Guía 7

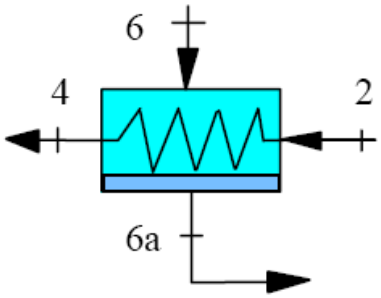
Andrés Yenes S

1. Borgnakke 7th, P 11.50.

A closed feedwater heater in a regenerative steam power cycle heats 20 kg/s of water from 100°C, 20 MPa to 250°C, 20 MPa. The extraction steam from the turbine enters the heater at 4 MPa, 275°C, and leaves as saturated liquid. What is the required mass flow rate of the extraction steam?

Solution:

The schematic from Figure 11.11 has the feedwater from the pump coming at state 2 being heated by the extraction flow coming from the turbine state 6 so the feedwater leaves as saturated liquid state 4 and the extraction flow leaves as condensate state 6a.



From table B.1 h kJ/kg

B.1.4: 100°C, 20 MPa $h_2 = 434.06$

B.1.4: 250°C, 20 MPa $h_4 = 1086.75$

B.1.3: 4 MPa, 275°C $h_6 = 2886.2$

B.1.2: 4 MPa, sat. liq. $h_{6a} = 1087.31$

C.V. Feedwater Heater

Energy Eq.: $\dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_2 h_4 + \dot{m}_6 h_{6a}$

Since all four states are known we can solve for the extraction flow rate

$$\dot{m}_6 = \dot{m}_2 \frac{h_2 - h_4}{h_{6a} - h_6}$$

$$= 20 \frac{434.06 - 1086.75}{1087.31 - 2886.2} \text{ kg/s} = \mathbf{7.257\ kg/s}$$

2. Borgnakke 7th, P 11.129.

Steam enters the turbine of a power plant at 5 MPa and 400°C, and exhausts to the condenser at 10 kPa. The turbine produces a power output of 20 000 kW with an isentropic efficiency of 85%. What is the mass flow rate of steam around the cycle and the rate of heat rejection in the condenser? Find the thermal efficiency of the power plant.

Solution: $\dot{W}_T = 20\ 000$ kW and $\eta_{Ts} = 85\ \%$

State 3: $h_3 = 3195.6$ kJ/kg , $s_3 = 6.6458$ kJ/kgK

State 1: $P_1 = P_4 = 10$ kPa , sat liq , $x_1 = 0$

$T_1 = 45.8°C$, $h_1 = h_f = 191.8$ kJ/kg , $v_1 = v_f = 0.00101$ m^3/kg

C.V Turbine : Energy Eq.: $q_T + h_3 = h_4 + w_T$; $q_T = 0$

$w_T = h_3 - h_4$, Assume Turbine is isentropic

$s_{4s} = s_3 = 6.6458$ kJ/kgK , $s_{4s} = s_f + x_{4s}\, s_{fg}$, solve for $x_{4s} = 0.7994$

$h_{4s} = h_f + x_{4s} h_{fg} = 1091.0$ kJ/kg

$w_{Ts} = h_3 - h_{4s} = 1091$ kJ/kg , $w_T = \eta_{Ts} w_{Ts} = 927.3$ kJ/kg

$$\dot{m} = \frac{\dot{W}_T}{w_T} = \mathbf{21.568\ kg/s}\ , \qquad h_4 = h_3 - w_T = 2268.3\ \text{kJ/kg}$$

C.V. Condenser: Energy Eq.: $h_4 = h_1 + q_c + w_c$; $w_c = 0$

$q_c = h_4 - h_1 = 2076.5$ kJ/kg , $\dot{Q}_c = \dot{m}\, q_c = \mathbf{44786\ kW}$

C.V. Pump: Assume adiabatic, reversible and incompressible flow

$w_{ps} = \int v\, dP = v_1(P_2 - P_1) = 5.04$ kJ/kg

Energy Eq.: $h_2 = h_1 + w_p = 196.8$ kJ/kg

C.V Boiler : Energy Eq.: $q_B + h_2 = h_3 + w_B$; $w_B = 0$

$q_B = h_3 - h_2 = 2998.8$ kJ/kg

$w_{net} = w_T - w_P = 922.3$ kJ/kg

$\eta_{th} = w_{net} / q_B = \mathbf{0.307}$

3. Borgnakke 7th, P 11.90

A refrigerator using R-134a is located in a 20°C room. Consider the cycle to be ideal, except that the compressor is neither adiabatic nor reversible. Saturated vapor at -20°C enters the compressor, and the R-134a exits the compressor at 50°C. The condenser temperature is 40°C. The mass flow rate of refrigerant around the cycle is 0.2 kg/s, and the coefficient of performance is measured and found to be 2.3. Find the power input to the compressor and the rate of entropy generation in the compressor process.

Solution:

Table B.5: $P_2 = P_3 = P_{sat\ 40C} = 1017$ kPa, $h_4 = h_3 = 256.54$ kJ/kg

$s_2 \approx 1.7472$ kJ/kg K, $h_2 \approx 430.87$ kJ/kg;

$s_1 = 1.7395$ kJ/kg K, $h_1 = 386.08$ kJ/kg

$\beta = q_L / w_C \rightarrow w_C = q_L / \beta = (h_1 - h_4) / \beta = (386.08 - 256.54) / 2.3 = 56.32$

$$\dot{W}_C = \dot{m}\, w_C = \mathbf{11.26\ kW}$$

C.V. Compressor $h_1 + w_C + q = h_2 \rightarrow$

$q_{in} = h_2 - h_1 - w_C = 430.87 - 386.08 - 56.32 = -11.53$ kJ/kg i.e. a heat loss

$s_1 + \int dQ/T + s_{gen} = s_2$

$s_{gen} = s_2 - s_1 - q / T_o = 1.7472 - 1.7395 + (11.53 / 293.15) = 0.047$ kJ/kg K

$$\dot{S}_{gen} = \dot{m}\, s_{gen} = 0.2 \times 0.047 = \mathbf{0.0094\ kW / K}$$

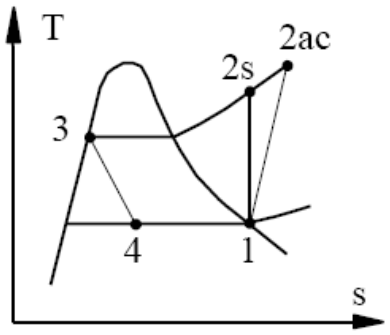
Ideal refrigeration cycle
with actual compressor

$T_{cond} = 40^oC$

$T_2 = 50^oC$

$T_{evap} = -20^oC = T_1$

Properties from Table B.5



4. Scahum's 2nd, Capítulo 10, Example 10.2

Solution: From Appendix D we find, using $T_3 = 38\,^{\circ}\text{C}$, that $h_3 = h_4 = 103.2$ kJ/kg. Also, from Table D-1 we observe that $P_1 \cong 0.18$ MPa. From Table D-3, at $P_1 = 0.18$ MPa and $T_1 = -13\,^{\circ}\text{C}$,

$$h_1 = 240 \text{ kJ/kg} \qquad s_1 = 0.930 \text{ kJ/kg·K}$$

If the compressor were isentropic, then, with $s_{2'} = s_1$ and $P_2 = 1.0$ MPa,

$$h_{2'} = \left(\frac{0.93 - 0.9066}{0.9428 - 0.9066}\right)(280.2 - 268.7) + 268.7 = 276 \text{ kJ/kg}$$

From the definition of efficiency, $\eta = w_s/w_a$, we have

$$0.8 = \frac{h_{2'} - h_1}{h_2 - h_1} = \frac{276 - 240}{h_2 - 240} \qquad \therefore h_2 = 285 \text{ kJ/kg}$$

The rate of refrigeration is $\dot{Q}_E = (0.6)(240 - 103.2) = 82$ kW. Note that the real effects have actually increased the capability to refrigerate a space. The coefficient of performance becomes

$$\text{COP}_R = \frac{82}{(0.6)(285 - 240)} = 3.04$$

The decrease in the COP_R occurs because the power input to the compressor has increased.

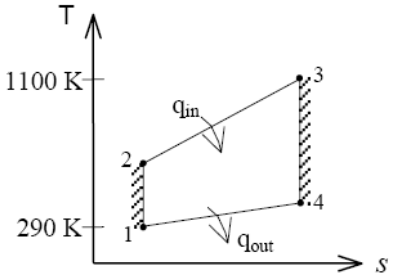
5. Lamentablemente no la pude encontrar.

6. Cengel 5th, P 9 – 77

9-77 A stationary gas-turbine power plant operates on a simple ideal Brayton cycle with air as the working fluid. The power delivered by this plant is to be determined assuming constant and variable specific heats.

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

Analysis (*a*) Assuming constant specific heats,



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

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

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

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

(*b*) Assuming variable specific heats (Table A-17),

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

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

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

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

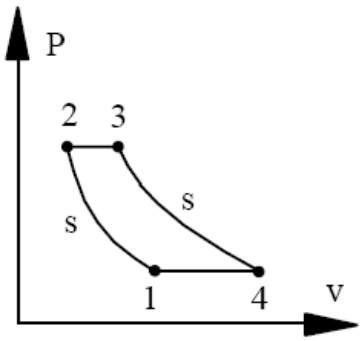
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} == 1 - \frac{h_4 - h_1}{h_3 - h_2} = 1 - \frac{651.37 - 290.16}{1161.07 - 526.11} = 0.431$$

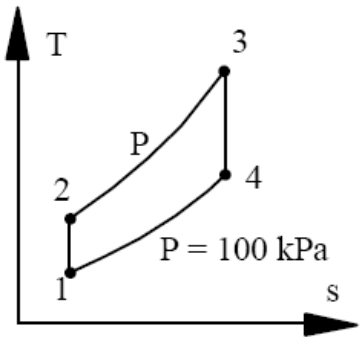
$$\dot{W}_{net,out} = \eta_T\dot{Q}_{in} = (0.431)(35{,}000\text{ kW}) = \mathbf{15{,}085\text{ kW}}$$

7. Borgnakke 7th, 12.17

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

Solution:





Compression ratio

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

Max temperature

$T_3 = 1100°C$

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

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

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

Energy equation with compressor work in

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

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

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

Energy equation with turbine work out

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

Scale the work with the mass flow rate

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

Energy added by the combustion process

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

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

8. Borgnakke 7th, P 12.51

The turbine in a jet engine receives air at 1250 K, 1.5 MPa. It exhausts to a nozzle at 250 kPa, which in turn exhausts to the atmosphere at 100 kPa. The isentropic efficiency of the turbine is 85% and the nozzle efficiency is 95%. Find the nozzle inlet temperature and the nozzle exit velocity. Assume negligible kinetic energy out of the turbine.

C.V. Turbine: First the ideal (reversible and adiabatic i.e. constant s, Eq.8.32)

$$T_{es} = T_i\,(P_e/P_i) = 1250\,(250/1500)^{(k-1)/k} = 749.2\ \text{K}$$

Energy Eq.: $w_{T,s} = h_i - h_{es} = C_p\,(T_i - T_{es}) = 1.004\,(1250 - 749.2) = 502.8\ \text{kJ/kg}$

Eq.9.27: $w_{T,AC} = w_{T,s} \times \eta_T = 427.4\ \text{kJ/kg} = h_i - h_{e,AC} = C_p\,(T_i - T_{e,AC})$

$$\Rightarrow\ T_{e,AC} = 1250 - 427.4 / 1.004 = \mathbf{824.3\ K},$$

C.V. Nozzle: First the ideal (reversible and adiabatic i.e. constant s, Eq.8.32)

$$\Rightarrow\ T_{es} = T_i\,(P_e/P_i)^{(k-1)/k} = 824.3\,(100/250)^{(k-1)/k} = 634.4\ \text{K}$$

Energy Eq.: $(1/2)\mathbf{V}_{e,s}^2 = h_i - h_{e,s} = 1.004(824.3 - 634.4) = 190.66\ \text{kJ/kg}$

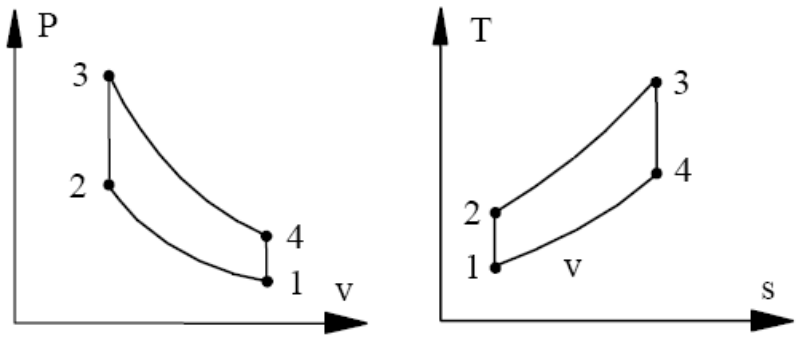
Eq.9.30: $(1/2)\mathbf{V}_{e,AC}^2 = (1/2)\mathbf{V}_{e,s}^2 \times \eta_{NOZ} = 181.13\ \text{kJ/kg}$

$$\mathbf{V}_{e,AC} = \sqrt{2 \times 1000 \times 181.13} = \mathbf{602\ m/s}$$

9. Borgnakke 7th, P 12.67

To approximate an actual spark-ignition engine consider an air-standard Otto cycle that has a heat addition of 1800 kJ/kg of air, a compression ratio of 7, and a pressure and temperature at the beginning of the compression process of 90 kPa, 10°C. Assuming constant specific heat, with the value from Table A.5, determine the maximum pressure and temperature of the cycle, the thermal efficiency of the cycle and the mean effective pressure.

Solution:



Compression: Reversible and adiabatic so constant s from Eq.8.33-34

$$P_2 = P_1(v_1/v_2)^k = 90(7)^{1.4} = 1372 \text{ kPa}$$

$$T_2 = T_1(v_1/v_2)^{k-1} = 283.2 \times (7)^{0.4} = 616.6 \text{ K}$$

Combustion: constant volume

$$T_3 = T_2 + q_H/C_{V0} = 616.6 + 1800/0.717 = \mathbf{3127\ K}$$

$$P_3 = P_2T_3/T_2 = 1372 \times 3127 / 616.6 = \mathbf{6958\ kPa}$$

Efficiency and net work

$$\eta_{TH} = 1 - T_1/T_2 = 1 - 283.2/616.5 = \mathbf{0.541}$$

$$w_{net} = \eta_{TH} \times q_H = 0.541 \times 1800 = 973.8 \text{ kJ/kg}$$

Displacement and P_{meff}

$$v_1 = RT_1/P_1 = (0.287 \times 283.2)/90 = 0.9029 \text{ m}^3/\text{kg}$$

$$v_2 = (1/7)\, v_1 = 0.1290 \text{ m}^3/\text{kg}$$

$$P_{meff} = \frac{w_{NET}}{v_1 - v_2} = \frac{973.8}{0.9029 - 0.129} = \mathbf{1258\ kPa}$$

10. Cengel 5th, P 9 – 34

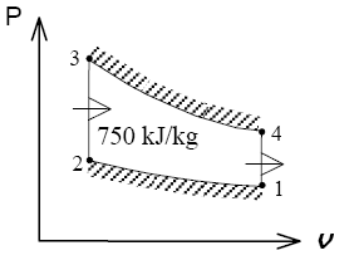
9-34 An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The pressure and temperature at the end of the heat addition process, the net work output, the thermal efficiency, and the mean effective pressure for the cycle are to be determined.

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

Properties The gas constant of air is $R = 0.287$ kJ/kg.K. The properties of air are given in Table A-17.

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

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



$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{8}(621.2) = 77.65 \longrightarrow \begin{matrix} T_2 = 673.1\,\text{K} \\ u_2 = 491.2\ \text{kJ/kg} \end{matrix}$$

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

Process 2-3: v = constant heat addition.

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

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

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

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

Process 4-1: v = constant heat rejection.

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

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 750 - 357.62 = \mathbf{392.4\ kJ/kg}$$

(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{392.4\ \text{kJ/kg}}{750\ \text{kJ/kg}} = \mathbf{52.3\%}$$

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

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

$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1-1/r)} = \frac{392.4\ \text{kJ/kg}}{(0.906\ \text{m}^3/\text{kg})(1-1/8)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{495.0\ kPa}$$

11. Borgnakke 7th, P 12.66

A gasoline engine has a volumetric compression ratio of 8 and before compression has air at 280 K, 85 kPa. The combustion generates a peak pressure of 6500 kPa. Find the peak temperature, the energy added by the combustion process and the exhaust temperature.

Solution:

Solve the problem with cold air properties.

Compression. Isentropic so we use Eqs.8.33-8.34

$$P_2 = P_1(v_1/v_2)^k = 85(8)^{1.4} = 1562 \text{ kPa}$$

$$T_2 = T_1(v_1/v_2)^{k-1} = 280(8)^{0.4} = 643.3 \text{ K}$$

Combustion. Constant volume

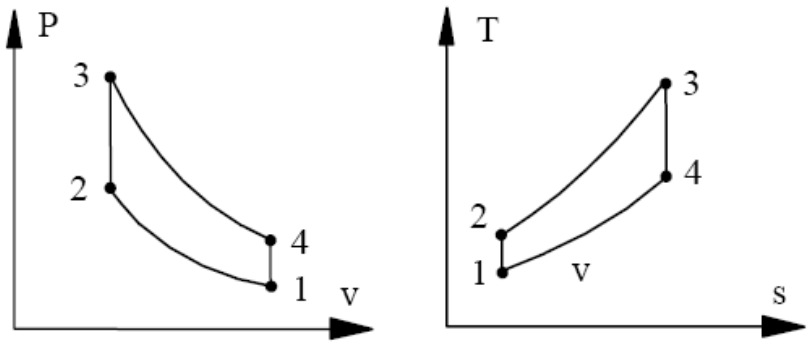
$$T_3 = T_2 (P_3/P_2) = 643.3 \times 6500/1562 = \mathbf{2677\ K}$$

$$q_H = u_3 - u_2 \approx C_v(T_3 - T_2)$$

$$= 0.717\ (2677 - 643.3) = \mathbf{1458\ kJ/kg}$$

Exhaust. Isentropic expansion so from Eq.8.33

$$T_4 = T_3/8^{0.4} = 2677/2.2974 = \mathbf{1165\ K}$$



12. Schaum's 2nd, Capítulo 8, Example 8.4

maximum possible thermal efficiency and compare with that of a Carnot cycle. [illegible]

Solution: The Otto cycle provides the model for this engine. The maximum possible thermal efficiency for the engine would be

$$\eta = 1 - \frac{1}{r^{k-1}} = 1 - \frac{1}{(10)^{0.4}} = 0.602 \quad \text{or } 60.2\%$$

Since process $1 \rightarrow 2$ is isentropic, we find that

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (473)(10)^{0.4} = 1188\ \text{K}$$

The net work for the cycle is given by

$$w_{\text{net}} = w_{1-2} + \cancel{w^0_{2-3}} + w_{3-4} + \cancel{w^0_{4-1}} = c_v(T_1 - T_2) + c_v(T_3 - T_4) \quad \text{or}$$
$$1000 = (0.717)(473 - 1188 + T_3 - T_4)$$

But, for the isentropic process $3 \rightarrow 4$,

$$T_3 = T_4\left(\frac{v_4}{v_3}\right)^{k-1} = (T_4)(10)^{0.4} = 2.512\,T_4$$

Solving the last two equations simultaneously, we find $T_3 = 3508$ K and $T_4 = 1397$ K, so that

$$\eta_{\text{carnot}} = 1 - \frac{T_L}{T_H} = 1 - \frac{473}{3508} = 0.865 \quad \text{or } 86.5\%$$

The Otto cycle efficiency is less than that of a Carnot cycle operating between the limiting temperatures because the heat transfer processes in the Otto cycle are not reversible.

The MEP is found by using the equation

$$w_{\text{net}} = (\text{MEP})(v_1 - v_2)$$

We have

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287)(473)}{200} = 0.6788\ \text{m}^3/\text{kg} \qquad \text{and} \qquad v_2 = \frac{v_1}{10}$$

Thus

$$\text{MEP} = \frac{w_{\text{net}}}{v_1 - v_2} = \frac{1000}{(0.9)(0.6788)} = 1640\ \text{kPa}$$

13. Cengel 5th, P 9 – 155

9-155 A spark-ignition engine operates on an ideal Otto cycle with a compression ratio of 11. The maximum temperature and pressure in the cycle, the net work per cycle and per cylinder, the thermal efficiency, the mean effective pressure, and the power output for a specified engine speed 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*) First the mass in one cylinder is determined as follows

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 11 = \frac{V_c + (0.0018)/4}{V_c} \longrightarrow V_c = 0.000045\ \text{m}^3 \text{ for one cylinder}$$

$$V_1 = V_c + V_d = 0.000045 + 0.00045 = 0.000495\ \text{m}^3$$

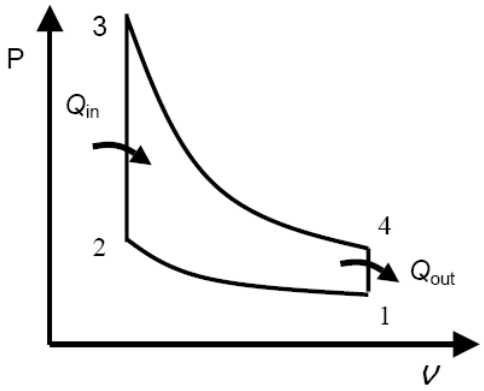
$$m = \frac{P_1 V_1}{RT_1} = \frac{(90\ \text{kPa})(0.000495\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323\ \text{K})} = 0.0004805\ \text{kg}$$

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: Isentropic compression

$$T_1 = 50°\text{C} = 323\ \text{K} \longrightarrow u_1 = 230.88\ \text{kJ/kg}$$

$$\left.\begin{matrix} T_1 = 50°\text{C} = 323\ \text{K} \\ P_1 = 90\ \text{kPa} \end{matrix}\right\} s_1 = 5.8100\ \text{kJ/kg}\cdot\text{K}$$



$$v_1 = \frac{V_1}{m} = \frac{0.000495\ \text{m}^3}{0.0004805\ \text{kg}} = 1.0302\ \text{m}^3/\text{kg}$$

$$V_2 = V_c = 0.000045\ \text{m}^3$$

$$v_2 = \frac{V_2}{m} = \frac{0.000045\ \text{m}^3}{0.0004805\ \text{kg}} = 0.09364\ \text{m}^3/\text{kg}$$

$$\left.\begin{matrix} s_2 = s_1 = 5.8100\ \text{kJ/kg.K} \\ v_2 = 0.09364\ \text{m}^3/\text{kg} \end{matrix}\right\} T_2 = 807.3\ \text{K}$$

$$T_2 = 807.3\ \text{K} \longrightarrow u_2 = 598.33\ \text{kJ/kg}$$

$$P_2 = P_1 \frac{V_1}{V_2}\frac{T_2}{T_1} = (90\ \text{kPa})\left(\frac{0.000495\ \text{m}^3}{0.000045\ \text{m}^3}\right)\left(\frac{807.3\ \text{K}}{323\ \text{K}}\right) = 2474\ \text{kPa}$$

Process 2-3: constant volume heat addition

$$Q_{in} = m(u_3 - u_2) \longrightarrow 1.5\ \text{kJ} = (0.0004805\ \text{kg})(u_3 - 598.33)\text{kJ/kg} \longrightarrow u_3 = 3719.8\ \text{kJ/kg}$$

$$u_2 = 598.33\ \text{kJ/kg} \longrightarrow T_3 = \mathbf{4037\ K}$$

$$P_3 = P_2\left(\frac{T_3}{T_2}\right) = (2474\ \text{kPa})\left(\frac{4037\ \text{K}}{807.3\ \text{K}}\right) = \mathbf{12{,}375\ kPa}$$

$$\left.\begin{matrix} T_3 = 4037\ \text{K} \\ P_3 = 12{,}375\ \text{kPa} \end{matrix}\right\} s_3 = 7.3218\ \text{kJ/kg}\cdot\text{K}$$

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

$$\left.\begin{aligned} s_4 &= s_3 = 7.3218\ \text{kJ/kg.K} \\ v_4 &= v_1 = 1.0302\ \text{m}^3\ \text{/kg} \end{aligned}\right\} T_4 = 2028\ \text{K}$$

$$T_4 = 2028\ \text{K} \longrightarrow u_4 = 1703.6\ \text{kJ/kg}$$

$$P_4 = P_3 \frac{v_3}{v_4}\frac{T_4}{T_3} = (12{,}375\ \text{kPa})\left(\frac{1}{11}\right)\left(\frac{2028\ \text{K}}{4037\ \text{K}}\right) = 565\ \text{kPa}$$

Process 4-1: constant voume heat rejection

$$Q_{\text{out}} = m(u_4 - u_1) = (0.0004805\ \text{kg})(1703.6 - 230.88)\text{kJ/kg} = 0.7077\ \text{kJ}$$

The net work output and the thermal efficiency are

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 1.5 - 0.7077 = \mathbf{0.792\ kJ}\ \text{(per cycle per cylinder)}$$

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_{\text{in}}} = \frac{0.792\ \text{kJ}}{1.5\ \text{kJ}} = \mathbf{0.528}$$

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

$$\text{MEP} = \frac{W_{\text{net,out}}}{v_1 - v_2} = \frac{0.7923\ \text{kJ}}{(0.000495 - 0.000045)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{1761\ kPa}$$

(*d*) The power for engine speed of 3000 rpm is

$$\dot{W}_{\text{net}} = n_{\text{cyl}} W_{\text{net}} \frac{\dot{n}}{2} = (4\ \text{cylinder})(0.792\ \text{kJ/cylinder - cycle})\frac{(3000\ \text{rev/min})}{(2\ \text{rev/cycle})}\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = \mathbf{79.2\ kW}$$

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