9–14 An air-standard cycle with variable specific heats is executed in a closed system and is composed of the following four processes:

1-2 Isentropic compression from 100 kPa and 27°C to 800 kPa

2-3 v = *constant* heat addition to 1800 K

3-4 Isentropic expansion to 100 kPa

4-1 P = *constant* heat rejection to initial state

(*a*) Show the cycle on P-v and T-s diagrams.

(*b*) Calculate the net work output per unit mass.

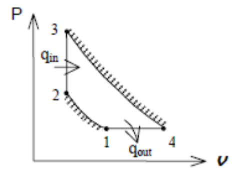
(*c*) Determine the thermal efficiency.

9-14 The four processes of an air-standard cycle are described. The cycle is to be shown on P-v and T-s diagrams, and the net work output and the thermal efficiency are to be determined.

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

Properties The properties of air are given in Table A-17.

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



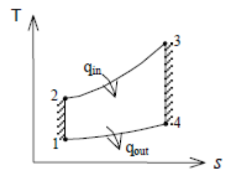
$$T_1 = 300\text{K} \longrightarrow \begin{array}{l} h_1 = 300.19 \text{ kJ/kg} \\ P_{r_1} = 1.386 \end{array}$$

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

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

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

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



From energy balances,

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

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

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

(*c*) Then the thermal efficiency becomes

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

9–16 An air-standard cycle is executed in a closed system and is composed of the following four processes:

1-2 Isentropic compression from 100 kPa and 27°C to 1 MPa

2-3 P = *constant* heat addition in amount of 2800 kJ/kg

3-4 v = *constant* heat rejection to 100 kPa

4-1 P = *constant* heat rejection to initial state

(*a*) Show the cycle on P-v and T-s diagrams.
(*b*) Calculate the maximum temperature in the cycle.
(*c*) Determine the thermal efficiency.

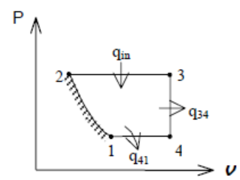
Assume constant specific heats at room temperature.
Answers: (*b*) 3360 K, (*c*) 21.0 percent

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 maximum temperature in the cycle and the thermal efficiency are to be determined.

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

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

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



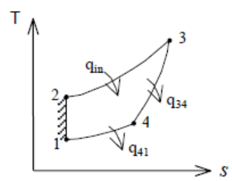
$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{1000\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} = 579.2\text{ K}$$

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

$$2800\text{ kJ/kg} = (1.005\text{ kJ/kg}\cdot\text{K})(T_3 - 579.2) \longrightarrow T_{max} = T_3 = \mathbf{3360\ \ K}$$

(*c*)

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



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

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

Discussion The assumption of constant specific heats at room temperature is not realistic in this case the temperature changes involved are too large.

9–19 An air-standard cycle is executed in a closed system with 0.004 kg of air and consists of the following three processes:

1-2 Isentropic compression from 100 kPa and 27°C to 1 MPa

2-3 P = *constant* heat addition in the amount of 2.76 kJ

3-1 $P = c_1 v + c_2$ heat rejection to initial state (c_1 and c_2 are constants)

(*a*) Show the cycle on *P-v* and *T-s* diagrams.

(*b*) Calculate the heat rejected.

(*c*) Determine the thermal efficiency.

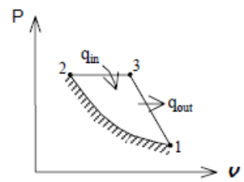
Assume constant specific heats at room temperature.

Answers: (*b*) 1.679 kJ, (*c*) 39.2 percent

9-19 The three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the heat rejected and the thermal efficiency are to be determined.

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

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



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

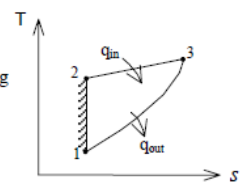
$$Q_{in} = m(h_3 - h_2) = mc_p(T_3 - T_2)$$

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

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

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

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

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

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

9–20 An air-standard cycle with variable specific heats is executed in a closed system with 0.003 kg of air and consists of the following three processes:

1-2 v = *constant* heat addition from 95 kPa and 17°C to 380 kPa

2-3 Isentropic expansion to 95 kPa

3-1 P = *constant* heat rejection to initial state

(*a*) Show the cycle on P-v and T-s diagrams.

(*b*) Calculate the net work per cycle, in kJ.

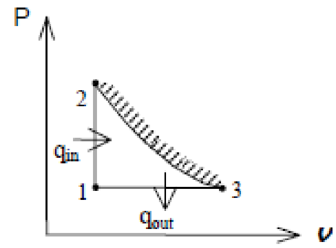
(*c*) Determine the thermal efficiency.

9-20 The three processes of an air-standard cycle are described. The cycle is to be shown on P-v and T-s diagrams, and the net work per cycle and the thermal efficiency are to be determined.

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

Properties The properties of air are given in Table A-17.

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

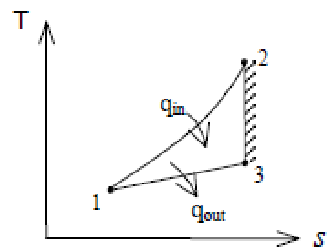


$$T_1 = 290 \text{ K} \longrightarrow \begin{array}{l} u_1 = 206.91 \text{ kJ/kg} \\ h_1 = 290.16 \text{ kJ/kg} \end{array}$$

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

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

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



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

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

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

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

9–22 Consider a Carnot cycle executed in a closed system with 0.003 kg of air. The temperature limits of the cycle are 300 and 900 K, and the minimum and maximum pressures that occur during the cycle are 20 and 2000 kPa. Assuming constant specific heats, determine the net work output per cycle.

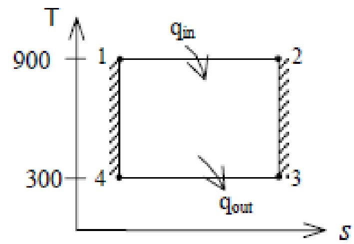
9-22 A Carnot cycle with the specified temperature limits is considered. The net work output per cycle is to be determined.

Assumptions Air is an ideal gas with constant specific heats.

Properties The properties of air at room temperature are c_p = 1.005 kJ/kg.K, c_v = 0.718 kJ/kg·K, 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,

$$P_2 = P_3\left(\frac{T_2}{T_3}\right)^{k/(k-1)} = (20 \text{ kPa})\left(\frac{900 \text{ K}}{300 \text{ K}}\right)^{1.4/0.4} = 935.3 \text{ kPa}$$

The heat input is determined from

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{935.3 \text{ kPa}}{2000 \text{ kPa}} = 0.2181 \text{ kJ/kg}\cdot\text{K}$$

$$Q_{in} = mT_H(s_2 - s_1) = (0.003 \text{ kg})(900 \text{ K})(0.2181 \text{ kJ/kg}\cdot\text{K}) = 0.5889 \text{ kJ}$$

Then,

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{900 \text{ K}} = 66.7\%$$

$$W_{net,out} = \eta_{th} Q_{in} = (0.667)(0.5889 \text{ kJ}) = \mathbf{0.393 \ kJ}$$

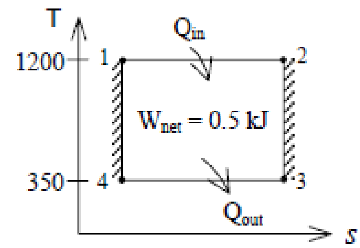
9–23 An air-standard Carnot cycle is executed in a closed system between the temperature limits of 350 and 1200 K. The pressures before and after the isothermal compression are 150 and 300 kPa, respectively. If the net work output per cycle is 0.5 kJ, determine (*a*) the maximum pressure in the cycle, (*b*) the heat transfer to air, and (*c*) the mass of air. Assume variable specific heats for air. *Answers:* (*a*) 30,013 kPa, (*b*) 0.706 kJ, (*c*) 0.00296 kg

9-23 A Carnot cycle with specified temperature limits is considered. The maximum pressure in the cycle, the heat transfer to the working fluid, and the mass of the working fluid are to be determined.

Assumptions Air is an ideal gas with variable specific heats.

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

$$T_1 = 1200\text{ K} \longrightarrow P_{r_1} = 238$$
$$T_4 = 350\text{ K} \longrightarrow P_{r_4} = 2.379 \quad \text{(Table A-17)}$$



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

(*b*) The heat input is determined from

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

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

(*c*) The mass of air is

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

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

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

OTTO CYCLE

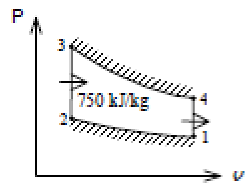
9–34 An ideal Otto cycle has a compression ratio of 8. At the beginning of the compression process, air is at 95 kPa and 27°C, and 750 kJ/kg of heat is transferred to air during the constant-volume heat-addition process. Taking into account the variation of specific heats with temperature, determine (*a*) the pressure and temperature at the end of the heat-addition process, (*b*) the net work output, (*c*) the thermal efficiency, and (*d*) the mean effective pressure for the cycle. *Answers:* (*a*) 3898 kPa, 1539 K, (*b*) 392.4 kJ/kg, (*c*) 52.3 percent, (*d*) 495 kPa

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.



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

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

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

Process 2-3: v = constant heat addition.

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

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

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

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

Process 4-1: v = constant heat rejection.

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

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

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

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

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

9–36 Repeat Problem 9–34 using constant specific heats at room temperature.

9-36 An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The pressure and temperature at the end of the heat addition process, the net work output, the thermal efficiency, and the mean effective pressure for the cycle are to be determined.

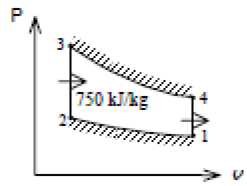
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

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

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

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

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



Process 2-3: v = constant heat addition.

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

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

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

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

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

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

Process 4-1: v = constant heat rejection.

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

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

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

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

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

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

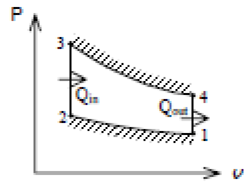
9–37 The compression ratio of an air-standard Otto cycle is 9.5. Prior to the isentropic compression process, the air is at 100 kPa, 35°C, and 600 cm³. The temperature at the end of the isentropic expansion process is 800 K. Using specific heat values at room temperature, determine (*a*) the highest temperature and pressure in the cycle; (*b*) the amount of heat transferred in, in kJ; (*c*) the thermal efficiency; and (*d*) the mean effective pressure. *Answers:* (*a*) 1969 K, 6072 kPa, (*b*) 0.59 kJ, (c) 59.4 percent, (*d*) 652 kPa

9-37 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.

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

Process 2-3: v = constant heat addition.

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

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

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

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

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

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

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

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

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

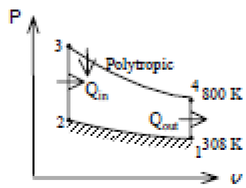
9–38 Repeat Problem 9–37, but replace the isentropic expansion process by a polytropic expansion process with the polytropic exponent $n = 1.35$.

9-38 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*) $$V_{min} = V_2 = \frac{V_{max}}{r}$$

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

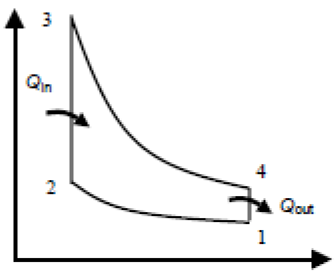
9–41 A four-cylinder, four-stroke, 2.2-L gasoline engine operates on the Otto cycle with a compression ratio of 10. The air is at 100 kPa and 60°C at the beginning of the compression process, and the maximum pressure in the cycle is 8 MPa. The compression and expansion processes may be modeled as polytropic with a polytropic constant of 1.3. Using constant specific heats at 850 K, determine (*a*) the temperature at the end of the expansion process, (*b*) the net work output and the thermal efficiency, (*c*) the mean effective pressure, (*d*) the engine speed for a net power output of 70 kW, and (*e*) the specific fuel consumption, in g/kWh, defined as the ratio of the mass of the fuel consumed to the net work produced. The air–fuel ratio, defined as the amount of air divided by the amount of fuel intake, is 16.

9-41 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



$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = (333\text{ K})(10)^{1.3-1} = 664.4\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{n} = (100\text{ kPa})(10)^{1.3} = 1995\text{ kPa}$$

Process 2-3: constant volume heat addition

$$T_3 = T_2\left(\frac{P_3}{P_2}\right) = (664.4\text{ K})\left(\frac{8000\text{ kPa}}{1995\text{ kPa}}\right) = 2664\text{ K}$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$$
$$= (0.823\text{ kJ/kg}\cdot\text{K})(2664 - 664.4)\text{K} = 1646\text{ kJ/kg}$$

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = (2664\text{ K})\left(\frac{1}{10}\right)^{1.3-1} = \mathbf{1335\ K}$$

$$P_4 = P_3\left(\frac{v_2}{v_1}\right)^{n} = (8000\text{ kPa})\left(\frac{1}{10}\right)^{1.3} = 400.9\text{ kPa}$$

Process 4-1: constant voume heat rejection.

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) = (0.823\text{ kJ/kg}\cdot\text{K})(1335 - 333)\text{K} = 824.8\text{ kJ/kg}$$

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

$$w_{net,out} = q_{in} - q_{out} = 1646 - 824.8 = \mathbf{820.9\ kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{820.9\text{ kJ/kg}}{1646\text{ kJ/kg}} = \mathbf{0.499}$$

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(333\text{ K})}{100\text{ kPa}} = 0.9557\text{ m}^3/\text{kg} = v_{max}$$

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

$$\text{MEP} = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1 - 1/r)} = \frac{820.9\text{ kJ/kg}}{(0.9557\text{ m}^3/\text{kg})(1 - 1/10)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{954.3\ kPa}$$

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

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 10 = \frac{V_c + 0.0022\ \text{m}^3}{V_c} \longrightarrow V_c = 0.0002444\ \text{m}^3$$

$$V_1 = V_c + V_d = 0.0002444 + 0.0022 = 0.002444\ \text{m}^3$$

The total mass contained in the cylinder is

$$m_t = \frac{P_1 V_1}{RT_1} = \frac{(100\ \text{kPa})(0.002444\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(333\ \text{K})} = 0.002558\ \text{kg}$$

The engine speed for a net power output of 70 kW is

$$\dot{n} = 2\frac{\dot{W}_{\text{net}}}{m_t w_{\text{net}}} = (2\ \text{rev/cycle})\frac{70\ \text{kJ/s}}{(0.002558\ \text{kg})(820.9\ \text{kJ/kg}\cdot\text{cycle})}\left(\frac{60\ \text{s}}{1\ \text{min}}\right) = \mathbf{4001\ rev/min}$$

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

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

$$\text{AF} = \frac{m_a}{m_f} = \frac{m_t - m_f}{m_f} \longrightarrow 16 = \frac{(0.002558\ \text{kg}) - m_f}{m_f} \longrightarrow m_f = 0.0001505\ \text{kg}$$

Finally, the specific fuel consumption is

$$\text{sfc} = \frac{m_f}{m_t w_{\text{net}}} = \frac{0.0001505\ \text{kg}}{(0.002558\ \text{kg})(820.9\ \text{kJ/kg})}\left(\frac{1000\ \text{g}}{1\ \text{kg}}\right)\left(\frac{3600\ \text{kJ}}{1\ \text{kWh}}\right) = \mathbf{258.0\ g/kWh}$$

DIESEL CYCLE

9–42C How does a diesel engine differ from a gasoline engine?

9–43C How does the ideal Diesel cycle differ from the ideal Otto cycle?

9–44C For a specified compression ratio, is a diesel or gasoline engine more efficient?

9–45C Do diesel or gasoline engines operate at higher compression ratios? Why?

9–46C What is the cutoff ratio? How does it affect the thermal efficiency of a Diesel cycle?

9-42C 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.

9-43C 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.

9-44C The gasoline engine.

9-45C Diesel engines operate at high compression ratios because the diesel engines do not have the engine knock problem.

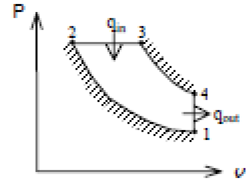
9-46C Cutoff ratio is the ratio of the cylinder volumes after and before the combustion process. As the cutoff ratio decreases, the efficiency of the diesel cycle increases.

9–47 An air-standard Diesel cycle has a compression ratio of 16 and a cutoff ratio of 2. At the beginning of the compression process, air is at 95 kPa and 27°C. Accounting for the variation of specific heats with temperature, determine (*a*) the temperature after the heat-addition process, (*b*) the thermal efficiency, and (*c*) the mean effective pressure.
Answers: (*a*) 1724.8 K, (*b*) 56.3 percent, (*c*) 675.9 kPa

9-47 An air-standard Diesel cycle with a compression ratio of 16 and a cutoff ratio of 2 is considered. The temperature after the heat addition process, the thermal efficiency, and the mean effective pressure are to be determined.

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

Properties The 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.

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

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

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2T_2 = (2)(862.4\text{ K}) = \mathbf{1724.8\text{ K}} \longrightarrow \begin{array}{l} h_3 = 1910.6\text{ kJ/kg} \\ v_{r_3} = 4.546 \end{array}$$

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

Process 3-4: isentropic expansion.

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

Process 4-1: v = constant heat rejection.

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

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

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

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

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

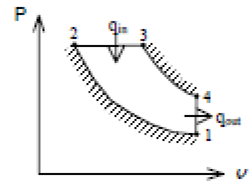
$$\text{MEP} = \frac{w_{net,out}}{v_1 - v_2} = \frac{w_{net,out}}{v_1(1-1/r)} = \frac{574.07\text{ kJ/kg}}{(0.906\text{ m}^3/\text{kg})(1-1/16)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{675.9\text{ kPa}}$$

9-48 An air-standard Diesel cycle with a compression ratio of 16 and a cutoff ratio of 2 is considered. The temperature after the heat addition process, the thermal efficiency, and the mean effective pressure are to be determined.

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

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

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



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

Process 2-3: P = constant heat addition.

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

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

Process 3-4: isentropic expansion.

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

Process 4-1: v = constant heat rejection.

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

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

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

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

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

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

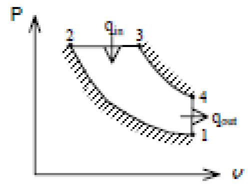
9–48 Repeat Problem 9–47 using constant specific heats at room temperature.

9-51 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*) $$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{95\text{ kPa}} = 0.885\text{ m}^3/\text{kg} = v_{max}$$

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

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

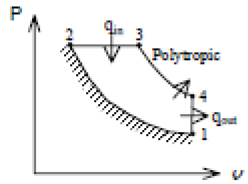
9–52 Repeat Problem 9–51, but replace the isentropic expansion process by polytropic expansion process with the polytropic exponent $n = 1.35$.

9-52 A diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency and the mean effective pressure are to be determined.

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

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

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



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

Process 2-3: P = constant heat addition.

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

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{n-1} = T_3\left(\frac{2.265V_2}{V_4}\right)^{n-1} = T_3\left(\frac{2.265}{r}\right)^{n-1} = (2200\text{ K})\left(\frac{2.265}{20}\right)^{0.35} = 1026\text{ K}$$

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

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

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

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

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

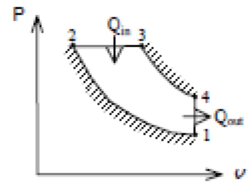
9–54 A four-cylinder two-stroke 2.4-L diesel engine that operates on an ideal Diesel cycle has a compression ratio of 17 and a cutoff ratio of 2.2. Air is at 55°C and 97 kPa at the beginning of the compression process. Using the cold-air-standard assumptions, determine how much power the engine will deliver at 1500 rpm.

9-54 A four-cylinder ideal diesel engine with air as the working fluid has a compression ratio of 17 and a cutoff ratio of 2.2. The power the engine will deliver at 1500 rpm is to be determined.

Assumptions **1** The 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.



$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (328\text{ K})(17)^{0.4} = 1019\text{ K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2}T_2 = 2.2T_2 = (2.2)(1019\text{ K}) = 2241\text{ K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{2.2V_2}{V_4}\right)^{k-1} = T_3\left(\frac{2.2}{r}\right)^{k-1} = (2241\text{ K})\left(\frac{2.2}{17}\right)^{0.4} = 989.2\text{ K}$$

$$m = \frac{P_1 V_1}{RT_1} = \frac{(97\text{ kPa})(0.0024\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(328\text{ K})} = 2.473\times10^{-3}\text{kg}$$

$$Q_{in} = m(h_3 - h_2) = mc_p(T_3 - T_2)$$
$$= (2.473\times10^{-3}\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(2241-1019)\text{K} = 3.038\text{ kJ}$$

$$Q_{out} = m(u_4 - u_1) = mc_v(T_4 - T_1)$$
$$= (2.473\times10^{-3}\text{kg})(0.718\text{ kJ/kg}\cdot\text{K})(989.2-328)\text{K} = 1.174\text{ kJ}$$

$$W_{net,out} = Q_{in} - Q_{out} = 3.038 - 1.174 = 1.864\text{ kJ/rev}$$

$$\dot{W}_{net,out} = \dot{n}W_{net,out} = (1500/60\text{ rev/s})(1.864\text{ kJ/rev}) = \mathbf{46.6\text{ kW}}$$

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

9–59 A six-cylinder, four-stroke, 4.5-L compression-ignition engine operates on the ideal diesel cycle with a compression ratio of 17. The air is at 95 kPa and 55°C at the beginning of the compression process and the engine speed is 2000 rpm. The engine uses light diesel fuel with a heating value of 42,500 kJ/kg, an air–fuel ratio of 24, and a combustion efficiency of 98 percent. Using constant specific heats at 850 K, determine (*a*) the maximum temperature in the cycle and the cutoff ratio (*b*) the net work output per cycle and the thermal efficiency, (*c*) the mean effective pressure, (*d*) the net power output, and (*e*) the specific fuel consumption, in g/kWh, defined as the ratio of the mass of the fuel consumed to the net work produced. *Answers:* (*a*) 2383 K, 2.7 (*b*) 4.36 kJ, 0.543, (*c*) 969 kPa, (*d*) 72.7 kW, (*e*) 159 g/kWh

9-59 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.

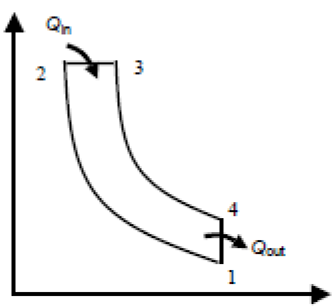
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

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

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (328\text{ K})(17)^{1.349-1} = 881.7\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = (95\text{ kPa})(17)^{1.349} = 4341\text{ kPa}$$



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

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 17 = \frac{V_c + 0.0045\text{ m}^3}{V_c}$$

$$V_c = 0.0002813\text{ m}^3$$

$$V_1 = V_c + V_d = 0.0002813 + 0.0045 = 0.004781\text{ m}^3$$

The total mass contained in the cylinder is

$$m = \frac{P_1V_1}{RT_1} = \frac{(95\text{ kPa})(0.004781\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(328\text{ K})} = 0.004825\text{ kg}$$

The mass of fuel burned during one cycle is

$$AF = \frac{m_a}{m_f} = \frac{m - m_f}{m_f} \longrightarrow 24 = \frac{(0.004825\text{ kg}) - m_f}{m_f} \longrightarrow m_f = 0.000193\text{ kg}$$

Process 2-3: constant pressure heat addition

$$Q_{in} = m_f q_{HV}\eta_c = (0.000193\text{ kg})(42{,}500\text{ kJ/kg})(0.98) = 8.039\text{ kJ}$$

$$Q_{in} = mc_v(T_3 - T_2) \longrightarrow 8.039\text{ kJ} = (0.004825\text{ kg})(0.823\text{ kJ/kg.K})(T_3 - 881.7)\text{K} \longrightarrow T_3 = \mathbf{2383\text{ K}}$$

The cutoff ratio is

$$\beta = \frac{T_3}{T_2} = \frac{2383\text{ K}}{881.7\text{ K}} = \mathbf{2.7}$$

(*b*)
$$V_2 = \frac{V_1}{r} = \frac{0.004781\text{ m}^3}{17} = 0.0002813\text{ m}^3$$

$$V_3 = \beta V_2 = (2.70)(0.0002813\text{ m}^3) = 0.00076\text{ m}^3$$

$$V_4 = V_1$$

$$P_3 = P_2$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = (2383\ \text{K})\left(\frac{0.00076\ \text{m}^3}{0.004781\ \text{m}^3}\right)^{1.349-1} = 1254\ \text{K}$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = (4341\ \text{kPa})\left(\frac{0.00076\ \text{m}^3}{0.004781\ \text{m}^3}\right)^{1.349} = 363.2\ \text{kPa}$$

Process 4-1: constant voume heat rejection.

$$Q_{out} = mc_v(T_4 - T_1) = (0.004825\ \text{kg})(0.823\ \text{kJ/kg}\cdot\text{K})(1254 - 328)\text{K} = 3.677\ \text{kJ}$$

The net work output and the thermal efficiency are

$$W_{net,out} = Q_{in} - Q_{out} = 8.039 - 3.677 = \mathbf{4.361\ kJ}$$

$$\eta_{th} = \frac{W_{net,out}}{Q_{in}} = \frac{4.361\ \text{kJ}}{8.039\ \text{kJ}} = \mathbf{0.543}$$

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

$$\text{MEP} = \frac{W_{net,out}}{V_1 - V_2} = \frac{4.361\ \text{kJ}}{(0.004781 - 0.0002813)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{969.2\ kPa}$$

(*d*) The power for engine speed of 2000 rpm is

$$\dot{W}_{net} = W_{net}\frac{\dot{n}}{2} = (4.361\ \text{kJ/cycle})\frac{2000\ \text{(rev/min)}}{(2\ \text{rev/cycle})}\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = \mathbf{72.7\ kW}$$

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

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

$$\text{sfc} = \frac{m_f}{W_{net}} = \frac{0.000193\ \text{kg}}{4.361\ \text{kJ/kg}}\left(\frac{1000\ \text{g}}{1\ \text{kg}}\right)\left(\frac{3600\ \text{kJ}}{1\ \text{kWh}}\right) = \mathbf{159.3\ \ g/kWh}$$

STIRLING AND ERICSSON CYCLES

9–60C Consider the ideal Otto, Stirling, and Carnot cycles operating between the same temperature limits. How would you compare the thermal efficiencies of these three cycles?

9–61C Consider the ideal Diesel, Ericsson, and Carnot cycles operating between the same temperature limits. How would you compare the thermal efficiencies of these three cycles?

9–62C What cycle is composed of two isothermal and two constant-volume processes?

9–63C How does the ideal Ericsson cycle differ from the Carnot cycle?

9-60C The efficiencies of the Carnot and the Stirling cycles would be the same, the efficiency of the Otto cycle would be less.

9-61C The efficiencies of the Carnot and the Ericsson cycles would be the same, the efficiency of the Diesel cycle would be less.

9-62C The Stirling cycle.

9-63C The two isentropic processes of the Carnot cycle are replaced by two constant pressure regeneration processes in the Ericsson cycle.

9–65 Consider an ideal Ericsson cycle with air as the working fluid executed in a steady-flow system. Air is at 27°C and 120 kPa at the beginning of the isothermal compression process, during which 150 kJ/kg of heat is rejected. Heat transfer to air occurs at 1200 K. Determine (*a*) the maximum pressure in the cycle, (*b*) the net work output per unit mass of air, and (*c*) the thermal efficiency of the cycle. *Answers:* (*a*) 685 kPa, (*b*) 450 kJ/kg, (*c*) 75 percent

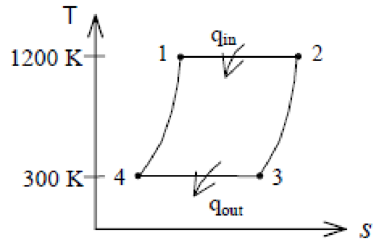
9-65 An ideal steady-flow Ericsson engine with air as the working fluid is considered. The maximum pressure in the cycle, the net work output, and the thermal efficiency of the cycle are to be determined.

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 $\qquad P_4 = \mathbf{685.2\ kPa}$

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

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

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

9–66 An ideal Stirling engine using helium as the working fluid operates between temperature limits of 300 and 2000 K and pressure limits of 150 kPa and 3 MPa. Assuming the mass of the helium used in the cycle is 0.12 kg, determine (*a*) the thermal efficiency of the cycle, (*b*) the amount of heat transfer in the regenerator, and (*c*) the work output per cycle.

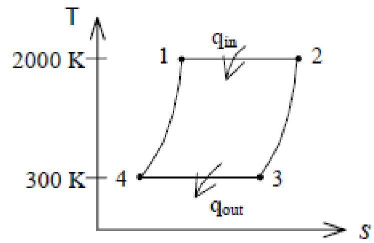
9-66 An ideal Stirling engine with helium as the working fluid operates between the specified temperature and pressure limits. The thermal efficiency of the cycle, the amount of heat transfer in the regenerator, and the work output per cycle are to be determined.

Assumptions Helium is an ideal gas with constant specific heats.

Properties The gas constant and the specific heat of helium at room temperature are $R = 2.0769$ kJ/kg.K, $c_v = 3.1156$ kJ/kg.K and $c_p = 5.1926$ kJ/kg.K (Table A-2).

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

$$\eta_{th} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{2000\text{ K}} = \mathbf{85.0\%}$$



(*b*) The amount of heat transferred in the regenerator is

$$Q_{regen} = Q_{41,in} = m(u_1 - u_4) = mc_v(T_1 - T_4)$$
$$= (0.12\text{ kg})(3.1156\text{ kJ/kg}\cdot\text{K})(2000 - 300)\text{K}$$
$$= \mathbf{635.6\ kJ}$$

(*c*) The net work output is determined from

$$\frac{P_3 \boldsymbol{v}_3}{T_3} = \frac{P_1 \boldsymbol{v}_1}{T_1} \longrightarrow \frac{\boldsymbol{v}_3}{\boldsymbol{v}_1} = \frac{T_3 P_1}{T_1 P_3} = \frac{(300\text{ K})(3000\text{ kPa})}{(2000\text{ K})(150\text{ kPa})} = 3 = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}$$

$$s_2 - s_1 = c_v \ln\frac{T_2}{T_1}^{\nearrow 0} + R\ln\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1} = (2.0769\text{ kJ/kg}\cdot\text{K})\ln(3) = 2.282\text{ kJ/kg}\cdot\text{K}$$

$$Q_{in} = mT_H(s_2 - s_1) = (0.12\text{ kg})(2000\text{ K})(2.282\text{ kJ/kg}\cdot\text{K}) = 547.6\text{ kJ}$$

$$W_{net,out} = \eta_{th} Q_{in} = (0.85)(547.6\text{ kJ}) = \mathbf{465.5\ kJ}$$

9–67C Why are the back work ratios relatively high in gas-turbine engines?

9–68C What four processes make up the simple ideal Brayton cycle?

9–69C For fixed maximum and minimum temperatures, what is the effect of the pressure ratio on (*a*) the thermal efficiency and (*b*) the net work output of a simple ideal Brayton cycle?

9–70C What is the back work ratio? What are typical back work ratio values for gas-turbine engines?

9–71C How do the inefficiencies of the turbine and the compressor affect (*a*) the back work ratio and (*b*) the thermal efficiency of a gas-turbine engine?

9-67C 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.

9-68C 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.

9-69C 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.

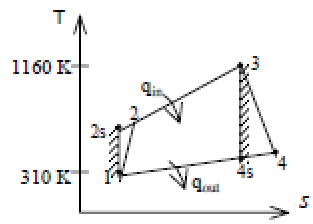
9-70C Back work ratio is the ratio of the compressor (or pump) work input to the turbine work output. It is usually between 0.40 and 0.6 for gas turbine engines.

9-71C As a result of turbine and compressor inefficiencies, (a) the back work ratio increases, and (b) the thermal efficiency decreases.

9–73 A simple Brayton cycle using air as the working fluid has a pressure ratio of 8. The minimum and maximum temperatures in the cycle are 310 and 1160 K. Assuming an isentropic efficiency of 75 percent for the compressor and 82 percent for the turbine, determine (*a*) the air temperature at the turbine exit, (*b*) the net work output, and (*c*) the thermal efficiency.

9-73 [*Also solved by EES on enclosed CD*] A simple Brayton cycle with air as the working fluid has a pressure ratio of 8. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

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



Properties The properties of air are given in Table A-17.

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

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

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

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

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

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

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

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

$$= 1230.92 - (0.82)(1230.92 - 692.19)$$

$$= 789.16 \text{ kJ/kg}$$

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

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

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

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

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

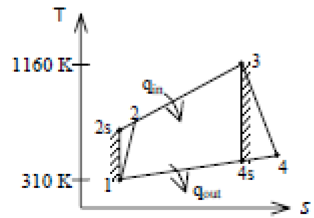
9–75 Repeat Problem 9–73 using constant specific heats at room temperature.

9-75 A simple Brayton cycle with air as the working fluid has a pressure ratio of 8. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

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

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

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



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

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

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

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

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

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

$$= \mathbf{733.9\text{ K}}$$

(*b*)

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

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

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

(*c*)

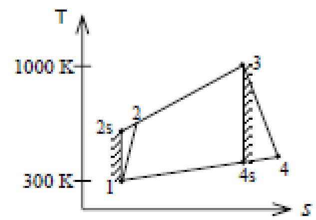
$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{91.3\text{ kJ/kg}}{517.3\text{ kJ/kg}} = \mathbf{17.6\%}$$

9–76 Air is used as the working fluid in a simple ideal Brayton cycle that has a pressure ratio of 12, a compressor inlet temperature of 300 K, and a turbine inlet temperature of 1000 K. Determine the required mass flow rate of air for a net power output of 70 MW, assuming both the compressor and the turbine have an isentropic efficiency of (*a*) 100 percent and (*b*) 85 percent. Assume constant specific heats at room temperature. *Answers:* (*a*) 352 kg/s, (*b*) 1037 kg/s

9-76 A gas turbine power plant that operates on the simple Brayton cycle with air as the working fluid has a specified pressure ratio. The required mass flow rate of air is to be determined for two cases.

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

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



Analysis (*a*) Using the isentropic relations,

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

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

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

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

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

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

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

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

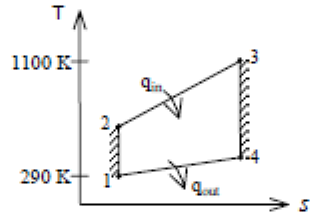
$$\dot{m}_a = \frac{\dot{W}_{net,out}}{w_{a,net,out}} = \frac{70{,}000\text{ kJ/s}}{67.5\text{ kJ/kg}} = \mathbf{1037\ kg/s}$$

9–77 A stationary gas-turbine power plant operates on a simple ideal Brayton cycle with air as the working fluid. The air enters the compressor at 95 kPa and 290 K and the turbine at 760 kPa and 1100 K. Heat is transferred to air at a rate of 35,000 kJ/s. Determine the power delivered by this plant (*a*) assuming constant specific heats at room temperature and (*b*) accounting for the variation of specific heats with temperature.

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),

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

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

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

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

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

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

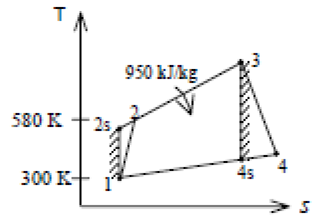
9–78 Air enters the compressor of a gas-turbine engine at 300 K and 100 kPa, where it is compressed to 700 kPa and 580 K. Heat is transferred to air in the amount of 950 kJ/kg before it enters the turbine. For a turbine efficiency of 86 percent, determine (*a*) the fraction of the turbine work output used to drive the compressor and (*b*) the thermal efficiency. Assume variable specific heats for air.

9-78 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,



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

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

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

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

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

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

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

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

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

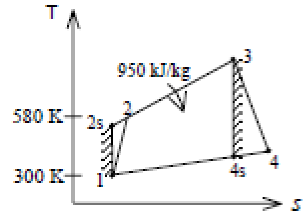
$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{256.15\text{ kJ/kg}}{950\text{ kJ/kg}} = \mathbf{27.0\%}$$

9–79 Repeat Problem 9–78 using constant specific heats at room temperature.

9-79 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) $$w_{net,out} = w_{T,out} - w_{C,in} = 562.2 - 281.4 = 280.8\text{ kJ/kg}$$

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

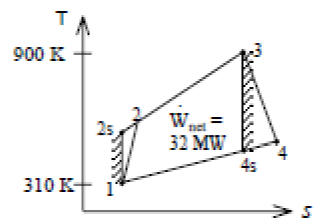
9–82 A gas-turbine power plant operates on the simple Brayton cycle with air as the working fluid and delivers 32 MW of power. The minimum and maximum temperatures in the cycle are 310 and 900 K, and the pressure of air at the compressor exit is 8 times the value at the compressor inlet. Assuming an isentropic efficiency of 80 percent for the compressor and 86 percent for the turbine, determine the mass flow rate of air through the cycle. Account for the variation of specific heats with temperature.

9-82 A 32-MW gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The mass flow rate of air through the cycle is to be determined.

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

Properties The properties of air are given in Table A-17.

Analysis Using variable specific heats,



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

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

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

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

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

and $$\dot{m} = \frac{\dot{W}_{\text{net,out}}}{w_{\text{net,out}}} = \frac{32{,}000 \text{ kJ/s}}{40.68 \text{ kJ/kg}} = \mathbf{786.6\ kg/s}$$

9–84 A gas-turbine power plant operates on the simple Brayton cycle between the pressure limits of 100 and 1200 kPa. The working fluid is air, which enters the compressor at 30°C at a rate of 150 m³/min and leaves the turbine at 500°C. Using variable specific heats for air and assuming a compressor isentropic efficiency of 82 percent and a turbine isentropic efficiency of 88 percent, determine (*a*) the net power output, (*b*) the back work ratio, and (*c*) the thermal efficiency. *Answers:* (*a*) 659 kW, (*b*) 0.625, (*c*) 0.319

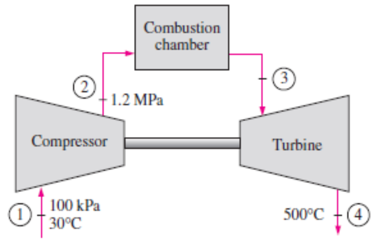


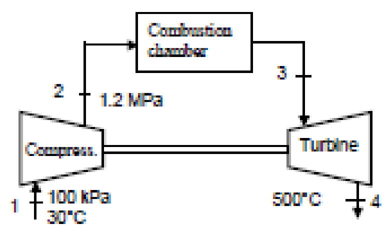
FIGURE P9–84

9-84 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

$$T_1 = 30°\text{C} \longrightarrow h_1 = 303.60 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 30°\text{C} \\ P_1 = 100 \text{ kPa} \end{array}\right\} s_1 = 5.7159 \text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 1200 \text{ kPa} \\ s_2 = s_1 = 5.7159 \text{ kJ/kg.K} \end{array}\right\} h_{2s} = 617.37 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.82 = \frac{617.37 - 303.60}{h_2 - 303.60} \longrightarrow h_2 = 686.24 \text{ kJ/kg}$$

Process 3-4: Expansion

$$T_4 = 500°\text{C} \longrightarrow h_4 = 792.62 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.88 = \frac{h_3 - 792.62}{h_3 - h_{4s}}$$

We cannot find the enthalpy at state 3 directly. However, using the following lines in EES together with the isentropic efficiency relation, we find $h_3 = 1404.7$ kJ/kg, $T_3 = 1034$°C, $s_3 = 6.5699$ 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

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(100 \text{ kPa})(150/60 \text{ m}^3/\text{s})}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(30+273 \text{ K})} = 2.875 \text{ kg/s}$$

The net power output is

$$\dot{W}_{C,in} = \dot{m}(h_2 - h_1) = (2.875 \text{ kg/s})(686.24 - 303.60)\text{kJ/kg} = 1100 \text{ kW}$$

$$\dot{W}_{T,out} = \dot{m}(h_3 - h_4) = (2.875 \text{ kg/s})(1404.7 - 792.62)\text{kJ/kg} = 1759 \text{ kW}$$

$$\dot{W}_{net} = \dot{W}_{T,out} - \dot{W}_{C,in} = 1759 - 1100 = \mathbf{659 \text{ kW}}$$

(*b*) The back work ratio is

$$r_{bw} = \frac{\dot{W}_{C,in}}{\dot{W}_{T,out}} = \frac{1100 \text{ kW}}{1759 \text{ kW}} = \mathbf{0.625}$$

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

$$\dot{Q}_{in} = \dot{m}(h_3 - h_2) = (2.875 \text{ kg/s})(1404.7 - 686.24)\text{kJ/kg} = 2065 \text{ kW}$$

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{659 \text{ kW}}{2065 \text{ kW}} = \mathbf{0.319}$$

Brayton Cycle with Regeneration

9–85C How does regeneration affect the efficiency of a Brayton cycle, and how does it accomplish it?

9–86C Somebody claims that at very high pressure ratios, the use of regeneration actually decreases the thermal efficiency of a gas-turbine engine. Is there any truth in this claim? Explain.

9–87C Define the effectiveness of a regenerator used in gas-turbine cycles.

9–88C In an ideal regenerator, is the air leaving the compressor heated to the temperature at (*a*) turbine inlet, (*b*) turbine exit, (*c*) slightly above turbine exit?

9–89C In 1903, Aegidius Elling of Norway designed and built an 11-hp gas turbine that used steam injection between the combustion chamber and the turbine to cool the combustion gases to a safe temperature for the materials available at the time. Currently there are several gas-turbine power plants that use steam injection to augment power and improve thermal efficiency. For example, the thermal efficiency of the General Electric LM5000 gas turbine is reported to increase from 35.8 percent in simple-cycle operation to 43 percent when steam injection is used. Explain why steam injection increases the power output and the efficiency of gas turbines. Also, explain how you would obtain the steam.

9-85C 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.

9-86C 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.

9-87C The extent to which a regenerator approaches an ideal regenerator is called the effectiveness ε, and is defined as $\varepsilon = q_{regen, act} / q_{regen, max}$.

9-88C (b) turbine exit.

9-89C The steam injected increases the mass flow rate through the turbine and thus the power output. This, in turn, increases the thermal efficiency since $\eta = W / Q_{in}$ and W increases while Q_{in} remains constant. Steam can be obtained by utilizing the hot exhaust gases.

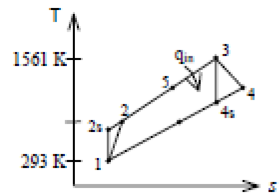
9–91 The 7FA gas turbine manufactured by General Electric is reported to have an efficiency of 35.9 percent in the simple-cycle mode and to produce 159 MW of net power. The pressure ratio is 14.7 and the turbine inlet temperature is 1288°C. The mass flow rate through the turbine is 1,536,000 kg/h. Taking the ambient conditions to be 20°C and 100 kPa, determine the isentropic efficiency of the turbine and the compressor. Also, determine the thermal efficiency of this gas turbine if a regenerator with an effectiveness of 80 percent is added.

9-91 [*Also solved by EES on enclosed CD*] The thermal efficiency and power output of an actual gas turbine are given. The isentropic efficiency of the turbine and of the compressor, and the thermal efficiency of the gas turbine modified with a regenerator are to be determined.

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

Properties The properties of air are given in Table A-17.

Analysis The properties at various states are



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

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

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

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

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

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

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

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

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

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

Then the compressor and turbine efficiencies become

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

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

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

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

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

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

9–93 An ideal Brayton cycle with regeneration has a pressure ratio of 10. Air enters the compressor at 300 K and the turbine at 1200 K. If the effectiveness of the regenerator is 100 percent, determine the net work output and the thermal efficiency of the cycle. Account for the variation of specific heats with temperature.

9-93 An ideal Brayton cycle with regeneration is considered. The effectiveness of the regenerator is 100%. The net work output and the thermal efficiency of the cycle are to be determined.

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

Properties The properties of air are given in Table A-17.

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

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

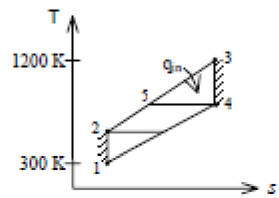
$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (10)(1.386) = 13.86 \longrightarrow h_2 = 579.87 \text{ kJ/kg}$$

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

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

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

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



Thus,

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

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

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

and

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

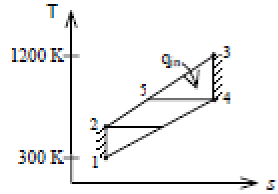
9–95 Repeat Problem 9–93 using constant specific heats at room temperature.

9-95 An ideal Brayton cycle with regeneration is considered. The effectiveness of the regenerator is 100%. The net work output and the thermal efficiency of the cycle are to be determined.

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

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

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



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

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

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

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

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

Then,

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

or,

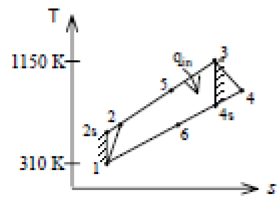
$$\begin{aligned} w_{net} &= \eta_{th} q_{in} \\ &= \eta_{th}(h_3 - h_5) \\ &= \eta_{th} c_p (T_3 - T_5) \\ &= (0.517)(1.005\text{ kJ/kg.K})(1200 - 621.5) \\ &= 300.6\text{ kJ/kg} \end{aligned}$$

9–96 A Brayton cycle with regeneration using air as the working fluid has a pressure ratio of 7. The minimum and maximum temperatures in the cycle are 310 and 1150 K. Assuming an isentropic efficiency of 75 percent for the compressor and 82 percent for the turbine and an effectiveness of 65 percent for the regenerator, determine (*a*) the air temperature at the turbine exit, (*b*) the net work output, and (*c*) the thermal efficiency. *Answers:* (*a*) 783 K, (*b*) 108.1 kJ/kg, (*c*) 22.5 percent

9-96 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*)

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

(*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,

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

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

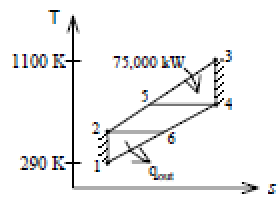
9–97 A stationary gas-turbine power plant operates on an ideal regenerative Brayton cycle (ϵ = 100 percent) with air as the working fluid. Air enters the compressor at 95 kPa and 290 K and the turbine at 760 kPa and 1100 K. Heat is transferred to air from an external source at a rate of 75,000 kJ/s. Determine the power delivered by this plant (*a*) assuming constant specific heats for air at room temperature and (*b*) accounting for the variation of specific heats with temperature.

9-97 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,



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

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

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

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

$$\dot{W}_{net} = \eta_T \dot{Q}_{in} = (0.5225)(75{,}000\text{ kW}) = \mathbf{39{,}188\ \ kW}$$

(*b*) Assuming variable specific heats,

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

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

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

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

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

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

$$\dot{W}_{net} = \eta_T \dot{Q}_{in} = (0.5371)(75{,}000\text{ kW}) = \mathbf{40{,}283\ \ kW}$$

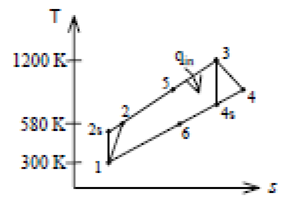
9–98 Air enters the compressor of a regenerative gas-turbine engine at 300 K and 100 kPa, where it is compressed to 800 kPa and 580 K. The regenerator has an effectiveness of 72 percent, and the air enters the turbine at 1200 K. For a turbine efficiency of 86 percent, determine (*a*) the amount of heat transfer in the regenerator and (*b*) the thermal efficiency. Assume variable specific heats for air. *Answers:* (*a*) 152.5 kJ/kg, (*b*) 36.0 percent

9-98 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



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

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

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

$$T_3 = 1200\text{ K} \longrightarrow h_3 = 1277.79\text{ kJ/kg}$$

$$P_{r_3} = 238.0$$

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

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

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

(*b*)

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

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

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

9–99 Repeat Problem 9–98 using constant specific heats at room temperature.

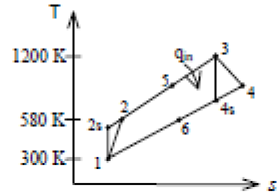
9-99 A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

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

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

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

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



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

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

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

(b)

$$w_{net} = w_{T,out} - w_{C,in} = c_p(T_3 - T_4) - c_p(T_2 - T_1) = (1.005\text{ kJ/kg}\cdot\text{K})[(1200 - 737.8) - (580 - 300)]\text{K} = 183.1\text{ kJ/kg}$$

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

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

9–100 Repeat Problem 9–98 for a regenerator effectiveness of 70 percent.

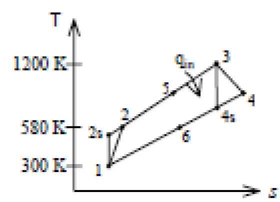
9-100 A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

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 of air at various states are

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



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

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

$$T_3 = 1200\text{K} \longrightarrow h_3 = 1277.79\text{kJ/kg}$$

$$P_{r_3} = 238.0$$

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

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

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

(*b*)

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

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

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

9–101C Under what modifications will the ideal simple gas-turbine cycle approach the Ericsson cycle?

9–102C The single-stage compression process of an ideal Brayton cycle without regeneration is replaced by a multistage compression process with intercooling between the same pressure limits. As a result of this modification,

(*a*) Does the compressor work increase, decrease, or remain the same?

(*b*) Does the back work ratio increase, decrease, or remain the same?

(*c*) Does the thermal efficiency increase, decrease, or remain the same?

9–103C The single-stage expansion process of an ideal Brayton cycle without regeneration is replaced by a multistage expansion process with reheating between the same pressure limits. As a result of this modification,

(*a*) Does the turbine work increase, decrease, or remain the same?

(*b*) Does the back work ratio increase, decrease, or remain the same?

(*c*) Does the thermal efficiency increase, decrease, or remain the same?

9–104C A simple ideal Brayton cycle without regeneration is modified to incorporate multistage compression with intercooling and multistage expansion with reheating, without changing the pressure or temperature limits of the cycle. As a result of these two modifications,

(*a*) Does the net work output increase, decrease, or remain the same?

(*b*) Does the back work ratio increase, decrease, or remain the same?

(*c*) Does the thermal efficiency increase, decrease, or remain the same?

(*d*) Does the heat rejected increase, decrease, or remain the same?

9–105C A simple ideal Brayton cycle is modified to incorporate multistage compression with intercooling, multistage expansion with reheating, and regeneration without changing the pressure limits of the cycle. As a result of these modifications,

(*a*) Does the net work output increase, decrease, or remain the same?

(*b*) Does the back work ratio increase, decrease, or remain the same?

(*c*) Does the thermal efficiency increase, decrease, or remain the same?

(*d*) Does the heat rejected increase, decrease, or remain the same?

9–106C For a specified pressure ratio, why does multistage compression with intercooling decrease the compressor work, and multistage expansion with reheating increase the turbine work?

9–107C In an ideal gas-turbine cycle with intercooling, reheating, and regeneration, as the number of compression and expansion stages is increased, the cycle thermal efficiency approaches (*a*) 100 percent, (*b*) the Otto cycle efficiency, or (*c*) the Carnot cycle efficiency.

9-101C As the number of compression and expansion stages are increased and regeneration is employed, the ideal Brayton cycle will approach the Ericsson cycle.

9-102C (a) decrease, (b) decrease, and (c) decrease.

9-103C (a) increase, (b) decrease, and (c) decrease.

9-104C (a) increase, (b) decrease, (c) decrease, and (d) increase.

9-105C (a) increase, (b) decrease, (c) increase, and (d) decrease.

9-106C 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.

9-107C (c) The Carnot (or Ericsson) cycle efficiency.

9–108 Consider an ideal gas-turbine cycle with two stages o compression and two stages of expansion. The pressure rati across each stage of the compressor and turbine is 3. The ai enters each stage of the compressor at 300 K and each stage o the turbine at 1200 K. Determine the back work ratio and th thermal efficiency of the cycle, assuming (*a*) no regenerator i used and (*b*) a regenerator with 75 percent effectiveness i used. Use variable specific heats.

9-108 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.

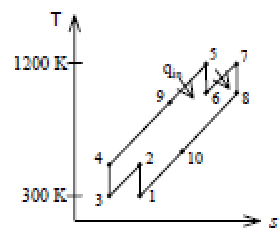
Analysis (*a*) The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine since this is an ideal cycle. Then,

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

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_2 = h_4 = 411.26\text{ kJ/kg}$$

$$T_5 = 1200\text{ K} \longrightarrow \begin{array}{l} h_5 = h_7 = 1277.79\text{ kJ/kg} \\ P_{r_5} = 238 \end{array}$$

$$P_{r_6} = \frac{P_6}{P_5} P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_6 = h_8 = 946.36\text{ kJ/kg}$$



$$w_{C,in} = 2(h_2 - h_1) = 2(411.26 - 300.19) = 222.14\text{ kJ/kg}$$

$$w_{T,out} = 2(h_5 - h_6) = 2(1277.79 - 946.36) = 662.86\text{ kJ/kg}$$

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

$$q_{regen} = \varepsilon(h_8 - h_4) = (0.75)(946.36 - 411.26) = 401.33\text{ kJ/kg}$$

$$q_{in} = q_{in,old} - q_{regen} = 1197.96 - 401.33 = 796.63\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{440.72\text{ kJ/kg}}{796.63\text{ kJ/kg}} = \mathbf{55.3\%}$$

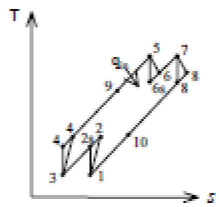
9–109 Repeat Problem 9–108, assuming an efficiency of 80 percent for each compressor stage and an efficiency of 85 percent for each turbine stage.

9-109 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,



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

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_{2s} = h_{4s} = 411.26\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_4 = h_1 + (h_{2s} - h_1)/\eta_C$$
$$= 300.19 + (411.26 - 300.19)/(0.80)$$
$$= 439.03\text{ kJ/kg}$$

$$T_5 = 1200\text{ K} \longrightarrow h_5 = h_7 = 1277.79\text{ kJ/kg}$$
$$P_{r_5} = 238$$

$$P_{r_6} = \frac{P_6}{P_5} P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_{6s} = h_{8s} = 946.36\text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_8 = h_5 - \eta_T (h_5 - h_{6s})$$
$$= 1277.79 - (0.85)(1277.79 - 946.36)$$
$$= 996.07\text{ kJ/kg}$$

$$w_{C,\text{in}} = 2(h_2 - h_1) = 2(439.03 - 300.19) = 277.68\text{ kJ/kg}$$
$$w_{T,\text{out}} = 2(h_5 - h_6) = 2(1277.79 - 996.07) = 563.44\text{ kJ/kg}$$

Thus,
$$r_{bw} = \frac{w_{C,\text{in}}}{w_{T,\text{out}}} = \frac{277.68\text{ kJ/kg}}{563.44\text{ kJ/kg}} = \mathbf{49.3\%}$$

$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6) = (1277.79 - 439.03) + (1277.79 - 996.07) = 1120.48\text{ kJ/kg}$$

$$w_{\text{net}} = w_{T,\text{out}} - w_{C,\text{in}} = 563.44 - 277.68 = 285.76\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{285.76\text{ kJ/kg}}{1120.48\text{ kJ/kg}} = \mathbf{25.5\%}$$

(*b*) When a regenerator is used, r_{bw} remains the same. The thermal efficiency in this case becomes

$$q_{\text{regen}} = \varepsilon(h_8 - h_4) = (0.75)(996.07 - 439.03) = 417.78\text{ kJ/kg}$$
$$q_{\text{in}} = q_{\text{in,old}} - q_{\text{regen}} = 1120.48 - 417.78 = 702.70\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{285.76\text{ kJ/kg}}{702.70\text{ kJ/kg}} = \mathbf{40.7\%}$$

9–110 Consider a regenerative gas-turbine power plant with two stages of compression and two stages of expansion. The overall pressure ratio of the cycle is 9. The air enters each stage of the compressor at 300 K and each stage of the turbine at 1200 K. Accounting for the variation of specific heats with temperature, determine the minimum mass flow rate of air needed to develop a net power output of 110 MW.
Answer: 250 kg/s

9-110 A regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The minimum mass flow rate of air needed to develop a specified net power output is 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 The mass flow rate will be a minimum when the cycle is ideal. That is, the turbine and the compressors are isentropic, the regenerator has an effectiveness of 100%, and the compression ratios across each compression or expansion stage are identical. In our case it is $r_p = \sqrt{9} = 3$. Then the work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.

$$T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}, \quad P_{r_1} = 1.386$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_2 = h_4 = 411.26 \text{ kJ/kg}$$

$$T_5 = 1200 \text{ K} \longrightarrow h_5 = h_7 = 1277.79 \text{ kJ/kg}, \quad P_{r_5} = 238$$

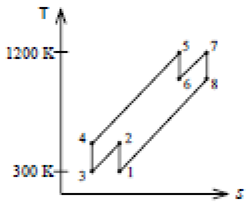
$$P_{r_6} = \frac{P_6}{P_5} P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_6 = h_8 = 946.36 \text{ kJ/kg}$$

$$w_{C,\text{in}} = 2(h_2 - h_1) = 2(411.26 - 300.19) = 222.14 \text{ kJ/kg}$$

$$w_{T,\text{out}} = 2(h_5 - h_6) = 2(1277.79 - 946.36) = 662.86 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{T,\text{out}} - w_{C,\text{in}} = 662.86 - 222.14 = 440.72 \text{ kJ/kg}$$

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{110{,}000 \text{ kJ/s}}{440.72 \text{ kJ/kg}} = \mathbf{249.6 \text{ kg/s}}$$



Jet Propulsion Cycles

9–112C What is propulsive power? How is it related to thrust?

9–113C What is propulsive efficiency? How is it determined?

9–114C Is the effect of turbine and compressor irreversibilities of a turbojet engine to reduce (*a*) the net work, (*b*) the thrust, or (*c*) the fuel consumption rate?

9-112C The power developed from the thrust of the engine is called the propulsive power. It is equal to thrust times the aircraft velocity.

9-113C 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.

9-114C It reduces the exit velocity, and thus the thrust.

9–117 A turbojet aircraft is flying with a velocity of 320 m/s at an altitude of 9150 m, where the ambient conditions are 32 kPa and −32°C. The pressure ratio across the compressor is 12, and the temperature at the turbine inlet is 1400 K. Air enters the compressor at a rate of 60 kg/s, and the jet fuel has a heating value of 42,700 kJ/kg. Assuming ideal operation for all components and constant specific heats for air at room temperature, determine (*a*) the velocity of the exhaust gases, (*b*) the propulsive power developed, and (*c*) the rate of fuel consumption.

9-117 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.

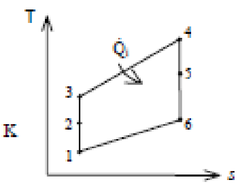
Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** Kinetic and potential energies are negligible except at the diffuser inlet and the nozzle exit. **5** The turbine work 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:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}} \longrightarrow \dot{E}_{in} = \dot{E}_{out}$$



$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^{2\nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 241\text{ K} + \frac{(320\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 291.9\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (32\text{ kPa})\left(\frac{291.9\text{ K}}{241\text{ K}}\right)^{1.4/0.4} = 62.6\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (12)(62.6\text{ kPa}) = 751.2\text{ kPa}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (291.9\text{ K})(12)^{0.4/1.4} = 593.7\text{ K}$$

Turbine:

$$w_{comp,in} = w_{turb,out} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or,

$$T_5 = T_4 - T_3 + T_2 = 1400 - 593.7 + 291.9 = 1098.2\text{K}$$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{32\text{ kPa}}{751.2\text{ kPa}}\right)^{0.4/1.4} = 568.2\text{ K}$$

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}} \longrightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\nearrow 0}}{2} \longrightarrow 0 = c_p(T_6 - T_5) + V_6^2/2$$

or,

$$V_6 = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(1098.2 - 568.2)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{1032\text{ m/s}}$$

(*b*)

$$\dot{W}_p = \dot{m}(V_{exit} - V_{inlet})V_{aircraft} = (60\text{ kg/s})(1032 - 320)\text{m/s}(320\text{ m/s})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{13{,}670\text{ kW}}$$

(*c*)

$$\dot{Q}_{in} = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3) = (60\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400 - 593.7)\text{K} = 48{,}620\text{ kJ/s}$$

$$\dot{m}_{fuel} = \frac{\dot{Q}_{in}}{\text{HV}} = \frac{48{,}620\text{ kJ/s}}{42{,}700\text{ kJ/kg}} = \mathbf{1.14\text{ kg/s}}$$

9–118 Repeat Problem 9–117 using a compressor efficiency of 80 percent and a turbine efficiency of 85 percent.

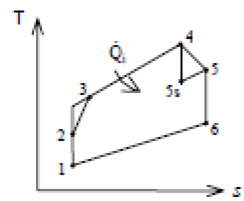
9-118 A turbojet aircraft is flying at an altitude of 9150 m. The velocity of exhaust gases, the propulsive power developed, and the rate of fuel consumption are to be determined.

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:

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (1022.7\text{ K})\left(\frac{32\text{ kPa}}{197.7\text{ kPa}}\right)^{0.4/1.4} = 607.8\text{ K}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\not{0}\text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\,\not{0}}}{2}$$

$$0 = c_p(T_6 - T_5) + V_6^2/2$$

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*)
$$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3) = (60\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400-669.2)\text{K} = 44{,}067\text{ kJ/s}$$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}_{\text{in}}}{\text{HV}} = \frac{44{,}067\text{ kJ/s}}{42{,}700\text{ kJ/kg}} = \mathbf{1.03\ kg/s}$$

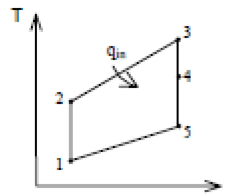
9–119 Consider an aircraft powered by a turbojet engine that has a pressure ratio of 12. The aircraft is stationary on the ground, held in position by its brakes. The ambient air is at 27°C and 95 kPa and enters the engine at a rate of 10 kg/s. The jet fuel has a heating value of 42,700 kJ/kg, and it is burned completely at a rate of 0.2 kg/s. Neglecting the effect of the diffuser and disregarding the slight increase in mass at the engine exit as well as the inefficiencies of engine components, determine the force that must be applied on the brakes to hold the plane stationary. *Answer:* 9089 N

9-119 A turbojet aircraft that has a pressure rate of 12 is stationary on the ground. The force that must b 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 a ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the nozzl exit.

Properties The properties of air are given in Table A17.

Analysis (*a*) Using variable specific heats for air,



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

$$P_{r_1} = 1.386$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (12)(1.386) = 16.63 \longrightarrow h_2 = 610.65 \text{ kJ/kg}$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{fuel}} \times \text{HV} = (0.2 \text{ kg/s})(42{,}700 \text{ kJ/kg}) = 8540 \text{ kJ/s}$$

$$q_{\text{in}} = \frac{\dot{Q}_{\text{in}}}{\dot{m}} = \frac{8540 \text{ kJ/s}}{10 \text{ kg/s}} = 854 \text{ kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 \longrightarrow h_3 = h_2 + q_{in} = 610.65 + 854 = 1464.65 \text{ kJ/kg}$$

$$\longrightarrow P_{r_3} = 396.27$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_2 - h_1 = h_3 - h_4$$

or,

$$h_4 = h_3 - h_2 + h_1 = 1464.65 - 610.65 + 300.19 = 741.17 \text{ kJ/kg}$$

Nozzle:

$$P_{r_5} = P_{r_3}\left(\frac{P_5}{P_3}\right) = (396.27)\left(\frac{1}{12}\right) = 33.02 \longrightarrow h_5 = 741.79 \text{ kJ/kg}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0 \text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_4 + V_4^2/2 = h_5 + V_5^2/2$$

$$0 = h_5 - h_4 + \frac{V_5^2 - V_4^{2\;\nearrow 0}}{2}$$

or,

$$V_5 = \sqrt{2(h_4 - h_5)} = \sqrt{(2)(1154.19 - 741.17)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 908.9 \text{ m/s}$$

$$\text{Brake force} = \text{Thrust} = \dot{m}(V_{\text{exit}} - V_{\text{inlet}}) = (10 \text{ kg/s})(908.9 - 0)\text{m/s}\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{9089 \ N}$$

9–121 Air at 7°C enters a turbojet engine at a rate of 16 kg/s and at a velocity of 300 m/s (relative to the engine).

Air is heated in the combustion chamber at a rate 15,000 kJ/s and it leaves the engine at 427°C. Determine the thrust produced by this turbojet engine. (*Hint:* Choose the entire engine as your control volume.)

9-121 Air enters a turbojet engine. The thrust produced by this turbojet engine is to be determined.

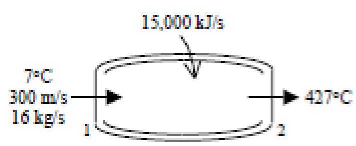
Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit.

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

$$T_1 = 280 \text{ K} \longrightarrow h_1 = 280.13 \text{ kJ/kg}$$

$$T_2 = 700 \text{ K} \longrightarrow h_2 = 713.27 \text{ kJ/kg}$$



$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$\dot{Q}_{in} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

$$15{,}000 \text{ kJ/s} = (16 \text{ kg/s})\left[713.27 - 280.13 + \frac{V_2^2 - (300 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right]$$

It gives V_2 = 1048 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

9–122 Determine the total exergy destruction associated with the Otto cycle described in Problem 9–34, assuming a source temperature of 2000 K and a sink temperature of 300 K. Also, determine the exergy at the end of the power stroke. *Answers: 245.12 kJ/kg, 145.2 kJ/kg*

9-122 The total exergy destruction associated with the Otto cycle described in Prob. 9-34 and the exergy at the end of the power stroke are to be determined.

Analysis From Prob. 9-34, $q_{in} = 750$, $q_{out} = 357.62$ kJ/kg, $T_1 = 300$ K, and $T_4 = 774.5$ K.

The total exergy destruction associated with this Otto cycle is determined from

$$x_{destroyed} = T_0\left(\frac{q_{out}}{T_L} - \frac{q_{in}}{T_H}\right) = (300\text{ K})\left(\frac{357.62\text{ kJ/kg}}{300\text{ K}} - \frac{750\text{ kJ/kg}}{2000\text{ K}}\right) = \mathbf{245.12\ kJ/kg}$$

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

$$\phi_4 = (u_4 - u_0) - T_0(s_4 - s_0) + P_0(v_4 - v_0)$$

where

$$u_4 - u_0 = u_4 - u_1 = q_{out} = 357.62\text{kJ/kg}$$

$$v_4 - v_0 = v_4 - v_1 = 0$$

$$s_4 - s_0 = s_4 - s_1 = s_4^\circ - s_1^\circ - R\ln\frac{P_4}{P_1} = s_4^\circ - s_1^\circ - R\ln\frac{T_4 v_1}{T_1 v_4} = s_4^\circ - s_1^\circ - R\ln\frac{T_4}{T_1}$$

$$= 2.6823 - 1.70203 - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{774.5\text{ K}}{300\text{ K}} = 0.7081\text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\phi_4 = (357.62\text{ kJ/kg}) - (300\text{ K})(0.7081\text{ kJ/kg}\cdot\text{K}) + 0 = \mathbf{145.2\ kJ/kg}$$

9–123 Determine the total exergy destruction associated with the Diesel cycle described in Problem 9–47, assuming a source temperature of 2000 K and a sink temperature of 300 K. Also, determine the exergy at the end of the isentropic compression process. *Answers: 292.7 kJ/kg, 348.6 kJ/kg*

9-123 The total exergy destruction associated with the Diesel cycle described in Prob. 9-47 and the exergy at the end of the compression stroke are to be determined.

Analysis From Prob. 9-47, q_{in} = 1019.7, q_{out} = 445.63 kJ/kg, T_1 = 300 K, v_1 = 0.906 m^3/kg, and $v_2 = v_1 / r$ = 0.906 / 12 = 0.0566 m^3/kg.

The total exergy destruction associated with this Otto cycle is determined from

$$x_{destroyed} = T_0\left(\frac{q_{out}}{T_L} - \frac{q_{in}}{T_H}\right) = (300\text{ K})\left(\frac{445.63\text{ kJ/kg}}{300\text{ K}} - \frac{1019.7\text{ kJ/kg}}{2000\text{ K}}\right) = \mathbf{292.7\ kJ/kg}$$

Noting that state 1 is identical to the state of the surroundings, the exergy at the end of the compression stroke (state 2) is determined from

$$\begin{aligned}\phi_2 &= (u_2 - u_0) - T_0(s_2 - s_0) + P_0(v_2 - v_0)\\ &= (u_2 - u_1) - T_0(s_2 - s_1) + P_0(v_2 - v_1)\\ &= (643.3 - 214.07) - 0 + (95\text{ kPa})(0.0566 - 0.906)\text{m}^3\text{/kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)\\ &= \mathbf{348.6\ kJ/kg}\end{aligned}$$

9–125 Calculate the exergy destruction associated with each of the processes of the Brayton cycle described in Problem 9–73, assuming a source temperature of 1600 K and a sink temperature of 290 K.

9-125 The exergy destruction associated with each of the processes of the Brayton cycle described in Prob. 9-73 is to be determined.

Analysis From Prob. 9-73, q_{in} = 584.62 kJ/kg, q_{out} = 478.92 kJ/kg, and

$$T_1 = 310\text{K} \longrightarrow s_1^\circ = 1.73498\text{kJ/kg}\cdot\text{K}$$
$$h_2 = 646.3\text{kJ/kg} \longrightarrow s_2^\circ = 2.47256\text{kJ/kg}\cdot\text{K}$$
$$T_3 = 1160\text{K} \longrightarrow s_3^\circ = 3.13916\text{kJ/kg}\cdot\text{K}$$
$$h_4 = 789.16\text{kJ/kg} \longrightarrow s_4^\circ = 2.67602\text{kJ/kg}\cdot\text{K}$$

Thus,

$$\begin{aligned}x_{destroyed,12} &= T_0 s_{gen,12} = T_0(s_2 - s_1) = T_0\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right) =\\ &= (290\text{ K})(2.47256 - 1.73498 - (0.287\text{ kJ/kg}\cdot\text{K})\ln(8)) = \mathbf{40.83\ kJ/kg}\end{aligned}$$

$$\begin{aligned}x_{destroyed,23} &= T_0 s_{gen,23} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = T_0\left(s_3^\circ - s_2^\circ - R\ln\frac{P_3}{P_2}^{\nearrow 0} + \frac{-q_{in}}{T_H}\right)\\ &= (290\text{ K})\left(3.13916 - 2.47256 - \frac{584.62\text{ kJ/kg}}{1600\text{ K}}\right) = \mathbf{87.35\ kJ/kg}\end{aligned}$$

$$\begin{aligned}x_{destroyed,34} &= T_0 s_{gen,34} = T_0(s_4 - s_3) = T_0\left(s_4^\circ - s_3^\circ - R\ln\frac{P_4}{P_3}\right) =\\ &= (290\text{ K})(2.67602 - 3.13916 - (0.287\text{ kJ/kg}\cdot\text{K})\ln(1/8)) = \mathbf{38.76\ kJ/kg}\end{aligned}$$

$$\begin{aligned}x_{destroyed,41} &= T_0 s_{gen,41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = T_0\left(s_1^\circ - s_4^\circ - R\ln\frac{P_1}{P_4}^{\nearrow 0} + \frac{q_{out}}{T_L}\right)\\ &= (290\text{ K})\left(1.73498 - 2.67602 + \frac{478.92\text{ kJ/kg}}{310\text{ K}}\right) = \mathbf{206.0\ kJ/kg}\end{aligned}$$

9–126 Determine the total exergy destruction associated with the Brayton cycle described in Problem 9–93, assuming a source temperature of 1800 K and a sink temperature of 300 K. Also, determine the exergy of the exhaust gases at the exit of the regenerator.

9-126 The total exergy destruction associated with the Brayton cycle described in Prob. 9-93 and the exergy at the exhaust gases at the turbine exit are to be determined.

Analysis From Prob. 9-93, $q_{in} = 601.94$, $q_{out} = 279.68$ kJ/kg, and $h_6 = 579.87$ kJ/kg.

The total exergy destruction associated with this Otto cycle is determined from

$$x_{destroyed} = T_0\left(\frac{q_{out}}{T_L} - \frac{q_{in}}{T_H}\right) = (300\text{ K})\left(\frac{279.68\text{ kJ/kg}}{300\text{ K}} - \frac{601.94\text{ kJ/kg}}{1800\text{ K}}\right) = \mathbf{179.4\ kJ/kg}$$

Noting that $h_0 = h_{@\,300\text{ K}} = 300.19$ kJ/kg, 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 $s_6 - s_0 = s_6 - s_1 = s_6^\circ - s_1^\circ - R\ln\frac{P_6}{P_1}^{\nearrow 0} = 2.36275 - 1.70203 = 0.66072\text{ kJ/kg}\cdot\text{K}$

Thus, $\phi_6 = 579.87 - 300.19 - (300\text{ K})(0.66072\text{ kJ/kg}\cdot\text{K}) = \mathbf{81.5\ kJ/kg}$