Solucionario Guía 6

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1. Cengel, 10-19

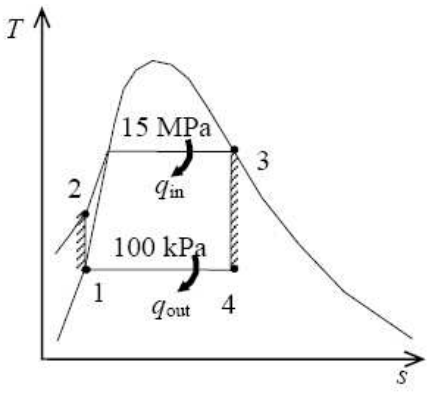
10-19 A simple ideal Rankine cycle with water as the working fluid operates between the specified pressure limits. The power produced by the turbine, the heat added in the boiler, and the thermal efficiency of the cycle are to be determined.

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

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

$$h_1 = h_{f@100\,\text{kPa}} = 417.51\,\text{kJ/kg}$$
$$v_1 = v_{f@100\,\text{kPa}} = 0.001043\,\text{m}^3/\text{kg}$$

$$w_{\text{p,in}} = v_1(P_2 - P_1)$$
$$= (0.001043\,\text{m}^3/\text{kg})(15{,}000-100)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 15.54\,\text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 417.51+15.54 = 433.05\,\text{kJ/kg}$$



$$\left.\begin{array}{l} P_3 = 15{,}000\,\text{kPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2610.8\,\text{kJ/kg} \\ s_3 = 5.3108\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 100\,\text{kPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{5.3108-1.3028}{6.0562} = 0.6618 \\ h_4 = h_f + x_4 h_{fg} = 417.51+(0.6618)(2257.5) = 1911.5\,\text{kJ/kg} \end{array}$$

Thus,

$$w_{\text{T,out}} = h_3 - h_4 = 2610.8-1911.5 = \mathbf{699.3\,kJ/kg}$$
$$q_{\text{in}} = h_3 - h_2 = 2610.8-433.05 = \mathbf{2177.8\,kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 1911.5-417.51 = 1494.0\,\text{kJ/kg}$$

The thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1494.0}{2177.8} = \mathbf{0.314}$$

2. Cengel 5th, P10-22

10-22 A steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The thermal efficiency of the cycle, the mass flow rate of the steam, and the temperature rise of the cooling water are to be determined.

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

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

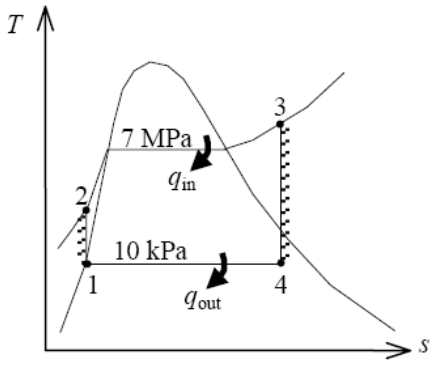
$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$

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

$$w_{p,\text{in}} = v_1(P_2 - P_1)$$

$$= (0.00101\ \text{m}^3/\text{kg})(7{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 7.06\text{ kJ/kg}$$



$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 7.06 = 198.87\ \text{kJ/kg}$$

$$\left.\begin{matrix} P_3 = 7\text{ MPa} \\ T_3 = 500°\text{C} \end{matrix}\right\} \begin{matrix} h_3 = 3411.4\text{ kJ/kg} \\ s_3 = 6.8000\text{ kJ/kg}\cdot\text{K} \end{matrix}$$

$$\left.\begin{matrix} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{matrix}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201$$

$$h_4 = h_f + x_4 h_{fg} = 191.81 + (0.8201)(2392.1) = 2153.6\text{ kJ/kg}$$

Thus,
$$q_{\text{in}} = h_3 - h_2 = 3411.4 - 198.87 = 3212.5\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2153.6 - 191.81 = 1961.8\text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3212.5 - 1961.8 = 1250.7\text{ kJ/kg}$$

and
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1250.7\text{ kJ/kg}}{3212.5\text{ kJ/kg}} = \mathbf{38.9\%}$$

(*b*)
$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000\text{ kJ/s}}{1250.7\text{ kJ/kg}} = \mathbf{36.0\ kg/s}$$

(*c*) The rate of heat rejection to the cooling water and its temperature rise are

$$\dot{Q}_{\text{out}} = \dot{m} q_{\text{out}} = (35.98\text{ kg/s})(1961.8\text{ kJ/kg}) = 70{,}586\ \text{kJ/s}$$

$$\Delta T_{\text{coolingwater}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}c)_{\text{coolingwater}}} = \frac{70{,}586\text{ kJ/s}}{(2000\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})} = \mathbf{8.4°C}$$

3. Schaum's 2nd, p9.7

9.7 For the regenerative cycle shown in Fig. 9-20 determine the thermal efficiency, the mass flux of steam, and the ratio of rejected heat to added heat. Neglect pump work.

Referring to Fig. 9-7*b* to identify the states and using the steam tables, we find

$$h_2 \cong h_1 = 191.8 \text{ kJ/kg} \qquad h_6 \cong h_7 = 762.8 \text{ kJ/kg} \qquad h_3 = 3625.3 \text{ kJ/kg}$$

The enthalpies of states 4 and 5 are determined by assuming an isentropic process as follows:

$$\left.\begin{aligned} s_5 = s_3 = 6.904 \text{ kJ/kg·K} \\ P_5 = 1 \text{ MPa} \end{aligned}\right\} \qquad \therefore h_5 = 2932 \text{ kJ/kg}$$

$$s_4 = s_3 = 6.904 = 0.6491 + 7.5019x_4 \qquad \therefore x_4 = 0.8338$$

$$\therefore h_4 = 191.8 + (0.8338)(2392.8) = 2187 \text{ kJ/kg}$$

An energy balance on the heater, which is assumed insulated, is $\dot{m}_5(h_5 - h_6) = \dot{m}_2(h_7 - h_2)$. A mass balance provides (see Fig. 9-10) $\dot{m}_7 = \dot{m}_5 + \dot{m}_2$. Assuming $\dot{m}_7 = 1$ kg/s, the above two equations are combined to give

$$\dot{m}_2 = \frac{h_5 - h_6}{h_7 - h_2 + h_5 - h_6} = \frac{2932 - 763}{763 - 192 + 2932 - 763} = 0.792 \text{ kg/s}$$

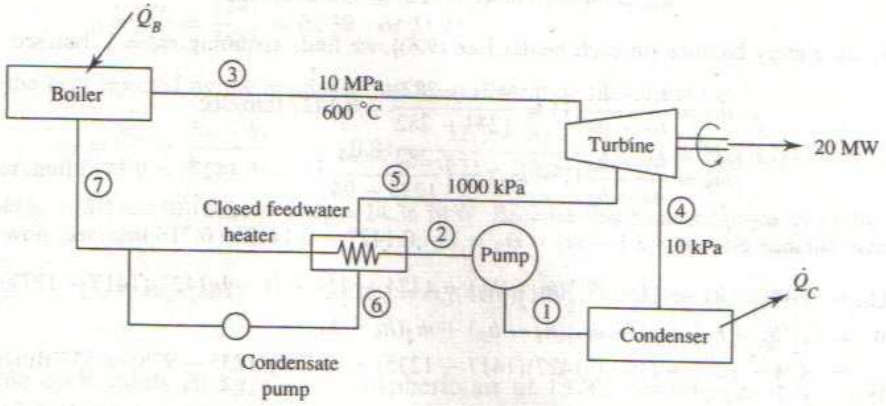


Fig. 9-20

We then have $\dot{m}_5 = 1 - \dot{m}_2 = 1 - 0.792 = 0.208$ kg/s. The turbine power (with $\dot{m}_7 = 1$ kg/s) can now be calculated to be

$$\dot{W}_T = \dot{m}_7(h_3 - h_5) + \dot{m}_2(h_5 - h_4) = (1.0)(3625 - 2932) + (0.792)(2932 - 2187) = 1283 \text{ kW}$$

The boiler heat rate is

$$\dot{Q}_B = \dot{m}_7(h_3 - h_7) = (1.0)(3625 - 763) = 2862 \text{ kW}$$

The cycle efficiency is calculated to be

$$\eta = \frac{\dot{W}_T}{\dot{Q}_B} = \frac{1283}{2862} = 0.448 \quad \text{or } 44.8\%$$

The mass flux of steam is found as

$$\dot{m}_7 = \frac{\dot{W}_T}{(\dot{W}_T)_{\text{with } \dot{m}_7 = 1}} = \frac{20}{1.283} = 15.59 \text{ kg/s}$$

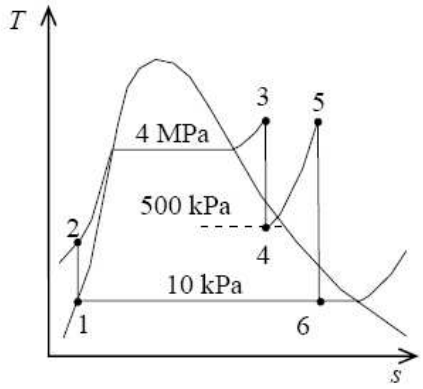
The ratio of rejected heat to added heat is

$$\frac{\dot{Q}_C}{\dot{Q}_B} = \frac{\dot{Q}_B - \dot{W}_T}{\dot{Q}_B} = 1 - \frac{\dot{W}_T}{\dot{Q}_B} = 1 - \frac{1283}{2862} = 0.552$$

4. Cengel 6th, p10-38

10-38 An ideal reheat Rankine with water as the working fluid is considered. The temperatures at the inlet of both turbines, and the thermal efficiency of the cycle are to be determined.

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



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

$$h_1 = h_{f@10\,\text{kPa}} = 191.81\,\text{kJ/kg}$$
$$v_1 = v_{f@10\,\text{kPa}} = 0.001010\,\text{m}^3/\text{kg}$$

$$w_{p,in} = v_1(P_2 - P_1)$$
$$= (0.001010\,\text{m}^3/\text{kg})(4000-10)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)$$
$$= 4.03\,\text{kJ/kg}$$
$$h_2 = h_1 + w_{p,in} = 191.81 + 4.03 = 195.84\,\text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 500\,\text{kPa} \\ x_4 = 0.90 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 640.09 + (0.90)(2108.0) = 2537.3\,\text{kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 1.8604 + (0.90)(4.9603) = 6.3247\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 4000\,\text{kPa} \\ s_3 = s_4 \end{array}\right\} \begin{array}{l} h_3 = 2939.4\,\text{kJ/kg} \\ T_3 = \mathbf{292.2°C} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10\,\text{kPa} \\ x_6 = 0.90 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.81 + (0.90)(2392.1) = 2344.7\,\text{kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6492 + (0.90)(7.4996) = 7.3989\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_5 = 500\,\text{kPa} \\ s_5 = s_6 \end{array}\right\} \begin{array}{l} h_5 = 3029.2\,\text{kJ/kg} \\ T_5 = \mathbf{282.9°C} \end{array}$$

Thus,

$$q_{in} = (h_3 - h_2) + (h_5 - h_4) = 2939.4 - 195.84 + 3029.2 - 2537.3 = 3235.4\,\text{kJ/kg}$$
$$q_{out} = h_6 - h_1 = 2344.7 - 191.81 = 2152.9\,\text{kJ/kg}$$

and

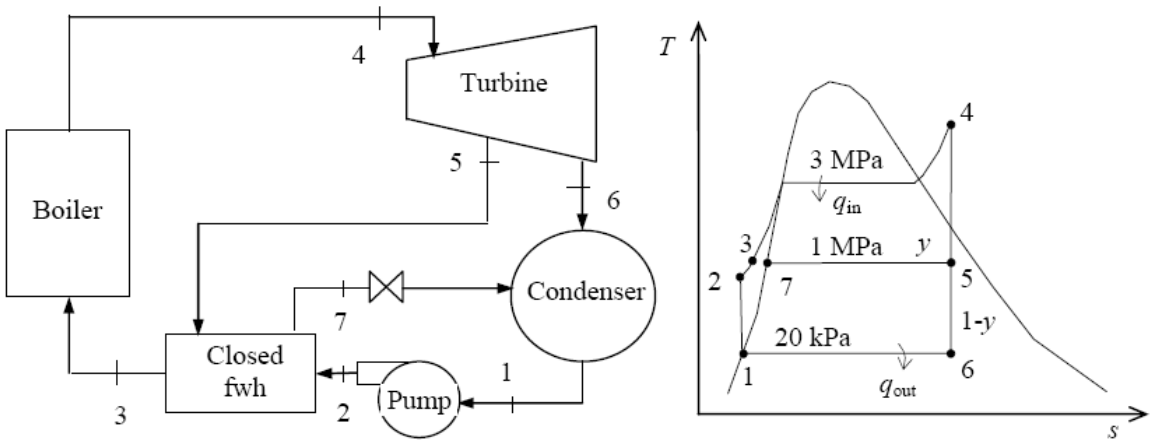
$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{2152.9}{3235.4} = \mathbf{0.335}$$

5. Cengel 6th, p10-55

10-55 An ideal regenerative Rankine cycle with a closed feedwater heater is considered. The work produced by the turbine, the work consumed by the pumps, and the heat added in the boiler are to be determined.

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

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



$$h_1 = h_{f@\,20\,\text{kPa}} = 251.42\,\text{kJ/kg}$$
$$\nu_1 = \nu_{f@\,20\,\text{kPa}} = 0.001017\,\text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{p,in}} &= \nu_1 (P_2 - P_1) \\ &= (0.001017\,\text{m}^3/\text{kg})(3000-20)\text{kPa}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) \\ &= 3.03\,\text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 3.03 = 254.45\,\text{kJ/kg}$$

$$\left.\begin{aligned} P_4 &= 3000\,\text{kPa} \\ T_4 &= 350°\text{C} \end{aligned}\right\} \begin{aligned} h_4 &= 3116.1\,\text{kJ/kg} \\ s_4 &= 6.7450\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_5 &= 1000\,\text{kPa} \\ s_5 &= s_4 \end{aligned}\right\} h_5 = 2851.9\,\text{kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 20\,\text{kPa} \\ s_6 &= s_4 \end{aligned}\right\} \begin{aligned} x_6 &= \frac{s_6 - s_f}{s_{fg}} = \frac{6.7450 - 0.8320}{7.0752} = 0.8357 \\ h_6 &= h_f + x_6 h_{fg} = 251.42 + (0.8357)(2357.5) = 2221.7\,\text{kJ/kg} \end{aligned}$$

For an ideal closed feedwater heater, the feedwater is heated to the exit temperature of the extracted steam, which ideally leaves the heater as a saturated liquid at the extraction pressure.

$$\left.\begin{aligned} P_7 &= 1000\,\text{kPa} \\ x_7 &= 0 \end{aligned}\right\} \begin{aligned} h_7 &= 762.51\,\text{kJ/kg} \\ T_7 &= 179.9°\text{C} \end{aligned}$$

$$\left.\begin{aligned} P_3 &= 3000\,\text{kPa} \\ T_3 &= T_7 = 209.9°\text{C} \end{aligned}\right\} h_3 = 763.53\,\text{kJ/kg}$$

An energy balance on the heat exchanger gives the fraction of steam extracted from the turbine ($= \dot{m}_5 / \dot{m}_4$) for closed feedwater heater:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_5 h_5 + \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7$$
$$y h_5 + 1 h_2 = 1 h_3 + y h_7$$

Rearranging,

$$y = \frac{h_3 - h_2}{h_5 - h_7} = \frac{763.53 - 254.45}{2851.9 - 762.51} = 0.2437$$

Then,

$$w_{T,out} = h_4 - h_5 + (1-y)(h_5 - h_6) = 3116.1 - 2851.9 + (1 - 0.2437)(2851.9 - 2221.7) = \mathbf{740.9\ kJ/kg}$$
$$w_{P,in} = \mathbf{3.03\ kJ/kg}$$
$$q_{in} = h_4 - h_3 = 3116.1 - 763.53 = \mathbf{2353\ kJ/kg}$$

Also,

$$w_{net} = w_{T,out} - w_{P,in} = 740.9 - 3.03 = 737.8\ \text{kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{737.8}{2353} = 0.3136$$

6. Schaum's 2nd, p8.3

8.3 An adiabatic compressor is supplied with 2 kg/s of atmospheric air at 15 °C and delivers it at 5 MPa. Calculate the efficiency and power input if the exiting temperature is 700 °C.

Assuming an isentropic process and an inlet temperature of 15 °C, the exit temperature would be

$$T_{2'} = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (288)\left(\frac{5000}{100}\right)^{0.2857} = 880.6\ \text{K}$$

The efficiency is then

$$\eta = \frac{w_s}{w_a} = \frac{C_p(T_{2'} - T_1)}{C_p(T_2 - T_1)} = \frac{880.6 - 288}{973 - 288} = 0.865 \text{ or } 86.5\%$$

The power input is $\dot{W}_{comp} = \dot{m} C_p (T_2 - T_1) = (2)(1.00)(973 - 288) = 1370$ kW.

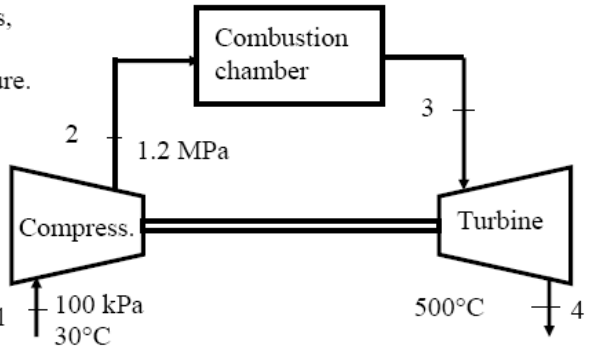
7. Cengel 6th, p9-105

9-105 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

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

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

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

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

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)
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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

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

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

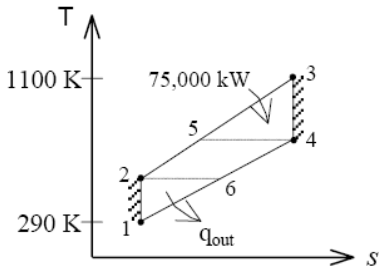
8. Cengel 5th, p9-97

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,

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

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

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

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

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

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

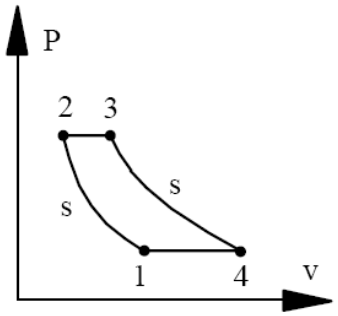
$$\dot{W}_{net} = \eta_T \dot{Q}_{in} = (0.5371)(75{,}000\text{ kW}) = \mathbf{40{,}283\ kW}$$

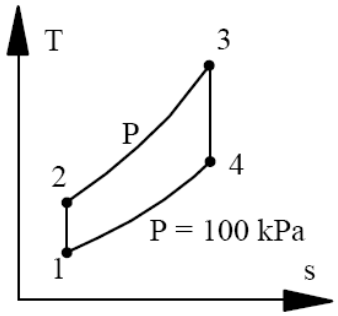
9. Borgnakke, p12.17

12.17

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

Solution:





Compression ratio

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

Max temperature

$T_3 = 1100^{o}C$

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

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

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

Energy equation with compressor work in

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

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

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

Energy equation with turbine work out

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

Scale the work with the mass flow rate

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

Energy added by the combustion process

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

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