Question 1A. A 1-m³ tank containing air at 10°C and 350 kPa is connected through a valve to another tank containing 3 kg of air at 35°C and 150 kPa. Now the valve is opened, and the entire system is allowed to reach thermal equilibrium with the surroundings, which are at 20°C. Determine the volume of the second tank and the final equilibrium pressure of air.

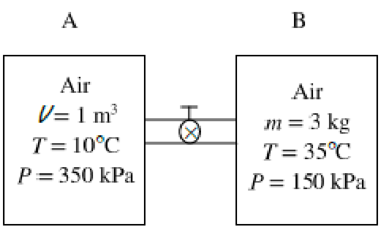
Assumptions At specified conditions, air behaves as an ideal gas.

Properties The gas constant of air is $R = 0.287$ kPa.m³/kg.K (Table A-1).

Analysis Let's call the first and the second tanks A and B. Treating air as an ideal gas, the volume of the second tank and the mass of air in the first tank are determined to be

$$V_B = \left(\frac{m_1 R T_1}{P_1}\right)_B = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})}{150\text{ kPa}} = \mathbf{1.768\ m^3}$$

$$m_A = \left(\frac{P_1 V}{R T_1}\right)_A = \frac{(350\text{ kPa})(1.0\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\text{ K})} = 4.309\text{ kg}$$



Thus,

$$V = V_A + V_B = 1.0 + 1.768 = 2.768\text{ m}^3$$
$$m = m_A + m_B = 4.309 + 3 = 7.309\text{ kg}$$

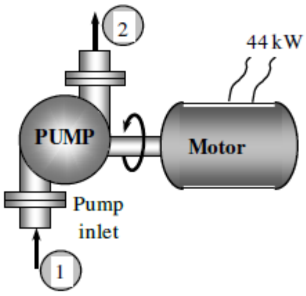
Then the final equilibrium pressure becomes

$$P_2 = \frac{m R T_2}{V} = \frac{(7.309\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{2.768\text{ m}^3} = \mathbf{222\ kPa}$$

Question 1B. An oil pump is drawing 44 kW of electric power while pumping oil with density 860 kg/m^3 at a rate of 0.1 m^3/s. The inlet and outlet diameters of the pipe are 8 cm and 12 cm, respectively. If the pressure rise of oil in the pump is measured to be 500 kPa and the motor efficiency is 90 percent, determine the mechanical efficiency of the pump.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference across the pump is negligible.

Properties The density of oil is given to be $\rho = 860$ kg/m^3.



Analysis Then the total mechanical energy of a fluid is the sum of the potential, flow, and kinetic energies, and is expressed per unit mass as $e_{mech} = gh + Pv + V^2/2$. To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta \dot{E}_{\text{mech, fluid}} = \dot{m}(e_{\text{mech, out}} - e_{\text{mech, in}}) = \dot{m}\left((Pv)_2 + \frac{V_2^2}{2} - (Pv)_1 - \frac{V_1^2}{2}\right) = \dot{V}\left((P_2 - P_1) + \rho\frac{V_2^2 - V_1^2}{2}\right)$$

since $\dot{m} = \rho\dot{V} = \dot{V}/v$, and there is no change in the potential energy of the fluid. Also,

$$V_1 = \frac{\dot{V}}{A_1} = \frac{\dot{V}}{\pi D_1^2/4} = \frac{0.1\ \text{m}^3/\text{s}}{\pi(0.08\ \text{m})^2/4} = 19.9\ \text{m/s}$$

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{0.1\ \text{m}^3/\text{s}}{\pi(0.12\ \text{m})^2/4} = 8.84\ \text{m/s}$$

Substituting, the useful pumping power is determined to be

$$\dot{W}_{\text{pump,u}} = \Delta\dot{E}_{\text{mech,fluid}}$$

$$= (0.1\ \text{m}^3/\text{s})\left(500\ \text{kN/m}^2 + (860\ \text{kg/m}^3)\frac{(8.84\ \text{m/s})^2 - (19.9\ \text{m/s})^2}{2}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\right)\left(\frac{1\ \text{kW}}{1\ \text{kN}\cdot\text{m/s}}\right)$$

$$= 36.3\ \text{kW}$$

Then the shaft power and the mechanical efficiency of the pump become

$$\dot{W}_{\text{pump,shaft}} = \eta_{\text{motor}}\dot{W}_{\text{electric}} = (0.90)(44\ \text{kW}) = 39.6\ \text{kW}$$

$$\eta_{\text{pump}} = \frac{\dot{W}_{\text{pump, u}}}{\dot{W}_{\text{pump, shaft}}} = \frac{36.3\ \text{kW}}{39.6\ \text{kW}} = 0.918 = \mathbf{91.8\%}$$

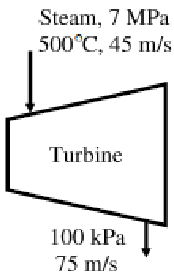
Question 2A. Steam enters an adiabatic turbine steadily at 7 MPa, 500°C, and 45 m/s and leaves at 100 kPa and 75 m/s. If the power output of the turbine is 5 MW and the isentropic efficiency is 77 percent, determine (a) the mass flow rate of steam through the turbine, (b) the temperature at the turbine exit, and (c) the rate of entropy generation during this process. Assume that the exit velocity of the steam from the turbine is not affected by irreversibilities.

Assumptions **1** Steady operating conditions exist. **2** Potential energy changes are negligible.

Analysis (*a*) The properties of the steam at the inlet of the turbine and the enthalpy at the exit for the isentropic case are (Table A-6)

$$\left.\begin{array}{l} P_1 = 7 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3411.4 \text{ kJ/kg} \\ s_1 = 6.8000 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ s_2 = s_1 = 6.8000 \text{ kJ/kg.K} \end{array}\right\} h_{2s} = 2466.6 \text{ kJ/kg}$$



The power output if the expansion was isentropic would be

$$\dot{W}_s = \frac{\dot{W}_a}{\eta_T} = \frac{5000 \text{ kW}}{0.77} = 6494 \text{ kW}$$

An energy balance on the turbine for the isentropic process may be used to determine the mass flow rate of the steam

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_{2s} + \frac{V_2^2}{2}\right) + \dot{W}_s$$

$$\dot{m}\left[3411.4 \text{ kJ/kg} + \frac{(45 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right] = \dot{m}\left[2466.6 \text{ kJ/kg} + \frac{(75 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right] + 6494 \text{ kW}$$

$$\dot{m} = \mathbf{6.886 \text{ kg/s}}$$

(*b*) An energy balance on the turbine for the actual process may be used to determine actual enthalpy at the exit

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{W}_a$$

$$(6.886 \text{ kg/s})\left[3411.4 \text{ kJ/kg} + \frac{(45 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right] = (6.886 \text{ kg/s})\left[h_2 + \frac{(75 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right] + 5000 \text{ kW}$$

$$h_2 = 2683.5 \text{ kJ/kg}$$

Now, other properties at the exit state may be obtained

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ h_2 = 2683.5 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{103.7°C} \\ s_2 = 7.3817 \text{ kJ/kg.K} \end{array}$$

(*c*) Since the turbine is adiabatic, the entropy generation is the entropy change of steam as it flows in the turbine

$$\dot{S}_{gen} = \dot{m}(s_2 - s_1) = (6.886 \text{ kg/s})(7.3817 - 6.8000) \text{ kJ/kg.K} = \mathbf{4.01 \text{ kW/K}}$$

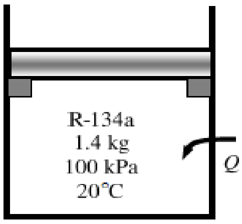
Question 2B. A piston–cylinder device initially contains 1.4 kg of refrigerant-134a at 100 kPa and 20°C. Heat is now transferred to the refrigerant from a source at 150°C, and the piston, which is resting on a set of stops, starts moving when the pressure inside reaches 120 kPa. Heat transfer continues until the temperature reaches 80°C. Assuming the surroundings to be at 25°C and 100 kPa, determine (a) the work done, (b) the heat transfer, (c) the exergy destroyed, and (d) the second-law efficiency of this process.

Assumptions **1** The device is stationary and kinetic and potential energy changes are zero. **2** There is no friction between the piston and the cylinder. **3** Heat is transferred to the refrigerant from a source at 150°C.

Analysis (*a*) The properties of the refrigerant at the initial and final states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.23373 \text{ m}^3/\text{kg} \\ u_1 = 248.81 \text{ kJ/kg} \\ s_1 = 1.0919 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 120 \text{ kPa} \\ T_2 = 80°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.23669 \text{ m}^3/\text{kg} \\ u_2 = 296.94 \text{ kJ/kg} \\ s_2 = 1.2419 \text{ kJ/kg.K} \end{array}$$



Noting that pressure remains constant at 120 kPa as the piston moves, the boundary work is determined to be

$$W_{\text{b,out}} = mP_2(v_2 - v_1) = (1.4 \text{ kg})(120 \text{ kPa})(0.23669 - 0.23373)\text{m}^3/\text{kg} = \mathbf{0.497\ kJ}$$

(*b*) The heat transfer can be determined from an energy balance on the system

$$Q_{\text{in}} = m(u_2 - u_1) + W_{\text{b,out}} = (1.4 \text{ kg})(296.94 - 248.81)\text{kJ/kg} + 0.497 \text{ kJ} = \mathbf{67.9\ kJ}$$

(*c*) The exergy destruction associated with this process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation is determined from an entropy balance on an *extended system* that includes the piston-cylinder device and the region in its immediate surroundings so that the boundary temperature of the extended system where heat transfer occurs is the source temperature,

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

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} + S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1) \longrightarrow S_{\text{gen}} = m(s_2 - s_1) - \frac{Q_{\text{in}}}{T_{\text{source}}}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298 \text{ K})\left[(1.4 \text{ kg})(1.2419 - 1.0919) \text{ kJ/kg}\cdot\text{K} - \frac{67.9 \text{ kJ}}{150 + 273 \text{ K}}\right] = \mathbf{14.8\ kJ}$$

(*d*) Exergy expended is the work potential of the heat extracted from the source at 150°C,

$$X_{\text{expended}} = X_Q = \eta_{\text{th,rev}} Q = \left(1 - \frac{T_L}{T_H}\right)Q = \left(1 - \frac{25 + 273 \text{ K}}{150 + 273 \text{ K}}\right)(67.9 \text{ kJ}) = 20.06 \text{ kJ}$$

Then the 2nd law efficiency becomes

$$\eta_{\text{II}} = \frac{X_{\text{recovered}}}{X_{\text{expended}}} = 1 - \frac{X_{\text{destroyed}}}{X_{\text{expended}}} = 1 - \frac{14.80 \text{ kJ}}{20.06 \text{ kJ}} = 0.262 \text{ or } \mathbf{26.2\%}$$

Discussion The second-law efficiency can also be determined as follows:

The exergy increase of the refrigerant is the exergy difference between the initial and final states,

$$\begin{aligned} \Delta X &= m\left[u_2 - u_1 - T_0(s_2 - s_1) + P_0(v_2 - v_1)\right] \\ &= (1.4 \text{ kg})\left[(296.94 - 248.81)\text{kJ/kg} - (298 \text{ K})(1.2419 - 1.0919)\text{kg.K} + (100 \text{ kPa})(0.23669 - 0.23373)\text{m}^3/\text{kg}\right] \\ &= 5.177 \text{ kJ} \end{aligned}$$

The useful work output for the process is

$$W_{\text{u,out}} = W_{\text{b,out}} - mP_0(v_2 - v_1) = 0.497 \text{ kJ} - (1.4 \text{ kg})(100 \text{ kPa})(0.23669 - 0.23373)\text{m}^3/\text{kg} = 0.0828 \text{ kJ}$$

The exergy recovered is the sum of the exergy increase of the refrigerant and the useful work output,

$$X_{\text{recovered}} = \Delta X + W_{\text{u,out}} = 5.177 + 0.0828 = 5.260 \text{ kJ}$$

Then the second-law efficiency becomes

$$\eta_{\text{II}} = \frac{X_{\text{recovered}}}{X_{\text{expended}}} = \frac{5.260 \text{ kJ}}{20.06 \text{ kJ}} = 0.262 \text{ or } \mathbf{26.2\%}$$

Question 3: A four-cylinder, four-stroke, 1.8-liter modern, high-speed compression-ignition engine operates on the ideal dual cycle with a compression ratio of 16. The air is at 95 kPa and 70°C at the beginning of the compression process and the engine speed is 2200 rpm. Equal amounts of fuel are burned at constant volume and at constant pressure. The maximum allowable pressure in the cycle is 7.5 MPa due to material strength limitations. Using constant specific heats at 1000 K, determine (a) the maximum temperature in the cycle, (b) the net work output and the thermal efficiency, (c) the mean effective pressure, and (d) the net power output.

Solution:

9-149 A modern compression ignition engine operates on the ideal dual cycle. The maximum temperature in the cycle, the net work output, the thermal efficiency, the mean effective pressure, the net power output, the second-law efficiency of the cycle, and the rate of exergy of the exhaust gases are to be determined.

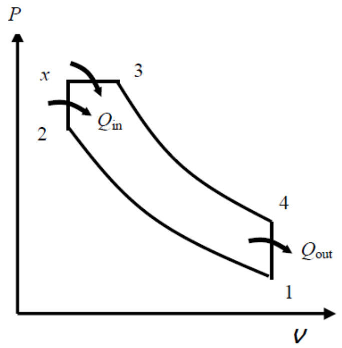
Assumptions **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

Properties The properties of air at 1000 K are $c_p = 1.142$ kJ/kg·K, $c_v = 0.855$ kJ/kg·K, $R = 0.287$ kJ/kg·K, and $k = 1.336$ (Table A-2b).

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

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 16 = \frac{V_c + 0.0018\,\text{m}^3}{V_c} \longrightarrow V_c = 0.00012\,\text{m}^3 = V_2 = V_x$$

$$V_1 = V_c + V_d = 0.00012 + 0.0018 = 0.00192\,\text{m}^3 = V_4$$



Process 1-2: Isentropic compression

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (343\,\text{K})(16)^{1.3361} = 870.7\,\text{K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = (95\,\text{kPa})(16)^{1.336} = 3859\,\text{kPa}$$

Process 2-x and x-3: Constant-volume and constant pressure heat addition processes:

$$T_x = T_2\frac{P_x}{P_2} = (870.7\,\text{K})\frac{7500\,\text{kPa}}{3859\,\text{kPa}} = 1692\,\text{K}$$

$$q_{2\text{-}x} = c_v(T_x - T_2) = (0.855\,\text{kJ/kg.K})(1692 - 870.7)\text{K} = 702.6\,\text{kJ/kg}$$

$$q_{2\text{-}x} = q_{x\text{-}3} = c_p(T_3 - T_x) \longrightarrow 702.6\,\text{kJ/kg} = (0.855\,\text{kJ/kg.K})(T_3 - 1692)\text{K} \longrightarrow T_3 = \mathbf{2308K}$$

(*b*)
$$q_{in} = q_{2\text{-}x} + q_{x\text{-}3} = 702.6 + 702.6 = 1405\,\text{kJ/kg}$$

$$V_3 = V_x\frac{T_3}{T_x} = (0.00012\,\text{m}^3)\frac{2308\,\text{K}}{1692\,\text{K}} = 0.0001636\,\text{m}^3$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = (2308\,\text{K})\left(\frac{0.0001636\,\text{m}^3}{0.00192\,\text{m}^3}\right)^{1.3361} = 1009\,\text{K}$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = (7500\,\text{kPa})\left(\frac{0.0001636\,\text{m}^3}{0.00192\,\text{m}^3}\right)^{1.336} = 279.4\,\text{kPa}$$

Process 4-1: constant voume heat rejection.

$$q_{out} = c_v(T_4 - T_1) = (0.855\,\text{kJ/kg}\cdot\text{K})(1009 - 343)\text{K} = 569.3\,\text{kJ/kg}$$

The net work output and the thermal efficiency are

$$w_{net,out} = q_{in} - q_{out} = 1405 - 569.3 = \mathbf{835.8kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{835.8\,\text{kJ/kg}}{1405\,\text{kJ/kg}} = 0.5948 = \mathbf{59.5\%}$$

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

$$m = \frac{P_1 V_1}{R T_1} = \frac{(95\,\text{kPa})(0.00192\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(343\,\text{K})} = 0.001853\,\text{kg}$$

$$\text{MEP} = \frac{m w_{\text{net,out}}}{V_1 - V_2} = \frac{(0.001853\,\text{kg})(835.8\text{kJ/kg})}{(0.00192 - 0.00012)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{860.4\,kPa}$$

(*d*) The power for engine speed of 2200 rpm is

$$\dot{W}_{\text{net}} = m w_{\text{net}} \frac{\dot{n}}{2} = (0.001853\,\text{kg})(835.8\,\text{kJ/kg}) \frac{2200\,\text{(rev/min)}}{(2\,\text{rev/cycle})}\left(\frac{1\,\text{min}}{60\,\text{s}}\right) = \mathbf{28.39 kW}$$

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

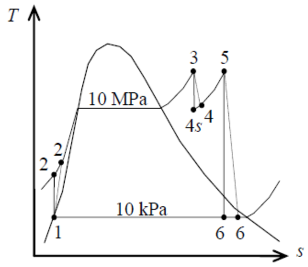
Question 4: Consider a steam power plant that operates on a reheat Rankine cycle and has a net power output of 80 MW. Steam enters the high-pressure turbine at 10 MPa and 500°C and the low-pressure turbine at 1 MPa and 500°C. Steam leaves the condenser as a saturated liquid at a pressure of 10 kPa. The isentropic efficiency of the turbine is 80 percent, and that of the pump is 95 percent. Show the cycle on a T-s diagram with respect to saturation lines and determine (a) the quality (or temperature, if superheated) of the steam at the turbine exit, (b) the thermal efficiency of the cycle, and (c) the mass flow rate of the steam.

Solution:

10-37 A steam power plant that operates on a reheat Rankine cycle is considered. The quality (or temperature, if superheated) of the steam at the turbine exit, the thermal efficiency of the cycle, and the mass flow rate of the steam 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),



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

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@10\text{ kPa}} = 0.001010 \text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= (0.00101 \text{ m}^3/\text{kg})(10{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 10.62 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 10.62 = 202.43 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 10 \text{ MPa} \\ T_3 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3375.1 \text{ kJ/kg} \\ s_3 &= 6.5995 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_{4s} &= 1 \text{ MPa} \\ s_{4s} &= s_3 \end{aligned}\right\} h_{4s} = 2783.8 \text{ kJ/kg}$$

$$\begin{aligned} \eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 &= h_3 - \eta_T(h_3 - h_{4s}) \\ &= 3375.1 - (0.80)(3375.1 - 2783.7) = 2902.0 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_5 &= 1 \text{ MPa} \\ T_5 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_5 &= 3479.1 \text{ kJ/kg} \\ s_5 &= 7.7642 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_{6s} &= 10 \text{ kPa} \\ s_{6s} &= s_5 \end{aligned}\right\} \begin{aligned} x_{6s} &= \frac{s_{6s} - s_f}{s_{fg}} = \frac{7.7642 - 0.6492}{7.4996} = 0.9487 \quad \text{(at turbine exit)} \\ h_{6s} &= h_f + x_{6s}h_{fg} = 191.81 + (0.9487)(2392.1) = 2461.2 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} \eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 &= h_5 - \eta_T(h_5 - h_{6s}) \\ &= 3479.1 - (0.80)(3479.1 - 2461.2) \\ &= 2664.8 \text{ kJ/kg} > h_g \text{ (superheated vapor)} \end{aligned}$$

From steam tables at 10 kPa we read $T_6 = \mathbf{88.1°C}$.

(*b*) $w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3375.1 - 2902.0 + 3479.1 - 2664.8 = 1287.4 \text{ kJ/kg}$

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3375.1 - 202.43 + 3479.1 - 2902.0 = 3749.8 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{p,in}} = 1287.4 - 10.62 = 1276.8 \text{ kJ/kg}$$

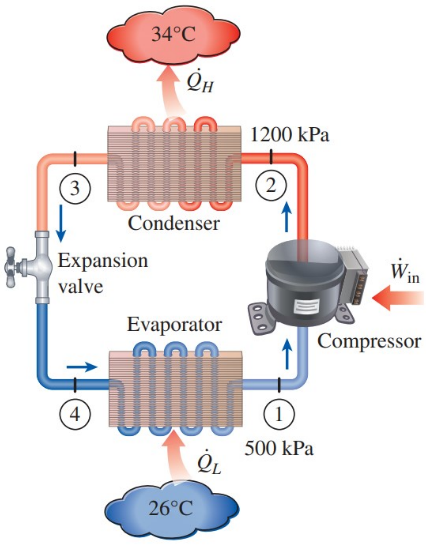
Thus the thermal efficiency is

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1276.8 \text{ kJ/kg}}{3749.8 \text{ kJ/kg}} = \mathbf{34.1\%}$$

(*c*) The mass flow rate of the steam is

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{80{,}000 \text{ kJ/s}}{1276.9 \text{ kJ/kg}} = \mathbf{62.7 \text{ kg/s}}$$

Question 5: An air conditioner with refrigerant-134a as the working fluid is used to keep a room at 26°C by rejecting the waste heat to the outside air at 34°C. The room is gaining heat through the walls and the windows at a rate of 250 kJ/min while the heat generated by the computer, TV, and lights amounts to 900 W. An unknown amount of heat is also generated by the people in the room. The condenser and evaporator pressures are 1200 and 500 kPa, respectively. The refrigerant is saturated liquid at the condenser exit and saturated vapor at the compressor inlet. If the refrigerant enters the compressor at a rate of 100 L/min and the isentropic efficiency of the compressor is 75 percent, determine (a) the temperature of the refrigerant at the compressor exit, (b) the rate of heat generation by the people in the room, (c) the COP of the air conditioner, and (d) the minimum volume flow rate of the refrigerant at the compressor inlet for the same compressor inlet and exit conditions.

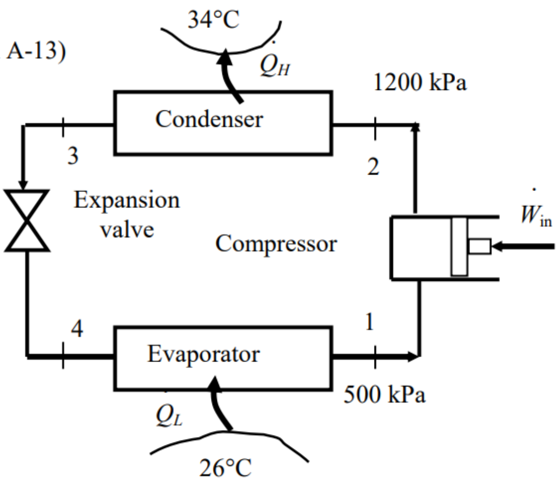


Solution:

11-111 An air-conditioner with refrigerant-134a as the refrigerant is considered. The temperature of the refrigerant at the compressor exit, the rate of heat generated by the people in the room, the COP of the air-conditioner, and the minimum volume flow rate of the refrigerant at the compressor inlet are to be determined.

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)



$$\left.\begin{array}{l} P_1 = 500\text{ kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 259.36\text{ kJ/kg} \\ \boldsymbol{v}_1 = 0.04117\text{ kJ/kg} \\ s_1 = 0.9242\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1200\text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 277.45$$

$$h_3 = h_{f@1200\text{kPa}} = 117.79\text{ kJ/kg}$$

$$h_4 = h_3 = 117.79\text{ kJ/kg}$$

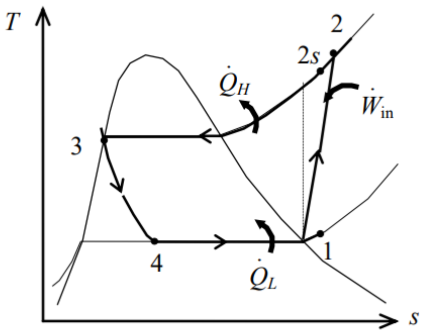
$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.75 = \frac{277.45 - 259.36}{h_2 - 259.36} \longrightarrow h_2 = 283.48\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1200\,\text{kPa} \\ h_2 = 283.48\,\text{kJ/kg} \end{array}\right\} T_2 = \mathbf{54.5^\circ C}$$

(*b*) The mass flow rate of the refrigerant is

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{(100\,\text{L/min})\left(\dfrac{1\,\text{m}^3}{1000\,\text{L}}\right)\left(\dfrac{1\,\text{min}}{60\,\text{s}}\right)}{0.04117\,\text{m}^3/\text{kg}} = 0.04048\,\text{kg/s}$$



The refrigeration load is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.04048\,\text{kg/s})(259.36 - 117.79)\text{kJ/kg} = 5.731\,\text{kW}$$

which is the total heat removed from the room. Then, the rate of heat generated by the people in the room is determined from

$$\dot{Q}_{\text{people}} = \dot{Q}_L - \dot{Q}_{\text{heat}} - \dot{Q}_{\text{equip}} = (5.731 - 250/60 - 0.9)\,\text{kW} = \mathbf{0.665 kW}$$

(*c*) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.04048\,\text{kg/s})(283.48 - 259.36)\text{kJ/kg} = 0.9764\,\text{kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.731}{0.9764} = \mathbf{5.87}$$

(*d*) The reversible COP of the cycle is

$$\text{COP}_{\text{rev}} = \frac{1}{T_H/T_L - 1} = \frac{1}{(34+273)/(26+273) - 1} = 37.38$$

The corresponding minimum power input is

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{rev}}} = \frac{5.731\,\text{kW}}{37.38} = 0.1533\,\text{kW}$$

The minimum mass and volume flow rates are

$$\dot{m}_{\text{min}} = \frac{\dot{W}_{\text{in,min}}}{h_2 - h_1} = \frac{0.1533\,\text{kW}}{(283.48 - 259.36)\text{kJ/kg}} = 0.006358\,\text{kg/s}$$

$$\dot{\boldsymbol{V}}_{1,\text{min}} = \dot{m}_{\text{min}}\boldsymbol{v}_1 = (0.006358\,\text{kg/s})(0.04117\,\text{m}^3/\text{kg}) = (0.0002618\,\text{m}^3/\text{s}) = \mathbf{15.7 L/min}$$

Question 6A: Show that: $c_p - c_v = T\left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial v}{\partial T}\right)_P$.

Solution:

12-43 It is to be shown that $c_p - c_v = T\left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial v}{\partial T}\right)_P$.

Analysis We begin by taking the entropy to be a function of specific volume and temperature. The differential of the entropy is then

$$ds = \left(\frac{\partial s}{\partial T}\right)_v dT + \left(\frac{\partial s}{\partial v}\right)_T dv$$

Substituting $\left(\frac{\partial s}{\partial T}\right)_v = \frac{c_v}{T}$ from Eq. 12-28 and the third Maxwell equation changes this to

$$ds = \frac{c_v}{T} dT + \left(\frac{\partial P}{\partial T}\right)_v dv$$

Taking the entropy to be a function of pressure and temperature,

$$ds = \left(\frac{\partial s}{\partial T}\right)_P dT + \left(\frac{\partial s}{\partial P}\right)_T dP$$

Combining this result with $\left(\frac{\partial s}{\partial T}\right)_P = \frac{c_p}{T}$ from Eq. 12-34 and the fourth Maxwell equation produces

$$ds = \frac{c_p}{T} dT - \left(\frac{\partial v}{\partial T}\right)_P dP$$

Equating the two previous ds expressions and solving the result for the specific heat difference,

$$(c_p - c_v)dT = T\left(\frac{\partial v}{\partial T}\right)_P dP + T\left(\frac{\partial P}{\partial T}\right)_v dv$$

Taking the pressure to be a function of temperature and volume,

$$dP = \left(\frac{\partial P}{\partial T}\right)_v dT + \left(\frac{\partial P}{\partial v}\right)_T dv$$

When this is substituted into the previous expression, the result is

$$(c_p - c_v)dT = T\left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial T}\right)_v dT + T\left[\left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial v}\right)_T + \left(\frac{\partial P}{\partial T}\right)_v\right] dv$$

According to the cyclic relation, the term in the bracket is zero. Then, canceling the common dT term,

$$c_p - c_v = T\left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial v}{\partial T}\right)_P$$

Question 6B: A fuel mixture of 60 percent by mass methane (CH_4) and 40 percent by mass ethanol (C_2H_6O), is burned completely with theoretical air. If the total flow rate of the fuel is 10 kg/s, determine the required mass flow rate of air. Please write the balanced chemical reaction for the complete combustion of the fuel-air mixture.

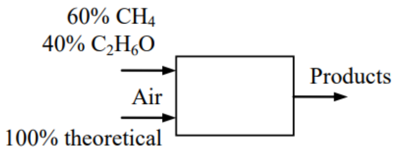
Solution:

15-25 A fuel mixture of 60% by mass methane, CH_4, and 40% by mass ethanol, C_2H_6O, is burned completely with theoretical air. The required flow rate of air is to be determined.

Assumptions **1** Combustion is complete. **2** The combustion products contain CO_2, H_2O, 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 For 100 kg of fuel mixture, the mole numbers are



$$N_{CH4} = \frac{mf_{CH4}}{M_{CH4}} = \frac{60\text{ kg}}{16\text{ kg/kmol}} = 3.75\text{ kmol}$$

$$N_{C2H6O} = \frac{mf_{C2H6O}}{M_{C2H6O}} = \frac{40\text{ kg}}{46\text{ kg/kmol}} = 0.8696\text{ kmol}$$

The mole fraction of methane and ethanol in the fuel mixture are

$$x = \frac{N_{CH4}}{N_{CH4} + N_{C2H6O}} = \frac{3.75\text{ kmol}}{(3.75 + 0.8696)\text{ kmol}} = 0.8118$$

$$y = \frac{N_{C2H6O}}{N_{CH4} + N_{C2H6O}} = \frac{0.8696\text{ kmol}}{(3.75 + 0.8696)\text{ kmol}} = 0.1882$$

The combustion equation in this case can be written as

$$x\,CH_4 + y\,C_2H_6O + a_{th}[O_2 + 3.76N_2] \longrightarrow B\,CO_2 + D\,H_2O + F\,N_2$$

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: $x + 2y = B$

Hydrogen balance: $4x + 6y = 2D$

Oxygen balance: $2a_{th} + y = 2B + D$

Nitrogen balance: $3.76a_{th} = F$

Substituting x and y values into the equations and solving, we find the coefficients as

$$x = 0.8118 \qquad B = 1.188$$
$$y = 0.1882 \qquad D = 2.188$$
$$a_{th} = 2.188 \qquad F = 8.228$$

Then, we write the balanced reaction equation as

$$0.8118\,CH_4 + 0.1882\,C_2H_6O + 2.188[O_2 + 3.76N_2] \longrightarrow 1.188\,CO_2 + 2.188\,H_2O + 8.228\,N_2$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\text{AF} = \frac{m_{air}}{m_{fuel}} = \frac{(2.188 \times 4.76\text{ kmol})(29\text{ kg/kmol})}{(0.8118\text{ kmol})(12 + 4 \times 1)\text{kg/kmol} + (0.1882\text{ kmol})(2 \times 12 + 6 \times 1 + 16)\text{kg/kmol}}$$
$$= 13.94\text{ kg air/kg fuel}$$

Then, the required flow rate of air becomes

$$\dot{m}_{air} = \text{AF}\dot{m}_{fuel} = (13.94)(10\text{ kg/s}) = \mathbf{139.4\,kg/s}$$