP
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the LP compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[1] + w_compisen_LP = 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_LP = h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

"HP Compressor:"
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 HP compressor"
P[4] = Pratio*P[3]
P[3] = P[2]
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])
"T_s[4] is the isentropic value of T[4] at compressor exit"
Eta_c = w_compisen_HP/w_comp_HP
"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[3] + w_compisen_HP = h_s[4]
h[3]=ENTHALPY(Air,T=T[3])
h_s[4]=ENTHALPY(Air,T=T_s[4])

"Actual compressor analysis:"
h[3] + w_comp_HP = h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])
```

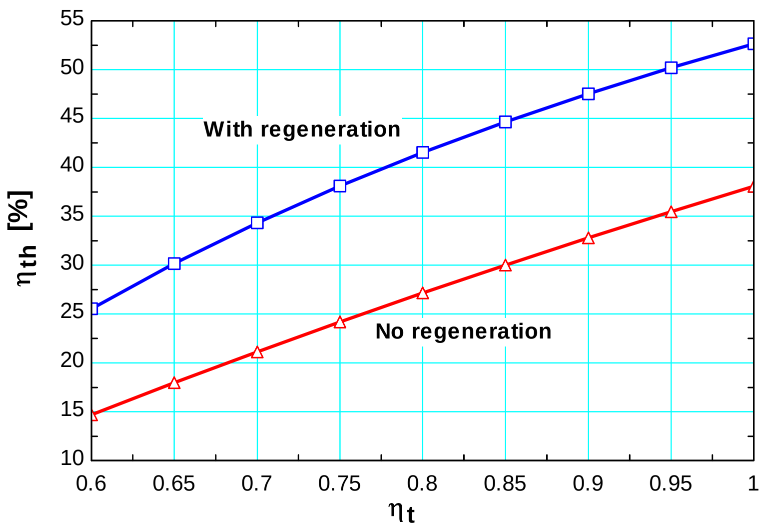
```
"Intercooling heat loss:"
h[2] = q_out_intercool + h[3]

"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[4] + q_in_noreg = h[6]

h[6]=ENTHALPY(Air,T=T[6])
P[6]=P[4]"process 4-6 is SSSF constant pressure"

"HP Turbine analysis"

s[6]=ENTROPY(Air,T=T[6],P=P[6])
s_s[7]=s[6] "For the ideal case the entropies are constant across the turbine"
P[7] = P[6] /Pratio
s_s[7]=ENTROPY(Air,T=T_s[7],P=P[7])"T_s[7] is the isentropic value of T[7] at HP turbine exit"
Eta_t = w_turb_HP /w_turbisen_HP "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[6] = w_turbisen_HP + h_s[7]
h_s[7]=ENTHALPY(Air,T=T_s[7])
"Actual Turbine analysis:"
h[6] = w_turb_HP + h[7]
h[7]=ENTHALPY(Air,T=T[7])
s[7]=ENTROPY(Air,T=T[7], P=P[7])

"Reheat Q_in:"
h[7] + q_in_reheat = h[8]
h[8]=ENTHALPY(Air,T=T[8])

"HL Turbine analysis"

P[8]=P[7]
s[8]=ENTROPY(Air,T=T[8],P=P[8])
s_s[9]=s[8] "For the ideal case the entropies are constant across the turbine"
P[9] = P[8] /Pratio
s_s[9]=ENTROPY(Air,T=T_s[9],P=P[9])"T_s[9] is the isentropic value of T[9] at LP turbine exit"
Eta_t = w_turb_LP /w_turbisen_LP "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[8] = w_turbisen_LP + h_s[9]
h_s[9]=ENTHALPY(Air,T=T_s[9])
"Actual Turbine analysis:"
h[8] = w_turb_LP + h[9]
h[9]=ENTHALPY(Air,T=T[9])
s[9]=ENTROPY(Air,T=T[9], P=P[9])

"Cycle analysis"
w_net=w_turb_HP+w_turb_LP - w_comp_HP - w_comp_LP
q_in_total_noreg=q_in_noreg+q_in_reheat
Eta_th_noreg=w_net/(q_in_total_noreg)*Convert(, %) "[%]"    "Cycle thermal efficiency"
Bwr=(w_comp_HP + w_comp_LP)/(w_turb_HP+w_turb_LP)"Back work ratio"

"With the regenerator, the heat added in the external heat exchanger is"
h[5] + q_in_withreg = h[6]
h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[4]
```

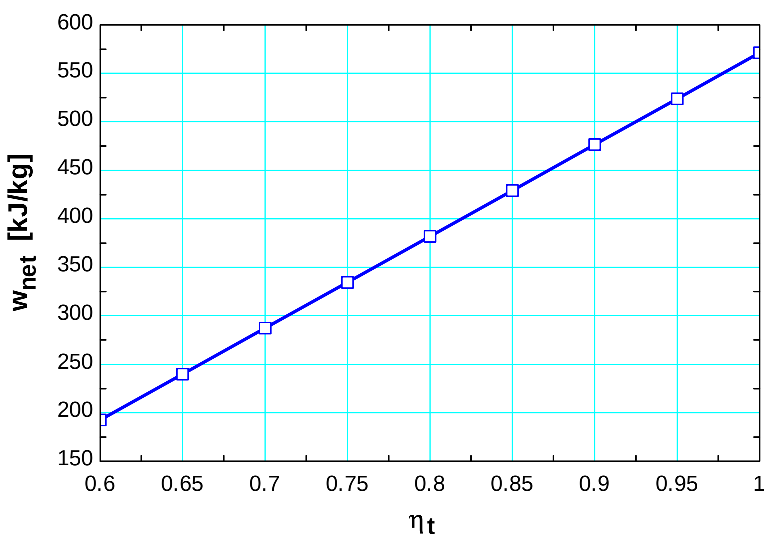
```
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[4])/(h[9]-h[4])
"Energy balance on regenerator gives h[10] and thus T[10] as:"
h[4] + h[9]=h[5]  + h[10]
h[10]=ENTHALPY(Air, T=T[10])
s[10]=ENTROPY(Air,T=T[10], P=P[10])
P[10]=P[9]

"Cycle thermal efficiency with regenerator"
q_in_total_withreg=q_in_withreg+q_in_reheat
Eta_th_withreg=w_net/(q_in_total_withreg)*Convert(, %) "[%]"

"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]
s_s[8]=s[8]
T_s[8]=T[8]
s_s[10]=s[10]
T_s[10]=T[10]
```

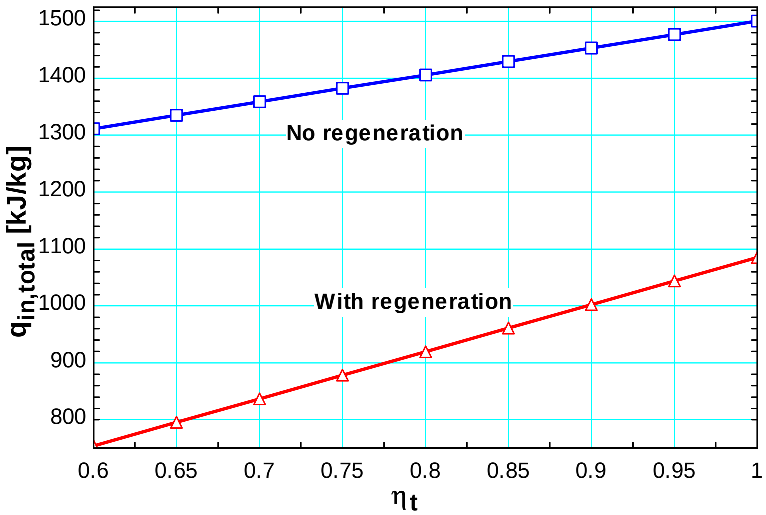
η_{reg}	η_C	η_t	$\eta_{th,noreg}$ [%]	$\eta_{th,withreg}$ [%]	$q_{in,total,noreg}$ [kJ/kg]	$q_{in,total,withreg}$ [kJ/kg]	w_{net} [kJ/kg]
0.6	0.78	0.86	30.57	41.29	1434	1062	438.5
0.65	0.78	0.86	30.57	42.53	1434	1031	438.5
0.7	0.78	0.86	30.57	43.85	1434	1000	438.5
0.75	0.78	0.86	30.57	45.25	1434	969.1	438.5
0.8	0.78	0.86	30.57	46.75	1434	938.1	438.5
0.85	0.78	0.86	30.57	48.34	1434	907.1	438.5
0.9	0.78	0.86	30.57	50.06	1434	876.1	438.5
0.95	0.78	0.86	30.57	51.89	1434	845.1	438.5
1	0.78	0.86	30.57	53.87	1434	814.1	438.5

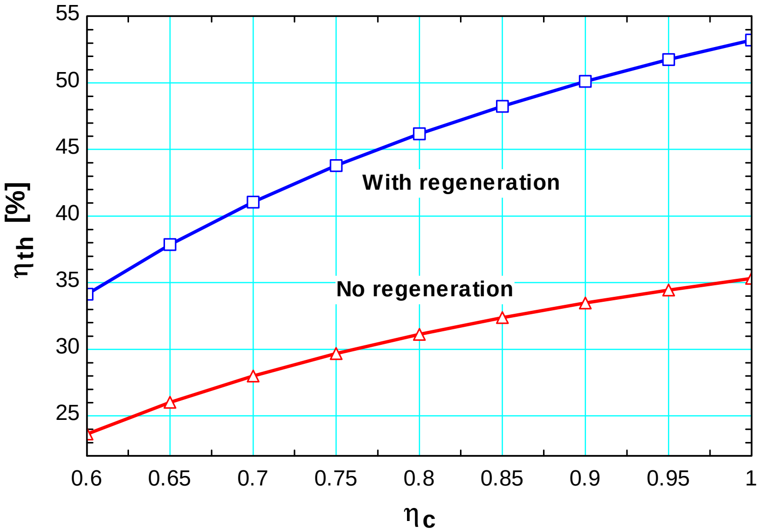



With regeneration
No regeneration
ηth [%]
ηreg
52
48
44
40
36
32
28
0.6
0.65
0.7
0.75
0.8
0.85
0.9
0.95
1


With regeneration
No regeneration
ηth [%]
ηt
55
50
45
40
35
30
25
20
15
10
0.6
0.65
0.7
0.75
0.8
0.85
0.9
0.95
1

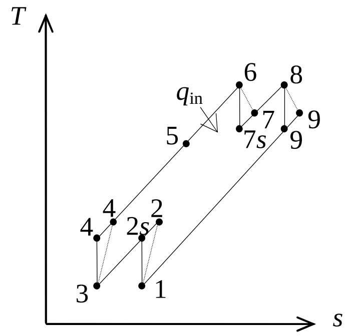

wnet [kJ/kg]
ηt
600
550
500
450
400
350
300
250
200
150
0.6
0.65
0.7
0.75
0.8
0.85
0.9
0.95
1


1500
1400
1300
1200
1100
1000
900
800
$q_{in,total}$ [kJ/kg]
No regeneration
With regeneration
0.6
0.65
0.7
0.75
0.8
0.85
0.9
0.95
1
η_t


55
50
45
40
35
30
25
η_{th} [%]
With regeneration
No regeneration
0.6
0.65
0.7
0.75
0.8
0.85
0.9
0.95
1
η_c

9-177 A regenerative gas-turbine engine operating with two stages of compression and two stages of expansion is considered. The back work ratio 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** 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 The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.

$$T_{4s} = T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(4)^{0.667/1.667} = 522.4\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_4 = T_2 = T_1 + (T_{2s}-T_1)/\eta_C$$
$$= 300 + (522.4-300)/(0.78)$$
$$= 585.2\text{ K}$$

$$T_{9s} = T_{7s} = T_6\left(\frac{P_7}{P_6}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{1}{4}\right)^{0.667/1.667} = 804.0\text{ K}$$

$$\eta_T = \frac{h_6-h_7}{h_6-h_{7s}} = \frac{c_p(T_6-T_7)}{c_p(T_6-T_{7s})} \longrightarrow T_9 = T_7 = T_6 - \eta_T(T_6-T_{7s})$$
$$= 1400 - (0.86)(1400-804.0)$$
$$= 887.4\text{ K}$$

$$\varepsilon = \frac{h_5-h_4}{h_9-h_4} = \frac{c_p(T_5-T_4)}{c_p(T_9-T_4)} \longrightarrow T_5 = T_4 + \varepsilon(T_9-T_4)$$
$$= 585.2 + (0.75)(887.4-585.2)$$
$$= 811.8\text{ K}$$

$$w_{\text{C,in}} = 2(h_2-h_1) = 2c_p(T_2-T_1) = 2(5.1926\text{ kJ/kg}\cdot\text{K})(585.2-300)\text{K} = 2961\text{ kJ/kg}$$

$$w_{\text{T,out}} = 2(h_6-h_7) = 2c_p(T_6-T_7) = 2(5.1926\text{ kJ/kg}\cdot\text{K})(1400-887.4)\text{K} = 5323\text{ kJ/kg}$$

Thus,

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{2961\text{ kJ/kg}}{5323\text{ kJ/kg}} = \mathbf{0.556}$$

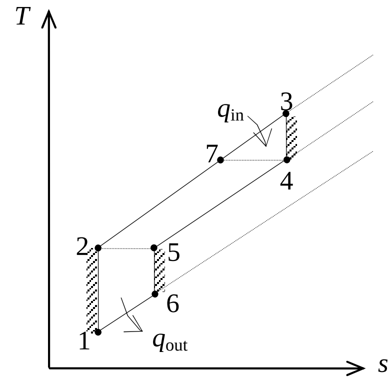
$$q_{\text{in}} = (h_6-h_5)+(h_8-h_7) = c_p[(T_6-T_5)+(T_8-T_7)]$$
$$= (5.1926\text{ kJ/kg}\cdot\text{K})[(1400-811.8)+(1400-887.4)]\text{K} = 5716\text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 5323 - 2961 = 2362\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{2362\text{ kJ/kg}}{5716\text{ kJ/kg}} = 0.4133 = \mathbf{41.3\%}$$

9-178 An ideal gas-turbine cycle with one stage of compression and two stages of expansion and regeneration is considered. The thermal efficiency of the cycle as a function of the compressor pressure ratio and the high-pressure turbine to compressor inlet temperature ratio is to be determined, and to be compared with the efficiency of the standard regenerative cycle.

Analysis The *T-s* diagram of the cycle is as shown in the figure. If the overall pressure ratio of the cycle is r_p, which is the pressure ratio across the compressor, then the pressure ratio across each turbine stage in the ideal case becomes $\sqrt{r_p}$. Using the isentropic relations, the temperatures at the compressor and turbine exit can be expressed as



$$T_5 = T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1\left(r_p\right)^{(k-1)/k}$$

$$T_7 = T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{\sqrt{r_p}}\right)^{(k-1)/k} = T_3 r_p^{(1-k)/2k}$$

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = T_5\left(\frac{1}{\sqrt{r_p}}\right)^{(k-1)/k} = T_2 r_p^{(1-k)/2k} = T_1 r_p^{(k-1)/k} r_p^{(1-k)/2k} = T_1 r_p^{(k-1)/2k}$$

Then,

$$q_{\text{in}} = h_3 - h_7 = c_p\left(T_3 - T_7\right) = c_p T_3\left(1 - r_p^{(1-k)/2k}\right)$$

$$q_{\text{out}} = h_6 - h_1 = c_p\left(T_6 - T_1\right) = c_p T_1\left(r_p^{(k-1)/2k} - 1\right)$$

and thus

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_p T_1\left(r_p^{(k-1)/2k} - 1\right)}{c_p T_3\left(1 - r_p^{(1-k)/2k}\right)}$$

which simplifies to

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_3} r_p^{(k-1)/2k}$$

The thermal efficiency of the single stage ideal regenerative cycle is given as

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_3} r_p^{(k-1)/k}$$

Therefore, the regenerative cycle with two stages of expansion has a higher thermal efficiency than the standard regenerative cycle with a single stage of expansion for any given value of the pressure ratio r_p.

9-179 A gas-turbine plant operates on the regenerative Brayton cycle with reheating and intercooling. The back work ratio, the net work output, the thermal efficiency, the second-law efficiency, and the exergies at the exits of the combustion chamber and the regenerator are to be determined.

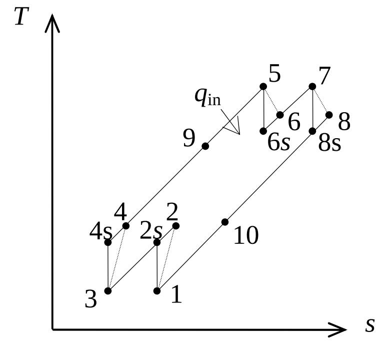
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is R = 0.287 kJ/kg·K.

Analysis (a) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.

Optimum intercooling and reheating pressure is

$$P_2 = \sqrt{P_1 P_4} = \sqrt{(100)(1200)} = 346.4\text{ kPa}$$



Process 1-2, 3-4: Compression

$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.43\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 300\text{ K} \\ P_1 = 100\text{ kPa} \end{array}\right\} s_1 = 5.7054\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 346.4\text{ kPa} \\ s_2 = s_1 = 5.7054\text{ kJ/kg.K} \end{array}\right\} h_{2s} = 428.79\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.80 = \frac{428.79 - 300.43}{h_2 - 300.43} \longrightarrow h_2 = 460.88\text{ kJ/kg}$$

$$T_3 = 350\text{ K} \longrightarrow h_3 = 350.78\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_3 = 350\text{ K} \\ P_3 = 346.4\text{ kPa} \end{array}\right\} s_3 = 5.5040\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_4 = 1200\text{ kPa} \\ s_4 = s_3 = 5.5040\text{ kJ/kg.K} \end{array}\right\} h_{4s} = 500.42\text{ kJ/kg}$$

$$\eta_C = \frac{h_{4s} - h_3}{h_4 - h_3} \longrightarrow 0.80 = \frac{500.42 - 350.78}{h_4 - 350.78} \longrightarrow h_4 = 537.83\text{ kJ/kg}$$

Process 6-7, 8-9: Expansion

$$T_6 = 1400\text{ K} \longrightarrow h_6 = 1514.9\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_6 = 1400\text{ K} \\ P_6 = 1200\text{ kPa} \end{array}\right\} s_6 = 6.6514\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_7 = 346.4\text{ kPa} \\ s_7 = s_6 = 6.6514\text{ kJ/kg.K} \end{array}\right\} h_{7s} = 1083.9\text{ kJ/kg}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} \longrightarrow 0.80 = \frac{1514.9 - h_7}{1514.9 - 1083.9} \longrightarrow h_7 = 1170.1\text{ kJ/kg}$$

$$T_8 = 1300\text{ K} \longrightarrow h_8 = 1395.6\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_8 = 1300\text{ K} \\ P_8 = 346.4\text{ kPa} \end{array}\right\} s_8 = 6.9196\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_9 = 100\text{ kPa} \\ s_9 = s_8 = 6.9196\text{ kJ/kg.K} \end{array}\right\} h_{9s} = 996.00\text{ kJ/kg}$$

$$\eta_T = \frac{h_8 - h_9}{h_8 - h_{9s}} \longrightarrow 0.80 = \frac{1395.6 - h_9}{1395.6 - 996.00} \longrightarrow h_9 = 1075.9 \text{ kJ/kg}$$

Cycle analysis:

$$w_{C,in} = h_2 - h_1 + h_4 - h_3 = 460.88 - 300.43 + 537.83 - 350.78 = 347.50 \text{ kJ/kg}$$

$$w_{T,out} = h_6 - h_7 + h_8 - h_9 = 1514.9 - 1170.1 + 1395.6 - 1075.9 = 664.50 \text{ kJ/kg}$$

$$r_{bw} = \frac{w_{C,in}}{w_{T,out}} = \frac{347.50}{664.50} = \mathbf{0.523}$$

$$w_{net} = w_{T,out} - w_{C,in} = 664.50 - 347.50 = \mathbf{317.0 \text{ kJ/kg}}$$

Regenerator analysis:

$$\varepsilon_{regen} = \frac{h_9 - h_{10}}{h_9 - h_4} \longrightarrow 0.75 = \frac{1075.9 - h_{10}}{1075.9 - 537.83} \longrightarrow h_{10} = 672.36 \text{ kJ/kg}$$

$$\left.\begin{array}{l} h_{10} = 672.36 \text{ K} \\ P_{10} = 100 \text{ kPa} \end{array}\right\} s_{10} = 6.5157 \text{ kJ/kg} \cdot \text{K}$$

$$q_{regen} = h_9 - h_{10} = h_5 - h_4 \longrightarrow 1075.9 - 672.36 = h_5 - 537.83 \longrightarrow h_5 = 941.40 \text{ kJ/kg}$$

(*b*) $$q_{in} = h_6 - h_5 = 1514.9 - 941.40 = 573.54 \text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{317.0}{573.54} = \mathbf{0.553}$$

(*c*) The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). The maximum temperature for the cycle can be taken to be the turbine inlet temperature. That is,

$$\eta_{max} = 1 - \frac{T_1}{T_6} = 1 - \frac{300 \text{ K}}{1400 \text{ K}} = 0.786$$

and

$$\eta_{II} = \frac{\eta_{th}}{\eta_{max}} = \frac{0.553}{0.786} = \mathbf{0.704}$$

(*d*) The exergies at the combustion chamber exit and the regenerator exit are

$$\begin{aligned} x_6 &= h_6 - h_0 - T_0(s_6 - s_0) \\ &= (1514.9 - 300.43)\text{kJ/kg} - (300 \text{ K})(6.6514 - 5.7054)\text{kJ/kg.K} \\ &= \mathbf{930.7 \text{ kJ/kg}} \end{aligned}$$

$$\begin{aligned} x_{10} &= h_{10} - h_0 - T_0(s_{10} - s_0) \\ &= (672.36 - 300.43)\text{kJ/kg} - (300 \text{ K})(6.5157 - 5.7054)\text{kJ/kg.K} \\ &= \mathbf{128.8 \text{ kJ/kg}} \end{aligned}$$

9-180 The thermal efficiency of a two-stage gas turbine with regeneration, reheating and intercooling to that of a three-stage gas turbine is to be compared.

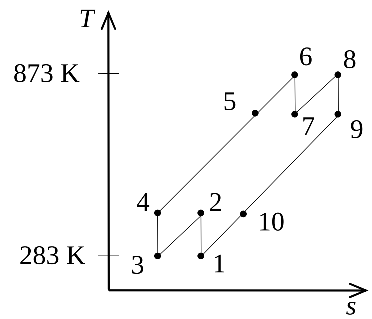
Assumptions **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

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

Analysis

Two Stages:

The pressure ratio across each stage is



$$r_p = \sqrt{16} = 4$$

The temperatures at the end of compression and expansion are

$$T_c = T_{\min} r_p^{(k-1)/k} = (283\text{ K})(4)^{0.4/1.4} = 420.5\text{ K}$$

$$T_e = T_{\max}\left(\frac{1}{r_p}\right)^{(k-1)/k} = (873\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 587.5\text{ K}$$

The heat input and heat output are

$$q_{\text{in}} = 2c_p(T_{\max} - T_e) = 2(1.005\text{ kJ/kg}\cdot\text{K})(873 - 587.5)\text{ K} = 573.9\text{ kJ/kg}$$

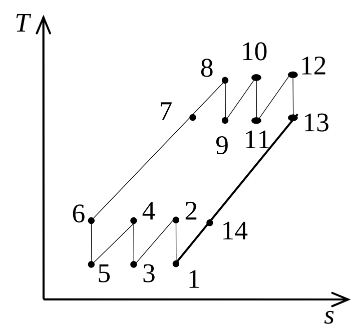
$$q_{\text{out}} = 2c_p(T_c - T_{\min}) = 2(1.005\text{ kJ/kg}\cdot\text{K})(420.5 - 283)\text{ K} = 276.4\text{ kJ/kg}$$

The thermal efficiency of the cycle is then

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{276.4}{573.9} = \mathbf{0.518}$$

Three Stages:

The pressure ratio across each stage is



$$r_p = 16^{1/3} = 2.520$$

The temperatures at the end of compression and expansion are

$$T_c = T_{\min} r_p^{(k-1)/k} = (283\text{ K})(2.520)^{0.4/1.4} = 368.5\text{ K}$$

$$T_e = T_{\max}\left(\frac{1}{r_p}\right)^{(k-1)/k} = (873\text{ K})\left(\frac{1}{2.520}\right)^{0.4/1.4} = 670.4\text{ K}$$

The heat input and heat output are

$$q_{\text{in}} = 3c_p(T_{\max} - T_e) = 3(1.005\text{ kJ/kg}\cdot\text{K})(873 - 670.4)\text{ K} = 610.8\text{ kJ/kg}$$

$$q_{\text{out}} = 3c_p(T_c - T_{\min}) = 3(1.005\text{ kJ/kg}\cdot\text{K})(368.5 - 283)\text{ K} = 257.8\text{ kJ/kg}$$

The thermal efficiency of the cycle is then

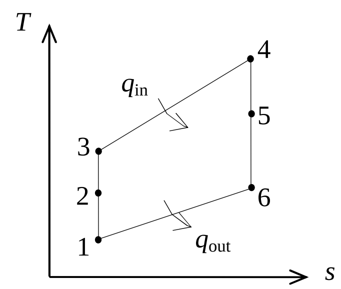
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{257.8}{610.8} = \mathbf{0.578}$$

9-181E A pure jet engine operating on an ideal cycle is considered. The thrust force produced per unit mass flow rate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

Properties The properties of air at room temperature are R = 0.3704 psia·ft^3/lbm·R (Table A-1E), c_p = 0.24 Btu/lbm·R and k = 1.4 (Table A-2Ea).

Analysis (a) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 1200 ft/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).



Diffuser:

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

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^{2\ \nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 490\text{ R} + \frac{(1200\text{ ft/s})^2}{(2)(0.24\text{ BtuJlbm}\cdot\text{R})}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 609.8\text{ R}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (10\text{ psia})\left(\frac{609.8\text{ R}}{490\text{ R}}\right)^{1.4/0.4} = 21.5\text{ psia}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (9)(21.5\text{ psia}) = 193.5\text{ psia}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (609.8\text{ R})(9)^{0.4/1.4} = 1142.4\text{ R}$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or $\quad T_5 = T_4 - T_3 + T_2 = 1160 - 1142.4 + 609.8 = 627.4\text{ R}$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (1160\text{ R})\left(\frac{10\text{ psia}}{193.5\text{ psia}}\right)^{0.4/1.4} = 497.5\text{ R}$$

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

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\ \nearrow 0}}{2} \longrightarrow 0 = c_p(T_6 - T_5) + V_6^2/2$$

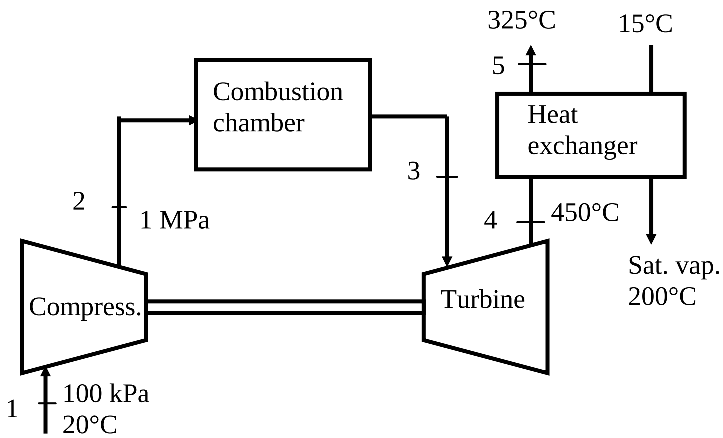
or, $\quad V_6 = V_{exit} = \sqrt{(2)(0.24\text{ Btu/lbm}\cdot\text{R})(627.4 - 497.5)\text{R}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)} = 1249\text{ ft/s}$

The specific impulse is then

$$\frac{F}{\dot{m}} = V_{\text{exit}} - V_{\text{inlet}} = 1249 - 1200 = \mathbf{49\ m/s}$$

9-182 The electricity and the process heat requirements of a manufacturing facility are to be met by a cogeneration plant consisting of a gas-turbine and a heat exchanger for steam production. The mass flow rate of the air in the cycle, the back work ratio, the thermal efficiency, the rate at which steam is produced in the heat exchanger, and the utilization efficiency of the cogeneration plant 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.



Analysis (*a*) For this problem, we use the properties of air from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.

Process 1-2: Compression

$$T_1 = 20°\text{C} \longrightarrow h_1 = 293.5 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 20°\text{C} \\ P_1 = 100 \text{ kPa} \end{array}\right\} s_1 = 5.682 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{array}{l} P_2 = 1000 \text{ kPa} \\ s_2 = s_1 = 5.682 \text{ kJ/kg.K} \end{array}\right\} h_{2s} = 567.2 \text{ kJ/kg}$$

$$\eta_\text{C} = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.86 = \frac{567.2 - 293.5}{h_2 - 293.5} \longrightarrow h_2 = 611.8 \text{ kJ/kg}$$

Process 3-4: Expansion

$$T_4 = 450°\text{C} \longrightarrow h_4 = 738.5 \text{ kJ/kg}$$

$$\eta_\text{T} = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.88 = \frac{h_3 - 738.5}{h_3 - h_{4s}}$$

We cannot find the enthalpy at state 3 directly. However, using the following lines in EES together with the isentropic efficiency relation, we find h_3 = 1262 kJ/kg, T_3 = 913.2ºC, s_3 = 6.507 kJ/kg.K. The solution by hand would require a trial-error approach.

```
h_3=enthalpy(Air, T=T_3)
s_3=entropy(Air, T=T_3, P=P_2)
h_4s=enthalpy(Air, P=P_1, s=s_3)
```

Also,

$$T_5 = 325°\text{C} \longrightarrow h_5 = 605.4 \text{ kJ/kg}$$

The inlet water is compressed liquid at 15ºC and at the saturation pressure of steam at 200ºC (1555 kPa). This is not available in the tables but we can obtain it in EES. The alternative is to use saturated liquid enthalpy at the given temperature.

$$\left.\begin{array}{l} T_{w1} = 15°\text{C} \\ P_1 = 1555 \text{ kPa} \end{array}\right\} h_{w1} = 64.47 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{w2} = 200°\text{C} \\ x_2 = 1 \end{array}\right\} h_{w2} = 2792 \text{ kJ/kg}$$

The net work output is

$$w_{\text{C,in}} = h_2 - h_1 = 611.8 - 293.5 = 318.2 \text{ kJ/kg}$$

$$w_{\text{T,out}} = h_3 - h_4 = 1262 - 738.5 = 523.4 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 523.4 - 318.2 = 205.2 \text{ kJ/kg}$$

The mass flow rate of air is

$$\dot{m}_a = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{1500 \text{ kJ/s}}{205.2 \text{ kJ/kg}} = \mathbf{7.311 \text{ kg/s}}$$

(b) The back work ratio is

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{318.2}{523.4} = \mathbf{0.608}$$

The rate of heat input and the thermal efficiency are

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_3 - h_2) = (7.311 \text{ kg/s})(1262 - 611.8)\text{kJ/kg} = 4753 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1500 \text{ kW}}{4753 \text{ kW}} = 0.3156 = \mathbf{31.6\%}$$

(c) An energy balance on the heat exchanger gives

$$\dot{m}_a (h_4 - h_5) = \dot{m}_w (h_{w2} - h_{w1})$$

$$(7.311 \text{ kg/s})(738.5 - 605.4)\text{kJ/kg} = \dot{m}_w (2792 - 64.47)\text{kJ/kg} \longrightarrow \dot{m}_w = \mathbf{0.3569 \text{ kg/s}}$$

(d) The heat supplied to the water in the heat exchanger (process heat) and the utilization efficiency are

$$\dot{Q}_{\text{p}} = \dot{m}_w (h_{w2} - h_{w1}) = (0.3569 \text{ kg/s})(2792 - 64.47)\text{kJ/kg} = 973.5 \text{ kW}$$

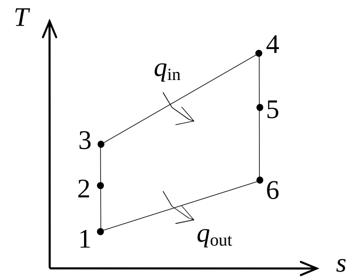
$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{p}}}{\dot{Q}_{\text{in}}} = \frac{1500 + 973.5}{4753 \text{ kW}} = 0.5204 = \mathbf{52.0\%}$$

9-183 A turbojet aircraft flying is considered. The pressure of the gases at the turbine exit, the mass flow rate of the air through the compressor, the velocity of the gases at the nozzle exit, the propulsive power, and the propulsive efficiency of the cycle are to be determined.

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

Properties The gas constant of air is R = 0.287 kJ/kg·K (Table A-1).

Analysis (a) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.



Diffuser, Process 1-2:

$$T_1 = -35°\text{C} \longrightarrow h_1 = 238.23 \text{ kJ/kg}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$(238.23 \text{ kJ/kg}) + \frac{(900/3.6 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = h_2 + \frac{(15 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) \longrightarrow h_2 = 269.37 \text{ kJ/kg}$$

$$\left.\begin{array}{l} h_2 = 269.37 \text{ kJ/kg} \\ P_2 = 50 \text{ kPa} \end{array}\right\} s_2 = 5.7951 \text{ kJ/kg} \cdot \text{K}$$

Compressor, Process 2-3:

$$\left.\begin{array}{l} P_3 = 450 \text{ kPa} \\ s_3 = s_2 = 5.7951 \text{ kJ/kg.K} \end{array}\right\} h_{3s} = 505.19 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{3s} - h_2}{h_3 - h_2} \longrightarrow 0.83 = \frac{505.19 - 269.37}{h_3 - 269.37} \longrightarrow h_3 = 553.50 \text{ kJ/kg}$$

Turbine, Process 3-4:

$$T_4 = 950°\text{C} \longrightarrow h_4 = 1304.8 \text{ kJ/kg}$$

$$h_3 - h_2 = h_4 - h_5 \longrightarrow 553.50 - 269.37 = 1304.8 - h_5 \longrightarrow h_5 = 1020.6 \text{ kJ/kg}$$

where the mass flow rates through the compressor and the turbine are assumed equal.

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow 0.83 = \frac{1304.8 - 1020.6}{1304.8 - h_{5s}} \longrightarrow h_{5s} = 962.45 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_4 = 950°\text{C} \\ P_4 = 450 \text{ kPa} \end{array}\right\} s_4 = 6.7725 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{array}{l} h_{5s} = 962.45 \text{ kJ/kg} \\ s_5 = s_4 = 6.7725 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} P_5 = \mathbf{147.4 \text{ kPa}}$$

(b) The mass flow rate of the air through the compressor is

$$\dot{m} = \frac{\dot{W}_C}{h_3 - h_2} = \frac{500 \text{ kJ/s}}{(553.50 - 269.37) \text{ kJ/kg}} = \mathbf{1.760 \text{ kg/s}}$$

(c) *Nozzle, Process 5-6:*

$$\left.\begin{array}{l} h_5 = 1020.6 \text{ kJ/kg} \\ P_5 = 147.4 \text{ kPa} \end{array}\right\} s_5 = 6.8336 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{array}{l} P_6 = 40 \text{ kPa} \\ s_6 = s_5 = 6.8336 \text{ kJ/kg.K} \end{array}\right\} h_{6s} = 709.66 \text{ kJ/kg}$$

$$\eta_N = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow 0.83 = \frac{1020.6 - h_6}{1020.6 - 709.66} \longrightarrow h_6 = 762.52 \text{ kJ/kg}$$

$$h_5 + \frac{V_5^2}{2} = h_6 + \frac{V_6^2}{2}$$

$$(1020.6 \text{ kJ/kg}) + 0 = 762.52 \text{ kJ/kg} + \frac{V_6^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) \longrightarrow V_6 = \mathbf{718.5 \text{ m/s}}$$

where the velocity at nozzle inlet is assumed zero.

(*d*) The propulsive power and the propulsive efficiency are

$$\dot{W}_p = \dot{m}(V_6 - V_1)V_1 = (1.76 \text{ kg/s})(718.5 \text{ m/s} - 250 \text{ m/s})(250 \text{ m/s})\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = \mathbf{206.1 \text{ kW}}$$

$$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = (1.76 \text{ kg/s})(1304.8 - 553.50)\text{kJ/kg} = 1322 \text{ kW}$$

$$\eta_p = \frac{\dot{W}_p}{\dot{Q}_{\text{in}}} = \frac{206.1 \text{ kW}}{1322 \text{ kW}} = \mathbf{0.156}$$

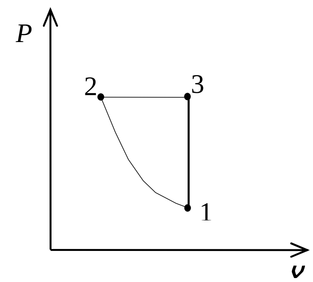
9-184 The three processes of an air standard cycle are described. The cycle is to be shown on the *P*-*v* and *T*-*s* diagrams, and the expressions for back work ratio and the thermal efficiency are to be obtained.

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.

Analysis (a) The *P*-*v* and *T*-*s* diagrams for this cycle are as shown.

(b) The work of compression is found by the first law for process 1-2:

$$q_{1-2} - w_{1-2} = \Delta u_{1-2}$$
$$q_{1-2} = 0(isentropic\ \ process)$$
$$w_{1-2} = -\Delta u_{1-2} = -C_v\left(T_2 - T_1\right)$$
$$w_{comp} = -w_{1-2} = C_v\left(T_2 - T_1\right)$$

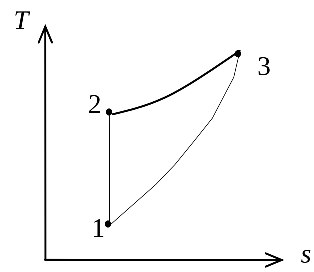


The expansion work is found by

$$w_{\exp} = w_{2-3} = \int_2^3 Pdv = P\left(v_3 - v_2\right) = R\left(T_3 - T_2\right)$$

The back work ratio is

$$\frac{w_{comp}}{w_{\exp}} = \frac{C_v\left(T_3 - T_1\right)}{R\left(T_3 - T_2\right)} = \frac{C_v}{R}\frac{T_1}{T_2}\frac{\left(T_3/T_1 - 1\right)}{\left(T_3/T_2 - 1\right)}$$



Process 1-2 is isentropic; therefore,

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \frac{1}{r^{k-1}} \text{ and } \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^k = r^k$$

Process 2-3 is constant pressure; therefore,

$$\frac{P_3 V_3}{T_3} = \frac{P_2 V_2}{T_2} \Rightarrow \frac{T_3}{T_2} = \frac{V_3}{V_2} = \frac{V_1}{V_2} = r$$

Process 3-1 is constant volume; therefore,

$$\frac{P_3 V_3}{T_3} = \frac{P_1 V_1}{T_1} \Rightarrow \frac{T_3}{T_1} = \frac{P_3}{P_1} = \frac{P_2}{P_1} = r^k$$

The back work ratio becomes (C_v=R/(k-1))

$$\frac{w_{comp}}{w_{\exp}} = \frac{1}{k-1}\frac{1}{r^{k-1}}\frac{r^{k-1} - 1}{r - 1}$$

(c) Apply first law to the closed system for processes 2-3 and 3-1 to show:

$$q_{in} = C_p\left(T_3 - T_2\right)$$
$$q_{out} = C_v\left(T_3 - T_1\right)$$

The cycle thermal efficiency is given by

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_v\left(T_3 - T_1\right)}{C_p\left(T_3 - T_2\right)} = 1 - \frac{1}{k}\frac{T_1\left(T_3/T_1 - 1\right)}{T_2\left(T_3/T_2 - 1\right)}$$

The efficiency becomes

$$\eta_{th} = 1 - \frac{1}{k} \frac{1}{r^{k-1}} \frac{r^k - 1}{r - 1}$$

(d) Determine the value of the back work ratio and efficiency as r goes to unity.

$$\frac{w_{comp}}{w_{\exp}} = \frac{1}{k-1} \frac{1}{r^{k-1}} \frac{r^{k-1} - 1}{r - 1}$$

$$\lim_{r \to 1} \frac{w_{comp}}{w_{\exp}} = \frac{1}{k-1} \left\{ \lim_{r \to 1} \frac{1}{r^{k-1}} \frac{r^{k-1} - 1}{r - 1} \right\} = \frac{1}{k-1} \left\{ \lim_{r \to 1} \frac{r^{k-1} - 1}{r^k - r^{k-1}} \right\} = \frac{1}{k-1} \left\{ \lim_{r \to 1} \frac{(k-1) r^{k-2}}{k r^{k-1} - (k-1) r^{k-2}} \right\}$$

$$\lim_{r \to 1} \frac{w_{comp}}{w_{\exp}} = \frac{1}{k-1} \left\{ \frac{k-1}{k-k+1} \right\} = \frac{1}{k-1} \left\{ \frac{k-1}{1} \right\} = 1$$

$$\eta_{th} = 1 - \frac{1}{k} \frac{1}{r^{k-1}} \frac{r^k - 1}{r - 1}$$

$$\lim_{r \to 1} \eta_{th} = 1 - \frac{1}{k} \left\{ \lim_{r \to 1} \frac{1}{r^{k-1}} \frac{r^k - 1}{r - 1} \right\} = 1 - \frac{1}{k} \left\{ \lim_{r \to 1} \frac{r^k - 1}{r^k - r^{k-1}} \right\} = 1 - \frac{1}{k} \left\{ \lim_{r \to 1} \frac{k r^{k-1}}{k r^{k-1} - (k-1) r^{k-2}} \right\}$$

$$\lim_{r \to 1} \eta_{th} = 1 - \frac{1}{k} \left\{ \frac{k}{k-k+1} \right\} = 1 - \frac{1}{k} \left\{ \frac{k}{1} \right\} = 0$$

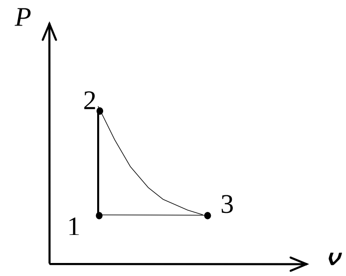
These results show that if there is no compression (i.e. $r = 1$), there can be no expansion and no net work will be done even though heat may be added to the system.

9-185 The three processes of an air standard cycle are described. The cycle is to be shown on the *P*-*v* and *T*-*s* diagrams, and the expressions for back work ratio and the thermal efficiency are to be obtained.

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.

Analysis (a) The *P*-*v* and *T*-*s* diagrams for this cycle are as shown.

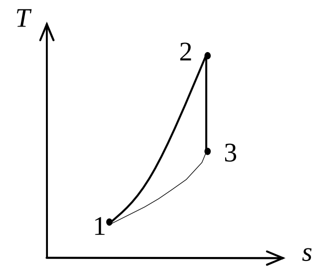
(b) The work of expansion is found by the first law for process 2-3:



$$q_{2-3} - w_{2-3} = \Delta u_{2-3}$$
$$q_{2-3} = 0(isentropic\ \ process)$$
$$w_{2-3} = -\Delta u_{2-3} = -C_v\left(T_3 - T_2\right)$$
$$w_{\exp} = w_{2-3} = C_v\left(T_2 - T_3\right)$$

The compression work is found by

$$w_{comp} = -w_{3-1} = -\int_3^1 Pdv = -P\left(v_1 - v_3\right) = R\left(T_3 - T_1\right)$$



The back work ratio is

$$\frac{w_{comp}}{w_{\exp}} = \frac{C_v\left(T_3 - T_1\right)}{R\left(T_2 - T_3\right)} = \frac{C_v}{R}\frac{T_3}{T_3}\frac{\left(1 - T_1/T_3\right)}{\left(T_2/T_3 - 1\right)} = \frac{C_v}{R}\frac{\left(1 - T_1/T_3\right)}{\left(T_2/T_3 - 1\right)}$$

Process 3-1 is constant pressure; therefore,

$$\frac{P_3V_3}{T_3} = \frac{P_1V_1}{T_1} \Rightarrow \frac{T_1}{T_3} = \frac{V_1}{V_3} = \frac{V_2}{V_3} = \frac{1}{r}$$

Process 2-3 is isentropic; therefore,

$$\frac{T_2}{T_3} = \left(\frac{V_2}{V_2}\right)^{k-1} = r^{k-1} \text{ and } \frac{P_2}{P_3} = \left(\frac{V_3}{V_2}\right)^{k} = r^{k}$$

The back work ratio becomes (C_v=R/(k-1))

$$\frac{w_{comp}}{w_{\exp}} = \left(k-1\right)\frac{1-\frac{1}{r}}{r^{k-1}-1} = \frac{k-1}{r}\frac{r-1}{r^{k-1}-1}$$

(c) Apply first law to the closed system for processes 1-2 and 3-1 to show:

$$q_{in} = C_v\left(T_2 - T_1\right)$$
$$q_{out} = C_p\left(T_3 - T_1\right)$$

The cycle thermal efficiency is given by

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p\left(T_3 - T_1\right)}{C_v\left(T_2 - T_1\right)} = 1 - k\frac{T_1\left(T_3/T_1 - 1\right)}{T_1\left(T_2/T_1 - 1\right)}$$

Process 1-2 is constant volume; therefore,

$$\frac{P_2V_2}{T_2} = \frac{P_1V_1}{T_1} \Rightarrow \frac{T_2}{T_1} = \frac{P_2}{P_1} = \frac{P_2}{P_3} = r^k$$

The efficiency becomes

$$\eta_{th} = 1 - k\frac{r-1}{r^k - 1}$$

(d) Determine the value of the back work ratio and efficiency as r goes to unity.

$$\frac{w_{comp}}{w_{\exp}} = (k-1)\frac{1-\frac{1}{r}}{r^{k-1}-1} = \frac{k-1}{r}\frac{r-1}{r^{k-1}-1}$$

$$\lim_{r\to 1}\frac{w_{comp}}{w_{\exp}} = (k-1)\left\{\lim_{r\to 1}\frac{r-1}{r^k - r}\right\} = (k-1)\left\{\lim_{r\to 1}\frac{1}{kr^{k-1}-1}\right\}$$

$$\lim_{r\to 1}\frac{w_{comp}}{w_{\exp}} = (k-1)\left\{\frac{1}{k-1}\right\} = 1$$

$$\eta_{th} = 1 - k\frac{r-1}{r^k - 1}$$

$$\lim_{r\to 1}\eta_{th} = 1 - k\left\{\lim_{r\to 1}\frac{r-1}{r^k - 1}\right\} = 1 - k\left\{\lim_{r\to 1}\frac{1}{kr^{k-1}}\right\}$$

$$\lim_{r\to 1}\eta_{th} = 1 - k\left\{\frac{1}{k}\right\} = 0$$

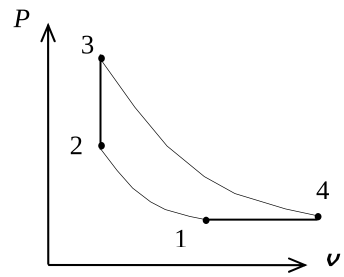
These results show that if there is no compression (i.e. r = 1), there can be no expansion and no net work will be done even though heat may be added to the system.

9-186 The four processes of an air-standard cycle are described. The cycle is to be shown on the *P*-*v* and *T*-*s* diagrams; an expression for the cycle thermal efficiency is to be obtained; and the limit of the efficiency as the volume ratio during heat rejection approaches unity is to be evaluated.

Analysis (a) The *P*-*v* and *T*-*s* diagrams of the cycle are shown in the figures.

(b) Apply first law to the closed system for processes 2-3 and 4-1 to show:

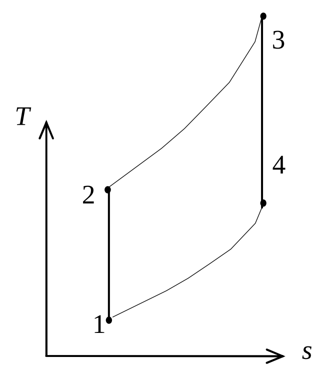
$$q_{in} = C_v\left(T_3 - T_2\right)$$
$$q_{out} = C_p\left(T_4 - T_1\right)$$



The cycle thermal efficiency is given by

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{C_p\left(T_4 - T_1\right)}{C_v\left(T_3 - T_2\right)} = 1 - k\frac{T_1\left(T_4/T_1 - 1\right)}{T_2\left(T_3/T_2 - 1\right)}$$

Process 1-2 is isentropic; therefore, $\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \frac{1}{r^{k-1}}$



Process 3-4 is isentropic; therefore, $\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{k-1} = r_e^{k-1}$

Process 4-1 is constant pressure; therefore,

$$\frac{P_4 V_4}{T_4} = \frac{P_1 V_1}{T_1} \Rightarrow \frac{T_4}{T_1} = \frac{V_4}{V_1} = r_p$$

$$\frac{T_3}{T_2} = \frac{T_3}{T_4}\frac{T_4}{T_1}\frac{T_1}{T_2} = r_e^{k-1} r_p \frac{1}{r^{k-1}} = \left(\frac{r_e}{r}\right)^{k-1} r_p$$

Since process 2-3 is constant volume and $V_3 = V_2$,

$$r_e = \frac{V_4}{V_3} = \frac{V_4}{V_2} = \frac{V_4}{V_1}\frac{V_1}{V_2} = r_p r$$

$$\frac{T_3}{T_2} = \left(\frac{r_p r}{r}\right)^{k-1} r_p = r_p^k$$

The efficiency becomes

$$\eta_{th} = 1 - k\frac{1}{r^{k-1}}\frac{r_p - 1}{r_p^k - 1}$$

(c) In the limit as r_p approaches unity, the cycle thermal efficiency becomes

$$\lim_{r_p \to 1} \eta_{th} = 1 - k\frac{1}{r^{k-1}}\left\{\lim_{r_p \to 1}\frac{r_p - 1}{r_p^k - 1}\right\} = 1 - k\frac{1}{r^{k-1}}\left\{\lim_{r_p \to 1}\frac{1}{k r_p^{k-1}}\right\}$$

$$\lim_{r_p \to 1} \eta_{th} = 1 - k\frac{1}{r^{k-1}}\left\{\frac{1}{k}\right\} = 1 - \frac{1}{r^{k-1}} = \eta_{th\ Otto}$$

9-187 The four processes of an air-standard cycle are described. The back work ratio and its limit as r_p goes to unity are to be determined, and the result is to be compared to the expression for the Otto cycle.

Analysis The work of compression for process 1-2 is found by the first law:

$$q_{1-2} - w_{1-2} = \Delta u_{1-2}$$
$$q_{1-2} = 0(isentropic\ \ process)$$
$$w_{1-2} = -\Delta u_{1-2} = -C_v\left(T_2 - T_1\right)$$
$$w_{comp,1-2} = -w_{1-2} = C_v\left(T_2 - T_1\right)$$

The work of compression for process 4-1 is found by

$$w_{comp,4-1} = -w_{4-1} = -\int_4^1 P dv = -P\left(v_1 - v_4\right) = R\left(T_4 - T_1\right)$$

The work of expansion for process 3-4 is found by the first law:

$$q_{3-4} - w_{3-4} = \Delta u_{3-4}$$
$$q_{3-4} = 0(isentropic\ \ process)$$
$$w_{3-4} = -\Delta u_{3-4} = -C_v\left(T_4 - T_3\right)$$
$$w_{\exp,3-4} = -w_{3-4} = C_v\left(T_3 - T_4\right)$$

The back work ratio is

$$\frac{w_{comp}}{w_{\exp}} = \frac{R\left(T_4 - T_1\right) + C_v\left(T_2 - T_1\right)}{C_v\left(T_3 - T_4\right)} = \frac{T_1}{T_3}\frac{\dfrac{R}{C_v}\left(T_4/T_1 - 1\right) + \left(T_2/T_1 - 1\right)}{\left(1 - T_4/T_3\right)}$$

Using data from the previous problem and $C_v = R/(k-1)$

$$\frac{w_{comp}}{w_{\exp}} = \frac{T_1}{T_3}\frac{(k-1)\left(r_p - 1\right) + \left(r^{k-1} - 1\right)}{\left(1 - \dfrac{1}{r_p^{k-1} r^{k-1}}\right)}$$

$$\lim_{r_p \to 1}\frac{w_{comp}}{w_{\exp}} = \frac{T_1}{T_3}\left\{\lim_{r_p \to 1}\frac{(k-1)\left(r_p - 1\right) + \left(r^{k-1} - 1\right)}{\left(1 - \dfrac{1}{r_p^{k-1} r^{k-1}}\right)}\right\} = \frac{T_1}{T_3}\left\{\frac{(k-1)(0) + \left(r^{k-1} - 1\right)}{1 - \dfrac{1}{r^{k-1}}}\right\}$$

$$\lim_{r_p \to 1}\frac{w_{comp}}{w_{\exp}} = \frac{T_1}{T_3}\left\{\frac{r^{k-1} - 1}{1 - \dfrac{1}{r^{k-1}}}\right\}$$

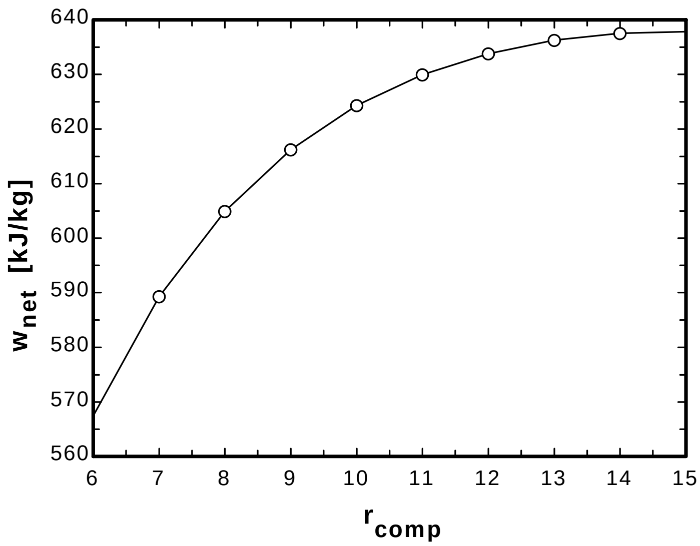
This result is the same expression for the back work ratio for the Otto cycle.

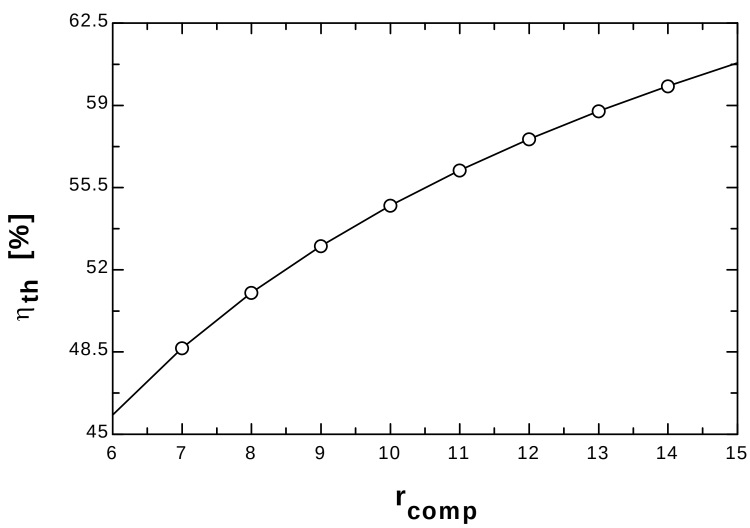
9-188 The effects of compression ratio on the net work output and the thermal efficiency of the Otto cycle for given operating conditions is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
"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]=R*T[1]
R=0.287 [kJ/kg-K]
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]=R*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]=R*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*Convert(, %) "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



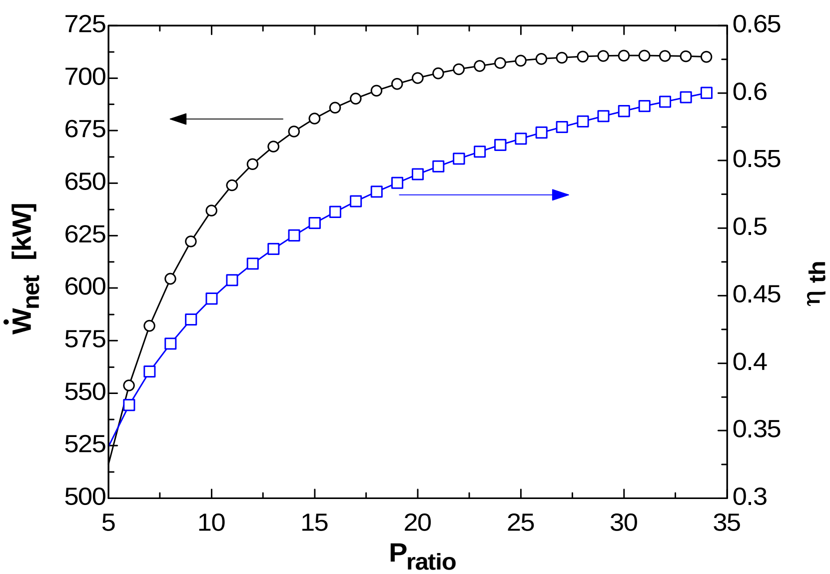


9-189 The effects of pressure ratio on the net work output and the thermal efficiency of a simple Brayton cycle is to be investigated. The pressure ratios at which the net work output and the thermal efficiency are maximum are to be determined.

Analysis Using EES, the problem is solved as follows:

```
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 anaysis"
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]
0.254	0.3383	5	175.8	516.3	692.1	1526
0.2665	0.3689	6	201.2	553.7	754.9	1501
0.2776	0.3938	7	223.7	582.2	805.9	1478
0.2876	0.4146	8	244.1	604.5	848.5	1458
0.2968	0.4324	9	262.6	622.4	885	1439
0.3052	0.4478	10	279.7	637	916.7	1422
0.313	0.4615	11	295.7	649	944.7	1406
0.3203	0.4736	12	310.6	659.1	969.6	1392
0.3272	0.4846	13	324.6	667.5	992.1	1378
0.3337	0.4945	14	337.8	674.7	1013	1364
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

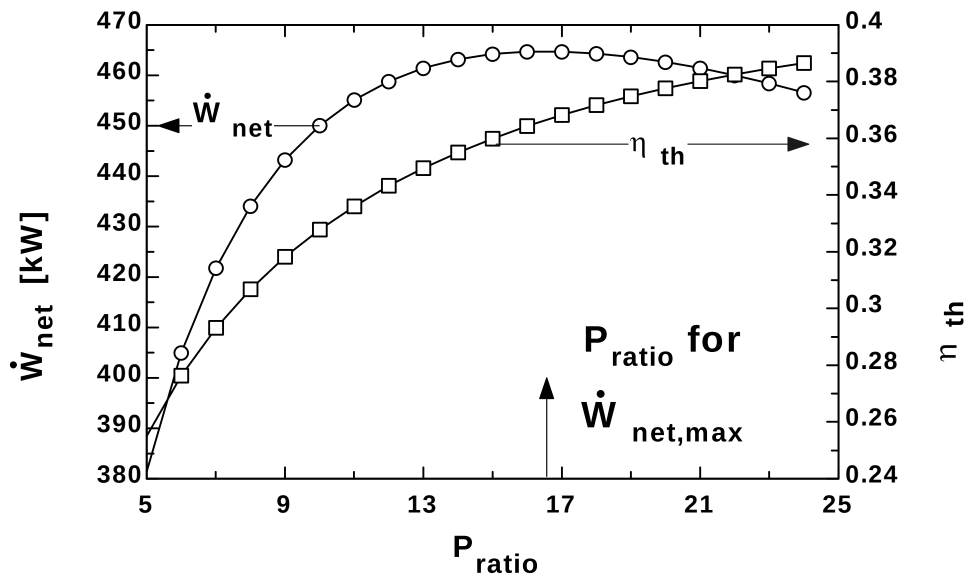


9-190 The effects of pressure ratio on the net work output and the thermal efficiency of a simple Brayton cycle is to be investigated assuming adiabatic efficiencies of 85 percent for both the turbine and the compressor. The pressure ratios at which the net work output and the thermal efficiency are maximum are to be determined.

Analysis Using EES, the problem is solved as follows:

```
P_ratio = 8
T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 1800 [K]
m_dot = 1 [kg/s]
Eta_c = 80/100
Eta_t = 80/100

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



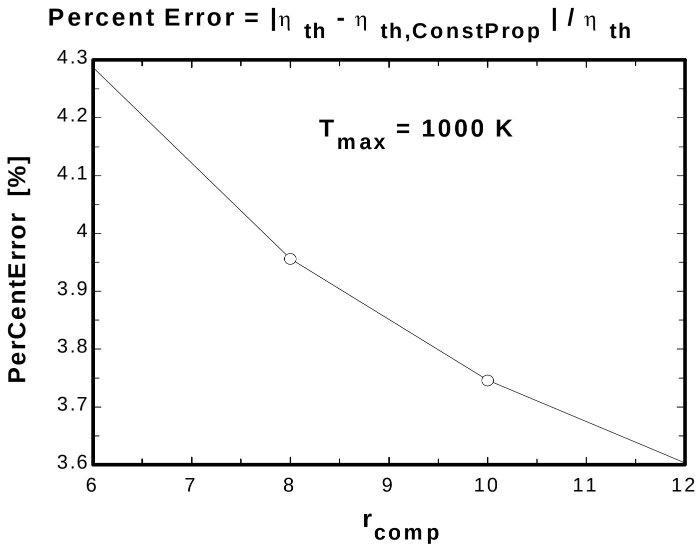
9-191 The effects of pressure ratio, maximum cycle temperature, and compressor and turbine inefficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with air as the working fluid is to be investigated. Constant specific heats at room temperature are to be used.

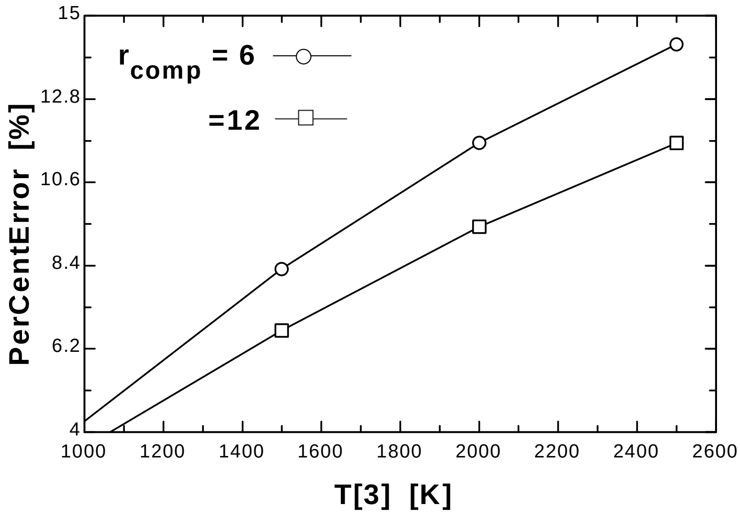
Analysis Using EES, the problem is solved as follows:

```
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)*Convert(, %) "[%]"
"The Easy Way to calculate the constant property Otto cycle efficiency is:"
Eta_th_easy = (1 - 1/r_comp^(k-1))*Convert(, %) "[%]"
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]=R*T[1]
R=0.287 [kJ/kg-K]
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]=R*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]=R*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*Convert(, %) "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*Convert(, %) "[%]"
```

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



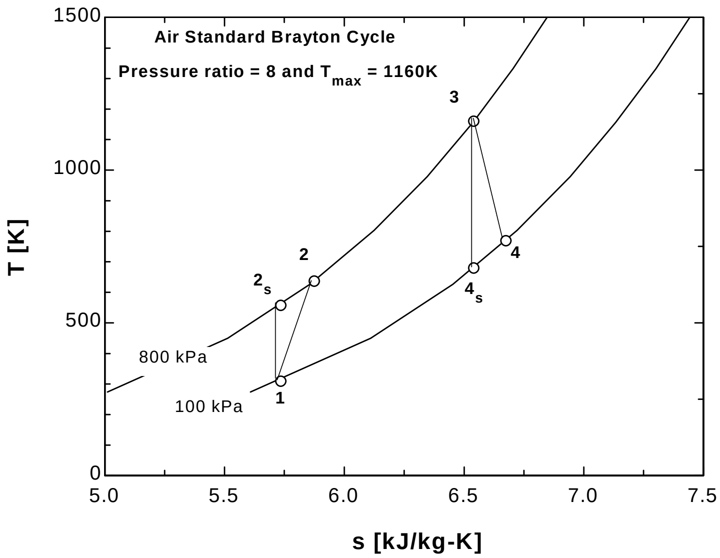


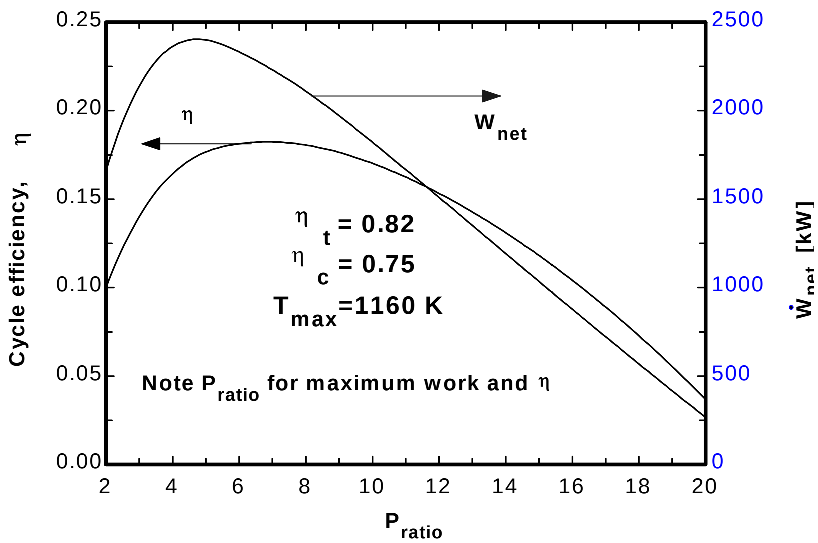
9-192 The effects of pressure ratio, maximum cycle temperature, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with air as the working fluid is to be investigated. Variable specific heats are to be used.

Analysis Using EES, the problem is solved as follows:

```
"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 anaysis"
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



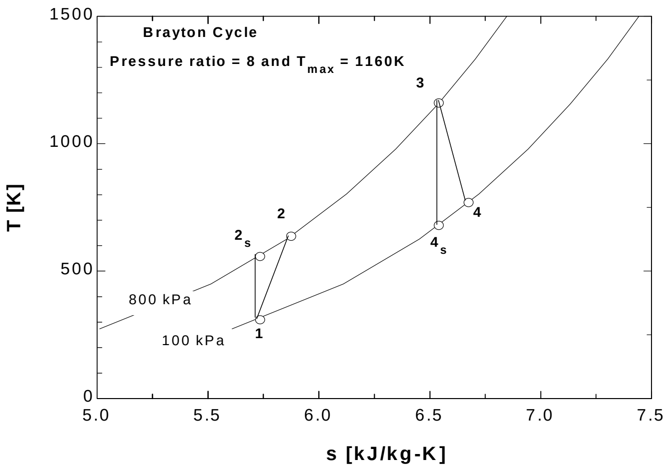


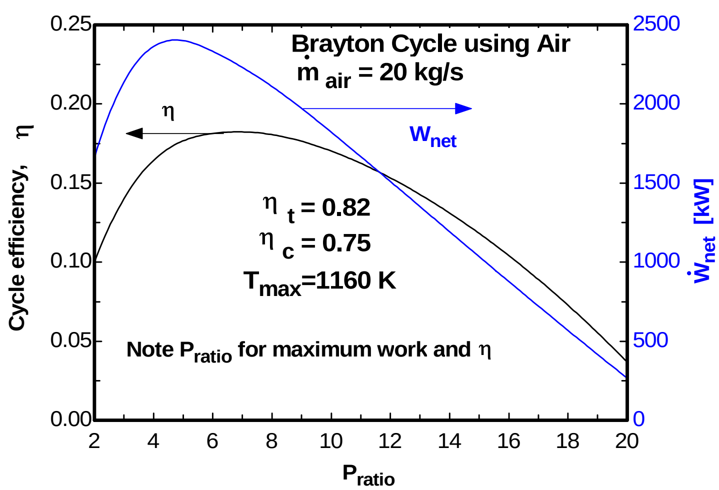
9-193 The effects of pressure ratio, maximum cycle temperature, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with helium as the working fluid is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
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 anaysis"
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





9-194 The effect of the number of compression and expansion stages on the thermal efficiency of an ideal regenerative Brayton cycle with multistage compression and expansion and air as the working fluid is to be investigated.

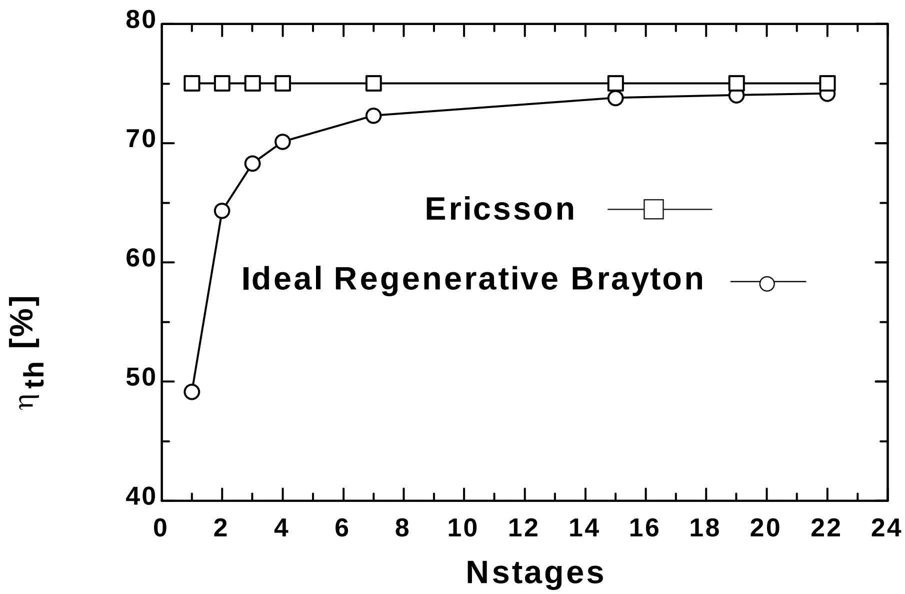
Analysis Using EES, the problem is solved as follows:

```
"Input data for air"
C_P = 1.005 [kJ/kg-K]
k = 1.4
"Nstages is the number of compression and expansion stages"
Nstages = 1
T_6 = 1200 [K]
Pratio = 12
T_1 = 300 [K]
P_1= 100 [kPa]
Eta_reg = 1.0 "regenerator effectiveness"
Eta_c =1.0 "Compressor isentorpic efficiency"
Eta_t =1.0 "Turbine isentropic efficiency"
R_p = Pratio^(1/Nstages)
"Isentropic Compressor anaysis"
T_2s = T_1*R_p^((k-1)/k)
P_2 = R_p*P_1
"T_2s 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_2s-T_1)
"Actual compressor analysis:"
 w_comp = C_P*(T_2 - T_1)

"Since intercooling is assumed to occur such that T_3 = T_1 and the compressors have the same pressure
ratio, the work input to each compressor is the same.  The total compressor work is:"
w_comp_total = Nstages*w_comp
"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"
"The heat added in the external heat exchanger + the reheat between turbines is"
q_in_total = C_P*(T_6 - T_5) +(Nstages - 1)*C_P*(T_8 - T_7)
"Reheat is assumed to occur until:"
T_8 = T_6
"Turbine analysis"
P_7 = P_6 /R_p
"T_7s is the isentropic value of T_7 at turbine exit"
T_7s = T_6*(1/R_p)^((k-1)/k)
"Turbine adiabatic efficiency, w_turbisen > w_turb"
Eta_t = w_turb /w_turbisen
"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_6 -  T_7s)
"Actual Turbine analysis:"
w_turb = C_P*(T_6 -  T_7)
w_turb_total = Nstages*w_turb

"Cycle analysis"
w_net=w_turb_total-w_comp_total "[kJ/kg]"
Bwr=w_comp/w_turb "Back work ratio"
```

P_4=P_2
P_5=P_4
P_6=P_5
T_4 = T_2
"The regenerator effectiveness gives T_5 as:"
Eta_reg = (T_5 - T_4)/(T_9 - T_4)
T_9 = T_7
"Energy balance on regenerator gives T_10 as:"
T_4 + T_9=T_5 + T_10
"Cycle thermal efficiency with regenerator"
Eta_th_regenerative=w_net/q_in_total*Convert(, %) "[%]"

"The efficiency of the Ericsson cycle is the same as the Carnot cycle operating between the same max and min temperatures, T_6 and T_1 for this problem."
Eta_th_Ericsson = (1 - T_1/T_6)*Convert(, %) "[%]"

$\eta_{th,Ericksson}$ [%]	$\eta_{th,Regenerative}$ [%]	Nstages
75	49.15	1
75	64.35	2
75	68.32	3
75	70.14	4
75	72.33	7
75	73.79	15
75	74.05	19
75	74.18	22



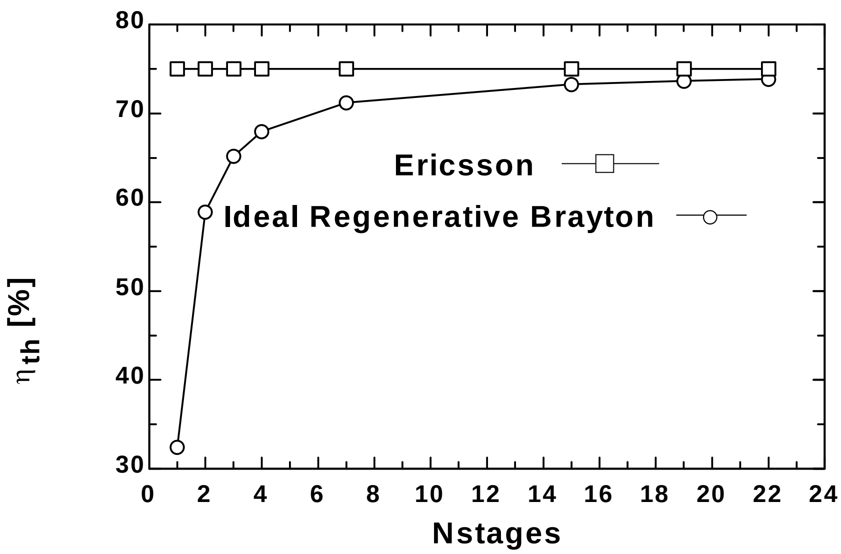
9-195 The effect of the number of compression and expansion stages on the thermal efficiency of an ideal regenerative Brayton cycle with multistage compression and expansion and helium as the working fluid is to be investigated.

Analysis Using EES, the problem is solved as follows:

```
"Input data for Helium"
C_P = 5.1926 [kJ/kg-K]
k = 1.667
"Nstages is the number of compression and expansion stages"
{Nstages = 1}
T_6 = 1200 [K]
Pratio = 12
T_1 = 300 [K]
P_1= 100 [kPa]
Eta_reg = 1.0 "regenerator effectiveness"
Eta_c =1.0 "Compressor isentorpic efficiency"
Eta_t =1.0 "Turbine isentropic efficiency"
R_p = Pratio^(1/Nstages)
"Isentropic Compressor anaysis"

T_2s = T_1*R_p^((k-1)/k)
P_2 = R_p*P_1
"T_2s 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_2s-T_1)
"Actual compressor analysis:"
 w_comp = C_P*(T_2 - T_1)
"Since intercooling is assumed to occur such that T_3 = T_1 and the compressors have the same pressure
ratio, the work input to each compressor is the same.  The total compressor work is:"
w_comp_total = Nstages*w_comp
"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"

"The heat added in the external heat exchanger + the reheat between turbines is"
q_in_total = C_P*(T_6 - T_5) +(Nstages - 1)*C_P*(T_8 - T_7)
"Reheat is assumed to occur until:"
T_8 = T_6
"Turbine analysis"
P_7 = P_6 /R_p
"T_7s is the isentropic value of T_7 at turbine exit"
T_7s = T_6*(1/R_p)^((k-1)/k)
"Turbine adiabatic efficiency, w_turbisen > w_turb"
Eta_t = w_turb /w_turbisen
"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_6 -  T_7s)

"Actual Turbine analysis:"
w_turb = C_P*(T_6 -  T_7)
w_turb_total = Nstages*w_turb

"Cycle analysis"
w_net=w_turb_total-w_comp_total
```

```
Bwr=w_comp/w_turb "Back work ratio"

P_4=P_2
P_5=P_4
P_6=P_5
T_4 = T_2
"The regenerator effectiveness gives T_5 as:"
Eta_reg = (T_5 - T_4)/(T_9 - T_4)
T_9 = T_7
"Energy balance on regenerator gives T_10 as:"
T_4 + T_9=T_5  + T_10

"Cycle thermal efficiency with regenerator"
Eta_th_regenerative=w_net/q_in_total*Convert(, %) "[%]"

"The efficiency of the Ericsson cycle is the same as the Carnot cycle operating between the same max and min temperatures, T_6 and T_1 for this problem."
Eta_th_Ericsson = (1 - T_1/T_6)*Convert(, %) "[%]"
```

$\eta_{th,Ericksson}$ [%]	$\eta_{th,Regenerative}$ [%]	Nstages
75	32.43	1
75	58.9	2
75	65.18	3
75	67.95	4
75	71.18	7
75	73.29	15
75	73.66	19
75	73.84	22



Fundamentals of Engineering (FE) Exam Problems

9-196 An Otto cycle with air as the working fluid has a compression ratio of 10.4. Under cold air standard conditions, the thermal efficiency of this cycle is

(a) 10% (b) 39% (c) 61% (d) 79% (e) 82%

Answer (c) 61%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
r=10.4
k=1.4
Eta_Otto=1-1/r^(k-1)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1/r "Taking efficiency to be 1/r"
W2_Eta = 1/r^(k-1)  "Using incorrect relation"
W3_Eta =  1-1/r^(k1-1); k1=1.667 "Using wrong k value"
```

9-197 For specified limits for the maximum and minimum temperatures, the ideal cycle with the lowest thermal efficiency is

(a) Carnot (b) Stirling (c) Ericsson (d) Otto (e) All are the same

Answer (d) Otto

9-198 A Carnot cycle operates between the temperatures limits of 300 K and 2000 K, and produces 600 kW of net power. The rate of entropy change of the working fluid during the heat addition process is

(a) 0 (b) 0.300 kW/K (c) 0.353 kW/K (d) 0.261 kW/K (e) 2.0 kW/K

Answer (c) 0.353 kW/K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=300 "K"
TH=2000 "K"
Wnet=600 "kJ/s"
Wnet= (TH-TL)*DS

"Some Wrong Solutions with Common Mistakes:"
W1_DS = Wnet/TH "Using TH instead of TH-TL"
W2_DS = Wnet/TL "Using TL instead of TH-TL"
W3_DS =  Wnet/(TH+TL) "Using TH+TL  instead of TH-TL"
```

9-199 Air in an ideal Diesel cycle is compressed from 2 L to 0.13 L, and then it expands during the constant pressure heat addition process to 0.30 L. Under cold air standard conditions, the thermal efficiency of this cycle is

(a) 41% (b) 59% (c) 66% (d) 70% (e) 78%

Answer (b) 59%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V1=2 "L"
V2= 0.13 "L"
V3= 0.30 "L"
r=V1/V2
rc=V3/V2
k=1.4
Eta_Diesel=1-(1/r^(k-1))*(rc^k-1)/k/(rc-1)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1-(1/r1^(k-1))*(rc^k-1)/k/(rc-1); r1=V1/V3 "Wrong r value"
W2_Eta = 1-Eta_Diesel  "Using incorrect relation"
W3_Eta =  1-(1/r^(k1-1))*(rc^k1-1)/k1/(rc-1); k1=1.667 "Using wrong k value"
W4_Eta = 1-1/r^(k-1) "Using Otto cycle efficiency"
```

9-200 Helium gas in an ideal Otto cycle is compressed from 20°C and 2.5 L to 0.25 L, and its temperature increases by an additional 700°C during the heat addition process. The temperature of helium before the expansion process is

(a) 1790°C (b) 2060°C (c) 1240°C (d) 620°C (e) 820°C

Answer (a) 1790°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
V1=2.5
V2=0.25
r=V1/V2
T1=20+273 "K"
T2=T1*r^(k-1)
T3=T2+700-273 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T3 =T22+700-273; T22=T1*r^(k1-1); k1=1.4 "Using wrong k value"
W2_T3 = T3+273  "Using K instead of C"
W3_T3 = T1+700-273 "Disregarding temp rise during compression"
W4_T3 = T222+700-273; T222=(T1-273)*r^(k-1) "Using C for T1 instead of K"
```

9-201 In an ideal Otto cycle, air is compressed from 1.20 kg/m^3 and 2.2 L to 0.26 L, and the net work output of the cycle is 440 kJ/kg. The mean effective pressure (MEP) for this cycle is

(a) 612 kPa (b) 599 kPa (c) 528 kPa (d) 416 kPa (e) 367 kPa

Answer (b) 599 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho1=1.20 "kg/m^3"
k=1.4
V1=2.2
V2=0.26
m=rho1*V1/1000 "kg"
w_net=440 "kJ/kg"
Wtotal=m*w_net
MEP=Wtotal/((V1-V2)/1000)

"Some Wrong Solutions with Common Mistakes:"
W1_MEP = w_net/((V1-V2)/1000) "Disregarding mass"
W2_MEP = Wtotal/(V1/1000)  "Using V1 instead of V1-V2"
W3_MEP = (rho1*V2/1000)*w_net/((V1-V2)/1000);  "Finding mass using V2 instead of V1"
W4_MEP = Wtotal/((V1+V2)/1000) "Adding V1 and V2 instead of subtracting"
```

9-202 In an ideal Brayton cycle, air is compressed from 95 kPa and 25°C to 1100 kPa. Under cold air standard conditions, the thermal efficiency of this cycle is

(a) 45% (b) 50% (c) 62% (d) 73% (e) 86%

Answer (b) 50%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=95 "kPa"
P2=1100 "kPa"
T1=25+273 "K"
rp=P2/P1
k=1.4
Eta_Brayton=1-1/rp^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1/rp "Taking efficiency to be 1/rp"
W2_Eta = 1/rp^((k-1)/k)  "Using incorrect relation"
W3_Eta = 1-1/rp^((k1-1)/k1); k1=1.667 "Using wrong k value"
```

9-203 Consider an ideal Brayton cycle executed between the pressure limits of 1200 kPa and 100 kPa and temperature limits of 20°C and 1000°C with argon as the working fluid. The net work output of the cycle is

(a) 68 kJ/kg (b) 93 kJ/kg (c) 158 kJ/kg (d) 186 kJ/kg (e) 310 kJ/kg

Answer (c) 158 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 "kPa"
P2=1200 "kPa"
T1=20+273 "K"
T3=1000+273 "K"
rp=P2/P1
k=1.667
Cp=0.5203 "kJ/kg.K"
Cv=0.3122 "kJ/kg.K"
T2=T1*rp^((k-1)/k)
q_in=Cp*(T3-T2)
Eta_Brayton=1-1/rp^((k-1)/k)
w_net=Eta_Brayton*q_in

"Some Wrong Solutions with Common Mistakes:"
W1_wnet = (1-1/rp^((k-1)/k))*qin1;  qin1=Cv*(T3-T2) "Using Cv instead of Cp"
W2_wnet = (1-1/rp^((k-1)/k))*qin2;  qin2=1.005*(T3-T2)  "Using Cp of air instead of argon"
W3_wnet = (1-1/rp^((k1-1)/k1))*Cp*(T3-T22); T22=T1*rp^((k1-1)/k1); k1=1.4  "Using k of air instead of argon"
W4_wnet = (1-1/rp^((k-1)/k))*Cp*(T3-T222); T222=(T1-273)*rp^((k-1)/k) "Using C for T1 instead of K"
```

9-204 An ideal Brayton cycle has a net work output of 150 kJ/kg and a backwork ratio of 0.4. If both the turbine and the compressor had an isentropic efficiency of 85%, the net work output of the cycle would be

(a) 74 kJ/kg (b) 95 kJ/kg (c) 109 kJ/kg (d) 128 kJ/kg (e) 177 kJ/kg

Answer (b) 95 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
wcomp/wturb=0.4
wturb-wcomp=150 "kJ/kg"
Eff=0.85
w_net=Eff*wturb-wcomp/Eff

"Some Wrong Solutions with Common Mistakes:"
W1_wnet = Eff*wturb-wcomp*Eff "Making a mistake in Wnet relation"
W2_wnet = (wturb-wcomp)/Eff "Using a wrong relation"
W3_wnet = wturb/eff-wcomp*Eff "Using a wrong relation"
```

9-205 In an ideal Brayton cycle, air is compressed from 100 kPa and 25°C to 1 MPa, and then heated to 927°C before entering the turbine. Under cold air standard conditions, the air temperature at the turbine exit is

(a) 349°C (b) 426°C (c) 622°C (d) 733°C (e) 825°C

Answer (a) 349°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 "kPa"
P2=1000 "kPa"
T1=25+273 "K"
T3=900+273 "K"
rp=P2/P1
k=1.4
T4=T3*(1/rp)^((k-1)/k)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T4 = T3/rp "Using wrong relation"
W2_T4 = (T3-273)/rp "Using wrong relation"
W3_T4 = T4+273  "Using K instead of C"
W4_T4 = T1+800-273 "Disregarding temp rise during compression"
```

9-206 In an ideal Brayton cycle with regeneration, argon gas is compressed from 100 kPa and 25°C to 400 kPa, and then heated to 1200°C before entering the turbine. The highest temperature that argon can be heated in the regenerator is

(a) 246°C (b) 846°C (c) 689°C (d) 368°C (e) 573°C

Answer (e) 573°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=0.5203 "kJ/kg.K"
P1=100 "kPa"
P2=400 "kPa"
T1=25+273 "K"
T3=1200+273 "K"
"The highest temperature that argon can be heated in the regenerator is the turbine exit temperature,"
rp=P2/P1
T2=T1*rp^((k-1)/k)
T4=T3/rp^((k-1)/k)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T4 = T3/rp "Using wrong relation"
W2_T4 = (T3-273)/rp^((k-1)/k) "Using C instead of K for T3"
W3_T4 = T4+273  "Using K instead of C"
W4_T4 = T2-273 "Taking compressor exit temp as the answer"
```

9-207 In an ideal Brayton cycle with regeneration, air is compressed from 80 kPa and 10°C to 400 kPa and 175°C, is heated to 450°C in the regenerator, and then further heated to 1000°C before entering the turbine. Under cold air standard conditions, the effectiveness of the regenerator is

(a) 33% (b) 44% (c) 62% (d) 77% (e) 89%

Answer (d) 77%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
P1=80 "kPa"
P2=400 "kPa"
T1=10+273 "K"
T2=175+273 "K"
T3=1000+273 "K"
T5=450+273 "K"
"The highest temperature that the gas can be heated in the regenerator is the turbine exit temperature,"
rp=P2/P1
T2check=T1*rp^((k-1)/k) "Checking the given value of T2. It checks."
T4=T3/rp^((k-1)/k)
Effective=(T5-T2)/(T4-T2)

"Some Wrong Solutions with Common Mistakes:"
W1_eff = (T5-T2)/(T3-T2) "Using wrong relation"
W2_eff = (T5-T2)/(T44-T2); T44=(T3-273)/rp^((k-1)/k) "Using C instead of K for T3"
W3_eff = (T5-T2)/(T444-T2);  T444=T3/rp "Using wrong relation for T4"
```

9-208 Consider a gas turbine that has a pressure ratio of 6 and operates on the Brayton cycle with regeneration between the temperature limits of 20°C and 900°C. If the specific heat ratio of the working fluid is 1.3, the highest thermal efficiency this gas turbine can have is

(a) 38% (b) 46% (c) 62% (d) 58% (e) 97%

Answer (c) 62%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.3
rp=6
T1=20+273 "K"
T3=900+273 "K"
Eta_regen=1-(T1/T3)*rp^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1-((T1-273)/(T3-273))*rp^((k-1)/k) "Using C for temperatures instead of K"
W2_Eta = (T1/T3)*rp^((k-1)/k)  "Using incorrect relation"
W3_Eta = 1-(T1/T3)*rp^((k1-1)/k1); k1=1.4 "Using wrong k value (the one for air)"
```

9-209 An ideal gas turbine cycle with many stages of compression and expansion and a regenerator of 100 percent effectiveness has an overall pressure ratio of 10. Air enters every stage of compressor at 290 K, and every stage of turbine at 1200 K. The thermal efficiency of this gas-turbine cycle is

(a) 36% (b) 40% (c) 52% (d) 64% (e) 76%

Answer (e) 76%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
rp=10
T1=290 "K"
T3=1200 "K"
Eff=1-T1/T3

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 100
W2_Eta = 1-1/rp^((k-1)/k) "Using incorrect relation"
W3_Eta = 1-(T1/T3)*rp^((k-1)/k) "Using wrong relation"
W4_Eta = T1/T3 "Using wrong relation"
```

9-210 Air enters a turbojet engine at 320 m/s at a rate of 30 kg/s, and exits at 650 m/s relative to the aircraft. The thrust developed by the engine is

(a) 5 kN (b) 10 kN (c) 15 kN (d) 20 kN (e) 26 kN

Answer (b) 10 kN

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel1=320 "m/s"
Vel2=650 "m/s"
Thrust=m*(Vel2-Vel1)/1000 "kN"
m= 30 "kg/s"

"Some Wrong Solutions with Common Mistakes:"
W1_thrust = (Vel2-Vel1)/1000 "Disregarding mass flow rate"
W2_thrust = m*Vel2/1000 "Using incorrect relation"
```

9-211 ··· 9-219 Design and Essay Problems.