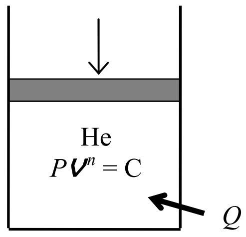
4-131 A cylinder is initially filled with helium gas at a specified state. Helium is compressed polytropically to a specified temperature and pressure. The heat transfer during the process is to be determined.

Assumptions **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Properties The gas constant of helium is R = 2.0769 kPa.m^3/kg.K (Table A-1). Also, c_v = 3.1156 kJ/kg.K (Table A-2).

Analysis The mass of helium and the exponent n are determined to be



$$m = \frac{P_1 V_1}{RT_1} = \frac{(150\text{ kPa})(0.5\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 0.123\text{ kg}$$

$$\frac{P_1 V_1}{RT_1} = \frac{P_2 V_2}{RT_2} \longrightarrow V_2 = \frac{T_2 P_1}{T_1 P_2} V_1 = \frac{413\text{ K}}{293\text{ K}} \times \frac{150\text{ kPa}}{400\text{ kPa}} \times 0.5\text{ m}^3 = 0.264\text{ m}^3$$

$$P_2 V_2^n = P_1 V_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{V_1}{V_2}\right)^n \longrightarrow \frac{400}{150} = \left(\frac{0.5}{0.264}\right)^n \longrightarrow n = 1.536$$

Then the boundary work for this polytropic process can be determined from

$$\begin{aligned} W_{\text{b,in}} &= -\int_1^2 P\,dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n} \\ &= -\frac{(0.123\text{ kg})(2.0769\text{ kJ/kg}\cdot\text{K})(413-293)\text{K}}{1-1.536} = 57.2\text{ kJ} \end{aligned}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Taking the direction of heat transfer to be to the cylinder, the energy balance for this stationary closed system can be expressed as

$$\begin{aligned} \underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} &= \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \\ Q_{\text{in}} + W_{\text{b,in}} &= \Delta U = m(u_2 - u_1) \\ Q_{\text{in}} &= m(u_2 - u_1) - W_{\text{b,in}} \\ &= mc_v(T_2 - T_1) - W_{\text{b,in}} \end{aligned}$$

Substituting,

Q_{in} = (0.123 kg)(3.1156 kJ/kg·K)(413 - 293)K - (57.2 kJ) = **-11.2 kJ**

The negative sign indicates that heat is lost from the system.

4-143 A well-insulated room is heated by a steam radiator, and the warm air is distributed by a fan. The average temperature in the room after 30 min is to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats at room temperature. **2** The kinetic and potential energy changes are negligible. **3** The air pressure in the room remains constant and thus the air expands as it is heated, and some warm air escapes.

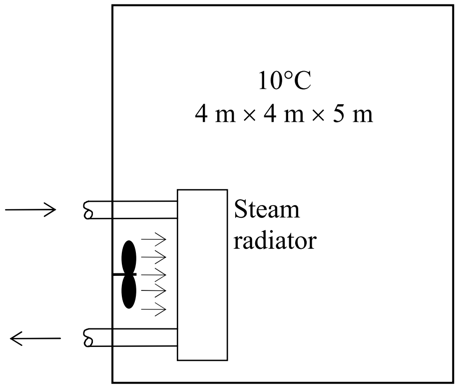
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_p = 1.005 kJ/kg.K for air at room temperature (Table A-2).

Analysis We first take the radiator as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = m(u_1 - u_2)$$



Using data from the steam tables (Tables A-4 through A-6), some properties are determined to be

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ (\boldsymbol{v}_2 = \boldsymbol{v}_1) \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001043, & \boldsymbol{v}_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40, & u_{fg} = 2088.2 \text{ kJ/kg} \end{array}$$

$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{1.08049 - 0.001043}{1.6941 - 0.001043} = 0.6376$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + 0.6376 \times 2088.2 = 1748.7 \text{ kJ/kg}$$

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.015 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.0139 \text{ kg}$$

Substituting, Q_{out} = (0.0139 kg)(2654.6 — 1748.7)kJ/kg = 12.58 kJ

The volume and the mass of the air in the room are $\boldsymbol{V}$= 4×4×5 = 80 m^3 and

$$m_{\text{air}} = \frac{P_1\boldsymbol{V}_1}{RT_1} = \frac{(100 \text{ kPa})(80 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283 \text{ K})} = 98.5 \text{ kg}$$

The amount of fan work done in 30 min is

$$W_{\text{fan,in}} = \dot{W}_{\text{fan,in}}\Delta t = (0.120 \text{ kJ/s})(30 \times 60 \text{ s}) = 216 \text{kJ}$$

We now take the air in the room as the system. The energy balance for this closed system is expressed as

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

$$Q_{\text{in}} + W_{\text{fan,in}} - W_{\text{b,out}} = \Delta U$$

$$Q_{\text{in}} + W_{\text{fan,in}} = \Delta H \cong mc_p(T_2 - T_1)$$

since the boundary work and ΔU combine into ΔH for a constant pressure expansion or compression process. It can also be expressed as

$$(\dot{Q}_{\text{in}} + \dot{W}_{\text{fan,in}})\Delta t = mc_{p,\text{avg}}(T_2 - T_1)$$

Substituting, (12.58 kJ) + (216 kJ) = (98.5 kg)(1.005 kJ/kg°C)(T_2 - 10)°C

which yields

T_2 = **12.3°C**

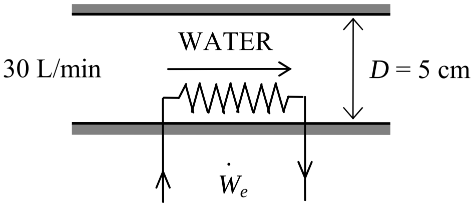
Therefore, the air temperature in the room rises from 10°C to 12.3°C in 30 min.

5-190 Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{\text{CV}} = 0$ and $\Delta E_{\text{CV}} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** The pipe is insulated and thus the heat losses are negligible.

Properties The density and specific heat of water at room temperature are ρ = 1000 kg/m^3 and $c = 4.18$ kJ/kg·°C (Table A-3).

Analysis (*a*) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. Also, there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{W}_{\text{e,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{\text{out}} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

The mass flow rate of water through the pipe is

$$\dot{m} = \rho\dot{V}_1 = (1000 \text{ kg/m}^3)(0.030 \text{ m}^3/\text{min}) = 30 \text{ kg/min}$$

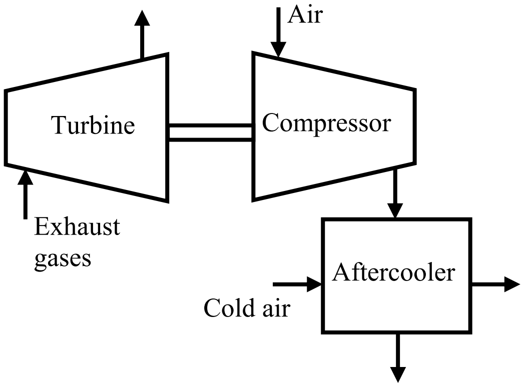
Therefore,

$$\dot{W}_{\text{e,in}} = \dot{m}c(T_2 - T_1) = (30/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(55 - 20)^\circ\text{C} = \mathbf{73.2\ kW}$$

(*b*) The average velocity of water through the pipe is determined from

$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\pi r^2} = \frac{0.030 \text{ m}^3/\text{min}}{\pi(0.025 \text{ m})^2} = \mathbf{15.3\ m/min}$$

5-195 The turbocharger of an internal combustion engine consisting of a turbine, a compressor, and an aftercooler is considered. The temperature of the air at the compressor outlet and the minimum flow rate of ambient air are to be determined.



Assumptions **1** All processes are steady since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air properties are used for exhaust gases. **4** Air is an ideal gas with constant specific heats. **5** The mechanical efficiency between the turbine and the compressor is 100%. **6** All devices are adiabatic. **7** The local atmospheric pressure is 100 kPa.

Properties The constant pressure specific heats of exhaust gases, warm air, and cold ambient air are taken to be c_p = 1.063, 1.008, and 1.005 kJ/kg·K, respectively (Table A-2b).

Analysis (*a*) An energy balance on turbine gives

$$\dot{W}_\text{T} = \dot{m}_\text{exh} c_{p,\text{exh}} \left(T_\text{exh,1} - T_\text{exh,2}\right) = (0.02\text{ kg/s})(1.063\text{ kJ/kg}\cdot\text{K})(400-350)\text{K} = 1.063\text{ kW}$$

This is also the power input to the compressor since the mechanical efficiency between the turbine and the compressor is assumed to be 100%. An energy balance on the compressor gives the air temperature at the compressor outlet

$$\dot{W}_\text{C} = \dot{m}_\text{a} c_{p,\text{a}} (T_\text{a,2} - T_\text{a,1})$$

$$1.063\text{ kW} = (0.018\text{ kg/s})(1.008\text{ kJ/kg}\cdot\text{K})(T_\text{a,2} - 50)\text{K} \longrightarrow T_\text{a,2} = \mathbf{108.6\ ^\circ C}$$

(*b*) An energy balance on the aftercooler gives the mass flow rate of cold ambient air

$$\dot{m}_\text{a} c_{p,\text{a}} (T_\text{a,2} - T_\text{a,3}) = \dot{m}_\text{ca} c_{p,\text{ca}} (T_\text{ca,2} - T_\text{ca,1})$$

$$(0.018\text{ kg/s})(1.008\text{ kJ/kg}\cdot{}^\circ\text{C})(108.6-80)^\circ\text{C} = \dot{m}_\text{ca}(1.005\text{ kJ/kg}\cdot{}^\circ\text{C})(40-30)^\circ\text{C}$$

$$\dot{m}_\text{ca} = 0.05161\text{ kg/s}$$

The volume flow rate may be determined if we first calculate specific volume of cold ambient air at the inlet of aftercooler. That is,

$$v_\text{ca} = \frac{RT}{P} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(30+273\text{ K})}{100\text{ kPa}} = 0.8696\text{ m}^3/\text{kg}$$

$$\dot{V}_\text{ca} = \dot{m} v_\text{ca} = (0.05161\text{ kg/s})(0.8696\text{ m}^3/\text{kg}) = \mathbf{0.0449\ m^3/s} = \mathbf{44.9\ L/s}$$

7-163 A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water stream and the rate of entropy generation are to be determined.

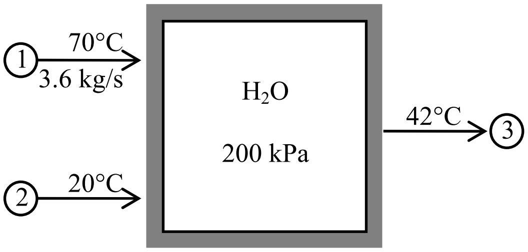
Assumptions **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

Properties Noting that $T < T_{\text{sat @ 200 kPa}} = 120.21°\text{C}$, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus from Table A-4,

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = 70°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f@70°\text{C}} = 293.07\text{ kJ/kg} \\ s_1 \cong s_{f@70°\text{C}} = 0.9551\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200\text{ kPa} \\ T_2 = 20°\text{C} \end{array}\right\} \begin{array}{l} h_2 \cong h_{f@20°\text{C}} = 83.91\text{ kJ/kg} \\ s_2 \cong s_{f@20°\text{C}} = 0.2965\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 200\text{ kPa} \\ T_3 = 42°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@42°\text{C}} = 175.90\text{ kJ/kg} \\ s_3 \cong s_{f@42°\text{C}} = 0.5990\text{ kJ/kg}\cdot\text{K} \end{array}$$



Analysis (*a*) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}h_1 + \dot{m}_2h_2 = \dot{m}_3h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two relations gives

$$\dot{m}_1h_1 + \dot{m}_2h_2 = (\dot{m}_1 + \dot{m}_2)h_3$$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2}\dot{m}_1 = \frac{(293.07 - 175.90)\text{kJ/kg}}{(175.90 - 83.91)\text{kJ/kg}}(3.6\text{ kg/s}) = \mathbf{4.586\ kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 3.6 + 4.586 = 8.186\text{ kg/s}$

(*b*) Noting that the mixing chamber is adiabatic and thus there is no heat transfer to the surroundings, the entropy balance of the steady-flow system (the mixing chamber) can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}_1s_1 + \dot{m}_2s_2 - \dot{m}_3s_3 + \dot{S}_{\text{gen}} = 0$$

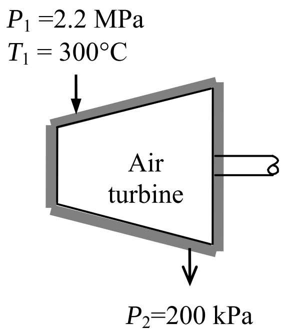
Substituting, the total rate of entropy generation during this process becomes

$$\begin{aligned}\dot{S}_{\text{gen}} &= \dot{m}_3s_3 - \dot{m}_2s_2 - \dot{m}_1s_1 \\ &= (8.186\text{ kg/s})(0.5990\text{ kJ/kg}\cdot\text{K}) - (4.586\text{ kg/s})(0.2965\text{ kJ/kg}\cdot\text{K}) - (3.6\text{ kg/s})(0.9551\text{ kJ/kg}\cdot\text{K}) \\ &= \mathbf{0.1054\ kW/K}\end{aligned}$$

7-209 Air is expanded by an adiabatic turbine with an isentropic efficiency of 85%. The outlet temperature, the work produced, and the entropy generation are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

Properties The properties of air at the anticipated average temperature of 400 K are c_p = 1.013 kJ/kg·°C and k = 1.395 (Table A-2b). Also, R = 0.287 kJ/kg·K (Table A-2a).



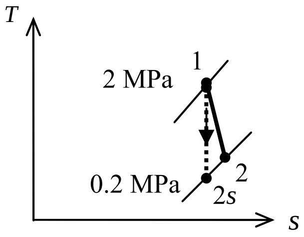
Analysis We take the turbine as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters and leaves the turbine, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\;\cancel{\nearrow}0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}h_1 = \dot{W}_{a,\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,\text{out}} = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$



The isentropic exit temperature is

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (300+273\text{ K})\left(\frac{200\text{ kPa}}{2200\text{ kPa}}\right)^{0.395/1.395} = 290.6\text{ K}$$

From the definition of the isentropic efficiency,

$$w_{a,\text{out}} = \eta_T w_{s,\text{out}} = \eta_T c_p(T_1 - T_{2s}) = (0.90)(1.013\text{ kJ/kg}\cdot\text{K})(573 - 290.6)\text{K} = \mathbf{257.5\text{ kJ/kg}}$$

The actual exit temperature is then

$$w_{a,\text{out}} = c_p(T_1 - T_{2a}) \longrightarrow T_{2a} = T_1 - \frac{w_{a,\text{out}}}{c_p} = T_1 - \frac{w_{a,\text{out}}}{c_p} = 573\text{ K} - \frac{257.5\text{ kJ/kg}}{1.013\text{ kJ/kg}\cdot\text{K}} = \mathbf{318.8\text{ K}}$$

The rate of entropy generation in the turbine is determined by applying the rate form of the entropy balance on the turbine:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\;\cancel{\nearrow}0\text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } \dot{Q} = 0)$$

$$\dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

$$s_{\text{gen}} = s_2 - s_1$$

Then, from the entropy change relation of an ideal gas,

$$\begin{aligned} s_{\text{gen}} = s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.013\text{ kJ/kg}\cdot\text{K})\ln\frac{318.8\text{ K}}{573\text{ K}} - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{200\text{ kPa}}{2200\text{ kPa}} \\ &= \mathbf{0.0944\text{ kJ/kg}\cdot\text{K}} \end{aligned}$$

7-215 The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted steam to the feedwater and entropy generation per unit mass of feedwater are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the device to the surroundings is negligible.

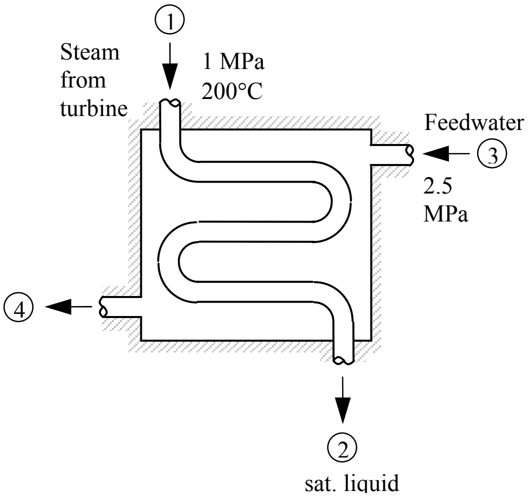
Properties The properties of steam and feedwater are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 2828.3 \text{ kJ/kg} \\ s_1 = 6.6956 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_2 = h_{f@1\text{ MPa}} = 762.51 \text{ kJ/kg} \\ s_2 = s_{f@1\text{ MPa}} = 2.1381 \text{ kJ/kg}\cdot\text{K} \\ T_2 = 179.88°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 2.5 \text{ MPa} \\ T_3 = 50°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@50^\circ\text{C}} = 209.34 \text{ kJ/kg} \\ s_3 \cong s_{f@50^\circ\text{C}} = 0.7038 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 2.5 \text{ MPa} \\ T_4 = T_2 - 10^\circ\text{C} \cong 170^\circ\text{C} \end{array}\right\} \begin{array}{l} h_4 \cong h_{f@170^\circ\text{C}} = 719.08 \text{ kJ/kg} \\ s_4 \cong s_{f@170^\circ\text{C}} = 2.0417 \text{ kJ/kg}\cdot\text{K} \end{array}$$



Analysis (*a*) We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

Mass balance (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_s \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_{fw}$$

Energy balance (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_s(h_2 - h_1) = \dot{m}_{fw}(h_3 - h_4)$

Dividing by $\dot{m}_{fw}$ and substituting,

$$\frac{\dot{m}_s}{\dot{m}_{fw}} = \frac{h_4 - h_3}{h_1 - h_2} = \frac{(719.08 - 209.34) \text{ kJ/kg}}{(2828.3 - 762.51) \text{ kJ/kg}} = \mathbf{0.247}$$

(*b*) The total entropy change (or entropy generation) during this process per unit mass of feedwater can be determined from an entropy balance expressed in the rate form as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{m}_s(s_1 - s_2) + \dot{m}_{fw}(s_3 - s_4) + \dot{S}_{\text{gen}} = 0$$

$$\frac{\dot{S}_{\text{gen}}}{\dot{m}_{fw}} = \frac{\dot{m}_s}{\dot{m}_{fw}}(s_2 - s_1) + (s_4 - s_3) = (0.247)(2.1381 - 6.6956) + (2.0417 - 0.7038)$$

$$= \mathbf{0.213 \text{ kJ/K}} \text{ per kg of feedwater}$$

7-226 Water is heated from 16°C to 43°C by an electric resistance heater placed in the water pipe as it flows through a showerhead steadily at a rate of 10 L/min. The electric power input to the heater and the rate of entropy generation are to be determined. The reduction in power input and entropy generation as a result of installing a 50% efficient regenerator are also to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** Heat losses from the pipe are negligible.

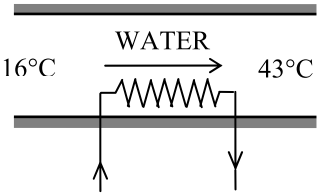
Properties The density of water is given to be ρ = 1 kg/L. The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

Analysis (*a*) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot m_1 = \dot m_2 = \dot m$. Then the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot E_{\rm in} - \dot E_{\rm out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\rm system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot E_{\rm in} = \dot E_{\rm out}$$

$$\dot W_{\rm e,in} + \dot m h_1 = \dot m h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot W_{\rm e,in} = \dot m(h_2 - h_1) = \dot m c(T_2 - T_1)$$



where

$$\dot m = \rho \dot{\mathbf{V}} = (1\ \text{kg/L})(10\ \text{L/min}) = 10\ \text{kg/min}$$

Substituting,

$$\dot W_{\rm e,in} = (10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(43-16)^\circ\text{C} = \mathbf{18.8\ kW}$$

The rate of entropy generation in the heating section during this process is determined by applying the entropy balance on the heating section. Noting that this is a steady-flow process and heat transfer from the heating section is negligible,

$$\underbrace{\dot S_{\rm in} - \dot S_{\rm out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot S_{\rm gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot S_{\rm system}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot m s_1 - \dot m s_2 + \dot S_{\rm gen} = 0 \longrightarrow \dot S_{\rm gen} = \dot m(s_2 - s_1)$$

Noting that water is an incompressible substance and substituting,

$$\dot S_{\rm gen} = \dot m c \ln\frac{T_2}{T_1} = (10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot\text{K})\ln\frac{316\ \text{K}}{289\ \text{K}} = \mathbf{0.0622\ kJ/K}$$

(*b*) The energy recovered by the heat exchanger is

$$\dot Q_{\rm saved} = \varepsilon \dot Q_{\rm max} = \varepsilon \dot m C(T_{\rm max} - T_{\rm min}) = 0.5(10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(39-16)^\circ\text{C} = 8.0\ \text{kJ/s} = 8.0\ \text{kW}$$

Therefore, 8.0 kW less energy is needed in this case, and the required electric power in this case reduces to

$$\dot W_{\rm in,new} = \dot W_{\rm in,old} - \dot Q_{\rm saved} = 18.8 - 8.0 = \mathbf{10.8\ kW}$$

Taking the cold water stream in the heat exchanger as our control volume (a steady-flow system), the temperature at which the cold water leaves the heat exchanger and enters the electric resistance heating section is determined from

$$\dot Q = \dot m c(T_{\rm c,out} - T_{\rm c,in})$$

Substituting,

$$8\ \text{kJ/s} = (10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(T_{\rm c,out} - 16^\circ\text{C})$$

It yields

$$T_{\text{c,out}} = 27.5^\circ\text{C} = 300.5\text{K}$$

The rate of entropy generation in the heating section in this case is determined similarly to be

$$\dot{S}_{\text{gen}} = \dot{m}c\ln\frac{T_2}{T_1} = (10/60\text{ kg/s})(4.18\text{ kJ/kg}\cdot\text{K})\ln\frac{316\text{ K}}{300.5\text{ K}} = 0.0350\text{ kJ/K}$$

Thus the reduction in the rate of entropy generation within the heating section is

$$\dot{S}_{\text{reduction}} = 0.0622 - 0.0350 = \mathbf{0.0272\ kW/K}$$

8-31 Steam and R-134a at the same states are considered. The fluid with the higher exergy content is to be identified.

Assumptions Kinetic and potential energy changes are negligible.

Analysis The properties of water at the given state and at the dead state are

Steam
1 kg
800 kPa
180°C

$$\left.\begin{array}{l} P = 800\text{ kPa} \\ T = 180°\text{C} \end{array}\right\} \begin{array}{l} u = 2594.7\text{ kJ/kg} \\ \boldsymbol{v} = 0.24720\text{ m}^3\text{/kg} \quad \text{(Table A - 6)} \\ s = 6.7155\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ P_0 = 100\text{ kPa} \end{array}\right\} \begin{array}{l} u_0 \cong u_{f\,@\,25°\text{C}} = 104.83\text{ kJ/kg} \\ \boldsymbol{v}_0 \cong \boldsymbol{v}_{f\,@\,25°\text{C}} = 0.001003\text{ m}^3\text{/kg} \quad \text{(Table A - 4)} \\ s_0 \cong s_{f\,@\,25°\text{C}} = 0.3672\text{ kJ/kg}\cdot\text{K} \end{array}$$

The exergy of steam is

$$\begin{aligned} \Phi &= m\left[u - u_0 + P_0(\boldsymbol{v} - \boldsymbol{v}_0) - T_0(s - s_0)\right] \\ &= (1\text{ kg})\left[\begin{array}{l} (2594.7 - 104.83)\text{kJ/kg} + (100\text{ kPa})(0.24720 - 0.001003)\text{m}^3\text{/kg}\left(\dfrac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ -(298\text{ K})(6.7155 - 0.3672)\text{kJ/kg}\cdot\text{K} \end{array}\right] \\ &= \mathbf{622.7\ kJ} \end{aligned}$$

For R-134a;

R-134a
1 kg
800 kPa
180°C

$$\left.\begin{array}{l} P = 800\text{ kPa} \\ T = 180°\text{C} \end{array}\right\} \begin{array}{l} u = 386.99\text{ kJ/kg} \\ \boldsymbol{v} = 0.044554\text{ m}^3\text{/kg} \quad \text{(Table A -13)} \\ s = 1.3327\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ P_0 = 100\text{ kPa} \end{array}\right\} \begin{array}{l} u_0 \cong u_{f\,@\,25°\text{C}} = 85.85\text{ kJ/kg} \\ \boldsymbol{v}_0 \cong \boldsymbol{v}_{f\,@\,25°\text{C}} = 0.0008286\text{ m}^3\text{/kg} \quad \text{(Table A -11)} \\ s_0 \cong s_{f\,@\,25°\text{C}} = 0.32432\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\begin{aligned} \Phi &= m\left[u - u_0 + P_0(\boldsymbol{v} - \boldsymbol{v}_0) - T_0(s - s_0)\right] \\ &= (1\text{ kg})\left[\begin{array}{l} (386.99 - 85.85)\text{kJ/kg} + (100\text{ kPa})(0.044554 - 0.0008286)\text{m}^3\text{/kg}\left(\dfrac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ -(298\text{ K})(1.3327 - 0.32432)\text{kJ/kg}\cdot\text{K} \end{array}\right] \\ &= \mathbf{5.02\ kJ} \end{aligned}$$

The steam can therefore has more work potential than the R-134a.

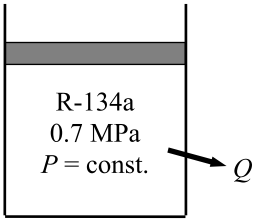
8-32 A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled and condensed at constant pressure. The exergy of the refrigerant at the initial and final states, and the exergy destroyed during this process are to be determined.

Assumptions The kinetic and potential energies are negligible.

Properties From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 0.7\,\text{MPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.034875\,\text{m}^3/\text{kg} \\ u_1 = 274.01\,\text{kJ/kg} \\ s_1 = 1.0256\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\,\text{MPa} \\ T_2 = 24°\text{C} \end{array}\right\} \begin{array}{l} v_2 \cong v_{f@24°\text{C}} = 0.0008261\,\text{m}^3/\text{kg} \\ u_2 \cong u_{f@24°\text{C}} = 84.44\,\text{kJ/kg} \\ s_2 \cong s_{f@24°\text{C}} = 0.31958\,\text{kJ/kg}\cdot\text{K} \end{array}$$



$$\left.\begin{array}{l} P_0 = 0.1\,\text{MPa} \\ T_0 = 24°\text{C} \end{array}\right\} \begin{array}{l} v_0 = 0.23718\,\text{m}^3/\text{kg} \\ u_0 = 251.84\,\text{kJ/kg} \\ s_0 = 1.1033\,\text{kJ/kg}\cdot\text{K} \end{array}$$

Analysis (*a*) From the closed system exergy relation,

$$\begin{aligned} X_1 = \Phi_1 &= m\{(u_1 - u_0) - T_0(s_1 - s_0) + P_0(v_1 - v_0)\} \\ &= (5\,\text{kg})\{(274.01 - 251.84)\,\text{kJ/kg} - (297\,\text{K})(1.0256 - 1.1033)\,\text{kJ/kg}\cdot\text{K} \\ &\quad + (100\,\text{kPa})(0.034875 - 0.23718)\text{m}^3/\text{kg}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)\} \\ &= \mathbf{125.1\ kJ} \end{aligned}$$

and

$$\begin{aligned} X_2 = \Phi_2 &= m\{(u_2 - u_0) - T_0(s_2 - s_0) + P_0(v_2 - v_0)\} \\ &= (5\,\text{kg})\{(84.44 - 251.84)\,\text{kJ/kg} - (297\,\text{K})(0.31958 - 1.1033)\,\text{kJ/kg}\cdot\text{K} \\ &\quad + (100\,\text{kPa})(0.0008261 - 0.23718)\text{m}^3/\text{kg}\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right)\} \\ &= \mathbf{208.6\ kJ} \end{aligned}$$

(*b*) The reversible work input, which represents the minimum work input $W_{\text{rev,in}}$ in this case can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- X_{\text{destroyed}}^{\nearrow 0\ (\text{reversible})}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$W_{\text{rev,in}} = X_2 - X_1 = 208.6 - 125.1 = 83.5\,\text{kJ}$$

Noting that the process involves only boundary work, the useful work input during this process is simply the boundary work in excess of the work done by the surrounding air,

$$\begin{aligned} W_{\text{u,in}} &= W_{\text{in}} - W_{\text{surr,in}} = W_{\text{in}} - P_0(V_1 - V_2) = P(V_1 - V_2) - P_0 m(v_1 - v_2) \\ &= m(P - P_0)(v_1 - v_2) \\ &= (5\,\text{kg})(700 - 100\,\text{kPa})(0.034875 - 0.0008261\,\text{m}^3/\text{kg})\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) = 102.1\,\text{kJ} \end{aligned}$$

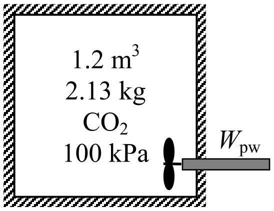
Knowing both the actual useful and reversible work inputs, the exergy destruction or irreversibility that is the difference between the two is determined from its definition to be

$$X_{\text{destroyed}} = I = W_{\text{u,in}} - W_{\text{rev,in}} = 102.1 - 83.5 = \mathbf{18.6\ kJ}$$

8-41 An insulated tank contains CO_2 gas at a specified pressure and volume. A paddle-wheel in the tank stirs the gas, and the pressure and temperature of CO_2 rises. The actual paddle-wheel work and the minimum paddle-wheel work by which this process can be accomplished are to be determined.

Assumptions **1** At specified conditions, CO_2 can be treated as an ideal gas with constant specific heats at the average temperature. **2** The surroundings temperature is 298 K.

Properties The gas constant of CO_2 is 0.1889 kJ/kg·K (Table A-1)



Analysis (a) The initial and final temperature of CO_2 are

$$T_1 = \frac{P_1\boldsymbol{V}_1}{mR} = \frac{(100\,\text{kPa})(1.2\,\text{m}^3)}{(2.13\,\text{kg})(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 298.2\,\text{K}$$

$$T_2 = \frac{P_2\boldsymbol{V}_2}{mR} = \frac{(120\,\text{kPa})(1.2\,\text{m}^3)}{(2.13\,\text{kg})(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 357.9\,\text{K}$$

$$T_{\text{avg}} = (T_1+T_2)/2 = (298.2+357.9)/2 = 328\,\text{K} \longrightarrow c_{\boldsymbol{v},\text{avg}} = 0.684\,\text{kJ/kg}\cdot\text{K} \;\; \text{(Table A-2}b)$$

The actual paddle-wheel work done is determined from the energy balance on the CO gas in the tank,

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} = \Delta U = mc_{\boldsymbol{v}}(T_2 - T_1)$$

or

$$W_{\text{pw,in}} = (2.13\,\text{kg})(0.684\,\text{kJ/kg}\cdot\text{K})(357.9-298.2)\text{K} = \mathbf{87.0\,kJ}$$

(b) The minimum paddle-wheel work with which this process can be accomplished is the reversible work,

which can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} \underbrace{- \overset{\nearrow 0 \text{ (reversible)}}{X_{\text{destroyed}}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}} \rightarrow W_{\text{rev,in}} = X_2 - X_1$$

Substituting the closed system exergy relation, the reversible work input for this process is determined to be

$$\begin{aligned} W_{\text{rev,in}} &= m\left[(u_2-u_1) - T_0(s_2-s_1) + P_0(\overset{\nearrow 0}{\boldsymbol{v}_2} - \boldsymbol{v}_1)\right] \\ &= m\left[c_{\boldsymbol{v},\text{avg}}(T_2-T_1) - T_0(s_2-s_1)\right] \\ &= (2.13\,\text{kg})\left[(0.684\,\text{kJ/kg}\cdot\text{K})(357.9-298.2)\text{K} - (298.2)(0.1253\,\text{kJ/kg}\cdot\text{K})\right] \\ &= \mathbf{7.74\ kJ} \end{aligned}$$

since

$$s_2 - s_1 = c_{\boldsymbol{v},\text{avg}} \ln\frac{T_2}{T_1} + R\ln\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}^{\nearrow 0} = (0.684\,\text{kJ/kg}\cdot\text{K})\ln\left(\frac{357.9\,\text{K}}{298.2\,\text{K}}\right) = 0.1253\,\text{kJ/kg}\cdot\text{K}$$

8-49 Carbon steel balls are to be annealed at a rate of 2500/h by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The total rate of heat transfer from the balls to the ambient air and the rate of exergy destruction due to this heat transfer are to be determined.

Assumptions 1 The thermal properties of the balls are constant. 2 There are no changes in kinetic and potential energies. 3 The balls are at a uniform temperature at the end of the process.

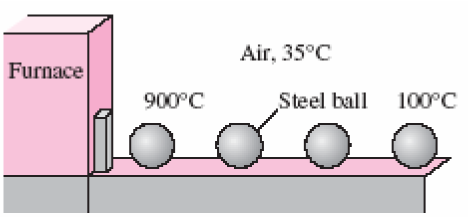
Properties The density and specific heat of the balls are given to be ρ = 7833 kg/m^3 and c_p = 0.465 kJ/kg.°C.

Analysis (*a*) We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc_p(T_1 - T_2)$$



The amount of heat transfer from a single ball is

$$m = \rho \mathbf{V} = \rho \frac{\pi D^3}{6} = (7833\ \text{kg/m}^3)\frac{\pi(0.008\ \text{m})^3}{6} = 0.00210\ \text{kg}$$

$$Q_{\text{out}} = mc_p(T_1 - T_2) = (0.0021\ \text{kg})(0.465\ \text{kJ/kg.°C})(900-100)\text{°C} = 781\ \text{J} = 0.781\ \text{kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

$$\dot{Q}_{\text{out}} = \dot{n}_{\text{ball}} Q_{\text{out}} = (1200\ \text{balls/h}) \times (0.781\ \text{kJ/ball}) = 936\ \text{kJ/h} = \mathbf{260\ \ W}$$

(*b*) The exergy destruction (or irreversibility) can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the ball and its immediate surroundings so that the boundary temperature of the extended system is at 35°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \rightarrow S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_p \ln\frac{T_2}{T_1} = (0.00210\ \text{kg})(0.465\ \text{kJ/kg.K})\ln\frac{100+273}{900+273} = -0.00112\ \text{kJ/K}$$

Substituting,

$$S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}} = \frac{0.781\ \text{kJ}}{308\ \text{K}} - 0.00112\ \text{kJ/K} = 0.00142\ \text{kJ/K} \quad \text{(per ball)}$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = S_{\text{gen}} \dot{n}_{\text{ball}} = (0.00142\ \text{kJ/K} \cdot \text{ball})(1200\ \text{balls/h}) = 1.704\ \text{kJ/h.K} = 0.000473\ \ \text{kW/K}$$

Finally,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (308\ \text{K})(0.000473\ \text{kW/K}) = 0.146\ \text{kW} = \mathbf{146\ \ W}$$

8-61 Steam expands in a turbine from a specified state to another specified state. The actual power output of the turbine is given. The reversible power output and the second-law efficiency are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** The temperature of the surroundings is given to be 25°C.

Properties From the steam tables (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 6\text{ MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3658.8\text{ kJ/kg} \\ s_1 = 7.1693\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50\text{ kPa} \\ T_2 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 2682.4\text{ kJ/kg} \\ s_2 = 7.6953\text{ kJ/kg}\cdot\text{K} \end{array}$$

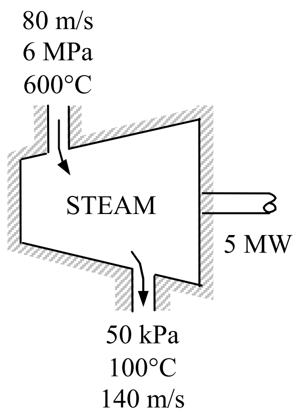
Analysis (*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2)$$

$$\dot{W}_{\text{out}} = \dot{m}\left[h_1 - h_2 + \frac{V_1^2 - V_2^2}{2}\right]$$



Substituting,

$$5000\text{ kJ/s} = \dot{m}\left(3658.8 - 2682.4 + \frac{(80\text{ m/s})^2 - (140\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right)$$

$$\dot{m} = 5.156\ \text{ kg/s}$$

The reversible (or maximum) power output is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0 \text{ (reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

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

$$\dot{m}\psi_1 = \dot{W}_{\text{rev,out}} + \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,out}} = \dot{m}(\psi_1 - \psi_2) = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta\text{ke} - \Delta\text{pe}^{\nearrow 0}]$$

Substituting,

$$\begin{aligned} \dot{W}_{\text{rev,out}} &= \dot{W}_{\text{out}} - \dot{m}T_0(s_1 - s_2) \\ &= 5000\text{ kW} - (5.156\text{ kg/s})(298\text{ K})(7.1693 - 7.6953)\text{ kJ/kg}\cdot\text{K} = \mathbf{5808\ \ kW} \end{aligned}$$

(*b*) The second-law efficiency of a turbine is the ratio of the actual work output to the reversible work,

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{out}}}{\dot{W}_{\text{rev,out}}} = \frac{5\text{ MW}}{5.808\text{ MW}} = \mathbf{86.1\%}$$

Discussion Note that 13.9% percent of the work potential of the steam is wasted as it flows through the turbine during this process.

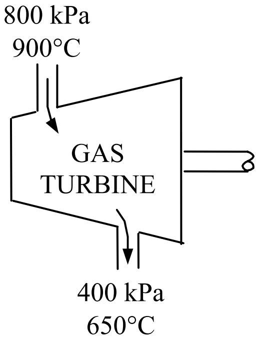
8-64 Combustion gases expand in a turbine from a specified state to another specified state. The exergy of the gases at the inlet and the reversible work output of the turbine are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C. **4** The combustion gases are ideal gases with constant specific heats.

Properties The constant pressure specific heat and the specific heat ratio are given to be $c_p = 1.15$ kJ/kg.K and $k = 1.3$. The gas constant R is determined from

$$R = c_p - c_v = c_p - c_p / k = c_p(1 - 1/k) = (1.15\,\text{kJ/kg}\cdot\text{K})(1 - 1/1.3) = 0.265\,\text{kJ/kg}\cdot\text{K}$$

Analysis (*a*) The exergy of the gases at the turbine inlet is simply the flow exergy,



$$\psi_1 = h_1 - h_0 - T_0(s_1 - s_0) + \frac{V_1^2}{2} + \cancelto{0}{gz_1}$$

where

$$\begin{aligned} s_1 - s_0 &= c_p \ln\frac{T_1}{T_0} - R\ln\frac{P_1}{P_0} \\ &= (1.15\,\text{kJ/kg}\cdot\text{K})\ln\frac{1173\,\text{K}}{298\,\text{K}} - (0.265\,\text{kJ/kg}\cdot\text{K})\ln\frac{800\,\text{kPa}}{100\,\text{kPa}} \\ &= 1.025\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

Thus,

$$\psi_1 = (1.15\,\text{kJ/kg.K})(900-25)^\circ\text{C} - (298\,\text{K})(1.025\,\text{kJ/kg}\cdot\text{K}) + \frac{(100\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{705.8\,kJ/kg}$$

(*b*) The reversible (or maximum) work output is determined from an exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} \underbrace{- \cancelto{0\text{ (reversible)}}{\dot{X}_{\text{destroyed}}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\cancelto{0\text{ (steady)}}{\Delta\dot{X}_{\text{system}}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\begin{aligned} \dot{X}_{\text{in}} &= \dot{X}_{\text{out}} \\ \dot{m}\psi_1 &= \dot{W}_{\text{rev,out}} + \dot{m}\psi_2 \\ \dot{W}_{\text{rev,out}} &= \dot{m}(\psi_1 - \psi_2) = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta\text{ke} - \cancelto{0}{\Delta\text{pe}}] \end{aligned}$$

where

$$\Delta\text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(220\,\text{m/s})^2 - (100\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 19.2\,\text{kJ/kg}$$

and

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.15\,\text{kJ/kg}\cdot\text{K})\ln\frac{923\,\text{K}}{1173\,\text{K}} - (0.265\,\text{kJ/kg}\cdot\text{K})\ln\frac{400\,\text{kPa}}{800\,\text{kPa}} \\ &= -0.09196\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

Then the reversible work output on a unit mass basis becomes

$$\begin{aligned} w_{\text{rev,out}} &= h_1 - h_2 + T_0(s_2 - s_1) - \Delta\text{ke} = c_p(T_1 - T_2) + T_0(s_2 - s_1) - \Delta\text{ke} \\ &= (1.15\,\text{kJ/kg}\cdot\text{K})(900-650)^\circ\text{C} + (298\,\text{K})(-0.09196\,\text{kJ/kg}\cdot\text{K}) - 19.2\,\text{kJ/kg} \\ &= \mathbf{240.9\,kJ/kg} \end{aligned}$$

8-79 A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The mass of the R-134a that entered the tank and the exergy destroyed during this process are to be determined.

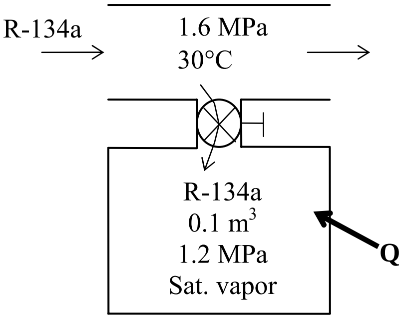
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

Properties The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1.2\text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@1.2\text{ MPa}} = 0.01672\text{ m}^3/\text{kg} \\ u_1 = u_{g@1.2\text{ MPa}} = 253.81\text{ kJ/kg} \\ s_1 = s_{g@1.2\text{ MPa}} = 0.91303\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 1.4\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{f@1.4\text{ MPa}} = 0.0009166\text{ m}^3/\text{kg} \\ u_2 = u_{f@1.4\text{ MPa}} = 125.94\text{ kJ/kg} \\ s_2 = s_{f@1.4\text{ MPa}} = 0.45315\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.6\text{ MPa} \\ T_i = 30°\text{C} \end{array}\right\} \begin{array}{l} h_i = 93.56\text{ kJ/kg} \\ s_i = 0.34554\text{ kJ/kg}\cdot\text{K} \end{array}$$



Analysis We take the tank as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance:

$$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

(*a*) The initial and the final masses in the tank are

$$m_1 = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.1\text{ m}^3}{0.01672\text{ m}^3/\text{kg}} = 5.983\text{ kg}$$

$$m_2 = \frac{\boldsymbol{V}_2}{\boldsymbol{v}_2} = \frac{0.1\text{ m}^3}{0.0009166\text{ m}^3/\text{kg}} = 109.10\text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 109.10 - 5.983 = \mathbf{103.11\text{ kg}}$$

The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(103.11\text{ kg})(93.56\text{ kJ/kg}) + (109.10)(125.94\text{ kJ/kg}) - (5.983\text{ kg})(253.81\text{ kJ/kg}) \\ &= 2573\text{ kJ} \end{aligned}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation S_{gen} in this case is determined from an entropy balance on an *extended system* that includes the tank and its immediate surroundings so that the boundary temperature of the extended system is the surroundings temperature T_{surr} at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} + m_i s_i + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$ Substituting, the exergy destruction

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 - m_i s_i - \frac{Q_{\text{in}}}{T_0}$$

is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 \left[m_2 s_2 - m_1 s_1 - m_i s_i - \frac{Q_{\text{in}}}{T_0} \right] \\ &= (318\ \text{K})\left[109.10 \times 0.45315 - 5.983 \times 0.91303 - 103.11 \times 0.34554 - (2573\ \text{kJ})/(318\ \text{K})\right] \\ &= \mathbf{80.3\ kJ} \end{aligned}$$

8-86 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of exergy destruction are to be determined.

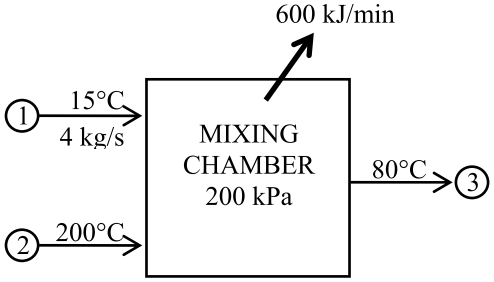
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Properties Noting that $T < T_{\text{sat @ 200 kPa}} = 120.23°\text{C}$, the cold water and the exit mixture streams exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. From Tables A-4 through A-6,

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 15°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f@15^\circ\text{C}} = 62.98 \text{ kJ/kg} \\ s_1 \cong s_{f@15^\circ\text{C}} = 0.22447 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ T_2 = 200°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 2870.4 \text{ kJ/kg} \\ s_2 = 7.5081 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 200 \text{ kPa} \\ T_3 = 80°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@80°\text{C}} = 335.02 \text{ kJ/kg} \\ s_3 \cong s_{f@80°\text{C}} = 1.0756 \text{ kJ/kg}\cdot\text{K} \end{array}$$



Analysis (a) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}h_1 + \dot{m}_2 h_2 = \dot{Q}_{\text{out}} + \dot{m}_3 h_3$$

Combining the two relations gives $\dot{Q}_{\text{out}} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2)h_3 = \dot{m}_1(h_1 - h_3) + \dot{m}_2(h_2 - h_3)$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of the superheated steam is determined to be

$$\dot{m}_2 = \frac{\dot{Q}_{\text{out}} - \dot{m}_1(h_1 - h_3)}{h_2 - h_3} = \frac{(600/60 \text{ kJ/s}) - (4 \text{ kg/s})(62.98 - 335.02)\text{kJ/kg}}{(2870.4 - 335.02)\text{kJ/kg}} = \mathbf{0.429\ kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 4 + 0.429 = 4.429 \text{ kg/s}$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation S_{gen} is determined from an entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \ \rightarrow \dot{S}_{\text{gen}} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{\text{out}}}{T_0}$$

Substituting, the exergy destruction is determined to be

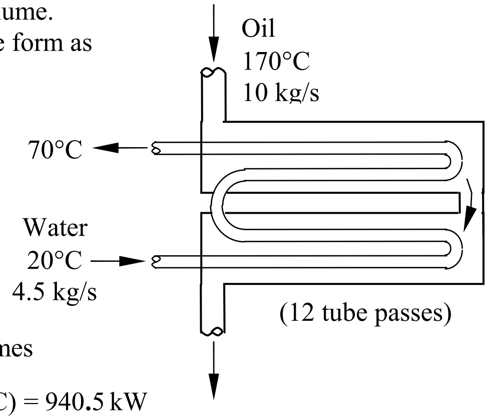
$$\begin{aligned} \dot{X}_{\text{destroyed}} &= T_0 \dot{S}_{\text{gen}} = T_0\left(\dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{out}}{T_{b,surr}}\right) \\ &= (298 \text{ K})(4.429 \times 1.0756 - 0.429 \times 7.5081 - 4 \times 0.22447 + 10/298)\text{kW/K} \\ &= \mathbf{202\ kW} \end{aligned}$$

8-88 Water is heated by hot oil in a heat exchanger. The outlet temperature of the oil and the rate of exergy destruction within the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} = (4.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70\text{°C} - 20\text{°C}) = 940.5 \text{ kW}$$

Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{oil} \rightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170\text{°C} - \frac{940.5 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg.°C})} = \mathbf{129.1°C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{water} s_1 + \dot{m}_{oil} s_3 - \dot{m}_{water} s_2 - \dot{m}_{oil} s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{water}(s_2 - s_1) + \dot{m}_{oil}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{gen} = \dot{m}_{water} c_p \ln\frac{T_2}{T_1} + \dot{m}_{oil} c_p \ln\frac{T_4}{T_3}$$

$$= (4.5 \text{ kg/s})(4.18 \text{ kJ/kg.K}) \ln\frac{70 + 273}{20 + 273} + (10 \text{ kg/s})(2.3 \text{ kJ/kg.K}) \ln\frac{129.1 + 273}{170 + 273} = 0.736 \text{ kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$,

$$\dot{X}_{destroyed} = T_0 \dot{S}_{gen} = (298 \text{ K})(0.736 \text{ kW/K}) = \mathbf{219 \text{ kW}}$$

8-101 Hot exhaust gases leaving an internal combustion engine is to be used to obtain saturated steam in an adiabatic heat exchanger. The rate at which the steam is obtained, the rate of exergy destruction, and the second-law efficiency are to be determined.

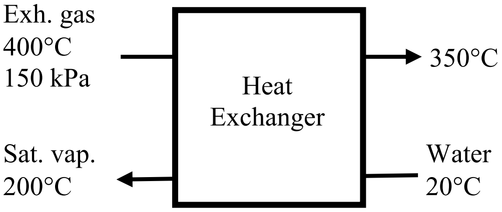
Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air properties are used for exhaust gases. **4** Pressure drops in the heat exchanger are negligible.

Properties The gas constant of air is R = 0.287 kJkg.K. The specific heat of air at the average temperature of exhaust gases (650 K) is c_p = 1.063 kJ/kg.K (Table A-2).

Analysis (*a*) We denote the inlet and exit states of exhaust gases by (1) and (2) and that of the water by (3) and (4). The properties of water are (Table A-4)

$$\left.\begin{array}{l} T_3 = 20°\text{C} \\ x_3 = 0 \end{array}\right\} \begin{array}{l} h_3 = 83.91 \text{ kJ/kg} \\ s_3 = 0.29649 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_4 = 200°\text{C} \\ x_4 = 1 \end{array}\right\} \begin{array}{l} h_4 = 2792.0 \text{ kJ/kg} \\ s_4 = 6.4302 \text{ kJ/kg.K} \end{array}$$



An energy balance on the heat exchanger gives

$$\dot{m}_a h_1 + \dot{m}_w h_3 = \dot{m}_a h_2 + \dot{m}_w h_4$$

$$\dot{m}_a c_p (T_1 - T_2) = \dot{m}_w (h_4 - h_3)$$

$$(0.8 \text{ kg/s})(1.063 \text{ kJ/kg°C})(400 - 350)°\text{C} = \dot{m}_w (2792.0 - 83.91)\text{kJ/kg}$$

$$\dot{m}_w = \mathbf{0.01570 \text{ kg/s}}$$

(*b*) The specific exergy changes of each stream as it flows in the heat exchanger is

$$\Delta s_a = c_p \ln\frac{T_2}{T_1} = (0.8 \text{ kg/s})(1.063 \text{ kJ/kg.K})\ln\frac{(350+273)\text{ K}}{(400+273)\text{ K}} = -0.08206 \text{ kJ/kg.K}$$

$$\begin{aligned} \Delta\psi_a &= c_p (T_2 - T_1) - T_0 \Delta s_a \\ &= (1.063 \text{ kJ/kg.°C})(350 - 400)°\text{C} - (20 + 273 \text{ K})(-0.08206 \text{ kJ/kg.K}) \\ &= -29.106 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} \Delta\psi_w &= h_4 - h_3 - T_0 (s_4 - s_3) \\ &= (2792.0 - 83.91)\text{kJ/kg} - (20 + 273 \text{ K})(6.4302 - 0.29649)\text{kJ/kg.K} \\ &= 910.913 \text{ kJ/kg} \end{aligned}$$

The exergy destruction is determined from an exergy balance on the heat exchanger to be

$$-\dot{X}_{\text{dest}} = \dot{m}_a \Delta\psi_a + \dot{m}_w \Delta\psi_w = (0.8 \text{ kg/s})(-29.106 \text{ kJ/kg}) + (0.01570 \text{ kg/s})(910.913) \text{ kJ/kg} = -8.98 \text{ kW}$$

or

$$\dot{X}_{\text{dest}} = \mathbf{8.98 \text{ kW}}$$

(*c*) The second-law efficiency for a heat exchanger may be defined as the exergy increase of the cold fluid divided by the exergy decrease of the hot fluid. That is,

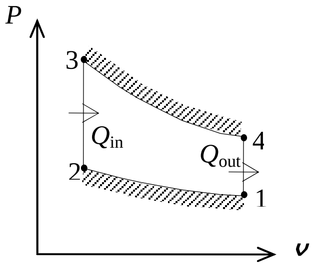
$$\eta_{\text{II}} = \frac{\dot{m}_w \Delta\psi_w}{-\dot{m}_a \Delta\psi_a} = \frac{(0.01570 \text{ kg/s})(910.913 \text{ kJ/kg})}{-(0.8 \text{ kg/s})(-29.106 \text{ kJ/kg})} = \mathbf{0.614}$$

9-39 An ideal Otto cycle with air as the working fluid has a compression ratio of 9.5. The highest pressure and temperature in the cycle, the amount of heat transferred, the thermal efficiency, and the mean effective pressure are to be determined.

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

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



$$T_2 = T_1\left(\frac{\upsilon_1}{\upsilon_2}\right)^{k-1} = (308\text{ K})(9.5)^{0.4} = 757.9\text{ K}$$

$$\frac{P_2\upsilon_2}{T_2} = \frac{P_1\upsilon_1}{T_1} \longrightarrow P_2 = \frac{\upsilon_1}{\upsilon_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{757.9\text{ K}}{308\text{ K}}\right)(100\text{ kPa}) = 2338\text{ kPa}$$

Process 3-4: isentropic expansion.

$$T_3 = T_4\left(\frac{\upsilon_4}{\upsilon_3}\right)^{k-1} = (800\text{ K})(9.5)^{0.4} = \mathbf{1969\ K}$$

Process 2-3: υ = constant heat addition.

$$\frac{P_3\upsilon_3}{T_3} = \frac{P_2\upsilon_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1969\text{ K}}{757.9\text{ K}}\right)(2338\text{ kPa}) = \mathbf{6072\ kPa}$$

(*b*)

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

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

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

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

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

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

(*d*)

$$V_{\min} = V_2 = \frac{V_{\max}}{r}$$

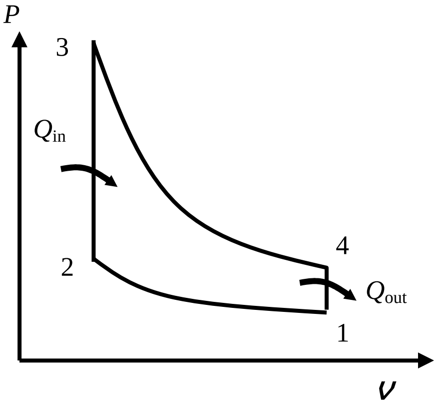
$$\text{MEP} = \frac{W_{\text{net,out}}}{V_1 - V_2} = \frac{W_{\text{net,out}}}{V_1(1 - 1/r)} = \frac{0.350\text{ kJ}}{(0.0006\text{ m}^3)(1 - 1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{652\ kPa}$$

9-43 A gasoline engine operates on an Otto cycle. The compression and expansion processes are modeled as polytropic. The temperature at the end of expansion process, the net work output, the thermal efficiency, the mean effective pressure, the engine speed for a given net power, and the specific fuel consumption are to be determined.

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

Properties The properties of air at 850 K are c_p = 1.110 kJ/kg·K, c_v = 0.823 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.349 (Table A-2b).

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



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

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

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

Process 2-3: constant volume heat addition

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

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

Process 3-4: polytropic expansion.

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

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

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

Process 4-1: constant volume heat rejection.

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

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

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

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

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

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

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

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

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

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

The total mass contained in the cylinder is

$$m_t=\frac{P_1V_1}{RT_1}=\frac{(100\,\text{kPa})/0.00176\,\text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(310\ \text{K})}=0.001978\,\text{kg}$$

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

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

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

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

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

Finally, the specific fuel consumption is

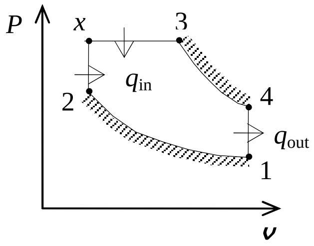
$$\text{sfc}=\frac{m_f}{m_t w_{\text{net}}}=\frac{0.0001164\,\text{kg}}{(0.001978\,\text{kg})(794\,\text{kJ/kg})}\left(\frac{1000\ \text{g}}{1\,\text{kg}}\right)\left(\frac{3600\ \text{kJ}}{1\,\text{kWh}}\right)=\mathbf{267\ g/kWh}$$

9-53 An ideal dual cycle has a compression ratio of 14 and cutoff ratio of 1.2. The thermal efficiency, amount of heat added, and the maximum gas pressure and temperature are to be determined.

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

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

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



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

and the specific volume at the end of the compression is

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

The pressure at the end of the compression is

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

and the maximum pressure is

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

The temperature at the end of the compression is

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

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

From the definition of cutoff ratio

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

The remaining state temperatures are then

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

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

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

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

The heat rejected is

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

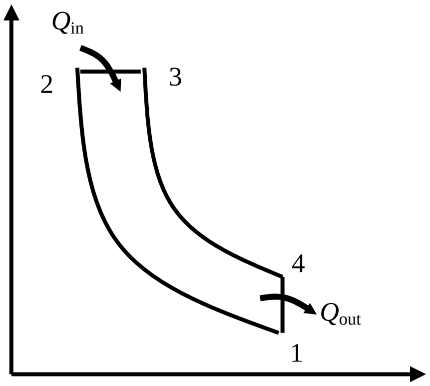
Then, $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{197.1\,\text{kJ/kg}}{556.5\,\text{kJ/kg}} = \mathbf{0.646}$$

9-65 A six-cylinder compression ignition engine operates on the ideal Diesel cycle. The maximum temperature in the cycle, the cutoff ratio, the net work output per cycle, the thermal efficiency, the mean effective pressure, the net power output, and the specific fuel consumption are to be determined.

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

Properties The properties of air at 850 K are c_p = 1.110 kJ/kg·K, c_v = 0.823 kJ/kg·K, R = 0.287 kJ/kg·K, and k = 1.349 (Table A-2b).

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



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

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

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

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

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

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

The total mass contained in the cylinder is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95\,\text{kPa})(0.003378\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(340\,\text{K})} = 0.003288\,\text{kg}$$

The mass of fuel burned during one cycle is

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

Process 2-3: constant pressure heat addition

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

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

The cutoff ratio is

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

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

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

$$V_4 = V_1$$

$$P_3 = P_2$$

Process 3-4: isentropic expansion.

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

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

Process 4-1: constant voume heat rejection.

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

The net work output and the thermal efficiency are

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

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

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

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

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

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

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

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

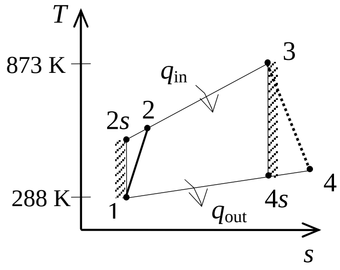
$$\mathrm{sfc} = \frac{m_f}{W_{\mathrm{net}}} = \frac{0.0001134\,\mathrm{kg}}{2.710\,\mathrm{kJ/kg}}\left(\frac{1000\,\mathrm{g}}{1\,\mathrm{kg}}\right)\left(\frac{3600\,\mathrm{kJ}}{1\,\mathrm{kWh}}\right) = \mathbf{151\,g/kWh}$$

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

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

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

Analysis For the compression process,



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

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

For the expansion process,

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

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

The isentropic and actual work of compressor and turbine are

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

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

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

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

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

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

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

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

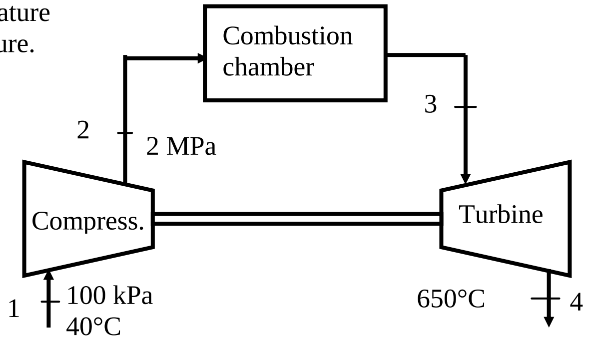
The two results are almost identical.

9-100 A gas-turbine plant operates on the simple Brayton cycle. The net power output, the back work ratio, and the thermal efficiency are to be determined.

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

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

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



Process 1-2: Compression

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

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

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

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

Process 3-4: Expansion

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

$$\eta_\text{T} = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.88 = \frac{h_3 - 959.2}{h_3 - h_{4s}}$$

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

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

The mass flow rate is determined from

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

The net power output is

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

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

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

(*b*) The back work ratio is

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

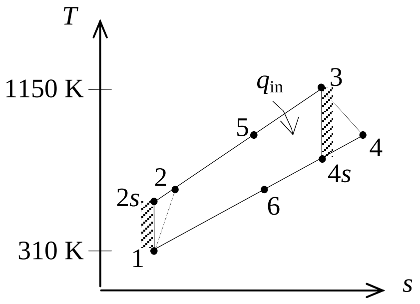
(*c*) The rate of heat input and the thermal efficiency are

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

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

9-116 A Brayton cycle with regeneration using air as the working fluid is considered. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

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



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

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

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

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

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

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

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

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

Thus,

$$T_4 = \mathbf{782.8\text{ K}}$$

(*b*)

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

(*c*)

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

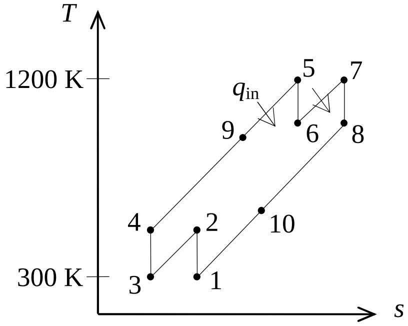
Then,

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

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

9-129 An ideal gas-turbine cycle with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency of the cycle are to be determined for the cases of with and without a regenerator.

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



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

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

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

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

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

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

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

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

Thus,

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{222.14\text{ kJ/kg}}{662.86\text{ kJ/kg}} = \mathbf{33.5\%}$$

$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6) = (1277.79 - 411.26) + (1277.79 - 946.36) = 1197.96\text{ kJ/kg}$$

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

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{440.72\text{ kJ/kg}}{1197.96\text{ kJ/kg}} = \mathbf{36.8\%}$$

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

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

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

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

9-141 A turbofan engine operating on an ideal cycle produces 50,000 N of thrust. The air temperature at the fan outlet needed to produce this thrust is to be determined.

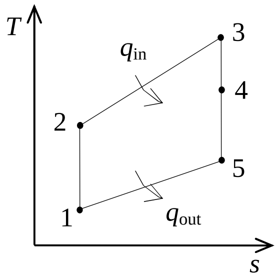
Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

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

Analysis The total mass flow rate is

$$\boldsymbol{v}_1 = \frac{RT}{P} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3)(253\ \text{K})}{50\ \text{kPa}} = 1.452\ \text{m}^3/\text{kg}$$

$$\dot m = \frac{AV_1}{\boldsymbol{v}_1} = \frac{\pi D^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi(2.5\ \text{m})^2}{4}\frac{200\ \text{m/s}}{1.452\ \text{m}^3/\text{kg}} = 676.1\ \text{kg/s}$$



Now,

$$\dot m_e = \frac{\dot m}{8} = \frac{676.1\ \text{kg/s}}{8} = 84.51\ \text{kg/s}$$

The mass flow rate through the fan is

$$\dot m_f = \dot m - \dot m_e = 676.1 - 84.51 = 591.6\ \text{kg/s}$$

In order to produce the specified thrust force, the velocity at the fan exit will be

$$F = \dot m_f (V_{\text{exit}} - V_{\text{inlet}})$$

$$V_{\text{exit}} = V_{\text{inlet}} + \frac{F}{\dot m_f} = (200\ \text{m/s}) + \frac{50{,}000\ \text{N}}{591.6\ \text{kg/s}}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 284.5\ \text{m/s}$$

An energy balance on the stream passing through the fan gives

$$c_p(T_4 - T_5) = \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2}$$

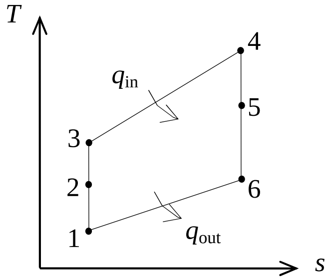
$$\begin{aligned} T_5 &= T_4 - \frac{V_{\text{exit}}^2 - V_{\text{inlet}}^2}{2c_p} \\ &= 253\ \text{K} - \frac{(284.5\ \text{m/s})^2 - (200\ \text{m/s})^2}{2(1.005\ \text{kJ/kg}\cdot\text{K})}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) \\ &= \mathbf{232.6\ K} \end{aligned}$$

9-142 A pure jet engine operating on an ideal cycle is considered. The velocity at the nozzle exit and the thrust produced are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** The turbine work output is equal to the compressor work input.

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

Analysis (*a*) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 240 m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).



Diffuser:

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

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

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

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 260\text{ K} + \frac{(240\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 288.7\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (45\text{ kPa})\left(\frac{288.7\text{ K}}{260\text{ K}}\right)^{1.4/0.4} = 64.88\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (13)(64.88\text{ kPa}) = 843.5\text{ kPa}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (288.7\text{ K})(13)^{0.4/1.4} = 600.7\text{ K}$$

Turbine:

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

or $\quad T_5 = T_4 - T_3 + T_2 = 830 - 600.7 + 288.7 = 518.0\text{ K}$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (830\text{ K})\left(\frac{45\text{ kPa}}{843.5\text{ kPa}}\right)^{0.4/1.4} = 359.3\text{ K}$$

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

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

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

or $\quad V_6 = V_{exit} = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(518.0 - 359.3)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{564.8\ m/s}$

The mass flow rate through the engine is

$$\boldsymbol{v}_1 = \frac{RT}{P} = \frac{(0.287\text{ kPa}\cdot\text{m}^3)(260\text{ K})}{45\text{ kPa}} = 1.658\text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{AV_1}{\boldsymbol{v}_1} = \frac{\pi D^2}{4}\frac{V_1}{\boldsymbol{v}_1} = \frac{\pi(1.6\text{ m})^2}{4}\frac{240\text{ m/s}}{1.658\text{ m}^3/\text{kg}} = 291.0\text{ kg/s}$$

The thrust force generated is then

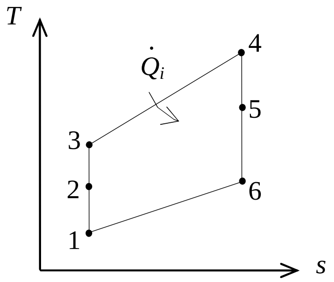
$$F = \dot{m}(V_{\text{exit}} - V_{\text{inlet}}) = (291.0\text{ kg/s})(564.8 - 240)\text{m/s}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{94{,}520\ N}$$

9-143 A turbojet aircraft flying at an altitude of 9150 m is operating on the ideal jet propulsion cycle. The velocity of exhaust gases, the propulsive power developed, and the rate of fuel consumption are to be determined.

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

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

Analysis (*a*) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of V_1 = 320 m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).



Diffuser:

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

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

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

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

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

Compressor:

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

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

Turbine:

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

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

Nozzle:

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

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

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

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

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

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

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

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

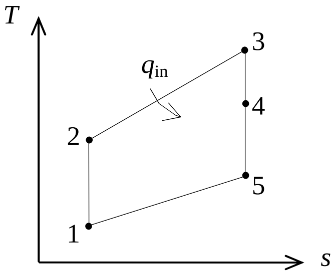
9-145 A turbojet aircraft that has a pressure rate of 9 is stationary on the ground. The force that must be applied on the brakes to hold the plane stationary is to be determined.

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

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

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

Compressor:



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

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = (9)(1.2311) = 11.08 \longrightarrow h_2 = 544.07\text{ kJ/kg}$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{fuel}} \times \text{HV} = (0.5\text{ kg/s})(42{,}700\text{ kJ/kg}) = 21{,}350\text{ kJ/s}$$

$$q_{\text{in}} = \frac{\dot{Q}_{\text{in}}}{\dot{m}} = \frac{21{,}350\text{ kJ/s}}{20\text{ kg/s}} = 1067.5\text{ kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 \longrightarrow h_3 = h_2 + q_{in} = 544.07 + 1067.5 = 1611.6\text{ kJ/kg} \longrightarrow P_{r_3} = 568.5$$

Turbine:

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

or

$$h_4 = h_3 - h_2 + h_1 = 1611.6 - 544.07 + 290.16 = 1357.7\text{ kJ/kg}$$

Nozzle:

$$P_{r_5} = P_{r_3}\left(\frac{P_5}{P_3}\right) = (568.5)\left(\frac{1}{9}\right) = 63.17 \longrightarrow h_5 = 888.56\text{ kJ/kg}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$h_4 + V_4^2/2 = h_5 + V_5^2/2$$
$$0 = h_5 - h_4 + \frac{V_5^2 - V_4^{2\,\nearrow 0}}{2}$$

or

$$V_5 = \sqrt{2(h_4 - h_5)} = \sqrt{(2)(1357.7 - 888.56)\text{kJ/kg}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 968.6\text{ m/s}$$

$$\text{Brake force} = \text{Thrust} = \dot{m}(V_{\text{exit}} - V_{\text{inlet}}) = (20\text{ kg/s})(968.6 - 0)\text{m/s}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{19{,}370\ N}$$

9-154 The total exergy destruction associated with the Brayton cycle described in Prob. 9-116 and the exergy at the exhaust gases at the turbine exit are to be determined.

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

Analysis From Prob. 9-116, q_{in} = 480.82, q_{out} = 372.73 kJ/kg, and

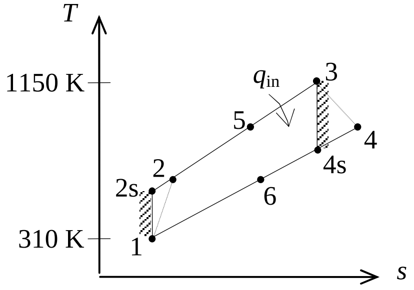
$$T_1 = 310 \text{ K} \longrightarrow s_1^\circ = 1.73498 \text{ kJ/kg}\cdot\text{K}$$

$$h_2 = 618.26 \text{ kJ/kg} \longrightarrow s_2^\circ = 2.42763 \text{ kJ/kg}\cdot\text{K}$$

$$T_3 = 1150 \text{ K} \longrightarrow s_3^\circ = 3.12900 \text{ kJ/kg}\cdot\text{K}$$

$$h_4 = 803.14 \text{ kJ/kg} \longrightarrow s_4^\circ = 2.69407 \text{ kJ/kg}\cdot\text{K}$$

$$h_5 = 738.43 \text{ kJ/kg} \longrightarrow s_5^\circ = 2.60815 \text{ kJ/kg}\cdot\text{K}$$



and, from an energy balance on the heat exchanger,

$$h_5 - h_2 = h_4 - h_6 \longrightarrow h_6 = 803.14 - (738.43 - 618.26) = 682.97 \text{ kJ/kg}$$

$$\longrightarrow s_6^\circ = 2.52861 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$x_{\text{destroyed},12} = T_0 s_{\text{gen},12} = T_0(s_2 - s_1) = T_0\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right)$$

$$= (290 \text{ K})(2.42763 - 1.73498 - (0.287 \text{ kJ/kg}\cdot\text{K})\ln(7)) = \mathbf{38.91 \text{ kJ/kg}}$$

$$x_{\text{destroyed},34} = T_0 s_{\text{gen},34} = T_0(s_4 - s_3) = T_0\left(s_4^\circ - s_3^\circ - R\ln\frac{P_4}{P_3}\right)$$

$$= (290 \text{ K})(2.69407 - 3.12900 - (0.287\text{kJ/kg}\cdot\text{K})\ln(1/7)) = \mathbf{35.83 \text{ kJ/kg}}$$

$$x_{\text{destroyed, regen}} = T_0 s_{\text{gen,regen}} = T_0[(s_5 - s_2) + (s_6 - s_4)] = T_0[(s_5^\circ - s_2^\circ) + (s_6^\circ - s_4^\circ)]$$

$$= (290 \text{ K})(2.60815 - 2.42763 + 2.52861 - 2.69407) = \mathbf{4.37 \text{ kJ/kg}}$$

$$x_{\text{destroyed},53} = T_0 s_{\text{gen},53} = T_0\left(s_3 - s_5 - \frac{q_{R,53}}{T_R}\right) = T_0\left(s_3^\circ - s_5^\circ - \cancelto{0}{R\ln\frac{P_3}{P_5}} - \frac{q_{in}}{T_H}\right)$$

$$= (290 \text{ K})\left(3.12900 - 2.60815 - \frac{480.82 \text{ kJ/kg}}{1500 \text{ K}}\right) = \mathbf{58.09 \text{ kJ/kg}}$$

$$x_{\text{destroyed},61} = T_0 s_{\text{gen},61} = T_0\left(s_1 - s_6 + \frac{q_{R,61}}{T_R}\right) = T_0\left(s_1^\circ - s_6^\circ - \cancelto{0}{R\ln\frac{P_1}{P_6}} + \frac{q_{\text{out}}}{T_L}\right)$$

$$= (290 \text{ K})\left(1.73498 - 2.52861 + \frac{372.73 \text{ kJ/kg}}{290 \text{ K}}\right) = \mathbf{142.6 \text{ kJ/kg}}$$

Noting that $h_0 = h_{@\,290\text{ K}}$ = 290.16 kJ/kg and $T_0 = 290 \text{ K} \longrightarrow s_1^\circ = 1.66802 \text{ kJ/kg}\cdot\text{K}$, the stream exergy at the exit of the regenerator (state 6) is determined from

$$\phi_6 = (h_6 - h_0) - T_0(s_6 - s_0) + \cancelto{0}{\frac{V_6^2}{2}} + \cancelto{0}{gz_6}$$

where

$$s_6 - s_0 = s_6 - s_1 = s_6^\circ - s_1^\circ - \cancelto{0}{R\ln\frac{P_6}{P_1}} = 2.52861 - 1.66802 = 0.86059 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\phi_6 = 682.97 - 290.16 - (290 \text{ K})(0.86059 \text{ kJ/kg}\cdot\text{K}) = \mathbf{143.2 \text{ kJ/kg}}$$

9-179 A gas-turbine plant operates on the regenerative Brayton cycle with reheating and intercooling. The back work ratio, the net work output, the thermal efficiency, the second-law efficiency, and the exergies at the exits of the combustion chamber and the regenerator are to be determined.

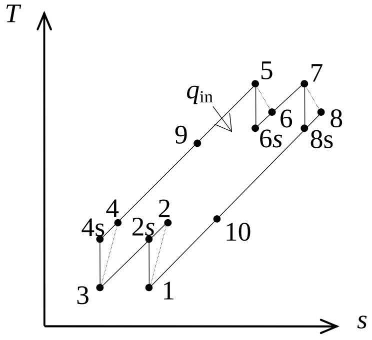
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

Properties The gas constant of air is R = 0.287 kJ/kg·K.

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

Optimum intercooling and reheating pressure is

$$P_2 = \sqrt{P_1 P_4} = \sqrt{(100)(1200)} = 346.4\text{ kPa}$$



Process 1-2, 3-4: Compression

$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.43\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 300\text{ K} \\ P_1 = 100\text{ kPa} \end{array}\right\} s_1 = 5.7054\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 346.4\text{ kPa} \\ s_2 = s_1 = 5.7054\text{ kJ/kg.K} \end{array}\right\} h_{2s} = 428.79\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.80 = \frac{428.79 - 300.43}{h_2 - 300.43} \longrightarrow h_2 = 460.88\text{ kJ/kg}$$

$$T_3 = 350\text{ K} \longrightarrow h_3 = 350.78\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_3 = 350\text{ K} \\ P_3 = 346.4\text{ kPa} \end{array}\right\} s_3 = 5.5040\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_4 = 1200\text{ kPa} \\ s_4 = s_3 = 5.5040\text{ kJ/kg.K} \end{array}\right\} h_{4s} = 500.42\text{ kJ/kg}$$

$$\eta_C = \frac{h_{4s} - h_3}{h_4 - h_3} \longrightarrow 0.80 = \frac{500.42 - 350.78}{h_4 - 350.78} \longrightarrow h_4 = 537.83\text{ kJ/kg}$$

Process 6-7, 8-9: Expansion

$$T_6 = 1400\text{ K} \longrightarrow h_6 = 1514.9\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_6 = 1400\text{ K} \\ P_6 = 1200\text{ kPa} \end{array}\right\} s_6 = 6.6514\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_7 = 346.4\text{ kPa} \\ s_7 = s_6 = 6.6514\text{ kJ/kg.K} \end{array}\right\} h_{7s} = 1083.9\text{ kJ/kg}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} \longrightarrow 0.80 = \frac{1514.9 - h_7}{1514.9 - 1083.9} \longrightarrow h_7 = 1170.1\text{ kJ/kg}$$

$$T_8 = 1300\text{ K} \longrightarrow h_8 = 1395.6\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_8 = 1300\text{ K} \\ P_8 = 346.4\text{ kPa} \end{array}\right\} s_8 = 6.9196\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_9 = 100\text{ kPa} \\ s_9 = s_8 = 6.9196\text{ kJ/kg.K} \end{array}\right\} h_{9s} = 996.00\text{ kJ/kg}$$

$$\eta_T = \frac{h_8 - h_9}{h_8 - h_{9s}} \longrightarrow 0.80 = \frac{1395.6 - h_9}{1395.6 - 996.00} \longrightarrow h_9 = 1075.9 \text{ kJ/kg}$$

Cycle analysis:

$$w_{\text{C,in}} = h_2 - h_1 + h_4 - h_3 = 460.88 - 300.43 + 537.83 - 350.78 = 347.50 \text{ kJ/kg}$$

$$w_{\text{T,out}} = h_6 - h_7 + h_8 - h_9 = 1514.9 - 1170.1 + 1395.6 - 1075.9 = 664.50 \text{ kJ/kg}$$

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{347.50}{664.50} = \mathbf{0.523}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 664.50 - 347.50 = \mathbf{317.0 \ kJ/kg}$$

Regenerator analysis:

$$\varepsilon_{\text{regen}} = \frac{h_9 - h_{10}}{h_9 - h_4} \longrightarrow 0.75 = \frac{1075.9 - h_{10}}{1075.9 - 537.83} \longrightarrow h_{10} = 672.36 \text{ kJ/kg}$$

$$\left.\begin{aligned} h_{10} &= 672.36 \text{ K} \\ P_{10} &= 100 \text{ kPa} \end{aligned}\right\} s_{10} = 6.5157 \text{ kJ/kg} \cdot \text{K}$$

$$q_{\text{regen}} = h_9 - h_{10} = h_5 - h_4 \longrightarrow 1075.9 - 672.36 = h_5 - 537.83 \longrightarrow h_5 = 941.40 \text{ kJ/kg}$$

(*b*) $q_{\text{in}} = h_6 - h_5 = 1514.9 - 941.40 = 573.54 \text{ kJ/kg}$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{317.0}{573.54} = \mathbf{0.553}$$

(*c*) The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). The maximum temperature for the cycle can be taken to be the turbine inlet temperature. That is,

$$\eta_{\max} = 1 - \frac{T_1}{T_6} = 1 - \frac{300 \text{ K}}{1400 \text{ K}} = 0.786$$

and

$$\eta_{II} = \frac{\eta_{th}}{\eta_{\max}} = \frac{0.553}{0.786} = \mathbf{0.704}$$

(*d*) The exergies at the combustion chamber exit and the regenerator exit are

$$\begin{aligned} x_6 &= h_6 - h_0 - T_0(s_6 - s_0) \\ &= (1514.9 - 300.43)\text{kJ/kg} - (300 \text{ K})(6.6514 - 5.7054)\text{kJ/kg.K} \\ &= \mathbf{930.7 \ kJ/kg} \end{aligned}$$

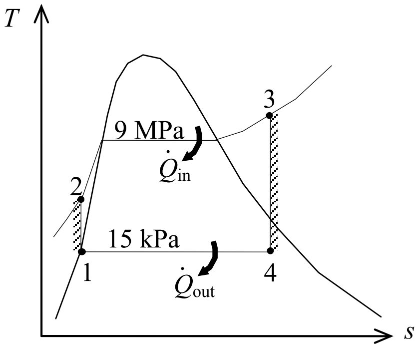
$$\begin{aligned} x_{10} &= h_{10} - h_0 - T_0(s_{10} - s_0) \\ &= (672.36 - 300.43)\text{kJ/kg} - (300 \text{ K})(6.5157 - 5.7054)\text{kJ/kg.K} \\ &= \mathbf{128.8 \ kJ/kg} \end{aligned}$$

10-26 A 120-MW coal-fired steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The overall plant efficiency and the required rate of the coal supply are to be determined.

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

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

$$h_1 = h_{f@\,15\text{ kPa}} = 225.94\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,15\text{ kPa}} = 0.0010140\text{ m}^3\text{/kg}$$



$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001014\text{ m}^3\text{/kg})(9000 - 15\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 9.11\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 225.94 + 9.11 = 235.05\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 9\text{ MPa} \\ T_3 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3512.0\text{ kJ/kg} \\ s_3 = 6.8164\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 15\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8164 - 0.7549}{7.2522} = 0.8358$$
$$h_4 = h_f + x_4 h_{fg} = 225.94 + (0.8358)(2372.4) = 2208.8\text{ kJ/kg}$$

The thermal efficiency is determined from

$$q_\text{in} = h_3 - h_2 = 3512.0 - 235.05 = 3276.9\text{ kJ/kg}$$
$$q_\text{out} = h_4 - h_1 = 2208.8 - 225.94 = 1982.9\text{ kJ/kg}$$

and

$$\eta_\text{th} = 1 - \frac{q_\text{out}}{q_\text{in}} = 1 - \frac{1982.9}{3276.9} = 0.3949$$

Thus,

$$\eta_\text{overall} = \eta_\text{th} \times \eta_\text{comb} \times \eta_\text{gen} = (0.3949)(0.75)(0.96) = 0.2843 = \mathbf{28.4\%}$$

(*b*) Then the required rate of coal supply becomes

$$\dot{Q}_\text{in} = \frac{\dot{W}_\text{net}}{\eta_\text{overall}} = \frac{120{,}000\ \text{kJ/s}}{0.2843} = 422{,}050\ \text{kJ/s}$$

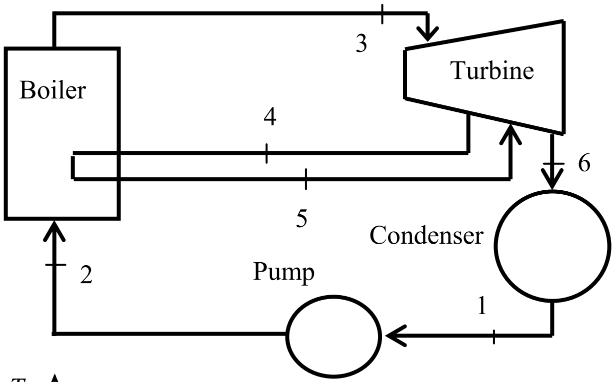
and

$$\dot{m}_\text{coal} = \frac{\dot{Q}_\text{in}}{C_\text{coal}} = \frac{422{,}050\ \text{kJ/s}}{29{,}300\ \text{kJ/kg}} = 14.404\ \text{kg/s} = \mathbf{51.9\ tons/h}$$

10-41 A steam power plant that operates on a reheat Rankine cycle is considered. The condenser pressure, the net power output, and the thermal efficiency 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),



$$\left.\begin{array}{l} P_3 = 12.5\text{ MPa} \\ T_3 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3476.5\text{ kJ/kg} \\ s_3 = 6.6317\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 2\text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} h_{4s} = 2948.1\text{ kJ/kg}$$

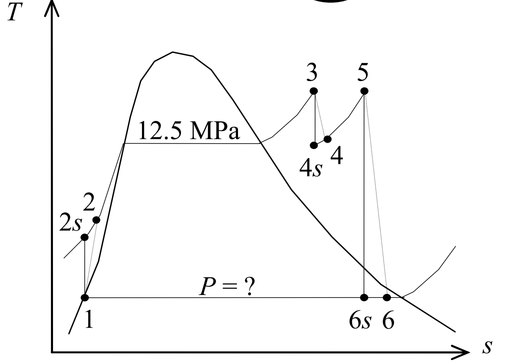
$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$\begin{aligned} \rightarrow h_4 &= h_3 - \eta_T\left(h_3 - h_{4s}\right) \\ &= 3476.5 - (0.85)(3476.5 - 2948.1) \\ &= 3027.3\text{ kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_5 = 2\text{ MPa} \\ T_5 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3358.2\text{ kJ/kg} \\ s_5 = 7.2815\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = ? \\ x_6 = 0.95 \end{array}\right\} h_6 = \quad \text{(Eq. 1)}$$

$$\left.\begin{array}{l} P_6 = ? \\ s_6 = s_5 \end{array}\right\} h_{6s} = \quad \text{(Eq. 2)}$$



$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T\left(h_5 - h_{6s}\right) = 3358.2 - (0.85)\left(3358.2 - h_{6s}\right) \quad \text{(Eq. 3)}$$

The pressure at state 6 may be determined by a trial-error approach from the steam tables or by using EES from the above three equations:

$P_6 =$ **9.73 kPa**, $h_6 = 2463.3$ kJ/kg,

(*b*) Then,

$$\begin{aligned} h_1 &= h_{f@\,9.73\text{ kPa}} = 189.57\text{ kJ/kg} \\ v_1 &= v_{f@\,10\text{ kPa}} = 0.001010\text{ m}^3\text{/kg} \\ w_{p,\text{in}} &= v_1\left(P_2 - P_1\right)/\eta_p \\ &= \left(0.00101\text{ m}^3\text{/kg}\right)\left(12{,}500 - 9.73\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/(0.90) \\ &= 14.02\text{ kJ/kg} \\ h_2 &= h_1 + w_{p,\text{in}} = 189.57 + 14.02 = 203.59\text{ kJ/kg} \end{aligned}$$

Cycle analysis:

$$q_{\text{in}} = \left(h_3 - h_2\right) + \left(h_5 - h_4\right) = 3476.5 - 203.59 + 3358.2 - 2463.3 = 3603.8\text{ kJ/kg}$$

$$q_{\text{out}} = h_6 - h_1 = 2463.3 - 189.57 = 2273.7\text{ kJ/kg}$$

$$\dot{W}_{\text{net}} = \dot{m}(q_{\text{in}} - q_{\text{out}}) = (7.7\text{ kg/s})(3603.8 - 2273.7)\text{kJ/kg} = \mathbf{10{,}242\text{ kW}}$$

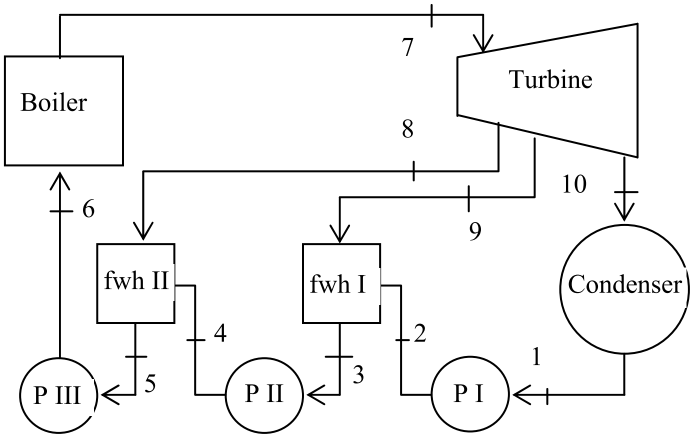
(*c*) The thermal efficiency is

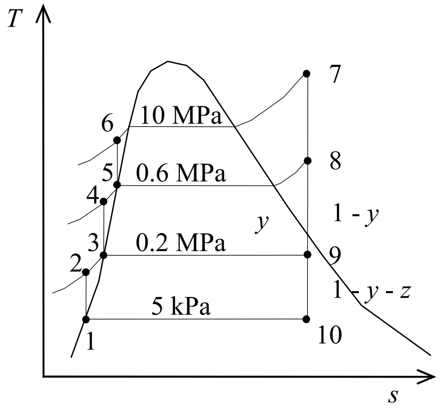
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2273.7\text{ kJ/kg}}{3603.8\text{ kJ/kg}} = 0.369 = \mathbf{36.9\%}$$

10-52 A steam power plant operates on an ideal regenerative Rankine cycle with two open feedwater heaters. The net power output of the power plant and the thermal efficiency of the cycle are to be determined.

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

Analysis





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

$$h_1 = h_{f @ 5\text{ kPa}} = 137.75 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f @ 5\text{ kPa}} = 0.001005 \text{ m}^3/\text{kg}$$

$$w_{pI,\text{in}} = \boldsymbol{v}_1(P_2 - P_1) = (0.001005 \text{ m}^3/\text{kg})(200 - 5 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = 0.20 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 137.75 + 0.20 = 137.95 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.2 \text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f @ 0.2\text{ MPa}} = 504.71 \text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f @ 0.2\text{ MPa}} = 0.001061 \text{ m}^3/\text{kg} \end{array}$$

$$w_{pII,\text{in}} = \boldsymbol{v}_3(P_4 - P_3) = (0.001061 \text{ m}^3/\text{kg})(600 - 200 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$
$$= 0.42 \text{ kJ/kg}$$

$$h_4 = h_3 + w_{pII,\text{in}} = 504.71 + 0.42 = 505.13 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 0.6 \text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_5 = h_{f @ 0.6\text{ MPa}} = 670.38 \text{ kJ/kg} \\ \boldsymbol{v}_5 = \boldsymbol{v}_{f @ 0.6\text{ MPa}} = 0.001101 \text{ m}^3/\text{kg} \end{array}$$

$$w_{pIII,\text{in}} = \boldsymbol{v}_5(P_6 - P_5) = (0.001101 \text{ m}^3/\text{kg})(10{,}000 - 600 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$
$$= 10.35 \text{ kJ/kg}$$

$$h_6 = h_5 + w_{pIII,\text{in}} = 670.38 + 10.35 = 680.73 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 10 \text{ MPa} \\ T_7 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3625.8 \text{ kJ/kg} \\ s_7 = 6.9045 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 0.6 \text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_8 = 2821.8 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 0.2 \text{ MPa} \\ s_9 = s_7 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{6.9045 - 1.5302}{5.5968} = 0.9602 \\ h_9 = h_f + x_9 h_{fg} = 504.71 + (0.9602)(2201.6) = 2618.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_{10} = 5 \text{ kPa} \\ s_{10} = s_7 \end{array}\right\} \begin{array}{l} x_{10} = \dfrac{s_{10} - s_f}{s_{fg}} = \dfrac{6.9045 - 0.4762}{7.9176} = 0.8119 \\ h_{10} = h_f + x_{10} h_{fg} = 137.75 + (0.8119)(2423.0) = 2105.0 \text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

FWH-2:

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

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_4 h_4 = \dot{m}_5 h_5 \longrightarrow y h_8 + (1 - y) h_4 = 1(h_5)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_5$). Solving for y,

$$y = \frac{h_5 - h_4}{h_8 - h_4} = \frac{670.38 - 505.13}{2821.8 - 505.13} = 0.07133$$

FWH-1:

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_9 h_9 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow z h_9 + (1 - y - z) h_2 = (1 - y) h_3$$

where z is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for z,

$$z = \frac{h_3 - h_2}{h_9 - h_2}(1 - y) = \frac{504.71 - 137.95}{2618.7 - 137.95}(1 - 0.07136) = 0.1373$$

Then,

$$q_{\text{in}} = h_7 - h_6 = 3625.8 - 680.73 = 2945.0 \text{ kJ/kg}$$

$$q_{\text{out}} = (1 - y - z)(h_{10} - h_1) = (1 - 0.07133 - 0.1373)(2105.0 - 137.75) = 1556.8 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2945.0 - 1556.8 = 1388.2 \text{ kJ/kg}$$

and

$$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (22 \text{ kg/s})(1388.2 \text{ kJ/kg}) = 30{,}540 \text{ kW} \cong \mathbf{30.5 \text{ MW}}$$

(*b*) The thermal efficiency is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1556.8 \text{ kJ/kg}}{2945.0 \text{ kJ/kg}} = \mathbf{47.1\%}$$

10-53 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.

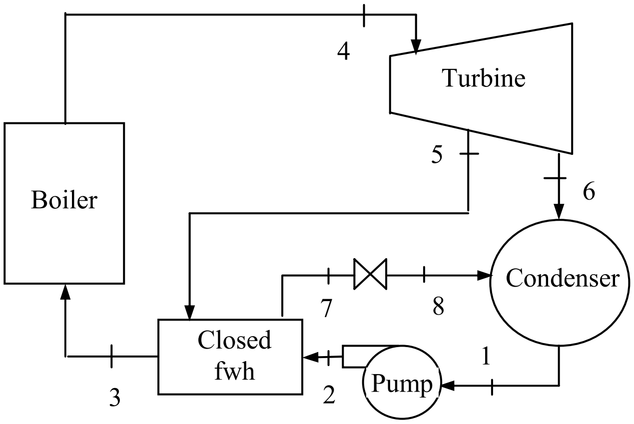
Analysis From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,20\,\text{kPa}} = 251.42\ \text{kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,20\,\text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{p,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017\ \text{m}^3/\text{kg})(3000-20)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 3.03\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 251.42 + 3.03 = 254.45\ \text{kJ/kg}$$

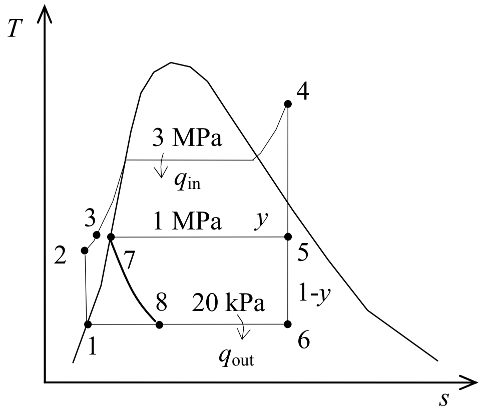


$$\left.\begin{aligned} P_4 &= 3000\ \text{kPa} \\ T_4 &= 350°\text{C} \end{aligned}\right\} \begin{aligned} h_4 &= 3116.1\ \text{kJ/kg} \\ s_4 &= 6.7450\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

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

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

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.



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

$$h_8 = h_7 = 762.51\ \text{kJ/kg}$$

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

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:

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

Rearranging,

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

Then,

$$w_{\text{T,out}} = h_4 - h_5 + (1-y)(h_5 - h_6) = 3116.1 - 2851.9 + (1-0.2437)(2851.9 - 2221.7) = \mathbf{740.9\ kJ/kg}$$

$$w_{\text{P,in}} = \mathbf{3.03\ kJ/kg}$$

$$q_{\text{in}} = h_4 - h_3 = 3116.1 - 763.53 = \mathbf{2353\ kJ/kg}$$

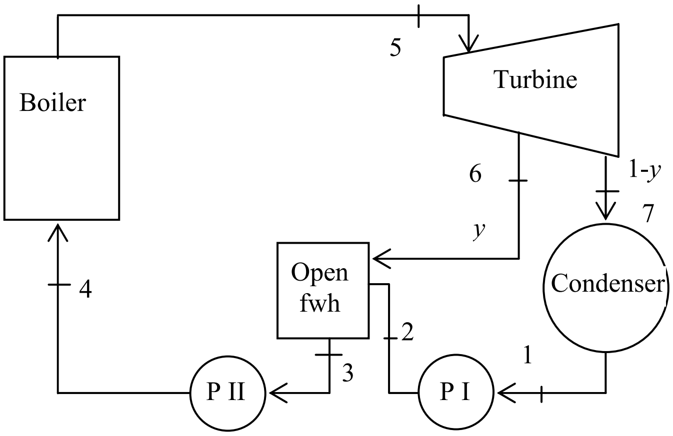
Also, $w_{\text{net}} = w_{\text{T,out}} - w_{\text{P,in}} = 740.9 - 3.03 = 737.8\ \text{kJ/kg}$

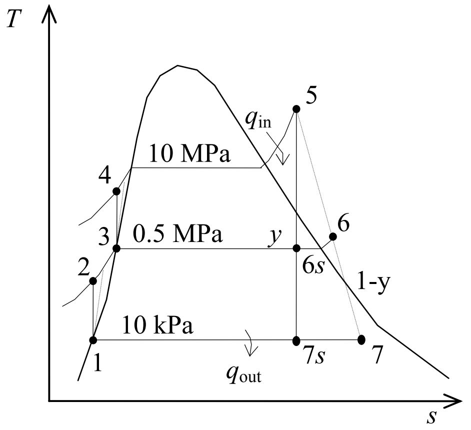
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{737.8}{2353} = 0.3136$$

10-100 An 150-MW steam power plant operating on a regenerative Rankine cycle with an open feedwater heater is considered. The mass flow rate of steam through the boiler, the thermal efficiency of the cycle, and the irreversibility associated with the regeneration process 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),

$$h_1 = h_{f\,@\,10\,\text{kPa}} = 191.81\ \text{kJ/kg}$$
$$v_1 = v_{f\,@\,10\,\text{kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= v_1(P_2 - P_1)/\eta_p \\ &= \left(0.00101\ \text{m}^3/\text{kg}\right)\left(500 - 10\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 0.52\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.52 = 192.33\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.5\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.5\,\text{MPa}} = 640.09\ \text{kJ/kg} \\ v_3 = v_{f\,@\,0.5\,\text{MPa}} = 0.001093\ \text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= v_3(P_4 - P_3)/\eta_p \\ &= \left(0.001093\ \text{m}^3/\text{kg}\right)\left(10{,}000 - 500\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 10.93\ \text{kJ/kg} \end{aligned}$$
$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.93 = 651.02\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\ \text{MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\ \text{kJ/kg} \\ s_5 = 6.5995\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$x_{6s} = \frac{s_{6s} - s_f}{s_{fg}} = \frac{6.5995 - 1.8604}{4.9603} = 0.9554$$
$$\left.\begin{array}{l} P_{6s} = 0.5\ \text{MPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{aligned} h_{6s} &= h_f + x_{6s}h_{fg} = 640.09 + (0.9554)(2108.0) \\ &= 2654.1\ \text{kJ/kg} \end{aligned}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow \begin{aligned} h_6 &= h_5 - \eta_T(h_5 - h_{6s}) \\ &= 3375.1 - (0.80)(3375.1 - 2654.1) \\ &= 2798.3\ \text{kJ/kg} \end{aligned}$$

$$x_{7s} = \frac{s_{7s} - s_f}{s_{fg}} = \frac{6.5995 - 0.6492}{7.4996} = 0.7934$$

$$\left.\begin{array}{l} P_{7s} = 10 \text{ kPa} \\ s_{7s} = s_5 \end{array}\right\} \begin{array}{l} h_{7s} = h_f + x_{7s} h_{fg} = 191.81 + (0.7934)(2392.1) \\ \quad = 2089.7 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_7}{h_5 - h_{7s}} \longrightarrow \begin{array}{l} h_7 = h_5 - \eta_T (h_5 - h_{7s}) \\ \quad = 3375.1 - (0.80)(3375.1 - 2089.7) \\ \quad = 2346.8 \text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

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

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1 - y) h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for y,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{640.09 - 192.33}{2798.3 - 192.33} = 0.1718$$

Then, $q_{\text{in}} = h_5 - h_4 = 3375.1 - 651.02 = 2724.1 \text{ kJ/kg}$

$$q_{\text{out}} = (1 - y)(h_7 - h_1) = (1 - 0.1718)(2346.8 - 191.81) = 1784.7 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2724.1 - 1784.7 = 939.4 \text{ kJ/kg}$$

and

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150{,}000 \text{ kJ/s}}{939.4 \text{ kJ/kg}} = \mathbf{159.7 \text{ kg/s}}$$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1784.7 \text{ kJ/kg}}{2724.1 \text{ kJ/kg}} = \mathbf{34.5\%}$$

Also,

$$\left.\begin{array}{l} P_6 = 0.5 \text{ MPa} \\ h_6 = 2798.3 \text{ kJ/kg} \end{array}\right\} s_6 = 6.9453 \text{ kJ/kg} \cdot \text{K}$$

$$s_3 = s_{f\,@\,0.5\text{ MPa}} = 1.8604 \text{ kJ/kg} \cdot \text{K}$$

$$s_2 = s_1 = s_{f\,@\,10\text{ kPa}} = 0.6492 \text{ kJ/kg} \cdot \text{K}$$

Then the irreversibility (or exergy destruction) associated with this regeneration process is

$$i_{\text{regen}} = T_0 s_{\text{gen}} = T_0 \left(\sum m_e s_e - \sum m_i s_i + \frac{q_{\text{surr}}}{T_L}^{\nearrow 0} \right) = T_0 \left[s_3 - y s_6 - (1 - y) s_2 \right]$$

$$= (303 \text{ K})[1.8604 - (0.1718)(6.9453) - (1 - 0.1718)(0.6492)]$$

$$= \mathbf{39.25 \text{ kJ/kg}}$$

11-26 A vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of cooling, the power input, and the COP are to be determined. Also, the same parameters are to be determined if the cycle operated on the ideal vapor-compression refrigeration cycle between the same pressure limits.

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

Analysis (a) From the refrigerant-134a tables (Tables A-11 through A-13)

$$T_{\text{sat@200 kPa}} = -10.1°\text{C}$$

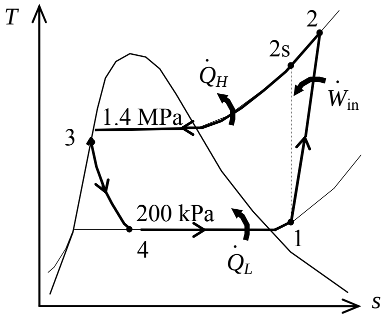
$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = -10.1 + 10.1 = 0°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 253.05\text{ kJ/kg} \\ s_1 = 0.9698\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1400\text{ kPa} \\ s_1 = s_1 \end{array}\right\} h_{2s} = 295.90\text{ kJ/kg}$$

$$T_{\text{sat@1400 kPa}} = 52.4°\text{C}$$

$$\left.\begin{array}{l} P_3 = 1400\text{ kPa} \\ T_3 = 52.4 - 4.4 = 48°\text{C} \end{array}\right\} h_3 \cong h_{f@48°\text{C}} = 120.39\text{ kJ/kg}$$

$$h_4 = h_3 = 120.39\text{ kJ/kg}$$



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

$$0.88 = \frac{295.90 - 253.05}{h_2 - 253.05} \longrightarrow h_2 = 301.74\text{ kJ/kg}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.025\text{ kg/s})(253.05 - 120.39) = \mathbf{3.317\ kW}$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.025\text{ kg/s})(301.74 - 120.39) = \mathbf{4.534\ kW}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.025\text{ kg/s})(301.74 - 253.05) = \mathbf{1.217\ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{3.317\text{ kW}}{1.217\text{ kW}} = \mathbf{2.725}$$

(b) Ideal vapor-compression refrigeration cycle solution

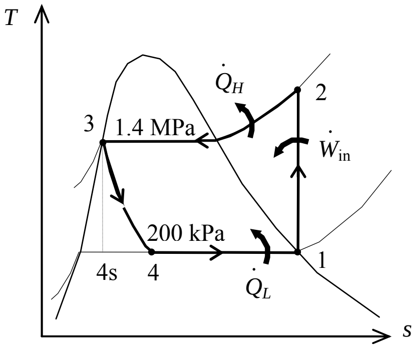
From the refrigerant-134a tables (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 244.46\text{ kJ/kg} \\ s_1 = 0.9377\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1400\text{ kPa} \\ s_1 = s_1 \end{array}\right\} h_2 = 285.08\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1400\text{ kPa} \\ x_3 = 0 \end{array}\right\} h_3 = 127.22\text{ kJ/kg}$$

$$h_4 = h_3 = 127.22\text{ kJ/kg}$$



$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.025\text{ kg/s})(244.46 - 127.22) = \mathbf{3.931kW}$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.025\text{ kg/s})(285.08 - 127.22) = \mathbf{3.947\ kW}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.025\text{ kg/s})(285.08 - 244.46) = \mathbf{1.016\ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{3.931\text{ kW}}{1.016\text{ kW}} = \mathbf{2.886}$$

Discussion The cooling load increases by 18.5% while the COP increases by 5.9% when the cycle operates on the ideal vapor-compression cycle.

11-35 An ideal vapor-compression refrigeration cycle is used to keep a space at a low temperature. The cooling load, the COP, the exergy destruction in each component, the total exergy destruction, and the second-law efficiency are to be determined.

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

Analysis (a) The properties of R-134a are (Tables A-11 through A-13)

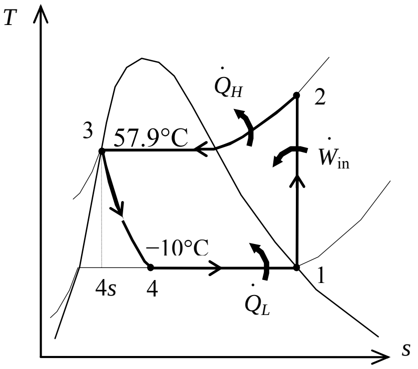
$$\left.\begin{array}{l} T_1 = -10°\text{C} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 244.51 \text{ kJ/kg} \\ s_1 = 0.9377 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = P_{\text{sat@57.9°C}} = 1600 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 287.85 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1600 \text{ kPa} \\ x_3 = 0 \end{array}\right\} \begin{array}{l} h_3 = 135.93 \text{ kJ/kg} \\ s_3 = 0.4791 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$h_4 = h_3 = 135.93 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_4 = -10°\text{C} \\ h_4 = 135.93 \text{ kJ/kg} \end{array}\right\} s_4 = 0.5251 \text{ kJ/kg}\cdot\text{K}$$



The energy interactions in the components and the COP are

$$q_L = h_1 - h_4 = 244.51 - 135.93 = \mathbf{108.6 \text{ kJ/kg}}$$

$$q_H = h_2 - h_3 = 287.85 - 135.93 = 151.9 \text{ kJ/kg}$$

$$w_{\text{in}} = h_2 - h_1 = 287.85 - 244.51 = 43.33 \text{ kJ/kg}$$

$$\text{COP} = \frac{q_L}{w_{\text{in}}} = \frac{108.6 \text{ kJ/kg}}{43.33 \text{ kJ/kg}} = \mathbf{2.506}$$

(b) The exergy destruction in each component of the cycle is determined as follows

Compressor:

$$s_{\text{gen},1-2} = s_2 - s_1 = 0$$

$$Ex_{\text{dest,1-2}} = T_0 s_{\text{gen,1-2}} = \mathbf{0}$$

Condenser:

$$s_{\text{gen},2-3} = s_3 - s_2 + \frac{q_H}{T_H} = (0.4791 - 0.9377) \text{ kJ/kg}\cdot\text{K} + \frac{151.9 \text{ kJ/kg}}{298 \text{ K}} = 0.05124 \text{ kJ/kg}\cdot\text{K}$$

$$Ex_{\text{dest,2-3}} = T_0 s_{\text{gen,2-3}} = (298 \text{ K})(0.05124 \text{ kJ/kg}\cdot\text{K}) = \mathbf{15.27 \text{ kJ/kg}}$$

Expansion valve:

$$s_{\text{gen},3-4} = s_4 - s_3 = 0.5251 - 0.4791 = 0.04595 \text{ kJ/kg}\cdot\text{K}$$

$$Ex_{\text{dest,3-4}} = T_0 s_{\text{gen,3-4}} = (298 \text{ K})(0.04595 \text{ kJ/kg}\cdot\text{K}) = \mathbf{13.69 \text{ kJ/kg}}$$

Evaporator:

$$s_{\text{gen},4-1} = s_1 - s_4 - \frac{q_L}{T_L} = (0.9377 - 0.5251) \text{ kJ/kg}\cdot\text{K} - \frac{108.6 \text{ kJ/kg}}{278 \text{ K}} = 0.02201 \text{ kJ/kg}\cdot\text{K}$$

$$Ex_{\text{dest,4-1}} = T_0 s_{\text{gen,4-1}} = (298 \text{ K})(0.02201 \text{ kJ/kg}\cdot\text{K}) = \mathbf{6.56 \text{ kJ/kg}}$$

The total exergy destruction can be determined by adding exergy destructions in each component:

$$\begin{aligned} \dot{E}x_{\text{dest,total}} &= \dot{E}x_{\text{dest,1-2}} + \dot{E}x_{\text{dest,2-3}} + \dot{E}x_{\text{dest,3-4}} + \dot{E}x_{\text{dest,4-1}} \\ &= 0 + 15.27 + 13.69 + 6.56 = \mathbf{35.52 \text{ kJ/kg}} \end{aligned}$$

(c) The exergy of the heat transferred from the low-temperature medium is

$$Ex_{q_L} = -q_L\left(1-\frac{T_0}{T_L}\right) = -(108.6\text{ kJ/kg})\left(1-\frac{298}{278}\right) = 7.812\text{ kJ/kg}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{Ex_{q_L}}{w_{\text{in}}} = \frac{7.812}{43.33} = 0.1803 = \mathbf{18.0\%}$$

The total exergy destruction in the cycle can also be determined from

$$Ex_{\text{dest,total}} = w_{\text{in}} - Ex_{q_L} = 43.33 - 7.812 = 35.52\text{ kJ/kg}$$

The result is identical as expected.

The second-law efficiency of the compressor is determined from

$$\eta_{\text{II,Comp}} = \frac{\dot{X}_{\text{recovered}}}{\dot{X}_{\text{expended}}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{act, in}}} = \frac{\dot{m}[h_2 - h_1 - T_0(s_2 - s_1)]}{\dot{m}(h_2 - h_1)}$$

since the compression through the compressor is isentropic ($s_2 = s_1$), the second-law efficiency is

$$\eta_{\text{II,Comp}} = 1 = \mathbf{100\%}$$

The second-law efficiency of the evaporator is determined from

$$\eta_{\text{II, Evap}} = \frac{\dot{X}_{\text{recovered}}}{\dot{X}_{\text{expended}}} = \frac{\dot{Q}_L(T_0 - T_L)/T_L}{\dot{m}[h_4 - h_1 - T_0(s_4 - s_1)]} = 1 - \frac{\dot{X}_{\text{dest,4-1}}}{\dot{X}_4 - \dot{X}_1}$$

where

$$\begin{aligned} x_4 - x_1 &= h_4 - h_1 - T_0(s_4 - s_1) \\ &= (135.93 - 244.51)\text{ kJ/kg} - (298\text{ K})(0.5251 - 0.9377)\text{ kJ/kg}\cdot\text{K} \\ &= 14.37\text{ kJ/kg} \end{aligned}$$

Substituting,

$$\eta_{\text{II, Evap}} = 1 - \frac{x_{\text{dest,4-1}}}{x_4 - x_1} = 1 - \frac{6.56\text{ kJ/kg}}{14.37\text{ kJ/kg}} = 0.544 = \mathbf{54.4\%}$$

11-64 A two-stage cascade refrigeration system is considered. Each stage operates on the ideal vapor-compression cycle with upper cycle using water and lower cycle using refrigerant-134a as the working fluids. The mass flow rate of R-134a and water in their respective cycles and the overall COP of this system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The heat exchanger is adiabatic.

Analysis From the water and refrigerant tables (Tables A-4, A-5, A-6, A-11, A-12, and A-13),

$$\left.\begin{array}{l} T_1 = 5^\circ\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,5^\circ\text{C}} = 2510.1\ \text{kJ/kg} \\ s_1 = s_{g\,@\,5^\circ\text{C}} = 9.0249\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.6\ \text{MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 5083.4\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.6\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,1.6\ \text{MPa}} = 858.44\ \text{kJ/kg}$$

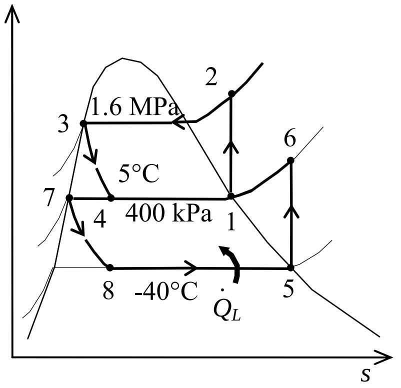
$$h_4 \cong h_3 = 858.44\ \text{kJ/kg} \quad \text{(throttling)}$$

$$\left.\begin{array}{l} T_5 = -40^\circ\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_5 = h_{g\,@\,-40^\circ\text{C}} = 225.86\ \text{kJ/kg} \\ s_5 = s_{g\,@\,-40^\circ\text{C}} = 0.96866\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 400\ \text{kPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 267.59\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 400\ \text{kPa} \\ \text{sat. liquid} \end{array}\right\} h_7 = h_{f\,@\,400\ \text{kPa}} = 63.94\ \text{kJ/kg}$$

$$h_8 \cong h_7 = 63.94\ \text{kJ/kg} \quad \text{(throttling)}$$



The mass flow rate of R-134a is determined from

$$\dot{Q}_L = \dot{m}_R(h_5 - h_8) \longrightarrow \dot{m}_R = \frac{\dot{Q}_L}{h_5 - h_8} = \frac{20\ \text{kJ/s}}{(225.86 - 63.94)\ \text{kJ/kg}} = \mathbf{0.1235\ kg/s}$$

An energy balance on the heat exchanger gives the mass flow rate of water

$$\dot{m}_R(h_6 - h_7) = \dot{m}_w(h_1 - h_4)$$

$$\longrightarrow \dot{m}_w = \dot{m}_R \frac{h_6 - h_7}{h_1 - h_4} = (0.1235\ \text{kg/s})\frac{267.59 - 63.94}{2510.1 - 858.44} = \mathbf{0.01523\ kg/s}$$

The total power input to the compressors is

$$\begin{aligned} \dot{W}_\text{in} &= \dot{m}_R(h_6 - h_5) + \dot{m}_w(h_2 - h_1) \\ &= (0.1235\ \text{kg/s})(267.59 - 225.86)\ \text{kJ/kg} + (0.01523\ \text{kg/s})(5083.4 - 2510.1)\ \text{kJ/kg} \\ &= 44.35\ \text{kJ/s} \end{aligned}$$

The COP of this refrigeration system is determined from its definition,

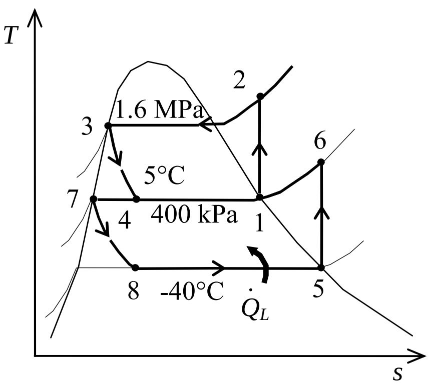
$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_\text{in}} = \frac{20\ \text{kJ/s}}{44.35\ \text{kJ/s}} = \mathbf{0.451}$$

11-65 A two-stage vapor-compression refrigeration system with refrigerant-134a as the working fluid is considered. The process with the greatest exergy destruction is to be determined.

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

Analysis From Prob. 11-55 and the water and refrigerant tables (Tables A-4, A-5, A-6, A-11, A-12, and A-13),

$$\begin{aligned}
s_1 &= s_2 = 9.0249 \text{ kJ/kg}\cdot\text{K}\\
s_3 &= 2.3435 \text{ kJ/kg}\cdot\text{K}\\
s_4 &= 3.0869 \text{ kJ/kg}\cdot\text{K}\\
s_5 &= s_6 = 0.96866 \text{ kJ/kg}\cdot\text{K}\\
s_7 &= 0.24757 \text{ kJ/kg}\cdot\text{K}\\
s_8 &= 0.27423 \text{ kJ/kg}\cdot\text{K}\\
\dot{m}_R &= 0.1235 \text{ kg/s}\\
\dot{m}_w &= 0.01523 \text{ kg/s}\\
q_L &= h_5 - h_8 = 161.92 \text{ kJ/kg}\\
q_H &= h_2 - h_3 = 4225.0 \text{ kJ/kg}\\
T_L &= -30°\text{C} = 243 \text{ K}\\
T_H &= 30°\text{C} = 303 \text{ K}\\
T_0 &= 30°\text{C} = 303 \text{ K}
\end{aligned}$$



The exergy destruction during a process of a stream from an inlet state to exit state is given by

$$x_{\text{dest}} = T_0 s_{\text{gen}} = T_0\left(s_e - s_i - \frac{q_{\text{in}}}{T_{\text{source}}} + \frac{q_{\text{out}}}{T_{\text{sink}}}\right)$$

Application of this equation for each process of the cycle gives

$$\begin{aligned}
\dot{X}_{\text{destroyed},23} &= \dot{m}_w T_0\left(s_3 - s_2 + \frac{q_H}{T_H}\right)\\
&= (0.01523)(303\text{ K})\left(2.3435 - 9.0249 + \frac{4225.0}{303}\right) = 33.52 \text{ kJ/s}\\
\dot{X}_{\text{destroyed},34} &= \dot{m}_\text{w} T_0 (s_4 - s_3) = (0.01523)(303)(3.0869 - 2.3435) = 3.43 \text{ kJ/s}\\
\dot{X}_{\text{destroyed},78} &= \dot{m}_\text{R} T_0 (s_8 - s_7) = (0.1235)(303)(0.27423 - 0.24757) = 0.996 \text{ kJ/s}\\
\dot{X}_{\text{destroyed},85} &= \dot{m}_\text{R} T_0\left(s_5 - s_8 - \frac{q_L}{T_L}\right)\\
&= (0.1235)(303)\left(0.96866 - 0.27423 - \frac{161.92}{243}\right) = 1.05 \text{ kJ/s}\\
\dot{X}_{\text{destroyed, heat exch}} &= T_0\left[\dot{m}_\text{w}(s_1 - s_4) + \dot{m}_\text{R}(s_7 - s_6)\right]\\
&= (303)\left[(0.01523)(9.0249 - 3.0869) + (0.1235)(0.24757 - 0.96866)\right] = 0.417 \text{ kJ/s}
\end{aligned}$$

For isentropic processes, the exergy destruction is zero:

$$\dot{X}_{\text{destroyed},12} = 0$$

$$\dot{X}_{\text{destroyed},56} = 0$$

Note that heat is absorbed from a reservoir at -30°C (243 K) and rejected to a reservoir at 30°C (303 K), which is also taken as the dead state temperature. Alternatively, one may use the standard 25°C (298 K) as the dead state temperature, and perform the calculations accordingly. The greatest exergy destruction occurs in the condenser.

11-116 An air conditioner operates on the vapor-compression refrigeration cycle. The rate of cooling provided to the space, the COP, the isentropic efficiency and the exergetic efficiency of the compressor, the exergy destruction in each component of the cycle, the total exergy destruction, the minimum power input, and the second-law efficiency of the cycle are to be determined.

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

Analysis (a) The properties of R-134a are (Tables A-11 through A-13)

$$T_{\text{sat@180 kPa}} = -12.7°\text{C}$$

$$\left.\begin{array}{l} P_1 = 180\text{ kPa} \\ T_1 = -12.7 + 2.7 = 10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 245.14\text{ kJ/kg} \\ s_1 = 0.9483\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1200\text{ kPa} \\ s_1 = s_1 \end{array}\right\} h_{2s} = 285.32\text{ kJ/kg}$$

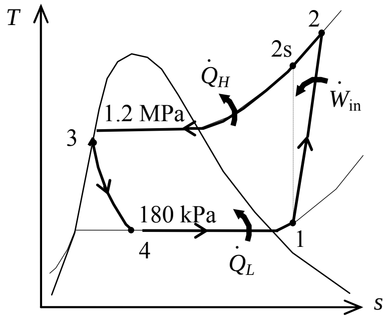
$$\left.\begin{array}{l} P_2 = 1200\text{ kPa} \\ T_2 = 60°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 289.64\text{ kJ/kg} \\ s_2 = 0.9614\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_{\text{sat@1200 kPa}} = 46.3°\text{C}$$

$$\left.\begin{array}{l} P_3 = 1200\text{ kPa} \\ T_3 = 46.3 - 6.3 = 40°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@40°\text{C}} = 108.26\text{ kJ/kg} \\ s_3 \cong s_{f@40°\text{C}} = 0.3948\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$h_4 = h_3 = 108.26\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 180\text{ kPa} \\ h_4 = 108.26\text{ kJ/kg} \end{array}\right\} s_4 = 0.4228\text{ kJ/kg}\cdot\text{K}$$



The cooling load and the COP are

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.06\text{ kg/s})(245.14 - 108.26)\text{kJ/kg} = 8.213\text{ kW}$$

$$= (8.213\text{ kW})\left(\frac{3412\text{ Btu/h}}{1\text{ kW}}\right) = \mathbf{28{,}020\ Btu/h}$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.06\text{ kg/s})(289.64 - 108.26)\text{kJ/kg} = 10.88\text{ kW}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.06\text{ kg/s})(289.64 - 245.14)\text{kJ/kg} = 2.670\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{8.213\text{ kW}}{2.670\text{ kW}} = \mathbf{3.076}$$

(b) The isentropic efficiency of the compressor is

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{285.32 - 245.14}{289.64 - 245.14} = 0.9029 = \mathbf{90.3\%}$$

The reversible power and the exergy efficiency for the compressor are

$$\begin{aligned} \dot{W}_{rev} &= \dot{m}\left[(h_2 - h_1) - T_0(s_2 - s_1)\right] \\ &= (0.06\text{ kg/s})\left[(289.64 - 245.14)\text{kJ/kg} - (310\text{ K})(0.9614 - 0.9483)\text{kJ/kg}\cdot\text{K}\right] \\ &= 2.428\text{ kW} \end{aligned}$$

$$\eta_{ex,C} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{in}}} = \frac{2.428\text{ kW}}{2.670\text{ kW}} = 0.9091 = \mathbf{90.9\%}$$

(c) The exergy destruction in each component of the cycle is determined as follows

Compressor:

$$\dot{S}_{\text{gen},1-2} = \dot{m}(s_2 - s_1) = (0.06\text{ kg/s})(0.9614 - 0.9483)\text{ kJ/kg}\cdot\text{K} = 0.0007827\text{ kW/K}$$

$$\dot{E}x_{\text{dest,1-2}} = T_0\dot{S}_{\text{gen,1-2}} = (310\text{ K})(0.0007827\text{ kW/K}) = \mathbf{0.2426\ kW}$$

Condenser:

$$\dot{S}_{\text{gen,2-3}} = \dot{m}(s_3 - s_2) + \frac{\dot{Q}_H}{T_H} = (0.06 \text{ kg/s})(0.3948 - 0.9614) \text{ kJ/kg}\cdot\text{K} + \frac{10.88 \text{ kW}}{310 \text{ K}} = 0.001114 \text{ kW/K}$$

$$\dot{E}x_{\text{dest,2-3}} = T_0 \dot{S}_{\text{gen,2-3}} = (310 \text{ K})(0.001114 \text{ kJ/kg}\cdot\text{K}) = \mathbf{0.3452\ kW}$$

Expansion valve:

$$\dot{S}_{\text{gen,3-4}} = \dot{m}(s_4 - s_3) = (0.06 \text{ kg/s})(0.4228 - 00.3948) \text{ kJ/kg}\cdot\text{K} = 0.001678 \text{ kW/K}$$

$$\dot{E}x_{\text{dest,3-4}} = T_0 \dot{S}_{\text{gen,3-4}} = (310 \text{ K})(0.001678 \text{ kJ/kg}\cdot\text{K}) = \mathbf{0.5203\ kJ/kg}$$

Evaporator:

$$\dot{S}_{\text{gen,4-1}} = \dot{m}(s_1 - s_4) - \frac{\dot{Q}_L}{T_L} = (0.06 \text{ kg/s})(0.9483 - 0.4228) \text{ kJ/kg}\cdot\text{K} - \frac{8.213 \text{ kW}}{294 \text{ K}} = 0.003597 \text{ kW/K}$$

$$\dot{E}x_{\text{dest,4-1}} = T_0 \dot{S}_{\text{gen,4-1}} = (310 \text{ K})(0.003597 \text{ kJ/kg}\cdot\text{K}) = \mathbf{1.115\ kW}$$

The total exergy destruction can be determined by adding exergy destructions in each component:

$$\begin{aligned}\dot{E}x_{\text{dest,total}} &= \dot{E}x_{\text{dest,1-2}} + \dot{E}x_{\text{dest,2-3}} + \dot{E}x_{\text{dest,3-4}} + \dot{E}x_{\text{dest,4-1}} \\ &= 0.2426 + 0.3452 + 0.5203 + 1.115 \\ &= \mathbf{2.223\ kW}\end{aligned}$$

(d) The exergy of the heat transferred from the low-temperature medium is

$$\dot{E}x_{\dot{Q}_L} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right) = -(8.213 \text{ kW})\left(1 - \frac{310}{294}\right) = 0.4470 \text{ kW}$$

This is the minimum power input to the cycle:

$$\dot{W}_{\text{in,min}} = \dot{E}x_{\dot{Q}_L} = \mathbf{0.4470\ kW}$$

The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{in,min}}}{\dot{W}_{\text{in}}} = \frac{0.4470}{2.670} = 0.1674 = \mathbf{16.7\%}$$

The total exergy destruction in the cycle can also be determined from

$$\dot{E}x_{\text{dest,total}} = \dot{W}_{\text{in}} - Ex_{\dot{Q}_L} = 2.670 - 0.4470 = 2.223 \text{ kW}$$

The result is the same as expected.

13-31 The partial pressures of a gas mixture are given. The mole fractions, the mass fractions, the mixture molar mass, the apparent gas constant, the constant-volume specific heat, and the specific heat ratio are to be determined.

Properties The molar masses of CO_2, O_2 and N_2 are 44.0, 32.0, and 28.0 kg/kmol, respectively (Table A-1). The constant-volume specific heats of these gases at 300 K are 0.657, 0.658, and 0.743 kJ/kg·K, respectively (Table A-2a).

Analysis The total pressure is

$$P_{\text{total}} = P_{\text{CO2}} + P_{\text{O2}} + P_{\text{N2}} = 12.5 + 37.5 + 50 = 100\ \text{kPa}$$

Partial pressures
CO_2, 12.5 kPa
O_2, 37.5 kPa
N_2, 50 kPa

The volume fractions are equal to the pressure fractions. Then,

$$y_{\text{CO2}} = \frac{P_{\text{CO2}}}{P_{\text{total}}} = \frac{12.5}{100} = \mathbf{0.125}$$

$$y_{\text{O2}} = \frac{P_{\text{O2}}}{P_{\text{total}}} = \frac{37.5}{100} = \mathbf{0.375}$$

$$y_{\text{N2}} = \frac{P_{\text{N2}}}{P_{\text{total}}} = \frac{50}{100} = \mathbf{0.50}$$

We consider 100 kmol of this mixture. Then the mass of each component are

$$m_{\text{CO2}} = N_{\text{CO2}} M_{\text{CO2}} = (12.5\ \text{kmol})(44\ \text{kg/kmol}) = 550\ \text{kg}$$
$$m_{\text{O2}} = N_{\text{O2}} M_{\text{O2}} = (37.5\ \text{kmol})(32\ \text{kg/kmol}) = 1200\ \text{kg}$$
$$m_{\text{N2}} = N_{\text{N2}} M_{\text{N2}} = (50\ \text{kmol})(28\ \text{kg/kmol}) = 1400\ \text{kg}$$

The total mass is

$$m_m = m_{\text{N2}} + m_{\text{O2}} + m_{\text{Ar}} = 550 + 1200 + 1400 = 3150\ \text{kg}$$

Then the mass fractions are

$$\text{mf}_{\text{CO2}} = \frac{m_{\text{CO2}}}{m_m} = \frac{550\ \text{kg}}{3150\ \text{kg}} = \mathbf{0.1746}$$

$$\text{mf}_{\text{O2}} = \frac{m_{\text{O2}}}{m_m} = \frac{1200\ \text{kg}}{3150\ \text{kg}} = \mathbf{0.3810}$$

$$\text{mf}_{\text{N2}} = \frac{m_{\text{N2}}}{m_m} = \frac{1400\ \text{kg}}{3150\ \text{kg}} = \mathbf{0.4444}$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{3150\ \text{kg}}{100\ \text{kmol}} = \mathbf{31.50\ kg/kmol}$$

The constant-volume specific heat of the mixture is determined from

$$\begin{aligned} c_v &= \text{mf}_{\text{Co2}} c_{v,\text{CO2}} + \text{mf}_{\text{O2}} c_{v,\text{O2}} + \text{mf}_{\text{N2}} c_{v,\text{N2}} \\ &= 0.1746 \times 0.657 + 0.3810 \times 0.658 + 0.4444 \times 0.743 \\ &= \mathbf{0.6956\ kJ/kg \cdot K} \end{aligned}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314\ \text{kJ/kmol} \cdot \text{K}}{31.50\ \text{kg/kmol}} = \mathbf{0.2639\ kJ/kg \cdot K}$$

The constant-pressure specific heat of the mixture and the specific heat ratio are

$$c_p = c_v + R = 0.6956 + 0.2639 = \mathbf{0.9595\ kJ/kg \cdot K}$$

$$k = \frac{c_p}{c_v} = \frac{0.9595\ \text{kJ/kg} \cdot \text{K}}{0.6956\ \text{kJ/kg} \cdot \text{K}} = \mathbf{1.379}$$

13-34 The masses, temperatures, and pressures of two gases contained in two tanks connected to each other are given. The valve connecting the tanks is opened and the final temperature is measured. The volume of each tank and the final pressure are to be determined.

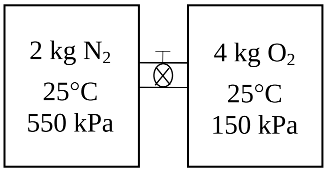
Assumptions Under specified conditions both N_2 and O_2 can be treated as ideal gases, and the mixture as an ideal gas mixture

Properties The molar masses of N_2 and O_2 are 28.0 and 32.0 kg/kmol, respectively. The gas constants of N_2 and O_2 are 0.2968 and 0.2598 kPa·m^3/kg·K, respectively (Table A-1).

Analysis The volumes of the tanks are

$$V_{N_2} = \left(\frac{mRT}{P}\right)_{N_2} = \frac{(2\text{ kg})(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})}{550\text{ kPa}} = \mathbf{0.322\ m^3}$$

$$V_{O_2} = \left(\frac{mRT}{P}\right)_{O_2} = \frac{(4\text{ kg})(0.2598\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})}{150\text{ kPa}} = \mathbf{2.065\ m^3}$$



$$V_{\text{total}} = V_{N_2} + V_{O_2} = 0.322\text{ m}^3 + 2.065\text{ m}^3 = 2.386\text{ m}^3$$

Also,

$$N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{2\text{ kg}}{28\text{ kg/kmol}} = 0.07143\text{ kmol}$$

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{4\text{ kg}}{32\text{ kg/kmol}} = 0.125\text{ kmol}$$

$$N_m = N_{N_2} + N_{O_2} = 0.07143\text{ kmol} + 0.125\text{ kmol} = 0.1964\text{ kmol}$$

Thus,

$$P_m = \left(\frac{NR_uT}{V}\right)_m = \frac{(0.1964\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(298\text{ K})}{2.386\text{ m}^3} = \mathbf{204\ kPa}$$

13-55 The volume fractions of components of a gas mixture are given. This mixture is expanded isentropically to a specified pressure. The work produced per unit mass of the mixture is to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

Properties The molar masses of H_2, He, and N_2 are 2.0, 4.0, and 28.0 kg/kmol, respectively (Table A-1). The constant-pressure specific heats of these gases at room temperature are 14.307, 5.1926, and 1.039 kJ/kg·K, respectively (Table A-2a).

Analysis We consider 100 kmol of this mixture. Noting that volume fractions are equal to the mole fractions, mass of each component are

$$\begin{aligned} m_{\text{H2}} &= N_{\text{H2}} M_{\text{H2}} = (30\ \text{kmol})(2\ \text{kg/kmol}) = 60\ \text{kg} \\ m_{\text{He}} &= N_{\text{He}} M_{\text{He}} = (40\ \text{kmol})(4\ \text{kg/kmol}) = 160\ \text{kg} \\ m_{\text{N2}} &= N_{\text{N2}} M_{\text{N2}} = (30\ \text{kmol})(28\ \text{kg/kmol}) = 840\ \text{kg} \end{aligned}$$

30% H_2
40% He
30% N_2
(by volume)
5 MPa, 600°C

The total mass is

$$m_m = m_{\text{H2}} + m_{\text{He}} + m_{\text{N2}} = 60 + 160 + 840 = 1060\ \text{kg}$$

Then the mass fractions are

$$\text{mf}_{\text{H2}} = \frac{m_{\text{H2}}}{m_m} = \frac{60\ \text{kg}}{1060\ \text{kg}} = 0.05660$$

$$\text{mf}_{\text{He}} = \frac{m_{\text{He}}}{m_m} = \frac{160\ \text{kg}}{1060\ \text{kg}} = 0.1509$$

$$\text{mf}_{\text{N2}} = \frac{m_{\text{N2}}}{m_m} = \frac{840\ \text{kg}}{1060\ \text{kg}} = 0.7925$$

The apparent molecular weight of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{1060\ \text{kg}}{100\ \text{kmol}} = 10.60\ \text{kg/kmol}$$

The constant-pressure specific heat of the mixture is determined from

$$\begin{aligned} c_p &= \text{mf}_{\text{H2}} c_{p,\text{H2}} + \text{mf}_{\text{He}} c_{p,\text{He}} + \text{mf}_{\text{N2}} c_{p,\text{N2}} \\ &= 0.05660 \times 14.307 + 0.1509 \times 5.1926 + 0.7925 \times 1.039 \\ &= 2.417\ \text{kJ/kg} \cdot \text{K} \end{aligned}$$

The apparent gas constant of the mixture is

$$R = \frac{R_u}{M_m} = \frac{8.314\ \text{kJ/kmol} \cdot \text{K}}{10.60\ \text{kg/kmol}} = 0.7843\ \text{kJ/kg} \cdot \text{K}$$

Then the constant-volume specific heat is

$$c_v = c_p - R = 2.417 - 0.7843 = 1.633\ \text{kJ/kg} \cdot \text{K}$$

The specific heat ratio is

$$k = \frac{c_p}{c_v} = \frac{2.417}{1.633} = 1.480$$

The temperature at the end of the expansion is

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{(k-1)/k} = (873\ \text{K}) \left(\frac{200\ \text{kPa}}{5000\ \text{kPa}} \right)^{0.48/1.48} = 307\ \text{K}$$

An energy balance on the adiabatic expansion process gives

$$w_{\text{out}} = c_p (T_1 - T_2) = (2.417\ \text{kJ/kg} \cdot \text{K})(873 - 307)\ \text{K} = \mathbf{1368\ kJ/kg}$$

13-56 The mass fractions of components of a gas mixture are given. This mixture is enclosed in a rigid, well-insulated vessel, and a paddle wheel in the vessel is turned until specified amount of work have been done on the mixture. The mixture's final pressure and temperature are to be determined.

Assumptions All gases will be modeled as ideal gases with constant specific heats.

Properties The molar masses of N_2, He, CH_4, and C_2H_6 are 28.0, 4.0, 16.0, and 30.0 kg/kmol, respectively (Table A-1). The constant-pressure specific heats of these gases at room temperature are 1.039, 5.1926, 2.2537, and 1.7662 kJ/kg·K, respectively (Table A-2a).

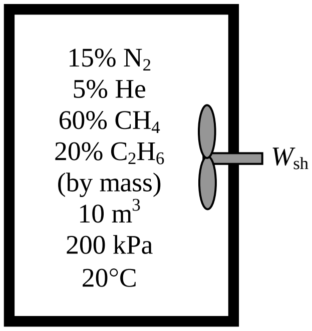
Analysis We consider 100 kg of this mixture. The mole numbers of each component are

$$N_{\mathrm{N2}}=\frac{m_{\mathrm{N2}}}{M_{\mathrm{N2}}}=\frac{15\ \mathrm{kg}}{28\ \mathrm{kg/kmol}}=0.5357\ \mathrm{kmol}$$

$$N_{\mathrm{He}}=\frac{m_{\mathrm{He}}}{M_{\mathrm{He}}}=\frac{5\ \mathrm{kg}}{4\ \mathrm{kg/kmol}}=1.25\ \mathrm{kmol}$$

$$N_{\mathrm{CH4}}=\frac{m_{\mathrm{CH4}}}{M_{\mathrm{CH4}}}=\frac{60\ \mathrm{kg}}{16\ \mathrm{kg/kmol}}=3.75\ \mathrm{kmol}$$

$$N_{\mathrm{C2H6}}=\frac{m_{\mathrm{C2H6}}}{M_{\mathrm{C2H6}}}=\frac{20\ \mathrm{kg}}{30\ \mathrm{kg/kmol}}=0.6667\ \mathrm{kmol}$$



The mole number of the mixture is

$$N_m=N_{\mathrm{N2}}+N_{\mathrm{He}}+N_{\mathrm{CH4}}+N_{\mathrm{C2H6}}=0.5357+1.25+3.75+0.6667=6.2024\ \mathrm{kmol}$$

The apparent molecular weight of the mixture is

$$M_m=\frac{m_m}{N_m}=\frac{100\ \mathrm{kg}}{6.2024\ \mathrm{kmol}}=16.12\ \mathrm{kg/kmol}$$

The constant-pressure specific heat of the mixture is determined from

$$\begin{aligned}c_p&=\mathrm{mf}_{\mathrm{N2}}c_{p,\mathrm{N2}}+\mathrm{mf}_{\mathrm{He}}c_{p,\mathrm{He}}+\mathrm{mf}_{\mathrm{CH4}}c_{p,\mathrm{CH4}}+\mathrm{mf}_{\mathrm{C2H6}}c_{p,\mathrm{C2H6}}\\&=0.15\times1.039+0.05\times5.1926+0.60\times2.2537+0.20\times1.7662\\&=2.121\ \mathrm{kJ/kg\cdot K}\end{aligned}$$

The apparent gas constant of the mixture is

$$R=\frac{R_u}{M_m}=\frac{8.134\ \mathrm{kJ/kmol\cdot K}}{16.12\ \mathrm{kg/kmol}}=0.5158\ \mathrm{kJ/kg\cdot K}$$

Then the constant-volume specific heat is

$$c_v=c_p-R=2.121-0.5158=1.605\ \mathrm{kJ/kg\cdot K}$$

The mass in the container is

$$m_m=\frac{P_1V_m}{RT_1}=\frac{(200\ \mathrm{kPa})(10\ \mathrm{m^3})}{(0.5158\ \mathrm{kPa\cdot m^3/kg\cdot K})(293\ \mathrm{K})}=13.23\ \mathrm{kg}$$

An energy balance on the system gives

$$W_{\mathrm{sh,in}}=m_mc_v(T_2-T_1)\longrightarrow T_2=T_1+\frac{W_{\mathrm{sh,in}}}{m_mc_v}=(293\ \mathrm{K})+\frac{100\ \mathrm{kJ}}{(13.23\ \mathrm{kg})(1.605\ \mathrm{kJ/kg\cdot K})}=\mathbf{297.7\ K}$$

Since the volume remains constant and this is an ideal gas,

$$P_2=P_1\frac{T_2}{T_1}=(200\ \mathrm{kPa})\frac{297.7\ \mathrm{K}}{293\ \mathrm{K}}=\mathbf{203.2\ kPa}$$

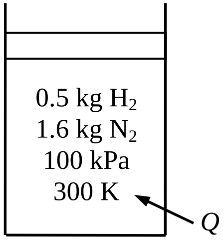
13-68 A piston-cylinder device contains a gas mixture at a given state. Heat is transferred to the mixture. The amount of heat transfer and the entropy change of the mixture are to be determined.

Assumptions **1** Under specified conditions both H_2 and N_2 can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** Kinetic and potential energy changes are negligible.

Properties The constant pressure specific heats of H_2 and N_2 at 450 K are 14.501 kJ/kg.K and 1.049 kJ/kg.K, respectively. (Table A-2b).

Analysis (*a*) Noting that $P_2 = P_1$ and $V_2 = 2V_1$,

$$\frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1} \longrightarrow T_2 = \frac{2V_1}{V_1} T_1 = 2T_1 = (2)(300\text{ K}) = 600\text{ K}$$



From the closed system energy balance relation,

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

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U \quad \to \quad Q_{\text{in}} = \Delta H$$

since W_b and ΔU combine into ΔH for quasi-equilibrium constant pressure processes.

$$\begin{aligned} Q_{\text{in}} &= \Delta H = \Delta H_{H_2} + \Delta H_{N_2} = \left[mc_{p,\text{avg}}(T_2 - T_1)\right]_{H_2} + \left[mc_{p,\text{avg}}(T_2 - T_1)\right]_{N_2} \\ &= (0.5\text{ kg})(14.501\text{ kJ/kg}\cdot\text{K})(600-300)\text{K} + (1.6\text{ kg})(1.049\text{ kJ/kg}\cdot\text{K})(600-300)\text{K} \\ &= \mathbf{2679\ kJ} \end{aligned}$$

(*b*) Noting that the total mixture pressure, and thus the partial pressure of each gas, remains constant, the entropy change of the mixture during this process is

$$\begin{aligned} \Delta S_{H_2} &= \left[m(s_2 - s_1)\right]_{H_2} = m_{H_2}\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}^{\nearrow 0}\right)_{H_2} = m_{H_2}\left(c_p \ln\frac{T_2}{T_1}\right)_{H_2} \\ &= (0.5\text{ kg})(14.501\text{ kJ/kg}\cdot\text{K})\ln\frac{600\text{ K}}{300\text{ K}} \\ &= 5.026\text{ kJ/K} \end{aligned}$$

$$\begin{aligned} \Delta S_{N_2} &= \left[m(s_2 - s_1)\right]_{N_2} = m_{N_2}\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}^{\nearrow 0}\right)_{N_2} = m_{N_2}\left(c_p \ln\frac{T_2}{T_1}\right)_{N_2} \\ &= (1.6\text{ kg})(1.049\text{ kJ/kg}\cdot\text{K})\ln\frac{600\text{ K}}{300\text{ K}} \\ &= 1.163\text{ kJ/K} \end{aligned}$$

$$\Delta S_{\text{total}} = \Delta S_{H_2} + \Delta S_{N_2} = 5.026\text{ kJ/K} + 1.163\text{ kJ/K} = \mathbf{6.19\ kJ/K}$$

14-73 Saturated humid air at a specified state is heated to a specified temperature. The relative humidity at the exit and the rate of heat transfer are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

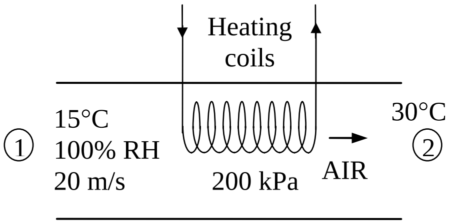
Analysis The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 200 kPa. The properties of the air at the inlet and exit states are determined to be

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 15°C}} = (1.0)(1.7057\text{ kPa}) = 1.7057\text{ kPa}$$

$$h_{g1} = h_{g\text{ @15°C}} = 2528.3\text{ kJ/kg}$$

$$P_{a1} = P_1 - P_{v1} = 200 - 1.7057 = 198.29\text{ kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\text{ K})}{198.29\text{ kPa}} = 0.4168\text{ m}^3/\text{kg dry air}$$



$$\omega_1 = \frac{0.622 P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.7057\text{ kPa})}{(200-1.7057)\text{ kPa}} = 0.005350\text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot\text{°C})(15\text{°C}) + (0.005350)(2528.3\text{ kJ/kg}) = 28.60\text{ kJ/kg dry air}$$

$$P_{v2} = P_{v1} = 1.7057\text{ kPa}$$

$$P_{g2} = P_{\text{sat @ 30°C}} = 4.2469\text{ kPa}$$

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{1.7057\text{ kPa}}{4.2469\text{ kPa}} = 0.402 = \mathbf{40.2\%}$$

$$h_{g2} = h_{g\text{ @ 30°C}} = 2555.6\text{ kJ/kg}$$

$$\omega_2 = \omega_1$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot\text{°C})(30\text{°C}) + (0.005350)(2555.6\text{ kJ/kg}) = 43.82\text{ kJ/kg dry air}$$

Then,

$$\dot{\boldsymbol{V}}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (20\text{ m/s})\left(\frac{\pi(0.04\text{ m})^2}{4}\right) = 0.02513\text{ m}^3/\text{s}$$

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.02513\text{ m}^3/\text{s}}{0.4168\text{ m}^3/\text{kg dry air}} = 0.06029\text{ kg/s}$$

From the energy balance on air in the heating section,

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1) = (0.06029\text{ kg/s})(43.82 - 28.60)\text{kJ/kg} = \mathbf{0.918\text{ kW}}$$

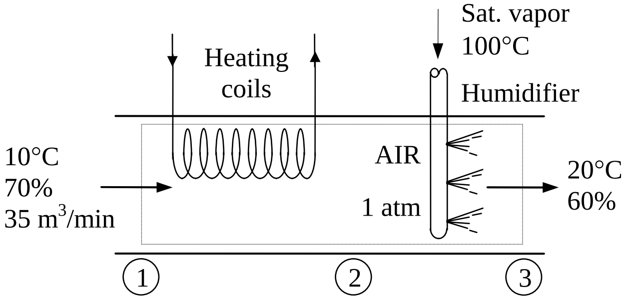
14-78 Air is first heated and then humidified by wet steam. The temperature and relative humidity of air at the exit of heating section, the rate of heat transfer, and the rate at which water is added to the air are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 23.5 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0053 \text{ kg H}_2\text{O/kg dry air} \ (= \omega_2)$$
$$v_1 = 0.809 \text{ m}^3\text{/kg dry air}$$

$$h_3 = 42.3 \text{ kJ/kg dry air}$$
$$\omega_3 = 0.0087 \text{ kg H}_2\text{O/kg dry air}$$



Analysis (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{35 \text{ m}^3\text{/min}}{0.809 \text{ m}^3\text{/kg}} = 43.3 \text{ kg/min}$$

Noting that $Q = W = 0$, the energy balance on the humidifying section can be expressed as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad \longrightarrow \quad \dot{m}_w h_w + \dot{m}_{a2} h_2 = \dot{m}_a h_3$$
$$(\omega_3 - \omega_2) h_w + h_2 = h_3$$

Solving for h_2,

$$h_2 = h_3 - (\omega_3 - \omega_2) h_{g\,@100°\text{C}} = 42.3 - (0.0087 - 0.0053)(2675.6) = 33.2 \text{ kJ/kg dry air}$$

Thus at the exit of the heating section we have ω_2 = 0.0053 kg H_2O dry air and h_2 = 33.2 kJ/kg dry air, which completely fixes the state. Then from the psychrometric chart we read

$$T_2 = \mathbf{19.5°C}$$
$$\phi_2 = \mathbf{37.8\%}$$

(*b*) The rate of heat transfer to the air in the heating section is

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (43.3 \text{ kg/min})(33.2 - 23.5) \text{ kJ/kg} = \mathbf{420 \text{ kJ/min}}$$

(*c*) The amount of water added to the air in the humidifying section is determined from the conservation of mass equation of water in the humidifying section,

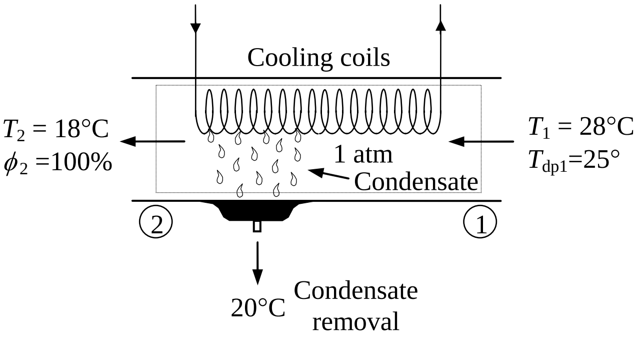
$$\dot{m}_w = \dot{m}_a (\omega_3 - \omega_2) = (43.3 \text{ kg/min})(0.0087 - 0.0053) = \mathbf{0.15 \text{ kg/min}}$$

14-82 Air is cooled and dehumidified at constant pressure. The amount of water removed from the air and the rate of cooling are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Properties The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 79.6 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0202 \text{ kg H}_2\text{O/kg dry air}$$
$$\boldsymbol{v}_1 = 0.881 \text{ m}^3\text{/kg dry air}$$



and

$$\phi_2 = 1.0$$
$$h_2 = 51.0 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0130 \text{ kg H}_2\text{O/kg dry air}$$

Also,

$$h_w \cong h_{f\,@\,20°\text{C}} = 83.915 \text{ kJ/kg} \qquad \text{(Table A-4)}$$

Analysis The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass flow rate of air is

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{(10{,}000/3600) \text{ m}^3/\text{s}}{0.881 \text{ m}^3/\text{kg dry air}} = 3.153 \text{ kg/s}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (3.153 \text{ kg/s})(0.0202 - 0.0130) = \mathbf{0.0227\ kg/s}$$

Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (3.153 \text{ kg/s})(79.6 - 51.0)\text{kJ/kg} - (0.02227 \text{ kg/s})(83.915 \text{ kJ/kg})$$

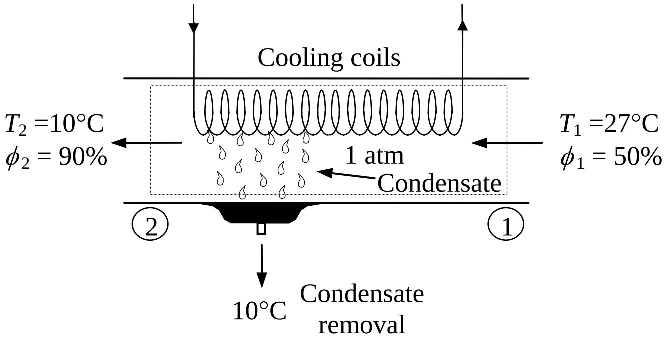
$$= \mathbf{88.3\ kW}$$

14-90 Atmospheric air enters the evaporator of an automobile air conditioner at a specified pressure, temperature, and relative humidity. The dew point and wet bulb temperatures at the inlet to the evaporator section, the required heat transfer rate from the atmospheric air to the evaporator fluid, and the rate of condensation of water vapor in the evaporator section are to be determined.

Assumptions **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

Analysis The inlet and exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at the inlet and exit states may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$T_{\text{dp1}} = \mathbf{15.7^\circ C}$$
$$T_{\text{wb1}} = \mathbf{19.5^\circ C}$$
$$h_1 = 55.60 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01115 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.8655 \text{ m}^3\text{ / kg dry air}$$
$$h_2 = 27.35 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.00686 \text{ kg H}_2\text{O/kg dry air}$$



The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{V_{\text{car}}\text{ACH}}{v_1} = \frac{(2\text{ m}^3\text{/change})(5\text{ changes/min})}{0.8655\text{ m}^3} = 11.55\text{ kg/min}$$

The mass flow rates of vapor at the inlet and exit are

$$\dot{m}_{v1} = \omega_1 \dot{m}_a = (0.01115)(11.55\text{ kg/min}) = 0.1288\text{ kg/min}$$

$$\dot{m}_{v2} = \omega_2 \dot{m}_a = (0.00686)(11.55\text{ kg/min}) = 0.07926\text{ kg/min}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$

where the the enthalpy of condensate water is

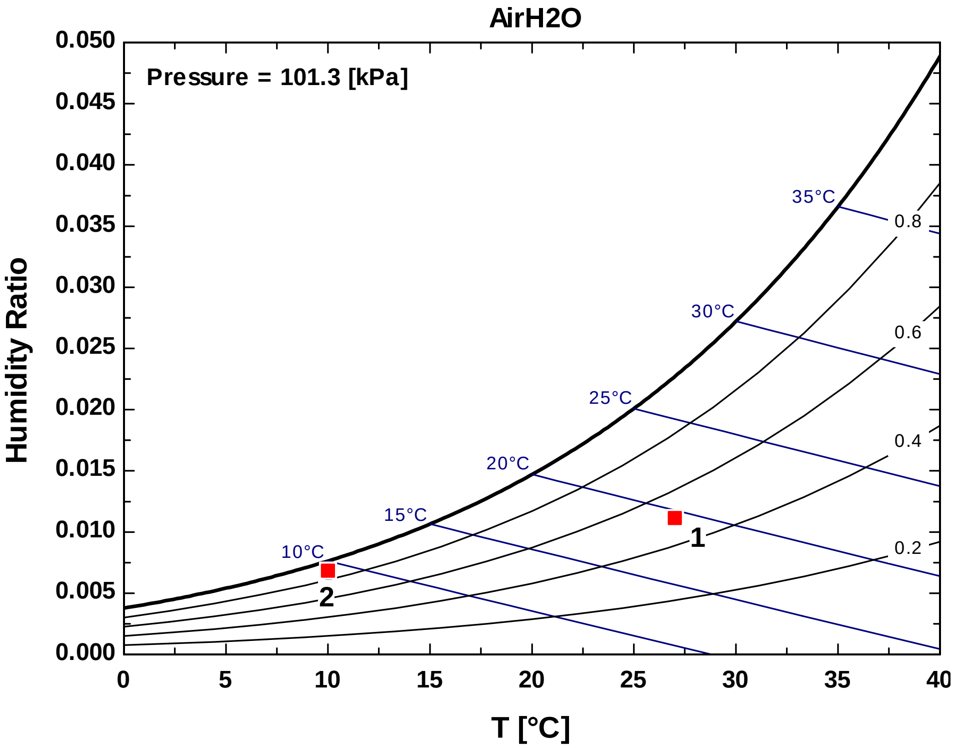
$$h_{w2} = h_{f@10^\circ\text{C}} = 42.02\text{ kJ/kg} \quad \text{(Table A-4)}$$

and the rate of condensation of water vapor is

$$\dot{m}_w = \dot{m}_{v1} - \dot{m}_{v2} = 0.1288 - 0.07926 = \mathbf{0.0495\text{ kg/min}}$$

Substituting,

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(11.55\text{ kg/min})(55.60\text{ kJ/kg}) = \dot{Q}_{\text{out}} + (11.55\text{ kg/min})(27.35\text{ kJ/kg}) + (0.0495\text{ kg/min})(42.02\text{ kJ/kg})$$
$$\dot{Q}_{\text{out}} = 324.4\text{ kJ/min} = \mathbf{5.41\,kW}$$



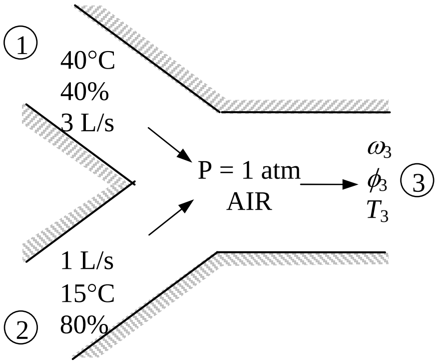
Discussion We we could not show the process line between the states 1 and 2 because we do not know the process path.

14-110 Two airstreams are mixed steadily. The temperature and the relative humidity of the mixture are to be determined.

Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31 or from EES) to be

$$h_1 = 88.5 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0187 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.914 \text{ m}^3\text{/kg dry air}$$



and

$$h_2 = 36.7 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0085 \text{ kg H}_2\text{O/kg dry air}$$
$$v_2 = 0.828 \text{ m}^3\text{/kg dry air}$$

Analysis The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{0.003 \text{ m}^3/\text{s}}{0.914 \text{ m}^3/\text{kg dry air}} = 0.003282 \text{ kg/s}$$

$$\dot{m}_{a2} = \frac{\dot{V}_2}{v_2} = \frac{0.001 \text{ m}^3/\text{s}}{0.828 \text{ m}^3/\text{kg dry air}} = 0.001208 \text{ kg/s}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (0.003282 + 0.001208) \text{ kg/s} = 0.00449 \text{ kg/s}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{0.003282}{0.001208} = \frac{0.0085 - \omega_3}{\omega_3 - 0.0187} = \frac{36.7 - h_3}{h_3 - 88.5}$$

which yields

$$\omega_3 = 0.0160 \text{ kg H}_2\text{O/kg dry air}$$
$$h_3 = 74.6 \text{ kJ/kg dry air}$$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

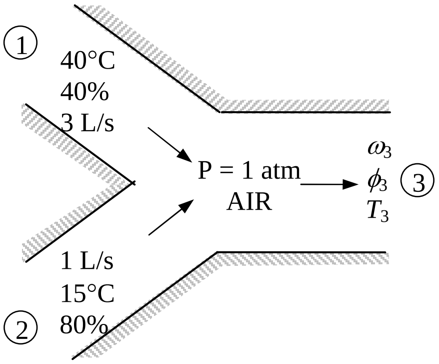
$$T_3 = \mathbf{33.4°C}$$
$$\phi_3 = 0.493 = \mathbf{49.3\%}$$

14-111 Two airstreams are mixed steadily. The rate of exergy destruction is to be determined.

Assumptions **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

Properties Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31 or from EES) to be

$$\begin{aligned} h_1 &= 88.5\ \text{kJ/kg dry air} \\ \omega_1 &= 0.0187\ \text{kg H}_2\text{O/kg dry air} \\ \boldsymbol{v}_1 &= 0.914\ \text{m}^3\text{/kg dry air} \end{aligned}$$



and

$$\begin{aligned} h_2 &= 36.7\ \text{kJ/kg dry air} \\ \omega_2 &= 0.0085\ \text{kg H}_2\text{O/kg dry air} \\ \boldsymbol{v}_2 &= 0.828\ \text{m}^3\text{/kg dry air} \end{aligned}$$

The entropies of water vapor in the air streams are

$$\begin{aligned} s_{g1} &= s_{g\,@\,40°\text{C}} = 8.2556\ \text{kJ/kg}\cdot\text{K} \\ s_{g2} &= s_{g\,@15°\text{C}} = 8.7803\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Analysis The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.003\ \text{m}^3\ /\ \text{s}}{0.914\ \text{m}^3\ /\ \text{kg dry air}} = 0.003282\ \text{kg/s}$$

$$\dot{m}_{a2} = \frac{\dot{\boldsymbol{V}}_2}{\boldsymbol{v}_2} = \frac{0.001\ \text{m}^3\ /\ \text{s}}{0.828\ \text{m}^3\ /\ \text{kg dry air}} = 0.001208\ \text{kg/s}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (0.003282 + 0.001208)\ \text{kg/s} = 0.00449\ \text{kg/s}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{0.003282}{0.001208} = \frac{0.0085 - \omega_3}{\omega_3 - 0.0187} = \frac{36.7 - h_3}{h_3 - 88.5}$$

which yields

$$\begin{aligned} \omega_3 &= 0.0160\ \text{kg H}_2\text{O/kg dry air} \\ h_3 &= 74.6\ \text{kJ/kg dry air} \end{aligned}$$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

$$\begin{aligned} T_3 &= 33.4°\text{C} \\ \phi_3 &= 0.493 \end{aligned}$$

The entropy of water vapor in the mixture is

$$s_{g3} = s_{g\,@33.4°\text{C}} = 8.3833\ \text{kJ/kg}\cdot\text{K}$$

An entropy balance on the mixing chamber for the water gives

$$\begin{aligned}\Delta\dot{S}_w &= \dot{m}_{a3}\omega_3 s_3 - \dot{m}_{a1}\omega_1 s_1 - \dot{m}_{a2}\omega_2 s_2\\ &= 0.00449\times 0.0160\times 8.3833 - 0.003282\times 0.0187\times 8.2556 - 0.001208\times 0.0085\times 8.7803\\ &= 5.426\times 10^{-6}\ \text{kW/K}\end{aligned}$$

The partial pressures of water vapor and dry air for all three air streams are

$$\begin{aligned}P_{v1} &= \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 40°C}} = (0.40)(7.3851\ \text{kPa}) = 2.954\ \text{kPa}\\ P_{a1} &= P_1 - P_{v1} = 101.325 - 2.954 = 98.37\ \text{kPa}\\ P_{v2} &= \phi_2 P_{g2} = \phi_2 P_{\text{sat @ 15°C}} = (0.80)(1.7057\ \text{kPa}) = 1.365\ \text{kPa}\\ P_{a2} &= P_2 - P_{v2} = 101.325 - 1.365 = 99.96\ \text{kPa}\\ P_{v3} &= \phi_3 P_{g3} = \phi_3 P_{\text{sat @ 33.4°C}} = (0.493)(5.150\ \text{kPa}) = 2.539\ \text{kPa}\\ P_{a3} &= P_3 - P_{v3} = 101.325 - 2.539 = 98.79\ \text{kPa}\end{aligned}$$

An entropy balance on the mixing chamber for the dry air gives

$$\begin{aligned}\Delta\dot{S}_a &= \dot{m}_{a1}(s_3 - s_1) + \dot{m}_{a2}(s_3 - s_2)\\ &= \dot{m}_{a1}\left(c_p \ln\frac{T_3}{T_1} - R\ln\frac{P_{a3}}{P_{a1}}\right) + \dot{m}_{a2}\left(c_p \ln\frac{T_3}{T_2} - R\ln\frac{P_{a3}}{P_{a2}}\right)\\ &= 0.003282\left[(1.005)\ln\frac{306.4}{313} - (0.287)\ln\frac{98.79}{98.37}\right] + 0.001208\left[(1.005)\ln\frac{306.4}{288} - (0.287)\ln\frac{98.79}{99.96}\right]\\ &= (0.003282)(-0.02264) + (0.001208)(0.06562)\\ &= 4.964\times 10^{-6}\ \text{kW/K}\end{aligned}$$

The rate of entropy generation is

$$\dot{S}_{\text{gen}} = \Delta\dot{S}_a + \Delta\dot{S}_w = 4.964\times 10^{-6} + 5.426\times 10^{-6} = 10.39\times 10^{-6}\ \text{kW/K}$$

Finally, the rate of exergy destruction is

$$\dot{X}_{\text{dest}} = T_0\dot{S}_{\text{gen}} = (298\ \text{K})(10.39\times 10^{-6}\ \text{kW/K}) = \mathbf{0.0031\ kW}$$

14-115 Water is cooled by air in a cooling tower. The relative humidity of the air at the exit and the water's exit temperature are to be determined.

Assumptions **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

Analysis The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$

Energy Balance:

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

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

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

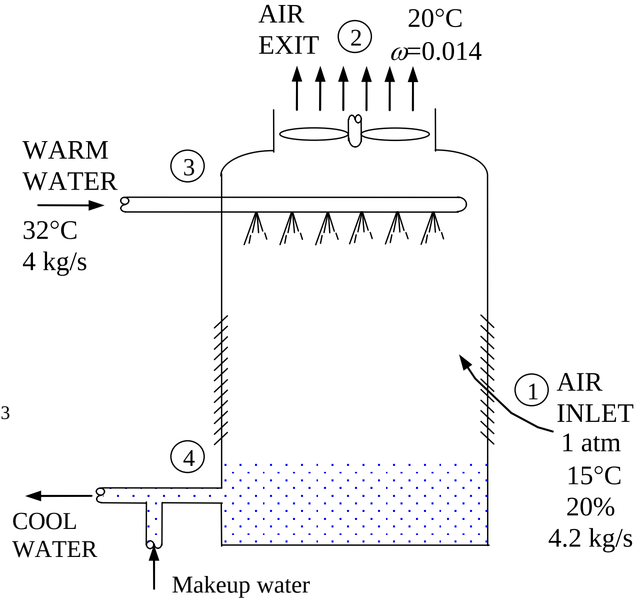
$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$

Solving for h_4,

$$h_4 = \frac{\dot{m}_3 h_3 - \dot{m}_a(h_2 - h_1)}{\dot{m}_3 - \dot{m}_{\text{makeup}}}$$



From the psychrometric chart (Fig. A-31),

$$h_1 = 20.4 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.00211 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.819 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 55.7 \text{ kJ/kg dry air}$$

$$\phi_2 = 1 = \mathbf{100\%}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,32^\circ\text{C}} = 134.1 \text{ kJ/kg H}_2\text{O}$$

Also, $\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (4.2 \text{ kg/s})(0.014 - 0.00211) = 0.050 \text{ kg/s}$

Substituting,

$$h_4 = \frac{\dot{m}_3 h_3 - \dot{m}_a(h_2 - h_1)}{\dot{m}_3 - \dot{m}_{\text{makeup}}} = \frac{(4)(134.1) - (0.12)(55.7 - 20.4)}{4 - 0.050} = 98.31 \text{ kJ/kg H}_2\text{O}$$

The exit temperature of the water is then (Table A-4)

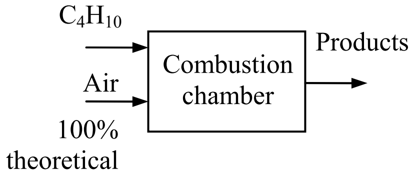
$$T_4 = T_{\text{sat}\,@\,h_f = 98.31\text{ kJ/kg}} = \mathbf{23.4^\circ C}$$

15-18 n-Octane is burned with stoichiometric amount of air. The mass fraction of each product, the mass of water in the products and the mass fraction of each reactant are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction equation for 100% theoretical air is



$$\mathrm{C_8H_{18}} + a_{\mathrm{th}}\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow B\,\mathrm{CO_2} + D\,\mathrm{H_2O} + E\,\mathrm{N_2}$$

where a_{th} is the stoichiometric coefficient for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

Carbon balance: $B = 8$

Hydrogen balance: $2D = 18 \longrightarrow D = 9$

Oxygen balance: $2a_{\mathrm{th}} = 2B + D \longrightarrow a_{\mathrm{th}} = 0.5(2\times 8 + 9) = 12.5$

Nitrogen balance: $a_{\mathrm{th}} \times 3.76 = E \longrightarrow E = 12.5 \times 3.76 = 47$

Substituting, the balanced reaction equation is

$$\mathrm{C_8H_{18}} + 12.5\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow 8\,\mathrm{CO_2} + 9\,\mathrm{H_2O} + 47\,\mathrm{N_2}$$

The mass of each product and the total mass are

$$\begin{aligned} m_{\mathrm{CO2}} &= N_{\mathrm{CO2}} M_{\mathrm{CO2}} = (8\text{ kmol})(44\text{ kg/kmol}) = 352\text{ kg} \\ m_{\mathrm{H2O}} &= N_{\mathrm{H2O}} M_{\mathrm{H2O}} = (9\text{ kmol})(18\text{ kg/kmol}) = 162\text{ kg} \\ m_{\mathrm{N2}} &= N_{\mathrm{N2}} M_{\mathrm{N2}} = (47\text{ kmol})(28\text{ kg/kmol}) = 1316\text{ kg} \\ m_{\mathrm{total}} &= m_{\mathrm{CO2}} + m_{\mathrm{N2}} + m_{\mathrm{H2O}} = 352 + 162 + 1316 = 1830\text{ kg} \end{aligned}$$

Then the mass fractions are

$$\mathrm{mf}_{\mathrm{CO2}} = \frac{m_{\mathrm{CO2}}}{m_{\mathrm{total}}} = \frac{352\text{ kg}}{1830\text{ kg}} = \mathbf{0.1923}$$

$$\mathrm{mf}_{\mathrm{H2O}} = \frac{m_{\mathrm{H2O}}}{m_{\mathrm{total}}} = \frac{162\text{ kg}}{1830\text{ kg}} = \mathbf{0.0885}$$

$$\mathrm{mf}_{\mathrm{N2}} = \frac{m_{\mathrm{N2}}}{m_{\mathrm{total}}} = \frac{1316\text{ kg}}{1830\text{ kg}} = \mathbf{0.7191}$$

The mass of water per unit mass of fuel burned is

$$\frac{m_{\mathrm{H2O}}}{m_{\mathrm{C8H18}}} = \frac{(9\times 18)\text{ kg}}{(1\times 114)\text{ kg}} = \mathbf{1.421\,kg\,H_2O/kg\,C_8H_{18}}$$

The mass of each reactant and the total mass are

$$\begin{aligned} m_{\mathrm{C8H18}} &= N_{\mathrm{C8H18}} M_{\mathrm{C8H18}} = (1\text{ kmol})(114\text{ kg/kmol}) = 114\text{ kg} \\ m_{\mathrm{air}} &= N_{\mathrm{air}} M_{\mathrm{air}} = (12.5\times 4.76\text{ kmol})(29\text{ kg/kmol}) = 1725.5\text{ kg} \\ m_{\mathrm{total}} &= m_{\mathrm{C8H18}} + m_{\mathrm{air}} = 114 + 1725.5 = 1839.5\text{ kg} \end{aligned}$$

Then the mass fractions of reactants are

$$\mathrm{mf}_{\mathrm{C8H18}} = \frac{m_{\mathrm{C8H18}}}{m_{\mathrm{total}}} = \frac{114\text{ kg}}{1839.5\text{ kg}} = \mathbf{0.0620}$$

$$\mathrm{mf}_{\mathrm{air}} = \frac{m_{\mathrm{air}}}{m_{\mathrm{total}}} = \frac{1725.5\text{ kg}}{1839.5\text{ kg}} = \mathbf{0.9380}$$

15-27 Butane C_4H_{10} is burned with 200 percent theoretical air. The kmol of water that needs to be sprayed into the combustion chamber per kmol of fuel is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction equation for 200% theoretical air without the additional water is

$$\mathrm{C_4H_{10}} + 2a_{\mathrm{th}}\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow B\,\mathrm{CO_2} + D\,\mathrm{H_2O} + E\,\mathrm{O_2} + F\,\mathrm{N_2}$$

where a_{th} is the stoichiometric coefficient for air. We have automatically accounted for the 100% excess air by using the factor $2a_{\mathrm{th}}$ instead of a_{th} for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

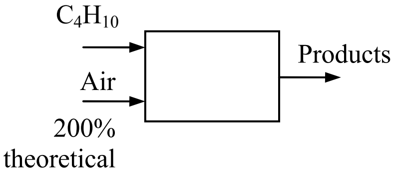
Carbon balance: $B = 4$

Hydrogen balance: $2D = 10 \longrightarrow D = 5$

Oxygen balance: $2 \times 2a_{\mathrm{th}} = 2B + D + 2E$

$a_{\mathrm{th}} = E$

Nitrogen balance: $2a_{\mathrm{th}} \times 3.76 = F$



Solving the above equations, we find the coefficients (E = 6.5, F = 48.88, and a_{th} = 6.5) and write the balanced reaction equation as

$$\mathrm{C_4H_{10}} + 13\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow 4\,\mathrm{CO_2} + 5\,\mathrm{H_2O} + 6.5\,\mathrm{O_2} + 48.88\,\mathrm{N_2}$$

With the additional water sprayed into the combustion chamber, the balanced reaction equation is

$$\mathrm{C_4H_{10}} + 13\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] + N_v\,\mathrm{H_2O} \longrightarrow 4\,\mathrm{CO_2} + (5 + N_v)\,\mathrm{H_2O} + 6.5\,\mathrm{O_2} + 48.88\,\mathrm{N_2}$$

The partial pressure of water in the saturated product mixture at the dew point is

$$P_{v,\mathrm{prod}} = P_{\mathrm{sat@60^\circ C}} = 19.95\ \mathrm{kPa}$$

The vapor mole fraction is

$$y_v = \frac{P_{v,\mathrm{prod}}}{P_{\mathrm{prod}}} = \frac{19.95\ \mathrm{kPa}}{100\ \mathrm{kPa}} = 0.1995$$

The amount of water that needs to be sprayed into the combustion chamber can be determined from

$$y_v = \frac{N_{\mathrm{water}}}{N_{\mathrm{total,product}}} \longrightarrow 0.1995 = \frac{5 + N_v}{4 + 5 + N_v + 6.5 + 48.88} \longrightarrow N_v = \mathbf{9.796\ kmol}$$

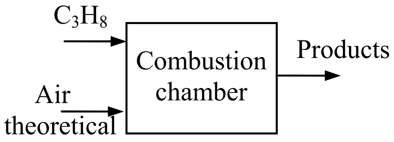
15-51 The higher and lower heating values of liquid propane are to be determined and compared to the listed values.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, and N_2. **3** Combustion gases are ideal gases.

Properties The molar masses of C, O_2, H_2, and air are 12, 32, 2, and 29 kg/kmol, respectively (Table A-1).

Analysis The combustion reaction with stoichiometric air is

$$C_3H_8(l) + 5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 18.8N_2$$



Both the reactants and the products are taken to be at the standard reference state of 25°C and 1 atm for the calculation of heating values. The heat transfer for this process is equal to enthalpy of combustion. Note that N_2 and O_2 are stable elements, and thus their enthalpy of formation is zero. Then,

$$q = h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{\mathrm{CO2}} + \left(N\bar{h}^\circ_f\right)_{\mathrm{H2O}} - \left(N\bar{h}^\circ_f\right)_{\mathrm{C3H8}}$$

The $\bar{h}^\circ_f$ of liquid propane is obtained by adding $\bar{h}_{fg}$ of propane at 25°C to $\bar{h}^\circ_f$ of gas propane (103,850 + 44.097 × 335 = 118,620 kJ/kmol). For the HHV, the water in the products is taken to be liquid. Then,

$$\begin{aligned} h_C &= (3\text{ kmol})(-393{,}520\text{ kJ/kmol}) + (4\text{ kmol})(-285{,}830\text{ kJ/kmol}) - (1\text{ kmol})(-118{,}620\text{ kJ/kmol}) \\ &= -2{,}205{,}260\text{ kJ/kmol propane} \end{aligned}$$

The HHV of the liquid propane is

$$\mathrm{HHV} = \frac{-h_C}{M_m} = \frac{2{,}205{,}260\text{ kJ/kmol } C_3H_8}{44.097\text{ kg/kmol } C_3H_8} = \mathbf{50{,}010\ kJ/kg\ C_3H_8}$$

The listed value from Table A-27 is **50,330 kJ/kg**. For the LHV, the water in the products is taken to be vapor. Then,

$$\begin{aligned} h_C &= (3\text{ kmol})(-393{,}520\text{ kJ/kmol}) + (4\text{ kmol})(-241{,}820\text{ kJ/kmol}) - (1\text{ kmol})(-118{,}620\text{ kJ/kmol}) \\ &= -2{,}029{,}220\text{ kJ/kmol propane} \end{aligned}$$

The LHV of the propane is then

$$\mathrm{LHV} = \frac{-h_C}{M_m} = \frac{2{,}029{,}220\text{ kJ/kmol } C_3H_8}{44.097\text{ kg/kmol } C_3H_8} = \mathbf{46{,}020\ kJ/kg\ C_3H_8}$$

The listed value from Table A-27 is **46,340 kJ/kg**. The calculated and listed values are practically identical.

15-66 A mixture of propane and methane is burned with theoretical air. The balanced chemical reaction is to be written, and the amount of water vapor condensed and the the required air flow rate for a given heat transfer rate are to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, CO, H_2O, O_2, and N_2 only.

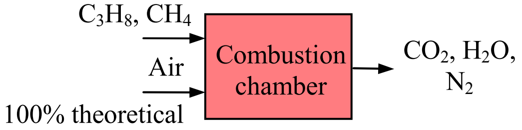
Properties The molar masses of C, H_2, O_2, N_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis (a) The balanced reaction equation for stoichiometric air is

$$0.4\,C_3H_8 + 0.6\,CH_4 + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow 1.8\,CO_2 + 2.8\,H_2O + a_{th} \times 3.76\,N_2$$

The stoicihiometric coefficient a_{th} is determined from an O_2 balance:

$$a_{th} = 1.8 + 1.4 = 3.2$$



Substituting,

$$0.4\,C_3H_8 + 0.6\,CH_4 + 3.2\left[O_2 + 3.76N_2\right] \longrightarrow 1.8\,CO_2 + 2.8\,H_2O + 12.032\,N_2$$

(b) The partial pressure of water vapor is

$$P_v = \frac{N_{H2O}}{N_{total}} P_{total} = \frac{2.8}{1.8 + 2.8 + 12.032}(100\text{ kPa}) = \frac{2.8\text{ kmol}}{16.632\text{ kmol}}(100\text{ kPa}) = 16.84\text{ kPa}$$

The dew point temperature of the product gases is the saturation temperature of water at this pressure:

$$T_{dp} = T_{sat@16.84\,kPa} = 56.2°\text{C} \quad \text{(Table A-5)}$$

Since the temperature of the product gases are at 398 K (125°C), there will be no condensation of water vapor.

(c) The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

The products are at 125 °C, and the enthalpy of products can be expressed as

$$\left(\bar{h} - \bar{h}^\circ\right) = \bar{c}_p \Delta T$$

where $\Delta T = 125 - 25 = 100°\text{C} = 100\text{ K}$. Then, using the values given in the table,

$$\begin{aligned}-Q_{out} &= (1.8)(-393{,}520 + 41.16 \times 100) + (2.8)(-241{,}820 + 34.28 \times 100) + (12.032)(0 + 29.27 \times 100) \\ &\quad - (0.4)(-103{,}850) - (0.6)(-74{,}850) \\ &= -1{,}246{,}760\text{ kJ/kmol fuel}\end{aligned}$$

or $\quad Q_{out} = 1{,}246{,}760\text{ kJ/kmol fuel}$

For a heat transfer rate of 97,000 kJ/h, the molar flow rate of fuel is

$$\dot{N}_{fuel} = \frac{\dot{Q}_{out}}{Q_{out}} = \frac{97{,}000\text{ kJ/h}}{1{,}246{,}760\text{ kJ/kmol fuel}} = 0.07780\text{ kmol fuel/h}$$

The molar mass of the fuel mixture is

$$M_{fuel} = 0.4 \times 44 + 0.6 \times 16 = 27.2\text{ kg/kmol}$$

The mass flow rate of fuel is

$$\dot{m}_{fuel} = \dot{N}_{fuel} M_{fuel} = (0.07780\text{ kmol/h})(27.2\text{ kg/kmol}) = 2.116\text{ kg/h}$$

The air-fuel ratio is

$$\text{AF} = \frac{m_{air}}{m_{fuel}} = \frac{(3.2 \times 4.76 \times 29)\text{ kg}}{(0.4 \times 44 + 0.6 \times 16)\text{ kg}} = 16.24\text{ kg air/kg fuel}$$

The mass flow rate of air is then

$$\dot{m}_{air} = \dot{m}_{fuel}\text{AF} = (2.116\text{ kg/h})16.24) = \mathbf{34.4\text{ kg/h}}$$

15-73 A high efficiency gas furnace burns gaseous propane C_3H_8 with 140 percent theoretical air. The volume flow rate of water condensed from the product gases is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, O_2, and N_2 only.

Properties The molar masses of C, H_2, O_2 and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

Analysis The reaction equation for 40% excess air (140% theoretical air) is

$$C_3H_8 + 1.4a_{th}[O_2 + 3.76N_2] \longrightarrow B\,CO_2 + D\,H_2O + E\,O_2 + F\,N_2$$

where a_{th} is the stoichiometric coefficient for air. We have automatically accounted for the 40% excess air by using the factor $1.4a_{th}$ instead of a_{th} for air. The coefficient a_{th} and other coefficients are to be determined from the mass balances

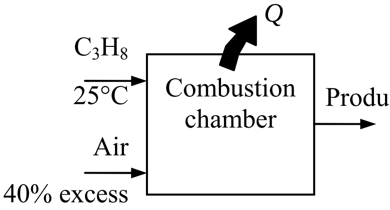
Carbon balance: $B = 3$

Hydrogen balance: $2D = 8 \longrightarrow D = 4$

Oxygen balance: $2\times 1.4a_{th} = 2B + D + 2E$

$0.4a_{th} = E$

Nitrogen balance: $1.4a_{th} \times 3.76 = F$



Solving the above equations, we find the coefficients (E = 2, F = 26.32, and a_{th} = 5) and write the balanced reaction equation as

$$C_3H_8 + 7[O_2 + 3.76N_2] \longrightarrow 3\,CO_2 + 4\,H_2O + 2\,O_2 + 26.32\,N_2$$

The partial pressure of water in the saturated product mixture at the dew point is

$$P_{v,\text{prod}} = P_{\text{sat@40°C}} = 7.3851\,\text{kPa}$$

The vapor mole fraction is

$$y_v = \frac{P_{v,\text{prod}}}{P_{\text{prod}}} = \frac{7.3851\,\text{kPa}}{100\,\text{kPa}} = 0.07385$$

The kmoles of water condensed is determined from

$$y_v = \frac{N_{\text{water}}}{N_{\text{total,product}}} \longrightarrow 0.07385 = \frac{4 - N_w}{3 + 4 - N_w + 2 + 26.32} \longrightarrow N_w = 1.503\,\text{kmol}$$

The steady-flow energy balance is expressed as

$$\dot N_{\text{fuel}} H_R = \dot Q_{\text{fuel}} + \dot N_{\text{fuel}} H_P$$

where $\dot Q_{\text{fuel}} = \dfrac{\dot Q_{\text{out}}}{\eta_{\text{furnace}}} = \dfrac{31{,}650\,\text{kJ/h}}{0.96} = 32{,}969\,\text{kJ/h}$

$$H_R = \bar h^o_{f\,\text{fuel@25°C}} + 7\bar h_{\text{O2@25°C}} + 26.32\bar h_{\text{N2@25°C}}$$
$$= (-103{,}847\,\text{kJ/kmol}) + 7(0) + 26.32(0) = -103{,}847\,\text{kJ/kmol}$$

$$H_P = 3\bar h_{\text{CO2@25°C}} + 4\bar h_{\text{H2O@25°C}} + 2\bar h_{\text{O2@25°C}} + 26.32\bar h_{\text{N2@25°C}} + N_w(\bar h^o_{f\,\text{H2O(liq)}})$$
$$= 3(-393{,}520\,\text{kJ/kmol}) + 4(\text{-}241{,}820\,\text{kJ/kmol}) + 2(0) + 26.32(0) + 1.503(-285{,}830\,\text{kJ/kmol})$$
$$= -2.577\times 10^6\,\text{kJ/kmol}$$

Substituting into the energy balance equation,

$$\dot N_{\text{fuel}} H_R = \dot Q_{\text{fuel}} + \dot N_{\text{fuel}} H_P$$

$$\dot N_{\text{fuel}}(-103{,}847\,\text{kJ/kmol}) = 32{,}969\,\text{kJ/h} + \dot N_{\text{fuel}}(-2.577\times 10^6\,\text{kJ/kmol}) \longrightarrow \dot N_{\text{fuel}} = 0.01333\,\text{kmol/h}$$

The molar and mass flow rates of the liquid water are

$$\dot N_w = N_w \dot N_{\text{fuel}} = (1.503\ \text{kmol/kmol fuel})(0.01333\ \text{kmol fuel/h}) = 0.02003\ \text{kmol/h}$$

$$\dot m_w = \dot N_w M_w = (0.02003\ \text{kmol/h})(18\ \text{kg/kmol}) = 0.3608\ \text{kg/h}$$

The volume flow rate of liquid water is

$$\dot V_w = (v_{f\,@25°C})\dot m_w = (0.001003\ \text{m}^3/\text{kg})(0.3608\ \text{kg/h}) = 0.0003619\ \text{m}^3/\text{h} = \mathbf{8.7\ L/day}$$

15-80 Propane is burned with stoichiometric and 50 percent excess air. The adiabatic flame temperature is to be determined for both cases.

Assumptions **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

Analysis Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $Q = W = 0$ reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \bar{h}_{f,R}^\circ$$

since all the reactants are at the standard reference temperature of 25°C. Then, for the stoicihiometric air

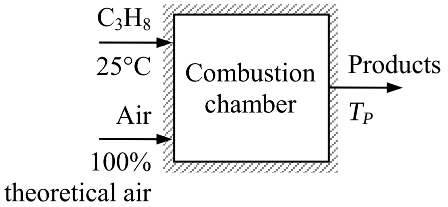
$$C_3H_8 + a_{\text{th}}(O_2 + 3.76\,N_2) \longrightarrow 3\,CO_2 + 4\,H_2O + a_{\text{th}} \times 3.76\,N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$a_{\text{th}} = 3 + 2 = 5$$

Thus,

$$C_3H_8 + 5\,(O_2 + 3.76\,N_2) \longrightarrow 3\,CO_2 + 4\,H_2O + 18.8\,N_2$$



From the tables,

Substance	$\bar{h}_f^\circ$ **kJ/kmol**	$\bar{h}_{298K}$ **kJ/kmol**
C_3H_8 (*g*)	-103,850	---
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO_2	-393,520	9364

Thus,

$$(3)\left(-393{,}520 + \bar{h}_{CO2} - 9364\right) + (4)\left(-241{,}820 + \bar{h}_{H2O} - 9904\right) + (18.8)\left(0 + \bar{h}_{N2} - 8669\right) = (1)\left(-103{,}850\right) + 0 + 0$$

It yields $$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} = 2{,}274{,}680 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,274,680/(3 + 4 + 18.8) = 88,166 kJ/kmol. This enthalpy value corresponds to about 2650 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 2650 K, but somewhat under it because of the higher specific heat of H_2O.

At 2500 K:

$$\begin{aligned} 3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} &= 3 \times 131{,}290 + 4 \times 108{,}868 + 18.8 \times 82{,}981 \\ &= 2{,}389{,}380 \text{ kJ} \ \ (\text{Higher than } 2{,}274{,}680 \text{ kJ}) \end{aligned}$$

At 2450 K:

$$\begin{aligned} 3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} &= 3 \times 128{,}219 + 4 \times 106{,}183 + 18.8 \times 81{,}149 \\ &= 2{,}334{,}990 \text{ kJ} \ \ (\text{Higher than } 2{,}274{,}680 \text{ kJ}) \end{aligned}$$

At 2400 K:

$$\begin{aligned} 3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} &= 3 \times 125{,}152 + 4 \times 103{,}508 + 18.8 \times 79{,}320 \\ &= 2{,}280{,}704 \text{ kJ} \ \ (\text{Higher than } 2{,}274{,}680 \text{ kJ}) \end{aligned}$$

At 2350 K:

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 18.8\bar{h}_{N2} = 3\times 122{,}091 + 4\times 100{,}846 + 18.8\times 77{,}496$$
$$= 2{,}226{,}580 \text{ kJ (Lower than 2,274,680 kJ)}$$

By interpolation of the two results,

T_P = **2394 K = 2121°C**

When propane is burned with 50% excess air, the reaction equation may be written as

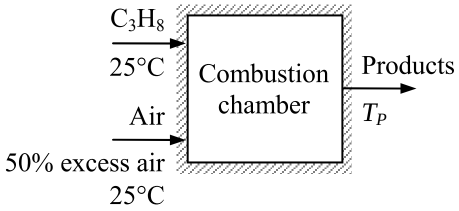
$$C_3H_8 + 1.5\times a_{th}(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 0.5\times a_{th}O_2 + 1.5\times a_{th}\times 3.76N_2$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.5a_{th} = 3 + 2 + 0.5a_{th} \longrightarrow a_{th} = 5$$

Thus,

$$C_3H_8 + 7.5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 2.5O_2 + 28.2N_2$$



Using the values in the table,

$$(3)(-393{,}520 + \bar{h}_{CO2} - 9364) + (4)(-241{,}820 + \bar{h}_{H2O} - 9904) + (2.5)(0 + \bar{h}_{O2} - 8682)$$
$$+ (28.2)(0 + \bar{h}_{N2} - 8669) = (1)(-103{,}850) + 0 + 0$$

It yields $3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 2.5\bar{h}_{O2} + 28.2\bar{h}_{N2} = 2{,}377{,}870 \text{ kJ}$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,377,870/(3+4+2.5+28.2) = 63,073 kJ/kmol. This enthalpy value corresponds to about 1960 K for N_2. Noting that the majority of the moles are N_2, T_P will be close to 1960 K, but somewhat under it because of the higher specific heat of H_2O.

At 1800 K:

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 2.5\bar{h}_{O2} + 28.2\bar{h}_{N2} = 3\times 88{,}806 + 4\times 72{,}513 + 2.5\times 60{,}371 + 28.2\times 57{,}651$$
$$= 2{,}333{,}160 \text{ kJ (Lower than 2,377,870 kJ)}$$

At 1840 K:

$$3\bar{h}_{CO2} + 4\bar{h}_{H2O} + 2.5\bar{h}_{O2} + 28.2\bar{h}_{N2} = 3\times 91{,}196 + 4\times 74{,}506 + 2.5\times 61{,}866 + 28.2\times 59{,}075$$
$$= 2{,}392{,}190 \text{ kJ (Higher than 2,377,870 kJ)}$$

By interpolation,

T_P = **1830 K = 1557°C**

15-89 Ethylene gas is burned steadily with 20 percent excess air. The temperature of products, the entropy generation, and the exergy destruction (or irreversibility) are to be determined.

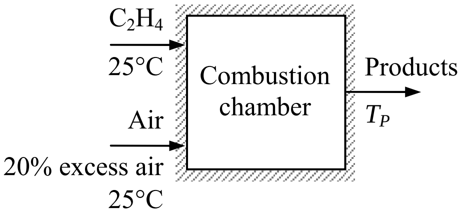
Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis (*a*) The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol of C_2H_4, the combustion equation can be written as

$$\mathrm{C_2H_4(g)} + 1.2a_{\mathrm{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 2\mathrm{CO_2} + 2\mathrm{H_2O} + 0.2a_{\mathrm{th}}\mathrm{O_2} + (1.2)(3.76)a_{\mathrm{th}}\mathrm{N_2}$$

where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$1.2a_{\mathrm{th}} = 2 + 1 + 0.2a_{\mathrm{th}} \longrightarrow a_{\mathrm{th}} = 3$$



Thus,

$$\mathrm{C_2H_4}(g) + 3.6(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 2\mathrm{CO_2} + 2\mathrm{H_2O} + 0.6\mathrm{O_2} + 13.54\mathrm{N_2}$$

Under steady-flow conditions, the exit temperature of the product gases can be determined from the steady-flow energy equation, which reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \bar{h}_{f,R}^\circ = \left(N\bar{h}_f^\circ\right)_{\mathrm{C_2H_4}}$$

since all the reactants are at the standard reference state, and for O_2 and N_2. From the tables,

Substance	$\bar{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol**	$\bar{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol**
C_2H_4 (*g*)	52,280	---
O_2	0	8682
N_2	0	8669
H_2O (*g*)	-241,820	9904
CO_2	-393,520	9364

Substituting,

$$\begin{aligned}(2)(-393{,}520 + \bar{h}_{\mathrm{CO_2}} - 9364) + (2)(-241{,}820 + \bar{h}_{\mathrm{H_2O}} - 9904)\\ + (0.6)(0 + \bar{h}_{\mathrm{O_2}} - 8682) + (13.54)(0 + \bar{h}_{\mathrm{N_2}} - 8669) = (1)(52{,}280)\end{aligned}$$

or, $$2\bar{h}_{\mathrm{CO_2}} + 2\bar{h}_{\mathrm{H_2O}} + 0.6\bar{h}_{\mathrm{O_2}} + 13.54\bar{h}_{\mathrm{N_2}} = 1{,}484{,}083 \text{ kJ}$$

By trial and error,

$$T_P = \mathbf{2269.6\ K}$$

(*b*) The entropy generation during this adiabatic process is determined from

$$S_{\mathrm{gen}} = S_P - S_R = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R$$

The C_2H_4 is at 25°C and 1 atm, and thus its absolute entropy is 219.83 kJ/kmol·K (Table A-26). The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\mathrm{total}}$, where y_i is the mole fraction of component *i* . Also,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

	N_i	y_i	$\bar{s}_i^\circ(T,1\text{atm})$	$R_u \ln(y_i P_m)$	$N_i \bar{s}_i$
C_2H_4	1	1.00	219.83	---	219.83
O_2	3.6	0.21	205.14	-12.98	784.87
N_2	13.54	0.79	191.61	-1.96	2620.94
					S_R = 3625.64 kJ/K
CO_2	2	0.1103	316.881	-18.329	670.42
H_2O	2	0.1103	271.134	-18.329	578.93
O_2	0.6	0.0331	273.467	-28.336	181.08
N_2	13.54	0.7464	256.541	-2.432	3506.49
					S_P = 4936.92 kJ/K

Thus,

$$S_{\text{gen}} = S_P - S_R = 4936.92 - 3625.64 = \mathbf{1311.28\ kJ/kmol \cdot K}$$

and

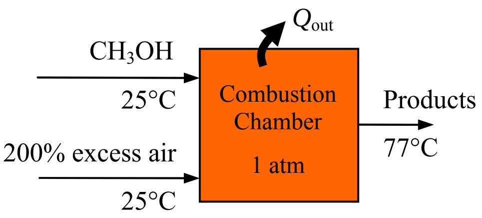
(*c*) $$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\text{ K})(1311.28\text{ kJ/kmol}\cdot\text{K C}_2\text{H}_4) = \mathbf{390{,}760\ \ kJ}\ \ (\text{per kmol C}_2\text{H}_4)$$

15-95 Methyl alcohol is burned steadily with 200 percent excess air in an automobile engine. The maximum amount of work that can be produced by this engine is to be determined.

Assumptions **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

Analysis The fuel is burned completely with the excess air, and thus the products will contain only CO_2, H_2O, N_2, and some free O_2. Considering 1 kmol CH_3OH the combustion equation can be written as

$$\mathrm{CH_3OH} + 3a_{\mathrm{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{CO_2} + 2\mathrm{H_2O} + 2a_{\mathrm{th}}\mathrm{O_2} + 3a_{\mathrm{th}} \times 3.76\mathrm{N_2}$$



where a_{th} is the stoichiometric coefficient and is determined from the O_2 balance,

$$0.5 + 3a_{\mathrm{th}} = 1 + 1 + 2a_{\mathrm{th}} \longrightarrow a_{\mathrm{th}} = 1.5$$

Thus,

$$\mathrm{CH_3OH} + 4.5(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{CO_2} + 2\mathrm{H_2O} + 3\mathrm{O_2} + 16.92\mathrm{N_2}$$

Under steady-flow conditions the energy balance $E_{\mathrm{in}} - E_{\mathrm{out}} = \Delta E_{\mathrm{system}}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{\mathrm{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

Substance	$\bar{\mathbf{h}}_{\mathbf{f}}^\circ$ **kJ/kmol**	$\bar{\mathbf{h}}_{\mathbf{298K}}$ **kJ/kmol**	$\bar{\mathbf{h}}_{\mathbf{350K}}$ **kJ/kmol**
CH_3OH	-200,670	---	---
O_2	0	8682	10,213
N_2	0	8669	10,180
H_2O (*g*)	-241,820	9904	11,652
CO_2	-393,520	9364	11,351

Thus,

$$\begin{aligned}-Q_{\mathrm{out}} &= (1)(-393{,}520 + 11{,}351 - 9364) + (2)(-241{,}820 + 11{,}652 - 9904) \\ &\quad + (3)(0 + 10{,}213 - 8682) + (16.92)(0 + 10{,}180 - 8669) - (1)(-200{,}670) \\ &= -663{,}550 \text{ kJ/kmol of fuel}\end{aligned}$$

The entropy generation during this process is determined from

$$S_{\mathrm{gen}} = S_P - S_R + \frac{Q_{\mathrm{out}}}{T_{\mathrm{surr}}} = \sum N_P\bar{s}_P - \sum N_R\bar{s}_R + \frac{Q_{\mathrm{out}}}{T_{\mathrm{surr}}}$$

The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\mathrm{total}}$, where y_i is the mole fraction of component *i*. Then,

$$S_i = N_i\bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

	N_i	y_i	$\bar{s}_i^\circ(T,1\text{atm})$	$R_u\ln(y_iP_m)$	$N_i\bar{s}_i$
CH_3OH	1	---	239.70	---	239.70
O_2	4.5	0.21	205.04	-12.98	981.09
N_2	16.92	0.79	191.61	-1.960	3275.20
					S_R = 4496 kJ/K
CO_2	1	0.0436	219.831	-26.05	245.88
H_2O (*g*)	2	0.0873	194.125	-20.27	428.79
O_2	3	0.1309	209.765	-16.91	680.03
N_2	16.92	0.7382	196.173	-2.52	3361.89
					S_P = 4717 kJ/K

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = 4717 - 4496 + \frac{663{,}550}{298} = 2448\ \text{kJ/K}\ (\text{per kmol fuel})$$

The maximum work is equal to the exergy destruction

$$W_{\max} = X_{\text{dest}} = T_0 S_{\text{gen}} = (298)(2448\,\text{kJ/K}) = 729{,}400\,\text{kJ/K}\ (\text{per kmol fuel})$$

Per unit mass basis,

$$W_{\max} = \frac{729{,}400\,\text{kJ/K}\cdot\text{kmol}}{32\,\text{kg/kmol}} = \mathbf{22{,}794\ kJ/kg\ fuel}$$